MARINE BOILER MANAGEMENT AND CONSTRUCTION

BEING A TREATISE ON

BOILER TROUBLES AND REPAIRS, CORROSION, FUELS AND HEAT ON THE PROPERTIES OF IRON AND STEEL, ON BOILER MECHANICS WORKSHOP PRACTICES AND BOILER DESIGN

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PREFACE

TO

FOURTH EDITION

The results of experimental researches into a variety of subjects are accumulating which affect either directly or indirectly the management or construction of boilers; these have been referred to in the various chapters of this edition. Thus amongst physical researches may be mentioned Stefan and Bolzmann's law of radiation which seems to explain why the high temperature flames which are associated with perfect combustion lead to furnace troubles, then there are Nicholson's experiments on the influence of the speed of a gas on its heat transmission, Coker's optical measurements of strains, and my own experiments on fatigue of metals—as yet unpublished—on brittleness caused by the presence of nitrogen in steel, and on water hammers in steam pipes. Heyn & Bauer's exhaustive experiments on corrosion have also been carefully summarised in this work.

On account of the increasing use of wrought iron and steel instead of copper for steam pipes, a new chapter has been added on this subject. It reviews all available experiments on the losses of pressure due to friction and to radiation, explains the methods of manufacture and discusses steam pipe explosions, of which about two hundred have been reported upon by the Board of Trade as being due either to water hammer, to inelastic arrangement or to bad material. The question of elasticity of pipe bends has necessitated the addition of a few mathematical remarks on curved beams, and a comparison of these with Prof. Bautlin's valuable experiments on full sized pipes. They were found to be of great use in estimating the stresses in the flanges of end plates of boilers. This subject and the staying of flat plates is deserving of greater attention than is generally bestowed on it by engineers.

Fortunately a large number of experiments with flat plates have now been made both by the Board of Trade, the German Admiralty, Prof. Bach, and others; they have been carefully summarised and are found to harmonise better than might have been expected. The staying of annular water spaces of donkey boilers is also dealt with, but only mathematically, as no experiments have been carried out.

The increasing use of high speed tool steel, and of electric and oxy-acetylene welding and a few other matters are also mentioned in the present edition.

The re-arrangement of the Board of Trade Rules in the early part this year has necessitated a similar re-arrangement of the last chapter in the present edition, but the other leading chapters on Construction and on Management, have received comparatively few additions or alterations.

C. E. S.

WEST DIDSBURY,
December, 1913.

PREFACE

TO

THE FIRST EDITION

While reading through these pages for the last time, and thereby completing a task which has proved a heavier one than could at first have been imagined, numerous passages have recalled to mind friendly discussions of which they are the outcome, or valuable hints and sometimes exhaustive criticisms from those friends to whom doubtful points were submitted. Should they find that their views have not in all cases been adopted, the text will doubtless also reveal to them the reasons why this could not be done; and to these friends I wish to convey my warmest thanks for the encouragement which their personal interest in this work has afforded me.

While collecting the material for this work a feeling that many problems yet remain to be solved has rarely been absent from my mind, more especially when the scientific side of a question was being inquired into. Not being in a position to satisfactorily discuss such problems, it seemed necessary at least to state them concisely, so that scientists might be induced to solve them for us. It would have been very easy to ignore such difficulties altogether; but this course would have been contrary to my purpose, which was to produce a work of a practical character. If it should be objected that just because of this object even the simplest mathematics ought to have been omitted, such critics should remember that true practice, unlike abstract science, is unscrupulous in the choice of means, and avails itself of the gratuitous labours of scholars and scientists as readily as it adopts the more costly experiences gained by repeated failures. Besides, as the object of science is truth, practical men, whose sole aim is success, dare not remain in the dark as regards its discoveries and deductions.

C. E. S.

Glasgow: *August* 1893.

INTRODUCTION

The information contained in this work has been collected for the use of people interested in the manufacture and management of marine boilers, and it is hoped that the summaries of the experiences gained in one of the branches with which it deals will assist those whose attention has been confined more particularly to the other. Thus, for the information of manufacturers, it was necessary to discuss the troubles to be expected from the use of defective materials, as well as the dangers to which a boiler is exposed after it leaves their hands; while for steam users, descriptions as to the processes of construction, and scientific inquiries about corrosion, fuels, and similar subjects, had to be brought together.

In spite of a very exhaustive search, extending over several years, little information either about the management or about workshop practices could be found, most books and papers professing to deal therewith doing so only in vague terms. The Author had, therefore, to rely mainly on his own experiences when explaining the various practices and manipulations. Although it was impossible to enter into every minute detail, it is hoped that no points have been omitted to which attention should be drawn.

In writing the chapter on 'Mechanics' the Author also had to rely mainly on his own resources when trying to present such problems as occur in marine boilers in as simple and yet as comprehensive a form as possible. Special attention has there been paid to the relations existing between elastic stresses and those which make their appearance just before rupture takes place, whereby it is hoped that the term 'factor of safety' has acquired a more precise meaning than it at present possesses. A graphic method for resolving stresses, a method for estimating the shearing strengths of a material from torsion experiments, and a discussion on the irregular distribution of stresses in riveted joints, are some of the subjects herein discussed, and about which little will be found in earlier books.

In the chapter on 'Corrosion' attention is drawn, amongst other matters, to the strange influence which apparently harmless salts exert on the harmful activity of weak acids; the action of air in feed-water and the much-debated question of galvanic currents in boilers have been treated in some detail.

While examining the numerous experiments on 'Heat Transmission,' and before Mr. Durston's excellent paper was read, the Author was confronted with the serious difficulty that nearly all these experiments are incomplete as regards certain essential points. Sometimes the heating value of the coal was not stated; sometimes the steam pressure, funnel or feed temperature was forgotten, or the ashes not weighed; but by having brought together various experiments in one chapter a better idea can now be formed than has yet been possible about the resistance encountered by heat when it travels from the flame through the iron and scale into the water.

Somewhat similar remarks might be made on the subject of 'Strength of Materials,' for metallurgists are still unable to explain why a metal which can be stretched from 20 to 30 per cent. in a testing machine sometimes shows no plasticity, and cracks spontaneously, when fitted in a boiler. Pains have been taken to collect all references to such influences as might cause trouble when using steel, and it is hoped that the Author's experience, both in engineering works and in numerous English and foreign steelworks, has led him to touch upon everything that is essential.

The chapter on 'Fuels and Combustion' ought to be of value to those who are entrusted with the carrying out of accurate experiments on the performances of engines and boilers. Not only has it been explained there how necessary it is to know the heating value of the fuel used, and whether it has been properly burnt, but explanations have been added showing how to carry out the experiments, and as they are comparatively simple, seagoing engineers possessed of a knowledge of chemistry might easily inform themselves on questions such as how much heat and unconsumed fuel are escaping up the funnel, and how much priming water is carried over with the steam, about which points reliable information is still wanting.

For the convenience of draughtsmen the rules of the Board of Trade and of Lloyd's Register on the scantlings of boilers have all been placed near the last pages, and much trouble has been taken to make the tables which are based on them not only reliable and comprehensive, but also compact. But as these rules are liable to occasional amendments, the tables may also have to be revised from time to time.

These rules and tables are preceded by a short chapter on 'Design,' in which such hints have been given as will, it is hoped, facilitate drawing-office work. It has not been thought advisable to reproduce any drawings of complete boilers, but a long list has been compiled of publications where they can be looked up, together with their principal dimensions, their performances, and the builders' names.

These and the other references to publications were found to be so numerous that it was necessary to abridge their titles to a few initial letters, which have been rearranged in alphabetical order in the following list, together with the necessary particulars for easily finding the required volume. It was at first intended to quote only such papers or books as had been printed in the English language; but many of these turned out to be translations, and it soon became evident that they could not always be relied upon. Not only were the misprints sometimes of the most serious nature, but in several instances it was found impossible to trace the original paper, because of the date, volume, or page being wrongly given. The plan had therefore to be adopted of mentioning only the original notice of experiments on investigation, no matter in what language they were first described.

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MARINE BOILER MANAGEMENT

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CHAPTER I

BOILER MANAGEMENT

SUCCESSFUL management of a boiler consists in getting the desired amount of work out of it, and at the same time keeping the expenses as low as possible. To be able to do this, a boiler must above all things be well designed for its work; it should be handled with the proper amount of care, and any defects which may show themselves should be made good at once and their causes removed.

Lighting Fires.—The fires in the main boiler are usually lighted by throwing some burning fuel from the donkey boiler into the main furnaces; but if this cannot be done, a small fire is kindled on one of the grates and the others lighted from it. In either case the whole of the fire bars are first covered with coal, so as to restrict the draught to that point where the fuel is burning; gradually, as the fire increases, the hot fuel is raked back, igniting the remainder. If the coal is very dusty it may be necessary to cover the bars with old matting or with paper.

If steam has to be raised quickly, the fires are started in all the furnaces at once; but this should never be done unless there is steam in the donkey boiler, and appliances are available for producing artificial circulation. Even where this is the case, and steam has to be raised with the utmost possible despatch, it is safer to force only one boiler, or such a number as are sufficient to drive the engine; for in all cases of great hurry, things are likely to go wrong, and it has happened over and over again that circumferential seams of boiler bottoms have cracked when not sufficiently warmed during this period. The circulating appliances used to be hydrokineters; now it is customary to fit a donkey suction to each boiler bottom, by which means the cold water is drawn off there and reintroduced through the feed near the water level.[1] Where none of these arrangements are fitted, the heating of the boiler must progress slowly, taking about twelve hours. Many engineers start fires only in one furnace of each boiler, usually the lowest one, in the hope that the cold water, which remains undisturbed

[1] These pipes should be so fitted that water cannot accidentally be forced from one boiler into another if their pressures should differ.

at the bottom, will get warmed. A better plan is to light fires in one of the wing furnaces, F, fig. 1, of each boiler, which sets up a natural circulation, the water rising on one side and descending on the other.

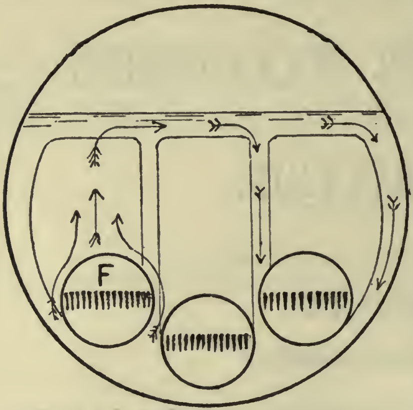

Fig. 1

If steam must be raised quickly, a good plan is to fill the boilers to nearly full glass, to light all the fires at once, and to blow off all the cold water below the furnaces as soon as that above them is boiling. From fear of subsequent leakage, the safety valves are usually kept closed, and the result is that the heated air in the steam space shows a pressure on the gauge before steam has been generated, which is most misleading, because it all disappears as soon as the engines are started. This air pressure on the water surface also prevents the formation of steam bubbles when 212° F. has been reached, which would materially assist the circulation, by carrying water with them as they rise. It is, therefore, better to keep the main or safety valves full open till steam is blowing off freely, and only to close them when the boiler bottom has grown hot. If closed earlier, the chances are, even with slow firing, that the boiler bottoms remain cold long after the full working pressure has been reached, and boiler shells being constructed mainly to resist the steam pressure, are then sometimes unable to bear the additional straining to which they are subjected by the very serious differences of temperature between the upper two-thirds and lower one-third of their circumference. (See fig. 93, p. 75.)

Cracked Shell Plates.—This difference of temperature can amount to as much as 270° F., and, as iron expands about one-thousandth of its length between 32° F. and 212° F., the bottom of the boiler shell would tend to be about one-seven-hundredth of its length shorter than the top, which, in a double-ended boiler of 17½ feet length, would amount to more than ¼ inch. Of course the upper two-thirds of the boiler would be slightly compressed, but the amount of metal in the lower third being the smaller, suffers the severest stress, probably quite two-thirds of that which the difference of temperature would warrant. But if iron or steel is prevented from contracting one-thousandth of its length, which is the same thing as stretching the metal by that amount, a stress of 13 tons per square inch is set up, and this, then, is approximately the extra stress which the lower third of the boiler shell has to resist.

As the percentage of strength of the circumferential joints is comparatively low, it is not to be wondered at that either the rivets shear or that the metal tears. In either case there is no immediate danger, for the leakage is slight and the water cold, at least at first. However, sometimes the solid plate cracks circumferentially, and then the rush of water is considerable; but it is only amongst iron boilers that this happens, which is doubtless due to the fact that this material is decidedly weaker across the grain than with it. As might be expected, double-ended boilers are more often injured in this way than single-ended ones.

Those who wish to obtain numerical results on this subject should fit the following arrangement to the backs of a boiler, one near the water level and the other near the bottom, or wherever they think it most convenient. A short iron tube (fig. 2), about $\frac{3}{4}$ of an inch in diameter, and closed at one end, is screwed at an angle of about 30° into the back plate, and a little mercury or oil poured into it. When desired a thermometer can be inserted, and the temperature measured. If mercury is used the tubes must not be of brass or gun metal, otherwise they will be eaten away.

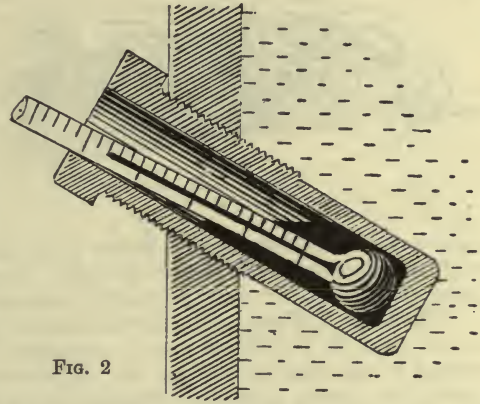

FIG. 2

Stoking.—A perfect knowledge of stoking can only be gained by a practical experience, which does not fall to the lot of most engineers. It is not difficult to throw shovelsful of coal through the comparatively small fire doors, even when they are 4 feet above the floors, but generally inexperienced hands will not be able to place the fuel on those parts of the grate where it is wanted, even when the conditions are favourable—and that is one of the chief secrets of good stoking, particularly with Welsh and similar coal, which may not be disturbed after it has been put on the grate. North-country coal has a tendency to cake, whereby the air passages are choked. While breaking up this fuel with a rake or slicer an excellent opportunity is afforded for levelling it.

While burning Welsh coal the case is different. The depressions where the air meets with least resistance, are burnt away quickest, even to such an extent that the upper edge of the bars becomes exposed, admitting a damaging excess of air (fig. 3). On account of this short cut very much less air finds its way through the thick fuel, and the combustion is reduced there. Uncovered bars and thin fires are readily discovered on trials by holding an anemometer in each ashpit. On recoaling, the hollows will most likely be more than filled up; but even if levelled, the combustion will be fiercest where the hot fuel was lying thickest, for there it is

FIG. 3

all aglow (fig. 4), and these are the parts which will now burn away quickest. Continued care is therefore necessary.

What is true of a single fire is true of a number, and unless all the fires connected to one funnel are kept equally thick, the thin ones will burn away fastest, and if there are several stokers on one watch, he will have the lightest job who chokes his furnace with fuel. When steaming under strong forced draught with closed stoke holes, speed in coaling is of the utmost importance, so as not to keep the doors open too long, and it is customary to have a right- and left-handed stoker for each fire, with a leading man to direct them where to throw the fuel. In order to see better which parts of the grate are bare he should wear green-glassed spectacles.

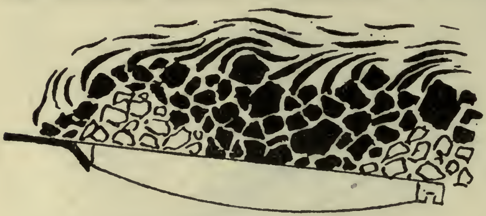

Fig. 4

Thickness of Fires.—The question as to how thick the fires ought to be kept is a difficult one to answer. Less resistance will be offered to the passage of the air, and the combustion will be more rapid, the thinner they are. Thick fires, or rather heavy firing, offer the advantage that the furnace doors have to be opened less frequently, and the loss and injury due to inrushes of cold air are not so great; but such fires produce an enormous amount of volatile products, which are often so cold that they cannot ignite, as they should do, when mixed with the air which is admitted through the doors. This loss is, of course, greater with bituminous coal.

Under natural draught the fires should be kept about 6 and 8 ins. thick respectively for North-country and for Welsh coal. Under forced draught they are sometimes 12 ins. thick.

Smoke Consumption.—In order to consume the volatile gases some firemen coal first one side of a grate and then the other, half of the upper surface being thus always in a glowing condition. Another plan—but this can only be carried out with caking coal—is to throw the green coal on the front end of the grate, and to rake it back when it is well alight. The combustible gases which are at first given off are thoroughly mixed with the air which reaches them through the doors, and passing over the red-hot coal the mixture is bound to ignite.

This plan has the advantage of allowing very long grates to be used. However, the ashes and clinker will be driven to the back ends, against the firebrick, whence it is most inconvenient to remove them. With dirty coals this trouble is so serious that firemen prefer to throw them as far back as possible, and to rake them forward when partly consumed, accumulating the ashes at the front end. This is an uneconomical proceeding.

Side Firing consists in throwing coal first on one side of a grate, and, when this has been burnt through throwing coal on the other side, with the result that the radiant heat of the hot side not only causes complete combustion of the smoke produced on the black side, but it also ignites the surface of the newly added coal and checks the production of smoke. Side firing, however, has the disadvantage, that the firemen have to open the fire-doors twice as often as is now customary, and this frequent opening results in more air being admitted above the grates than is necessary. Nevertheless, there can be no doubt but that this method of firing is very economical. It might be simplified and improved, leading to a saving of labour if a leaf were to be taken from the gasworks practice of charging long retorts. In these works the coal is shovelled into long semi-cylindrical troughs, which are lifted up to the retorts, shoved in, turned round, and withdrawn. Fig. 4*a* contains a suggestion how this method might be applied to the firing of marine boilers. The fire-door would have to be provided with two large circular holes, closed by flaps, slides, or by a revolving disc. When the left side of the furnace has to be charged with coal, the left-hand hole is opened and a semicircular trough filled with coal inserted, turned, round and quickly withdrawn. The filling of the trough is a much easier operation than throwing coal on a grate which may be 6 ft. long and 3 ft. above the floor, and as the hole need not be uncovered until the trough is in a position to be inserted, no unnecessary air is admitted and the economy would be increased.

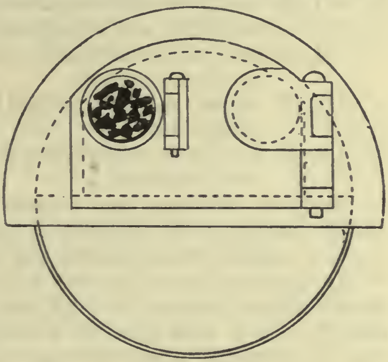

FIG. 4*a*

Air Admission above the Grate.—It is conceded on all hands that a certain amount of air must be admitted above the grate, and that it is difficult to determine how much; but it will not be out of place to illustrate the subject by drawing attention to the behaviour of the flame of an ordinary paraffin lamp. If the wick is turned too low disagreeable smells are produced, filling the room in a very few minutes; if too high, other smells and much soot are produced. In the one case we have incomplete combustion, due to the cooling action of an excess of air; in the other case too little was supplied. Smoke and smells are emitted from funnels, showing that here too unconsumed gases are escaping; for carbonic acid, as well as steam, is without odour.

With present arrangements it is almost impossible to regulate the air supply, or at any rate to fix it, so that the ratio of the excess to the total shall have a definite value, and consequently we find in practice that it varies from 25°/ₒ to 200°/ₒ. (See pp. 94 and 101.)

It is clear that when the fires are thick less air passes through them, less fuel is consumed, and relatively more air is drawn in through the doors. With thin fires the draught through them is stronger, more fuel is burnt, and less excess air drawn in. Now, although the products from thick fires contain more inflammable gases than those from thin ones, and consequently require more excess air than these, it is difficult to believe that they will receive just their correct share, whatever the sizes and number of holes in the doors, whatever the state of the grate, and whatever the intensity of the draught or the quality of the coal; and it is only reasonable to assume that the best results will be obtained—

I. If no air is admitted through the doors when the fires are very thin and all aglow.

II. If much air is admitted immediately after coaling and when the fires are thick.

III. If more air is admitted with North-country than with Welsh coal.

IV. If less air is admitted as the fires get dirty and the combustion is reduced.

But even with the most correct proportions the combustion will not be perfect, if for no other reason than that the flame comes in contact with the boiler plates, and gets cooled, before it is burnt out.

As regards the excess air, it is perhaps of more importance to decide where to admit it than how much to admit, provided it be enough. If it were possible to keep the fires very thin, it might be well to admit all the air through the bars, and none above the fire; but this is impossible. To admit air through the furnace doors stimulates combustion at this point, so that, before mixing with the distilled furnace gases, it has already been robbed of much of its oxygen. Just after coaling this is not the case; but being even colder than the products of distillation, it will cool instead of igniting them. As already mentioned, this result can be evaded by throwing the green coal only on the front ends; but another plan is to have much brick-work at or behind the bridge, which soon grows very hot, and the mixed gases ignite on coming in contact with it, and burn in the combustion chamber. Unfortunately, the distance to the tubes being short, the flame is immediately extinguished on entering them.

The plans have also been tried of admitting air at the bridge, or through a tube passing from the shell direct into the combustion chamber; but both are open to many objections, not the least being that the passages get choked and are certainly not under control, and can neither be cleaned nor closed when desired.

With those systems of forced draught in which the ashpits are closed, the funnel damper can be set so as to retard the draught above the bars to such an extent that just the right amount of air is admitted, even when the doors are wide open.

Flames.—Interesting experiments have been made to show that on mountain tops, where the air pressure is much reduced, and also in partial vacuums and bad atmospheres, ordinary flames lose their luminosity, while under high pressures they grow smoky, and even the hydrogen and carbonic oxide flames grow luminous. From this it has been argued that draught-retarders and the closing of the funnel damper, when using forced draught, will improve combustion. But those very experiments (E. Frankland, 1877, p. 876) go far to show that the combustion is not accelerated; besides, the increase of pressure attainable by these means is so slight in comparison with the atmospheric pressure variations, that the influence would have shown itself before now if it were true that more heat can be got out of coal with a high than a low atmospheric pressure.

Some interesting facts as to the igniting temperatures of various substances will be found in the chapter on 'Fuels and Combustion.' There will also be found a good deal of information on the heating-power of fuels, and on various methods of measuring it, as well as for determining, from the funnel gases, whether the combustion is perfect or not. Some of these tests are so simple that they can easily be carried out at sea.

Furnace Doors.—As regards economy of fuel, furnace doors unquestionably rank amongst the most important attachments to a boiler, and innumerable are the patents in connection with them; but the very fact of numberless ideas having been published makes it impossible to deal with any of them as exhaustively as might be wished, and only those will be mentioned which have found their way into the stoke-hole.

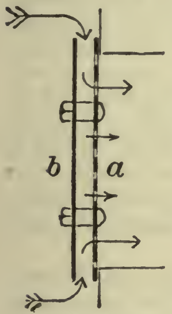

Fig. 5

The door may consist of a single plate, fitted with hinges and a latch, and perforated, chiefly at the upper edge, or fitted with a gridiron or other contrivance for regulating the admission of air. The objection to this arrangement is that, on account of the direct radiation from the fuel, the door very soon gets excessively hot and warps, and even cracks. The presence of an air regulator is an advantage, if properly used, but the very reverse if its action is not understood (see p. 5). Some doors are hinged, so that they can be kept partly open. In order to protect the door from the heat, an inside screen should be fitted, which can be renewed when burnt away; for, as it cannot be kept as cool as the outer one, it suffers more severely. A simple arrangement is to rivet a plate, *a*, fig. 5, having a number of holes, to a solid one, *b*. The air then enters at the circumference and passes through the various holes, as shown by the arrows.

In other cases the internal plate is so fitted that the current of air is directed either upwards or downwards, according to the views of the respective engineers. (See figs. 6 and 7.) An idea seems to prevail that by leading the air through complicated passages it collects heat, thereby facilitating combustion; but this warming is so slight that it does not justify expensive arrangements. Wide dead plates in combination with doors which admit air only at the top keep the latter cool. As already mentioned, hinges are sometimes constructed so as to keep the doors partly open; and certainly they should all be arranged so that they will keep quite open when coaling, particularly during rough weather. Some of the contrivances used for this purpose are very simple, as

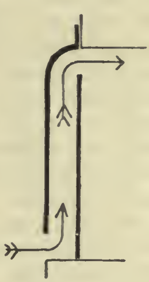
Fig. 6

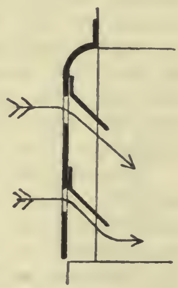
Fig. 7

well as fairly efficient. All of them should be strong, and capable of being worked in the easiest possible manner, for a fireman's chief tool is his shovel. Besides, on account of the heat, anything belonging to a fire door cannot well be touched by hand. One way of keeping the doors open is to balance them by either weights or springs, but with most of these arrangements the ashpit gets closed during firing, whereby that part of the air supply which passes through the fuel is lessened, and that which passes through the door increased, doing harm where it chills the various plates.

Door Frames suffer in the same way as the doors to which they are attached, unless they too are properly protected from the heat, either by baffle plates with air admissions at the back, or more generally by firebricks. It is never good to make the door frame in one piece, as it is sure to crack. With large furnaces, of 40 ins. diameter and above, two doors are sometimes fitted. Attempts have also been made to fit feed heaters at these points, but evidently without success.

Fire Bars.—It cannot be said that this subject has been neglected by inventors, for the patents in connection with it are innumerable; but it is certainly very unsatisfactory that, after trying various novelties, engineers always fall back on the old pattern, viz. length $2\frac{1}{2}$ to 4 ft., air spaces $\frac{1}{2}$ to $\frac{3}{4}$ in., depth $3\frac{1}{2}$ to 5 ins., and thickness $\frac{3}{4}$ to 1 in. at top, tapering 1 to $1\frac{1}{2}$ in. per foot of depth. Even for forced draught, if not excessive, the above dimensions give good results, although the Admiralty, who use wrought iron or steel instead of cast iron, make the bars about $\frac{5}{8}$ in. thick, while the air spaces are reduced to $\frac{3}{8}$ and even $\frac{1}{4}$ in.

It has been argued that by reducing the upper surface of the bars a smaller area is exposed to the heat of the fire, and that such bars will keep cooler; but a glance under the grate of a furnace is sufficient to convince anybody that as much if not more heat is received by radiation by the sides of the bars as by their upper surfaces. A closer examination will also show that each bar is surrounded by a visible layer of trembling hot air, which is moving upwards and seems to be about $\frac{1}{16}$ in. thick. It is the heating of this thin film of air which keeps the bars cool. These facts might lead to the conclusion that a distinct advantage would be gained by reducing the thickness of each bar, and fitting more of them, because thereby relatively more cooling surface is obtained. But if, as seems necessary, the air spaces are left as wide as before, each bar will receive an extra amount of heat, so that the thin bars will probably grow quite as hot as the thick ones, and if that is the case they are at a great disadvantage, for being thin they are sure to get bent sideways. In fact, the only way to use them is to pack them tightly into the furnace, so that they can neither bend nor twist.

Where great trouble is experienced water in trays is placed in the ashpits. This seems capable of abstracting sufficient heat, but the air channels under the bars are seriously reduced, and salt water in the furnaces is not a desirable object, as it causes corrosion, though on forced-draught trial trips it is often necessary to spray sea water into the ashpits. No doubt the accumulation of ashes and red-hot small coal in the ashpits keeps the bars hotter than they should be, and, as they also seriously interfere with the draught, it would be a great advantage if means could be devised for removing them. Steam blasts are used ashore to keep the bars cool.

Another plan for keeping the bars cool is to make them deeper. Heat travels so very quickly in metals that the small extra distance which it has to go before reaching the lower edge hardly affects the result, which, roughly stated, is, that the temperature of the bars is inversely proportional to their depths, or more correctly to their exposed surface, and that their rigidity (horizontally) is proportional to their depth and to the square of their thicknesses. The horizontal deflection of a bar, to which a definite curvature has been given, is proportional to the square of its length. These views lead to the following formulæ, with whose help the small table has been compiled. If not numerically correct, it can at least be used for making comparisons.

I. For a given coal consumption, and for a given length of fire bar, the sum of the sectional areas of the bars contained within 12 ins. of the furnace diameter should be a constant value.

$$n.t.d. = C_1.$$

II. For a given coal consumption the sum of the sectional areas of the bars contained within 12 ins. of the furnace diameter should be proportional to their lengths.

$$n.t.d. = C_2.l.$$

III. For equal lengths of fire bars the square of their depths should be proportional to the weight of fuel burnt per square foot per hour.

$$d^2. = C_3.Q.$$

In the above formulæ the various letters have the following meanings:—

n stands for number of fire bars per foot of furnace diameter.
t stands for thickness of fire bars at their upper edges.
d stands for depth of fire bars at their centres.
l stands for length of fire bars.
C_1, C_2, C_3 are constants.
Q is consumption of coal per square foot per hour.

Values of the Products n.t.d. (This is the Sum of the Sectional Areas in Square Inches of Cast-Iron Fire Bars per Foot of Furnace Diameter.)

Coal Consumption per Sq. Ft. of Grate per Hour	20 lbs.	40 lbs.	80 lbs.	100 lbs.
Length of fire bars = 6 ft.	72	100	—	—
" " 5 ft.	40	55	80	100
" " 4 ft.	32	45	64	80
" " 3 ft.	24	34	48	60
" " 2 ft. 6 ins.	20	28	40	50
" " 2 ft.	16	24	32	40
" " 1 ft. 6 ins.	12	17	24	30

With bars whose air spaces are one-half of their thickness—i.e. 4 ins. per foot of furnace front—the depths would be found by dividing any of the numbers in the table by 8 ins. Thus, with bars 2 ft.

6 ins. long, burning 40 lbs. per hour, the number is 28, and the minimum depths of such bars would be $3\frac{1}{2}$ ins., while the thickness might be made $\frac{3}{4}$ in., with $\frac{3}{8}$-in. air spaces, or $\frac{1}{2}$ in. with $\frac{1}{4}$-in. air spaces.

In cases of forced draught these two values are very often equal—say $\frac{3}{8}$-in. bars and $\frac{3}{8}$-in. air spaces. To obtain the depth in their case, the number in the table would have to be divided by 6 ins., so that with a consumption of 40 lbs. the 2 ft. 6 in. bars would have to be $4\frac{2}{3}$ ins. deep, and 18-in. bars would have to be 3 ins. deep.

Naturally these values are only approximate, and depend very much on the fuel, but they may serve as a guide when making alterations.

Furnace Diameters.—It will be noticed that as soon as the usual practice is departed from, either by increasing the length of bars or the coal consumption, then their depth grows so great that it seriously interferes with the draught. This influence is particularly noticeable in boilers with small furnaces. Compare, for instance, two flues, the one being 33 ins. in diameter and the other 48 ins. If, as is usual, the lines of the dead plates pass through their centres, then the sectional areas below and above these lines are 3 sq. ft. in the one furnace and 6·3 sq. ft. in the other. With five bars which are 3 ins. deep, the ashpit areas are reduced to 2·3 and 5·3 sq. ft. respectively, or ·85 and 1·33 sq. ft. per foot of furnace front. Under ordinary conditions this means that in the one case the air entering the furnace has to travel with a velocity of 12 ft. per second, in the other case its velocity is only $7\frac{1}{2}$ ft., and the resistances would be as 3 to 1. An extra inch added to the depth of the bars would increase the one resistance 25%, and the other only about 6%, so that in cases where the performance is low there is much less chance of efficient alterations if the furnaces are small than if they are large.

The sum of the air spaces between the fire bars usually amounts to 33% of the width of the grate, so that if the length is 5 ft. we have $1\frac{2}{3}$ sq. ft. of air passage for every foot of furnace front. This being about twice as large as the ashpit area of the small furnace, little improvement would be effected in the draught and combustion of the small furnace by giving wider air spaces, for the depth of the bars would also have to be increased, whereas in the large furnace, where the ashpit area is sufficient, the alteration might increase the combustion.

On trial trips, and whenever it is desired to obtain the highest performances, the ashpits should be kept clear of ashes at all times, for every inch of piled-up material restricts the draught. Ash ejectors might be used with advantage.

With forced draught the case is different. Except on such trial trips, where the air pressure is limited, there is practically no restriction as to the pressure which may be applied, and all the above reasons for allowing large air spaces fall to the ground, and there seem to be no objections against reducing them to the very narrowest limits. In fact, very good results are said to have been obtained in some foreign vessels where the air spaces have been reduced to $\frac{1}{25}$ in. and a high-pressure blast introduced into the ashpits. The necessity for opening the lower doors to remove the ashes does not exist, for none can fall through; they are all fused and form clinker on the bars. Of course such bars could not be used for natural draught; but natural draught in the Navy is simply another name for a very low fan pressure, which could be increased if desired.

FIG. 8

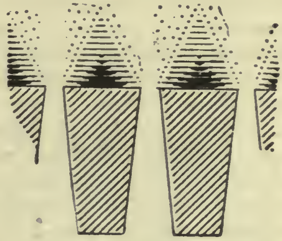

FIG. 9

The following advantages should not be overlooked:—By reducing the air spaces till they are mere slits, the chief resistance to the air passage is found at these points and not in the fuel, and the draught will hardly be affected by considerable variations of thickness of the fires. The chief combustion takes place just over the slits, and little, if any, over the centre of the bars, which remain covered with comparatively cool fuel or ashes, and are therefore not exposed to so much heat as the narrower bars with wide air spaces. An attempt to indicate this difference is shown in figs. 8 and 9.

The Burning of Fire Bars is often the consequence of irregularities in the upper surface of the grate. It is but natural that if several air spaces are blocked (fig. 10), either by clinker or when the bars are bent, then the air can cool only one side of such bars, and these must grow hotter than the others. If, in addition, fuel should get wedged into the wide space, nothing will prevent the corners of the bar from burning, and as these waste away the fuel sinks lower and lower, and it is only a question of time as to when it will have effected the destruction of the bars; naturally the adjoining ones, which are then exposed to the same action, will suffer in the same way.

FIG. 10

FIG. 11

An examination of the grates of a boiler which has been worked hard will show that the bars have all sagged more or less, and that some are bent sideways, but that the upper edges are all uniformly wasted away, presenting a regular surface. On removing and then replacing them this condition is entirely altered. Possibly bars from the sides have got placed near the centre, and *vice versâ*, and a section across the grate might have the appearance of fig. 11, with the inevitable result that the tops of the projecting bars would be quickly burnt away. Possibly this may lead to the worst ones falling out, but at any rate the bars will generally be more quickly wasted if placed together in this manner than if kept level. Where the grate is made up of two lengths, it is always best to place any new bars at the back end.

Ashes and Clinker.—Another great trouble, and one which, like all others, is aggravated under the conditions of forced draught, is the accumulation of clinker on the grate. Each ton of coal contains from $\frac{3}{4}$ cwt. to 3 cwt. of ashes, according to the quality, and although a portion drops into the ashpit, a considerable part remains on the grate. It also depends upon the nature of the fire and on the composition of the ashes whether much—say $\frac{2}{3}$—is blown up the funnel, or whether only a small quantity—say 25%—is disposed of in this way. If the fires are dull and the ashes refractory the latter remain dust and are carried away; but if they melt easily, or if the fire is a fierce one, they form clinker, which remains.

The following minerals are found in ashes of coal :—

Silicic acid	from	2%	to	60%
Calcic oxide	,,	1	,,	20
Ferric ,,	,,	15	,,	75
Alumina	,,	2	,,	40

The most refractory ashes are those containing the smallest quantities of silicic acid, and the most fusible those where it amounts to about 50%, with 25% calcic oxide. If the alkalies were absent the ashes would remain dust under the most trying circumstances. Salt increases the fusibility of clinker, and the hydrochloric acid, which it loses while melting, is a very active corrosive agent and injurious to the boiler plates. Sea water should, therefore, not be mixed with the coal, and leakages in the furnaces and combustion chambers should be prevented as much as possible.

Two qualities of coal, whose respective ashes are fairly refractory, will occasionally produce very fusible clinker, if used together. In such cases the one probably contains an excess of silica, and the other an excess of the other ingredients. The melting away of the firebricks of the bridges is sometimes due to the great heat, but more often to an excess of the alkalies in the clinker. They readily attack the silicic acid, which is the chief constituent of the firebricks. Basic bricks would not suffer under these circumstances, but cannot be used at sea, as they readily absorb moisture when out of use, and then fall to pieces.

Cinders.—When the ashes, which have not been blown away, melt, they trickle down the sides of the partly consumed small pieces of coke, and by glazing them prevent their quick combustion. This may be a slight advantage, as the point of highest temperature of the fire (see fig. 105, p. 97) is thereby raised a little above the grate surface. When these small pieces of fuel ultimately find themselves over an air space between two bars, they drop through and are wasted; those which, at the end of their downward motion, arrive on the top of a bar are consumed there; therefore, if a sufficient amount of forced draught is available, it would be more economical to reduce the air spaces to the smallest possible dimensions.

Clinker.—The slag which had adhered to the small coal is now resting on the fire bars, and congeals, or at any rate its lower part, as well as its extremities where it is in contact with the comparatively cool air.

As more and more slag is added, troughs of thick slag are formed, whose edges project over the air spaces and gradually close them (fig. 12), seriously interfering with the draught and combustion. With worn bars (rounded tops) the closing is effected more quickly, as the

Fig. 12

Fig. 13

slag trickles down the sides (fig. 13); and the very fact that this happens is a sign that the bars are equally hot; and this is not to be wondered at, for the slag is sure to be very thin on the top, affording no protection from the heat, and is very thick at the sides, preventing the cold air from cooling the bars.

Fig. 14

The necessity for guarding against this trouble has in America been the cause of the adoption of fire bars with hollow tops (fig. 14). The channels are soon filled with ashes or clinker, and it is only their two edges which absorb heat from the fire. It is stated that these fire bars, which are easily cast if the pattern is made in two halves, have a longer life than the ordinary ones, and also that the clinker does not adhere to them so firmly.

When fires have been burning for twenty-four hours they will be thoroughly dirty, the weight of mineral matter resting on one square foot of grate surface amounting to from 10 to 40 lbs., or from $\frac{3}{4}$ to 3 ins. in thickness. Of course the air spaces can then only be kept open by frequently using the slicer and pricker (figs. 17 and 19), but much can be done to break up the slag, which adheres to the bars, by moving them as indicated in fig. 15; this is only possible if they are very loosely packed, and it is customary to fit one bar less than the number which might possibly be got into a furnace diameter. They should be supported at the highest possible point, so that the air spaces at the top are not altered, but only the angles.

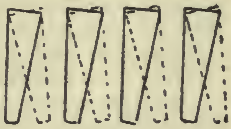

Fig. 15

In order to allow for longitudinal expansion of the fire bars, one end, where they rest on the dead plate, should be made slanting. If there are two lengths, the other slant should rest on the bridge plate (fig. 16). Any bodily motion is prevented by notching the other end of each bar where it rests on the cross bar.

If the lengths exceed 3 feet, small distance pieces, A, are cast on at the centres; they prevent excessive distortions. The bars are also made decidedly taper, 1 in. to 1½ in. per foot of depth, so that the fuel cannot be jammed between them, which would certainly happen if they were parallel and loosely fitted. The navy fire bars, which are made parallel, have therefore to be packed tight.

FIG. 16

Replacing Fire Bars.—In spite of all these precautions it will happen that fire bars burn, or under forced draught a whole grate may sometimes drop into the ashpit; but this only happens when a large amount of red-hot ashes has accumulated there. Nothing can then be done but to clear out the furnace and rebuild the grate. When only one or two bars have dropped out, these can be replaced by new ones without drawing the fires. At the front end this is an easy matter, and some men are even sufficiently skilled to be able to throw a fire bar into its right position at the back of a grate; but the safer plan is to tie it to a slicer, and then to place it in position. The yarn will very soon be burnt, and the slicer can then be withdrawn. Wrought-iron and steel fire bars, with their high melting temperatures, are said to last longer than cast-iron ones, but it is at present difficult to obtain them of a taper section, as very few rolling mills produce it, so that a large spare stock would have to be carried. Secondly, it is difficult, or rather expensive, to thicken the ends, though it would seem that with a suitable die and a steam hammer the problem could be readily solved. The plan has been successfully tried of notching the dead plate, bridge plate, and the cross bars, and using fire bars without thickened ends. In this case the air spaces cannot be kept equal at the upper edges, unless all parts are firmly fixed, which has already been shown to lead to trouble.

Cleaning Fires is one of those operations which, although necessary for the working, may cause a good deal of damage. Leakages in the combustion chambers are often attributed to the rush of cold air while the door is open and the grate bare. To close the funnel damper seriously interferes with the generation of steam from the other furnaces, and some ships have therefore been fitted with separate dampers over each nest of tubes (see fig. 30, p. 19), and the results are said to be satisfactory.

Another plan is to clean only half a fire at a time: one side of the fire is allowed to burn down, the clinker and ashes are raked out, and the bare bars covered with green coal, which is ignited by the fire at the other side, which, after a time, is treated in the same way. On account of the double operation firemen are not fond of this plan.

Another plan, which, to a certain extent, is indulged in by every fireman when burning caking coal, is to draw the clinker forward with a heavy rake (fig. 18) about one or two feet between each firing, taking

FIG. 17

FIG. 18

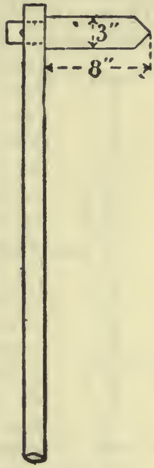

FIG. 19

particular care to throw the green coal on those parts of the grate which have been bared. The clinker should not be allowed to get cool, otherwise it will be difficult to loosen it from the bars, which is done with the slicer (fig. 17).

Removal of Ashes.—Having got the ashes out of the furnaces, they naturally fall against the boiler, and are there slaked with sea water. The noxious vapours which escape are an indication that corrosive acids are being generated, and it is only too well established that the most serious wasting to which the external parts of a boiler are subjected takes place here. Far better arrangements than those usually adopted should be made for preventing the ashes from resting against the boiler plates, or from getting into the narrow crevices which are found here. Besides cementing the inside of the end seams of boilers, it would not be amiss to cement the outsides as well where this corrosive action may be expected. When guard plates are fitted, no spaces should be allowed where ashes and moisture could find a lodgment.

Occasional efforts have been made to remove the ashes out of the stokehold by means of ejectors, and although successful for this purpose they quickly corrode, but other mechanical contrivances might be tried.

The amount of ashes to be dealt with in twenty-four hours can be roughly estimated as equal to the amount of mineral matter in the coal burnt during that time.

Mechanical Stokers.—It is unnecessary to make any remarks about mechanical stokers, as every one that has yet been tried at sea has been given up; but for those who wish to study the subject the following references may be of value:

J. Daglish, 'M. E.,' 1868, p. 155, mentions Stanley's and Vicar's patents. J. F. Spencer, 'C. E.,' 1891, vol. civ. p. 54, mentions about a dozen different systems. F. Colyer, 1886, also mentions several. New types are perpetually appearing. Chain grates are now very common for watertube boilers.

Banked Fires.—An interesting table, containing the amount of coal consumed under banked fires, will be found in C. Busley's work, 1883, p. 157, from which it appears that from two to three cwt. per furnace are consumed in twenty-four hours, and that the quantity required to raise steam is slightly less than this amount. He does not mention whether the fires were shoved back or drawn forward. The former is the more correct method, because it chokes the bridges, and the little air which passes there is hot; but, as the dampers can never be closed sufficiently, more heat would be developed than is required to maintain steam, and the more general practice is to draw the fires forward, and to leave the bars bare at the back. Much cold air is thus admitted, which keeps down the steam and reduces the draught. Of course, the boilers suffer injury under both treatments, but particularly under the last-mentioned one.

When the engines have to be stopped without previous warning, the dampers are at once closed and the smoke-box doors opened: or (but this should not be allowed) all the furnace doors are thrown open. This is certainly a very effective means of reducing the heat of the fires, but the cold inrush of air may easily cause leakage. Should the steam still be rising fast the supplementary feed should be turned on before the safety valves lift.

Fig. 20

Fig. 21

Sweeping Tubes.—On account of the soot, which adheres to the tops and sides of the tubes, and the ashes, which settle along their bottoms, it is necessary to sweep them about once a week. Wire brushes (fig. 20) are used for the soot and ashes, and the split scrapers (fig. 21) to remove salt. The latter tool is of little use at sea, because a tube which is once plugged up with salt closes up again after half an hour's steaming, and need therefore not have been cleaned. The brush should be strong enough to remove all the dust. These tools are hinged at their centres for convenience of handling. The sweepings of each tube naturally fall into the combustion chamber, but the draught at once carries the greater part into the adjoining tubes and up the funnel. In order not to get smothered the firemen keep the damper open, and very light fires on the grates, just sufficient to produce a draught. Except in large steamers, or in vessels fitted with several funnels, it is usual to clean the tubes of all the boilers at one operation, the whole watch being sent below for the purpose; for, as the draught is nearly non-existing during this time, little steam can be generated, and the sooner the operation is completed the better.

The efficiency of a boiler is seriously affected by accumulation of soot on the heating surfaces, so much so that all land economisers are fitted with scrapers which are constantly moved up and down the tubes by machinery. It is found that whenever this motion

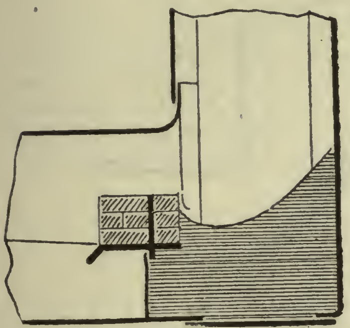

FIG. 23

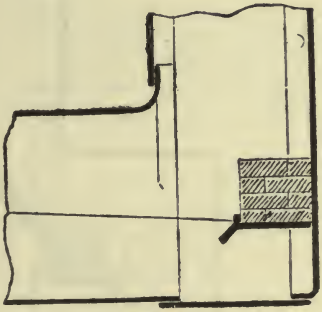

FIG. 24

ceases the efficiency of the economisers is much reduced. If a similar device could be introduced for the constant sweeping of boiler tubes, the efficiency of the tube surface would be increased, and the sizes of boilers might be reduced. Economisers probably suffer more in this respect than boiler tubes would do, because they contain cold water and are therefore likely to condense the tarry products of combustion.

Recently attempts have been made to clean the tubes by means of steam jets which are manipulated from the back ends of boilers if single ended.

Soot is also a great inconvenience to water-tube boilers, and may one day lead to similar appliances being employed on them: at present their tubes are mostly cleaned by steam blasts, which can of course only remove the loose soot, whereas it is believed that the hard tarry deposit nearest the metal is a most efficient non-conductor.

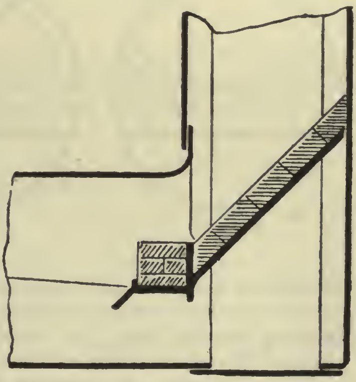

FIG. 25

Furnace Bridges.—The object of fitting bridges in furnaces is partly to keep the fuel from falling into combustion chambers, and partly to reduce the air channel, so that, by imparting a momentarily high velocity to the flame, a thorough mixture of its gases is effected, and the combustion completed. Brick fire-bridges are arranged as shown in figs. 23-26. The space at the back of the bridge shown in fig. 23 is usually quite full of ashes, even though a door is fitted at the bottom by which they could be removed.

The heat which the firebricks collect may occasionally do a considerable amount of harm to the plates against which they rest if, as sometimes happens, the water is let out of the boiler before it is cold.

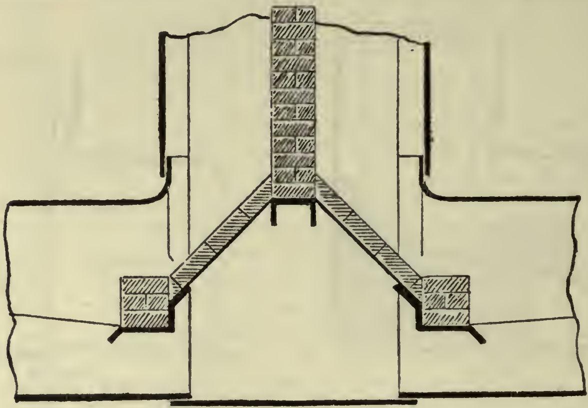

Fig. 26

Viewed from the front, the bricks are generally placed horizontally; but where much trouble is experienced by their falling out of place, which happens when clinkers are allowed to adhere to them and have to be broken, or when the fires are forced, or when the furnaces are large, the bricks are sometimes arranged in the form of an inverted arch (fig. 27). In double-ended boilers, with through combustion

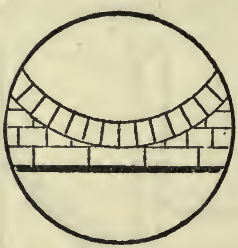

Fig. 27

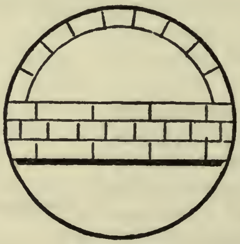

Fig. 28

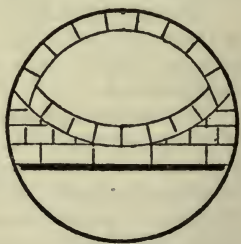

Fig. 29

chambers, the same plan has been tried on the central partition wall; but no advantage seems to have been gained, and where it is desired to carry the wall high up, it is safer to bolt two angle irons down the sides of the combustion chamber.

In cases where the furnace saddles have been cropped, or where the tube plate has been flanged to meet the furnace, exposing the back end seam to the direct action of the flames, it is sometimes necessary to protect it by a firebrick arch (fig. 28). Here, too, it is an advantage if the bridge is curved, as it gives a better support to the arch. (See fig. 29.)

Dampers.—One damper is usually fitted to each funnel, rarely one damper to each boiler, and still more rarely one damper to each furnace, but damper plates are provided for each ashpit. Ashore a pair of dampers is always fitted to each boiler, so that the air admission over the bars is reduced to a minimum during the cleaning of fires. With closed stokeholds the inrush of air during these cleaning periods is very great, and in many cases an arrangement similar to that shown in fig. 30 would be an advantage.

If forced draught is used with closed ashpits, dampers are not necessary, for the suction above the bars is not always equal to the pressure which is transmitted from below. In fact, when firing, it is customary to restrict the blast, because otherwise so much gas is generated that the funnel draught is unable to carry it off, and the fumes would enter the stokehold.

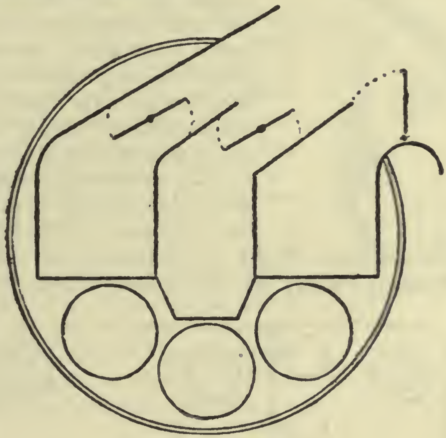

FIG. 30

Flames in Stokeholds.—The most series mishaps may be caused if steam is suddenly generated in the combustion chambers, which is a powerful reason why fusible plugs should not be fitted. With some boilers in the Navy sad loss of life has several times been caused by sudden leakages of the back ends of the tubes. It has been suggested that this happened with H.M.S. 'Barracouta,' but a similar effect might be produced by stopping the fan in one of the stokeholds, the flames being at once driven into it over the bridge by the fan in the other stokehold, while the funnel is unable to draw them away. The difference of pressure would no doubt be increased by men anxious to render assistance blocking the stokehold passages. Similar accidents have also happened in single-ended and in navy-type boilers, in which this action is impossible.

Alternate Heating and Cooling of Plates.—Very exhaustive and interesting experiments on this subject were made by E. Wehrenfennig ('Organ,' 1884, vol. xxi. p. 216, &c.). He found that heating various metals and cooling them had certain effects, which were reproduced on repeating the experiment, and which were intensified the longer the heating lasted and the higher the temperatures. His limits were boiling water and red heat. (See p. 30.)

His results are that steel and iron bars and plates shorten, but grow thicker if heated and cooled, about $\frac{1}{20}$ °/₀ to $\frac{1}{10}$ °/₀ for one red heat. Cast-iron and copper bars and plates lengthen, but grow thinner if heated, say, $\frac{1}{20}$ °/₀ and $\frac{1}{5}$ °/₀ respectively. Col. H. Clerk ('Proceedings,' 1863, vol. xii. p. 452) and H. Caron ('Comp. Rend.,' 1863, vol. lvi. p. 828) arrive at conclusions opposed to the above, but in their cases the cooling was effected suddenly.

E. Wehrenfennig also refers to experiences in connection with locomotives—for instance, that certain fittings grow tighter after a short use if exposed to heat; that iron nuts cannot be unscrewed from copper stays after heating, but brass nuts can; that blisters crack on the fire side, being exposed to greater changes of temperature; that rivet holes of seams in iron or steel furnaces crack, in copper plates they do not do so. Possibly his most interesting fact is that a land boiler contracted so much in length that this led to various troubles about the fittings. On the other hand, land boiler furnaces slowly lengthen, sometimes as much as $\frac{1}{2}$ inch in 30 ft.

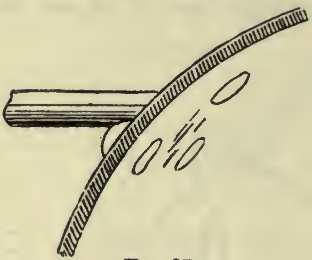

FIG. 37

The same thing evidently happens with the through combustion chambers of double-ended boilers, and cases have repeatedly occurred where, on removing the rivets in the circumferential furnace seams, the holes were found to be somewhat

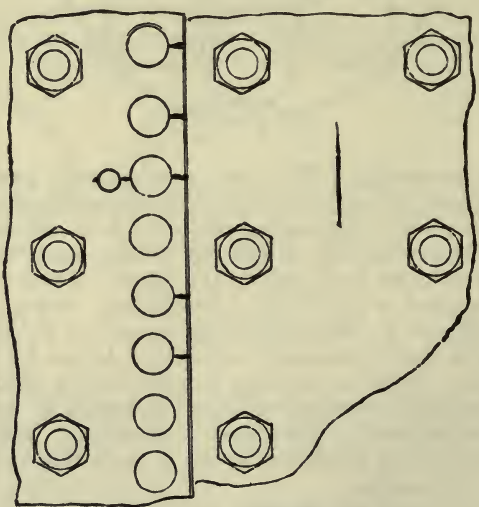

FIG. 38 FIG. 39

blind, although drilled in place, and then, having been rimered fair and re-riveted, they soon grew as blind as before.

Even flanged furnace saddle seams show the same wandering of rivet holes, but, for fear that this statement might be used to shield bad workmanship, it is as well to point out that in all such cases there is a regularity in the blindness of the holes which does not exist if the holes were drilled carelessly out of place.

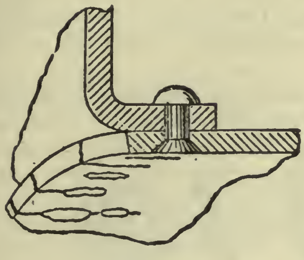
FIG. 40

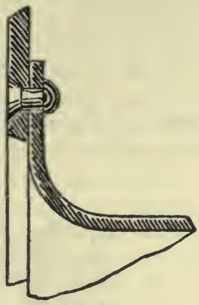
FIG. 41

It has been suggested that the cracking of rivet holes is caused by the carbonising of the iron and steel, due to their contact with fuel or hydrocarbons of the gases; but as an addition of carbon increases the volume of the metal, it is difficult to see how this can set up any tension stresses. Recent experiences suggest that these cracks are more likely to occur in steels rich in phosphorus and nitrogen than in purer ones. The four accompanying sketches show cracks as they occur under palm stays (fig. 37), in the flat parts of combustion chamber backs when covered with scale (fig. 39), in combustion chamber back plate seams (fig. 38), and in furnace saddle seams (fig. 40).

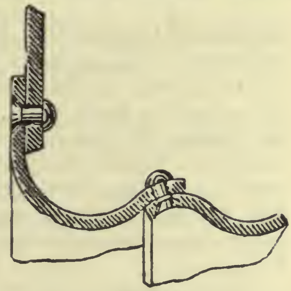
FIG. 42


FIG. 43


FIG. 44

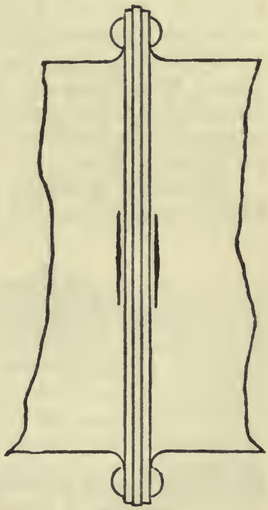
FIG. 45

Deterioration of Materials.—That this cracking of rivet holes is due more to the heat than to the impinging action of the flame, is shown by the fact that the joint of the tube plate with the furnace saddle does not give trouble if kept out of the fire, even though, as in fig. 41, the landing is exposed to the impact of the flame, while joints which are removed from this action by the depth of a furnace corrugation (fig. 42) suffer. Cracks like those shown in figs. 43 and 44 are due to severe alternating strains.

Seams in Furnaces.—Riveted joints should not be exposed to the direct action of the flame. Where this cannot be helped, as in

Fig. 46

the case of patches, the greatest care should be taken to keep the seams cool. Scale ought not to be allowed to accumulate there, and the two plates should, if possible, be brought into metallic contact, by removing the black scale, by thoroughly washing away the oil used for drilling (for this is one of the worst conductors of heat), and by fitting and bolting the plates so firmly together during riveting that caulking is almost superfluous. It is perhaps due to the more perfect contact between the plates that double-riveted seams behave better than single-riveted ones when exposed to the direction of the flame.

Experience has shown that for furnace saddle seams all these precautions are unavailing under the action of forced draught, and it has been found that when they must be repaired in this way, instead of using countersunk rivets, very large snap-headed ones should be fitted (fig. 46), with the object of reducing the exposed surface of the outer landing as much as possible, and at the same time carrying off all the heat through the rivets into the water. (See p. 120.) Seaton quotes Wye Williams, 'N.A.,' 1894, v. 35, p. 284, as having fitted 6-inch studs through his furnace plates, and found that the fire ends burnt off to a length of $2\frac{1}{2}$ inches, so that the much shorter rivet heads can take no harm.

Whenever possible, patches should be fitted on the fire sides of defective plates, so that when the rivet holes crack, as shown in fig.

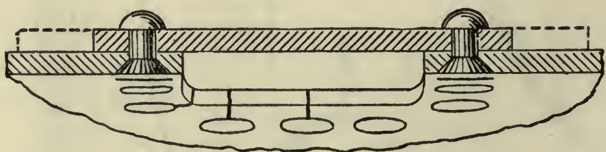

Fig. 47

47, it will not be necessary to cut out more material and fit a larger patch, as shown in dotted lines, when a renewal is necessary.

Cracked Rivet Holes.—One of the chief causes of these rivet holes cracking is scale, and whenever possible it should be removed from these patches. In the chapter on 'Heat Transmission' explanations will be found why the general efficiency of a boiler is not seriously affected by accumulations of scale, but that this substance will cause a considerable rise in the temperature of the plates it covers, which leads, as has just been explained, to cracking or to other troubles, such as leaky tubes and collapsed flues. The idea that scale prevents corrosion may still prevail, and may even be a true one; but the more rational view, that there are other and better means of stopping this waste, is gaining ground, and evaporators, for the supply of distilled water, are coming more and more into use. The removal of one evil often produces another, and recently much damage has been done when using fresh water by oil deposits, which, like all scales and other non-conductors, effect a considerable rise in the temperature of the plates they cover. This necessitates the addition of oil filters, which are inserted between the pumps and the boilers. A description of one of these, together with numerous facts and analyses of boiler deposits, will be found in a paper read by Mr. Edminston, 'N. E. C. I.,' 1892, vol. viii.

Recently the design of these filters has been much modified, and very favourable results are being obtained by means of appliances which separate the grease from the steam.

Boiler Scale.—Sea-water should not get into a marine boiler, but when it does enter, either by accident or intentionally to form a protecting scale, it meets with grease and iron rust, its own production, and the result is scale and deposit. Professor V. Lewis, 'N.A.,' 1891, v. 32, p. 67, deals with such a case, and gives analyses of scale from various parts; these have been recalculated in terms of sulphate of lime—calcic sulphate.

Position of Scale or Deposit	Scale. Furnace Top	Scale. Furnace Bottom	Scale. Tubes	Deposit Tubes	Deposit Boiler Bottom
Calcic sulphate . . .	100·0	100·0	100·0	100·0	100·0
Calcic carbonate . . .	6·9	10·2	8·2	7·0	0·0
Magnesic hydrate . . .	3·3	19·1	27·7	193·0	31·4
Iron, alum, &c. . . .	2·8	4·8	14·7	78·6	155·0
Organic matter. Oil . .	3·8	33·1	41·4	433·0	124·2
Moisture	0·9	1·9	2·2	36·8	25·7
Alkalies	0·0	0·0	2·0	15·5	8·0

It will at once be seen that the composition of the scale depends largely on the position from which it has been taken. Thus, for some reason or other, calcic carbonate is not found in the deposits on the boiler bottom, while grease is found chiefly in the deposits on the tubes, showing that it has a buoyant effect on the constituents to which it attaches itself. It seems to have most affinity for magnesic hydrate and also much affinity for iron rust. When the latter predominates the compound is so heavy that it sinks to the boiler bottom; when magnesia is in excess a deposit is formed on the tubes. Magnesic hydrate, as is well known, is the very dangerous deposit known as 'floury deposit,' and is generally due to an excess of alkalies, which, as will be seen, seem to accompany the magnesia in a fairly constant proportion. The buoyant action of the grease shows itself in the scale on the furnace bottom and on the tubes. It is strange that there is so little grease on the furnace top—possibly heat has driven it away; but then, one would expect to find an excess of magnesia.

The Effect of Scale.—It is generally believed that scale causes collapses and bulges simply because it is a bad heat conductor; but a recent experience ('M. S. U. A.,' 1899, p. 6) throws some unexpected light on this subject. An accident is there described in which a furnace collapsed and rent although the scale was intact over the bulged part. Near the rent there was another bulge which could only be seen from the fire side; on the water side it was covered by an intact sheet of scale about $\frac{3}{8}$ in. thick. The hollow space between the scale and the bulged plate must, while the boiler was at work, have been filled with superheated steam, which of course is an excellent non-conductor of heat. When once the plate separated from the scale the hollow thus formed would effectually shield the plate from the cooling contact with water, and it would grow red hot and bulge. This experience suggests that hard scale is more dangerous than soft scale, also that irregular working of boilers, which, on account of changes of temperature, would cause the hard scale to separate from the iron plate, should be avoided; and further, that a boiler which has once been emptied of water should not be refilled before at least all loose scale has been removed. The suggestion that hard scale may be more dangerous than soft scale derives additional support from another accident to a boiler, whose furnace crowns were covered 2 inches deep with a thick muddy deposit; the sides were of course bare, yet it was the sides which collapsed when the water grew too dense through salt.

Sea Water.—A most elaborate series of chemical analyses of sea water has been carried out by Prof. W. Dittmar for the 'Challenger' Report; see article Sea Water, 'Encycl. Brit.,' 9th edition, v. xxi., p. 611, from which it appears that sea water consists of

Chloride of Sodium	3·5990 per cent.
Chloride of Magnesium . . .	·5034 ,, ,,
Sulphate of Magnesia	·2192 ,, ,,
Sulphate of Lime	·1666 ,, ,,
Sulphate of Potash	·1141 ,, ,,
Bromide of Magnesium . . .	·0100 ,, ,,
Carbonate of Lime	·0160 ,, ,,
Total . .	4·6283 ,, ,,

Sulphate of lime is practically the only constituent of sea water which goes to form boiler scale, as can be seen by referring to the several analyses mentioned above. Oxide of magnesium is present in small quantities, say from 5 to 10 % of the sulphate of lime. The moisture in the scale should of course not be counted, nor the iron rust, which is due to the wasting of the boiler material. It is thus evident that boiler water contains nearly all the sea water salts except the sulphate of lime. These salts are: common salt, chloride of potash and sulphate and chloride of magnesia. The weights of these salts per 1,000 parts of water will be approximately as follows:—

Chloride and Bromide of Sodium and of Potassium about	36·0
Magnesium Chloride	5·0
Magnesium Sulphate	2·2
Sulphate of Potash	1·1
Total	44·3

In former days these soluble salts, which were of course being concentrated in the boiler, often led to overheating, and even now with leaky condensers or negligent feeding such accidents sometimes occur. Engineers are generally instructed not to allow their boiler water to exceed $\frac{4}{32}$, which is about four times the density of sea water; but salt water does not commence to crystallise until the density when cold exceeds 1·2045 or about $\frac{8 \cdot 4}{32}$. As will be seen from the following table, the amount of salt dissolved by water increases with the temperature, and at 350° F.—a not uncommon one in modern boilers—the weight of salt in water probably exceeds 57 ounces per gallon, whereas when cold it falls to about 50 ounces.

The following table has been calculated from Dr. H. Landholt and Dr. R. Bornstein's tables, 1912, p. 485.

Saltness of Water at Saturation Point

Temperature		Amount of Salt				Density when Cold
		Added to Water	Contained in the Solution			
° F.	° C.	%	%	Oz. per Gallon	$\frac{}{32}$	
32	0	35·5	26·2	50·3	8·4	1·2027
50	10	35·8	23·3	50·5	,,	1·2036
68	20	36·0	26·4	50·7	,,	1·2045
86	30	36·3	26·6	51·1	8·5	,,
104	40	36·6	26·8	51·4	8·6	,,
122	50	37·0	27·0	51·8	,,	,,
140	60	37·2	27·2	52·2	8·7	,,
158	70	37·9	27·4	52·6	,,	,,
176	80	38·2	27·6	53·0	8·8	,,
194	90	38·8	27·9	53·6	8·9	,,
212	100	39·6	28·3	54·4	9·0	,,

Salt Deposits.—The beautiful salt crystals which are found in boilers, in which the density was so great as to cause collapse, are not the cause of the accident; they have been formed during the period of cooling. The real injury has been done by the dense salt scale, which contains only fine crystals, or still more probably by a local deposit of salt which may have re-dissolved. One case is known in which the furnaces did not collapse nor showed any signs of weakness, although covered with large salt crystals about ½ in. cube. The density must have exceeded 51 ounces per gallon. This boiler was not worked hard.

The density as detailed in the above table is not a perfectly correct guide, for boiler water also contains magnesium salts, which add to the density, but which will not crystallise out at almost any temperature; therefore the percentages given in the table, which refer only to pure salt water, should be multiplied by $44{\cdot}3 : 36{\cdot}0 = 1{\cdot}23$ (see table, p. 25), so that in modern high-pressure boilers the water must, at least locally, contain about 70 ounces of various salts before deposits occur. These remarks are not intended to encourage the use of high-density water, but to enable engineers to fully understand such cases. Collapses have certainly occurred before the density reached even $\frac{8}{32}$, but this may possibly have been the result of purely local deposits of salt, due to hard firing and bad circulation. On standing idle after the accident the salt would dissolve away and leave no evidence except the harm it has done.

Floury Deposit consists chiefly of magnesia, and is formed in boilers if their waters are treated with caustic soda, generally for the purpose of preventing corrosion. This chemical certainly deposits the chloride of magnesia, and unless it is carefully introduced, and only in small quantities, the danger of a collapse is great. If there is grease in the boiler, the magnesia and the grease and water combine to form a substance of the same density as water, which can and does adhere to any part of the boiler and may cause overheating. See p. 122.

Grease in Boilers.—After the surface condenser came into use, engineers were troubled with furnace collapses, which are now explained as being due to deposits of grease, especially as it is known that grease is a ten times worse conductor of heat than scale; but even now the subject is a mysterious one, for whenever a furnace does collapse, due to grease in the boiler, the collapsed part never shows any grease at all, and even the other parts of the boiler are generally only covered with sufficient grease to dirty one's clothes, its thickness being probably less than $\frac{1}{1000}$ inch. Seeing that this film disappears from the collapsed parts, the only explanation which at present suggests itself is that at the temperature which exists in a boiler this grease changes its nature and becomes capable of forming tough bubbles wherever the heat is greatest, while either hydro-carbon vapour or super-heated steam is formed between the boiler plate and the skin of the grease bubbles, thus preventing the water from coming in contact with the already red-hot plate and cooling it. One objection to this view is that grease acts in a very positive manner, it being found that if one surface in a boiler has been overheated due to grease, so have the others. In one case nine furnaces in three boilers came down together; it also seems that the growth of collapses is a slow one, extending sometimes over days and even months. When furnaces collapse due to scale they do this very suddenly, and generally, too, it is only one furnace in a boiler which comes down at a time.

Grease and Scale.—It has been noticed that grease alone is far more dangerous than grease and scale together; probably the grease adheres so firmly to the scale that it cannot form bubbles. This may account for the experience that the furnaces of new boilers have now and then come down, sometimes even on trial trips. In all such cases the cause was grease without scale. On the other hand steamers are running without mishap whose boilers are coated with scale, and in addition with a filthy covering of grease.

Collapsed Furnaces.—One of the greatest mishaps to a boiler which can occur at sea is undoubtedly the collapsing of its furnaces. Generally, but not always, this is due to ignorance or carelessness of the engineer in charge. For instance, it may happen that things have been left in the boiler which, by means of the water circulation, are landed on some plate exposed to the direct action of the flame, and may cause a local bulge, a ludicrous instance being where the head of a dead horse had been put into a Cornish boiler to prevent corrosion, and had settled on the furnace crown. With steamers it has happened that while ashore, the supplementary feed which had to be put on carried with it mud, sand, or infusorial earth, and this settled on the furnace saddles, or near the upper part of the flanges of Adamson's rings. Under specially favourable conditions the bulges which have then been produced were shaped similarly to the section shown in fig. 48.

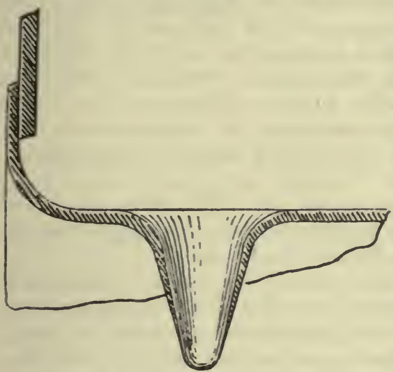

Fig. 48

A very serious case of collapsed furnaces occurred in a ship with four elliptical boilers. One of them had given trouble, had just been repaired at sea, and when put into use again, all the furnaces in the three other boilers came down together. No doubt the indicated pressure of the repaired boiler was fictitious (see p. 2), and due to air in the steam space, although the water was still fairly cold. On connecting the boilers it disappeared, and was perhaps reduced to a partial vacuum, into which steam suddenly rushed from the other boilers. The consequent excessive ebullition, aggravated as it was in these boilers by very restricted water spaces, and very little clearance below the tubes, must have raised the water away from the furnace crowns, and these collapsed.

Very mysterious collapses are sometimes caused by emptying and refilling a boiler quickly in port, without removing the manhole doors or cleaning its inside. It would appear as if, under these conditions, the scale on the tubes falls off, lodges on the furnace crowns, and, as it is not removed, causes overheating. It also happens that on blowing out the boiler, the scum and oily matter which floated at the water level adhere to the heating surfaces, even when the new water is admitted.

By far the greatest number of collapses are due to scale, salt, or greasy matter; but then, instead of the furnace crowns coming down, it is the sides which come in; for, although the deposit may be uniformly distributed, the heat of the fire is greatest on either side just over the burning fuel. Shortness of water and local deposits bring down the crowns, not the sides.

The collapse of furnaces may on occasion be due to very heavy firing combined with one or the other above-mentioned causes. Mr. Yarrow's experiments on tube plates (p. 121) show that, on account of the fire side of a plate being hotter than the water side, it curls towards the water. With the same fierce heat which he was using on flat plates, a furnace top of 20 ins. radius would tend to acquire a curvature of $20\frac{3}{4}$ ins. radius, and this would tend to increase the horizontal diameter by $1\frac{1}{2}$ ins. and reduce the vertical diameter by the same amount, making the furnace 3 ins. oval. The same influence is at work longitudinally. If free to move, the apex of the furnace crown, heated as above, would acquire a longitudinal curvature, of which the depression would be about $1\frac{1}{4}$ in. in a 6 ft. length. Of course, the end plates and the stiffness of the cool furnace bottom are opposed to these movements, but with the least sediment, grease or dense water, the metal is weakened and this support is much reduced and the furnace collapses. As these curvatures and depressions are independent of the thickness, it is well to make unstrengthened furnaces very thick if exposed to fierce heat, so as to give them extra strength.

In Mr. L. E. Fletcher's experiments on 1889 red-hot furnace crowns there is a case of a furnace hogging $\frac{3}{4}$ inch in a length of 20 ft. while raising steam, although the fires were very light.

Bulging of Flat Plates.—Another part of the boiler where scale deposits produce visible deformations is the flat plates of the combustion chambers. With narrow water spaces the presence of much steam assists in causing the plates to get hot and bulge, by keeping the scale partially dry. If this treatment did not tend to make the plates brittle, and cause them to crack between the stays, little harm would be done, because in their bulged shape plates are stronger than when flat. Possibly, too, the stretching of the plates enlarges the stay holes, and causes these to leak. When the bulging is serious the favourable conditions for further deformation are increased, because it is now more difficult than before to remove the scale.

In boilers with two or four furnaces, leading into one large combustion chamber, serious bulging sometimes takes place at the top of the saddle between the two central furnaces, but only if they are exposed to the heat of the fire, and if several angle irons have been fitted underneath, as is customary for strengthening these parts; the enclosed spaces form steam pockets, and the saddle plate above and between them gets overheated.

Tube-Plate Troubles in warship boilers once attracted very much attention, but it does not appear that any satisfactory explanations have been put forward. Doubtless here, too, it is a case of overheating caused by a high temperature in the combustion chamber, and an excess of steam bubbles near the plate on the water side. The phenomenon (see p. 30) that steel contracts permanently when heated and cooled again might account for the tubes shrinking and the tube plate drawing away from them, but it is difficult to suggest a remedy which has not been tried and failed.

Experiments on the subject have been made by A. F. Yarrow

('N. A.,' 1881, vol. xxxii. p. 106), who showed that tube plates tend to bulge towards the fire side, and thus to draw away from the tubes; but the strong curvatures which he was able to produce are not met with in practice. An idea of the temperature of a tube plate will be obtained by examining the following case. (Compare p. 122.)

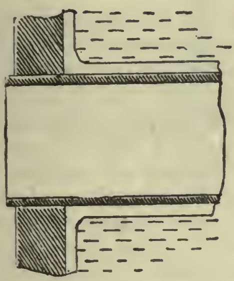

FIG. 49

Let the thickness of a tube plate (fig. 49) be 1 in., if perfectly clean its temperature would be 40° F. higher on the fire side than near the water, while transmitting sufficient heat to evaporate 20 lbs. of water per square foot per hour. The tube, being only $\frac{1}{10}$ in. thick, will be 4° F. hotter on the one side than on the other, and therefore its average will be only 18° F. less than that of the tube plate, which corresponds to a relative shrinkage of about $\frac{1}{3000}$ in. This can be neglected in comparison with the compression existing in the tube, due to the expanding. If covered with scale, as shown in fig. 49, or if in contact with a film of steam, so that the water can only reach the plate with difficulty, then the conditions are changed.

Suppose that the scale is $\frac{1}{10}$ in. thick, and that only half of the above-mentioned quantity of heat is being transmitted, then the metal of the tube and tube plate would be about 1,000° F. hotter than the boiler water. But it is difficult to believe that the heat transmitted through the tubes is equal to that passing through the plate, and certainly, within a very few inches from the end, the difference will exceed 50 %, so that it is not unreasonable to assume that the tubes are only 500° F. hotter than the water. With 3-in. diameters this corresponds to a relative difference of $\frac{1}{100}$ in. between the tube and hole. The heightened temperature will reduce the elastic limit to, say, 13 tons, so that the pressure, due to the expanding, cannot dilate them more than $\frac{3}{1000}$ in., thus leaving an opening of half the difference, viz. $\frac{7}{2000}$ in., or about $\frac{1}{300}$ in. all round the tube. This is sufficient to cause serious leakage. Salt from the boiler water is supposed to choke this opening as long as the heat is fierce, but dissolves out and renews the leakage when cold. (A. J. Durston, 'N. A.,' 1893, vol. xxxiv.)

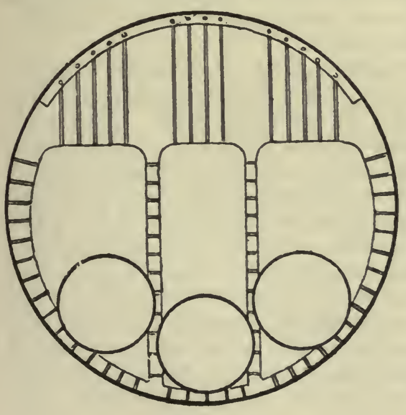

FIG. 50

In most boilers where this has occurred another force was at work intensifying the leakage. It is well known that in order to reduce the crushing stress on tube plates of double-ended boilers, not only the sides but also the tops and bottoms of the combustion chambers are stayed to the shell (fig. 50).

When pressure is raised in the boiler its diameter increases, and tension stresses are produced in the tube plates, which they, with their small effective sections (only about 25 % of the solid plate), are unable to resist, except by relieving the tubes of a little of the pressure which holds them in place. At the same time the tubes contract their diameters a little, due to the external steam pressure.

The chief difficulty in accepting these views is that the leakages are not diminished when the fires are drawn, and the phenomenon noticed by Wehrenfennig (see p. 20), that iron and steel contract permanently when heated, even if only up to boiling point, may help to explain the matter. The author's experiments on iron and steel bars which had been drawn out cold under a hammer, on cold rolled bars, on wire (drawn), and on stretched test pieces, show that they all contract permanently on being heated. The distortion which takes place during the annealing of flanged furnace front plates points to the same conclusion, viz. that leaky tubes are caused by a partial annealing of the expanded tube metal, due to excessive heat. The remedy which at once suggested itself is to anneal and re-expand the tube ends in place. According to this view, tubes which have leaked once (and have therefore been annealed) ought not to leak again after re-expanding, but although they do, the trouble is lessened. It is curious that tubes which have leaked on account of excessive scale grow tight when it has been removed, even if the ends are not re-expanded.

The only effective remedy for these troubles, if they were not caused by scale, appears to be the use of copper tube plates. Possibly a wider pitching of the tubes, or reducing their diameters at the back end, may do good. Or the tube ends might be subjected to a preliminary compression. This, however, would require an amount of care in boring the tube plate holes which is not often bestowed on them. Differences of $\frac{1}{16}$ in. in the diameters of various holes in one plate are not uncommon, and most of them are distinctly oval, sometimes as much as $\frac{1}{32}$ in. Taking more care to make the holes circular and of equal size may reduce the trouble.

In a recent experiment (discussion on A. J. Durston's paper, 'N.A.,' 1893, vol. xxxiv. p. 150) a compressed tube was compared with another which had been expanded in the usual way. Both were fixed in a tube plate which was placed over a smith's fire and heated to a temperature at which about 100 lbs. of water were evaporated per square foot per hour, but the plate was only occasionally covered with water, which then remained in a spheroidal condition. After a few hours' exposure the plate was allowed to grow dull red hot, when both tubes grew slack. Careful measurements showed that the expanded tube had contracted its diameter $\frac{1}{40}$ in., while the compressed one had not altered, but its hole in the tube plate had changed its taper from 1 : 12 to 1 : 10. This compressed tube could not be withdrawn out of the hole, for its projecting end ($\frac{1}{8}$ in.) had expanded during annealing. This goes far to prove that leaky tube ends are caused by raising their temperature to redness.

Attempts are being made to electrically weld tubes into the plates, but the risk of burning an internal part of a boiler which is nearly finished must deter all except the most venturesome from trying this plan.

As a makeshift ferrules are driven into the ends of boiler tubes (figs. 51 and 52), but they soon burn away. Better results are said to have been obtained by making the ferrules of cast instead of wrought iron, and leaving an air space between them and the tube (see fig. 53 and A. J. Durston's paper).

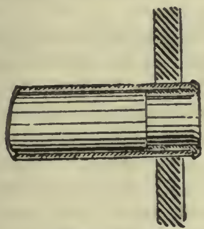

FIG. 51

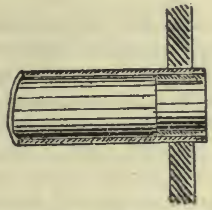

FIG. 52

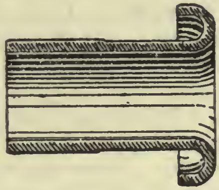

FIG. 53

Cleaning Boilers.—From the foregoing remarks it is clear that many troubles are caused by the accumulation of scale, grease, or even salt, and where this cannot be prevented, great care has to be taken that the boiler shall be cleaned out and scaled as often as possible. This work is never done well unless the various parts of the boiler are easily accessible, not only to boys, but to the engineer, who has to see that the work is properly done. None of the water spaces between the tubes should, therefore, be less than 10 ins., and if that amount of space cannot be provided over the wing surfaces, it ought to exist between the tubes and the shell, or a manhole should be fitted in the two wings of each boiler. They should not be made smaller than $10\frac{1}{2} \times 14$ ins., but 12×16 ins. is ample. A sufficiently large manhole should also be fitted between and under the furnaces, either at the front or back end of the boiler.

Removal of Scale.—There are several methods of removing scale. The general one is to chip it off. This work is very laborious on account of the hardness of the scale. Scale can be softened by adding a sufficient quantity of soda to the boiler a few days before reaching port. The soda converts the sulphate of lime into carbonate of lime and the change has a loosening effect on the scale. After a time, however, this scale would harden again and further additions of soda would have no effect. Mica powder is said to be a very effective scale loosener. When the boiler is empty, the scale has to be removed as rapidly as possible because air hardens it. Boilers should be cooled with all their water in them. The scale can then be removed with a hose pipe, brushes, and scrapers, and without chipping. The men should have oil-skins, and the water in the boilers should only be lowered as the cleaning progresses. Even old scale will yield to this treatment if repeated. Scale grows hard in about three hours after exposure to the atmosphere. Another method is to blow the boiler down while steam is up, care being taken that the brickwork of the firebridges is sufficiently cooled down, so as not to injure the furnace plates. The boiler is then allowed to cool, no doors being removed until the men are ready to enter the boiler. They will find the scale fairly soft and so well detached from the plates that it can be scraped off.

Removal of Grease.—The most effective method is to allow scale to accumulate and to remove it and the grease together. Naphtha and similar oils are often recommended with which the boiler surfaces have to be rubbed down. Of course naked lights may not be used, and as this plan is not very effective and said to be injurious to health, it should not be adopted.

Corrosion.—This subject is fully dealt with in a separate chapter, but here it will be necessary to make a few remarks on the external wasting away of plates. This never occurs except in the presence of moisture, and is found chiefly near manholes, near leaky joints, at the boiler bottoms and ends, where they come in contact with bilge water, and above all near the furnace fronts and combustion chamber bottoms. These are exposed to the very injurious action of moist ashes, containing large percentages of sulphuric and other acids, which are partly liberated by the ash-cock water. By keeping the boilers perfectly tight, and protecting the front plate from moist ashes, all these troubles can be avoided. Naturally also leaky tubes and seams cause external corrosion, for then a combination exists there of heat, moisture, salt, and noxious acids.

The leakage from valve and other flanges, the drainage from test cocks, and the moisture which collects in the wood of platforms adjoining the boiler backs, all tend to shorten the life of a boiler, and should be prevented as much as possible.

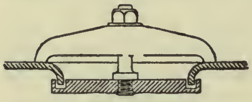

FIG. 54

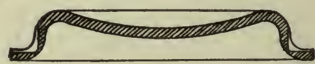

FIG. 54*a*

As regards manholes, an excellent practice is gaining ground of flanging them and facing the edges and grooving the door, as shown in fig. 54, which permits of their being faced up again when worn out.

Pressed manhole doors are now very largely used and generally with very satisfactory results, but it is desirable to make them of fairly thick plates, for when the fillets of the flange have been machined away, too little metal is sometimes left to resist the very severe thrust to which these doors are subjected. This weakness, in an exaggerated form, is indicated in fig. 54*a*.

Boiler Repairs.—Sooner or later even the best managed boilers will need repairs. If badly done they lead to further troubles, and may prematurely necessitate the renewal of the entire boiler. Care should therefore be taken that the workmanship is of the best, and that no structural defects are introduced, which would either prevent circulation or hinder the removal of scale.

Bulged Furnaces.—The most serious troubles with new boilers are the furnaces. A careless engineer, or other causes, which have been already mentioned, may suddenly bring them down. If the deformation is not great, i.e. only a few inches, and spread over a considerable area, the best remedy is to press the plates out again. Attempts have repeatedly been made to do this cold, but the deformations can never be perfectly removed in this way, and soon return. Besides, other parts of the furnaces are thereby strained, and it has happened that while pressing up a furnace crown, the welded seam at the bottom cracked and had to be patched. Formerly the necessary heating was done by coke fires and a blast, but it is more convenient to use a powerful Wells light, for with it there are no bricks to be removed. The hydraulic jack should be well bedded on timbers against the bottom of the furnace, and it should have a cast-iron head of the same radius as the furnace. The pressing out should be done slowly, and the plate should be reheated several times during these operations, and once afterwards for the purpose of annealing. If the operation is carried out too quickly, or in too few operations, the bulge is likely to reappear after a short time.

Collapsed Furnaces.—If the furnaces have collapsed thoroughly they must be renewed, and it depends very much upon the boiler design whether this can be done efficiently. If the furnace back end was flanged inside of the tube plate, as is now generally the case, there is hardly any other remedy than to cut the back end and to insert an unflanged furnace (figs. 55, 58). Sufficient width should be left for a double row of rivets, for such a joint seems to give better

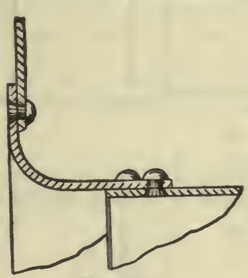

FIG. 55

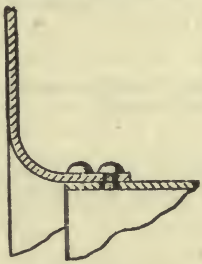

FIG. 56

results than a single-riveted one. This cutting out can be done very quickly and neatly with the help of the oxy-acetylene flame. A hole is drilled into the plate, its edge is raised to a white heat, and the acetylene is then shut off and only the oxygen blast maintained, which burns away a narrow slit of steel without much heating. Chiefly for cheapness' sake, but partly for facilitating the above-mentioned repairs, the tube plates, instead of the furnaces, are sometimes flanged (fig. 56). Should the furnace collapse, it is then only necessary to draw it and to fit a new one. Under any circumstances no expense or trouble should be spared on these seams to bring the two plates into metallic contact. It is well to wash the two surfaces in sal ammoniac, so as to remove the scale ; also, when all the holes have been drilled, which should be done in place, the furnace ought to be partly withdrawn, the burrs removed, as well as every trace of oil, because this is a very bad conductor of heat. The riveting should not be started from the bottom of the seam; otherwise, before the top is reached the plates at the crown will be far apart (see fig. 57). It is then practically impossible to make the seam permanently tight, and after being in use a short time it is sure to leak again. Sometimes it only opens on the fire side, and then, though the blade of a pocket knife can be inserted up to the other caulked edge, no leakage seems to take place. In order to be quite safe, it is best to fit and rivet the furnace crown into position before the furnace bottom is introduced.

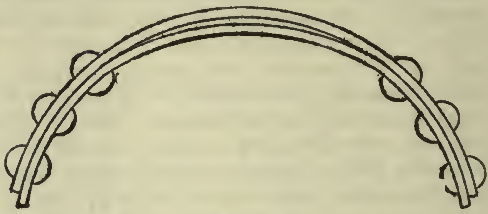

Fig. 57

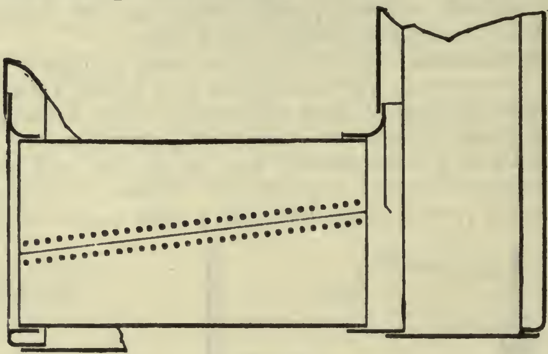

Fig. 58

The longitudinal seam should be slightly inclined, as shown in fig. 58, and then it is an easy matter to make it a very good fit. Another

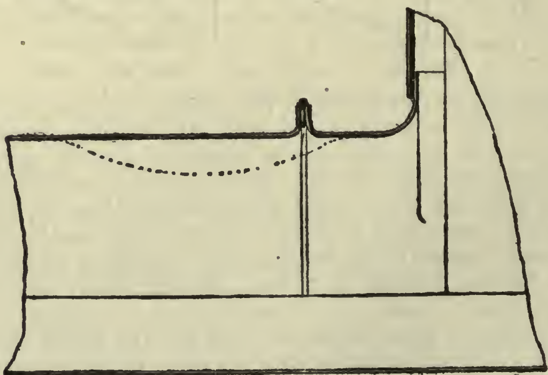

Fig. 59

way of repairing a collapsed furnace is shown in fig. 59. The furnace crown is cut in two, and the forward end is renewed, while the saddle is taken out, flanged, as shown, and refitted with an Adamson's ring.

With a furnace whose back end is not flanged, repairs may sometimes be effected by removing all the rivets and turning it round, so that the saddle seam is placed in the ashpit. The weld which was at the bottom of the furnace is now somewhere near the crown. Central

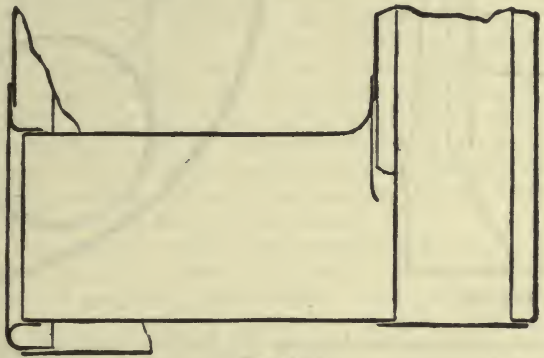

FIG. 60

furnaces can, under certain conditions of shape, be renewed by first removing the combustion chamber bottom plate.

If the furnace saddle is flanged over the tube plate, and if the furnace front plate is outside the front tube plate (see fig. 60), then both it and the furnace can be removed and the one renewed. But if the furnace front plate is placed inside the front tube plate, and has to be withdrawn, it is necessary to cut away its two corners (fig. 61), so as to be able to tilt it out of and into position. The amount of metal to be removed depends on the depth of the flange. This plan cannot be carried out on boilers with three furnaces, unless the horizontal seam intersects the central bundle of tubes.

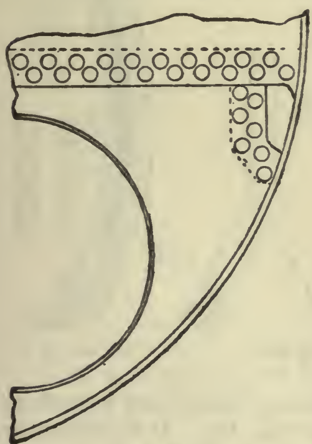

FIG. 61

The arrangement shown in fig. 60, which facilitates these repairs, has the further advantage that both the back and the front seams are very easily caulked from the outside. Its only drawback is that one loses about $1\frac{1}{2}$ in. of tube length.

Flanged furnaces can occasionally be removed and replaced intact by cutting away the lower part of their back ends (fig. 62). The wing combustion chambers would have to be specially constructed, so that no part of the furnace flange is wider than the front ends of the furnaces (fig. 63).

Before attempting such repairs, careful measurements should be taken, and, with the help of a board, representing the furnace front,

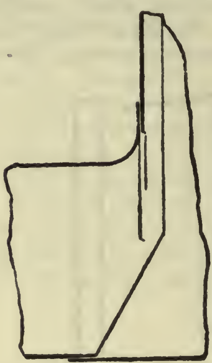

FIG. 62

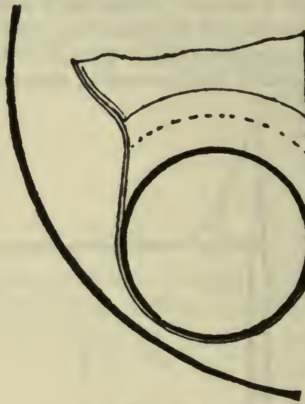

FIG. 63

and a wooden model of the furnace (fig 64), the action of passing the latter through the former should be performed. This will also enable one to determine how much of the back end has to be cut away.

Furnace Patches.—Furnaces sometimes suffer severely from pitting along the line of fire bars. If some of the pit holes should have pierced the plate, it is necessary to drill them out and fit screwed studs, or, better still, broad-headed rivets, which may be square, oval, or irregular in section, to suit the special case. Neither give trouble by leakage, but they do not add to the strength of the furnace, and, unless absolutely necessary, it is better not to perforate it at any point above the bars. If the pitting is uniformly distributed, as well as deep, there is danger of this part of the furnace giving way, but as yet no such case seems to have happened. To guard against this, or to prevent further pitting, some engineers bolt doubling plates, well bedded in red-lead cement, along the line of fire bars (fig. 65). It is difficult to imagine a more inappropriate remedy, for to place here, at the hottest part of the furnace, not only two thicknesses of plate, but actually to separate them by a highly non-conducting material, is little better than inviting disaster. If any action is necessary, the furnace crowns should be removed.

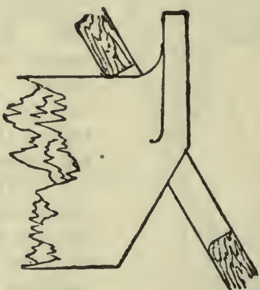

FIG. 64

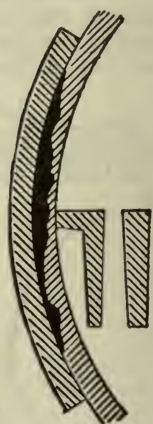

FIG. 65

Furnace bottoms are sometimes doubled in the same way; but this is not a permanent repair, and the plates should occasionally be removed for examination (see p. 38).

Blisters in the furnace crowns need only be pared away, unless they are so large and deep that there is fear of their weakening the furnace. In that case they have to be cut out and a patch fitted.

Cracked Furnaces.—Plain furnaces have hardly ever been known to crack except at the flanges or the seams, but every one of the patented forms has done so. If due to scale or grease, the cracks start on the fire side and sometimes run longitudinally; if due to unequal expansion or excessive strains, they sometimes show first on the water side, and generally in a circumferential direction. Similar cracks (see fig. 43, p. 21) are sometimes found in the sides of the furnace saddles, and can often be repaired by chain-pinning them. They make their appearance on the fire side, and sometimes, if not interfered with, do not penetrate through the plate.

When the cracks, which show themselves, are so serious that tapping pins into them would not prevent leakage, the effected parts have to be cut out and patches fitted. In fig. 66 the patch is fitted on the fire side, and can easily be renewed if the rivet holes should crack. In fig. 67 it is fitted on the water side. This is done with the object of keeping the seam turned away from the fire, but it is a most difficult piece of work, because the seams at A and B have to

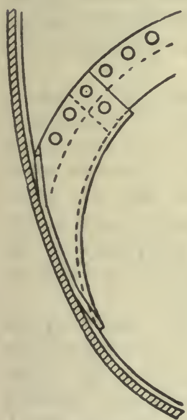

Fig. 66

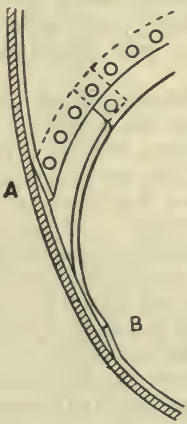

Fig. 67

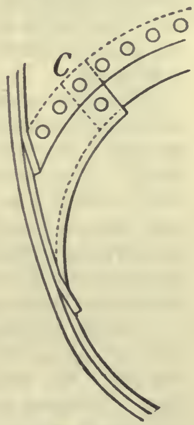

Fig. 67*a*

be sprung open. This patch is also not to be recommended, because if the rivet holes should crack, a larger corner would now have to be cut out and replaced (see fig. 47, p. 22). If the tube plate or furnace saddle is not too stiff, the repair shown in fig. 67*a* is a very convenient one.

Cracks like those shown in figs. 43-45 (p. 26) can be repaired by welding them with the oxy-acetylene flame, but such repairs will not be satisfactory unless the cracks are cut out or burned out, and the conditions explained on pp. 337 carried out; the flames should be applied to both sides of the plate while the new material is being gradually built up.

Combustion Chamber Bottoms.—These are often very seriously corroded on the fire side, particularly near leaky seams, and, as even the rivet heads retain their shape when wasted (see fig. 68), there is danger that such defects will be overlooked. The salt from leaky tubes when mixed with ashes causes corrosion at these points, and to prevent this, the bottoms are sometimes cemented. This is a very dangerous practice; not only does this prevent any leakage from being detected, but if there is one, or even if moisture finds its way between the plates and the cement, the furnace heat will convert it into steam and cause a small explosion, which may do damage to bridges and fire doors, &c. Fitting doubling plates to these parts may lead to the same result, but the consequences would be still more serious.

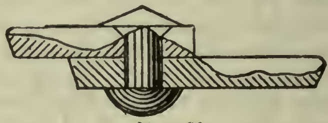

FIG. 68

The corrosion on the water side of the combustion chamber is often very evenly distributed, and if the bottoms are found to be thin, the backs are most likely worn away too, and before starting to cut out the one it is best to drill holes in the other for measuring the thickness.

The best means for detecting weak places in plates is to make a thorough examination when the plates are well scaled. The shadows produced by the lamp often indicate irregularities, particularly pit holes. The hammer will give no reliable indication of thickness when this exceeds $\frac{1}{8}$ in., and then it is of course possible to dent the plate and even to knock a hole into it. When general wasting and consequently structural weakness are suspected, drilling the plates is the only means of measuring their thickness, though magnetic and electric tests have been suggested and might be made available.

Before cutting out thin plates, it is well to drill test holes beyond the intended new seam, for if further thin places are revealed after the plate has been cut across, it will be necessary to repeat the operation. Patches on the lower parts of the combustion chamber usually include flanges, and care should be taken that these are properly fitted, for none of the seams or rivets can be caulked from the water side, as is the case with new boilers. Of course new stays will have to be fitted, and these should always be nutted, no matter whether this was originally the case or not.

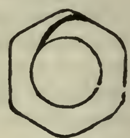

FIG. 69

Screw Stays very often lose their nuts by burning, or rather bad welds open under the influence of the heat (fig. 69). It is quite a common practice to cut off the greater part of the projecting stay and rivet it over. This should not be done, because it is due to the presence of the nuts that a reduction is allowed in the thickness of the plates. The better plan is to cut the thread deeper (fig. 70), for which special tools exist, and to fit new nuts.

This burning away of the stay nuts is generally due to their not being in metallic contact either with the stay or the plate; or, in other words, they are a loose fit on the thread, and are resting on a washer and two thicknesses of red-lead cement. When refitting the nuts, these mistakes should not be repeated, but where the stays are not normal to the inside plates, taper washers are a necessary evil. This arrangement should be objected to in new boilers.

With thin plates the stay ends often leak, particularly if their ends are riveted over. Instead of renewing them, some boiler-makers go to the trouble of making a small cap and bolting it over the head, as shown in fig. 71. Sometimes several stays in one combustion chamber are hidden away in this manner, and it is only a wonder that this practice has never led to a disaster. (See p. 222.)

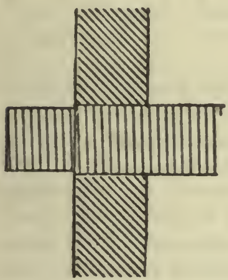

Fig. 70

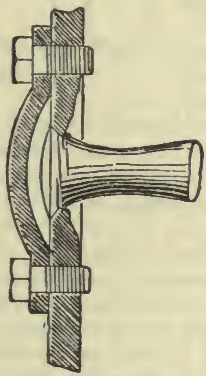

Fig. 71

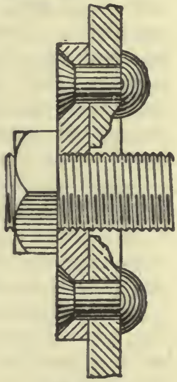

Fig. 72

The excuse made for fitting these patches is that the plates had become too thin for holding a new stay. If the thinning is only local, a very efficient repair is to replace the stay by one of a larger diameter. But in screwing up the nut its power will perhaps be so great that the thread in the plate gets stripped; to guard against this, a check nut should be fitted on the inside. It is bad for the transmission of heat, but is better than no support at all.

Often it will be necessary to remove the stay, and to fit a small patch and tap it, as shown in fig. 72. This is the best possible arrangement, particularly if the stay is not normal to the plate, because the patch could be made taper, bent or recessed so as to let the nut come into metallic contact with it.

Sometimes the plates remain uninjured while the screw stays waste away and will have to be renewed, generally of a larger diameter, the threads having been injured.

In order to effect these various repairs, and to stop external leakages, sufficient space should be left at the backs of boilers, so that a man could work there.

It will be noticed that those stays which are nearest the combustion chamber bottoms and sides give the most trouble. This is no doubt due to the very severe strains to which they are subjected through the unequal expansion of furnace and shell plate, and therefore they ought to be kept as far away from the flanges as the strength of the boiler back plate will allow (see p. 141).

Similar remarks apply to the top row of screw stays to boiler shells; if placed too near the combustion chamber top they also are subjected to very severe strains, sometimes leading to rupture.

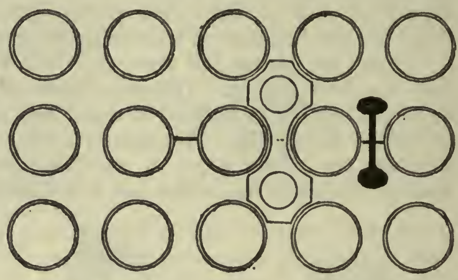

Fig. 73 *a* Fig. 73 *b*

Tube-Plate Troubles.—The last of the troubles in combustion chambers is the tube plate. An explanation as to the causes of leaky tubes has already been attempted, and some remedies suggested. The repeated re-expanding, together with the injury caused to the metal by getting hot, if covered with scale, ultimately causes the tube plate to crack. Fortunately, since steel and thicker tube plates have been introduced, such cracks are rarely met with. If caulking will not stop the leak, or if several spacings show cracks, it is necessary to bolt little spectacle patches (fig. 73*a*) over them, preferably on both sides. Keys, shaped like a dumb-bell (fig. 73*b*), are said to have given good results, but the fitting must be carefully done to make it a good job.

If the front tube plates of boilers are curved or bent back at the top, they are subjected to very serious tension stresses, which in one case led to a rupture, so that it was found necessary to fit vertical stays, as shown in fig. 74. It is, however, questionable whether the stays were of much value, as their lower extremities had only been bolted to the furnace tops, which, being thin, could not have offered a resistance if the large stays had been strained to their utmost.

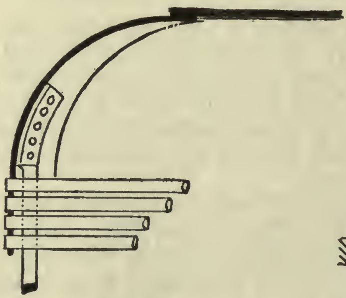

Fig. 74

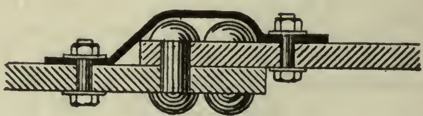

Fig. 75

Repairs to the Boiler Shells are sometimes necessary on account of the wasting away or cracking of circumferential seams. As such patches are not required for strength, but only for water-tightness, they need not be thick. At one time it was customary to make them of cast brass, a thick sheet of lead, which was hammered over the seam, being used as a pattern; but it was found that this gave rise to very serious local corrosion, caused, perhaps, by galvanic action. In their stead light iron cover plates, filled with cement, are bolted over these seams, as shown in fig. 75. They were very efficient in stopping leaks where caulking was unavailing. Of course if the shell plate is seriously weakened it is necessary either to fit an efficient doubling plate, or the defective part must be cut out, and a proper patch put in its place; but it is very seldom that this occurs, except perhaps with donkey boilers, whose bottoms cannot be properly protected from moisture.

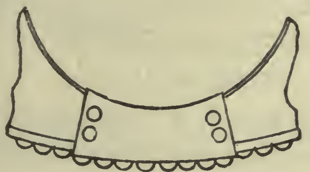

Fig. 76

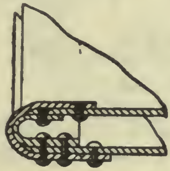

Fig. 77

It is different with the end plates of main boilers; they are exposed at one end to the hot and moist ashes, and at the other to leakages from flanges and manholes, &c. As these plates are originally very thick, and more than sufficiently supported, their wasting away is not so much a danger as a nuisance. Most of the patches to these parts are therefore simply doubling plates over the thinnest places, and it is only when no other remedy is thought effective that the bad parts are cut out.

The following are a few sketches of repairs which may occasionally be necessary. Figs. 76, 77 show a case in which the circumferential seams at the front ends of the shell and the furnace have wasted away and have been repaired by a covering patch. As it forms a ridge on the inside of the furnace, against which the rake will always be knocking, it is better to carry out this repair as shown in figs. 78, 79. Part of the furnace bottom is cut away, and the patch fitted in its place. Of course a strap will have to be placed on the water side, as shown in dotted lines (fig. 79).

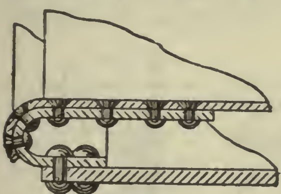

Fig. 78

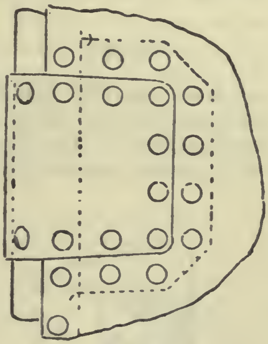

Fig. 79

If only the front plate is wasted away, and if this is only between the two furnaces, an internal patch can be fitted, as shown in figs. 80, 81. Generally a manhole or a few large stays will be found here, and as it is impossible to get a sufficiently large patch into the boiler to cover them, it will be necessary to fit it from the outside.

When the lower front seams both of the furnaces and the shell are in a bad condition, it may be necessary partly to renew the front plate, and to make its flanges sufficiently deep to take in another row of rivets. The joints should be placed where there is sufficient room for working at them. Lap joints, as shown in fig. 83, are very difficult to

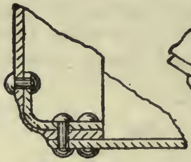

Fig. 80

Fig. 81

make, for if looked at from the back (fig. 82), it will be noticed that the patch has to be fitted under the remaining plate, which cannot be done efficiently.

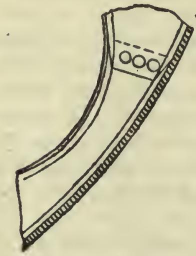

Fig. 82

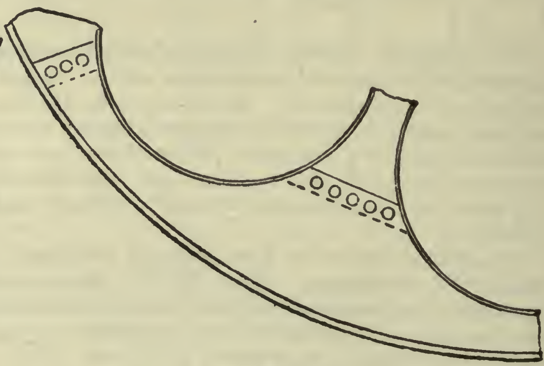

Fig. 83

A simpler plan is to butt the plates at these points, and to fit flanged butt straps internally (fig. 84), or, where the space is too narrow, a solid block (fig. 85). In either case this part has to be carefully fitted after all the other seams have been riveted up. Of course with the solid

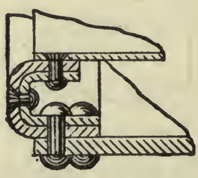

Fig. 84

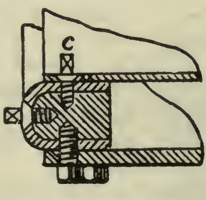

Fig. 85

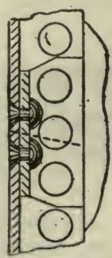

Fig. 86

block it will be necessary to use screwed studs and bolts, except perhaps for those points marked *c*, for which through rivets may be used, but only if the holes come opposite each other. The flanges should not be cut away parallel to the axis of the boiler and furnace, but should slant downwards (see dotted line, fig. 86); this will enable the plate edges to be driven tight together.

Before deciding which plan to adopt, a sketch should be made to see whether it is possible to insert the various rivets. Much space is gained if the rivet holes are countersunk inside so as to do away with the heads, and outside, so as to be able to use short rivets (see fig. 87).

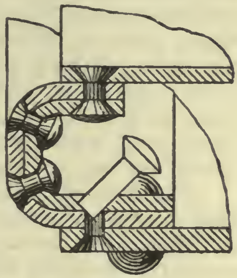

FIG. 87

BOILER FITTINGS.

Water Gauges. — Of the various boiler mountings, one of the most important is the gauge glass, and it ought always to be in good working order, and capable of being tested at any time. The material and construction should be of the very best. Asbestos packed cocks are a great advantage, but even with them, if the gun metal is bad, leakages cannot be prevented. Iron and steel plates are attacked by acids, whereas copper alloys suffer severely when in contact with caustic soda; care should therefore be taken never to make the boiler water too alkaline.

In some cases the gauges are fitted direct to the boiler plating, but this is not a good plan; not only does the panting of the plates lead to breakages of the glass tubes, but the scum cannot be kept out of the glass, and the oscillations due to the rolling of the vessel and to irregularities of ebullition make it difficult to obtain good readings.

The usual plan is to connect the gauge glass to the top and bottom of the boiler by means of long pipes, the one reaching well up into the steam space, and the lower one into the water, where little circulation is expected to be found. These pipes can be fitted either externally or internally. The latter arrangement is not always possible, on account of the smoke box, and some engineers object to it *in toto*, on the principle that no copper pipes should be fitted inside a boiler. From the following table it will also be seen that, if not

Hot Water in Boiler. Cold Water (70° F.) in 10-foot Gauge Pipe			Hot Salt Water in Boiler. Distilled Hot Water in 10-foot Gauge Pipe		
Boiler Pressure	Temperature in Boiler	Water Level in Boiler *above* Gauge Reading } H, fig. 31	Saltness of Boiler Water		Water Level in Boiler *below* Gauge Reading
Lbs.	° F.	Inches	Oz.	$\frac{1}{32}$	Inches
60	307	11	10	1·7	4½
100	338	13½	20	3·3	9
150	366	16	30	5·0	14
200	387	18	40	6·7	19

often blown through, gauges with long connections may indicate a different water level from that actually existing in the boiler. When fitted externally these pipes have the effect of cooling the water contained in them, and as its density is thereby very much increased, the indicated water level is lower than that in the boiler. Matters are still further complicated by the accumulation of distilled water in the upper parts of the connecting pipe, which, if there is much salt in the boiler, has the effect of making the water level appear higher than it really is. The possible differences caused by a connecting pipe 10 ft. long are shown in the above tables, which contain the corrections which should be applied to the water-gauge readings under the various conditions.

The first of these tables has been calculated on the assumption that the rate of expansion of water is independent of the pressure; but even if not strictly true, both tables show that a serious difference may exist between the gauge-glass indication and the water level, unless the cold or distilled water in the connecting pipe has first been blown out. Few gauges are so fitted that this can be readily done (see fig. 31), for if the water is low, the opening of the cock D, while B is closed, will not necessarily clear the pipe C D of its water, or at any rate not as quickly as if the cock A had been closed. This being inconvenient, some gauge stands are fitted with a large cock at F.

No double bends, capable of collecting water, should be allowed to exist in the gauge steam pipe, as they would cause the glass to show a wrong level.

Gauge-Glass Protectors.—On account of occasional personal injuries, when gauge glasses of high pressure boilers broke, they are now very generally protected by glass shields of which there are several patterns. These protectors should be made of stout and tough glass, for their splinters can do far more harm than the much smaller ones of the glass tubes. One of the safest protectors consisted of a semicircular sheet of stout glass in which wire netting was embedded, but this netting obstructed or rather confused the view of the water level. Where flat plate-glass protectors are used, care should be taken that there are no conspicuous horizontal objects opposite to them, whose reflection might be mistaken for water-level readings.

Test Cocks are intended to indicate the boiler water level. They are often fitted to the gauge-glass stand, but their proper position is on the boiler plating. One should be at or near the ordinary water level, a second a few inches higher, and a third on a level with the highest heating surface; for, should the water have been lost out of the gauge glass, it is very important to know whether a mishap can still be averted by putting on all available feed, or whether it is necessary to draw the fires. It is not always well to check the fires, for the immediate effect of stopping ebullition is to lower the water level. Sudden variations of water level are sometimes caused by the drain pipe from the superheater having been fitted below the water level of the boiler and not being closed on occasions when this was necessary; and more recently similar effects have been produced by negligence in not closing the feed-pipe suctions from the boiler bottoms, which are only intended to be open while in use, when steam is being raised.

In marine boilers shortness of water is not necessarily as dangerous as in land boilers, because the furnace crowns are many feet below the ordinary water level, and the upper heating surfaces are not exposed to a very high temperature; besides, in rough weather they are being constantly moistened by water splashing on them. But the case is different in calm weather, if the steamer has a strong list; for then, even if there is a reasonable height of water in the glass, one of the wing combustion chamber tops may be laid dry for a long period and get overheated. Furnace crown plates would grow red-hot in about five minutes after being laid bare, while combustion chamber top plates would require twice or four times as much time. Obviously this danger is intensified if the gauge glass is fitted to the boiler side, and even the trim of a vessel may expose parts of the heating surface to this influence if the boiler is a very long one. In such cases it is advisable to fit fluid levels in the engine room, which will indicate how much water ought to be in the glass. They are very simple instruments, and can easily be made with the help of a few glass tubes (see figs. 32, 33).

Fusible Plugs.—To guard against the troubles arising from shortness of water, some engineers fit fusible plugs to the combustion chamber tops, and in some countries this practice is compulsory; but there is this serious objection, that when a plug is once fused the boiler cannot be used again for some hours. On a ship with only one boiler such a mishap might lead to her total loss, and almost under any circumstances which are likely to occur at sea, it would be better to have the use of boilers which can be worked even if only at a very low pressure, than not to be able to use them at all till a new plug has been inserted. See p. 19.)

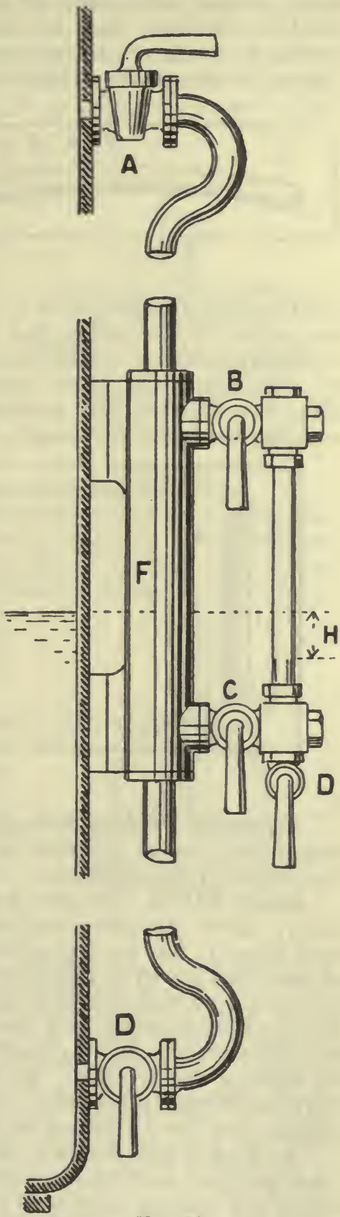

FIG. 31

Low Water Alarms.—The majority of land boilers are fitted with safety valves, which blow off when the minimum permissible water level is reached; they are inapplicable to marine boilers, and if modified to suit the altered circumstances would, though to a less degree, be open to the objections urged against fusible plugs. Water alarms, especially such as are arranged to ring an electric bell, situated if necessary in the engineer's berth, may in some steamers be found useful. These fittings seem to be reliable, but it would be well if they could be fitted with a cock or valve so as to produce artificially the conditions of low water in the boiler, and thus ascertain whether they are in working condition. In this form they would be a convenient means of summoning the chief engineer in case of emergency. Water alarms in which a float actuates a whistle work well, but would not combine the above advantages.

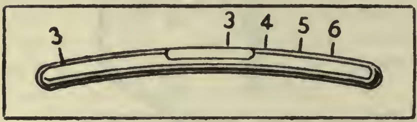

Fig. 32

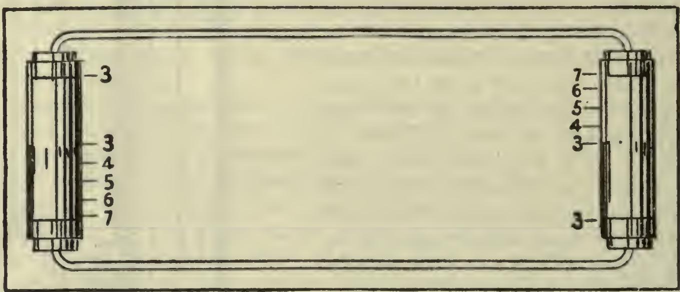

Fig. 33

Automatic Feeding Arrangements are as yet only being fitted to water-tube boilers, and are hardly likely to come into use in the boilers under consideration.

Safety Valves.—Another very important boiler mounting is the safety valve. Its action and design are so simple that little need be said about it, but being practically always out of use, there is a great danger that it will be found out of working order when it should act, and to ensure that this shall not happen, all the working parts should be lined with brass, and should be loose fits, and their working condition should often be tested by lifting the valves from their seats or turning them round. As it has sometimes happened that the valve seats have lifted with the valves, thus preventing any escape of steam, they should always be securely bolted or pinned down.

As long as low pressures were customary, weighted safety valves were common and worked satisfactorily; but now, since high pressures are almost universal, dead weights have been replaced by springs, for otherwise the working pressure would have to be reduced about 30 % in rough weather, on account of weights being thrown up under the combined action of the steam and the pitching of the vessel. (Wilson, 'Marine E.,' 1892.)

Formerly spring loaded valves were very inefficient, allowing the steam pressure to accumulate considerably, unless the springs were made exceptionally long. This difficulty has been more than overcome by utilising the reaction of the escaping steam, and, instead of allowing the pressure to rise after steam has commenced to blow off, there are many valves which will not close until the pressure has been seriously reduced.

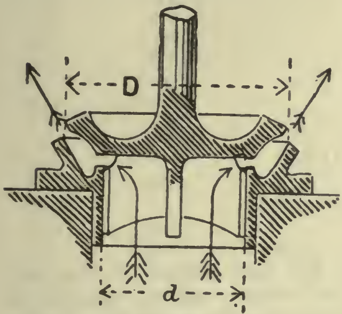

FIG. 34

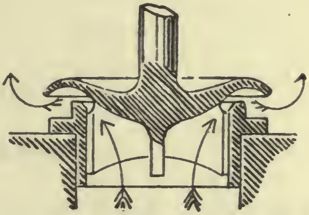

FIG. 35

The principles on which such safety valves can be constructed are shown in figs. 34, 35, 36. In the first of these the valve diameter is d, but when the steam is escaping it acts on the diameter D, and keeps the valve well open until the pressure has dropped considerably.

In fig. 35 the escaping steam reacts on the surrounding lip, while in fig. 36 it reacts, as shown, by striking against the valve top.

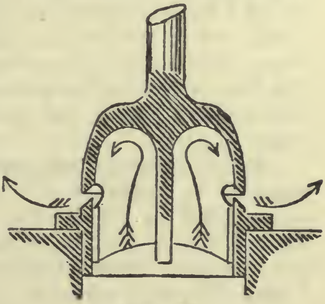

FIG. 36

The object of these various designs is that the valve should open fully when it does so at all, that it should not be closed till the pressure has dropped a pound or two below the working pressure, and that this should be effected with as short a spring as possible.

Obviously all these points are affected by the angle of the seat and the curvature and size of the lip; but, besides that, the diameter and length of the waste steam pipe are influential factors, and it often happens that safety valves which worked satisfactorily when tested at the works, will not do so when fitted on board. The chattering noise which they make is evidently due to the length of the pipe as well as to the weight of the valve and the elasticity of the spring. Every time that the valve comes down and stops the flow of steam, a partial vacuum is formed by the uprushing column, and the valve is lifted up again. The possible remedies are, to have an opening at the lower end of the waste pipe, through which air will flow whenever a vacuum is formed, or to fit frictional appliances in the lower part of the pipe, such as a series of diaphragms, wire brushes, or a box of pebbles.

CHAPTER II

STEAM AND WATER

This chapter, which did not appear in the first edition, is intended to give such information about steam and water, boiling, circulating, and superheating, as may be useful to boiler-makers and users. Some of these items may seem to be of no great practical importance, but when it is remembered that difficulties have been caused to contractors by such a simple question as whether the weight of water in a torpedo-boat boiler should be taken cold or when heated to, say, 360° F., i.e. whether a cubic foot of water was to count as weighing 62·4 lbs. or only 55 lbs., it is evident that this information at least should not be omitted from a work on boilers. Then, also, these and other matters affect the measurements of evaporation. It has therefore been attempted to give the latest information on the properties of steam and water, and most of these are contained in the table on pages 50 to 53.

Steam—Pressure, Temperature and Volume.—These subjects have engaged the attention of the following observers, whose results are summarised in 'Wied. Ann.,' 1907, vol. xxii. p. 625, and some are reproduced in the following tables :—Magnus, Regnault, Wüllner and Grotrian, Batelli, Cailletet and Colardeau, Marvin, Ramsay and Young, Wiebe, Knoblauch, Linde and Klebe. The results of Regnault and of Ramsay and Young are contained in cols. 9 and 11. As will be seen, the two are not in absolute agreement, but very nearly so. Cailletet and Colardeau's experiments were made by comparing steam pressures with compressed hydrogen pressure gauges which had first been tested against the mercury column on the Eiffel Tower. Although nearly ten years have elapsed since these experiments were first published, the only details are contained in a small diagram in which the height of the Eiffel Tower is represented by about three inches. In that paper the experimenters also state that their observations agree with the formula given by Clausius ('Wied. Ann.,' 1881, vol. xiv. p. 698). Their information was thought too vague to be embodied in the table, but the following are the values supplied to me by the authors.

Table of Steam Pressures

Temperatures	Steam Pressures		Temperatures	Steam Pressures	
	Clausius	Cailletet and Colardeau		Clausius	Cailletet and Colardeau
° F.	Lbs.	Lbs.	° F.	Lbs.	Lbs.
32	0·089	...	428	336·6	...
68	0·337	...	437	...	368·2
104	1·07	...	464	489·0	...
176	2·88	...	482	...	576·3
212	14·70	14·70	500	687·5	...
248	28·82	...	527	...	873·0
257	...	32·35	536	944·0	...
284	52·47	. .	572	1267·0	1266·0
302	...	69·10	608	1675·0	...
320	90·00	...	617	..	1787·0
347	...	129·20	630	1972·0	...
356	145·80	...	662	...	2462·0
392	225·70	224·9	689	...	2946·0

Steam Density.—W. Thompson ('Trans. R. Soc., Edinb.,' 1849, vol. xvi. p. 575) showed that the second thermo-dynamic law could be used to calculate the change of density which takes place during a change from the fluid to the gaseous condition provided the pressure temperature curve and the latent heat be known.

$$Vg - Vf = \frac{dT}{dP} \cdot \frac{J.\lambda}{T}$$

Here Vg and Vf are the volumes in c.c. of one gram of steam gas, and of fluid water at the absolute temperature T. J. is Joule's mechanical equivalent of heat, and equals 427·98 kilogram metres; it corresponds to 778·14 foot pounds per British Termal Unit (B.T.U.). See E. H. Griffiths ('Trans.,' 1893, vol. 184*a*, p. 496). λ is the latent heat of evaporation at and from the temperature T; $\frac{dT}{dP}$ is the increase of temperature with pressure.

Columns 14 and 15 contain the values calculated by Rankine with the help of this formula, he using Regnault's formula for steam pressure, which are not absolutely the same as those taken from his curve, col. 9. The values of λ are those given in col. 12 (p. 51). Direct experiments on density of steam have been made by Fairbairn and Tate ('Trans.,' 1860, vol. l. p. 192), and more recently by Ramsay and Young ('Trans.,' 1891, vol. clxxxiii). The capacities of Fairbairn and Tate's experimental glass globes were about 75 cubic inches, which is satisfactory, but the calibration of their glass stem, each division in each of the three globes being exactly 252·4 grains, seems to indicate that they took some details for granted; and, further, all air does not appear to have been extracted from the water; at any rate in six out of fourteen experiments the differences of level of the two mercury columns are such that, unless there was air in the glass globe, the experimental results are impossible, while in two other experiments there is similar indication that there must have been air in the

Table of Properties of Steam and Water

Steam Temperatures		Heats of Water (Regnault)			Volume of Water		Compressibility of Water per Million Atmospheres
		Specific		Total above 32° (3)			
° C.	° F.	At Temperature (1)	Mean (2)		Under one Atmosphere (4)	Nearly Boiling (5)	(6)
0	32	1·0000	1·0000	0·0	1·000,127	1·000,178	51·77
4	39·2	1·0002	1·0001	7·2007	1·000,000	1·000,051	50·33
10	50	1·0005	1·0002	18·004	1·000,260	1·000,313	48·54
20	68	1·0012	1·0006	36·022	1·001,751	1·001,796	46·09
30	86	1·0020	1·0009	54·049	1·004,314	1·004,356	44·14
40	104	1·0030	1·0013	72·094	1·007,73	1·007,77	42·65
50	122	1·0042	1·0018	90·162	1·012,01	1·012,05	41·68
60	140	1·0056	1·0023	108·25	1·016,97	1·017,00	41·15
70	158	1·0072	1·0029	126·37	1·022,60	1·022,63	41·3
80	176	1·0098	1·0037	144·53	1·028,90	1·028,92	42·2
90	194	1·0109	1·0044	162·72	1·035,74	1·035,75	43·9
100	212	1·0130	1·0052	180·94	1·043,15	1·043,15	46·3
110	230	1·0153	1·0061	199·17	...	...	49·5
120	248	1·0177	1·0069	217·49	...	1·059,30	53·4
130	266	1·0204	1·0079	235·85	...	...	58·2
140	284	1·0232	1·0089	254·25	...	1·078,86	63·0
150	302	1·0262	1·0100	272·70	...	...	69·8
160	320	1·0294	1·0111	291·20	...	1·100,91	75·9
170	338	1·0328	1·0123	309·76	...	...	84·6
180	356	1·0364	1·0136	328·40	...	1·126,43	93·2
190	374	1·0401	1·0149	347·10	...	1·140,8	102·4
200	392	...	...	...	...	1·156,8	112·4
210	410	...	...	...	...	1·173,8	123·3
220	428	...	...	...	...	1·192,2	134·9
230	446	...	...	...	...	1·214,3	147·0
240	464	...	...	...	...	1·235,8	160·1
250	482	...	...	...	...	1·255,8	173·9
260	500	...	...	...	...	1·284,5	188·5
270	518	...	...	...	...	...	...

Names of experimenters or calculators :—Reg. = Regnault ; R. & Y. = Ramsay and Young ; Griff. = Griffiths ; Rankine ; F. & T. = Fairbairn and Tate.

Column 2 is mean of column 1 ; column 3 is found from column 2 ; column 5 is found from column 4 with help of column 6. Values from 248° to 320° are Hirn's experiments ; the values for higher temperatures are by Ramsay and Young ; column 6 is from Landholt and Bornstein up to 140° ; Ramsay and Young values from 374° ; the intermediate values are interpolated.

Table of Properties of Steam and Water (cont.)

Steam Temperatures		Absolute Temperatures		Absolute Steam Pressures in Pounds per Square Inch		Total Heat in Steam above 32° F.	Latent Heat of Evaporation at Temperature	
° C.	° F.	° F. (7)	° C. (8)	Reg. (9)	R. & Y. (10)	Reg. B.T.U. (11)	Reg. B.T.U. (12)	Griff. B.T.U. (13)
0	32	491	273	·088	...	1091·7	1091·7	1074·1
4	39·2	498·2	277	·117	...	1093·9	1086·7	1071·7
10	50	509	283	·177	...	1097·2	1079·2	1063·3
20	68	527	293	·258	...	1102·7	1066·7	1052·5
30	86	545	303	·609	...	1108·2	1054·2	1041·6
40	104	563	313	1·06	...	1113·7	1041·5	1030·8
50	122	581	323	1·78	...	1119·2	1029·0	1020·0
60	140	599	333	2·88	...	1124·6	1016·1	1009·2
70	158	617	343	4·51	...	1130·1	1003·7	993·4
80	176	635	353	6·86	...	1135·6	991·1	987·6
90	194	653	363	10·14	...	1141·1	978·4	976·7
100	212	671	373	14·70	14·70	1146·6	965·7	965·9
110	230	689	383	20·78	20·76	1152·1	953·9	955·1
120	248	707	393	28·83	28·70	1157·6	940·1	944·3
130	266	725	403	39·25	39·03	1163·1	927·3	933·5
140	284	743	413	52·53	52·07	1168·6	914·4	927·6
150	302	761	423	69·23	69·00	1174·1	901·4	911·8
160	320	779	433	89·90	89·90	1179·5	888·3	901·0
170	338	797	443	115·2	114·7	1185·0	875·2	890·2
180	356	815	453	145·8	144·8	1190·5	862·1	879·4
190	374	833	463	182·5	181·7	1196·0	848·9	868·6
200	392	851	473	226·0	224·8	...	...	...
210	410	869	483	277·2	275·3	...	...	...
220	428	887	493	336·3	335·7	...	...	...
230	446	905	503	404·6	404·8	...	...	...
240	464	923	513	...	483·5	...	...	...
250	482	941	523	...	574·7	...	...	...
260	500	959	533	...	678·0	...	...	...
270	518	977	543	...	794·3	...	...	...

Columns 7 and 8. The absolute zero has been assumed 491° F. and 273 below freezing point. Column 11 gives Regnault's experiments; column 12 is found by subtracting column 3 from 11; column 13 is calculated by Griffiths from various data.

Table of Properties of Steam and Water (cont.)

Steam Temperatures		Rankine		Density of Steam compared with Hydrogen = 1								
				F. & T. (16)		Ramsay and Young (17)						
				Amount of Superheat ° F.								
° C.	° F.	Specific Volume of Steam (14)	Density of Steam (15)	0° Sat.	4°	0° Sat.	9°	27	36	54	63	81
0	32	211,131	8·74	...	...	...	...	...	...	...	...	...
4	39·2	...	...	...	...	...	...	...	...	...	...	...
10	50	108,731	8·83	...	...	...	...	...	...	...	...	...
20	68	58,868	8·89	...	...	...	...	9·67	9·53	9·36	...	...
30	86	33,385	8·94	...	...	...	...	9·61	9·48	9·33	...	...
40	104	19,723	8·98	...	...	...	...	9·55	9·44	9·30	...	...
50	122	12,079	9·04	...	...	...	...	9·51	9·40	9·27	9·24	...
60	140	7,672	9·06	...	...	...	9·83	9·47	9·37	9·25	9·22	...
70	158	5,027	9·10	...	...	...	9·73	9·44	9·35	9·23	9·20	...
80	176	3,387	9·13	...	...	...	9·65	9·40	9·32	9·22	9·19	...
90	194	2,340	9·18	...	...	9·76	9·60	9·38	9·30	9·20	9·17	...
100	212	1,658	9·20	...	...	9·69	9·56	9·36	9·29	9·19	9·17	...
110	230	1,192	9·29	9·93	9·84	9·64	9·52	9·34	9·28	9·19	9·16	...
120	248	878·5	9·32	9·80	9·63	9·59	9·49	9·33	9·27	9·19	9·16	9·12
130	266	657·2	9·39	9·67	9·42	9·58	9·48	9·33	9·27	9·19	9·16	9·12
140	284	500·1	9·44	9·54	9·21	9·57	9·47	9·33	9·28	9·20	9·17	9·13
150	302	385·8	9·52	...	...	9·58	9·48	9·34	9·29	9·21	9·18	9·13
160	320	301·8	9·58	...	...	9·60	9·50	9·35	9·31	9·23	9·20	9·15
170	338	239·2	9·65	...	...	9·64	9·53	9·38	9·34	9·26	9·23	9·18
180	356	191·7	9·73	...	...	9·70	9·59	9·43	9·38	9·31	9·28	9·22
190	374	155·3	9·81	...	...	9·77	9·65	9·49	9·44	9·36	9·33	9·28
200	392	127·3	9·88	...	...	9·87	9·75	9·57	9·52	9·43	9·40	9·35
210	410	105·2	9·96	...	...	10·01	9·86	9·68	9·63	9·53	9·50	9·44
220	428	87·7	10·05	...	...	10·18	10·01	9·82	9·75	9·66	9·63	9·57
230	446	73·8	10·13	...	...	10·38	10·19	9·98	9·92	9·82	9·79	...
240	464	...	...	...	...	10·64	10·41	10·18	10·12	10·02	9·98	...
250	482	...	...	...	...	10·97	10·70	10·43	10·35	10·26	...	...
260	500	...	...	...	...	...	...	...	...	...	...	...
270	518	...	...	...	...	...	...	...	...	...	...	...

Columns 14 and 15, see p. 49. Columns 16 give Fairbairn and Tate's experiments recalculated; columns 17 give Ramsay and Young's experiments calculated by author.

TABLE

The following table contains two sets of experiments on the specific heat of superheated steam. Those made by the National Physical Laboratory are limited to one pressure of 4·3 atmospheres, or 63·2 pounds per square inch, and to a range of 90° F. The experiments by Knoblauch and Jakob range from two to eight continental atmospheres (14·3 pounds per square inch), and from 300° to 673° F., but do not show as marked changes as might be expected from the first-named experiments.

Specific Heats of Steam

National Physical Laboratory				Knoblauch & Jakob			
Pressure Atm.	Temperature mean ° C.	Temperature mean ° F.	Specific Heat	Pressure Atm. Kg/cm.	Temperature mean ° C.	Temperature mean ° F.	Specific Heat
...	...	...	...	2·01	149·1	300·4	0·478
...	...	...	...	2·02	188·2	370·8	0·470
...	...	...	...	2·00	240·4	464·7	0·474
...	...	..	...	2·00	245·9	474·6	0·474
...	...	...	...	2·02	296·6	565·7	0·476
...	...	...	...	2·01	345·9	654·6	0·486
...	...	..	...	2·02	349·3	660·7	0·494
4·3	165	329	0·497	...	...	...	...
...	...	...	...	4·00	170·6	339·1	0·502
4·3	175	347	0·472	...	...	...	...
4·3	185	365	0·455	...	...	...	...
4·3	195	383	0·442	...	...	...	...
4·3	205	401	0·432	...	...	...	...
...	...	...	...	3·99	210·5	410·9	0·490
4·3	215	419	0·424	...	...		...
...	...	...	...	4·01	260·8	501·4	0·487
...	...	...	...	3·98	304·7	580·5	0·485
...	...	...	...	3·98	306·1	583·0	0·491
...	...	...	...	4·01	306·5	583·7	0·491
...	...	...	...	4·00	350·3	662·5	0·500
...	...	...	...	6·00	180·2	356·4	0·531
...	...	...	...	6·00	217·5	423·5	0·502
...	...	...	...	6·01	259·8	499·6	0·492
...	...	...	...	5·98	306·4	583·5	0·490
...	...	...	...	6·01	349·6	661·3	0·502
...	...	...	...	8·00	188·1	370·6	0·557
...	...	...	...	8·00	240·0	464·0	0·500
...	...	...	...	8·0C	295·0	563·0	0·490
...	...	...	...	8·00	295·2	563·4	0·488
...	...	...	...	8·00	295·7	564·3	0·498
...	...	...	...	7·99	344·9	652·8	0·501
...	...	...	...	7·98	349·8	661·6	0·504
...	...	...	...	8 00	356·0	672·8	0·509

surrounding boiler. In their summary Fairbairn and Tate also depart from the principles laid down by them, and instead of assuming the saturation point to have been reached when the first change of level was observed, they waited until a considerable difference of pressure showed itself. The fluctuations recorded by them of the levels of the two mercury columns are also not satisfactory; but the general principle on which the experiments were carried out is a good one, and, in spite of the above objections, sufficient reliance was placed on their results to cause these to be recalculated. The low-pressure tests were not made on the same principle and have been rejected; of the high-pressure ones only Nos. 1, 4, 7, 8, 9, 13 gave marked indications as to when the point of saturation had been reached, and only these have been used in col. 16. The values against 266° F. are the mean results.

Ramsay and Young's experiments extend over a much wider range of temperature, but the volume of steam experimented on was only a little larger than one cubic centimeter, or say one-thousandth of the volume employed by Fairbairn and Tate. It would, therefore, not be wise to rely on them absolutely, even although they may help to explain the great difference to be found between the weight of steam as measured on the indicator cards and the weight of feed water. These experiments are, however, of great comparative value, in so far as that they show a gradual decrease of density as the super-heat is increased. Their experiments extend from 284° F. to 518° F., and also from 59° F. to 122° F., but the lower temperature experiments are few, and not quite in accord with each other, and certainly are in direct conflict with Rankine's deductions. They were made in another apparatus. It is probable, therefore, that the increase of density with decreasing low temperature shown in the table may not be correct. Their experiments made at 392° F. and 464° F. are quite out of harmony with the rest, and have not been taken into account in the compilation of the table.

The densities in cols 16 and 17 are expressed in terms of hydrogen gas = 1 at 0° C. = 32° F. = 491° F. absolute, and 760 mm. pressure = 14·7 lbs. per square inch. The weight of hydrogen is 0·089551 gram per liter.

The specific volume is the volume in c.c. occupied by 1 gram steam and is easily found from the table.

According to the chemical analysis one would expect the density of steam to be 8·98. This low value is probably only reached when the steam is thoroughly superheated and under a low pressure. At any rate in Ramsay and Young's experiments the value was still above 9·0 when the pressure was 52 lbs., and the temperature 518° F., this being a superheat of 234° F.

Total Heat of Steam.—Regnault's experiments on the total heat of steam, from 145° F. to 381° F., adapt themselves very closely to the formula

$$H = 1101{\cdot}5 + 0{\cdot}305\,T$$

where T is the temperature in degrees Fah.

This formula has been used in the compilation of column 11.

The Latent Heat of evaporation at any temperature is found by subtracting the total heat in the water from the total heat in the steam. Regnault carried out experiments on the specific heat of water. Col. 1, the total heat in water, is the sum of these values reckoned from 32° F. (see col. 3). These values have to be subtracted from col. 11, the difference being the latent heat of steam. In recent years the specific heat of water has been much enquired into, but Regnault's values may be accepted as correct. Recently Griffiths has recalculated these values (see col. 13). As Regnault's results are those generally adopted, these have been accepted throughout this work.

Water Density and Compressibility.—Up to a temperature of 212° F. experiments on density of water have usually been made under a pressure of one atmosphere, whereas above this point the pressures have been various. It has therefore been necessary to correct the densities found by Hirn, and by Ramsay and Young, for pressure, and this involves a knowledge of the elastic compressibility of water at all temperatures. Col. 4 gives this value up to 500° F., from 32° F. to 140° F. being values by Pagliani and Vicentini, and above this point the values are taken from Ramsay and Young's experiments. With the help of these values col. 5 has been compiled. It contains the densities of water at the various boiling temperatures. It should, however, be noted that these values are not direct observations, and it is more than probable (see retarded ebullitions, p. 57) that just before bursting into steam there is a relatively large increase of water volume.

Critical Temperature and Pressure.—When a volatile fluid like CO_2 is warmed under pressure, it will be noticed that at a certain temperature, called the critical temperature, fluid and vapour cannot be distinguished from each other. No increase of pressure will effect condensation. The lowest pressure at which this change takes place is called the critical pressure. According to Cailletet and Colardeau the critical point for water is reached at 200·5 atmospheres and 680° F.; above this temperature water does not exist, only steam. At the critical temperature it is evident that there is no latent heat of evaporation, because there is no evaporation. This means that if the values of the latent heats in cols. 12 and 13 were extended to 680° F., they would be reduced to nothing. Regnault's values seem far more likely to fulfil this condition than the other two.

It is also evident that at the critical temperatures the volume of water must exactly equal the volume of steam, for the two are then identical; this means that if cols. 8 and 19 and 20 were extended their values ought to grow equal to each other when a temperature of 680° F. is reached.

Seeing that water on being heated under pressure approaches nearer and nearer to the conditions of steam, and that steam and water are undistinguishable from each other at the critical temperature, it is more than likely that the change from steam to water or *vice versâ* even at low temperature is not so abrupt as is generally believed.

The following further properties of water may on occasion be of service.

The Boiling Point of Salt Water is higher than that of pure water, so that steam evolved from brine must be slightly superheated; but it is only with the greatest difficulty, even in laboratory experiments, that this has been verified.

Boiling Temperatures of Salt Water

Percentage of salt added to 100 water	0	14·9	25·5	35·5	40·7
Boiling temperature at atmospheric pressure . . °F.	212	216·5	221	225 5	226·8

Gerlach, 'Zeitschr. f. analyt. Chemie,' 1887, vol. xxvi. p. 413.

The Specific Heat of Salt Water is less than that of pure water, as will be seen from the following table. As the density of water increases with the added salt, the heat capacity of water does not sink quite at the same rate as the specific heat.

Table of Specific Heat of Salt Water

Percentage of salt added to 100 water . .	0	1·6	4·9	11·5	12·3	24 3
Specific heat of salt water. . . .	1·00	0·978	0·945	0·877	0·871	0·791

J. Thomsen, 'Pogg. Ann.,' 1871, vol. cxlii. p. 337; A. Winkelmann, 'Wied. Ann.,' 1873, vol. cxlix. p. 1; Marignac, 'Ann. Ch.,' 1876, vol. viii. p. 41.

Boiling Phenomena.—If one had never seen water boiling, and if one were asked to describe what one would expect to occur if water of a boiling temperature were placed in contact with a hotter plate, one would be tempted to say that, at first, innumerable small bubbles would form all over the surface, like dampness on a window pane; that these bubbles, like the water drops on the glass, would increase in size, and would join together, and when an individual had acquired a sufficiently large size, it would break away and rise to the surface, the space thus left bare being at once re-covered with minute bubbles. It is, however, well known that water does not boil like this. When only little heat is applied to the plate one may have to wait for long periods between the appearances of bubbles, and when they do show they grow so suddenly and tear themselves away so quickly that the process cannot be watched. It will, however, be noticed that, on its upward journey, each bubble increases in size, especially at first. This cannot be due to the decreasing head of water over the bubble, for it would have to rise 30 ft. before its diameter is increased to 30%.

The increased size cannot be due to the inertia of the water; for assuming that the bubble is formed with such rapidity as to impart an appreciable velocity to the mass of water above it, and that in coming to rest the water exerts a suction on the bubble and thus increases its size, then clearly after the first increase the bubble would decrease again. This does not seem to be the case; besides a suction which would double the volume of the bubble would have to be about 7 lbs. per square inch, and for however short a period this reduced pressure might exist, it would cause the whole body of the nearly boiling water to burst into steam. Only one further explanation suggests itself, and that is retarded ebullition, which means that the water is superheated.

Retarded Ebullition has for generations done service as an explanation for boiler explosions (L. E. Fletcher, 'Experiments on Red-hot Furnaces,' Manchester; A. Witz, 'Comp. Rend.,' 1892, vol. cxiv. p. 41), but since these have been carefully enquired into the theory has never been confirmed. It is however well known that, if water be carefully heated in a very smooth vessel, the temperature may be raised several degrees above 212° F. before the water bursts into steam. Chemists are sometimes troubled by retarded ebullition, which now and then may eject the contents of test tubes, slight superheating will produce this phenomenon; for if ebullition takes place when only $\frac{1}{2}$° F. superheat has been reached, then the volume of steam which is evolved is about equal to the volume of water. Of course if this sudden ebullition takes place in a closed vessel hardly any bubbles would be seen, for the increase of pressure due to half a degree of temperature is only one-sixth of a pound, and this slight increase would at once check the sudden ebullition. Boiler explosions from this cause are therefore impossible. If cold water be heated in a glass vessel over a flame and properly illuminated, one can see the change of density of the locally heated water. If boiling water be gently heated, the irregular density is not very marked until boiling actually commences, then it will be seen that the heating surface is covered with a layer of what appears to be less dense water, and in all probability this is superheated water, though judging by the small bubbles which are occasionally evolved from it, the amount of superheat can hardly be more than $\frac{1}{1000}$° F. If now a bubble be formed on the heating surface, the surrounding superheated water gives up its steam, and rapidly enlarges the bubble, which then rises to the surface. The current which its passage produces draws more superheated water to the bubble's birthplace and along its path. Every succeeding bubble will therefore most likely start at the same point and travel along a path which has now become a channel for superheated water, and it is this water which gives up its steam to the bubbles and causes them visibly to increase in size as they rise to the surface.

After a time the superheated water will be distributed throughout the mass of hot water, and if small solid particles are floating about, they may form the starting points for a series of bubbles. A somewhat similar condition of things undoubtedly exists in effervescent waters which have just been poured into a glass. These are supersaturated with carbonic acid gas, which concentrates around small floating cells and is there liberated, but steam is also a gas, and superheated water is water supersaturated with steam. There is, however, this marked difference, that steam has a much greater latent heat of evaporation than carbonic acid, and its cooling effect is greater, but this is in a way balanced by its greater volume.

Erratic Ebullition.—If the above view be accepted, it is easy to understand why the generation of steam is a jerky performance, notably in water-tube boilers. Suppose that in a bottom tube of a Belleville element, of say 100 ft. total length, a steam bubble be suddenly formed; it will have to set the whole water column in motion; to arrest it a suction force would have to be applied in the lower column, which means a reduction of pressure. In the early days of this boiler, this action caused much trouble, for instead of setting the upper 100 ft. column of water in motion, it was easier for the steam to drive the water back into the downcast tube, and a backward circulation was sometimes set up. In modern Belleville boilers, non-return valves are fitted at the entrance to the lowest tubes, and the jerkiness of the steam generation is now an essential advantage to this boiler, for the suction which follows each outburst of steam draws fresh water into the lower tube and thus materially assists circulation.

However, this boiler has gone out of use, and other water-tube boilers have to depend on rapid circulation for the removal of steam bubbles. They are mentioned here because they illustrate more clearly than is possible with cylindrical boilers how pulsations occur. If in the lower rows of water tubes there are many large bubbles adhering to the upper surfaces of the tubes, these will act as powerful non-conductors, reduce the rate at which steam is generated and circulation maintained causing the tubes to overheat. Suddenly some of the stagnant steam escapes, and the local circulation which it sets up in one part of the boiler is communicated to other parts, and for a time all the bubbles are swept out of their tubes. This period is followed by one in which the overheated tubes produce much steam, and the good circulation is maintained until the tubes are comparatively cold again. In a cylindrical boiler such irregularities in the rate of ebullition may be brought about by the steam bubbles which cling to the furnaces and tubes retarding heat transmission and circulation, but on being liberated both are intensified. No experiments seem to have been made with a view of ascertaining from which plate surfaces bubbles are most easily liberated, though the impression prevails that the rougher they are the better.

Priming.—Information on this subject is as yet almost non-existent. The generally accepted views are that low-pressure boilers prime far more than high-pressure ones, and therefore require more steam space, or perhaps more steam height; also, that throttling the steam at the main stop valve reduces or stops priming, and that the injection of oil, particularly mineral oil, is a still more efficient remedy. Soda and salt seem to increase priming, probably by forming soaps with the oils. The action of mineral oils may be compared to the action of oil on troubled waters: it prevents the formation of light bubbles. That the partial closing of the boiler stop valve has an effect in reducing priming, while linking up the engine, so as to use the same reduced amount of steam, has not, clearly points to the necessity of imparting violent motion to the steam, or rather to the as yet uninjured froth, so as to burst the bubbles.

J. T. Thornycroft ('C. E.,' 1890, vol. xcix. p. 41) argues this point very clearly, and has effectively demonstrated that a very small steam space will suffice, if only the mixture of steam and water is properly guided. In his water-tube boiler he admits this mixture at the upper circumference of the dome (fig. 88), and dashes it against an internal baffle plate, whereupon the water falls to the level W and the steam is carried off through the pipe A. This small dome separated sufficient steam for 774 I.H.P., equal to about 10 cubic ft. of steam per second, while the internal diameter was only 26 ins., and the clear height from water level to crown of baffle plate only 15 ins. An ordinary double-ended marine boiler of the same power would require at least 48 ins. of steam-space height. It must, however, be admitted that in his more recent practice the steam drums are larger than they used to be.

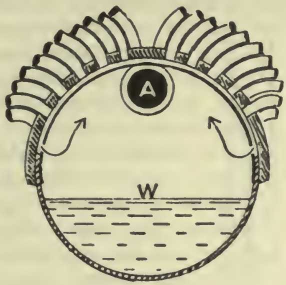

Fig. 88

That this large space is necessary need not be wondered at; for, in spite of the wide water space between the nests of tubes, which are primarily intended to act as downcast shafts, facilitating circulation, they are very far removed from doing their work properly. Perhaps

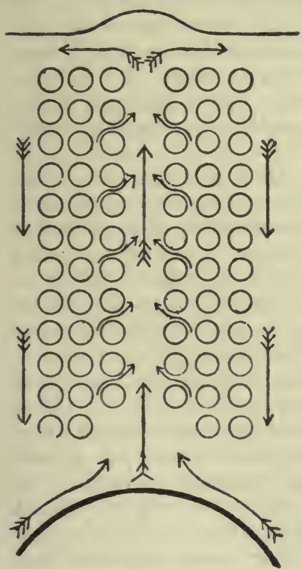

Fig. 89

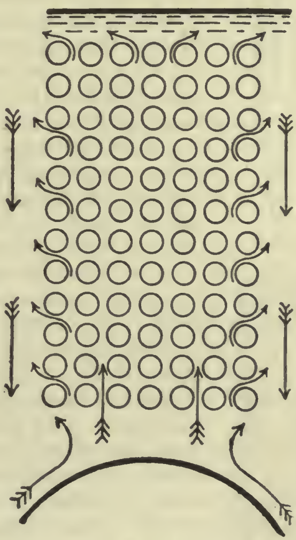

Fig. 90

more than one-third of the total evaporation takes place at the furnace crowns, but instead of allowing the steam to ascend to the water level as freely as possible, by removing a central row of tubes (fig. 89), it has to rise as best it can (fig. 90), and naturally struggles towards the water spaces, where it comes in conflict with the downward current. By closing up all but the wing water spaces, this struggle can be made to grow so severe that the imprisoned steam raises the upper water level, as, for instance, in locomotive and navy boilers. The consequence is that certain parts of the heating surface are more often exposed to steam than to water, and even though they may not get burnt, are occasionally ineffective.

A half-hearted attempt to direct the current of steam and water is sometimes made by bolting a few plates horizontally to the lower row of steam-space stays; but to be really effective they ought to cover the whole water level, with the exception of the wings, and no downward currents should be allowed at the centre water spaces.

In some boilers flat sheet-iron tubes are fitted to wings, reaching to the boiler bottoms, but it is to be feared that the inducement for the water to circulate through them, while there are wider water spaces on either side, is not a very strong one. Undoubtedly, however, the water which does enter them must fall below the furnaces.

Improvements in circulation, though not a remedy against priming, could sometimes be effected by removing a central row of tubes from each nest, and securing vertical plates to the outside rows of tubes, which would prevent any steam from entering the water spaces. By placing horizontal plates over each nest of tubes (fig. 90) the separation of steam and water would also be effected; but all loose parts in a boiler are a nuisance.

The notions as to what takes place inside a boiler are exceedingly vague, and it may be of interest to draw attention to a few points.

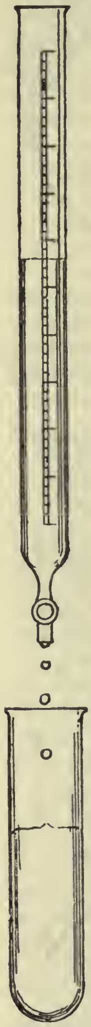

Fig. 91

Take, for instance, a boiler 15 ft. diameter and 11 ft. long. Its steam space will be about 4 ft. high and have a capacity of 450 cubic ft. It will be capable of evaporating about 10,000 lbs. of water per hour under natural, and double that quantity under forced draught. The total area of the water level is about 150 sq. ft., so that under forced-draught conditions every square foot of this water level emits 133 lbs. of steam per hour, or $\frac{1}{28}$ lb. per second. At 180 lbs. working pressure this is equal to a volume of $\frac{1}{12}$ cubic ft., or 144 cubic ins., per second, while at atmospheric pressure the volume would be more than 10 times as great. But even 144 cubic ins. of steam per square foot of water surface is a large quantity, amounting to about two bubbles 1 in. in diameter per second per square inch of surface, or 2,000 per second if only $\frac{1}{10}$ in. in diameter. No wonder, then, if the water contains any frothing substances, that the bubbles will not burst till they are carried into the steam pipe, particularly as it takes less than forty seconds to remove all the steam contained in the steam space.

It is well known that it is quite impossible to make soap bubbles in an electrified atmosphere, and it is very surprising that in the days of low pressures no attempts were ever made to turn this knowledge to practical account.

Attempts to measure the amount of priming water were first made by Professor Thurston ('Am. C. E.,' 1874), by blowing part of the steam intended for the engines into a calorimeter, and measuring its specific heat. Comparing it with Regnault's results, the amount of suspended water could be calculated. This method being too complicated, the author, while at sea, tried to determine the amount of salt carried over by the steam, first by means of a salinometer, and then by means of the chloride of silver test, but was unable to detect any. During the trials of the Research Committee (' M. E.,' 1890) and later C. J. Wilson has, with the help of a more delicate test, been able to measure the priming water with great accuracy, and in all future trials of boilers the tests, which are very simple, should be repeated. They can easily be carried out at sea, and are also of special value to determine whether the condenser is leaking or not. To a measured sample of condensed steam a drop of a solution of yellow chromate of potash is added, and a weak solution of nitrate of silver—say 1 per 1,000—is slowly added, until the water suddenly turns brown (fig. 91). Repeat the experiment with a much smaller sample of water drawn direct from the boiler. It may require relatively from 20 to 100 times as much nitrate of silver as in the last case, and the ratio between the two is the ratio of priming water to steam. When testing the condenser water, in order to ascertain whether there are any leaky tubes, the comparison must be made with sea and not with boiler water. As a check on these results add to the condensed water as much boiler water as, according to the test, was primed over, and repeat the experiment. The amount of nitrate of silver should then, of course, be twice as much as that used in the first test. The water which was drawn from the boiler has been partly evaporated by its own heat. The correction is made by ascertaining the temperature of the steam with the help of any reliable table. Subtract it from 1,178° F., then divide by 966, and multiply the previously found percentage of priming water by this quotient.

Great care has to be taken that no solid salt is contained in the brine cock, as this would materially affect the result; and it is also of importance to draw the steam out of the main steam pipe as near the stop valve as is possible, otherwise the priming water will have settled on the walls. For rough estimates it will suffice if the drain water from the high-pressure valve chest is taken. When the priming is to be measured of a marine boiler which uses no sea water, it is necessary to add a little pure salt—about $\frac{1}{3}$ oz. per gallon of water, or 5 lbs. per ton. This quantity is sufficient, yet it is so small that it cannot possibly do any harm. Probably the most convincing method is to use a Carpenter separating calorimeter. Steam is taken from the main steam pipe, passed through a separator, where it parts with its moisture, and is then condensed; the ratio of the two quantities of water is the amount of priming.

Circulation in Water-Tube Boilers.—Elaborate and instructive experiments have been made on this subject by Professor W. H. Watkins, 'N.A.,' 1896, vol. xxxvii. p. 267, who constructed a large number of models, using glass tubes instead of metal tubes and introducing oak sawdust into the water to show the circulation. It was then found that, especially in the large tube boilers, like the Belleville, the movements of the steam bubbles were no indication as to the movement of the water; that, in fact, the water remained fairly stationary while the bubbles moved through it. It could frequently be noticed that little heaps of sawdust collected in the more horizontal tubes just below the points where the flames were striking them, even although much steam was being generated a little further on just over the flames. Occasionally in the Belleville boiler model these heaps would be swept forward, due no doubt to a momentary suction assisted by the action of the non-return valve. This stagnation was not so marked in other large-tube boilers, and seemed to be a rare occurrence in the Babcock and Wilcox model boiler, but the finding of loose scale in the lower tubes of full-sized ones shows that their circulation is not as powerful as one would imagine. Certainly this boiler offers very much less resistance to water circulation than the Belleville boiler, but it suffers from the disadvantage that the water which passes through the lower tubes has overcome more resistance than that which passes through the upper ones.

In small-tubed boilers the sawdust and the bubbles travel together, and while steam is being generated a most violent circulation is set up. The motive force is the difference of the pressure of the column of mixed steam and water moving upward as compared with the pressure of the solid water in the downcast tubes. This motive force can of course never exceed a head of water corresponding to the height of the boiler, and a theoretical limit to the steaming power of small-tubed boilers is therefore fixed.

The limit is a high one, possibly exceeding an evaporation of 200 or 300 lbs. of water per square foot per hour with one-inch tubes in boilers of ordinary dimensions, but this consideration shows that attempts to reduce both the diameters of water tubes without increasing the heights of the boilers may lead to failures.

Circulation Reversed.—Another important matter revealed by experiments like the above is that the currents in water tubes can be reversed, that the bubbles instead of rising can be made to descend, and then to rise in the upcast tube. This condition can be easily reproduced if in a U tube the downcast is first heated until much steam is generated there, then if the flame be moved from this tube to the downcast, there will be no reversal of current, the bubbles will be seen to form near the flame in the old downcast, travel downwards and then up the old upcast tube, which is of course not being heated. On rare occasions a similar result is produced in experimental boilers, and is then doubtless due to the already mentioned jerky generation of steam. In some glass models, notably in that of the Sterling boiler, one can notice that when the flame is just giving the proper amount of heat the generation of steam in the front rows of tubes may be so sudden that the bubbles are shot both upwards and downwards. In water-tube boilers of the Yarrow type having no regular downcast tubes, it is probable that reversals of current often take place in those tubes not in immediate contact with the fire. This is perhaps no disadvantage.

Circulation Power of Bubbles.—The following simple considerations may be of some assistance in arriving at a correct view of the part which steam bubbles play in producing circulation. If a beaker with water be placed on a spring weighing machine, and a sphere of some heavy material be raised or lowered with a velocity of v feet per second, the balance will indicate an upward or downward pressure of about $\frac{v^2.d^2}{400}$ pounds, where d is the diameter of the sphere measured in inches. If the sphere be made to travel upwards with such a velocity that the pressure is exactly equal to the weight of water displaced by the sphere, then the maximum velocity is found with which bubbles of the diameter d can rise towards the surface.

$$v = 2{\cdot}7 \sqrt{d''}.$$

Bubbles of $\frac{1}{9}$ in. diameter could move upwards with a velocity of 0·9 ft. per second, the resistance encountered would be $\frac{1}{40000}$ lb., equal to a head of $\frac{1}{145}$ in. If eight such bubbles combine, forming one large one of $\frac{2}{9}$ in. diameter, its upward velocity would be 1·22 ft. per second, and its resistance equal to a head of $\frac{1}{8}$ inch water (C. H. R. Sankey, 'C. E.,' vol. cxli. p. 133, finds 1 ft. for $\frac{1}{8}$ inch bubbles as a limit; larger bubbles are probably too irregular in shape.) The total relative resistance offered to the eight small bubbles is the same as to the single large one of the same volume. The forces which oppose the upward movements of the bubbles force the water upwards, the available head of pressure being that just found divided by the sectional area of the channel. Assume that (fig. 50, p. 29) a vertical tube of 1 sq. ft. sectional area and 6 ft. high were placed over a furnace which is producing 144 cubic inches of steam per minute, and assume the bubbles to be $\frac{1}{9}$ inch diameter, then they would rise through 6 ft. in 6·7 seconds and at any time the tube would be filled with $6{\cdot}7 \times 144 = 860$ cubic inches of steam or 800,000 bubbles, of which each one would exert an upward force of 1 : 40,000 lbs., resulting in a total circulative force of 20 lbs. per sq. ft. or about $\frac{1}{4}$ inch head of water. Assume another case in which the bubbles are $\frac{2}{9}$ inch diameter, then their velocities will be 1·8 ft. per second and they will pass up the 6 ft. tube in 3·35 seconds, so that the tube will only contain half the above-mentioned volume of steam, viz. 430 cubic inches, with the result that the head of pressure is reduced to about $\frac{1}{8}$ inch and the circulation is correspondingly reduced. Seeing that the heat transmissions from a plate to water increases with increased circulation, the best results will be obtained in those boilers which produce the smallest bubbles. This is the most feasible explanation for the fact that locomotive boilers work far more satisfactorily when constantly trembling while running, than when used for stationary purposes.

In water-tube boilers the movement of bubbles is based on somewhat more complicated conditions. Assume that the bubbles in a vertical water tube are moving upwards, each bubble having a mean length l, and separated from its neighbour by a water space of the length L. The diameter d of the bubble is not quite as large as D, the bore of the tube, as could be seen if a glass tube and coloured water be used. The buoyancy of the bubble is $\gamma.l.d.^2 \frac{\pi}{4}$, where γ is the density of water. The sum of the pressure against its upper side and suction on its lower side is $p = \gamma.l.$ or $l. =$ head of pressure. This pressure, p, imparts a downward velocity, v, to the film of water round the bubble: v feet per second $= 5 . \sqrt{l}$ (approximately). The work performed by the bubble in its upward course at a velocity V is $V.\gamma.l.d.^2 \frac{\pi}{4}$, and this is equal to the frictional work done by the water on the walls of the tube: $F = \phi\,(v^3 l + w^5 l).\ \pi.\ d$, where ϕ is the coefficient of friction, and w is the upward velocity of the water above the bubble which is found as follows :—

$$(V - w).d^2.\ \frac{\pi}{4} = v\ (D^2 - d^2)\ \frac{\pi}{4}.$$

Combining these equations we get a cubic one which expresses the velocity V in terms of the ratio D : d or *vice versâ*; then finding the maximum value for V, the various other values can all be found approximately

$$V = 1{\cdot}5\ \frac{l}{\sqrt{l}\,.\,(22.d.l - L)}.$$

Here V is expressed in feet per second and the dimensions in inches. D. v and w are found from previous equations. If the rising bubbles have to produce a downward current in other tubes, and if any of the uptake tubes are not vertical, the problem grows a little more complicated.

Whatever deductions may be drawn from these formulæ, or from similar considerations as to the action of bubbles, as to the least height of boilers, smallest diameters of tubes and greatest inclination, experiments must be undertaken before any definite decision is arrived at. Only recently a case was mentioned to the author where a four-inch waterpipe swelled locally to about nine inches diameter before it burst, apparently without a cause. Here the material could certainly not be blamed, nor was scale or grease found, and the only other explanation which suggests itself is want of circulation.

Moist Steam.—It has been clearly demonstrated that any moisture carried over with the steam very materially reduces the efficiency of an engine, especially by increasing the initial condensation and re-evaporation; it is therefore desirable to fit steam traps to the main steam pipes whenever these are long and likely to cause condensation. On trials the moisture of steam should be measured; this should be done by a separator calorimeter, Carpenter's being a very serviceable one.

Superheated Steam.—The specific heat of superheated steam under constant pressure varies from 0·6 at 290° to 0·8 at 408° F., and should not be confounded with the co-efficient 0·305 in Regnault's formula for the total heat of evaporation, for it occupies a position between the two specific heats under constant pressure and under constant volume. After having expended much heat on the production of saturated steam, only a little more is required to superheat it. If this is done, all moisture is of course removed, and if sufficient heat has been added by this means, the condensation which would otherwise take place in the engine, due to work done, will not take place. It is very desirable to attain this object, but repeated attempts of a sustained nature both at sea and on land have borne no lasting fruit, the mechanical difficulties seeming more than to balance the economic results, which may be stated to be from 10 to 15 per cent.[1] The chief troubles have been: cutting of the high-pressure cylinders and valve faces, occasional burning of the superheater tubes and of the engine's cylinder lagging, and then there is of course the danger of melting the brazed seams in the copper pipes. These troubles are in the author's opinion due chiefly to the absence of suitable bye passes for the funnel gases. It should be possible to divert these from the superheater when the engines are not working, or when the steam temperature has grown too high.

Feed Heaters. — A study of properties of steam (see tables, p. 50) shows that by far the greatest amount of heat is spent in converting water into steam, only about one fifth of this heat being used for raising the temperature. All the heat of evaporation is carried away by the engine condenser water, unless before the steam enters the condenser a part is abstracted for feed-heating purposes. It would, of course, be useless to employ the exhaust steam from the low-pressure cylinder, for this is nearly as cool as the condenser and the feed; it is therefore customary to take the steam for the heaters from the valve chest of the low-pressure or intermediate cylinder. If the engines are of the triple expansion type, the abstracted steam has then done respectively two-thirds or one-third of its share of work, but the expenditure of heat is only about one-tenth of what it would have been if this steam had been passed on to the condenser. The saving is therefore considerable.[2] The steam for these heaters should be drawn from the bottoms of the valve chests, so as to carry with it any condensed water.

[1] See 'C. E.,' 1891, vol. cviii. p. 474.
[2] See J. A. Normand and others. 'N. A.,' 1895, vol. xxxvi. p. 35.

CHAPTER III

CORROSION

Experiments and papers on this subject are fairly numerous, and before discussing the various theories it will be advantageous to mention them. R. Mallet, 'Brit. Assoc.,' 1838, vol. viii. p. 253; 1840, vol. x. p. 221; 1843, vol. xiii. p. 1; and 'N. A.,' 1872, vol. xiii. p. 90. —In these experiments and papers the question of boiler corrosion is hardly touched upon, but galvanic action and related subjects are thoroughly discussed. Crace Calvert, 'Manch. L. Ph.,' 1871, vol. x. p. 99, shows that air in water causes corrosion.

'Parliamentary Reports,' Admiralty Committee on Corrosion in Boilers, appointed in 1874, three reports, 1874, 1878, and 1880, c. 2662. —These experiments were very exhaustive; they aimed at ascertaining whether there was a difference between the behaviour of iron and steel, whether the lubricants in the engines affected boiler corrosion, and at determining the influence of zinc, of galvanic action, of various fluids, and of air in water.

D. Phillips, 'C. E.,' 1881, vol. lxv. p. 73, discusses the above reports, and comes to the conclusion that iron does not corrode as fast as steel. W. Parker, 'I. and S. I.,' 1881, p. 39, like the Boiler Committee, exposed various materials in actual boilers, but isolated each plate. The results do not show a great difference between iron and steel. D. Phillips, 'Marine E.,' 1890-91, adds further experiments in support of the above-mentioned views.

Other experiments on corrosion will be found in the following papers:—M. Lodin, 'Comp. Rend.,' 1870, vol. lxx. p. 321. A. Mercier, 'An. Mines,' 1879, 7th ser. vol. xv. p. 234, gives experiments on the influence of fatty matter on corrosion of iron and steel. M. Lodin, 'Comp. Rend.,' 1880, vol. xci. p. 217, experiments on corrosion of various wires in hot fluids to which vegetable matter has been added. M. B. Jamieson, 'C. E.,' 1881, vol. lxv. p. 323. J. Norris, 'N. A.,' 1882, vol. xxiii. p. 151, determined the influence on corrosion of the air contained in water. L. Gruner, 'An. Mines,' 1883, 8th ser. vol. iii. p. 5, relative corrosion of 28 materials under 4 different conditions. This paper contains an appendix by Bustein, which shows that exposures of the samples for 112 days inside a boiler, and also in a boiler flue, affected the mechanical properties (see p. 145). J. R. Fothergill, 'M. E.,' 1884, p. 339; F. Marshall, p. 344. Professor Lewes, 'N. A.,' 1887, vol. xxviii. p. 247, influence on corrosion of the air contained in water, and general views on corrosion, chiefly in sea water. A. C. Brown, 'I. and S. I.,' 1888, ii. p. 129. Professor Lewes, 'N. A.,' 1889, vol. xxx. p. 340. T. Andrews, 'C. E.,' 1884, vol. lxxvii. p. 323, and 1885, vol. lxxxii. p. 281, gives experiments on corrosion in sea water. A. Wagner, 'Bayerisches Industrie u. Gewerbe Blatt,' 1875, p. 102, various theories on the chemistry of corrosion, and gives the results of his experiments on the action of various salts in hot and cold water. Bursteyn, 'Mitt. Pola,' 1879, vol. xii. p. 503, experiments on the influence of high pressure and fatty matter. H. Schnyder ('Berg.- H.-Z.,' vol. xxvii. p. 212), experiments on the behaviour of zinc under various conditions. J. Dewrance, 'C. E.,' 1900, vol. cxli. p. 107, deals with air in boilers, and the discussion refers chiefly to corrosion in H.M.S.'s water-tube boilers. Prof. Heyn and Bauer, Mitt. Berlin, 1908, 1909, determined the corrosive influence of about forty salt solutions. J. N. Friend, 'I. and S. I.,' 1911, repeated some of these experiments.

The main object of all these experiments is to ascertain the true causes of corrosion, and to discover means for preventing it; and since steel has been substituted for iron the question of its relative liability to corrode has repeatedly come to the front. No engineer with extended experience will hesitate to admit that steel does behave worse than iron. Fortunately this view has not only not led them to return to iron, with its numerous bad properties, but it has driven them to adopt preventives which have now reduced corrosion in both iron and steel boilers to very small proportions. These preventives are—

I. The substitution of mineral lubricants for animal or vegetable fats or oils. II. The use of fresh or even distilled water wherever obtainable instead of sea-water. III. The removal of air from the feed water. IV. Electric currents. V. The use of alkalies.

In order to understand how these practices have been arrived at it will be necessary to discuss the various theories of corrosion.

I. and V. **Action of Engine Lubricants.**—All vegetable and animal fats or oils are of the nature of salts, for they consist of bases and acids. In fats and oils the base is glycerine, and the acids have numerous names. Fats and oils, like some other neutral compounds —for instance, the carbonates of lime and magnesia and the sulphate of iron—can be split up into acids and bases by heating them. The temperature at which this splitting up takes place with fats is a little above 212° F. Thus stearine, which is the same substance as stearic acid, is manufactured by raising tallow and water to the proper temperature. Naturally the same process takes place in boilers, and the liberated fatty acids are now capable of corroding the same amount of iron as an equivalent weight of sulphuric acid would attack. The resultant compound is ferric soap, which makes up the greater part of the filthy greasy substance to be met with in some boilers. By bringing the fatty acids in contact with other substances, other soaps are formed. Thus with potash we get soft soap, with soda hard soap; with lime we get the very much harder soap called putty, while with the oxides of lead and zinc medical ointments are the products. Rosins which are the chief constituents of varnishes, behave very much like oils and should not be introduced into boilers.

Whereas glycerine is one of the weakest bases, lime, potash, and soda are the very strongest, and whereas the one is capable of retaining the fatty acids at a boiling temperature, the others will only part with them in presence of a stronger acid. Carbonic acid is in a sense weaker than some oil and fat acids, and can be replaced if they are brought into contact with carbonate of soda. Some of them are not strong enough to do this direct, but must first attack the iron, forming ferric soap. In such a case the soda does not prevent corrosion, while in the previous one the liberated carbonic acid might do just as much harm as the fat. When using organic lubricants it is, therefore, better to add caustic soda or lime instead of the carbonate. Mineral oils are not double compouuds, and, like water, consist of only two elements—viz. carbon and hydrogen—and these are harmless. To add soda where these are in use would be useless.

II. **The Use of Fresh Water.**—Since the introduction of surface condensers sea water has gone out of use, and now that evaporators are largely used the make up water is generally distilled. Nevertheless, sea water does leak past the condenser tube ends, and some of its salts are decidedly injurious. See p. 24.

Chloride of Magnesia.—Professor Lewes states that if sea water be distilled while in contact with iron it gives off hydrochloric gas when the volume of the water has been reduced to one-fifth, but it is only too probable that before it is produced in sufficient quantities to escape it must have been attacking the iron. He also mentions that magnesic chloride and calcic carbonate (lime) react on each other, and are converted into calcic chloride and magnesic oxide, the carbonic acid escaping at a boiling temperature. When exposed to the influence of the air magnesic oxide absorbs carbonic acid, so that on re-filling a boiler which has been opened for some time, and heating it, this acid gas is given off again. He looks on magnesium salts as being decidedly injurious to the life of a boiler; but, as all sea water contains them, this should, if possible, never be admitted.

The Boiler Committee tube No. 21 contained chloride of magnesia and attacked iron and steel very severely.

A. Wagner shows that if water contains chloride of magnesia, but no air, it commences to attack iron at a temperature of about 212° F., while the following chlorides will only attack it in presence of air:—they are arranged in the order of their corrosive power—ammonium, sodium, potassium, barium, calcium.

The experience which is accumulating with the Manchester Steam Users' Association does not confirm these conclusions; at any rate, pure chloride of magnesia and water will not attack iron either cold or at a boiling temperature even if oxygen be forced into the water, and as far as can be ascertained the results mentioned above are due to absence of precautions for the rigid exclusion of even a trace of carbonic acid, such as distilling caustic soda solution *in vacuo*. Such a condition cannot of course be attained in a boiler, and for practical purposes the advice to exclude chlorides of magnesia is a good one.

Chloride of Lime acts similarly to chloride of magnesia.

Sulphates of potassium, sodium, calcium and magnesium seem to protect iron, but sulphates of manganese and especially of ammonium seem to corrode it.

Ammonium salts attack iron very energetically.

Magnesium Carbonate deserves special mention, for although it is apparently inactive up to a temperature of about 350° F., corresponding to a pressure of about 120 lbs. per sq. inch, yet above this temperature this salt splits up into carbonic acid and oxide of magnesium which forms a floury deposit, a mud which settles on furnace crowns and causes collapses.

Carbonates.—According to Heyn and Bauer these salts are corrosive, and the sodium and potassium carbonates unquestionably produce pitting. According to the author's analysis of their results it would seem that the protective power of these salts is a local one or patchy, leaving the unprotected parts to be severely attacked (pitting) by galvanic currents which are set up between the protected and unprotected parts. All soda salts caused pitting, the severest actions being with mixtures of common salt and soda; the latter salt should therefore not be usedif there is a chance of sea-water leakages entering the boiler.

Alkalies are not absolute protectors, for in high pressure boilers they do not neutralise the free carbonic acid which is introduced with the feed. In fact carbonate of soda is slowly converted to caustic under high pressure conditions.

Oxygen and Carbonic Acid.—Many experiments have been made to prove that oxygen will or will not attack iron unless there is at least a trace of carbonic acid present in the water. These researches, including those of the Manchester Steam Users' Association leave no doubt but that corrosion in the presence of oxygen is very materially reduced if carbonic acid is excluded from the water. Nevertheless, corrosion usually does occur after a time in the presence of pure oxygen, but only very locally, as if some of the microscopic impurities in the steel or iron had taken the place of the carbonic acid; these and silica salts cannot be entirely excluded from boilers so that even when all carbonic acid is removed, corrosion may still be expected unless the oxygen too is excluded.

Common Salt, if pure, seems to do no harm to iron, but even the best commercial salt usually contains some chloride of magnesia.

Free Acids, which are sometimes proposed for boiler-scale solvents, should not be used, nor should the feed water ever be taken from a river near a chemical factory. Occasionally waste acids are discharged there, which may do a serious amount of injury. This was mentioned by Mr. Hallett ('M.E.,' 1884, p. 350).

Heyn and Bauer experimented with arsenic trioxide, which is occasionally used as an anti-corrosive, and found that it had a protective influence when very dilute. They also tried chromic acid, which, judging by their results with cold solutions, seems to give a protective coating to steel plates (it is known to make steel and iron passive). One ounce per 450 sq. ft. should afford complete protection, one ounce per 1,800 sq. ft. affords partial protection, and one ounce per 4,500 sq. ft. affords very little protection.

Oxidising Salts.—Potassium chromate, potassium bi-chromate, potassium chlorate are also fairly good protectors though the latter salt is rather doubtful, but potassium bromate and sodate cause corrosion.

Deoxidising Salts might be expected to be protectors, but the ferrous sulphate corrodes iron. Potassium cyanide and calcium sulphide are practically neutral, but sodium sulphide and beet sugar are fairly good protectors. Molasses according to reports from sugar works are strongly corrosive.

Copper Salts seem to be an unavoidable, though only a minute, constituent of all feed water, as is proved by the green scale in all old boilers. For some unaccountable reason it does not deposit itself uni-

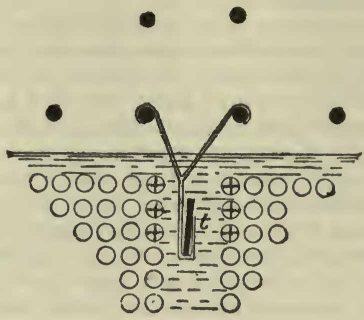

FIG. 92

formly over the inside of the boiler. Its presence is denied by many engineers, therefore little information could be obtained as to the points where it is most generally found. The author's experience is that patches of green scale will be found on the zinc slabs, and also near them on the iron. This is true even when the zinc *t* is suspended from the steam-space stays (fig. 92). It will be found that the tubes marked + sometimes contain thick patches of greenish-grey scale. Small quantities are also found near the water lines of boilers, particularly at both end plates. Larger quantities are found at the front end plates, between the nests of tubes, and on the lower part of the shell plates. The furnace bottoms usually contain the greatest number of patches, but it is difficult to detect them, as these parts are discoloured by grease. The furnace crowns and combustion chambers are nearly always quite free from them, from which it would appear that it is chiefly the non-heating surfaces of a boiler which get covered.

If the copper would only distribute itself uniformly, it would act as a protective scale, but as it is deposited on small areas, these may become sources of danger by producing galvanic currents, at any rate if there are any powerful chemicals in the boiler water, either acids or alkalies, or even neutral salts.

Unless copper pipes are fitted inside the boiler, the only sources of supply are the condenser tubes, the feed pipes, and the pumps. Here again it is the use of vegetable and animal lubricants which causes the mischief, for both readily attack copper or brass, as can be seen by their greenish colour if either has been in contact with these metals. Distilled water also seems to be a solvent; it certainly attacks lead. It is also believed that the minute particles which are worn off the working parts of the pumps are carried bodily into the boiler, and J. MacFarlane Gray ('N. A.,' 1861, vol. ii. p. 157) has detected specks at the bottom of pit holes, and attributes pitting to that cause; and Professor Lewes ('N. A.,' 1889, vol. xxx. p. 340) also believes that the presence of copper causes pitting. The author's observations do not support either view, for although he has been very careful to observe green scale patches, he never could detect signs of pitting near them, nor did the corrosion seem in any way to be increased at these points. These views are supported by Mr. H. W. Hirman's statement, in answer to Mr. MacFarlane Gray (see above), that land boilers using town water pitted, although here there could be no question of the presence of copper. Additional remarks on galvanic action will be found further on.

Zinc Salts.—There remains one set of salts which possibly play an important part in boilers of the present day, but of which no mention is made in any books or papers, and the following remarks are therefore purely speculative. It is a well-known fact that if boiler water is often renewed the reduction of the zinc slabs is more rapid than if the same water is used over and over again. A very obvious explanation would be, that the more zinc salts are dissolved in the water, the less corrosive it is. If this view is correct, the painting of the insides of boilers with zinc oxide, and the addition of some zinc salts—the chloride of course excepted—to the boiler water ought to have a beneficial effect, and recent limited experiences show this to be the case.

Neutral Salts.—The following experiments throw a little light on the part played by what would appear to be perfectly harmless salts in increasing the corrosive power of acids ('Journal of the Camera Club,' London, 1892, vol. vi. p. 52). M. Gourdon's experiments in 1873 show that exceedingly diluted sulphuric acid could still be made to attack zinc metal by adding to it various salts.

Added Salts of	Dilution of Acid in Water
Cobalt	1 : 10,000
Nickel	1 : 7,000
Platinum	1 : 7,000
Iron	1 : 7,000
Gold	1 : 5,000
Copper	1 : 4,000
Silver	1 : 3,500
Tin	1 : 1,500
Antimony	1 : 700
Bismuth	1 : 500
Lead	1 : 400

On the same page L. Warnerke gives a table showing that by adding various salts to a one per cent. solution of sulphuric acid, the speed with which it attacks zinc metal can be varied considerably. The amount of corrosion is given in decimals of millimeters of depth per hour.

Salts added to the Dilute Sulphuric Acid	Speed of Corrosion
Nickel ammonio-tartrate	·13 mm.
Cobalt chloride	·11
Iridium chloride	·09
Palladium chloride	·085
Nickel cyanide	·077
Chrom. chloride	·070
Gold chloride	·070
Silver ammonio-nitrate	·025
Lead nitrate	·010

The interesting point about these results is, that both lead and copper salts stand very low down in the two lists, showing that the accepted notion, that they are very injurious, is, in one sense at least, a wrong one; and the bare fact that these various salts can influence the behaviour of sulphuric acid makes it appear probable that zinc salts also have an influence on the action of corrosive substances found in a boiler—apparently a beneficial one.[1] Possibly experiments carried out on the above lines might show that manganese salts, which enter the boiler water as the steel plates corrode, act injuriously, and this might explain why the presence of this metal in steel has been looked upon as increasing its corrodibility.

The Absorption of Air by a fluid is stated to take place as follows: The volume of gas which a definite quantity of fluid can absorb is independent of the pressure, and decreases with rising temperature. Therefore, as 1,000 cubic ins. of water of 0° C. will absorb 48·9 cubic ins. of oxygen at atmospheric pressure, they will also absorb an equal volume at two atmospheres, and so on. Of course, as the pressure is increased the density of the oxygen is increased, so that the *weight* of absorbed gas is proportional to the pressure. This law also holds good for a mixture of gases.

Air and all gases are absorbed by water and change its density. Thus H. J. Chaney ('Trans.,' 1892, vol. 183*a*, p. 334) finds that 1 cubic foot of water saturated with air is 321 grains lighter than pure water, or about 0·075 per cent. (1 cubic foot weighs about 435,933 grains). It was formerly believed that all air could be expelled by boiling, and that at a given temperature a fluid absorbs a given volume of gas quite independent of the pressure; more recent experiments show that apparently neither view is quite correct. The weight of absorbed gas is therefore proportional to the gas pressure. Experiences as regards occluded gases in metals throw doubts even on this theory. In the following table the volumes absorbed at various temperatures are given, and also the weight of absorbed gas if the

Table of Gases absorbed by Water

Name of Gas		Amount of Gas absorbed by 1,000 c.c. Water under a Pressure of one Atmosphere							
		Oxygen		Nitrogen		Carbonic Acid		Carbon Monoxide	
Temperatures		Volume	Weight	Volume	Weight	Volume	Weight	Volume	Weight
°C.	°F.	c.c.	per 1,000	c.c.	per 1,000	c.c.	per 1,000	c.c.	per 1,000
0	32	48·9	0·070	23·5	0·029	1713	3·38	35·4	0·044
20	68	33·3	0·044	16·5	0·019	942	1·73	24·9	0·028
25	77	31·0	0·040	15·6	0·018	830	1·49	23·4	0·026
30	86	29·0	0·037	14·9	0·017	738	1·31	22·1	0·025
40	122	26·4	0·033	13·6	0·015	607	1·04	20·4	0·022
60	150	23·8	0·028	12·5	0·013	438	0·71	18·2	0·018
100	212	23·2	0·024	12·9	0·012	...	...	19·2	0·017

NOTE.—The above are a summary from various experiments by Bunsen, also A. Winkler, 'Zeitschr. f. phys. Chemie,' 1892, vol. ix. p. 171; and Bohn and Bock, 'Wied. Ann.,' 1891, vol. xlviii. p. 319. The volumes in this table are for the respective temperatures.

[1] Mercury salts are known to be efficient protectors against rust.

pressure is one atmosphere. When a mixture of several gases presses on a fluid, the effective pressure of one of these gases is the product of the combined pressure into the volumetric proportion of the gas. Thus for air, where the volumetric percentages of oxygen and nitrogen are as 20·8 to 79·2, oxygen only exerts a pressure of 0·208 atmosphere, and nitrogen 0·792 atmosphere, and the weight of oxygen absorbed by water when in contact with air is only 0·208 of what it would be if pure oxygen were in contact with the water. Air and steam behave in the same way.

By forcing gases into water, its temperature is slightly raised, the liberated heat being apparently equal to the latent heat of evaporation. This is certainly true for steam, and seems to be nearly true for other vapours, such as carbonic acid, whose latent heat is 134 calories (Muir, pp. 294 and 996). The speed with which oxygen is absorbed by water is according to M. Lodin ('Comp. Rend.,' 1880, vol. xci. p. 217) 0·000036 lb. per sq. lb. per hour at 64° to 68° F.; at 212° F. the rate is ten times faster. Lime water absorbs it at the rate of 0·000047 lb. and eight times faster when hot.

If a closed vessel, fig. 91*a*, contains 1,000 c.c. water and a large volume of oxygen at atmospheric pressure, the temperature being 0° C. or 32° F., then the water will hold 0·07 gram of oxygen in solution (see table). If the temperature be raised to 60° C. the pressure beings till one atmosphere, the water will hold only 0·028 gram of oxygen in solution, but at 100° C., = 212° F., the quantity is but slightly less, viz. 0·024 gram, and it may reasonably be assumed that at the higher temperatures which exist in a boiler the water will absorb about the same quantity. If in the above experiment the pressure were increased to two atmospheres the weights of absorbed oxygen would be doubled, but only if this increase be due to added oxygen. If, for instance, enough nitrogen were forced into the vessel to double the pressure, then there would still be a partial pressure of oxygen of only one atmosphere, and the water would not absorb more of this gas. It would, however, absorb a certain quantity of nitrogen. The same law holds good with a mixture of steam and a gas. Thus assume that the oxygen in the above vessel is at a temperature of 100° C., no water being present, and that water were then to be injected, it would add its own pressure of one atmosphere to that of the oxygen, but the water would not absorb more than 0 024 gram of this gas.


FIG. 91*a*

Feed pumps, if attached to the main engine, are usually designed to deliver about four times as much water as is condensed, and when working normally they naturally suck up a threefold volume of air. This air is subjected to a pressure of say 200 lbs., or about 14 atmospheres, and as the volumetric ratio of oxygen and nitrogen in the air is as 20 to 80, the water will at 20° C. absorb (see table) about $3 \times 0{\cdot}20 \times 14 \times 0{\cdot}0333 = 0{\cdot}28$ times its own volume of oxygen and $3 \times 0{\cdot}80 \times 14 \times 0{\cdot}0165 = 0{\cdot}55$ times its own volume of nitrogen, or a total gas volume of 0·83 times its own; the remaining 2·17 volumes of air being compressed to one fourteenth, or say 0·16 of the feed-water volume. This feed water, heavily charged with oxygen, say 2 ozs. per ton of water, enters the boiler. If it were sprayed into the steam space it would at once be heated to say 390° F., and as the partial pressure of the oxygen in the steam space would not be more than one-eightieth atmosphere, very little oxygen would remain in the heated water. If, however, the feed be allowed to fall to the bottom of the boiler, it will be heated gradually under conditions which prevent the escape of the absorbed air which is retained until the mixture comes into contact with a heating surface. Then while steam is being generated, which escapes, the liberated oxygen attaches itself to the plate and attacks it. If the feed amounts to one ton per hour it will, under the above conditions, contain enough oxygen to corrode 4 ozs. of iron or say 6 lbs. of iron per day.

Oxygen and Carbonic Acid.—It has already been mentioned that oxygen alone, or even when in company of neutral salts, will not attack iron. Carbonic acid will attack iron, forming a ferrous salt which is an active absorber of oxygen, and changes itself into a simple oxide to which the liberated CO_2 is attached mechanically and ready to attack the adjoining iron.

Red or Black Mud.—In new boilers one frequently notices much black or dark red mud, and the plates may appear to be rough or even covered with pit holes, leading one to believe that the boiler is corroding fast. The true explanation will generally be found to be that as in modern boilers the pressure and temperature are high, water acquires the power of dissolving the silica in the mill scale on the plates; this slag contains ferric and ferrous oxide of iron in a finely divided state which sink to the bottom of the boiler. The roughness of the exposed iron plate is therefore a natural condition brought about in the rolling mill and only shown up after the glossy scale has disappeared.

Position of Feed Discharge.—Having admitted the saturated feed into the boiler, there are three ways of dealing with it:—

1. Either lead it so that it enters the bottom of the boiler or easily falls there.
2. Admit it at some point in the boiler—say, over the back end of the tubes, so that it gets thoroughly mixed with the hot water and loses its air.
3. Lead it through pipes which are fixed inside the boiler, in order that it may become sufficiently heated to part with all its air.

All these plans are in use. In the case of No. 3 the internal pipes, if made of iron, suffer very seriously from pitting. If made of copper they also suffer; and in such cases much green scale will be found in the boiler.

No. 2 is the general practice. Few engineers have had the courage to discharge into the steam space, but where tried the plan works satisfactorily, and ought to assist the water circulation, on account of 20% more steam being generated and condensed. Boilers liable to prime could not be worked in this way, as was mentioned by J. H. Hallett ('M. E.,' 1884, p. 350); but at any rate the feed should then be introduced at a point where it will be heated as quickly as possible.

No plan could be worse than to discharge the feed at the bottom of a boiler, for it is clear that, in comparison with other parts, the spaces under and between the furnaces can have only a very restricted circulation. The defect is increased if they are filled with cold, and therefore heavy, water (see fig. 93: the dark spaces at the bottom represent cold water); then, as the fire bars, the ashes, and the inrushing cold air all prevent heat from reaching the bottoms of the

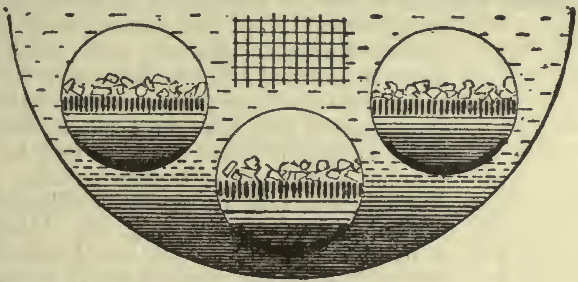

Fig. 93

furnaces, the water has absolutely no chance of rising except by the very slow process of being fed from below. This speed is about 5 ft. per hour, or 1 in. per minute. There is, therefore, no difficulty in heating all the feed slowly and along one particular zone, viz. along the line of the fire bars. Under these conditions the straining to which both the shell and furnaces, but particularly the latter, are subjected must be excessive, for not only is there a sudden jump from cold air below the grate to white heat above them, but on the water side there is also a sudden rise from about 100° F. to over 350° F.

Pitting.—It is, however, not with stresses and circulation that we are at present dealing, but with corrosion, due to air absorbed by the feed water. What takes place with this air is shown in fig. 94. As soon as the cold water comes in contact with the warm part of the furnace plate, F, it is heated and compelled to give up its air, and being in contact with the plate, the air settles on it. There being no circulation, it is only when the bubbles have grown sufficiently large that they rise. But during this period of rest the air which contains oxygen and carbonic acid will attack the iron, and having formed small irregularities, subsequent bubbles find a still better lodgment and speedily effect the formation of pit holes. If the feed is led into the bottom of the boiler, and if it is saturated with air, it can be shown that every inch of furnace length generates about four and a half cubic inches of air per hour. This is equal to about one bubble $\frac{1}{8}$ in. diameter per second. The excessive differences of temperature along this line of grate, and the consequent excessive straining of

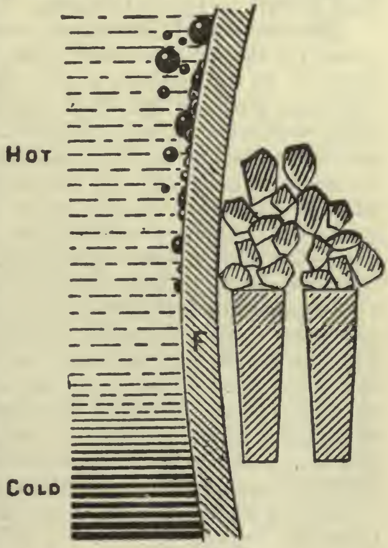

Fig. 94

these parts, quickly loosen all rust as it is formed, so that metallic iron or steel is always exposed to the air if allowed to be produced there. Certain it is that if there is any pitting going on in a boiler, the greater part is sure to be found along the line of fire bars. Another part which is also severely attacked is the under side of the furnace and combustion chamber, for here the air bubbles cannot rise if they have once been formed.

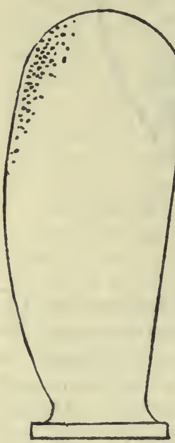

FIG. 95

In support of these views it may be pointed out that severe pitting frequently takes place on the forward side of propeller blades at the leading edge, as shown in fig. 95. Here the air is not driven out by heat, but is abstracted by the partial vacuum which is found there, and, in spite of the high velocity of the water, there seems to be sufficient time for the mischief to be done. Even bronze blades are sometimes pitted at this point.

Another confirmation is found in the severe pitting of internal iron feed pipes; even copper ones waste away. Here the air is liberated by the transmitted heat of the surrounding water, and it has been suggested that long internal iron feed pipes should be fitted to boilers, and renewed whenever they are eaten through, for, whether this is due to the air or some other corrosive agent, it is cheaper to lose a regular quantity of temporary piping than to have to renew furnaces. They last about eighteen months.

Distribution of Corrosion.—One peculiarity about corrosion in general is that boilers which suffer much at their lower parts are often found to be as good as new in their steam spaces. In others the stays and plates of the steam space suffer severely, while the lower parts are only slightly attacked. This happens chiefly when the feed is discharged near the water level, but, as it is next to impossible to say in what path the circulation in a boiler takes place, and whether the feed is carried up or down, definite conclusions cannot be drawn. Another curious fact is, that if a number of boilers are connected with a single feed pipe, that one which is farthest away from the pumps suffers more than the others.

Steam-Space Corrosion.—If it is the air which attacks the steam-space stays and superheater plates, there is perhaps no other remedy than not to admit it, unless it can be proved that zinc here also acts as a protector. It seems to do so, but this can hardly be through galvanic action. Possibly here again it is the presence of zinc salts, of which it is not difficult to imagine that they have been carried into the steam space. In some steamers these parts are whitewashed with zinc white, and the result seems to be a satisfactory one.

It is of course possible that this corrosion is due to steam alone, or to the hydrochloric acid which, as has been mentioned, escapes from sea water when evaporated in iron boilers; but it is safer to blame the air, as it has not yet been shown that steam can attack iron at temperatures ranging near the boiling point of water. If it did, the action ought to be equally strong in all boilers, and that is far from being the case. That this form of corrosion may lead to serious results is only to be expected, because of its rare occurrence and on account of the wasting being very uniform. Thus, rivet heads and flat plates in steam spaces retain their original shape even when most of their substance is gone. As an instance a case may be mentioned where the rivet heads of a steam dome were corroded as shown (fig. 96). The head seemed to be intact, but had in reality disappeared, and several of the rivets could be driven back with a hand hammer. The dotted line shows the original thickness of the plate and size of rivet head.

In another case the steam-space end plate of a boiler was wasted as shown (fig. 97), and it was only due to the necessity of renewing the stays that the corrosion of the plate was discovered, for the shoulder round the nut had been mistaken for the washer which is

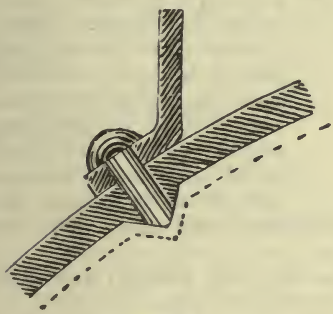

Fig. 96

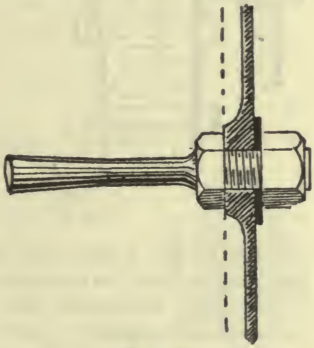

Fig. 97

sometimes placed there. The wasting of the front plates and front ends of the steam stays is aggravated by the heat of the uptake; but, as equally bad cases are met with when baffle plates are fitted, the temperature cannot be the sole cause.

The following is a curious illustration of this sort of corrosion. On examining a pair of boilers of a ship which was barely four years old, it was found that, although efficient baffle plates had been fitted, five of the steam-space stays, situated about 2 ft. above the water level, had lost nearly 30 % of their substance, but only close to the smoke-box end. The starboard boiler was in a perfect condition. Zinc had been used in both, but had been given up. There was only one difference in the structure of these boilers, which is so small that it would not be worth mentioning were it not in connection with the feed. It was found that the knee pipes, K (fig. 98), which had been screwed to the feed inlets, in order to produce downward currents, had fallen off in the port boiler, and therefore a possibility exists that the feed water of that boiler travelled in the direction shown by the arrow F, and would part with its air sooner and deliver it into the steam space.

Internal Feed Pipes burst, although their ends are open; this is no doubt due to water-hammer action. If the boiler water contains salt, its boiling point will be higher (see table, p. 56) than that of the pure feed temporarily stopped; there is every probability that the pipe will have contained steam. As soon as the feed supply was restarted the steam condensed, and the boiler water was sucked towards the valve with sufficient velocity to injure the joints or burst the pipe. A small hole near the flange or an uncemented joint entirely prevents this action.

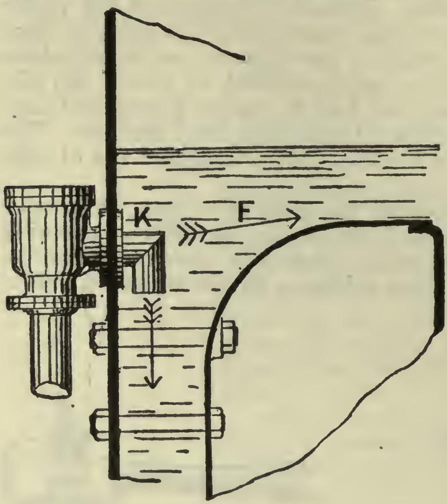

Fig. 98

A most salutary lesson was taught some years ago, when the admixture of air to the boiler feed was advocated as an improvement to the circulation. Its injurious effect was so great that the idea very soon died out. The Boiler Committee also helped to convince engineers on this point, and in all high-class engines care is taken to keep the air out of the boiler, this being the simplest and safest plan for guarding against its corrosive action.

Separate automatic feed pumps are the only efficient means of securing this object.

Galvanic Action.—The laws which have been discovered in this branch of science may be briefly stated as follows: A continuous electric current is only possible in a circuit. Its intensity, measured in ampères, is equal at all points of this circuit, and is proportional to the algebraical sum of the electromotive forces in the circuit (measured in volts), and inversely proportional to the sum of the electrical resistances of the circuit (measured in ohms).

The electromotive force makes its appearance under various circumstances, viz. when two different substances are brought into contact, or when changes of electricity or magnetism take place near the circuit, or when heat is made to travel along part of a circuit. Here only the former case will be dealt with.

If a circuit is constructed consisting only of solids, then no current will be generated, because the sum of electromotive forces equals nought. Therefore, if the electromotive force due to contact of one metal with several others is known, that between two of these is found by the difference. The following example will illustrate this.

The electromotive forces in volts due to contact with iron are—

Carbon	Platinum	Copper	Iron	Tin	Lead	Zinc
−·485	−·369	−·146	0	+·313	+·401	+·600

To find the value when, for instance, carbon and copper are brought into contact we have $- \cdot 485 - (- \cdot 146) = - \cdot 339$. These values change considerably with rises of temperature. If, therefore, one of the points of contact in a metallic circuit is heated, the electromotive force at this point will be altered, and a current produced. Thus, at 530° F. the electromotive force between copper and iron is reduced to 0, and a circuit consisting of only these two metals, but with one joint heated and the other cold, would show an excess of potential of $- \cdot 146$ volt in one direction. All thermo-electric piles are constructed on this principle. With fluids, or with solids and fluids, this simple law does not exist, as will be seen from the following values of electromotive force which appear on immersing any of the above solids in pure distilled water, and also in sea water.

Table.—Electromotive Forces

Fluids	Carbon	Platinum	Copper	Iron	Tin	Lead	Zinc
Distilled water	From +·010 to +·017	+·285 +·345	+·100 +·269	−·148	+·177	+·171	−·505 +·156
Sea water .	?	−·856	−·475	−·605	−·334	−·267	−·565

The electromotive force in the circuit (fig. 99) would therefore be as follows:—

Contact copper and iron . .	− ·146 volt
„ iron and salt water .	− ·605 „
„ salt water and copper	+ ·475 „
Electromotive force in circuit	− ·276 volt

FIG. 99

A Daniell's cell, consisting of copper, zinc, sulphate of zinc, porous cell, sulphate of copper, has an electromotive force of about 1·10 volt.

One effect of the electric current is to produce chemical changes in the fluids through which it passes, usually splitting them up into the elements. It has been found that the amounts are strictly proportional to their atomic weights. Thus a current of one ampère, passing through water during one second, will produce ·0001038 gram of hydrogen (H_2), and $0{\cdot}5 \times 15{\cdot}96$ times as much oxygen (O), viz. ·0008283 gram. If the electricity was produced by a Daniell's cell, which then of course forms part of the circuit, it would be found that during this second its zinc has lost ·003367 gram, and its copper gained ·003279 gram, these quantities being proportional to their respective atomic weights. Iron would have lost ·002900 gram.

This loss of metal at the electrodes is said to be a secondary action, being due to the dissolving power of the elements produced there. Oxygen and hydrogen would be liberated from salt water; but the former gas, being nascent, would combine with the iron, if that is the metal of the electrodes.

It is evident that the chemicals produced by these secondary actions must influence the electromotive forces in the circuit; this is called polarisation. Iron shows this property in a very marked degree.

If the positive electrode be made of iron, which, as has been stated, is a distinctly electropositive metal, it will very soon change its nature, and grow even more electronegative than copper. The unexposed parts will still be electropositive. If the whole piece of the iron is now placed in the fluid, a strong current is set up from its negative to its positive end, and through the fluid back to the negative part; but very soon the current ceases, and on examination it will be found that the electronegative property has spread over the whole piece. Other means for making iron electronegative are dipping it into chromic or nitric acid, or heating it in air—in fact, any process that will oxidise it; and there seems little doubt but that this form of polarisation is due to a scale of oxide of iron, which in some cases is so fine as to be invisible. This would lead to the conclusion that iron ought not to rust beyond its initial stage; but, as it does so, particularly in boilers, there is no alternative but to admit that this beneficial change is not possible at a boiling temperature, or that the galvanic action, which produces it and causes it to spread, does not exist in a boiler, or will not act in the same way on large surfaces as on small ones. Electronegative iron can be brought back to its primary condition by making it the negative electrode. This is probably due to the reducing action of the hydrogen on the oxide of iron.

The presence of hydrogen on the electrodes also influences the workings of a circuit, partly increasing the resistance and partly making the electrode more electropositive. It is, therefore, not surprising that the galvanic actions either do not occur, or remain unnoticed when the electromotive forces are weak. Some allowance should therefore be made for the density of a current; this is measured by dividing its intensity by the sectional area of any particular point of a circuit; and it is of interest to note that the denser it is at the negative electrode, the more ozone instead of oxygen is generated there.

Irregular Potentials on iron surfaces.—If a thoroughly cleaned piece of iron be placed in a neutral electrolit, as for instance, dilute brine, and if two short pieces of rubber tubing be placed on the iron, and polished iron terminals of a sensitive galvanometer be slipped into them, then, if the electromotive condition of the iron plate and the two terminals were the same, there would be no current. But as a matter of fact a current indicating differences of voltage will show itself, and if one of the rubber tubes with its terminal be moved about the plate, considerable variations of voltage will be discovered, indicating either that the cleaning process was not a perfect one or that there are voltage differences in the metal. These differences are spread over large areas, but the microscope reveals minute irregularities which, as they are selectively attacked by etching fluids, must give rise to minute local currents whenever any electrolitic fluid, one which is a good conductor of electricity, is present; which means that concentrated solutions of salts will assist corrosion unless they are alkaline or otherwise anti-corrosive.

Centres of galvanic action can be revealed as follows (see 'Corrosion of Iron and Steel' by Newton and Friend). Prepare a clear solution of gelatine or agar-agar (1·5 per cent.), add 2 per cent. of standard phenalphthalein solution (1 per cent. in strong alcohol), neutralise and add 7 per cent. of a 1 per cent. solution of potassium ferri-cyanide. Immerse a piece of warmed iron into this solution and allow it to cool and solidify, then some regions will develop red others blue tints. The experiment illustrates very clearly the nature of local corrosive action.

IV. Galvanic Action of Zinc.—If slabs of zinc are fitted in such a manner as to be in metallic contact with the iron (see fig. 100), a current passes from zinc to water, to iron, and back, zinc being dissolved and hydrogen evolved on the iron surface.

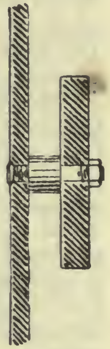

FIG. 100

If the zinc is connected to the iron plates by means of a copper wire, there will be two circuits, viz. one from the zinc to water, to copper, and one from iron to water, to copper; the one current wastes the zinc, the other the iron. If, therefore, the zinc does protect the iron from corrosion, it must be due to some other cause than galvanic action. Copper pipes should produce a current from iron to water, to copper, and the iron should grow electronegative.

The intensities of electric currents which might be expected in a boiler, if no polarisation took place, are very difficult to estimate, as the laws which govern the flow of electricity in conductors of large dimension have not yet been clearly formulated. The resistance of a conductor is proportional to its length, and inversely proportional to its sectional area. For a cubic centimeter and per cubic inch we have the following resistances measured in ohms:—

	Per Cubic Centimeter	Per Cubic Inch
Silver	·000,001,579	·000,000,621
Copper	·000,001,611	·000,000,635
Iron	·000,009,638	·000,003,81
Mercury	·000,094,340	·000,037,1
1 water + ·055 salt . . .	91·2	35·9
1 „ + ·0425 „ . . .	123·0	48·5
1 „ + ·0212 „ . . .	235·0	92·6
1 „ + ·0106 „ . . .	434·0	170·8

Prof. Heyn and Bauer (Mitt. Berlin, 1908), give full details of electromotive differences and of resistances for a large number of salt solutions.

This table shows that the resistance in the metal may be neglected, but it is also evident that, except when the electrodes are almost in contact, the fluid resistance will reduce the intensity of the current excessively.

Take, for instance, a strip of iron 1 in. wide (fig. 101), from which part of the scale, S, has been removed, exposing the iron at Fe to the salt water, W; then a current will flow from Fe to W, to S, and back. Restricting the observation to the small zone of the diameter D, the available section of the conductor is $\frac{D}{2}$—say, $\frac{1}{32}$ in.—and its mean length from Fe to S is $\frac{1\cdot507}{32}$ in. Assuming that the water contains 5·5%

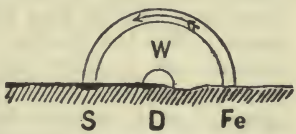

FIG. 101

salt, the resistance of this small element would be ($35 \cdot 9 \times 1 \cdot 507 =$) $54 \cdot 2$ ohms + the resistance of the scale and iron, say 60 ohms.

Assuming also that the electromotive force of iron scale in salt water is the same as for copper, the electromotive force of such an element would be ·276, which would give a current of $\frac{\cdot 276}{60} = \cdot 0046$, which would be capable of corroding ·0022 lb. per day, or at the rate of about $\frac{1}{10}$ in. in 60 days. At one inch distance from the edge of the scale (see fig. 101) the action would be about 3°/₀ of the above, showing how rapidly it diminishes with the distance.

The Galvanic Action of Zinc in Boilers.—If galvanic currents exist, the studs by which the slabs are secured ought to remain bright, at least within one-sixteenth of an inch of the metal; but it is well known to those who take much trouble in keeping up a good metallic connection that corrosion takes place even at the contact surfaces, which have to be scraped or filed after every voyage. The current produced by zinc, iron, and sea water is about twice as strong as that of scale and iron, but as it has no effect on its immediate surrounding, it is hardly probable that it will have much influence on the more distant parts, which are separated from it by several feet of salt water.

In spite of these estimates, it can be shown experimentally that zinc and iron (wire) if placed together in aërated water, but without touching each other, will both corrode; if the iron wire be attached to zinc, only the zinc corrodes. This, however, is only true when certain salts are present in the water, and that may explain the varied experiences as regards zinc.

Some light may perhaps be thrown on this very difficult subject by experiments similar to the following. Tie a small piece of zinc to a piece of rusty iron and immerse in a 5°/₀ solution of caustic soda; very soon the rust will fall off and a dull iron surface appear.

Protection by Galvanic Currents.—Views like the foregoing naturally suggest that protection against corrosion would be afforded by means of externally produced galvanic currents, which should be of such an intensity and should be so directed that they would more than balance any local currents. It is, however, only comparatively recently that Mr. Cumberland has perfected the idea. He suspends one or more iron anodes in the boilers, and passes currents of about 2 to 4 ampères (according to the area of the heating surface) through these anodes into the boiler water, thence to the heating surface and back to the dynamo. These currents are of course far cheaper than those produced by zinc slabs and more certain in their action, and should offer the further advantage of polarising the whole inner surface of a boiler, so that after a time comparatively weak currents could be used. The loss of iron from the anode should be about 1 to 4 ozs. per day. Carbon anodes should not be used on any account, because they do not destroy the hydrochloric acid or other corrosive agents which are formed on their surfaces. Suitable safeguards must be provided against passing the current in the wrong direction.

Composition of Boiler Plates and Corrosion.—A subject to which sufficient attention has not yet been paid is the influence of the composition of iron and steel on their behaviour in the presence of corrosive fluids. It is believed that manganese increases corrosion, while there can be no question that nickel reduces it. Carbon behaves very strangely, and it would seem as if in one form—viz. as found in annealed steel—it increased corrosion, whereas in hardened steel it reduces it. However there is little scope for improvement, because its percentage is fixed by other considerations.

A few remarks on some of the results of the experiments carried out by the Admiralty Boiler Committee will not be out of place here.

As in the case of Mr. Parker's experiments, small samples of steel and iron plates were exposed in various merchant and Government steamers, and their losses ascertained. The unit of comparison is 1 grain per square foot per 10 days, equal to about $\frac{1}{800}$ in. per annum.

It was found that when using jet condensers the corrosion was very erratic, ranging from 35·5 to 281·3 grains; but there were only four of these experiments.

Influence of Condenser Tubes.—In four cases copper-tubed surface condensers were used, and the corrosion was slight. There were 25 cases of brass condenser tubes. The corrosion in their boilers was slight, except in two cases, with which respectively colza and Rangoon oil were used as lubricants in the cylinders. With these two exceptions the maximum and minimum losses were 128·8 and 16·8 grains. In only two of these cases was zinc used, and its influence was not marked. The cylinder lubricants were nearly always mineral oils. There were seventeen cases of tinned condenser tubes. Their influence was very marked, the corrosion being far severer than under the other conditions. Mineral oil was used in four of these ships, and the corrosion varied from 204·6 to 362·2 grains. In these cases the use of zinc alone does not seem to do much good, the losses in three cases being 129·6, 215·2, and 503·2 grains. On the other hand, when chalk, Portland cement, and, strange to say, tallow were placed in the boilers whose condenser tubes were tinned, the losses were much reduced, two cases being as low as 32·7 and 39·7 grains. In two boilers soda was used. Its action is not a decided one, for the losses were 30·6 and 376·3 grains.

Consumption of Zinc.—In the reports of the Boiler Committee information will also be found as to the loss of zinc when fitted in boilers. In the case of H.M.S. 'Crocodile' 425 to 630 lbs. of rolled zinc were consumed per boiler per annum, while in H.M.S. 'Serapis' the quantity was 236 to 420 lbs. This is only the actual loss as found by weighing the zinc before and after each voyage. J. Morris, 'N. A.,' 1882, vol. xxiii. p. 151, states that one slab $12 \times 12 \times 1\frac{1}{2}$ inch was completely consumed in 6 weeks. The use of zinc as an anti-corrosive is first mentioned in 1875.

In addition to the general remarks about preventing corrosion (p. 67) the following have suggested themselves. Strong chemicals, either alkalies or acids, should not be used. Certain salts increase or reduce the speed with which iron is attacked. Zinc salts appear to reduce the action, tin salts to increase it. Air, being the probable cause of pitting, should be carefully excluded.

CHAPTER IV

FUELS AND COMBUSTION

Combustion.—When two substances, having a chemical affinity for each other, are brought together under favourable conditions, they combine, forming a new compound. During this combination heat is evolved, which, if the chemical affinity is sufficiently strong, will raise the temperature of the substances to such a height that they grow luminous. This is called combustion. The term is not restricted to the burning of coal, wood, oil, or gas in air, but is applicable to the burning of these or other substances in any other gas, such as chlorine or hydrogen, and is sometimes used to denote a similar process in which only solids or only fluids combine. It may be illustrated by igniting gunpowder, by dropping potassium metal into mercury, or by heating a mixture of iron filings and sulphur. In any of these cases luminous heat is generated.

Slow Combustion is a term which is commonly restricted to express decay of organic matter, but the process is met with on a large scale in every coal mine, where it is the cause of the increased temperature of the ventilating air as it leaves the shaft. In our lungs a process of slow combustion is continually proceeding, and in the tarnishing of metals the same action is manifested. Thus magnesium wire if lighted burns, if exposed to atmospheric influences it tarnishes. In both cases oxide of magnesium has been formed.

When burning coal in a boiler furnace, the main object is to obtain heat; slow and imperfect combustion have, therefore, to be prevented. The one takes place generally, but not always, if the air is in excess and too cold, while the other is caused by an insufficient supply.

Heat, as is well known, is not the same thing as temperature. It is a quantity and not a condition. It is measured in units of heat (called calories), of which each one will raise the temperature of one pound of water one degree Fahrenheit. It is often more convenient to use another measure, viz. the evaporative unit. This is equal to the amount of heat which will evaporate one pound of boiling water from and at 212° F. One evaporative unit equals 966 calories, or heat units, or thermal units (J. K. Cotterill, London, 1878, p. 314); it is also equal to 22·63 horse-power during one minute, while one calorie per second equals 1·4054 horse-power.

Heats of Combustion.—Exhaustive experiments have been made on numerous elements and chemical compounds to determine the amount of heat generated during the processes of combustion, chemical combination, absorption, and solution. Most of these results are contained in M. M. P. Muir's 'The Elements of Thermal Chemistry,' 1885. In this book the kilogram and the degree centigrade are employed, and in order to reduce the values to English measure they have to be multiplied by $\frac{9}{5}$, or divided by 536$\frac{2}{3}$, to convert them respectively into thermal or into evaporative units.

For convenience in calculating the heats of complicated chemical processes the values in that and similar books are not stated per unit of weight of each element or substance, but per atomic weight. Thus the value for pure carbon is given for 12 kils. and not for 1 kil. of carbon, because the atomic weight of this element is 12.

The following table contains a few of the most important determinations. See also W. Ostwald, 1887.

Table of Heats of Combustion of Elements and Compounds

Substances	Formula	Heats of Combustion				
		Per Atom. Weight	Per Pound of Fuel		Per Pound of Oxygen	
		B.T.U.	B.T.U.	Evaporative Units	B.T.U.	Evaporative Units
Diamond . .	$C + O_2$	168,000	14,000	14·50	5,250	5·44
Native graphite .	$C + O_2$	168,500	14,040	14·53	5,266	5·45
Carbon, amorph. .	$C + O_2$	174,500	14,540	15·07	5,454	5·65
Do., imperfectly burnt . .	$C + O$	52,100	4,340	4·50	3,257	3·37
Silicon, amorph. .	$Si + O_2$	395,000	14,100	14·60	8,229	8·51
Sulphur (melted) .	$S + O_2$	130,200	4,069	4·20	4,069	4·22
Phosphorus (yellow)	$P_2 + O_5$	666,000	21,500	22·30	8,326	8·61
Hydrogen to water	$H_2 + O$	123,000	61,156	63·70	7,688	7·96
„ to steam	$H_2 + O$	105,720	51,322	54·70	6,608	3·84
Iron . . .	$Fe_3 + O_4$	477,000	2,751	2·84	7,464	7·70
Carbonic acid .	$CO + O$	122,400	4,375	4·50	7,651	7·93
„ per 1 carbon	$CO + O$	122,400	10,200	10·57	...	...
Marsh gas . .	$CH_4 + O_4$	376,200	23,512	24·30	5,878	6·09

Partial Combustion.—The heat generated by the combustion of 28 lbs. of carbonic oxide ($CO = 12\,C + 16\,O$) amounts to 126·9 English evaporative units. If this heat be added to the 53·9, which are generated by imperfectly burning 12 lbs. of carbon, 180·8 units are obtained, which are exactly equal to the heat evolved by burning these 12 lbs. of carbon to carbonic acid in one operation. This shows that, as regards the final result, it does not matter whether carbon is burnt in one or in two stages, or, in other words, whether it is completely burnt at once, or is first converted into carbonic oxide gas and then burnt. Another feature of interest is that the partial burning of 12 carbon, to form 28 carbonic oxide, only produces 42·5% of the heat which will be evolved when 28 carbonic oxide are burnt to carbonic acid. The remaining 57·5% might be looked upon as the latent heat of evaporation of 12 lbs. of carbon. This would imply that it requires four and a half times as much heat to evaporate carbon as to evaporate water. But this simple relation is not supported by other experiments, and from a study of the heats of combustion of various chemical compounds it may be inferred that the calorific value of any fuel cannot be determined accurately by calculations based on chemical analysis alone.

Heating Power of Fuels.—There are two methods of determining the heating power of fuels: either by measuring the amount of heat generated during their combustion, or by chemically analysing them, and summing up the known amounts of heat which each element would generate if burnt separately. The serious difficulties which at one time beset the first of these methods have been successfully overcome by the use of the so-called

Bomb Calorimeter. (See Berthelot and Vieille, 'An. Ch. et Ph.,' 1887, vi. vol. x. p. 433; I. Mahler, 'I. and S. I.,' 1892, p. 183.)—The fuel is powdered and placed in a small platinum crucible inside a strong iron bomb. Moist oxygen, under a pressure of 350 lbs., is then admitted, and the whole is placed in a calorimeter, and allowed to cool till the water temperature has grown steady. The fuel is ignited by electricity, and is consumed rapidly and completely. The rise of temperature of the calorimeter, amounting to about 5° F., is then the measure of the heat evolved.

Although the combustion is a perfect one, a necessary correction has to be made for the condensation of the steam produced, for it must be remembered that at the temperature of the waste gases of a boiler all moisture contained in them exists as steam. Assume that the fuel contains 80°/ₒ carbon, 4°/ₒ moisture, and 5°/ₒ hydrogen, the latter combining with an eightfold weight of oxygen and producing 45°/ₒ additional moisture or steam, we would obtain in the bomb calorimeter from the carbon $0{\cdot}80 \times 14{,}540 = 11{,}632$ B.T.U.

and from the hydrogen $0{\cdot}05 \times 61{,}156 = 1{,}223$ B.T.U.

Total = 12,855 B.T.U.

But in a furnace 45 + 5 = 49°/ₒ of moisture abstract $0{\cdot}49 \times 1{,}016$ B.T.U. at 60° F. = 498 B.T.U.

leaving a net available heat at 60° F. of 12,357 B.T.U.

or nearly 4°/ₒ less than the uncorrected bomb determination which, it is evident, should always be accompanied by a chemical analysis.

All uncertainty about radiation, and the trouble due to the friction of the mercury in the thermometer, have been overcome by the arrangement adopted by C. J. Wilson, who during each experiment adjusts the temperature of the surrounding envelope so as to agree with the changes of the calorimeter water.

Formerly the only reliable calorific measurements had to be carried out in the elaborate instrument designed by MM. Faber and Silbermann, in which the fuel was consumed in an atmosphere of oxygen under normal pressure. The escaping gases were led through a long cooling tube, and their temperature measured. They were then collected, and analysis carried out, and accurate corrections were made for incomplete combustion.

W. Thompson Calorimeter.—Fig. 102 shows a simple, and, for comparative purposes, a reliable instrument. The broken (not powdered) fuel is weighed and placed in a small platinum crucible, F, which is supported in a fork by the lower lid of an open-ended bottle, C. This lid is pierced by a few tubes, D, having very small holes at their ends. A tube, O, of fireproof material reaches through the upper cork down to the fuel. Its one branch is connected by means of an india-rubber tube to an oxygen flask, and its vertical branch is closed by a nipple or stopper. T is a very sensitive thermometer.

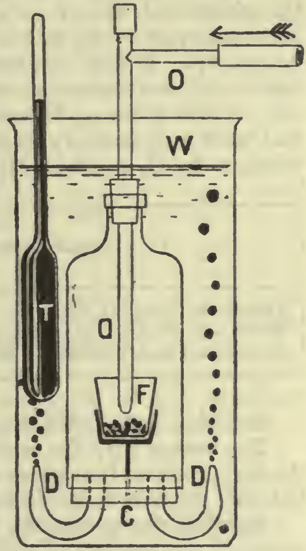

FIG. 102

When ready the bottle C is lowered into the calorimeter W, containing a weighed quantity of water, and is allowed to remain there till the temperature is steady. Care has to be taken not to allow any escape of air, otherwise water will enter through the tubes D and flood the fuel. When ready, a small piece of glowing tinder is dropped down the tube O, which is at once closed again, and the blast of oxygen turned on. The products of combustion escape through the tubes D, parting with most of the heat which they contain to the surrounding water. After five or ten minutes the combustion is completed; the water is allowed to enter the bottle, cooling the interior, and is then expelled, and the generated heat can now be estimated by measuring the total rise of temperature of the water, which amounts to from 3° to 6° F. To obtain accurate results various precautions must be taken, of which some of the more important are: reduction of losses by radiation; the use of moist oxygen of the same temperature as the calorimeter, so that none of the calorimeter water is evaporated. This water may be slightly acidulated, so that it will absorb no carbonic acid. The correction for complete absorption in water would be a subtraction of ·25 evaporative unit for each pound of gas absorbed, or ·92 for each pound of carbon burnt. If caustic soda were added to the water other allowances would have to be made. The amount of hydrogen and moisture in the coal, which is all converted into water, not steam, must be allowed for.

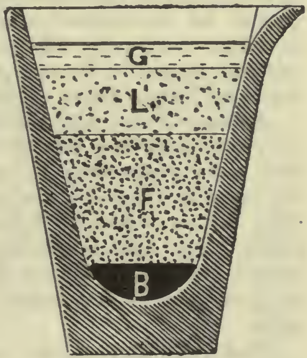

FIG. 103

Berthier's Method is based on the fallacious idea that the amount of heat generated by burning a fuel is proportional to the oxygen consumed, but being a fairly simple one, it may occasionally be of use. It is carried out as follows:—Accurately weigh, say, 25 grains of fuel, and mix them intimately with 2 oz. of pure litharge, and pour them into a crucible (fig. 103), B and F. They are then covered with 1½ oz. of pure litharge, L, and over all is placed a layer of ¼ in. of broken glass, G. The crucible is then heated to redness, care being taken that the top grows hot first, so as to melt the glass, and that it is well protected from smoke. After about three-quarters of an hour's heating, it is allowed to cool, and is then broken up. It will be found to contain a button of lead, B. This is hammered and washed to remove all impurities. It will weigh from 700 to 800 grains. Whatever the result, divide it by the weight of fuel used, and multiply by the constant C_1, so as to obtain its calorific value, or by C_2 to obtain its evaporative value.

Nature of Fuel	Constants	
	C_1	C_2
Anthracite, charcoal, &c.	543	·563
Welsh and similar coal, yielding, say, 90 % coke	518	·537
Bituminous coal, yielding 65–85 % coke	500	·517

See 'An. Ind.,' 1883, vol. i. p. 38.

The reason why the coefficients vary is because the hydrogen in the fuel produces four times as much heat as an equal weight of carbon, but only reduces three times as much lead.

Another way of roughly estimating the heating value of a fuel is to ascertain the amount of moisture, volatile compounds, coke and ashes, which it contains, and to credit every pound of the gases as being able to evaporate twenty times their weight of water, while the coke less ashes gives 15 evaporative units.

Coal Analysis.—The calculation of the calorific value from the chemical analysis is necessarily somewhat incorrect, for although the heats of combustion of the various elements have been accurately determined, practically nothing is known about the amount of heat required to split up the chemical compounds which constitute coal. It must also be remembered that the amount of oxygen in the coal cannot be determined analytically, but only by difference, and is therefore seriously affected by the chemical composition of the mineral matter in the coal, which is entirely altered, and its weight increased, while being converted into ashes.

Ashes.—These also affect the calorific value of the fuels in which they are contained, chiefly of course on account of the heat which they carry away when being removed, but partly also on account of the sulphur they contain, for in the fuel this element is combined with iron as pyrites (FeS_2) and in the ashes it is contained as sulphide (FeS). The heat necessary for this reduction has not been determined, but as the amount of combustible sulphur is a small quantity, no great error is introduced by not making a deduction from the heat value.

Coal Sampling.—Little attention has been paid to the question of how much coal should be put aside during a trial for purposes of analysis. It is usually assumed that this quantity should bear some proportion to the total quantity of coal burnt, and that if a sufficiently large sample be taken the analysis will be correct within a certain limit. That is not so, for the size of the sample depends entirely on the sizes of the largest lumps in the heap, and far from affording certainty, all that can be said of any sampling is that it merely determines the probability that a certain error as regards composition will not be exceeded. Thus if the lumps of coal in a heap are all of one size, and if the sample consists of 16,600 pieces, then the chances are that only one sample out of 100 is likely to be wrong as regards chemical composition to a greater extent than $\frac{1}{10}$°/$_0$; 166 pieces, or one hundredth of the above, would ensure that an error of 1°/$_0$ will not occur more than once in 100 samples. This is on the assumption that the irregularities in the compositions of the various layers of the seam from which the coal was taken differ chemically from each other to the extent of 20°/$_0$. In good coal seams the difference is less than 2°/$_0$, and the accuracy ten times greater. The chance of error is inversely proportional to the square root of the number of pieces. For mixed sizes of coal the total weight of the sample may be one-half of the product of the weight of the largest pieces by the numbers mentioned above.

The following analyses and calorific values, as determined by the bomb calorimeter, of the fuel used on the trials of the Research Committee and a few others have been kindly supplied by Mr. C. J. Wilson:—

Vessel's Name and Fuel	Composition of Fuel in %					Calorific Value	Evaporative Value
	C	H	O+S+N	Ash	Moist.		
'Colchester' { ⅔ Yorkshire, ⅓ Nottinghamshire }	71·89	5·48	14·36	4·08	4·25	13,176	13·63
'Tartar' { Penrikyber Welsh coal }	87·98	4·22	3·31	3·42	1·07	14,765	15·28
A sample of Welsh coal	81·76	3·81	2·46	9·32	2·65	13,802	14·30
A sample of coal	86·02	4·22	4·48	3·94	1·34	14,710	15·25
A sample of coke	76·94	0	0	14·23	8·83	11,032	11·40
A sample of coke	71·17	0	0	19·46	9·37	10,200	10·58

The following table contains summaries of some analyses and calorific determinations (W. Thompson, 'Soc. Ch. Ind.,' 1889, vol. viii. p. 525):—

Name of Fuel	Composition of Fuel in %							Experimental	
	C	H	O	S	N	Moist.	Ash	Calorific Value	Evaporative Value
Nixon Navigation Glam.	88·03	4·11	1·98	·66	·96	1·02	3·22	15,010	15·55
Tyldesley, Theperley & Co.	68·13	4·78	4·86	1·39	1·22	4·72	14·90	11,610	12·20
Tyldesley Coal Co.	74·46	5·10	8·25	·49	1·53	6·07	4·09	12,730	13·18
Upper Drumgray seam { Bottom	75·48	4·98	7·86	·75	1·59	2·78	6·55	13,300	13·77
Upper Drumgray seam { Mid rib	72·13	4·67	6·56	·54	1·25	2·26	12·58	12,520	12·96
Upper Drumgray seam { Top seam	75·05	5·12	9·39	·86	1·76	3·53	4·29	13,560	14·05
Bickshaw Main	78·93	4·90	7·24	1·04	1·56	4·36	1·96	13,430	13·91
Pemberton 5-ft	72·41	5·16	8·84	·93	1·41	6·70	4·55	13,040	13·50
Cramhawke	69·77	4·82	12·44	1·17	1·33	7·15	3·31	13,420	13·19
Wigan 4-ft.	76·49	4·96	8·46	1·07	1·44	4·84	2·75	13,590	14·07
Bickershaw 7-ft.	73·91	4·86	11·32	·68	1·67	6·60	·96	13,350	13·82
Pendleton 4-ft.	79·76	4·89	7·51	·59	1·43	3·90	1·91	13,920	14·42

These calorific values were obtained in a Thompson calorimeter and corrected, while the following, made by Scheurer-Kestner ('An. Ch.

Ph.,' 1886, vi. vol. viii. p. 271), were made in the more elaborate calorimeter of Faber and Silbermann.

Name of Fuel	Composition of Pure Fuel in %					Pure Coal	
	C	H	O+S+N	Ash to be added	Coke produced	Calorific Value	Evaporative Value
Bwlf . . .	91·08	3·83	5·09	3·32	82·1	15,800	16·4
Powels . . .	92·49	4·04	3·47	3·72	87·2	16,060	16·6
Altendorf . .	89·92	4·11	6·0	10 to 12	83·9	16,400	17·0
Ronchamp . .	89·09	5·09	5·82	—	78·8	16,420	17·0

Nixon's Coal.—Scheurer-Kestner ('Soc. I. Mul.,' 1888, vol. lviii. p. 313):—

C	H	O	S	N	Ash	Calorific Value	Evaporative Value
85·73	4·17	3·97	·66	·46	5·01	15,950	16·5

In 'Soc. I. Mul.,' 1891, p. 517, Scheurer-Kestner states that all his previous results are too high.

Prof. H. Fritz ('Dingler's J.,' 1876, vol. ccxix. p. 185) gives the following list:—

Name of Fuel	Calorific Value	Evaporative Value	Name of Fuel	Calorific Value	Evaporative Value
Hydrogen	62,100	64·3	Coke (15 % cinder) .	10,430	10·8
Coal gas	39,600	41·0	Alcohol (pure) . .	12,520	12·9
Petroleum	18,200	18·8	Peat coal	10,430	10·8
Olive oil	17,630	18·2	Lignite	9,000	9·3
Wax	15,660	16·2	Peat (dry) . . .	8,640	8·9
Tallow	14,930	15·4	Peat (20 % moisture) .	6,480	6·7
Anthracite	14,590	15·1	Red coal	7,160	7·4
Carbon	14,540	15·0	Dry wood	6,480	6·7
Coal (mean) . . .	13,500	14·0	Wood (20 % moisture).	5,040	5·2
Charcoal	12,600	13·0	Straw	3,360	3·5
Coal (clean) . . .	12,600	13·0	Tan	5,580	5·7

The following table is a summary of C. Grove and W. Thorpe's (1889, p. 54) analyses of 111 samples of fuel; it contains only the highest and lowest values, and clearly shows how much their compositions vary.

Coal Field	No. of Samples	—	C	H	O	S	N	Ash	Density	Coke
Welsh	37	Max.	91·4	5·8	17·9	5·1	2·2	14·7	1·38	92·2
		Min.	71·1	3·5	·4	·1	0	1·2	1·25	55·4
Newcastle	18	Max.	86·8	6·7	10·3	2·8	1·8	9·1	1·31	72·3
		Min.	78·0	4·5	2·4	·1	·7	·2	1·25	35·6
Derbyshire	7	Max.	81·9	5·6	12·4	1·3	2·1	4·6	1·32	62·5
		Min.	78·0	4·8	8·6	·7	·8	1·2	1·27	52·8
Lancashire	28	Max.	83·9	6·2	20·0	3·0	2·2	14·3	1·35	66·1
		Min.	72·9	4·8	4·9	·5	·5	1·1	1·23	51·1
Scotch	7	Max.	88·5	6·5	15·5	1·6	2·0	10·7	1·32	59·1
		Min.	74·5	4·8	·9	·4	0	1·2	1·23	49·3
Van Diemen's Land . .	8	Max.	70·4	4·2	9·1	1·3	1·6	30·4	...	...
		Min.	57·2	2·9	1·0	·7	·9	14·4	...	...
Chili	6	Max.	78·3	6·4	17·3	6·1	1·1	36·9	...	...
		Min.	39·0	4·0	8·4	·9	·5	5·7	...	...

Coal Trials.—Parliamentary Papers

Date	No.	Index		Remarks
		Vol.	No.	
1857–8	375	39	43	
1859	116	25	209	Welsh and Newcastle
1860	485	42	267	
1862	204	34	113	
,,	337	,,	11	
,,	364	,,	121	Australian
1863	79	35	151	
,,	159	,,	141	,,
1864	(80, 80–I.)	37	187, 211	Welsh and Newcastle
,,	375	,,	213	,, ,,
1865	365	35 II.	79	
1866	440	46	35	
1867	561	44	339	
,,	563	,,	491	Lancashire and Cheshire
1868–9	212	38	443	
,,	270	,,	447	,, ,,
1870–71	121	40	463	Newcastle
1872	165	39	349	
1873	—	42	537	Indian troopships
1876	280	47	713	
1877	397	52	515	
1878	396	49	541	
,,	233	47	495	
1878–9	380	45	493	

Spontaneous Combustion of Coals.—Parliamentary Papers

Date	No.	Index	
		Vol.	No.
1876	C. 1586–1	41	1
1878	366	47	33
1878–9	373	64	47
1880	C. 1586–1	41	1
1886	366	59	45

H. Poole gives very exhaustive information about coals of the world, his tables covering about forty-two pages, of which, however, English coal, coke, &c. cover about two pages.

The object of determining the heating power of a fuel is undoubtedly to obtain a correct measure of its commercial value; but heating power is not everything, and there are other qualities, such as liability to smoke, to flame, to cake, to ignite easily and even spontaneously, to break up into dust when exposed, and various others.

Burning Qualities.—Amongst these one of great importance is as to whether a coal will ignite easily or not. Thus Cannel coal can be lit by applying a match, while anthracite can only be kept alight under carefully regulated conditions of draught, &c. The success of explosives depends on a good knowledge of this subject; thus we find that sulphur, which ignites at a low temperature, has to be added to gunpowder in larger or smaller quantities, according as to whether it has to explode quickly or slowly. More recently it has been discovered that the temperature at which wood is converted into charcoal materially affects the temperature at which it ignites, and that where the process is carried out gently and over a long period, as is the case with wooden beams in houses placed close to flues, ignition ultimately takes place spontaneously. Gun cotton ignites so readily that it could not be used for ammunition until it was discovered that an admixture of camphor or nitro-glycerine raised this temperature, and also retards the burning process; chloride of tin still further reduces its liability to ignite. Some curious phenomena have been noted. Thus phosphorus, which ignites at a temperature of 140° F. in an ordinary atmosphere, will not burn if placed in absolutely pure oxygen. A mixture of hydrogen and chlorine gas ignites at a very low temperature if illuminated by actinic rays, but no amount of heating, short of redness, will effect their chemical combination in the dark.

Table of Temperatures of Ignition

Substance	Igniting Temperatures ° F.
Phosphoretted hydrogen	116
Phosphorus	140
Lignite dust	300
Sulphur	470
Carbon disulphide	300 or 440
Dried peat	435
Anthracite dust	570
Coal	600
Charcoal made at 550° to 700° F.	680
„ „ 2,200° to 2,300° F.	1,100 or 1,400
Cokes	Red Heat
Anthracite	„ 750
Carbonic oxide	„ 1,211
Hydrogen	1,030 or 1,290
Fire damp	1,200

With hydrocarbons the igniting temperature depends on the chemical composition; thus benzine ignites at about 500° F., while some of the dense oils will quench red-hot iron without igniting. Generally speaking hydrocarbons prevent both quick and spontaneous combustion. More precise information on igniting temperatures would greatly assist the understanding of fuel combustion. Much may be learnt about the behaviour of various coals by consulting the following papers:—'Memoirs of the Geological Survey of Great Britain' (London, 1848, vol. ii. p. 539); Report of Sir H. De la Bêche and Dr. Lyon Playfair.

Influence of Air Pressure on Combustion.—Frankland's experiments (1877, p. 863) show that, contrary to the accepted views, increased pressure (not draught) does not accelerate combustion, but that it increases the luminosity and smokiness of flames. At twenty atmospheres air-pressure the hydrogen flame gives a perfectly continuous spectrum, while the carbonic oxide flame is equally luminous at fourteen atmospheres. An ordinary candle gives hardly any light when burnt on the summit of Mont Blanc, but smokes under pressure.

The Specific Heat of a Gas can only be accurately estimated if the specific heats of individual substances are known. The following small list contains all the necessary data :—

Table of Specific Heats at Constant Pressure

ELEMENTS

Carbon (charcoal)	·1935	Oxygen	·2175
Hydrogen	3·4090	Nitrogen	·2438

COMPOUNDS

Air	·2375	Water	1·0000
Carbonic oxide (CO)	·2450	Marsh gas (CH_4)	·5929
„ acid (CO_2)	·2163	Sulphurous acid (SO_2)	·1540
Steam (H_2O)	·4805	See also steam tables	

There is some uncertainty as to the specific heat of steam, but as it does not enter very largely into the composition of burnt products (see p. 52), one per cent. of moisture adds about 0·003 to the specific heat of air.

Air being composed of 20·8 volumes of oxygen and 79·2 volumes of nitrogen, or by weight 23·07 oxygen and 76·93 nitrogen, its specific heat is

$$\frac{23{\cdot}07 \times {\cdot}2157 + 76{\cdot}93 \times {\cdot}2434}{100} = {\cdot}2375, \text{ as stated in the table.}$$

The quantity of oxygen required to consume 1 lb. of carbon is $\frac{32}{12} = 2{\cdot}667$, and this carries with it 8·92 lbs. of nitrogen.

The calculation is therefore as follows :—

$$\begin{array}{rll}
1{\cdot}000 & \text{carbon} & \\
2{\cdot}667 & \text{oxygen} & \\
\hline
3{\cdot}667 & \text{carbonic acid} & \times {\cdot}2163 = {\cdot}793 \\
8{\cdot}920 & \text{nitrogen} & \times {\cdot}2438 = 2{\cdot}171 \\
\hline
12{\cdot}587 & \text{products} & \times {\cdot}2355 = 2{\cdot}964 \\
\hline
\end{array}$$

So that the specific heat of this mixed gas is ·2355. But the amount of air supplied, viz. 11·6 times as much as the carbon, is only just sufficient for perfect combustion; in practice it is above 15 and sometimes even over 30. By adding 7·413 lbs. of air to the above the products of combustion are increased to 20 lbs., and the specific heat would be raised to ·2360.

Coal contains both hydrogen and some moisture, and, as the specific heat of steam is about twice as great as that of dry air, the average value is somewhat higher than the above. However, for practical purposes the value ·237 is sufficiently accurate.

Temperatures of Flames.—When no heat is lost by radiation or convection the entire quantity generated during combustion must have been used to warm the products, raising their temperature sufficiently to make them glow, and the result is a flame. Its temperature is calculated as follows:—If n pounds of air are supplied for every pound of carbon burnt, then the weight of the products will be $n+1$, the heat evolved will be 14,540 cals. per pound of fuel, and if σ is the specific heat of the gases, the temperature of the flame $T = \frac{14540}{\sigma . (n + I)}$.

It is now possible to determine the temperature T of the flame with the help of the previous formula. Let $n + 1 = 20$, then T = 3,080° F. above t, the mean temperature of the air and fuel. Roughly speaking, the furnace temperature multiplied into the ratio of products to coal ($= n + \cdot 9$) is 61,000° F.; but this is true only down to a limit of about $n = 15$. Below this point the combustion of the coal is not a perfect one, and instead of 14,540 only 4,340 calories are evolved.

Therefore, unless much hydrocarbon gas or smoke is produced we have the following three conditions:—

When $n = 5{\cdot}8$, carbonic oxide and nitrogen are the only products; the initial temperature is 2,350° F. + t.

When n is greater than 5·8 and less than 15, the products consist of carbonic oxide, carbonic acid, nitrogen, and oxygen, and the temperature grows higher till $n = 15$ is reached; then no carbonic oxide is evolved. The initial temperature is 4,130° F. + t.

Above this point the temperature grows less and less, as stated above, viz. $\frac{61,000}{n + \cdot 9}$.

When 1 lb. of hydrogen is being burnt, 8 times its weight of oxygen is required, which is accompanied by 26·8 lbs. of nitrogen, making a total of 35·8 lbs. of products of combustion. These consist of 9 lbs. of steam and 26·8 of nitrogen. The specific heat of this mixture is, say, $\frac{9 \times \cdot 4805 \times 26{\cdot}8 \times \cdot 2434}{35{\cdot}8} = \cdot 303$ (or ·278). The heat evolved being 52,860 cals., the temperature of this flame is 4,873° F. + t.

For such cases in which $n + 1$ exceeds, say, 50, divide 247,000 by $(7{\cdot}7 + n)$ and the quotient is the temperature.

The flame temperature of a compound containing carbon and hydrogen would of course depend on some as yet undiscovered law, but a fairly good guess can be made by taking a mean of the values just found. Roughly

$$T = \frac{61,000\ C + 247,000\ H}{(C + H)n + C{\cdot}9 + H7{\cdot}7}.$$

Of course the actual temperatures of flames are decidedly lower than those found by this formula, because loss of heat takes place by radiation during the process of combustion.[1]

[1] Moisture in the atmosphere seriously affects boiler performances.

The following are a few particulars of temperatures of flames —

('*Franklin Inst.*,' 1878, *vol. lxxvi. p.* 205.)

Bunsen burner using 1 vol. gas, 2·4 air gave	2,480° F.
" " " 1 vol. gas, 2·4 air, and 1 nitrogen gave	2,150 "
Do. do. and 2 nitrogen gave	1,890 "
Bunsen burner, 1 vol. gas, 2·4 air and CO_2, gave	2,010 "
" " " " $2CO_2$ "	1,434 "
Locatelli lamp	1,690 "
Stearine candle	1,723 "
Petroleum lamp with chimney . . .	1,885 "
" " without "	1,690 "
" " sooty envelope . . .	1,434 "
Alcohol (·912) lamp	2,140 "
" (·822) "	2,150 "

H. Le Chatelier, '*Comp. Rend.*,' 1892, *vol. cxiv. p.* 470.

Bessemer process of steel-making . . .	2,984° F.
Siemens " "	2,876 "
Glass-melting	2,390 "
Electric light	3,272–3,812 "

The Temperature of Dissociation.—When this temperature is reached chemical compounds are split up. For sympathetic ink and cupreous hydrate it is low, for gypsum it is higher, also for carbonate of lime, sulphate of iron, &c. For steam and for carbonic acid it is very high and has not yet been determined. From this it would follow that the higher the initial temperatures of the fuels and air the less heat is evolved, until a point is reached where no combustion takes place. This is far higher than any temperature to be met with even in a steel melting furnace, but its existence must not be lost sight of where heated air or gas is used, for the expected benefit will not be fully realised, nor will the flame temperatures be as high as expected.

Furnace Temperatures.—Hardly any experiments have yet been made to determine furnace temperatures, but since the introduction of MM. Mesuré and Noël's optical pyrometer ('I. and S. I.,' 1889, p. 251) at least one of the difficulties has been removed. The first of these instruments is a thermopile, the heated junction of the circuit consisting of wires made of two different alloys of platinum. The optical pyrometer consists of a tube in which a quartz plate is fixed near the centre, and a Nichol's prism at either end. One of these can be turned round its own axis. On looking at a heated object, and slowly turning the prism, it will be noticed that the colour changes from green to red and *vice versâ*. The angle at which this change takes place is noted, and is the measure of the temperature. Care has to be taken not to let any daylight be reflected from the hot object, otherwise the readings will be those of the temperature of the sun. Both these instruments have to be graduated according to some reliable standard.

Air Thermometer.—If properly manipulated there is probably no more accurate measure of temperature than an air thermometer. All the permanent gases expand at the rate of ·3665 between the freezing and the boiling point of water. This is at the rate of ·0020611 per °F. It has been assumed that the **absolute zero** is reached when the volume of air is reduced to nothing. At the above rate this would be found to be 491° below the freezing point of water, or $-459°$ F. Therefore if t is the Fahrenheit temperature of an object, its absolute temperature would be $T = t + 459°$. The instrument shown in fig. 104 is based on the above facts. It consists of a small pear-shaped platinum bulb, having a small pin-hole at its apex. It is carefully filled with nitrogen gas, enclosed in fire clay, and placed in the furnace whose temperature is to be measured. When sufficiently heated the whole instrument is plunged into cold or boiling water, and is kept there point downwards till it has acquired the temperature of its surroundings. The protecting scale of fire clay has of course fallen off. The bulk is now carefully weighed, first in its present condition, viz. partly filled with water, then when quite full, and then when quite empty. The ratio of the total internal volume of the bulb, which, while in the furnace, was full of hot air, to the volume of air remaining after cooling in the water bath, is exactly the ratio of the absolute furnace temperature to that of the bath. Any other pyrometer can be graduated by placing it alongside the above.

FIG. 104

Pyrometers.—Descriptions of various pyrometers will be found in the following publications. (See also p. 109.)

J. Wilson, 'M. E.,' 1852, p. 53, mentions one which consists of a piece of platinum, to be thrown into a calorimeter while hot.

M. Launy, 'Comp. Rend.,' 1869, vol. lxix. p. 347. The pressure exerted by the liberated carbonic acid gas from heated lime is measured.

T. Carnelly and T. Burton, 'I. and S. I.,' 1884, p. 195. Various pyrometers.

Prof. J. Wiborg, 'I. and S. I.,' 1888, ii. p. 110. An exposed bulb is filled with compressed air.

M. Santignon, 'Génie C.,' 1890, vol. xvi. p. 328. This pyrometer consists of a bent metal tube which is inserted in the heated space, and a steady current of water is made to flow through it.

Messrs. Heish and Folkard, 'Iron Age,' vol. xxxvii. p. 25. Platinum bulb air pyrometer.

Roberts Austin, 'C.E.,' 1892, vol. cx. p. 152, describes Callendar's electric resistance pyrometer sensitive to $\frac{1}{5}°$ C. up to a red heat. Le Chatelier's thermo-couple is sensitive to 2° F. up to 1,800° F. Le Chatelier's optical pyrometer, Mesuré and Nöel's optical pyrometer.

Landholt and Börnstein (1912) give a list of melting temperatures of various alloys and metals and convenient boiling temperatures (p. 324).

Temperatures in the Fuel.—It has already been explained that the temperature of a flame is proportional to the percentage of carbonic acid it contains till a point is reached where carbonic oxide is formed.

Fig. 105 illustrates, in a rough way, the conditions which may be expected in the interior of a thick fire.[1]

Estimated Ratio of Air to consumed Carbon	Thickness of Fire	Estimated Temperatures in the Interior of a Fire
20 : 1. Air admitted above fuel		2,960° F.
6·8 : 1. Chiefly CO . .		2,410° F.
10 : 1. Partly CO and CO_2		2,980° F.
15 : 1. { Complete combustion . .		3,870° F.
20 : 1. Do.. A. —		— A. 2,960° F.
43 : 1. Do . . .		1,550° F.
1 : 0.	Fig. 105	60° F.

Of course this picture is mere guesswork, but it shows that by measuring the temperatures at various depths, or by analysing the gases at these points, some valuable information might still be obtained as to the process of combustion (compare p. 298).

Prof. A. Ledebeur ('Stahl und Eisen,' 1882, vol. ii. p. 356) made the following interesting experiments, in which the temperatures of the burning charcoal (contained in a tube) were carefully regulated. To a certain extent these experiments modify the above calculations :—

Temperature	Percentage of Carbon burnt		Oxygen used
Degrees Fahrenheit	CO_2	CO	%
— 660	78·6	21·4	33·0
— 710	72·4	27·6	80·6
Dark red . . 1,150	71·4	28·6	87·9
Cherry red . 1,250	62·6	37·4	80·3
Yellow . . 2,010	1·3	98·7	100·0

With thin fires it is quite possible that sufficient air for complete combustion passes through the fuel. In this case the conditions above the line A do not exist, and it is not necessary to admit air above the bars ; but with thick fires, particularly with a weak draught, the upper layers of fuel effect the reduction of carbonic acid to carbonic oxide, and air must be admitted above. It will also readily suggest itself that if green coal is thrown on a fire which is still evolving much carbonic oxide, the latter will be cooled to such an extent that it cannot ignite when coming in contact with the upper air. Fires should, therefore, be allowed to burn as low as possible before recoaling.

[1] In blast furnaces no free oxygen and hardly any carbonic acid are found at a greater distance than two feet from each tuyer (p. 298).

It might seem to be an advantage, even in cases of forced draught, to keep the fires thin, and thereby prevent the formation of combustible gases. This, however, is not the case, for, unless the grate surface were very much increased, there would be a serious loss, due to the blowing away of unburnt coal particles. A direct test for CO is described in 'Comp. Rend.,' 1897, vol. cxxiv. p. 621, and 'C. E.,' vol. cxxix. p. 474.

Effects of Draught.—In cases of complete combustion the weight of the products is sometimes thirty times the weight of fuel burnt. Now, suppose that the consumption is raised from 20 lbs. per square foot per hour to 40 lbs.; then the velocity of the escaping gases, which amounts to 600 lbs. per hour = 10 cubic ft. per second, would be twice as great, and the power to carry away solid particles would have been increased about fourfold. If, however, the fires are kept thick and all the gases leave the fuel as carbonic oxide, their weight is reduced to 270 lbs., and the temperature being slightly lower, and the bulk proportionately less, the power to lift up and carry away small particles of coal is reduced to about one-fifth. Of course even the thickest fires in a marine boiler will allow free oxygen and much carbonic acid to pass, so that the above favourable condition is hardly approached; but at any rate it is clear that with forced draught fires should be kept thick, restricting the admission of air under the bars, keeping the temperatures high, and giving much fuel contact to the carbonic acid gases, so as to convert them into combustible ones. A very large proportion of air should then be admitted at the doors or bridges. Naturally the horizontal draught will be excessive unless the furnaces are very large.

The air which is admitted for the combustion of the gases should be as hot as possible, and when admitted at the bridges the utmost has probably been done in this direction by having passed it under the fire bars. When admitted at the doors artificial heating is the only means of warming it, but, as all draught over the fire bed has the effect of blowing away particles of coal and cinder, this trouble would only be aggravated if the air temperature and bulk were increased. An additional power to do harm is given to the draught if much air is also driven through the bed of coal, whereby all the smaller particles are carried to the top. This can only be prevented by a thick bed of coal, and this again is only possible with large furnaces.

Forced and Natural Draught.—A few formulæ are necessary for explaining this subject, but they are of the simplest.

$$v^2 = 2.g.h.$$

Here v is the velocity of air, h column of air measured in feet, corresponding to the draught pressure, and $g = 32{\cdot}2$ feet is the acceleration due to gravity. Instead of an air column or mercury column, it is usual to express draught pressure in inches of water. The respective weights of mercury, water, and air being 13·60, 1, and ·001293, the formula is changed into

$$v^2 = 4{,}140\,\mathrm{H} = 56{,}300\,\mathrm{M},$$

where H = pressure measured in inches of water, and M = pressure measured in inches of mercury. In practice the friction and bends will seriously reduce the velocity, and the co-efficients have to be reduced by about 25 %. It is also evident that with the same draught an incandescent, and therefore lighter, gas is more easily moved than a cold one. If T denotes its absolute temperature, while 491° is the absolute temperature at 32° F., then $v^2 = 8{\cdot}5 \,.\, \mathrm{H} \,.\, \mathrm{T}$ theoretically, or say $6 \,.\, \mathrm{H} \,.\, \mathrm{T}$ practically.

It is as well to mention that a more general formula, but without the above correction, would be

$$v^2 = 121{\cdot}5 \frac{\mathrm{H} \,.\, \mathrm{T}}{m}.$$

Here m is the molecular weight of a gas. This is 14·37 for air, about 16 for oxygen, 14 for nitrogen, 22 for carbonic acid, and 9 for steam.

In order to find V, the volume of air discharged per second, the sectional area A, in square feet, of the orifice or channel has to be multiplied by the velocity v.

$$\mathrm{V} = v \,.\, \mathrm{A} = \mathrm{A} \,.\, \sqrt{6 \,.\, \mathrm{H} \,.\, \mathrm{T}}.$$

The weight Q is found by multiplying this volume into the weight of a cubic foot of air, and introducing the correction for temperature.

$$\mathrm{Q} = \mathrm{A} \,.\, 97{\cdot}3 \sqrt{\frac{\mathrm{H}}{\mathrm{T}}}, \text{ or roughly } 100 \,.\, \mathrm{A} \sqrt{\frac{\mathrm{H}}{\mathrm{T}}},$$

from which it follows that the pressure H required for delivering Q lbs. of hot air per second through the section A is

$$\mathrm{H} = \frac{\mathrm{Q}^2}{\mathrm{A}^2} \frac{\mathrm{T}}{10{,}000}, \text{ measured in inches of water.}$$

Resistances in Furnaces.—This formula has been used in estimating some of the values in the following table, in which an attempt is made to illustrate what takes place in two furnaces, whose diameters are respectively 30 ins. and 45 ins., with 6-ft. grates. The sectional areas of the ashpits and over the fuel would be 1·6 and 3·7 sq. ft. 20 lbs. per square foot is to be the coal consumption under natural and 40 lbs. under forced draught. The products of combustion will weigh about 24 times as much as the fuel, and it is assumed that two-thirds of the air passes through the fuel, and the other third through the door. If the third is admitted at a temperature of 32° F., it will be reasonable to assume that the temperatures in the fuel and in the flame are about 2,500° F., and that as they pass into the tubes this has been reduced to 1,000° F., while on entering the funnel it has fallen to 600° F.

Doubtful though some of the following results are, they will at any rate serve to make comparisons, and then there can be no question that the large furnace has the advantage.

Estimated Temperatures, Volumes, Velocities, and Resistances of Gases in a Pair of Boiler Furnaces

Particulars		Draught			
		Ordinary		Forced	
Furnace diameters	ins.	30	45	30	45
Ashpit = $\frac{1}{3}$ sectional area of furnace	sq. ft.	1·6	3·7	1·6	3·7
Sum of sectional area of tubes	,,	4	6	2	3
Grate surface (6-ft. bars)	,,	15	$22\frac{1}{2}$	15	$22\frac{1}{2}$
Funnel area	,,	3	$4\frac{1}{2}$	3	$4\frac{1}{2}$
Coal consumption per hour	lbs.	300	450	600	900
Weight of products per second	,,	2	3	4	6
Volume of $\frac{2}{3}$ air (32° F.) through ashpit	cub. ft.	15·8	23·7	31·6	47·5
Mean velocity in ashpit	ft. per sec.	9·9	6·4	13·1	8·5
Resistance	ins.	·076	·032	·31	·13
Volume of $\frac{2}{3}$ products in interior of fuel (2,500° F.)	cub. ft.	96	144	192	289
Upward velocity in fuel	ft. per sec.	6·4	6·4	12·8	12·8
Resistance (*see footnote*[1])	ins.	·070	·070	·28	·28
Volume of all products above fuel (2,500° F.)	cub. ft.	144	216	289	433
Horizontal velocity over fuel	ft. per sec.	90	58	119	77
Resistance	ins.	·47	·20	1·68	·79
Volume of products at 1,000° F.	cub. ft.	72	108	144	216
Velocity on entering small tubes	ft. per sec.	18	18	72	72
Resistance	ins.	·037	·037	·60	·60
Volume of products at 600° F.	cub. ft.	53	79	105	158
Velocity in funnel	ft. per sec.	$17\frac{1}{2}$	$17\frac{1}{2}$	35	35
Resistance	ins.	·047	·047	·19	·19
Total Resistances		·700	·386	3·06	1·99

Closed Ashpits.—When air is blown under the bars it is, perhaps, an advantage to keep the air spaces between them very small, thereby increasing this resistance to such an extent that all the others sink into insignificance. This would ensure a fairly constant supply of air, no matter what condition the fires are in. With a fierce combustion and small furnaces it may then be found necessary to keep the furnace doors open, so as to admit sufficient air above the bars.

Minimum Air Admission above the Grate.—This question is one of the most difficult that presents itself in the working of boilers, and has engaged the attention of engineers for the last fifty years, with no better result than the evolution of an idea that air must be admitted above the bars, but as to how much and by what means has never yet

[1] The values in this division have been calculated on the assumption that the interstices in the fuel bed are one-fifth of the grate surfaces, the upward velocity of the gases would be increased fivefold, and their resistances twenty-five fold.

been settled. The question of how little can only be determined by chemistry corrected by experience. In the trials of the 'Colchester' ('M. E.,' 1890, p. 206) there are three readings in which from 1 to 2 % of carbonic oxide was found in the waste products. The amount of unconsumed oxygen ranged from 5·4 to 6·6 %, corresponding to from 17 to 18½ lbs. of waste products per lb. of coal. On the other hand, there is one sample of gas in this trial where the oxygen had been reduced to 6·43 %, without evolution of carbonic oxide, and in the case of the 'Ville de Douvres' ('M. E.,' 1892, p. 8) there is one with 5·23 % oxygen. This may be explained by only one of the furnaces of the 'Colchester' working badly, and there, perhaps, the oxygen may have been entirely consumed. To be on the safe side, 5 % of free oxygen, corresponding to 16½ lbs. of waste products per lb. of coal, may be looked upon as the lowest limit. Under such conditions the weight of the escaping gases will be a minimum, and, as their temperature is high, they will more readily part with their heat, and weighing less than when more air is admitted, they move more slowly in the tubes, and have more time to cool down. (See chapter on 'Heat Transmission.')

Maximum Air Admission above the Grate.—It would, therefore, appear that as little air should be admitted as is compatible with perfect combustion; but this is not borne out in practice, as will be seen from the following table, which contains some of the results of trials by the Research Committee of the Institute of Mechanical Engineers, and previously referred to:—

Name of Vessel	Ratio of Waste Products to Fuel	Heat carried away by Waste Products per Cent.	Weight of Fuel consumed divided by Total Heating Surface
'Colchester'	18·5	28·0	·987
'Ville de Douvres'	17·9	26·8	1·010
'Fusi Yama'	22·8	23·5	·437
'Tartar'	31·5	22·1	·367
'Iona'	25·5	16·2	·298

It would, in fact, appear as if the order of things had been reversed, and that the more air was admitted the more economical was the boiler; but this view leaves out of account the ratio of fuel burnt per hour to heating surface, which value will be found in the last column, and which seems to exert a far greater influence than the other condition. The only way of settling this question would be to carry out progressive trials with the same boiler. By a comparison of the results obtained on the 'Fusi Yama' and on the 'Tartar,' where the waste products were respectively 23 and 31½ times as heavy as the fuel burnt, and the performances of the boilers fairly equal, it will be found that, if anything, less heat was carried away when a large excess (about twice as much as the necessary quantity) of air had been admitted to the fires, showing that little, if any, harm is done by such proceedings.

These remarks apply, of course, only to the economical side of the question, for it is well known that by opening the doors two influences are at work to reduce the boiler performance—firstly, the uptake temperature, and with it the draught is reduced, and, on account of the relatively smaller resistance encountered by the air above the bars, less air passes through the fuel, and its consumption is therefore seriously checked. With forced draught the available pressure can be increased to practically an unlimited extent, and it ought not to be difficult to regulate its flow.

Velocity of Waste Products.—If anemometers are placed in the ashpits it will be found that their readings cannot be depended upon, as these vary considerably, according to the positions of the instruments. If placed in the uptakes they will soon get damaged by the heat and dirt, and would be difficult to read. Besides, it has not yet been ascertained at what point of the section of a funnel the velocity of the gases is just equal to the mean. These difficulties can, to a certain extent, be overcome by making the arms of the anemometer equal to one quarter of the diameter of the funnel (see fig. 106) and using four vanes, V, set at an angle of 45° to the axis. It will then be found that the time occupied by each vane in sweeping through part of any annular zone of air in the funnel is nearly proportional to the area of that zone, and therefore the readings will be very near the average velocity. A spring or weight should be attached to the clockwork, C, of the instrument, of just sufficient force to overcome the initial friction. The most accurate results would doubtless be obtained if the anemometer were made of the same diameter as the funnel, and its blades shaped like those of the old-fashioned screw propellers, their depth and pitch being kept constant and their widths increasing towards the extremities. In this case it can also be shown that the mean velocity of the gas for all the zones would be recorded.

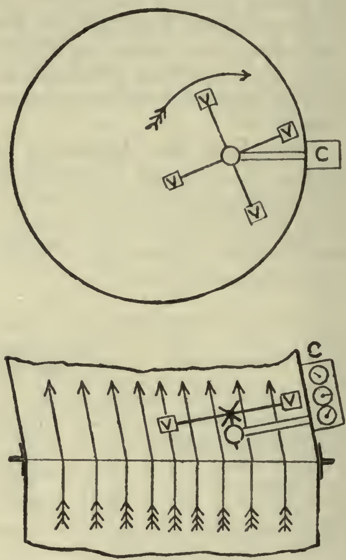

FIG. 106

Air Pressure Gauges usually consist of glass U tubes half filled with water, which may be stained. They are not very accurate, for the total difference of level sometimes does not exceed $\frac{1}{2}$ in. Differential gauges are therefore sometimes used; they consist of a U tube of say $\frac{1}{4}$ in. bore, with two cylindrical bulbs at their top ends. Two non-miscible fluids are put into the two arms; any difference of level in the two large top ends is then very much magnified by a rise or fall of the fluid boundary in one of the arms. The magnification depends partly on the difference of sectional area of the cylinders and U tube, and partly on the difference of density of the two fluids. Paraffin and alcohol are used, but water and aniline oil are more sensitive, as their densities are nearly equal.

Correction for Temperature.—If possible, two or more thermometers should be used, so as to obtain average results, and the necessary data are now available for calculating the weight of the waste products. Let V cubic feet be the measured volume which has passed away in one hour; then if T is its absolute temperature, while 491° is the absolute temperature at 0° F., the weight of air per hour is $Q = \frac{\cdot 0808 \cdot 491 \cdot V}{T} = 39{\cdot}7 \frac{V}{T}$. Here ·0808 is the weight of 1 cubic foot of dry air.

If $T = 918°$ and $V = 1{,}000{,}000$, then $Q = 43{,}200$ lbs.

The necessary allowance should of course be made for the height of the barometer, which is here assumed to be 30 ins., and for the specific weight of the waste gases.

The success which has attended the author's trials to determine the volume of large quantities of water by chemical means (see 'N. A.,' 1896, vol. xxxvii. p. 226, and 'M. S. U. A.,' 1900) encourages the hope that a similar method may be employed to measure the volume and weight of waste gases. In those experiments a carefully gauged stream of salt water was run into the engine hotwell, samples of water were drawn from the pumps, and chemical analyses of a comparative and therefore simple nature were made. The saltness thus found of the salted feed, as compared with that of the added salt water, would be in inverse proportion to the two volumes of water. The circulating water was also measured in the same way, but carbonate of soda was used instead of salt. To measure flue gases a vapour such as hydrochloric acid, or one of its organic combinations, would have to be injected into the escaping gases, and samples of waste gases would have to be taken from near the top of the funnel; these would have to be specially analysed for the injected vapour. Its dilution with waste gases would then determine this amount.

One quart of strong hydrochloric acid per ton of coal burnt would be sufficient to determine the weight of waste gases to within one per cent., provided that at least one cubic yard of gases is passed through the chemical absorption tube. With smaller volumes of gas, especially if greater accuracy is desired, proportionately more acid had to be used. Hydrochloric and hydroiodic acids have been used, but without success.

Chemical Estimate of Weight of Waste Products.—In outline the principle is the same as that just mentioned. The quantity of carbon burnt and the percentage of carbon in the waste products have to be ascertained; then their weight is equal to the weight of carbon burnt multiplied by 100 and divided by the percentage of carbon in the waste gases. The coal must therefore be weighed, a laborious proceeding, and carefully analysed for carbon. The gas analysis is a simple operation. It is also known (see p. 93) that one pound of coal requires 11·587 pounds of air for complete combustion, and that the waste products weigh 12·587 pounds, and further the oxygen consumed and the resulting carbon dioxide are nearly equal in volume, while the volumetric percentage of oxygen in the air is 20·8°/。. If therefore one finds 20·8°/。 carbonic acid and no oxygen it would follow that the waste products weigh 12·587 times the weight of fuel and for lesser quantities of carbonic acid the waste gases are proportionately heavier. The general rule is therefore to divide the percentage of carbonic acid found into 262·0, which is the product $20·8 \times 12·587$. This is the weight of waste products per lb. of fuel burnt. The result is, however, generally far from correct, as will appear from the following remarks.

Collection of Gas Samples.—In order to obtain really reliable results, great care has to be taken to obtain average samples of gas. It would be good if continuous supplies could slowly be drawn from a central tube of every smoke box, and each one analysed ; or, if that is impossible, the supplies might all be brought together in one tube, and an average sample taken.

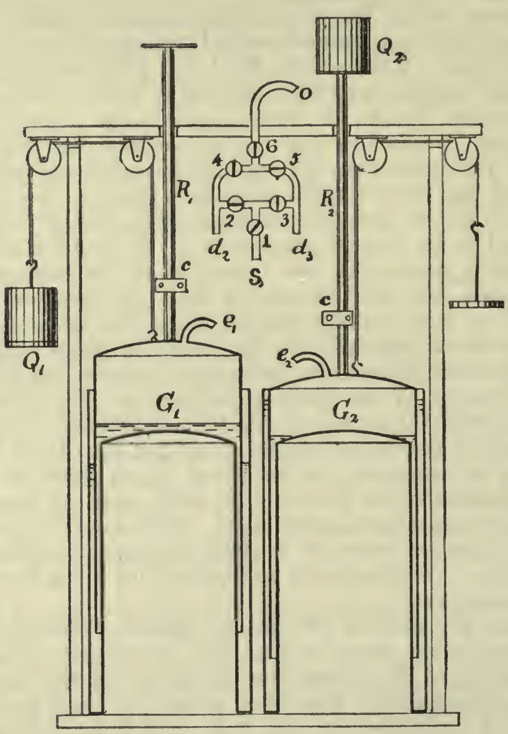

FIG. 108

In order that equal amounts may be drawn from each point, the sizes of the holes should all be equal (fire-proof gas burners can be used), and the section of the tubes which connect them ought to be fairly large, so as to reduce the internal friction. Before bottling a sample a considerable quantity of gas has to be sucked through the tubes, so as to remove all trace of atmospheric air, and a certain time allowance should be made for the volumetric capacity of the pipes, especially if they are long and of large diameter.

India rubber tubes should be used sparingly because CO_2 readily diffuses through them. A lead pipe of small bore is a safe and convenient connection to the gasometer (fig. 108). It is very desirable to introduce a small wash bottle close to the cocks S, for the speed with which the gases bubble through the water is an indication of the rate of suction. If it ceases then most likely the connecting nipples are choked. They can be cleared by a strong blast through either d_2 or d_3, whichever connection is out of use, but it is more convenient to have a three way-cock close to the funnel which should be so arranged that it will not admit air to the gasometer.

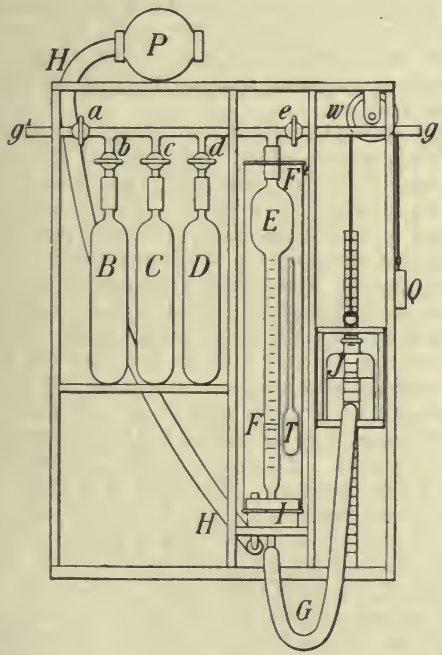

FIG. 110

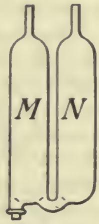

FIG. 111

In order to obtain average results a large number of samples should be collected, analysed, and the weight of gases determined, because one has to take the mean of the reciprocals of the analyses. It would not be correct to take the direct mean, because the percentages of carbonic acid appear in the denominators of previous equations. A convenient apparatus for collecting samples continuously is shown partly in section in fig. 108. Two gasometers G_1 G_2 are placed side by side, the air bells have guide rods, R_1 R_2, which can if necessary be loaded. The suction in G_1 is produced by the weight Q_1, and the connecting rope *c* is a clamp which prevents the bell from being lifted too high. An arrangement of cocks, nos. 1 to 6, is suspended in front of the gasometers. Indiarubber tubes connect d_2 to e_1 and d_3 to e_2, S is similarly connected to G of fig. 107, while O is connected to the Orsat gas analyser (fig. 110). The cock 1 is permanently fixed so as to admit gas at the desired rate, 3 and 4 are closed and 2 opened so that gas from the chimney passes through S, 1, 2, d_2 and e_1 into G_1 which should fill itself in about one hour's time. While this is going on Q_2 depresses the gasometer G_2, the cocks 5 and 6 are opened and the gas collected in G_2 passes to the Orsat apparatus. With a little expedition duplicate analyses could be made in one hour. The gasometers should be annular as shown so as to contain a minimum of water, which should have a film of oil on its upper surface.

If samples are taken at definite periods, these arrangements are not necessary. An air pump or aspirator draws the gases direct into the sample bottle. All water used in connection with this sampling should contain acidulated water which has been well shaken up with waste gases; otherwise some carbonic acid is likely to be absorbed and possibly other gases given off.

Gas Analysis.—It will be well to study the subject in such books as that by Winkler on Gas Analysis, which deals with all its branches. Doubtless the very convenient 'Orsat' analysing apparatus will then be the one selected for use. The following is a brief description:—E, fig. 110, is a glass vessel open at both ends; the lower part is a tube graduated from 0 to 50 cubic centimeters. For our purposes it would be better if this tube were smaller, and held only 25 cubic centimeters. The upper part of E is more bulky, and the total capacity down to the lowest mark of the tube is exactly 100 cubic centimeters. The tube G is attached to the bottom of E, and also to the bottle J, and the two contain sufficient acidulated water to more than fill E. The upper end of E is connected to a glass pipe g^1, a, e, g, with several stopcocks. To fill E with gas, g is connected to the gas supply, the cock a is opened and J lowered. The suction thus produced in E empties it of water and gas flows in. The cock a is then closed. The glass tube g^1, g, has three other cocks, b, c, d; these communicate with glass vessels B, C, D, of which a side view is given in fig. 111. M is filled with the necessary chemical up to the stopcock. This is generally closed, but after E has been filled with gas, and the cock d being open, while J is raised, all the gas in E is expelled into D or M, and the fluid in M passes into N. When absorption has taken place in M, J is lowered until the fluid in N passes back into M right up to the stopcock; the amount of gas absorbed is then read off on the scale on the tube of E. If it is desired to obtain sufficiently accurate results for determining the hydrogen, as previously explained, it is essential to surround E with a water jacket F F and to insert in it a very sensitive thermometer, whose indications have to be taken with each reading, for a difference of $\frac{1}{2}°$ F., which is frequently surpassed in half an hour, if working in a draughty place near a boiler would affect the readings by about $\frac{1}{10}$ cc. A convenient arrangement for circulating the water in F is to blow air into it from the bottom by means of the indiarubber ball B and the tube H. When the above-mentioned accuracy is aimed at it is almost essential to have an arrangement for adjusting the levels in E and J. This latter vessel should be suspended on a pulley and balanced by a weight Q (fig. 110) or a spring, and behind it should be placed a scale arranged as follows. The height of a column of water corresponding to the average barometric pressure is about 400 inches; this has to be expressed in terms of the scale on the gas-vessel E (fig. 110). If these divisions are ¼-inch the column would have a height of 1,600 of these divisions. The top of this scale should be placed where the zero point of E would be if parallel all the way up. Its position can also be fixed by subtracting 100 from the above, making it 1,500, and placing this mark on a level with the 100 mark near the bottom-end of the tube E. This scale is shown behind the bottle J. As ordinarily arranged the scale on E is in front; it should be behind and the figures should be turned as if viewed in a looking-glass, then they will appear in their right position viewed through the water. By raising or lowering J, the water-level in E can now be adjusted to any particular mark with the very greatest accuracy, which is not possible when the scale is in front of the glass. Now read off the water-level of J on the scale behind it, and add the reading of E. This sum is the pressure in the arbitrary scale on the gas in E. All subsequent readings have to be reduced to the first pressure. As the scale behind J will in this case be 16 times as long as the scale E, a tenth of one of the large divisions which are supposed to be ¼-inch, will correspond to 0·4 inch on the J scale, so that it is quite easy to read to decimals of the small divisions which is otherwise impossible, and then there is no difficulty as regards accurate results.

It is very essential that ample time should be allowed before each reading: firstly, to enable the gas which may have changed its temperature to acquire that of the jacket F; secondly, to allow the gas to regain its original humidity; thirdly, to allow all the water in E to drain down the walls. A deliberate wait of five minutes should be made before each reading. Some of the reactions, notably the oxygen and carbon monoxide absorption, require much time, so that with four absorbers each analysis takes up at least half an hour's time, perhaps more, and the suction into the gasometer should be so arranged that it fills in about one hour's time.

Chemicals Required for Analysis.—1. A solution of iodine in water. This converts the sulphurous into sulphuric acid which is absorbed by the fluid.

2. A solution of caustic potash in water for the absorption of the carbonic acid and of the sulphurous acid if the iodine is not used.

3. (i.) Sticks of phosphorus for the absorption of oxygen. The gas may be passed over these first and then through the caustic potash. (ii.) A solution of pyrogallic acid in caustic potash and water. This also absorbs oxygen, but is rather messy. It must not be used until the carbonic acid has been removed, because it absorbs it.

4. A strong solution of subchloride of copper in hydrochloric acid and water for the absorption of carbon monoxide. After treatment the gas has to come in contact again with caustic potash so as to remove the hydrochloric acid gas.

Both the pyrogallic and the subchloride of copper solution have to be carefully protected from the air as they both readily absorb oxygen.

Analytical Operations.—E (fig. 110) has to be filled from the gasometer by lowering J. After waiting a few minutes the first reading of E and J are taken, the water in the jacket F being frequently agitated by pressure on P which sends bubbles through it. The thermometer T has also to be read. The cocks *e* and *a* are closed. Now raise J and open *d*, leading to D, of which a side view is given in fig. 111. It contains say a solution of iodine in water up to the top of M, as well as glass rods, so as to give a large contact surface. N is nearly empty. After waiting a little, pass the gas into E and back into D; then re-introduce into E, wait a few minutes, agitating the water in the jacket, and take readings E, J and T. Now pass the gas into C, containing caustic potash, and treat in the same way. Then pass it into B and treat in the same way. As the absorption here is rather slow it will be found convenient to paste a small piece of paper on B with a mark which indicates when practically all the oxygen has been absorbed, then pass the gas into E and back again and wait at least one minute. If B can be viewed in the dark the end of the absorption is indicated by the disappearance of phosphorescence. The gas should now be passed into the vessel containing cuprous subchloride, not shown; into E and then into C and back to E. This chemical should be used last as it is a powerful oxygen absorber. The cock *e* is now opened, the gas in E blown out, and new gas admitted through *e*. If the cock *a* were opened the inert gas between *a* and *e* would be removed, which should not be done.

Continuous Combustion Indicators.—During the last few years the Ados CO_2 recorder has found great favour ashore amongst steam users, and could equally well be used on steamers. It works automatically by means of the chimney draught and makes an analysis and records the carbonic acid about every ten minutes. It can at any moment be switched on to any boiler, and thus enables a very good idea to be formed as to the efficiency of the stoking. In Mehling's gas composimeter the gas passes through a small orifice into a chamber containing caustic lime and soda, which absorbs all the carbonic acid. The volume of gas is now reduced, and its density diminished, and the suction required for drawing off this gas through another small orifice is less than the first. The two suctions are measured by mercury columns and indicate the percentage of CO_2 absorbed.

Mr. Craig's apparatus, made by Baird and Tatlock, consists of two gas meters with an absorption chamber between the two, but as the gas meters contain water they could not well be used at sea.

Funnel Temperatures.—The chief object of either measurement is the estimating of the heat wasted, and this can now be easily done if frequent readings are taken of the funnel temperatures. In this case, too, care should be taken to get good average results, for it is quite conceivable that if the thermometer is fixed in one position it will only record the temperature of the gases from one particular furnace, which may be hotter or colder than the rest.

For a list of pyrometers see p. 96, to which should be added that for funnel temperatures. A very convenient one is an ordinary mercurial thermometer, made of glass which can stand high temperatures. Quartz would be better. The vacuum end of the tube is filled with nitrogen, which prevents the mercury from boiling. Such instruments were used on the trials of the Research Committee ('M. E.,' 1889–92), and registered up to 860° F. This was not quite high enough for some of the trials, but the sodium and potassium thermometer of E. C. C. Baly and J. C. Chorley is said to register up to 1,100° F.; but of late Callendar's electric pyrometer is proving itself to be most suitable.

Great care must be taken to screen the thermometer bulb from radiant heat, and yet allow the air to come in contact with it; otherwise the readings will be wrong. Thus the temperature of waste gases will apparently be very much less at the top of the funnel than lower down, which is partly due to radiation into space of the heat acquired by the thermometer bulb, and partly due to the cooling of the gases as they travel upwards. In the same way the heated air which is admitted into some furnaces is incorrectly measured if this is done by an unprotected thermometer, because of the radiation, either from the hot fuel or from the heated iron. The screens should be at least twofold, and ought to be highly polished on their outer surfaces, so as to reflect all radiant heat.

If the collecting of the waste gases and the analyses have been correctly carried out then the ratio of carbon to sulphur, hydrogen and nitrogen ought to be the same in the fuel as in the waste gases

The following is one example from a series of twelve collections and analyses taken during one trial. The fuel used during the trial was analysed, after being air dried, with the following result:—

Carbon	74·65
Hydrogen	5·21
Sulphur	2·38
Mineral matter	3·59
Remaining moisture	5·11
By difference oxygen, nitrogen, etc.	9·06
Total	100·00
Add. moisture removed by air-drying	7·34

The ashes and clinker for the whole trial were weighed and analysed and accounted for 2·43 % mineral matter, and 4·37 % carbon, so that only 70·28 carbon were burnt. Possibly the missing 1·16 ashes may have carried off 2 % or 3 % more of carbon as flue dust, and ½ % may have produced smoke, it having been found that dense black smoke represents a loss of 1 % carbon. These doubtful losses will however not be taken into account.

When as in this case the nitrogen has not been determined it may be assumed to amount to 2⅔ % of the carbon, equal to 2 % in the above analysis, and equal to 1 % of the volume of carbonic acid found in the waste gases.

During the trial, which lasted from 9·15 until 5·8, the gases were collected continuously and analysed about every half hour. Here only the third sample will be dealt with, collected from 10·15 to 10·52. Corrections were made for changes of temperature during the half hour occupied in making the analysis. The nitrogen from the fuel, viz. $0·01 \times CO_2$, and the carbonic acid in the air, viz. 0·03 %, have also been subtracted.

One Flue-gas analysis collected from 10·15 *to* 10·52

Gas	Volume	Factors: Densities of Gases and Constituents—air (0·0809 pounds per cubic foot) = 1·0	Calculated Weights: N	O	C	S
	c.c.					
SO_2 . .	0·07	O = 1·132, S = 1·132	...	0·077	...	0·077
CO_2 . .	7·97	O = 1·10601, C = 0·41475	...	8·814	3·306	...
O_2 . .	10·80	O = 1·10535,	...	11·936	...	...
CO . .	0·15	O = 0·55267, C = 0·4145	...	0·083	0·062	...
N . .	80·02	N = 0·9725,	77·900	...	...	...
Totals .	99·01	——	77·900	20·910	3·368	0·077

The above factors have been used in all the calculations.

The weight of the nitrogen in 100 grams of dry air free from carbonic acid is 76·93 grams. All the above weights have therefore to be multiplied by the ratio 76·93 ÷ 77·90, and for convenience the constituents of the fuel, less carbon lost in ash, can be multiplied by a ratio of the carbons, viz. 3·368 ÷ 70·28.

Relative Weights

——	Nitrogen	Oxygen: Consumed	Oxygen: Free	Oxygen: Total	Carbon	Sulphur	Hydrogen
Gas . .	76·93	8·857	11·781	20·638	3·324	0·076	—
Fuel . .	·086	...	...	...	3·324	0·112	0·246

Seeing that the total oxygen should be 23·07 grams per 100 grams air it is evident that the 2·432 grams have disappeared. Now the 0·246 grams hydrogen will have consumed 7·98 times their weight, or 1·963 grams oxygen, so that 0·469 grams are still missing. These it may reasonably be assumed have disappeared as carbonic acid gas through the indiarubber tubing or have been absorbed by the fluid in the gasometer. They should be added to the consumed oxygen, making it 9·326 grams, and $\frac{3}{8}$, or 0·176 grams, being the equivalent loss of carbon, should be added to the 3·324 grams making it 3·500 grams.

The ratio of air supplied to air required is the ratio of the total to consumed oxygen—viz. 23·14 to 9·326 or 2·492 to 1. If the ratio of the total volumes of consumed oxygen, viz. 8·13 cc., had been compared with the original volume viz. 20·8, the ratio of the two air supplies would have been 2·56 to 1, and therefore not correct; but the result is nearer the truth than would have been the case if there had been either more or less leakage of carbonic acid, or if the effect of the hydrogen had been taken into account.

However, with serious leakages such a determination may be quite wrong. If the oxygen determination had been used the result would have been 2·11 to 1.

The difference between the two sulphur determinations is within the limits of observation, but may be due to the analytical combustion in oxygen being more searching than that in the furnaces.

The weight of the waste gases can now be determined as per following table:—

	Grams
Nitrogen in air	76·86
Oxygen in air	23·14
Carbonic acid in air	0 03
Moisture (by hygrometer)	5·10
Carbon (from fuel)	3·53
Hydrogen and sulphur from fuel	0·32
Oxygen, nitrogen, &c., from fuel	0·45
Moisture from fuel, 20·4 per cent. of C.	0·72
Total waste gases per lb. of pure air	110·15

Dividing this total by the carbon (3·53) and multiplying by the percentage (70·28) actually burnt the total weight of waste gases per pound of fuel fired is 21·93 lbs. This has to be multiplied by the temperature of the waste gases, which was 264° F. in excess of that of the air, and by 0·238 the specific heat of average waste gases; the product 720 B.T.U. is the heat carried up the chimney for every pound of fuel.

Although the above process appears complicated it works out fairly simply in practice, but the gas analysis has to be carefully done. As the sulphurous acid is always a very small volume and as it would be absorbed together with the carbonic acid if not already dealt with, it need not, generally speaking, be determined, but then $\frac{3}{8}$ of the sulphur in the fuel has to be added to the carbon when making the above estimate. An accurate gas analysis is, however, a desirable check on the fuel analysis.

It is evident that if the gas analyses have to be corrected for the hydrogen in the fuel it should be possible, by very carefully collecting and analysing the waste gases, to determine the ratio of hydrogen to carbon in the fuel, and thus dispensing with one rather costly analysis. The process is simple enough. Make the necessary allowances for the carbonic acid in the air and the nitrogen in the fuel, and then as above find the missing oxygen, divide it by 7·98 the molecular weight of oxygen, and also by the weight of carbon in the waste gases, and the result is the ratio of hydrogen to carbon in the fuel, or rather to that portion of it which has actually been consumed.

Now it will be noticed that in the above example it has been assumed that the only waste of carbon is that associated with the ashes and clinker, but occasionally, especially with forced draught, small particles of partly burnt fuel are carried away and these, instead of containing say 30 to 60 per cent. carbon, may contain much larger proportions, and as in nearly all trials a very large percentage of mineral matter disappears either into the flues or up the chimney, an even larger proportion of carbon is also carried away. Smoke, although the densest is responsible for a loss of only about 1 % of carbon, should not be entirely neglected.

Another uncertainty arises out of the very small amounts of fuel used for analysis, generally only 2 to 5 grams, or perhaps, one-millionth part of the fuel burnt, and what makes matters worse is that the fuel has to be ground to an impalpable powder and will almost certainly lose all volatile hydrocarbons which it may contain and may absorb a large volume of oxygen and nitrogen.

Most fuel analyses, like the present one, contain a large item: 'oxygen,' or 'oxygen &c., by difference,' but it does not appear to have been shown that this difference is really oxygen, and in the present example it has not been taken into account as such, otherwise the percentage of hydrogen in the fuel would have had to be reduced by one-eighth of this amount less the nitrogen.

On account of all these difficulties and uncertainties it would be very desirable to devise a reliable anemometer as suggested on page 102, and to carefully analyse the waste gases instead of as now relying on an analysis of a homœopathic dose of the fuel.

Variations in the Process of Combustion.—It is evident that when fresh fuel is thrown on a fire distillation takes place, that hydrocarbons are freely evolved and that a careful analysis of the waste products will reveal much missing oxygen, indicating that much hydrogen has been evolved and consumed. When the bed of fuel is all aglow, probably all hydrogen will have been driven off and only carbonic acid and perhaps carbon monoxide will be found in the waste gases. In the trial from which the above-mentioned example has been taken firing was reduced at 3.0 and ceased at about 3.30, but the fire was so thick that it continued to burn till after five o'clock. The ratios of the consumed hydrogen to carbon as found by gas analysis were as follows:—

Times of Collection	Ratio H÷C	Waste Gases per lb. of Fuel
h. m. h. m. m.	per cent.	
9.15– 9.45 = 30	11·14	20·8
9.45–10.15 = 30	10·31	21·7
10.15–10.52 = 37	10·88	21·9
10.52–11.30 = 38	9·60	22·0
11.30–12.0 = 30	6·73	21·5
12.0 –12.50 = 50	8·38	21·3
12.50– 1.35 = 35	10·56	21·3
1.35– 2.25 = 50	8·79	21·3
2.25– 3.0 = 35	10·68	21·8
3.0 – 3.35 = 35	7·21	22·0
3.35– 4.15 = 40	4·88	22·6
4.15– 5.8 = 53	0·00	22·8

The average ratio of H to C is 8·06 %, whereas according to the coal analysis it is 7·41 %. It will be noticed that there is considerable variation in the ratio, but that it is well above the average before three o'clock and fell off to nothing during the remaining two hours. In these and similar cases where the firing is irregular it may be desirable to estimate the initial flame temperatures, which are affected by the ratio of carbon to hydrogen and to the total waste products. The excess temperature over the air temperature can be found according to the formula given on page 94, but if the detailed determination of the weight of waste gases has already been carried out it will be simplest to write

$$T = \frac{1}{0{\cdot}237} \cdot \frac{C\,.\,14{,}540 + H\,.\,51{,}322}{\text{Weight of waste products}}$$

Then the efficiency e, neglecting radiation, is found by subtracting the temperature of the waste gases from T and dividing by T. Now let H_1, H_2 &c., be the heat produced in the furnaces during the times t_1, t_2 &c. Let S_1, S_2 &c. be the heats utilised in evaporating the water, and R the radiated heat which may be assumed to be a constant value, then

$$H_1\,.\,t_1\,.\,e_1 = S_1\,t_1 + R\,.\,t_1\,;\ H_2\,.\,t_2\,e_2 = S_2\,t_2 + R\,.\,t_2, \text{\&c.};$$

dividing by e, and summing up, we have

$$\Sigma(H\,.\,t) = \Sigma\left(\frac{s\,.\,t}{e}\right) + R\,\Sigma\left(\frac{t}{e}\right).$$

Here $\Sigma(H\,.\,t)$ is the total heat generated during the whole trial, which is of course known, the second term can be calculated and so can $\Sigma\left(\frac{t}{e}\right)$, the radiation R per unit of time can therefore be determined and then of course H_1, H_2, &c. It will thus be possible to determine the heat generated during each time interval and also the quantity of carbon and hydrogen burnt. The heat generated less the chimney loss during each interval, which can now be calculated from the estimated carbon consumption and other data, will, as a rule, not agree with S, the heat required for the steam production, unless this too has been carefully determined.

Weight of Steam Produced.—All the water which is pumped into a boiler is not necessarily evaporated at once: it may be in excess of the heat transmission or less. In the one case the waterlevel rises and the temperature and pressure sink, in the other case the reverse happens. It is therefore desirable to determine the waterlevels and pressures at definite intervals, or whenever the chemist has filled one of his gasometers. The weight of the water in the boiler for various levels should be known, the total heat contained in the water at any time for which the data are available can then of course be calculated. This is heat which passed through the heating surface and should be added to the head of evaporation during the past interval. The successive sums of heats should then be subtracted from each other. These are the values S, mentioned above.

CHAPTER V

HEAT TRANSMISSION

Nature of Heat.—At one time heat and light were looked upon as being two different substances or conditions; now it is universally admitted that they are identical, the only difference being an objective one, and depending solely on the incapability of our optic nerves to perceive heat rays. Our other nerves make no distinction between the two, and the back of the hand will as readily feel the radiation from a blackened stove as from an electric arc, the one emitting only heat and the other chiefly light.

By those who believe in the existence of a universal ether, light and heat are thought to be vibrations of this medium. Every known experiment on the subject points to the conclusion that these vibrations are of a transverse nature, like waves on the surface of fluids, and no relation seems to exist between these and the longitudinal vibrations called sound. The lengths of the waves of visible heat (light) range from 30,000 to 60,000 per inch. The non-luminous heat waves are longer, and the ultra-violet light, invisible to our eye but very active in photographic reactions and in vegetable life, consists of very much smaller waves. The speed with which heat and light travel through space is at the rate of about 187,000 miles per second, so that the number of vibrations per second has to be counted by hundreds of billions.

Radiant Heat of Gases and Solids.—Heat being invisible, many of the phenomena connected with radiation will be more readily understood if exemplified by the action of light, though Prof. Langley ('C. E.,' vol. c. p. 469) has done a good deal to confirm, by direct experiment, the conclusions which naturally suggest themselves when comparing light and heat.

It has been found that solids, when heated to incandescence, radiate heat and light of every colour, which is very clearly shown in the spectroscope by the fact that the spectrum of a luminous solid is a continuous one, light being visible continuously from the red end to the violet, and perceptible beyond both ends. Luminous gases do not radiate light of every colour, and some are even monochromatic. Thus the vapour of the metal cæsium has a spectrum consisting of only one green line. Indium shows three lines, chiefly blue; hydrogen shows four lines, placed in various parts of the entire spectrum. Arsenic and sodium show several lines near the yellow; potassium also shows comparatively few lines. For further information see W. M. Watts, 1872. Naturally all these gases also radiate definite colours of heat, if one may use this term; but the important point is that for the same intensity of colour of light or heat the heat thus radiated is very small as compared with the heat radiated by a solid which shows the same intensity of light and heat over the entire spectrum; so that it will take much longer to cool a gas than a solid. An idea of the enormous disparity existing between the two will be obtained by examining the spectrum lines of sodium with a good instrument. They will appear to be no wider than very fine inked lines. The full spectrum from red to violet, measured by yards, represents the amount of radiation from a solid of the same temperature as the gas, whereas the breadth of the lines represents the entire heat radiation of sodium vapour.

Absorption of Heat.—Gases absorb only such light as they can emit, but experiments as to the influence of temperature on this phenomenon are still wanting. According to present views it would appear that gaseous flames are only luminous at their outer surfaces, which is contrary to what one sees.

It may, therefore, be assumed that flames, unless they are quite luminous, radiate and absorb little heat, and none at all when they have grown cold and transparent. The heat which radiates from a glowing coke fire is not absorbed by the non-luminous flame over it. It also follows that when the gases are cooled below the point of luminosity they cannot transmit any of their heat to the boiler plates, except by actual contact, or—and this is of great importance—by contact with the suspended particles of soot and ashes, which, as they are solids, are capable of radiating heat even at low temperatures. It would therefore appear probable that a little smoke is a good admixture to the products of combustion. It is also probable that luminous flames, such as are produced from North-country coal, will be more efficient radiators in the furnace than those which are produced from coke, anthracite, &c. In the former case the radiating surface, i.e. the outer part of the flame, is very much larger than the exposed incandescent surface of the fuel on the grate.

Another matter which should be mentioned here is the influence of moisture and of carbonic acid on the radiating power of air. Prof. Tyndall (1870, pp. 320, 359) states that carbonic acid absorbs (and therefore also emits) 972 times as much heat as any of the permanent gases, including oxygen and nitrogen, and that humid air (he does not give the percentage of moisture) absorbs 90 times as much heat as dry air. As the products of combustion contain from 10 to 20 % of carbonic acid, this alone would raise the absorbing and radiating power about one or two hundredfold, and the moisture contained in the air and in the fuel would add considerably to this power. As these impurities, including smoke, vary considerably in every trial, it cannot be expected that much uniformity will be obtained in experiments which do not take them into account.

Radiating Power of Solids.—Physicists call those substances black as regards radiating power which emit a maximum amount of heat even when in an incandescent condition, and Stefan and Bolzman have discovered a law, which is fully confirmed by experiments, that this black substance, including the glowing particles in coal gas flame, emit a certain amount of heat per square foot per hour.

$$h = 16 \, . \, 10^{-10} \, . \, T^4$$

Here T is the absolute temperature, found by adding 459° to the temperature of the incandescent substance. The heat radiated by a bed of incandescent coke or anthracite is evidently proportional to the grate area, that radiated by a furnace flame is proportional to the heating surface of the furnace above the grate. Now the furnace temperature depends on the amount of excess air mixed with the flame and we have the following interesting results. See page 94.

CO_2 °/₀	Ratio of Air to Fuel	Excess Air	Flame Temperature		Radiated Heat = h
			° Fah.	Absolute	
16·1	15	30°/₀	3,830	4,290	542,000 B.T.U.
12·1	20	70 ,,	2,920	3,380	210,000 ,,
9·7	25	115 ,,	2,350	2,810	99,000 ,,
8·0	30	160 ,,	1,970	2,330	47,000 ,,
6·9	35	200 ,,	1,700	2,160	35,000 ,,

It will at once be seen that if the combustion in a furnace is improved from a CO_2 percentage of say 6·9 which is a low but not unusual result, to 12·1°/₀, which is a good result, then the quantity of heat imparted to the furnace plate is sixfolded, but considerable deductions must be made from the larger values on account of the extremely rapid loss of heat. Thus assuming that a grate is burning 30 lbs. of coal per sq. ft. per hour, or say 20 lbs. per sq. ft. of upper furnace surface, this combustion will produce $20 \times 15{,}000 = 300{,}000$ B.T.U. per sq. ft. of furnace surface, and if the combustion was so perfect as to result in a temperature of 2,920° F., and if this temperature could have been maintained 210,000 B.T.U. would have been abstracted by the furnace plates and only 90,000 B.T.U. would be found in the gases passing over the bridge, whose temperature would be reduced to about 1,300° F. Of course with the reductions of temperature the radiation will diminish and the above result will not be attained, but there is every reason to believe that very intense local heat productions are attained which would do much damage by local overheating (blowpipe effect) in scaly or greasy boilers if they were not of a temporary nature, at least in hand-fired furnaces. With mechanical stokers local productions of intense heat are of a more persistent nature and are the cause of occasional serious damage.

Surface Resistance.—The furnace plates radiate heat into the furnace according to the law mentioned above, but as the steam temperatures are low the maximum deductions to be made from the values in the above table do not amount to 1,000 B.T.U. and are negligible. The case is different when there is scale or grease in the boiler and the plates attain a high temperature, say 1,000° F. or 1,500° absolute, then the plate radiation, which has to be deducted from the flame radiation amounts to 8,000 B.T.U. which is quite appreciable against a low temperature flame or fire, and the net heat supplied to the boiler is materially reduced.

Coatings of Soot and Tar intensify this effect, for as they are excellent non-conductors their outer surfaces can be brought to an incandescent condition, when a flame unless it is very much heated will not be able to transfer much of its heat to the plate. Soot and tar deposit most readily on the cooler parts of boilers, i.e. in or on the tubes, and especially on the surfaces of feed economiser pipes, for which reason they have to be scraped continuously. It has not been found possible to introduce scrapers for boiler tubes, but attempts have been made to keep them clean by means of occasional steam blasts.

Surface Resistances.—In addition to the resistance which heat meets in its passage through the plates, there is the resistance to its entrance on one side and to its exit on the other side. This differs for different plate materials, perhaps also for different hot gases, and stands in no relation to the thermal conductivity. Thus cast iron, which has a much lower thermal conductivity than copper, is nevertheless a better transmitter of heat for all reasonable thicknesses, because it is the better absorber and emitter of heat. Even earthenware tiles $\frac{1}{8}$ in. thickness are 70 % better than copper. (See p. 126.) From various experiments and practical experiences it would appear that when air is in contact with metal the heat transmission is very low.

Some meagre results are to be gleaned from the performances of steamers fitted with Howden's system of hot-air draught, according to which about $2\frac{3}{4}$ units of heat are transmitted per hour through one square foot for every degree of difference. The amount would be double for each surface on the assumption that the metal acquires the mean temperature. Plates which are exposed to the direct action of the flame also receive heat by radiation. (See p. 115.)

Considerable light has recently been thrown on this subject by some very careful experiments which were carried out by Miss E. M. Bryant ('C. E.,' 1897, vol. clxxxii. p. 274), who measured the temperature gradient in the heated plate of an experimental boiler by electric methods. In the first set of experiments a copper plate was heated by a flame, in four other experiments the heating of steel and copper plates was done by radiation from a cast-iron hemisphere which was placed, concave side upwards, below the plate, while the heat was applied to the outside of this casting. Its temperature was measured electrically. The heated plates formed the bottoms of dishes filled with water. Comparing the amount of water evaporated with the measured temperature gradient in the plates, it appears that the thermal conductivity of steel (across the grain) is 0·093 G.C.S. This is the mean of 44 experiments, but, as will be seen on comparison with the previous table, it is a rather low value. For copper the average of 25 experiments gave 0·409 thermal conductivity. The ratio of the two conductivities has usually been found to be about 1 to 8; here it is 2 to 9. It is of course well known that impure copper is a very bad conductor of heat, and the experimental plate may have been a bad one.

According to Miss Bryant's summary, the heat radiated from the cast-iron hemisphere to the boiler plate is proportional to the square of the difference of temperature between the two.

$$H = k.(\Delta t)^2$$

in which H is expressed in evaporative units per square foot per hour. The values of k are as follows :—

Copper Plate

I.	Experiments.	Underside	oxidised . . .	$k = 0{\cdot}000{,}007{,}65$
I.	„	„	slightly smoked . .	$k = 0{\cdot}000{,}011{,}76$
I.	„	„	smoked . . .	$k = 0{\cdot}000{,}011{,}52$

Mild Steel Plate

III.	Experiments.	Underside	clean	$k = 0{\cdot}000{,}006{,}7$
IV.	„	„	smoked . . .	$k = 0{\cdot}000{,}010{,}9$
II.	„	„	rusted	$k = 0{\cdot}000{,}008{,}8$
II.	„	„	rusted and smoked .	$k = 0{\cdot}000{,}013{,}1$

It will be seen that smoked heating surfaces are much more efficient than clean ones, but contrary to expectations, rusted surfaces are very inefficient even if covered with soot. This matter deserves further consideration. It is also a practical experience that the scraping of heating surfaces whereby the tarry film is removed improves a heating surface.

Convection.—Miss Bryan's experiments, and in fact all practical experiments on heat transmission from flames, are mixed up with the question of convection or heat transmission by contact between gases and solids, which differs materially from the question of heat transmission in solids, because gases have very low thermal conductivities and also because they can be agitated, unused warm portions replacing those which have been in contact with the heating surface. The desirability of agitating flue gases has always been felt by boiler inventors and has led some of them to design marvellous complications, which look as if they were expected to trap or cheat the gases out of their heat. There have also not been wanting very forcible reminders, such as bulged and cracked plates over fire bridges and other obstructions in furnaces or flues, that heat transmission is intensified by disturbing obstructions, but the first man to suggest this idea to scientists of the day (1875) was Prof. Osb. Reynolds. Since then a large number of experiments have been made on the effect of velocity of water on the rate of heat transmission, and more recently Mr. P. Jordan ('M. E.' 1909, parts 3 and 4, page 1317) experimented on the velocity influence of air. His paper is a somewhat difficult one to study, on account of absence of actual measurements; his results have therefore had to be resummarised as follows :—

Five series of experiments were carried out: (E) on a copper pipe of $\frac{1}{2}$-inch bore, (F) on $1\frac{1}{4}$-inch bore, (D) on 2-inch bore, (C and B) on one or the other of these pipes with a central core, leaving an angular space. Let H be the heat (B.T.U.) transmitted per square foot per second from air of the temperature Ta to the copper tube whose surface temperature is Tc. Let V be the velocity of the air in feet per second $= Q \times 13{\cdot}141 \,.\, Ta : a \,.\, 491$, where Q is the weight in lbs. per second of the air passing through the tube section a, and where Ta is the absolute temperature of the air then

$$H = (Ta - Tc)\left(K_1 + C_1\frac{Q}{a}\right) \text{ or } H = (Ta - Tc)\,(K_2 + C_2V).$$

Table of Values of Constants Deduced from Experiments.

Experiment	Diameter Inch	K_1	C_1	K_2	C_2
E	$\frac{1}{2}$	0·001,43	0·000,80	0·002,36	0·000,036
F	$1\frac{1}{4}$	0·001,76	0·000,70	0·001,34	0·000,031
D	2	0·001,20	0·000,70	0·001,15	0·000,032
C	Annular	0·001,62	0·000,71	0·001,12	0·000,036
B	Annular	0·002,14	0·000,67	0·001,58	0·000,037
Mean of above		0·001,65	0·000,72	0·001,53	0·000,034

H can be converted into B.T.U. per square foot per hour by multiplying both K and C by 3,600 seconds, but if $\frac{Q}{a}$ and V are expressed respectively in pounds weight or feet per hour the C values remain as they are. In ordinary boiler practice V is about 10 ft. per second, so that $H = (Ta - Tc)\ 0·002$ B.T.U. per second. If it should be desired to reduce the heating surface to one-half by increasing the coefficient from 0·002 to 0·004, this would have to be done by increasing the velocity of the gases so that $V \times 0·000,039$ is about 0·002,5, which means that an air velocity of 70 ft. per second or 45 miles an hour must be reached instead of 10 ft. per second. Leaving the furnace diameter unchanged, the tubes which are usually 3 inches bore and 11 ft. long would have to be altered to $1\frac{1}{8}$ inch tubes 11 ft. long, or if the number of tubes be increased by 30°/$_\circ$ with diameters reduced to 1 inch, the lengths could be reduced to $7\frac{3}{4}$ feet. The very important fact must not be overlooked that the air friction increases as the square of the velocity, and that an increase from 10 to 70 ft. per second means that the resistance is fifty-folded. Even this resistance is no very serious matter in comparison with the resistance through the fuel, but ordinary chimney draft would now be out of the question, and forced draft would have to be resorted to.

Another matter which must not be overlooked is that an $\frac{1}{8}$ inch coating of soot, which would reduce the sectional area of a 3 inch tube by 16°/$_\circ$, would in 1-inch tube cause a reduction of 44°/$_\circ$, and the respective increases of friction would be 42 and 215°/$_\circ$, which would almost certainly mean that small tubes would have to be cleaned constantly as is done with economiser tubes of land boilers.

The Heat Transmission from the tube surface to the water is also affected by velocity say

$$H = (Te - Tw)\left(0·05 + 0·003\frac{Q}{a}\right),$$

but as the heat transmission for stationary water is very much better than that of air even when moving rapidly, practically no improvement could be attained by increasing the water velocity in ordinary boilers. In water-tube boilers rapid circulation is of importance because the steam bubbles have to be carried away.

Thermal Conductivities.—The law which regulates the flow of heat through plates is a very simple one. Let t_1 and t_2 be the temperatures inside of a plate, measured at two points which are l inches apart, then $\frac{t_2 - t_1}{l}$ is the temperature gradient, and if C is the thermal conductivity of the material of the plate, the amount of heat which passes across an area A of the plate in one hour is $H = C \cdot A \cdot \frac{t_2 - t_1}{l}$. The table on this page gives the values of C for various substances.

Thermal Conductivity of Solids=C.

Materials	G.C.S. Scale	Fahrenheit Scales		Observers
		Thermal Units	Evaporative Units	
Steel (hard) . .	·062	180	·186	Kohlrausch
Steel (soft) . .	·111	321	·333	,,
Iron at 32° F. .	·207 to ·154	603 to 449	·621 to ·462	Forbes
,, at 212° F. .	·157 to ·129	456 to 375	·472 to ·387	,,
,, at 527° F. .	·124 to ·112	361 to 297	·372 to ·306	,,
,, ...	·164	477	·492	Neumann
,, ...	$·199(1-·0029\,t)$	$610(1-·0015\,t)$	$·6(1-·0015\,t)$	Ångström
Copper . . .	$1·027(1-·0021\,t)$	$8140(1-·00115\,t)$	$3·08(1-·0015\,t)$	,,
,, . . .	1·108	3220	3·024	Neumann
Brass . . .	·302	878	·906	,,
,, . . .	·150	434	·450	Weber
Zinc . . .	·307	892	·921	Neumann
German silver .	·109	317	·327	,,
Slate, along cleavage	·0055 to ·0065	160 to 190	·0165 to ·0195	,,
,, across cleavage	·00315 to ·0036	92 to 105	·0095 to ·0108	,,
,, ...	·00081	23·5	·00243	Forbes
Clay, sun-dried .	·00223	6·6	·00669	Neumann
Chalk . . .	·0020 to ·0033	5·8 to 9·6	·006 to ·010	,,
Glass . . .	·00143 to ·00179	4·14 to 5·18	·00429 to ·00537	Meyer
Fire-Brick . .	·00174	5·1	·00522	Neumann
Plaster of Paris, wet	·00164	4·8	·00492	,,
Coal . . .	·00057 to ·00113	1·65 to 3·3	·0017 to ·0034	,,
Pumice stone .	·00055	1·60	·00165	,,
Sand (quartz) .	·00013	·376	·00039	Forbes
Various woods .	·00026 to ·00359	·76 to 1·71	·00078 to ·00177	Peclet
Caoutchouc . .	·00041	1·19	·00123	,,
,, vulcanised	·000089	·258	·000267	Forbes
Guttapercha .	·00048	1·40	·00144	Peclet
Powdered charcoal	·00022	·64	·00066	,,
,, coke .	·00044	1·28	·00132	,,
Charred wood .	·000122	·35	·000366	,,
Grey paper . .	·000094	·273	·000282	,,
Pasteboard . .	·000453	1·33	·00186	Forbes
Paraffin . .	·00014	·41	·00042	,,
Flannel . .	·0000335	·097	·0001	,,
Water . . .	·00136	4·00	·004	Weber

In this table the values given in the column headed 'Thermal Units' are the number of units of heat which one square foot of heating surface 1 in. thick will transmit per hour, if the difference of temperature of the two surfaces of the plate itself is one degree Fahrenheit. The values in the column 'Evaporative Units' are found from the last by dividing them by 966, which is the number of thermal units required to evaporate one pound of water from and at 212° F. These values can also be obtained direct from the G.C.S. values by multiplying them by 3.

It should be mentioned that all these experiments have been carried out on rods or rings, and that they are not absolutely reliable, because they are based on an imperfect knowledge of the changes of the specific heat and of the radiating power of the substances. If carried out on a plan similar to the one adopted by A. F. Yarrow for showing the curving of heated plates, these difficulties might be overcome, and, what is of almost greater importance, the coefficients of transmission of heat across the fibre of the material could thus be measured. (See below.)

Of more importance than the theories as to how the heat transmission takes place is the question of how much heat can be transmitted. This very soon resolves itself into the oft-debated subject as to the relative merits of the heating surfaces of the furnaces and of the tubes. Unfortunately few experiments have been carried out, and those few are very incomplete.

De Pambour was the first to raise the question ('Comp. Rend.,' 1840, vol. x. pp. 32, 111, 480), and to make comparisons, but these are valueless.

C. W. Williams ('Engineer,' 1858, vol. v. pp. 223, 243, and 'N.A.,' 1862, vol. iii. pp. 122) made experiments on the heat transmission of tubes.

J. Graham ('Manch. L. Ph.,' 1860, vol. xv. p. 8) experimented on the heat transmission of flat surfaces.

M. Geoffroy ('Couche,' 1877, vol. iii. p. 28) made experiments on a locomotive boiler whose length was subdivided.

P. Havrez ('An. Génie,' 1874, 2nd ser. vol. iii. p. 520) and J. A. Longridge ('C. E.,' 1878, vol. lii. p. 101) give analyses of these experiments. J. Durston, 'N. A.,' 1893, vol. xxxiv. p. 130.

Reference is also repeatedly made to M. Petit's experiments, but details are wanting. They seem to have consisted of two series: in the one lot coke was burnt, and in the other briquettes; and the draught was $\frac{3}{4}$ in. in the one case and 4 ins. in the other. Compare P. Havrez (see above), p. 550, and 'Chron. Ind.,' 1873, vol. ii. p. 425.

C. W. Williams ('Engineer,' 1858, vol. v. p. 206) also mentions experiments on a subdivided boiler carried out by Messrs. Wood and Dewrance, but all details are wanting.

The experiments detailed by A. F. Yarrow ('N. A.,' 1891, vol. xxxii. p. 108) can be used for determining the amount of heat transmitted through a plate. He measured the curvature of a tube plate which was covered with water and heated over a smith's fire. Its acquired radius was 550 ins., i.e. one of its surfaces had expanded $\frac{1}{550}$ of its length more than the other side (assuming the plate to have been 1 in. thick). But this can only have been brought about by the fire side of the plate being 330° F. hotter than the other, and as experiments made on the flow of heat in iron bars (see table, p. 120) show that this difference of heat per inch of distance can only exist if sufficient heat is being transmitted to evaporate 142 lbs. from and at 212° F., this must have been the evaporation under these conditions.

A. F. Yarrow also made similar comparative experiments on iron and copper plates placed over gas jets, and found that 57 lbs. per square foot were evaporated over the iron plates, and only 32 lbs. over the copper plates, which is in accordance with other experiments. (See pp. 118 and 126.)

Mean Temperatures of Heating Surfaces.—Another set of interesting experiments were carried out by J. Hirsh ('Soc. d'Enc.,' 1890, vol. v. p. 302), and are mentioned in the second part of his paper. He constructed a small kettle about 10 ins. in diameter, which had an iron bottom about $\frac{3}{8}$ in. thick. Arrangements were made for keeping the water level at a constant height, while a strong gas and air blast was directed against the bottom, of which a small area of 4 ins. in diameter was exposed. Twenty-four holes were drilled into the bottom of this plate $\frac{1}{8}$ in. in diameter and $\frac{1}{4}$ in. deep, and these were filled with lead and tin alloys. In all 38 experiments were carried out, of which the results of a few are given in the following table.

Conditions	Water Evaporated from and at 212° F. per Sq. Foot per Hour	Melting Temperature of Alloys		
		Above	Near	Below
	Lbs.	°F.	°F.	°F.
Distilled water	26·8	...	335	...
	35·6	...	369	...
	64·5	...	428	...
	82·5	428	...	482
	93·5	"	...	...
Water which contained $\frac{1}{2}$% starch	35·8	369	...	428
	59·2	...	428	...
	77·0	428	...	482
Distilled water. $\frac{1}{25}$ in. scale on plate	33·8	369	...	482
	46·1	...	428	...
Distilled water. $\frac{1}{5}$ in. scale on plate	34·6	428	...	482
	53·2	..	842	...
Distilled water. Plate greased with mineral oil	33·8	...	428	...
	34·3	...	428	...
	47·1	...	428	...
	62·6	482	...	635
	63·2	"	...	"

Sir John Durston ('N. A.,' 1893, vol. xxxiv. p. 133) has carried out similar experiments in open and closed vessels. See following tables:—

Conditions	Temperatures		
	Flame	Plate on Fire side	Water
	°F.	°F.	°F.
Dish $\frac{1}{4}$ in. thick			
Fresh water. Bunsen burners	1500	240	212
Do. Grease $\frac{1}{16}$ in. do.	1500	330	212
Fresh water. Forge fire	2200	280	212
Do. and 5 per cent. mineral oil. Forge fire	2300	310	212
Do. and $2\frac{1}{2}$ per cent. paraffin. Forge fire	2100	330	212
Do. and $2\frac{1}{2}$ per cent. methylated spirits. Forge fire	2500	300	212
Do. Greasy deposit $\frac{1}{16}$ in. thick. Forge fire	2500	550 +	212
Dish $\frac{3}{8}$ in. thick			
Fresh water. Bunsen burner	1500	430	363
Do. Forge fire	2000	430	344·5
Do. Greasy deposit $\frac{1}{16}$ in. Forge fire	2000	510	358
Do. Dry greasy deposit $\frac{1}{16}$ in. Forge fire	2000	550	351
Do. do. do.	2000	617	80

In the above experiments the fusible alloys were soldered to the fire sides of the plates; in the following experiments they were round or cubic plugs driven or placed in cavities in the plates. On account of the insufficient contact the recorded temperatures are probably too low. See Miss Bryant's paper ('C. E.,' 1897, vol. cxxxii. p. 274) for remarks on the reliability of measuring temperatures by means of alloys. It is also well known that fusible plugs in boilers are not as reliable as they should be.

Conditions	Temperatures			
	Flame	Plate on Fire side	Plate Centre	Water
	°F.	°F.	°F.	°F.
Dish with tubes. Fresh water	2000	...	290–336	212
Boiler tube plate "	3100	...	540	366
Do. Fresh water.	2750	690 +	...	363
Do. "	2500	750 +	...	362
Do. Fresh water and 0·07 per cent. mineral oil	3100	750–1060	...	360
Do. Fresh water and further 0·07 per cent. mineral oil	3200	1060	680–750	363

In order to compare these values with the estimated transmission in A. F. Yarrow's experiments, the excess temperatures above 212° must be divided by $\frac{3}{8}$, that being the ratio of the thicknesses of plates, and it will then be seen that in this case the heat transmitted is less, but grave doubts are entertained whether the indications of the alloys can be relied upon. Their outer surface is not iron, and they are not soldered to the bottoms of the holes, as they should be, and according to some supplementary experiments in which two thicknesses of plate were bolted together, the resistance to the passage of heat across the boundary of the two metals will have been very great, and the indicated temperatures must therefore be looked upon as excessive.

This is particularly the case where the plate has been covered with scale, and it is of interest to note that when this is $\frac{1}{8}$ in. thick the temperature of the fire side of the iron plate has to be increased an extra 460° when evaporating 55 lbs. of water per hour. According to this experiment, scale offers about five times as much resistance to the passage of heat as iron does, whereas laboratory experiments show the ratio between iron and plaster of Paris to be as 1 to 100. This may be due to the thermal conductivity of iron having been measured along the fibre, while in these experiments the heat travelled across the plate and across the various layers of fine slag. Then, too, the boiler scale is in a saturated condition.

A very important point to be noted is, that even a $\frac{3}{8}$-in. plate can be heated to above the melting temperature of lead, if coated with a little scale on the water side, provided the fire is so hot that 55 lbs. of water are evaporated per square foot per hour, and it is therefore not unreasonable to suppose that most of the late troubles with the tube plates of Navy boilers are due to over-heating, particularly when it is remembered that it is probably not water, but moist steam which is in contact with these plates.

The last set of experiments also show that an injurious effect is obtained by allowing grease to settle on heating surfaces.

Somewhat similar experiments have recently been made by the late Dr. Kirk ('Enging.,' 1892, vol. liv. p. 333). As in the above case, plugs of alloys were fitted into the bottom of a plate for determining its temperature. It was originally $2\frac{3}{4}$ ins. thick, and was gradually reduced to $\frac{13}{16}$ in. As no measurements were taken of the water evaporated, the results are of less value than the above.

Wye Williams ('N. A.,' 1894, vol. xxxv. p. 284) fitted thousands of studs into some furnace plates in order to increase the heating surface. They projected 3 ins. on both sides, but on the fire side they burnt away to $2\frac{1}{2}$ ins. As oxidation (blue) commences at about 500° F., the ends of these studs were doubtless heated to 1,000° F. or more and must have been effective as heat transmitters. Some patent boilers consist of tubes partly filled with water, sealed at both ends, and fitted like these studs, half in the fire, half in the water; the heat transmission is a more rapid one than by conductivity, but these boilers are not satisfactory.

Transmission of Heat.—The table on p. 125 contains the results of some interesting experiments, in which one side of the plate or tube was exposed to hot steam and the other side to the atmosphere.

Other experiments will be found in the following publications:—

M. Burnat, 'C. E.,' 1874, vol. xli. Straw and similar coverings.

W. J. Bird, 'N. Engl. T.,' 1879, vol. xxix. p. 7; 1881, vol. xxxi p. 77; 1882, vol. xxxii. p. 35; 1885, vol. xxxv. p. 159; 1886, vol. xxxvii p. 13. Steam losses in pipes exposed to atmospheric influences.

J. J. Coleman, 'Phil. Soc.,' Glasgow, 1883, p. 73.

H. Collins, 'Brit. Assoc.,' 1891, p. 780.

J. M. Ordway ('Am. M. E.,' 1883, vol. v. p. 95, 1884, vol. vi.) gives relative heat transmission through 1 inch of 50 substances, but, as he changes his standard of comparison, it is difficult to analyse his results.

'C. E.,' 1891, vol. cviii. states that the loss of heat through covered steam pipes through which super-heated steam was passing is at the rate of 1° F. per $3\frac{1}{4}$ square ft. of surface.

Comparing these values with those obtained for the transmission of heat through tube surfaces, it will be noticed that the external surface of a boiler would seem to be relatively about ten times more efficient as a heat dissipater than are the very much thinner tubes as heat absorbers. This is of importance, because it may happen that a boiler is made so long that the last foot of tube length supplies less heat than is given away by the last foot of shell. Generally the entire circumference of the boiler shell is about one-fourth to one-sixth of the sum of the circumferences of the boiler tubes, and if this boiler is unlagged there would be no advantage in allowing the gases to escape at any temperature less than about 400° F. above that of the water in the boiler, or say 750° F. If properly lagged the gases may be cooled much lower.

In the above-mentioned experiments heat was supplied to the pipes or plates by steam, and the cooling took place in air. With

Material and Conditions	Difference of Temperature	Steam Condensed per Square Foot per Hour	B.T.U. transmitted per Hour per Square Foot per Degree of Difference	Observer
	°F.	Lbs.		
Locomotive boiler	213	1·611	7·5	Isherwood [1]
$\frac{5}{16}$-in. iron plate	...	...	2·93	'Franklin Inst.' [2]
„ „ covered with cow-hair $\frac{1}{4}$ in.	...	...	1·05	„ „
Do. do. $\frac{1}{2}$ in.	...	...	·57	„ „
„ „ $\frac{3}{4}$ in.	...	...	·41	„ „
„ „ 1 in.	...	...	·31	„ „
„ „ $1\frac{1}{4}$ in.	...	...	·27	„ „
„ „ $1\frac{1}{2}$ in.	...	...	·25	„ „
Cast-iron pipe	...	·71	4·75	Meunier [3]
Wrought-iron pipe	...	·80	5·32	„
Copper pipe	...	·575	3·83	„
Cast-iron pipe	226	1·04	6·27	Meunier [4]
Wrought-iron pipe	222	1·04	6·42	„
Copper pipe	227	·828	4·96	„
Lagged cast-iron pipe	234	·465	2·67	„
„ wrought-iron pipe	217	·405	2·58	„
„ copper pipe	229	·541	3·20	„
Ribbed cast-iron pipe	203	1·84	8·75	E. Deny [5]
„ „ „	210	...	1·04	H. Fischer [6]
Cast-iron pipe	360	1·169	3·13	N. N.[7]
„ „	307	0·937	2·94	„
Do. and 2 in. papier mâché composition	360	0·294	0·79	„
„ and $2\frac{3}{4}$ in. hair-felt	360	0·216	0·58	„
„ and 2 in. Kieselguhr, 10% „ binding material	360	0·226	0·61	„
„ „ „ „	307	0·166	0·52	„
„ and $2\frac{3}{4}$ in. hair-felt	360	0·188	0·50	„
„ „ „ „	307	0·125	0·39	„
„ $\frac{3}{4}$ in. cork strips 2 layers canvas	360	0·198	0·53	„
„ „ „ „	307	0·152	0·48	„
Cast-iron pipe, Strong S.W. wind	250	1·19	4·60	D. K. Clark [8]
Do. East wind	247	0·732	2·85	„
„ average	250	1·00	3·86	„
„ and $1\frac{1}{2}$ in. Haake composition	„	0·230	0·89	„
„ 1 in. Haake composition	„	0·267	1·03	„
„ 1 in. Eagle composition	„	0·288	1·11	„
„ 1 in. Leroy composition	„	0·310	1·20	„
„ 1 in. Keenan composition	„	0·317	1·22	„
„ 1 in. Reid McFarlane composition	„	0·340	1·31	„
„ 1 in. McIvor composition	„	0·367	1·42	„
„ 1 in. Sutcliffe composition	„	0·374	1·45	„
Pipe. Bare	350	1·516	4·2	Capper [9]
„ 1·6 in. asbestos composition	„	·420	1·16	„
„ 1·25 in. Leroy composition	„	·399	1·10	„
„ 1·4 in. magnesia	„	·208	0·57	„
„ 1·6 in. mica flakes	„	·177	0·49	„

the exception of Miss Bryant's experiments, no direct measurements have yet been carried out to ascertain whether the chief resistance is encountered on the hot or cold side. This could best be done by repeating A. F. Yarrow's experiments on the curvature of heated plates and estimating the mean temperature of the plate by means of its lineal expansion. Some light might be thrown on the subject by comparing the previous experiments with the following (W. S. Hutton, 1887, p. 253). The units of heat which a $\frac{1}{8}$-in. plate will transmit per square foot per hour, if supplied with an unlimited amount of water on one side and steam on the other, are there given :—

Cast iron	265 units	Phosphor bronze	162 units
Wrought iron	252 „	Copper	155 „
Steel	246 „	Tin plate	142 „
White metal	207 „	Glass plate	259 „
Brass Plates	175 „	Tiles	246 „
Gun metal	168 „		

An interesting feature of W. S. Hutton's figures is that they show that the transmission of heat is less dependent on the conductivity of the material than on its other properties. Thus tiles and glass plates are nearly as good as cast iron, while copper and tin plates are the least efficient.

Smoke-box Radiation.—Marine boiler smoke-boxes are usually fitted with baffle plates, one inside and one out. The only advantage they offer is cheapness combined with a reduction of radiated heat; but heat lost by convection must be greater than with a plain plate, for a natural and very powerful current of air or hot gases is induced between the plates. To reduce the loss of heat from smoke-boxes the space between the baffle plates should be filled with non-conducting material, or if this is too heavy or too costly, the edges of the baffle plates should be flanged so as to press against the smoke-box plates. Only one baffle plate will then be needed—an inside one. If two are fitted, they should both be flanged and both fitted on the hottest side, so as to protect the casing and door from heat and thus prevent its warping. With water-tube boilers loss of heat, warping of the casing, and then leakage of air make themselves seriously felt.

From the foregoing it is evident that our knowledge of the transmission of heat is very limited, and possibly incorrect; nevertheless the collection of previous experiments into one chapter has the advantage of showing in what direction further information should be sought, and the following experiments readily suggest themselves :—

Notes to Table on p. 125

[1] Isherwood, *Experimental Researches*, vol. ii.
[2] *Franklin Inst.*, 1878, iii. vol. lxxv. p. 153.
[3] W. Meunier, *Rev. Ind.*, 1884.
[4] Ibid., *Soc. I. Mul.*, 1879, p. 730.
[5] E. Deny, ibid. 1883, vol. liii. p. 575, and 1834, vol. lv. p. 15.
[6] *Dingler's J.*, 1878, vol. ccxxviii. p. 1.
[7] N. N., *C. E.*, 1895, vol. cxxi. p. 300.
[8] D. K. Clark, *Engr.*, 1884, vol. lvii. p. 65.
[9] Prof. Capper, Report, 8 September, 1898.

1st. Experiments on the flexure of plates which are being heated on one side either by water, air, or radiant heat, and are being cooled on the other side by any of these methods.

2nd. A repetition of M. Geoffroy's experiments on a subdivided boiler, with accurate measurements of the temperatures of the waste products of combustion at various points, and also analyses of the gases, and calorific determinations of the fuels.

Temperature of Fire Bars.—The parts of the boiler which suffer most from the effects of slowness of transmission of heat are the fire bars. They are exposed to an intense heat at their upper surface, to radiation on part of their side surface, and the only available means of cooling them is by the air which passes over their sides. When the fires are thick, when much air is admitted above the bars, and when the draught is forced, the bars are naturally exposed to the very serious danger of overheating.

Assuming that the heat received by the upper surface of the bars is at the rate of 50,000 thermal units per hour per square foot, and assuming that this heat is being transmitted to the passing air at the high rate of ten units for every degree of difference of temperature, then if a is the width of the air space, h the depth of the fire bar, b its breadth, and t its excess temperature over that of air, we have

$$50000 \,.\, (b + a) = 10 \,.\, t \,.\, (b + 2h),$$

$$t = \frac{5000 \,.\, (b + a)}{(b + 2h)}.$$

In order to keep the temperature t below 1,000° F. it would be necessary to make $h = 2 \,.\, b + 2\tfrac{1}{2}\, a$, which is perhaps fairly near the truth; but this leaves out of account the quantity of air, which is certainly an important factor. With ordinary draught the air supply is at the rate of about 400 lbs. per hour per square foot of grate, or about 7 lbs. per minute, which is certainly not a large quantity. Comparing the temperatures t_1 and t_2 in two different sets of fire bars in which the various dimensions are a_1 and a_2, b_1 and b_2, h_1 and h_2, and the air supply Q_1 and Q_2, we have

$$\frac{t_1}{t_2} = \frac{b_1 + a_1 \,.\, (b_2 + 2h_2) \,.\, Q_2}{b_2 + a_2 \,.\, (b_1 + 2h_1) \,.\, Q_1}.$$

This would show that the more work a grate has got to do the deeper ought the bars to be made, and the greater should be their number. But as it is at first not so much the danger of burning the bars as of their being bent which has to be guarded against (see p. 10), making them very thin without reducing their length would increase this trouble. Until their temperatures have been taken under varying conditions, which is not difficult, it is idle to speculate any further on their behaviour, and at any rate the above formula should be looked upon more as an indication as to how much information is yet wanted than as a practical guide for new departures.

CHAPTER VI

STRENGTH OF MATERIALS

In this country, especially during the last twenty years, steel has almost entirely supplanted wrought iron for ship and boiler construction. No further excuse is therefore needed for discussing its peculiarities more exhaustively than those of the older metal. Besides, the two materials are so intimately related, that the large amount of research expended on one must assist in explaining the other.

Wrought Iron.—It is usually stated that steel is a homogeneous material, while wrought iron is a conglomerate of crystals, granules, or fibres cemented together by films of slag. Layers and threads of slag undoubtedly exist in iron, but it is unreasonable to suppose that they are as uniformly distributed as the above theory would make it appear. The manner of the production of wrought iron in the puddling furnace seems to be, that drops of molten pig are slowly converted into plastic wrought iron, their centres always retaining a slight excess of carbon and other impurities, while the flame and slag acting on their outer surfaces are converting them into almost pure iron, possessing, amongst other qualities, that of welding with the greatest ease. That these numerous drops of white-hot metal, which may now be called granules, or even lumps, should stick together is but natural, and the trouble to be expected—and it does exist—is that numerous cavities filled with slag will come into existence.

When drawn out into bars or plates wrought iron would therefore consist of numerous fibres whose individual outer surfaces are very pure and soft, and whose cores contain the small percentage of carbon which had been allowed to remain. Professor Wedding has shown how by the microscope we can distinguish between the hard and soft parts, for he found that the greater the percentage of carbon, the darker the colour if the iron or steel is raised to a blue heat ; and by carefully polishing and etching samples of iron, and then heating them sufficiently to make them appear of a uniform dark straw colour, the microscope will show that this uniformity is an illusion, and that the metal really consists of innumerable cells of soft iron surrounding hard cores.

Influence of Producing Temperatures.—Had the temperature of the puddling furnace been as great as that in any of the steel furnaces, each granule would have been melted, and the product would have been mild steel of the same tenacity as the puddled iron, but it is well known that the first of these metals requires quite twice as much horse-power for machining it as iron of the same tenacity, and there does not seem to be a better explanation than that this is due to the somewhat higher temperature at which it has been produced.

Steels from various makers, but of the same tenacity, are said to show great differences as regards the power required to chip them. An explanation for this might be sought for in variations in the casting temperatures, and these again would depend on the firebrick lining used.

J. W. Cabat ('American Inst. Mining Engs.,' vol. xiv. p. 85) deals with the influence of casting temperatures, and remarks that cold-blown charges of rail steel work better than hot-blown ones, and that open-hearth spring steel is similarly affected by the furnace temperature.

The Basic Bessemer Steel Process, using Phosphoric Pig Iron, and usually called the Thomas Gilchrist Process.—The pig is run into the converter in a molten state, and subjected from below to the action of a strong blast, which first removes the carbon and silicon, and then attacks the phosphorus. It was at one time believed that the development of heat in all the Bessemer processes was due to the burning of the carbon contained in the pig, but this has been disproved. Dr. F. C. G. Müller ('Deut. Ing.,' 1878, vol. vi. p. 387) mentions that silicon burns away first, and that for every per cent. consumed the temperature of the molten metal is raised 540° F.; that the carbon is not consumed until a temperature of 2,550° F. has been reached, and that its burning does not raise it. Phosphorus, like silicon, adds much heat to the bath, but will not burn until practically all the silicon and carbon have been consumed. It has also been found that the phosphorus cannot be consumed unless lime is present in the converter, and as this would attack and melt the ganister lining, which is nearly pure silicic acid, it is necessary to give the converters a *basic lining* (dolomite), from which the steel made by this process derives its name of '*basic steel.*' Only very little silicon may be tolerated in the pig intended for this process, as it attacks the dolomite, and in order to obtain a sufficiently high temperature, which cannot be done by burning carbon, phosphorus must be present in large proportions, viz. from 2 to 3%. In some pigs phosphorus is not sufficiently plentiful, and then, in order to obtain the right heat, it has to be melted in Siemens-Martin furnaces instead of cupolas.

During the process of manufacture the blast is kept up until all the phosphorus is consumed, and the metal in the converter is then almost absolutely pure iron. Spiegel and ferro-manganese are now added in the right proportions, and the charge is ready for casting.

At present the only available means for judging of the purity of the molten steel is to count the number of revolutions of the blowing engines from the time when the carbon lines in the spectrum have disappeared, but, in spite of assertions to the contrary, it does not appear that sufficient reliance can be placed on this proceeding, which is as follows :—

The composition of the pig iron is ascertained from samples if bought, or from daily returns if run direct from the blast furnace. The number of revolutions of the blowing engine required for supplying all the air necessary to consume all the carbon and silicon is estimated, and also how many extra revolutions will be required to remove the phosphorus. The weight of the charge, the temperature and moisture of the air, must necessarily affect the result somewhat; therefore the foreman first notes the number of revolutions up to the point when the spectroscope tells him that all the carbon has been burnt, and compares it with the estimated number. The difference, if any, is then applied to the calculated number of extra revolutions required for removing the phosphorus, and it depends upon the correctness of these several determinations whether the metal is pure or not. Even the temperature of the charge influences them, for the colder it is the more difficult is it to judge of the disappearance of the carbon lines; so that, generally speaking, it is very easy to be mistaken as to the time when the blast should stop. A danger to which the charge is also exposed is that due to the action of the highly phosphoric slag on the added spiegeleisen and ferro-manganese. It has been confidently stated, but not proved, that the affinity of carbon for oxygen, and of phosphorus for iron, is so great that, unless the latter stage of the process is hurried, phosphorus will leave the slag and re-enter the metal. This seems to be the reason why some works put the admixtures into the ladle and not into the converter. Where this is done the question arises whether the charge can then be thoroughly mixed. If not, this would explain occasional irregularities in the finished product, and these being more apparent in large plates than in bars and wires, may account for the strong dislike entertained towards basic Bessemer plates.

The Acid Bessemer Process differs from the one just described in so far as the lining of the converter is acid (silicic acid = ganister) and that no lime is added. The slag is generated by the combustion of the silicon (to silicic acid) and the iron (to iron oxide). Not a trace of phosphorus is removed by this process, and therefore the pig iron used must contain less than that to be allowed in the finished product. It must also contain a large percentage (2–3 %) of silicon for the production of sufficient heat.

The addition of spiegeleisen and ferro-manganese is effected just before casting, either in the converter or in the ladle.

The spectroscope enables the operator to judge, with reasonable accuracy, as to the percentage of carbon remaining. He could, therefore, either interrupt the blowing before all the carbon is burnt, which would save some of the costly admixtures, or he could wait till all the carbon has disappeared, and then reintroduce it with the necessary manganese. The latter of these two methods is the more reliable. An incidental difference between these processes is that in the basic process, during the necessary after-blow for removing the phosphorus, every trace of carbon disappears, while in the acid process this is not the case, so that by adding just sufficient ferro-manganese to remove all redshortness one would obtain a far weaker but also a more ductile material by the basic than by the acid process; the former is therefore used almost exclusively for producing the steel for soft wire.

The Open-Hearth Process was made available for producing mild steel only after Dr. Siemens had invented the regenerative chamber, and although the various shapes of the furnaces take the names of their respective designers, his name will always be associated with steel made by this process.

Heat regenerators consist of several vaults filled with loosely packed firebricks. Air is admitted through one of these chambers, and gas through another, and both are ignited in the furnace. The waste products are led to the chimney through the two other chambers, and heat them. The current is then reversed, and the air and the gas pass over the white-hot bricks of the latter chambers, and the waste products heat the former. The process is then continually repeated. Not only does this save nearly all the heat, which would have escaped with the hot gases, but the temperature of the furnace can easily be raised so high that even the firebrick lining would melt.

Within certain limits the temperatures can be regulated by admitting more or less excess air, and by altering the frequency of reversing the current. The flame might also be altered from an oxidising to a reducing one, but not without raising the temperature beyond the endurance of any firebrick. The gas used for firing these furnaces is produced at the works either by partly burning and partly distilling coal, in which case it consists of hydrocarbons, carbonic oxide, and nitrogen; or water gas is used, which contains carbonic oxide, hydrogen, and a smaller percentage of nitrogen. Recently part of the heat in the waste products has been used to distil the coal.

The Acid Siemens-Martin Process.—Pig iron is placed in the furnace, and when melted, iron ore and about 25 % scrap iron are added, until all the carbon and silicon are consumed, and the ore reduced to iron. Samples are repeatedly taken and tested mechanically, to judge of the condition of the bath, and when ready, spiegeleisen and ferro-manganese are added, and the charge is run. As in the Bessemer process, some of these additions may be saved by stopping the process of reduction at an early stage, and in some works the carbon is successfully reintroduced by adding it as powdered charcoal, or anthracite placed in the ladle or into the trough leading to it from the furnace.

During the early period of refining the various layers of the bath are of very different composition, as can easily be ascertained by taking a sample from the top or the bottom of the charge; but a natural mixing takes place, due to chemical action and to an evolution of gases. However, at the final stage, when the bath is nearly but not quite uniform, this action practically ceases. It is revived by the addition of iron ores, or of pig, or in some works by stirring with wooden poles, which evolve large quantities of gas.

The acid Siemens process does not remove either phosphorus or sulphur, and the pig and scrap used should therefore contain only traces of these impurities.

The Basic Siemens Process, using Pure Pig, is almost identical with the above, except that, on account of the use of a basic slag, almost every trace of phosphorus disappears. It is also found that both the carbon and manganese are very energetically attacked by the flame, and after the spiegel has been added, it is difficult to hit the right moment for running the charge. This difficulty is aggravated by the fact that the natural lowest limit of mild steel from these furnaces is below 20 tons, whereas with the acid Siemens furnace it is about 24 tons. When trying to make steel of 27 tons, a small error in the admixtures, or in the time of casting, will have two to three times as much effect in the one case as in the other. However, with care the very best material, and of the intended hardness, is obtained.

On account of the costliness of the basic lining this process is hurried as much as possible, and six or even seven charges are sometimes got out of a furnace in twenty-four hours. Of course this is only possible if the refining process is curtailed, and with this object in view not more than about 20 to 25 % of pig iron is used, while the rest is scrap. A further gain has been attempted by carrying out part of the process in a Bessemer converter where all the carbon and silicon are removed. The final reduction takes place in the basic Siemens furnace, where the last trace of phosphorus is abstracted.

'Basic Refined Steel.'—C. E. Stromeyer, 'Eng. Scot.,' 1897, vol. xli. p. 227, deals fully with this and the next material. E. Bertrand, 'I. and S. I.,' 1897, vol. i. p. 115.

The Basic Siemens Process, using Phosphoric Pig, is carried out at a few works in this country. It is a tedious one, lasting about fifteen hours per charge; this is made up of iron ores, lime, and about 75 to 80 % of pig, too poor in phosphorus for the basic converter and too rich for the acid Siemens process. The phosphoric acid which is generated is not volatile, and does not rise to the surface in bubbles like carbonic acid, and therefore cannot assist in mixing the charge, so that the process would be indefinitely prolonged if special means were not adopted, such as the occasional addition of pig and of iron ores, to produce gases. This is necessary up to the last stage, which, as in all other cases, consists in adding spiegeleisen and ferro-manganese.

The basic slag, which floats on the molten steel, is very thick, generally about 12 to 15 ins., and the furnace gases have little chance of acting on the metal. Few works use this process, and there all attempts to manufacture plates on which the same reliance can be placed as on the acid steel have as yet failed. With care good results could no doubt be obtained, but it seems that the effects of even slight carelessness on the part of the operator lead to bad results, and no tests have yet been devised which will detect them. Herr Knaut ('Stahl und Eisen,' 1900, vol. xx. p. 783) has put about 500 tons of this basic steel into boilers, and after ten years' experience condemns it.

The Puddling Process has been dying out fast. In it the pig iron is melted in the presence of iron ores and slags by means of flames, which can be changed from oxidising to reducing ones as occasion arises. The temperatures, being very much lower than in all the previous processes, appear to assist at removing a large percentage of phosphorus and sulphur. Much hard labour and considerable skill are required to obtain good results. The final operation consists in extracting the intermixed slag from the iron, which is done under a blooming hammer.

Crucible Steel, as its name implies, is produced in crucibles. These are filled with carefully weighed quantities of blister steel, pure iron, or scrap steel. It is said to be giving way to Siemens steel even for guns, and as it is a very costly process, and has never been used for boiler plates, it is unnecessary to enter into details, except to mention that when carrying out experiments on steel alloys it is of importance that the right fire-proof material should be used. Thus, when wishing to produce practically pure iron, basic crucibles must be employed; if a good percentage of carbon has to be retained, and for many alloys (manganese, silicon, aluminium), the melting must be done in plumbago, and for some other purposes acid crucibles are best, while for certain compositions the melting has to be carried out in various fire-resisting materials, and the molten metal mixed before casting.

The following books will be found to contain a very full account of the manufacture and properties of mild steel:—Dr. J. Percy, 'Iron and Steel,' London, 1864; M. H. Howe, 'The Metallurgy of Steel,' New York, 1890; Professor Ledebur, 'Handbuch für Eisenhüttenkunde,' Leipzig, 1884; V. Deshayes, 'Classement et Emploi des Aciers,' Paris, 1880; J. S. Jeans, 'Steel,' &c., 1890; Prof. A. Martens, 'Mitt. Berlin,' 1890, vol. ii.

The Influences of Impurities on the mechanical properties of mild steel.—It may be remarked that absolutely pure iron or steel has not yet been produced and experimented upon, and the following remarks, therefore, only apply to the effects of additions made to average qualities.

Recently the subject has shown itself to be more complicated than was at first assumed, but it is hoped that microscopic researches will elucidate the matter. Thus it is now known that carbon and iron combine in various proportions, forming compounds which are dissolved throughout the mass of steel. These compounds can be distinguished under the microscope and also by attacking the metal with various acids, when the gases given off or the residues enable one to estimate the amount of carbon in its different conditions. These compounds give varying qualities to steel. Phosphorus behaves in the same way, and instead of always producing coldshortness, it is now believed that only one of its ferric compounds does this, which when treated with certain acids evolves PH_3. The phosphide of iron, on the other hand, is not dissolved in the mass of metal, but exists in the form of minute crystals, which, being very hard, cause the edges of cutting tools to get blunt. This explains why some steels with much phosphorus are not coldshort. It has not yet been shown how the phosphides can be changed. Probably sulphur has the effect of dissolving the phosphide of iron, and therefore accentuating coldshortness. It is known that carbon can be affected by various impurities. Silicon, tin, and copper seem to drive it out of the iron. Manganese, chromium, and tungsten seem to assist at dissolving it, the two latter converting it into hardening carbon. For action of acids, see 'I. and S. I.,' 1888, p. 369, 1896, vol. i. p. 239, 1897, vol. i. p. 229; 'Comp. Rend.,' 1897, vol. cxxx. p. 148.

Hydrogen, even in small quantities, makes hard steels very brittle. It enters them when being pickled and can be driven out by heat.

Nitrogen increases the tenacity, but it shares with phosphorus the unenviable distinction of making steel unreliable. Neither element seems to affect the ductility as measured by mechanical tests, but when present in excess both elements are responsible for frequent failures of plates when in use. When the sum of the percentage of phosphorus plus five times the percentage of nitrogen (P + 5N) exceeds 0·08 % then trouble may be expected.

Nitrogen combines with iron under pressure, about 0·003 % under blast furnace conditions, and about 0·030 % under Bessemer converter conditions. Seeing that fluid Bessemer steel or large quantities of Bessemer scrap may be thrown into Basic Siemens' furnaces which use pure pig (see p. 132), this process may occasionally turn out unreliable steel. This may account for the relatively numerous steel plate failures in countries where basic steel is generally used.

Carbon increases the tenacity and tempering qualities (if it does not even create them); it reduces the elongation at the rate of about 30 % for every 1 % C., and the ductility and weldability and melting temperature. It increases liability to burn.

Boron increases tenacity, but only after tempering. It creates redshortness, but only at a white heat.

Silicon increases tenacity, reduces the elongation and melting temperature, prevents blowholes, increases red- and coldshortness, but only in presence of carbon. Reduces weldability, but only above 0·7 %.

Phosphorus has ascribed to it in company with nitrogen the chief blame for coldshortness and general treacherousness. It also increases the liability to get burnt. It reduces the melting temperature. In mild steel it should not exceed 0·07 %. In small quantities it increases corrosion, in large percentages it prevents it. It gives to fractures of nicked samples a stringy appearance. As it is believed to improve the machining qualities, phosphoric steels are now supplied for stays and other parts which are shaped by automatic tools. Up to 6·8 % it does not affect redshortness.

Arsenic reduces elongation and tempering qualities, increases coldshortness when above 0·17 %, but only slightly; 0·25 % prevents welding. In chemical analysis it has been mistaken for phosphorus. Like phosphorus and carbon, it reduces the melting temperature.

Bismuth causes red- and coldshortness.

Sulphur accentuates the bad effect of phosphorus, produces redshortness and greasiness as regards welding when it exceeds 0·06 %. In mild steel it should not exceed 0·07 %. It facilitates machining but segregates easily.

Manganese counteracts the hardening effects of carbon, and neutralises the redshortness of sulphur up to 0·1 %, and of silicon up to 0·5 %. In large quantities it is said to give a shaly appearance to fractures of torn or bent samples.

Nickel reduces tenacity and increases ductility, particularly as regards impact. It neutralises the influence of carbon, and perhaps of phosphorus, and it reduces corrosion.

Copper prevents blowholes, increases tenacity, reduces elongation, and produces redshortness, but only when above 3 %. Its influence on welding is doubtful; it reduces corrosion.

Chromium, like **Tungsten**, intensifies all the influences of carbon, particularly as regards the tempering qualities.

Aluminium prevents blowholes, reduces the melting temperature, and hardens cast iron, quiets the molten metal, and accentuates the effects of carbon.

Tin.—Authorities are conflicting. It seems to drive carbon out of cast iron, and 0·35°/$_\circ$ makes mild steel quite redshort. It enters steel as scrap tinplates.

Oxygen is not easily determined, and it cannot be said that it exists in iron either as dissolved or segregated oxide. The small traces of hydrogen which are always present in steel should effect a reduction when the steel is in the heating furnace, in the same way that hydrogen in a laboratory furnace is expected to reduce the oxides if heated to redness.

An attempt to summarise these remarks is made in the following table:—

Elements	C	P	S	As	Si	Sn	Al	Mn	Ni	Cr	Cu	N	H	B
Tenacity	+	+	+	0	+?	...	0	+	+	+	+	+	...	+
Elongation	−	−	0	−	−	...	−	+	−	−	−	−	...	0
Resistance to impact	−	−	...	−	−	...	+?	+	+	+	...	−	...	...
Pliability—cold	−	−	−	−	...	...	+	+	−	−	...	−	−	0
„ dull red	−	0	0	...	−	−	+	+	+	...	−?	...	...	0
„ white hot	−	−	−	0	−?	−	...	+	...	...	−	...	...	+
„ tempered	−	...	...	−	−	...	...	+	+	+	0	...	...	−
Weldability	−	?	−	−	−	−	...	0	−	...	?	...	...	−
Melting temperature	−	−	−	−	...	...	−	...	...	+	−	...	...	−
Corrodibility	?	−	...	...	...	...	...	+?	−	...	−	...	...	...
Tendency to burn	+	+	...	...	...	...	...	...	...	...	...	...	...	...

NOTE.—0 means that the property is not changed, + that it is increased, − that it is diminished by the impurity : ? means that the authorities are very conflicting ; ... means that no information

ADDENDUM.

For insertion after the table on page 135.

The ultimate tenacity in tons per square inch can, according to recent tests, be estimated to within 5 per cent. of its true value by inserting the percentages of the several chemical elements in the formula

$$19{\cdot}75 + 25\,(C + C^2) + 11{\cdot}5\,Si + 30\,P + 205\,N + 36\,AS - 9{\cdot}5\,S.$$

No definite relationship has yet been discovered for the elongation and other properties.

years suddenly developed serious cracks. On analysis it was found that its percentage of nitrogen was 0·0126, and the phosphorus 0·147°/$_\circ$, the sum 1P + 5N being 0·210 which is above the empyrical limit of 0·08°/$_\circ$. On enquiry it transpired that the front end plate and first ring had once figured in an exhibition and had been bought up by a boilermaker, who added the main length to this front portion, which had evidently been made of cheap Bessemer steel ; hence the high percentage of nitrogen in the first belt.

A vessel for holding compressed gas exploded at Innsbruck. The analysis showed that it contained 0·0120°/$_\circ$ nitrogen and 0·077°/$_\circ$ phosphorus, the sum 1P + 5N being 0·137°/$_\circ$.

A plate and an angle iron used in the construction of an open tank in which caustic liquor was concentrated cracked while in use, whereas older tanks remained intact. The analysis showed respectively 0·0062 failures of plates when in use. When the sum of the percentage of phosphorus plus five times the percentage of nitrogen (P + 5N) exceeds 0·08% then trouble may be expected.

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An attempt to summarise these remarks is made in the following table:—

Elements	C	P	S	As	Si	Sn	Al	Mn	Ni	Cr	Cu	N	H	B
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Resistance to impact	−	−	...	−	−	...	+?	+	+	+	...	−	...	...
Pliability—cold	−	−	−	−	...	...	+	+	−	−	...	−	−	0
" dull red	−	0	0	...	−	−	+	+	+	...	−?	...	...	0
" white hot	−	−	−	0	−?	−	...	+	...	...	−	...	...	+
" tempered	−	...	...	−	−	...	...	+	+	+	0	...	...	−
Weldability	−	?	−	−	−	−	...	0	−	...	?	...	...	−
Melting temperature	−	−	−	−	...	...	−	...	...	+	−	...	...	−
Corrodibility	?	−	...	...	...	...	...	+?	−	...	−	...	...	...
Tendency to burn	+	+	...	...	...	...	...	...	...	...	...	...	...	...

NOTE.—0 means that the property is not changed, + that it is increased, − that it is diminished by the impurity: ? means that the authorities are very conflicting; ... means that no information could be obtained.

Recent failures of steel due to impurities, and analysed in the laboratory of the Manchester Steam Users' Association, Manchester:—

Some Continental angle irons intended to be used in small portable locomotive boilers failed in the boiler shop. The analysis showed that this steel contained 0·0146°/$_{\circ}$ nitrogen, which fully accounts for the brittleness.

A furnace plate which had been giving trouble was being cut out, and was then found to be quite brittle; it contained 0·0180°/$_{\circ}$ nitrogen.

A ship was being built in a Continental port with steel plates which had satisfactorily withstood the usual tests, but these plates and angles broke up during construction in a most unaccountable manner. A sample of this steel was found to contain 0·0230°/$_{\circ}$ nitrogen, which is a maximum record.

The first ring of a shell of a boiler which had been in use for some years suddenly developed serious cracks. On analysis it was found that its percentage of nitrogen was 0·0126, and the phosphorus 0·147°/$_{\circ}$, the sum 1P + 5N being 0·210 which is above the empyrical limit of 0·08°/$_{\circ}$. On enquiry it transpired that the front end plate and first ring had once figured in an exhibition and had been bought up by a boilermaker, who added the main length to this front portion, which had evidently been made of cheap Bessemer steel; hence the high percentage of nitrogen in the first belt.

A vessel for holding compressed gas exploded at Innsbruck. The analysis showed that it contained 0·0120°/$_{\circ}$ nitrogen and 0·077°/$_{\circ}$ phosphorus, the sum 1P + 5N being 0·137°/$_{\circ}$.

A plate and an angle iron used in the construction of an open tank in which caustic liquor was concentrated cracked while in use, whereas older tanks remained intact. The analysis showed respectively 0·0062 and 0·0070°/₀ nitrogen, and 0·165 and 0·261°/₀ phosphorus, the sum 1P + 5N being respectively 0·196 and 0·296°/₀, which fully accounts for the brittleness, no matter whether the liquor was weak or strong. Drillings of another troublesome tank gave 0·006°/₀ nitrogen and 0·246°/₀ phosphorus: 1P + 5N = 0·276°/₀.

A shell plate of a fairly modern high-pressure boiler developed a circumferential crack outside the lap joint, which subsequently extended in a diagonal direction although covered with a patch. The analysis showed that the nitrogen and phosphorus percentages were respectively 0·0037 and 0·088, the sum 1P + 5N being 0·106°/₀.

The connecting rod bolt of an engine broke. In this material the percentages were 0·0091 nitrogen, and 0·050 phosphorus, 1P + 5N = 0·095°/₀.

The lower parts of the shell plate of a recently constructed boiler cracked where the scale, which falls from the furnaces, occasionally accumalates, and where therefore the plates may have become very hot. This material contained 0·0084°/₀ nitrogen, and 0·060°/₀ phosphorus, the sum 1P + 5N being 0·102°/₀.

Some furnace rivets in a recently constructed boiler gave trouble by snapping in two. This material contained 0·0151°/₀ nitrogen, and 0·163°/₀ phosphorus, the sum 1P + 5N being 0·238°/₀.

A marine boiler combustion-chamber plate which had cracked along the line of rivet holes showed 0·0041°/₀ nitrogen, and 0·092°/₀ phosphorus, the sum 1P + 5N being 0·112°/₀.

In a recently built marine boiler some rivet heads dropped off. On analysis they were found to contain 0·0036°/₀ nitrogen and 0·061°/₀ phosphorus. The sum 1P + 5N = 0·079°/₀ is just on the above-mentioned limit between good and bad steel.

Mention is made in "The Engineer" of the s.s. "Pahud," whose boiler exploded while under steam. No material was obtainable from this boiler, but the boiler of a sister ship, in which the butt straps leaked, and in which the shell plate was found to be dangerously cracked, was cut up and analysed. The results are as follows: nitrogen, 0·0060°/₀; phosphorus, 0·064°/₀; the sum 1P + 5N being 0·094°/₀

These last-mentioned failures are of special importance, because they are the only cases of steel marine boilers having exploded, although thousands have been built annually during the last thirty years. These exceptions may be looked upon as proving the rule that the process of manufacture (acid open-hearth steel) combined with the customary tests, ensure almost absolute safety. As this boiler was built on the Continent, it is possible that the steel was made by the basic open-hearth process, with which it is always possible to use up basic scrap, and that may contain much nitrogen It is therefore doubly desirable when using basic steel to analyse for nitrogen.

Mechanical Tests are called 'Qualitätsproben' in German, which word expresses very appropriately that they are mainly intended to determine the quality, and not, as is too often assumed in this country, that they should be an imitation of the conditions to which the material is to be subjected in the structure. Looked at in this light there is nothing unreasonable in bending a sample which has been cut from a plate which will never be subject to bending stresses, or to test a sample by tension, when the plate will certainly only be subjected to compression stresses. The object of the various tests is to ascertain the quality, and that is all, the correlation between a test and the quality best suited for a particular purpose having been determined by practical experience.

Thus, when iron came into use it was found that the better qualities—those which did not break with ordinary usage—could under the conditions of testing stand bending through a certain angle, and when fractured showed certain peculiarities of grain called fibre. Tests on this line, of a more or less severe nature, were then looked upon as a criterion of the quality. Later on tensile tests were added. In the early days of steel it was found that these tests did not suffice; that plates cracked without apparent cause when in use, even though samples which had been cut from them had withstood all the tests prescribed for the best iron in a more than satisfactory degree. It was found that this material was not reliable unless it would pass the temper-bending test, which was then adopted. As already mentioned, nitrogen and phosphorus impart treacherous qualities to steel, but no mechanical tests have yet been discovered which will detect such steels. The author is carrying out an elaborate comparison of all known tests, but has not discovered any correlation between them.

The following are the most important of these mechanical quality tests.

Cold Bending.—This test must be looked upon as a very important one, and more certain to detect a defective plate than either the temper-bending or tensile tests. To punch holes in the samples before bending does not appear to give a better indication as to the natural quality of the steel than if the sample is bent with sheared edges; but care should be taken that the shearing is done from opposite sides of the plate, and the two sharp edges should be kept on the inside of the bend; otherwise the results will be very erratic and unsatisfactory, depending not so much on the injury to the metal which has been caused by the shearing as on the shape of the ragged edge. If a tear commences from the sharp edge, it should not be looked upon as being very serious. There seems to be a difference, depending on whether the concave or convex side of the sample was the upper one while the ingot was heated for rolling, but nothing reliable is known about this subject.

Inferior qualities of steel will not behave well, and occasionally it happens that a strong steel bends better than a weak one. The bending of samples after annealing them is valueless, unless they crack even then, which would be an unmistakable sign that the material is quite bad. The thickness of the plate is an important factor and when 1 in. is exceeded only the best steel having sheared edges will stand even the slightest bending. For $\frac{3}{4}$-in. plates and upwards it is well to plane off $\frac{1}{2}$ in. of the edges, but then the samples should bend quite double. Care must be taken that the temperature is a reasonable one—say, 82° F. or 100° F.—for excessive cold has a bad influence, as will be seen from the following experiments. (See also p. 145.)

Out of ten samples cut from one $\frac{3}{8}$-in. plate six bent double at 82° F., while the four others broke at angles varying from 90° to 100°. These had been bent at 0° F. (i.e. 32° below freezing).

Two sets of samples of $\frac{1}{2}$-in. plates were cut from six separate charges. The first three were bent at 82° F. to $\frac{5}{8}$-in. radius, while at 0° F. the duplicate samples broke at $1\frac{1}{2}$-in. radius. Three others were bent at 82° F. and broke at 1-in. radius; their duplicates, tested at 0° F., broke off short. There was nothing in the analysis to indicate that the two qualities were different.

This cold-bending test is the only one which seems likely to detect whether the steel possesses the disagreeable quality of reverting to its (perhaps) naturally bad condition. When this is feared duplicate samples with sheared edges should be kept for a week or more, bent, and their curvatures compared with the original ones. Both samples should be bent till they crack, and at identical temperatures.

The Temper-Bending Test was devised to guard against the employment of steel liable to crack after it had been worked. Combined with the condition that the tenacity might reach thirty-two tons, it compelled manufacturers to replace part of the carbon in the steel by manganese, thereby maintaining the tenacity, but reducing the tempering qualities (see p. 135). The test is carried out as follows: Steel samples are heated to a cherry-red, and suddenly cooled in water of 82° F. (=28° C.) When cold they are bent. Better bending results

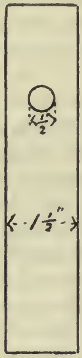


FIG. 112

FIG. 113

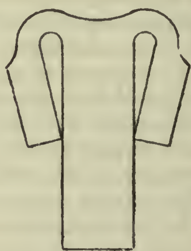


FIG. 114

are always obtained if a large batch of samples is operated upon, for if they are all thrown into the tank together their mass sufficiently retards the cooling and hardening effect of the water, while if thrown in singly none except the first few are exposed to the very cold water. Should there be any doubt about the quality of particular charges, it would not be difficult to retain such samples to the last, when the water has grown warm.

Another means of improving the bending results is to take the samples out of the water trough while still warm, and bend them in this condition. The effect of quenching the samples from a dull red heat seems to be a bad one, but the reverse is true if they are annealed thoroughly at a good red heat for a quarter of an hour, and then cooled slowly to a cherry-red before quenching. The use of a bending press instead of a hammer does not seem to be an advantage; in fact, experience points the other way, and an explanation may be sought for in the fact that most bending presses set up a greater tension stress in the outer fibres than is the case with a steam hammer. The removal of sheared edges before tempering—at least as regards the best qualities of steel—seems to be injurious. J. Riley arrived at a different conclusion ('I. and S. I., 1887, p. 121).

The Red-hot Bending Test is sometimes carried out on plates to be flanged, but seems to be unnecessary, except for iron. With steel this or a similar test is carried out on the baby ingot which is cast during the running of the charge, so as to be sure that the steel will roll well.

For mild steel plates the following test is occasionally adopted :—

While red hot, a half-inch hole is punched in a strip of metal about 1½ in. wide (see fig. 112), and is then drifted to 1 in. diameter and the end slit open, as shown in fig. 113 ; the two arms are then bent as shown in fig. 114, and the drifted surface should show no signs of cracks. This work is done in two heats.

The Drift Test consists in forcing a cylindrical cone into a drilled hole alternately from one side and then from the other. It is sometimes applied to steel intended for tube plates. It is a tedious and very uncertain test, and gives no indication of the quality of the material. This was very conclusively shown in W. Hackney's paper on test pieces ('C. E., 1883, vol. lxxv. p. 70). Locomotive tube plates are sometimes tested in this way.

Alternate Bending Tests are very tedious, and have to be carried out with much care, but then they are most valuable, as an indication of the ductility of the material. These tests are much used for wire, and they have given good comparative results in the author's investigations on blue heat. The plan there adopted was to bend the samples alternately from one side to the other through an angle of 45° over a radius of 1½ in. Professor Arnold has devised a machine which very rapidly bends samples many times through small but definite angles. Professor Wöhler devised a fatigue test which will be mentioned later, p. 160.

Bending Tests with Nicked Samples.—This test is a very valuable one if applied to iron, as it exposes the grain. With good steel the result is unsatisfactory, because it is difficult, even after nicking, to break it. In order to obtain a fracture the sample should be nicked all round, or bent through an angle of about 30° before nicking, and then broken. Fine grain indicates high carbon, coarse grain as well as silky fracture indicates low carbon. An excess of manganese is said to give a shaly and even a laminated fracture, and phosphorus in nicked samples gives a well recognisable stringy fracture. The shape of a fracture depends not only on the quality of the material, but also on the method of breaking; if done slowly fibre usually appears. Sudden blows produce crystals; their size increases with the softness of the material. It is a mistake to imagine that these crystals exist in the material; they are the shape of the fractured surface. The starting-point of most fractures can easily be found.

Samples Nicked on Edge.—This has proved itself to be a very discriminating test in the author's experiments on the ageing qualities of steel. The samples were planed on their edges, and nicked by means of a cold chisel, which penetrated to a definite depth, about

$\frac{1}{16}$ in., and then bent. The roughness near the nicked edge was filed flat, and after bending the curvature of this side, which was innermost, was measured near the nick with metal templates.

Percussion Bending Test is customary with rails and with tyres. The comparatively thin plates of a boiler are nearly always subjected to it as they are struck with a hammer, and, as already mentioned, the results are, if possible, more satisfactory than when the bending is done in a press.

Brinell's Method of Testing consists in placing a hard steel ball of, say, 10 mm. diameter on the polished surface of the sample, and loading and measuring the diameter of the impression under a microscope. The estimated depressions approximate very closely to the load applied and inversely to the tenacity of the material.

Tensile Tests.—Much importance is attached to this test. The ease with which it can be explained and the uniformity now obtained are powerful inducements for retaining it. It is useful for making comparisons of quality independent of tenacity, though in this respect it is not nearly as valuable as some of the bending tests; they, however, require observers possessed of much experience. The primary object of the tensile test is to ascertain the ultimate strength of the material; but the permanent elongations also give indications of the ductility, while with accurate instruments the modulus and limit of elasticity can also be ascertained. No relation between the contraction of area and general qualities has been detected, except that basic steel contracts more with equal elongation than acid steel. The temperature has a marked influence (Dr. J. Kollmann, 'Ver. Gew.,' 1880, vol. lix.).

A good deal has been written on the forms of test pieces, and the influence of length, breadth, and thickness, and on the presence and absence of scale on the surface—E. J. Reed, 1869, p. 401; P. D. Bennett, 'M. E.,' 1886, p. 27; E. Richards, 'I. and S. I.,' 1882, p. 11; Dr. H. Zimmermann, 'C. E.,' 1883, vol. lxxiv. p. 301; W. Hackney, 'C. E.,' 1883, vol. lxxv. pp. 70–159; D. Kirkaldy (various); J. Bauschinger, 'Mitt. Munich,' 1892, vol. xxi.

The general conclusions are, that long, thin test pieces show less strength and do not elongate nearly as much as short ones, but this is true only for materials which contract at the point of fracture. Mild steel contracts very much, so that results as to elongation would be misleading unless the length of the sample is mentioned. The general practices as to length are—

Board of Trade, 10 ins.
Admiralty, Lloyd's, and Continent, 8 ins., or 200 mm.
Artillery experts, Whitworth, and railway tyres } about 2 ins.

This short length of 2 ins. is due to the want of dimensions in the gun barrels. The removal of the scale reduces the elongation and increases the tenacity.

The Testing Machines all work on the principle that the sample is attached by one end to a weighted lever, and by the other end either to a screw or to a press which can be adjusted to make up for the elongation. With the older machines the screws can be worked only after the load has been removed, and the testing of ductile materials cannot be carried out in one operation, but this does not seem to affect the results.

In some machines the samples are placed horizontally, in others vertically, and the levers are either simple or compound, and the loading is done either by a jockey weight travelling on a lever, by weights added to the extremity of the lever, or by a fluid pressure or a pendulum weight.

The oldest machines are worked by added weights. On the Continent (chiefly Austria) water is run into a large bucket at the end of the lever. Generally a weighing machine is permanently stationed under the bucket, so that it can easily be weighed, but a gauge glass is also attached.

In England single levers and jockey weights are now customary. The stretch is taken up by steam power acting on a ram, and the speed is about 3 inches stretch per minute.

At some German works the lever is a loaded pendulum, which rises while a hydraulic ram stretches the sample.

In France and Belgium the lever is a crank, which presses on an iron disc resting on a sheet of leather, which covers a basin filled with mercury, and which communicates with a graduated vertical glass tube. The total load can be read off on a scale. This arrangement gives very accurate results, and can with advantage be used for ascertaining the drop (or elastic limit); but it is unsatisfactory in cases of dispute, because the column of mercury sinks back to zero at once when the test piece is broken. The defect could be remedied, or by attaching an ordinary steam indicator automatic diagrams could be obtained.

Descriptions and illustrations of testing machines will be found in the following publications: J. H. Wicksteed, 'M. E.,' 1882, p. 384, and 1886, p. 27; Prof. A. B. W. Kennedy, 'C. E.,' 1887, vol. lxxxviii. p. 1; U. R. Towne, 'M. E.,' 1888, pp. 206, 448. 'Engineering,' vol. xxxi. p. 57; vol. xxxiv. p. 254; vol. xxxv. p. 346; vol. xxxvi. p. 146; vol. xli. p. 180; vol. xliii. pp. 414, 572; vol. xliv. pp. 649, 652; vol. xlv. p. 458, &c.; vol. xlvi. p. 21.

Many testing machines are so arranged that they can be used for measuring compression-bending and torsion stresses. Appliances for bending samples can be attached to almost any machine, but that is not the case as regards compression and torsion. In the former case very careful adjustments are necessary, in order that the thrust may be perfectly central and the bearing perfectly normal. In both respects most machines are very imperfect. Almost any lathe can be used for the torsion test. It is a valuable one, but not carried out often enough, and then the lessons it teaches are not properly understood (see p. 157).

Strain Indicators.—When the elastic limit has to be ascertained strain indicators are indispensable. A description of these will be found in 'C. E.,' 1887, vol. lxxxviii. p. 1. The principle on which they are designed is the accurate measuring of the elongation or contraction of a test piece when loaded, but these changes of length are so minute that they have to be magnified at least 200 fold, better still 1,000 fold. This can be done with the help of microscopes; but they are very expensive instruments, and although accurate, are very inconvenient to work with. Micrometer screws have also been used, but unless they make and break contact electrically, they do not seem to be very accurate; besides, they are liable to have their threads stripped or damaged. Most of the strain indicators in practical use consist of some lever arrangement, and this always involves friction, which masks the effect of after-strains and may even show permanent sets before they occur. Rollers can of course be used instead of levers, but they are also not free from friction. The author has been very successful with rolling pins consisting of hard steel wires of diameters ranging down to $\frac{1}{100}$ in., to which are attached light balanced, straw pointers of 3 in. length; a magnification of 300 is obtained, which for many purposes is ample. The author has also adapted the phenomena of interference of monochromatic light waves to the measurement of strains. With such instruments a movement of one millionth inch can be detected; this is the elongation which occurs in one inch of steel when a stress of 30 lbs. per square in. is applied. The instrument is the only one that seems to have given satisfaction when used for the direct measurement of Poisson's Ratio. (C. E. Stromeyer, 'Proceedings,' 1894, vol. lv. p. 377.)

When using strain indicators one cannot be too careful to have the pull quite central in the test piece. Thus if in a test piece of $\frac{1}{2}$ in. thickness the pull falls $\frac{1}{100}$ in. to one side of the central line, or if the test piece is bent $\frac{1}{100}$ in., then a strain indicator placed on one side will give readings which are 6 per cent. too high or too low. For this reason wedges should not be used for holding test pieces when these delicate instruments are to be used; in fact, whatever the mode of attachment, the correct arrangement is to surround the test piece with three identical strain indicators. Their readings will tell one not only whether the pull is central or not, but also what the stress is at any particular section. This is of great importance when investigating limits of elasticity, for then it is not the average stress, but the maximum stress which is required to be known.

Speed of Testing does not seem to affect the final result, at least not unless the speed is excessively great, as in percussion tests, or unless the test is prolonged for hours or days. See J. Bauschinger, 'Mitt. München,' 1896, vol. xx.

Percussive Tensile Tests are carried out by securing the top end of a tensile test piece to a strong but narrow beam, and attaching a cross-bar to its lower end. A heavy weight, whose lower end is forked, is then dropped on it, striking the cross-bar. Another plan is to attach the weight to the lower end of the test piece and a cross-bar to its top end; the whole is then raised and dropped; the cross-bar is arrested by stops, while the weight tears the sample asunder. This is the more usual plan, and seems to be the better of the two.

Colonel Maitland ('C. E.,' 1887, vol. lxxxix. p. 114) was able to obtain a still more sudden rupture by shaping the test piece somewhat like a dumb-bell, surrounding the central, thinner part with gun cotton, and inserting both into a strong tube open at both ends. On firing the explosive, the two ends were driven out with great violence in opposite directions. It was found that during rupture the samples had elongated very much. This, however, might be due to combined action of the longitudinal pull and the surface pressure on the bar, acting like the forces which come into play when drawing wire (see p. 160).

Chipping, Machining, and Scratching have repeatedly been proposed for use, as they seem capable of giving valuable information.

Thus the turning tool of a lathe will show up very distinctly the various slabs and layers in an iron bar, exposing, as it were, very slight differences in the hardness or toughness of the material. With a hammer and chisel there is no difficulty in distinguishing between iron and steel, and some boiler-smiths profess to be able to tell, with the help of these tools, where a particular piece of steel has been manufactured. This test has not yet been made practical; the same may be said of *etching* and the *microscope*, and *magnetic tests*.

Micro Structure of Steel.—The sample to be examined should be filed flat, rough ground, and then polished. The rough and fine grinding can be done on carborundum discs of medium and finest grain. If the edges of the samples are to remain flat, the polishing should be done with carefully washed rouge on a hard surface, if necessary on pitch, but this substance has a tendency to tear small specs out of the steel. If the corners of the sample be round, the polishing may be done on parchment or cloth. If the polishing on cloth even without rouge be continued too long, a micro structure appears, which is called polish attack.

Heating the Samples in a Vacuum reveals the grain very clearly.

Heating in Air to straw colour or purple produces beautiful patterns of yellow, purple, and blue patches, but as yet no meaning attaches to these colours, though it is probable that they are in some way related to the angles at which the iron crystals are cut.

Annealed steels are conglomerations of **Pearlite** and **Ferrite.** If steel be heated to redness (700° C. = 1290° F.) and quenched in water, **Martensite** appears as needles which are coloured yellow to black when etched, according to the percentage of carbon. The needles in high carbon steels are shorter than in the low carbon ones. If steel be heated to 690° C. = (1270° F.) and quenched in water **Troostite** is formed. It consists of Martensite and Ferrite, and has a marbled structure.

Ferrite is iron free from carbon.

Cementite is a compound of iron and carbon, Fe_3C.

Pearlite is an intimate striated mixture of the two, containing about 0·9°/$_{\circ}$ carbon. The structure varies considerably in different steels, and when of a shelly or ropy structure this is supposed to indicate the presence of phosphorus.

Phosphorus segregations are revealed by etching the steel with an 8°/$_{\circ}$ solution of copper sulphate of ammonia, and gently washing off the deposited copper. The regions rich in phosphorus will appear dark, but only if the percentage of carbon is low. The darkenings are not strictly proportional to the percentage of phosphorus, but seem to be affected by other impurities.

Sulphur Segregations are revealed, but rather roughly, by Baumann's method. After fine grinding the steel surface place it for five minutes on a piece of photographic bromide paper which has been soaked in a 5°/$_{\circ}$ solution of sulphuric acid for five minutes. Wash the paper and fix in hypo. The sulphuretted hydrogen which is produced by the acid wherever there are minute specs of sulphide in the steel blackens the silver salts locally. The action is very rapid, and great care should be taken not to slide the sample on the paper. This test is affected by phosphorus. Dr. Heyn recommends the use of silk saturated with chloride of mercury.

Etching Fluids.—A very convenient etching fluid is picric acid crystals dissolved in pure alcohol. The sample should remain in this fluid for a few minutes, and slightly rocked, otherwise the edges will be attacked more severely than the central portions ; it should then be removed, washed, but not touched, dipped into alcohol, and when dry examined under the microscope. Irregular dark patches of pearlite will have made their appearance, which should be studied, and the sample should, if desired, be repeatedly etched until fine hair lines, the boundaries between the ferrite nodules, appear. Oval and long dove-coloured patches may also appear ; they are sulphide of manganese.

A 1°/₀ solution of nitric acid produces nearly the same effect as picric acid, but is said to develop the crystalline structure of the ferrite.

A weak solution of iodine in water is sometimes used ; it does not attack cementite, or, at least, it leaves it bright, but pearlite is darkened. This fluid is very troublesome.

An 8°/₀ of copper sulphate of ammonia is very effective. The section is at first covered with copper ; this will wash off unless a wrong percentage has been used, and then certain portions of the ferrite which contain much phosphorus will appear in a brownish colour. Segregations are thus clearly revealed. If the sample before polishing has been **severely strained**, this etching fluid will also darken the strained regions, which are said to show slip bands.

Hydrochloric acid, even as a 1°/₀ solution, is very troublesome, but, as it attacks the highly phosphoric segregations about eighty times more severely than others, it may yet become a useful reagent. The deep pit holes which it produces in some steels are now believed to be the localities of highly phosphoric nodules, and the difference of behaviour under this and the previous etching fluid, as well as other experiments, indicate that there are two phosphide of iron impurities, the one dissolved in the ferrite, is believed to be the cause of brittleness.

As yet the most important revelations which can be expected of an examination of the micro structure of a mild steel is with regard to the previous heat treatment, the pearlite segregations being few and large if the material has been wrongly heated, i.e. too long at a low temperature. If a sample of such a steel be heated to a suitable temperature, and cooled slowly, it should show much smaller segregations than before.

The Influence of Temperature, particularly of heat, on the strength of materials is a subject of the first importance. The relation existing between the boiler pressures and temperatures will be seen from the following table :—

Boiler Pressure	Lbs.	0	50	100	150	200	250	300	400	500
Temperature . . .	°F.	212	281	328	358	382	401	417	445	467
„ . . .	°C.	100	138	153	181	194	205	214	230	242

Original researches on this subject will be found in the following papers :—'Franklin Inst.,' 1836, ii. pp. 82–208 ; M. Baudrimont, 'An.

Ch. Ph.,' 1850, iii. vol. xxx. p. 304; W. Naylor, 'M. E.,' 1866, p. 76; W. Fairbairn, 1856, 2nd ser. p. 96; Sir W. Fairbairn, 'Manch. L. Ph.,' 1871, vol. x. p. 86; 'Portsmouth Dockyard Experiments,' 1877; C. Huston, 'Frankl. Inst.,' 1878, vol. lxxv. p. 93; 'N.,' 'Glaser's An.,' 1880, vol. vii. p. 165; Dr. J. Kollmann, 'Ver. Gew.,' 2nd ser., 1880, vol. lix. p. 92; J. F. Barnaby, 1881 and 1882; J. E. Howard (Watertown Arsenal), 'I. and S. I.,' 1889, ii. p. 460; Le Chatelier, 'Comp. Rend.,' 1889, vol. cix. p. 58; Board of Trade Report (No. 257) on 'Boiler Explosions'; Prof. Martens, 'Mitt. Berlin,' 1890, p. 159; A. Bleichenden, 'M. E.,' 1891, p. 320. See also p. 190 as regards elastic limits at high temperatures.

The Influence of Cold on the tenacity of metals is dealt with by the following experimenters:—Sir W. Fairbairn, 'Parl. Report of the Commissioners of Railway Structures,' p. 321; ibid. 'Brit. Assoc.,' 1857, p. 405; K. Styffe, 1869; 'N.,' 'Glaser's An.,' 1880, vol. vii. p. 165; Capt. Bernardo, 'Rev. d'Art.,' 1890, p. 485; W. Brockbank, 'Manch. L. Ph.,' 1871, vol. x. p. 77; W. W. Beaumont, 'C. E.,' 1876, vol. xlvii. p. 43; T. Andrews, 'C. E.,' 1887, vol. lxxxvii. p. 340; Spangenberg, 'Glaser's An.,' 1879, vol. v. p. 165; T. Andrews, 'C. E.,' 1891, vol. cv. pp. 161, 169; W. Rudeloff, 'Mitt. Berlin,' 1897.

Experiments on the influence of temperature on copper and other materials:—'Frankl. Inst.,' 1836, ii. p. 39; Dr. Kirk, 'Enging.,' 1887, vol. xliv. p. 661; W. Parker, 'N. A.,' 1889, vol. xxx. p. 47; Le Chatelier, 'Comp. Rend.,' 1889, vol. cix. p. 24; Le Chatelier, Paris, 1891.

Dr. J. Kollmann's experiments on iron and steel, 150 in number, seem to be the most reliable. His conclusions are that the ultimate strength of both materials decreases with rising temperature, and that it shows a very serious drop at about 450° to 500° F. The elongation is at a maximum at about 900° F. The contraction of the fractured sectional area steadily increases till it reaches 90% at a red heat. The limit of elasticity decreases steadily.

Another result is that the ductility, as measured by bending tests, increases with rising temperatures till 450° to 500° F. (blue heat) is reached, when the material is rotten. At higher temperatures it is pliable once more. It would seem as if medium quality steel is exceedingly brittle when cold (0° F.), while tougher qualities of the same tenacity remain ductile; at any rate, boiler-smiths and ship-platers have come to the conclusion that it is risky to handle steel or iron plates in cold weather, and where serious hammering or drifting is contemplated, heaters are almost invariably applied. If the steel contains much phosphorus it is exceedingly likely to break under the percussive bending test, if carried out in cold weather.

Sudden cooling, not necessarily from a red heat, is said to produce brittleness (see T. Andrews, 'C. E.,' 1891, vol. ciii. p. 231); and repeated heatings have also produced brittleness, but the exact conditions for effecting this change are not known. E. Wehrenfennig, 'O. I. A. V.,' 1879, vol. xxxi. p. 153; J. P. Barnaby, 1881 and 1882; A. Ledebur, 1884, p. 646; C. E. Stromeyer, 'C. E.,' 1886, vol. lxxxiv. p. 122; E. B. Martens, 'Ing. Civ.,' 1886, p. 607; E. Wehrenfennig, 'Organ,' 1884, vol. xxi. p. 216; A. E. Sherk, Enging.,' vol. xliv. p. 458; C. E. S., ibid. vol. xliv. p. 491; B. H. Thwaite, ibid. vol. xliv. pp. 505, 536; Th. Edington and Son, ibid. vol. xliv. p. 505.

Occlusion of Gases.—In some of the above-mentioned cases the brittleness may have been due to the absorption (occlusion) of injurious gases, and experiments prove that hydrogen is readily absorbed, and that it injures the material. Acids and other corrosive influences are also said to produce brittleness, but the real cause may be the hydrogen which is evolved during these processes.

Dr. Schahäntl ('Bayrisches Kunst- und Gewerbeblatt,' June 1863) gives an analysis of the various layers of an exploded boiler plate, and shows that on the water side it contained an excess of oxygen, while on the fire side occluded sulphurous acid was found. That solids do absorb gases is proved by the well-known fact that soaped window glasses turn a mauve colour after a time, which is due to a chemical action of the oxygen of the air on the manganese salts in the glass.

M. Bustein ('An. Mines,' 1883, viii. vol. iii. p. 28) gives the chemical analysis of three qualities of steel which had been exposed for 112 days in flue gases or in boiler water.

Previous Treatment of Sample	Tenacity—Tons			Elongation—per Cent.		
Original	56	50	42	17	19	24
Exposed 112 days in boiler . .	51	47	40	15	16	14
" " flue . .	44	41	37	16	21	"

Nitrogen.—The influence of nitrogen has already been mentioned. Prof. Hughes ('Tel. Eng.,' 1880) deals with this subject.

Influence of Pickling.—A. Ledebur ('Stahl und Eisen,' 1887, vol. vii. p. 682) gives analyses of eight qualities of steel (wire), and finds that pickling reduces the elongation about 15%, and the ductility 39%; that exposure to the atmosphere for two months reduces both qualities about 50%, but that annealing puts matters right again, though the ductility is not perfectly restored. The action of zinc in galvanic contact with the wires is worthy of notice; it prevented corrosion, but the wires grew very brittle, and on analysing them again they were found to contain from ·002 to ·005% of hydrogen. He points out that, on account of the low atomic weight of hydrogen, these percentages should be multiplied by, say, thirty to make them comparable with the volumes of phosphorus or of sulphur. Baedecker ('Deut. Ing.,' 1887, vol. xxxii. p. 187) confirms these views about the action of acids.

Heyn ('Stahl und Eisen,' 1900, vol. xx. p. 780) states that by heating and cooling steel in an atmosphere of hydrogen it becomes brittle.

The study of this subject is as yet in its infancy, and is certainly beset with many difficulties. It may, therefore, be of interest to mention where experiments on occluded gases can be found:—L. Troost and P. Hautefeuille, 'Comp. Rend.,' 1875, vol. lxxx. p. 788, and 'An. Chim. Ph.,' 1876, 5th ser. vol. vii. p. 155; M. Reynard, 'Ing. Civ.,' 1877, pp. 91 and 210; A. H. Allen, 'I. and S. I.,' 1879, p. 480, and 1880, p. 181; F. C. G. Müller, 'Deut. Ch. G.,' vol. xii. p. 11; ibid. 'Glaser's An.,' 1880, vol. vii. p. 138; ibid. 'Stahl und Eisen,' 1882, vol. ii. p. 531; N. Zyronski, 'Soc. I. Min.,' 1884, p. 101; H. M. Howe, 'Eng. Min. J.,' vol. xlv. p. 236. Nitrogen, Harbert and Tynam, 'I. and S. I.,' 1896, vol. ii. p. 161.

It would be wrong to compare occlusion by metals with the absorbing power of fluids—for instance, with water, which takes up very much larger quantities of carbonic acid when cold than when warm. On the contrary, metals behave in a most erratic manner, and usually absorb certain gases while in a molten state, which they give off again when cooling. Thus molten silver and copper absorb oxygen, and expel it with much force as they harden. Therefore it can hardly be expected that the annealing process will drive out all the gases occluded by iron or steel, even if carried out in a vacuum, although a certain proportion would disappear. The gas given off by steel in the soaking pits is said to be hydrogen. If this gas should be found to be the cause of brittleness, it may one day be necessary to dry the gases used in melting-furnaces.

Burnt Iron.—It has been attempted to explain the mystery of burnt iron as being due to the presence of a large percentage of occluded oxygen; but iron can unquestionably also be 'burnt' in a vacuum and in gases containing no oxygen. This has led to the view that the occluded gases in the iron or steel—chiefly hydrogen and nitrogen—leave the metal and form innumerable cavities, thereby destroying its continuity and making it rotten (burnt).

W. M. Williams ('Chem. S.,' 1870, vol. xxiv. p. 790) found oxide of iron in burnt iron.

H. Caron, 'Comp. Rend.,' 1872, vol. lxxiv. p. 662. Iron could be burnt in air, hydrogen or nitrogen gas.

Professor Ledebur, 'Jahrb. B. H.,' 1883, p. 19. Phosphoric iron is more easily burnt than pure qualities. He believes burning not to be due to oxygen, and mentions **dead iron**. See also 'I. and S. I.,' 1888, vol. i. p. 386; 1897, vol. i. p. 234; 1898, vol. i. p. 145.

In spite of this diversity of opinion there can be no doubt that iron and steel do get burnt if exposed to excessive heat, and are then both red- and coldshort; and also that in re-heating furnaces, and even in annealing furnaces, burning can very easily be brought about if the fire grates are left partly uncovered, and pure but heated air is allowed to impinge on the plates.

Recalescence.—This subject is a very complicated scientific one, it has lead to important conclusions as regards annealing.

Annealing.—It is well known that the annealed iron wire has an initial hardness, which disappears on straining it. Basic (Siemens) steel, made of pure pig and low in carbon, usually shows a slight increase in its ultimate tenacity after annealing, while tempering lowers its elastic limit. The reverse is the case with acid Siemens steel, which generally contains more carbon and less manganese. Internal strains are never entirely removed by annealing (p.251). Tool steel can be thoroughly annealed by heating it to redness and cooling slowly till it is just dark, and then quenching. Mild steel gives better bending tests if quenched at a cherry-red heat than if quenched at a very dull red heat. The literature on the subject is chiefly restricted to the question of removing the injury done by punching, and to the influence of quenching on the comparatively hard steel used for guns.

The well-known optic glass works, Schott Bros., Jena, have adopted a novel method of annealing. They argued that, however slowly a glass might be cooled, the outside would always be hotter than the inside until uniformly cold, and then the inside would be in tension. They therefore cool their glass masses step by step, taking care to maintain their annealing furnaces at constant temperatures for long periods at a time, and then cooling rather rapidly to the next step. By this means the glass has repeatedly passed through a period in which it was of an equal temperature throughout, and can now be produced so as to be without strains, which is proved by its optical properties.

According to practical experience and also theoretical investigations by J. A. Brinell ('Stahl und Eisen,' vol. v. p. 611, also abstract 'I. and S. I.,' 1886, vol. i. p. 365), annealing can produce both bad and good results. Steel or iron which is kept at a dull red heat for a long time becomes rotten; if kept at a bright red heat for a long time, it partakes of the nature of burnt material. The best results are obtained by heating to a good red heat for a short time and letting the object cool, rapidly at first and then slowly. See also 'I. and S. I.,' 1882, vol. i. p. 209; 1882, vol. ii. p. 532; 1898, vol. ii. p. 137, and W. Rudeloff, 'Mitt. Berlin,' 1891, p. 109.

Effects of Quenching Red-hot Steel.—For thin plates it seems advantageous to remove the sheared edges before tempering the samples, while for thick plates the reverse appears to be true. Quenching in oil gives more toughness, while quenching in water, and particularly in mercury, increases the hardness. Quenching at a dull red heat thoroughly anneals tool steel. Large objects like armour plates would become unequally chilled, warp and crack if plunged into water; they are therefore exposed to innumerable powerful jets of water which produce equal chilling effects all over the surfaces, and yet allow the resulting steam to escape.

Galvanised Steel.—Injury, possibly due to chemical action or to the absorption of vapours, is produced if iron or steel articles are galvanised at too high a temperature. In order to guard against this danger, the fires for heating the zinc baths should always be placed round their sides; this allows the impure zinc (containing iron) to fall to the bottom, where it floats on a layer of molten lead. If mixed with the other zinc, the temperature of the bath would have to be raised too high in order to keep it fluid.

Electroplating with zinc has a temporary anti-corrosive effect, it is of greatest value in revealing bad welds and other defects.

The Influence of Hammering or Cogging the Cast Ingots, or of Rolling them direct.—In England the practice is either to hammer or cog the ingots, while in Germany this is not always done, which is made possible by the different chemical compositions. Hammered ingots show a better surface when rolled, and are said to be a little denser. The question of waste is intimately bound up with the above, and there can be no doubt that there is less scrap from plates whose ingots were cogged or hammered than if no preliminary shaping had taken place. From a boiler-maker's point of view a large amount of scrap is an advantage, for the edges are sometimes overheated, and are never of exactly the same quality as the rest of the plate. As regards the final quality of the material several important investigations have been carried out, but being in each case confined to steel of one company, the deductions are not conclusive as regards other makers.

J. Riley, 'I. and S. I.,' 1887, p. 121. Influence of rolling, hammering, and annealing. This is a very exhaustive paper, and the conclusion to be drawn from it is that an excessive amount of hammering and rolling is not necessary. From one and the same charge thick plates are both weaker and more ductile than thin ones, which have received more work.

W. Parker ('I. and S. I.,' 1887, p. 134) mentions experiments to show that tenacity and elongation increase with rolling.

H. Allen, 'C. E.,' 1888, vol. xciv. p. 240. Rolled wire and drawn wire are neither much stronger nor more ductile than the billet unless tested in an unannealed condition.

Dr. Kirkaldy gives a few experiments.

W. H. Greenwood, 'C. E.,' 1889, vol. xcviii. p. 83. Fluid compressed steel. The benefit derived from this preliminary compression does not seem to dissipate during the subsequent manipulations.

The immediate effects of hammering, &c., are investigated in the following papers :—

H. Tresca, 'M. E.,' 1878, p. 315.

Ibid., 'Comp. Rend.,' 1883, vol. xcvii. pp. 222, 515, 928. Localisation of heat developed by a hammer blow.

M. Lau, 'Comp. Rend.,' 1882, vol. xciv. p. 952.

M. Osmond, 'Comp. Rend.,' 1885, vol. c. p. 1228. Motive power for rolling steel is 2½ times as much as for iron.

Dr. J. Kollmann, 'Ver. Gew.,' 1880, 2nd ser. vol. lix. p. 6.

Local Heating.—Serious injury is sometimes done to a steel plate by drawing out one of its corners, even though this be done at a red heat. Flanging, whether done by hand or presses, has also led to failures. It has never yet been reasonably demonstrated that this is due, as some contend, to the local heating alone, or, as others believe, to the straining produced by the unequal expansion of the locally heated parts; other influences, such as the chemical composition of the material and the processes through which it has passed, are probably important factors.

The following list contains published cases of cracked plates :—

Sir N. Barnaby, 'I. and S. I.,' 1879, p. 242. List of forty-three steel failures at Chatham in six months.

W. Denny, 'N. A.,' 1880, vol. xxi. p. 185. List of steel failures in his own yard.

W. Parker, 'N. A.,' 1881, vol. xxii. p. 12. Cracked plates of the steam yacht 'Livadia.'

A. C. Kirk, 'N. A.,' 1882, vol. xxiii. pp. 131 and 137. Cracked flanged plate.

J. F. Barnaby, 'Enging.,' 1883, April 20; D. S. Smart, 'C. E.,' 1884, vol. lxxx. p. 102. Sketches of cracked plates.

W. Parker, 'N. A.,' 1885, vol. xxvi. p. 253. Failures of thick steel plates.

H. Goodall, 'C. E.,' 1888, vol. xcii. p. 10. Sketches of some cracked plates. (See also p. 252.)

Of course this list contains only a very small fraction of the number of cracked plates, for as long as it is not settled whether these failures are due to the material, or to the mode of working the plates, it will be the custom, as now, for the steel-makers to replace the failed plates.

Blue Heat.—Perhaps some, though certainly not all, such failures are due to working the steel at a blue heat. This is another mysterious phenomenon connected with iron and steel. The salient features are that steel will bend without fracture at temperatures ranging from 0° F. to 450° F. (see Dr. J. Kollmann, p. 151), and again above 550° F., but it is quite rotten between these limits; and, further, a piece of steel which has been bent or hammered at this particular temperature, but without breaking, acquired and retained a permanent excessive brittleness (see p. 256). This brittleness can be removed by annealing, but time alone (8 years) does not restore the quality. Experiments on the subject, as well as failures, which are attributed to working steel at a blue heat, will be found in the following papers:—

M. Valton, 'Berg- H.-Z.,' 1877, vol. xxxi. p. 25; D. Adamson, 'I. and S. I.,' 1878, p. 402; W. Denny, 'N. A.,' 1880, vol. xxi. p. 185; C. E. Stromeyer, 'C. E.,' 1886, vol. lxxxiv. p. 114; ibid., 1888, vol. xciii. p. 89; W. Parker, 'I. and S. I.,' 1887, p. 136; G. B. Craig, 'N. A.,' 1888, vol. xxix. p. 113; Rudeloff, 'Mitt. Berlin,' 1889, p. 97; Board of Trade Report, August 31, 1886; 'Am. R. M. M. A.,' 1892; Prof. A. Ledebur, 'Glaser's An.,' 1886, vol. xviii. p. 205; A. Martens, 'Deut. Ing.,' 1892, p. 172.

Influence of Severe Stresses.—The limit of elasticity of a piece of mild steel, which has been stressed beyond its original limit, is raised even beyond its nominal ultimate strength, but this is no real improvement, for the severer the strain the more is the ductility reduced even down to brittleness. Possibly, however, the raising of the limit as regards one stress, for instance tension, may produce brittleness as regards compression and shear or *vice versâ*. It also seems as if steels rich in nitrogen and phosphorus are more liable to suffer in this respect than other steels.

Influence of Time.—The author's researches ('I. and S. I.,' 1907) have revealed that a majority out of 26 different qualities of mild steel grew brittle after a time if they had been locally injured; for instance, by nicking, and that heat accelerates this ageing process. There was no indication that the strengths had diminished, and that therefore old boilers might be weaker than new ones, but they would be more liable to suffer from any excessive strains, such as frequent proof tests, and actual failures are cited in support of this contention. Some Sheffield steel makers are of opinion that steel ingots improve by storing.

List of Spontaneous Failures.—Z. Colburn, 1860, p. 32. Old boiler stays said to be brittle.

Professor Thurston, 'I. and S. I.,' 1875, p. 342. Old rails had grown brittle, and improved on re-rolling.

Professor Thurston, 'Materials,' 1883, vol. ii. p. 576. Prolonged excessive straining causes rupture.

Collingwood, 'Am. C. E.,' 1880, vol. ix. p. 171. Tenacity of wire changes after a time.

L. Fletcher, 'C. E.,' 1884, vol. lxxx. p. 136.

A. J. Maginnis, 'Engr.,' 1885, vol. lx. p. 447; C. E. Stromeyer, 'C. E.,' 1886, vol. lxxxiv. p. 187. Sketches of plates which cracked spontaneously while not in use.

J. Harrison, 'Engr.,' 1886, vol. lxii.

'Army N. J.,' 1887, vol. xxiv. p. 65. A long list of steel armour plates which cracked spontaneously before being fitted.

Collingwood, 'Am. C. E.,' 1880, vol. ix. p. 171, and W. Hewitt and Felton, 'Am. M. E.,' 1888, vol. ix. p. 47. Strength and ductility of wires and plates change 24 hours after rolling.

H. M. Howe, 1890, p. 195, old armour plates are brittle; p. 210, time removes injury caused by cold working.

For other cases, see p. 136.

Although not as yet published, there are a few cases of boiler shells cracking under the hydraulic test, and particularly of furnace saddle corners cracking while out of use or if struck with a hammer. One of the most curious cases is perhaps the following: A steel steamer which had been on fire was being repaired, and several of the buckled plates had been taken out to be straightened. It was found that some of these cracked spontaneously while lying on the ground, although they had not cracked while being removed from the ship.

Besides these various failures other instances will readily suggest themselves about the influence which time has on the quality of iron and steel. There is the case of the gun of H.M.S. 'Collingwood,' which burst with a light charge after two years' rest, succeeding on its trial with several proof charges. These and similar failures are so very mysterious that they have been attributed to hidden or incipient flaws; but they have never been seen, and if the point is conceded that certain qualities of steel have a natural tendency to change—and this is stoutly maintained by experienced steel manufacturers—then the difficulty is somewhat reduced. That important changes are slowly occurring in steel is proved by the fact that ordinary test samples which have been successfully bent to a small radius crack spontaneously some time afterwards; that armour plates, after having been fired at, emit strange sounds for a long period, and that the elastic limit of steel test pieces which have been stretched a few per cent. slowly grows higher and higher when at rest.

Time can hardly be called a treatment, but the mystery attaching to its effects should be a sufficient excuse for most carefully investigating all such cases as may possibly have been produced by it.

Influence of Punching.—One treatment which produces injurious effects in steel is punching holes into the plates. The experiments on the subject are too numerous to be mentioned here, but many of them will be found in the chapter on 'Mechanics,' under 'Riveted Joints,' p. 219. The thickness of the plate, the diameters of the punch and die, as well as the hardness and chemical composition of the plate, affect the pliability of such samples. It has been found that rimering out $\frac{1}{16}$ in. of the holes removes all bad effects; but if it is true, as stated by Mr. Beck-Gerhard ('Gorni J.,' 1884, p. 347), that the curves of stress slowly extend at least 5 ins. away from the hole, then punched plates ought to be annealed or rimered at once. This experiment was as follows: A $\frac{5}{8}$-in. plate was polished on one side and punched (in a cold atmosphere). Spiral curves then showed themselves, which were first washed with aqua regia. The piece was then planed into several strips and each tested, when the spiral curves reappeared and were perceptible to the touch.

This may explain the curious curved markings near punched holes and sheared edges. In every one of these lines the surface scale has fallen off, showing that here stresses have been at work producing local deformations of at least $\frac{1}{2}°/_{o}$, for it is only after steel has been stretched this amount that the mill scale falls off the plates. Illustrations of similar effects will be found in Kirkaldy's works. (See p. 153.)

Modulus of Elasticity.—On loading a test piece it will be noticed that for steel and iron the elongation is almost proportional to the stress, being at the rate of $\frac{1}{1000}$ in. in 10 ins. for about every 1·3 ton per square in., from which it follows that the modulus of elasticity is about 13,000 tons, or 30,000,000 lbs. per square in., or 20,000 kils. per square millimeter. With cast iron and various other metals there seems to be a change in this modulus when the stress is increased beyond a certain point, and some people have called this the limit of elasticity, but a better name is limit of proportional elongation. The more accurate the strain indicators are, the more gradual does this change appear, and the obvious conclusion is that with these metals the modulus of elasticity is a variable quantity, growing smaller as the stress increases. It also decreases about $1\frac{1}{2}$ per cent. for every 100° F. rise of temperature (see H. Tomlinson, 'Phys. S.,' 1887, vol. viii. p. 171).

Elastic Limit.—It will be noticed that the pointers of the strain indicators oscillate slightly for every newly added load, coming to absolute rest only after a very long period. But when a certain stress has been reached this action ceases and the pointers acquire a slow onward motion. The elastic limit has now been reached, and to verify whether this is so or not, several or all of the weights are removed, and the pointers will either return to the positions previously occupied or not. This check is necessary, as the giving way of the attachments of the test pieces sometimes produces strange effects, and may even cause one of the pointers to travel backwards. For this reason also the determination of the elastic limit should always be made with the help of three strain indicators. Mild steel shows an elastic limit of about 15 tons. If very mild and previously tempered it is sometimes as low as 10 tons, but in such cases it is hardly perceptible and no breakdown point can be noticed.

Breakdown Point.—Continuing the straining of properly annealed samples, a point will be reached when the slow motion of the pointers gives way to a very rapid one, which is not stopped even by reducing the stress by several tons. This is called the 'breakdown point,' because of the behaviour of the material, or the 'drop,' the lever falling through a considerable angle. It has also been called the limit of plasticity, because above this point the material behaves as if it were plastic. This point is often mistaken for the limit of elasticity, and the two points sometimes fall together.

Irregular Stretching.—Even now it is of interest to watch the strain indicator, for it will be found that the plastic elongation proceeds very irregularly. Generally after adding a weight it commences slowly, increases, and then diminishes, until at last no further motion can be detected. Very often, particularly if only small additions are made to the load at one time, the pointers vary their speed repeatedly, increasing and diminishing their velocity several times without any additional weights being added to the lever. This seems to be due to local elongations taking place, particularly in long and thin test pieces, first at one point and then at another. A recently tested bar, $\frac{3}{8}$ in. square, showed four contractions in addition to the fracture in a length of 10 ins.

Irregular Elongations.—An explanation will suggest itself if several (say, four) short strain indicators are attached along the length of the sample, for it will then be noticed that first one and then the other span is elongating, showing that waves of plasticity pass along the samples. This may also be noticed when the breakdown point is reached, for then the mill scale falls off, first at the extremities, and then more towards the centre. The scale falls off when the stretch exceeds $\frac{1}{2}$ % of the length.

The same phenomena, but more marked, may be noticed when twisting wire in a torsion machine. Instead of proceeding uniformly, it will be noticed that the twist commences at one end (fig. 115), and that, like a wave, it travels to and fro till the sample breaks.

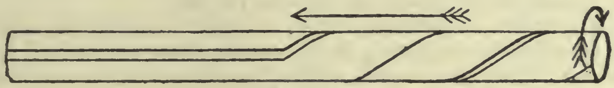


Fig. 115

Changes in the Limit of Elasticity.—The next point to be noticed in a test piece is that when once the elastic limit has been passed, and the sample then unloaded, it will not again elongate permanently until the previous stress has been reached. This is only natural, for the second testing is but a continuation of the first, and it is not difficult to accept the statement that the elastic limit of a sample is raised by preliminary testing.

R. H. Thurston, 1883, p. 601, has suggested that this behaviour would enable one to detect whether a broken structure had been overstrained. A test piece would have to be carefully cut from the plate and its limit of elasticity determined. Unfortunately experiments show that, even with the most careful handling during preparation, the elastic limit of the test piece has again fallen to its original value, or perhaps the shock of the rupture has produced this effect.

Influence of Time.—The test piece under consideration could now be stretched till it has elongated 5 %, which will raise the stress to about 25 tons. It may then be left in the machine overnight with the full load on it, or it may be put aside for a day or two. On re-testing it will be found that the elastic limit has risen considerably above 25 tons, which was the last stress to which it was subjected.

The following experimental results will illustrate this :—

Sample No. 1	
Elastic limit	19·3 tons
Ultimate strength	29·7 tons, 20·1 % elongation
Sample No. 2	
First elastic limit	16·3 tons
Then loaded to	24·6 tons, 5·2 % elongation
Second elastic limit	28·5 tons after an interval of 10 days
Ultimate strength	29·8 tons, 17 % elongation

A long list of experiments on this subject by Bauschinger is contained in 'Civil I.,' 1881, vol. xxvii. p. 1; also 'Mitt. Munich,' 1886, vol. xiii. 1891, vol. xx. He investigated the behaviour of 14 samples of iron and steel, and also copper and gunmetal. Unlike the author's experiments, none of his showed an increase of elastic limit beyond the ultimate strength of the material, but even with him the influence of time in raising it is very marked. With copper and gunmetal the elastic limits only rise as high as the preliminary stress. In all these cases the second elastic limit and breakdown point fall together, and the drop is now very much greater than with an annealed sample.

The question naturally arises, What would happen if the sample had first been subjected to a compression test? This experiment has also been carried out, and it was found that the elastic limit for tension had been reduced from 19·3 and 16·3 to 11·8 tons, and the ultimate tenacity raised to 30·7 tons, elongation 7·2 %.

A preliminary compression stress at right angles to the axis of the sample (produced by drawing it out under a hammer) raised the limit to 20·5 tons. This also increased the tenacity to 32 tons, and reduced the elongation to 12 %.

These experiments readily suggest that, as the elastic limit is a changeable value, it cannot be a reliable measure of the working strength of a material. When a preliminary test has raised the elastic limit considerably it may be dangerous to repeat it, because if the new elastic limit is accidentally exceeded a very considerable breakdown occurs, which may lead to rupture. Those parts of a structure which have been subjected to excessive compression stresses should not be exposed to severe tension stresses, as their elastic limits of tension have been lowered and their ductility reduced. The reverse is also probably true.

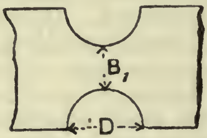


FIG. 116

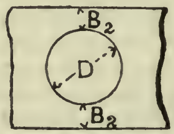


FIG. 117

Contraction of Test Pieces.—Continuing to follow the behaviour of a test piece, and this time until fracture takes place, it will be found that some materials, such as hard steel, manganese steel, bad iron, and gunmetal, show no contraction previous to rupture, while mild steel, iron, and brass and copper, do. The reasons for this contraction are not known. It has been suggested that contraction is due to local weakness, so that those metals which contract most are least uniform as regards tenacity; but if that were the case tensile tests with drilled samples would be very irregular, according as to whether the hole was near a weak or a strong place. The following experiments on a mild steel plate whose tenacity was 28 tons show that this is not the case:—

The samples were shaped as shown in figs. 116, 117. Thickness of plates $\frac{1}{2}$ in.; diameter of hole 1 in., drilled.

(See also E. Richards, 'I. and S. I.,' 1882, p. 43.)

Another suggestion is, that the rise of temperature of a test piece weakens the part which first contracts more than the others which have accidentally not contracted. Dr. J. Kollmann's experiments ('Ver. Gew.,' 1880, 2nd ser. vol. lix. p. 104) confirm this, for there it will be found that the contraction steadily increases from about 20 % at an ordinary temperature to 90 % at a red heat. Unfortunately for this view the tenacity does not show the same regularity. An explanation of this phenomenon is, therefore, still required. The subject will be referred to again when discussing compound stresses.

Breadth		Ult. Strength
B_1. Inches	B_2 Inches	Tons per Square Inch
1·85	...	30·1
1·25	...	30·5
·71	...	30·3
...	·53	29·6
·49	...	30·5
...	·43	29·8
...	·33	29·1
...	·28	28·6
...	·23	30·5
...	·18	28·7
...	·13	28·9
...	·075	28·6

Fractures.—The next thing to be noticed in a fractured test piece of mild steel is, that when placed together only the edges touch, leaving a hollow, as shown in fig. 118. There can be but one explanation, viz. that the rupture started near the centre of the section, and that the outside fibres continued to stretch after this point was reached. This suggests the view that the stresses to be found in the centre of a test piece differ from those on the outside surfaces, and are also more injurious. An examination of almost any torn sample of mild steel will show that near the edge the surfaces of rupture are very much on the slant and have every appearance of having been produced by partial shear, suggesting the idea that this material gives way more readily under shearing stress than under tension (see pp. 163).

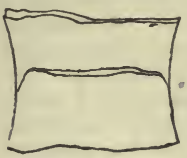


Fig. 118

The same tendency is noticed when tearing a sample perforated as in fig. 119. The lines of fracture, instead of running as horizontal as possible, will be distinctly steeper than the angle at which the holes were drilled.

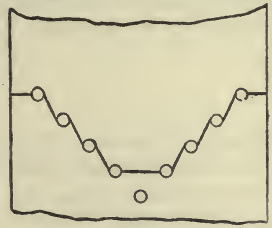


Fig. 119

A few other matters which have been noticed in fractures are: White specks, which are due to a local excess of phosphorus and other impurities (Stubbs, 'I. and S. I.,' 1881, p. 379; H. Eccles, ibid., 1888, p. 72; Prof. Ledebur, 'Stahl und Eisen,' 1889, vol. ix. p. 13). A smell of ammonia, said to be due to occluded nitrogen, is sometimes noticed; the colour also varies from bluish grey to salmon-colour tints or yellow ones.

Shearing Stresses.—The investigation of shearing stresses is beset with various difficulties, one of the most important being the smallness of the strains. Direct experiments have therefore only been useful in determining the ultimate shearing strength of various materials.

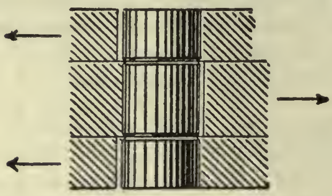


Fig. 120

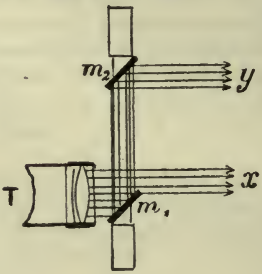


Fig. 121

It has been found that the hardness of the metal into which the holes are drilled influences the results. It is therefore usual to groove the samples to be tested as shown in fig. 120. Most experiments on riveted joints include some carried out as above, and the general conclusion is that the shearing strength is less than the tensile strength.

More interesting results are obtained if the material is exposed to torsion stresses. One end of a cylindrical bar is secured to the head of a lathe, while the other end is supported, if possible, on a knife-edge, and carries a lever, which is gradually loaded while the lathe head is being turned round. The twisting of the bar is measured with the help of a pair of mirrors m_1, m_2, arranged as shown in fig. 121. A graduated scale is placed at a considerable distance from the test piece, and examined partly direct (x), and partly by doubly reflected light (y). Two scales instead of one are then visible through the telescope T, and it is their relative displacement which measures the angular deflection. This arrangement has been devised by the author and has proved to be very accurate.

By plotting down the readings, curves are obtained, which may be called torsion diagrams, and represent the amount of twist which various torsion moments impart to a test bar. It is usual to reduce all these values so as to obtain the shearing stresses of the outside fibres and their angular displacement.

The latter value is found by multiplying the angle of twist into half the diameter of the bar, and dividing by the distance of centres.

The shearing stress of the outside fibre is found by the formula

$$\sigma_0 = \frac{16 \, . \, M}{\pi \, . \, d^3}.$$

Here M is the torsion moment, and d is the diameter of the bar. This formula is only correct as long as the moments are strictly proportional to the twist; where this is not the case, and particularly when the limit of elasticity is passed, or the point of rupture reached, it gives wrong results.

The actual shearing stress σ in the outer fibre is then found as follows: Let the heights of the line O B C (fig. 122) represent the distribution of the circumferential shearing stresses over the radius of a bar whose diameter is $d = 2r$, and which has been twisted through an angle θ in a length l, while the torsion moment is M. The sheer angle is β.

D C represents the sheering stress in the outer fibre. Now if a thin film of metal, of the thickness dr, be machined off the circumference of the bar, and if it be twisted once more, till the stress in the (now reduced) outer fibre is again equal to D C, then, as the new curve O P C is similar to, but shorter than, the original one, the new twisting moment M_1 will have to be somewhat weaker than M, viz. $M_1 = M \cdot \frac{(r - dr)^3}{r^3}$, while the angle θ has increased from θ to $\theta + d\theta = \theta \cdot \frac{r}{r - dr}$. Now let a film of material, of the thickness dr, be placed round the bar, so that it is once more equal to its original diameter, $2r$. Let this outer film be twisted till it has acquired a stress of

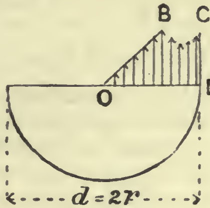


FIG. 122

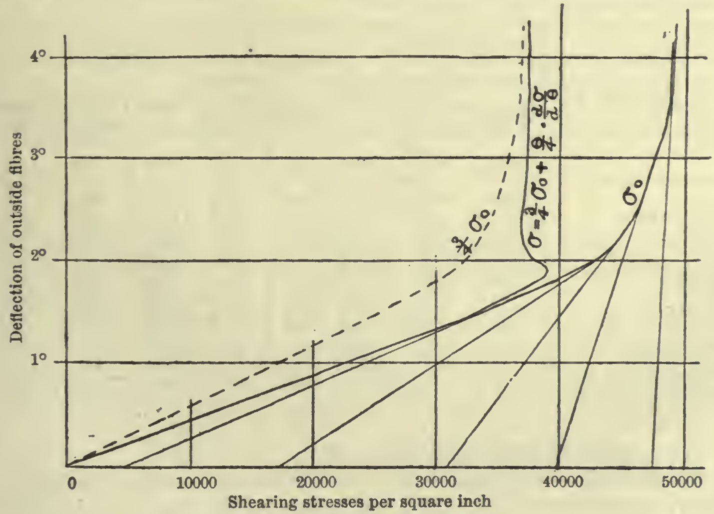


FIG. 123

$\sigma + \frac{d\sigma}{dr} \cdot \frac{dr}{2}$; then the torsion moment of the film is $\left(\quad + \frac{d\sigma}{2}\right) 2\pi r^2 dr$. The second term in the bracket is negligible and it follows that

$$\frac{dM}{dr} = -3 \cdot \frac{M}{r} + 2 \cdot \pi \cdot r^2 \cdot \sigma; \text{ but, as } \frac{dr}{r} = \frac{d\theta}{\theta}, \text{ we have}$$

$$\sigma = \frac{1}{2 \cdot \pi \cdot r^3} \cdot \left(3 \cdot M + \theta \frac{dM}{d\theta}\right) = \frac{4}{\pi \cdot d^3}\left(3 \cdot M + \theta \frac{dM}{d\theta}\right),$$

which formula can easily be converted into $\sigma = \frac{3}{4}\sigma_\circ + \frac{\beta}{4}\left(\frac{d\sigma^\circ}{d\beta}\right)$, $\sigma_\circ$ being $16M : \pi d^3$.

The way to construct a curve which will show the actual shearing stress σ of the outside fibre, when the value $\sigma_\circ$ has been previously determined, can be carried out as shown in fig. 123. The line σ_0 is copied from Platt and Haywards's paper ('C. E.,' 1887, vol. xc. plate 10, fig. 6). The faint tangential lines have been drawn to measure the value $\frac{d\sigma_0}{d\theta}$. The line $\frac{3}{4}\sigma_0$ has been constructed by proportional reduction, and to this has been added the respective values of $\frac{\theta}{4} \cdot \frac{d\sigma_0}{d\theta}$, with the help of which the curve σ has been constructed. The experimenters state that for this particular sample the elastic limit of shearing stress was 46,400 lbs.; but an examination of the curve σ shows that it was already reached at 35,000 lbs., and that at 38,000 lbs. a very serious drop took place, and the material of the outside fibre had not recovered even when the shear angle had increased to $\frac{1}{2}°$. Beyond this point it is always sufficiently accurate to adopt the value $\sigma = \frac{3}{4} = \sigma_0 = \frac{12 \,.\, M}{\pi \,.\, d^3}$. That this is fairly correct will be seen from the following experiments on tension, torsion, and direct shear (Platt and Haywards, 'C. E.,' 1887, vol. xc. p. 408):—

Materials	Limits of Elasticity			Ultimate Strengths				
	Tension (1)	Estimated from Torsion $\sigma = \frac{16 \,.\, M}{\pi d^3}$ (2)	Ratio 2 : 1 (3)	Tension (4)	Estimated from Torsion $\sigma = \frac{12 \,.\, M}{\pi d^3}$ (5)	Ratio 5 : 4 (6)	Experimental Shear on Rivets (7)	Ratio 7 : 4 (8)
	Tons	Tons		Tons	Tons		Tons	
Wrought iron	15·04	8·99	0·597	21·60	17·63	0·816	18·76	0·869
" "	16·82	10·36	0·616	25·00	20·20	0·809	21·21	0·849
" "	17·13	10·22	0·597	24·56	20·65	0·841	20·72	0·844
Bessemer steel	31·14	20·28	0·651	52·20	31·25	0·599	35·21	0·675
Crucible "	31·06	19·36	0·627	52·16	29·65	0·568	33·30	0·639
Cast "	17·22	10·40	0·604	38·04	24·30	0·639	27·60	0·726
Siemens "	17·85	10·20	0·571	28·40	20·90	0·736	23·00	0·810
" "	16·82	10·16	0·507	25·75	19·70	0·765	21·05	0·818
Mean	...	...	0·609	...	...	0·722	...	0·779
Muntz metal	11·20	8·70	0·777	25·46	18·26	0·717	18·60	0·731
Gunmetal Cu 64, Sn 8, Zn 2	7·25	5·40	0·745	13·68	11·06	0·809	12·47	0·917

For additional references, see p. 163.

These, D. Kirkaldy's, and some similar experiments by G. Berkley, 1868, V. Appleby ('C. E.,' 1883, vol. lxxiv. p. 268), the latter including compression tests, are apparently the most exhaustive ones that have yet been made to ascertain the relation which exists between compression, tension, and shearing stresses.

The latter are certainly always smaller than either of the two former, amounting to from 50 to 90 %.

Torsion tests carried out on bars from which the black scale had not been removed showed that it falls off when the limit of elasticity is reached, but even then only along a few axial lines, which implies that the elastic limit and breakdown point fall together.

Bending Stress.—After the foregoing it will be unnecessary to analyse the behaviour of beams subjected to stresses beyond their elastic limit; suffice it to say that as the limits of elasticity of the top and bottom fibres are not necessarily reached at the same time, it is impossible to separate the one from the other; but, assuming that they did agree, then the stress in the outside fibre of a narrow, rectangular bar would be found by the following formula :

$$S = \frac{6}{h^2 b}\left(\tfrac{2}{3}M + \frac{c \, . \, dM}{3 \, . \, dc}\right).$$ Here c stands for curvature.

From an examination of fig. 124, which represents a beam subjected to an excessive load, Q, it is evident that $\frac{dM}{dx} = Q$, and, by carefully measuring the various curvatures and changes of curvatures of such a sample, it would be possible to construct a strain stress diagram with the help of the following formula :

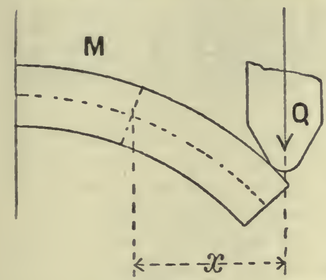


Fig. 124

$$S = \frac{Q}{h^2 b} \, . \, \left(4 \, . \, x - 2 \, . \, \rho \, . \frac{dx}{d\rho}\right).$$

It must not be forgotten that ρ, the radius of curvature, grows smaller with increasing x, and that $\frac{d\rho}{dx}$ is negative.

As in the case of torsion experiments, the second term in the bracket is very small when the point of rupture is reached, and to estimate the ultimate strength of a plastic material, including cast iron, but not glass or other brittle substances, it is sufficiently accurate to use the formula

$$S = \frac{4.Q.l}{h^2.b}.$$

This is 33 % less than obtained by the generally accepted elastic formula, and accounts for the fact that beams are apparently so much stronger than they should be.

Compound Stresses.—It has already been pointed out that the elastic limit and ultimate strength of a material are very much less in shear than in tension, or in compression. But it is well known that a **shearing stress** is composed of a tension and a compression stress, each of equal intensity, as shown in fig. 125, and it is therefore of importance to ascertain what other compound stresses exist.

There are, firstly, the simple stresses, (I.) **tension**, and (II.) **compression.**

A combination of two of these acting at right angles produces a (III.) **shearing stress** (fig. 125).

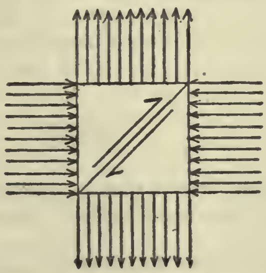


Fig. 125

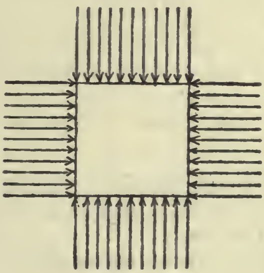


Fig. 126

IV. Two equal compression stresses are met with chiefly in railway axles when the wheel boss has been shrunk on them. This combination might be called a **shrinking** or **strangling stress**, to distinguish it from a compression stress, which acts only in one direction. (See fig. 126.)

V. The reverse of this stress might be called **drum tension** (fig. 127), as it is best represented by that case; it is also met with in thin spherical shells subjected to internal pressure. (See also p. 191.)

By adding stresses at right angles to the planes in which the last two are acting four others are obtained.

VI. **Fluid Pressure** (fig. 128).—In this case there are only compression stresses. If they are all changed into tension stresses we get a combination which might be called (VII.) **negative fluid pressure**, or **solid tension** (fig. 129).

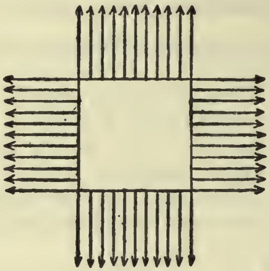


Fig. 127

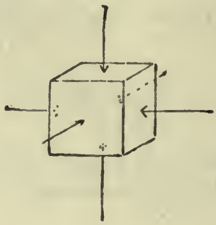


Fig. 128

VIII. When two of the stresses are compression, and the other one tension, we have the case of wire-drawing. This might be called **a draw stress** (fig. 130). (See p. 142.)

IX. By combining two tension stresses with one compression we reproduce a condition which is found on the inner spherical surfaces of very thick-walled exploding shells. This might be called a **bomb stress** (figs. 131 and 181). Of the last six combinations there is only one

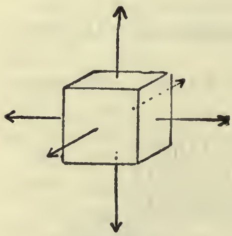


Fig. 129

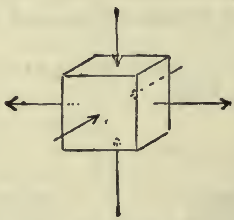


Fig. 130

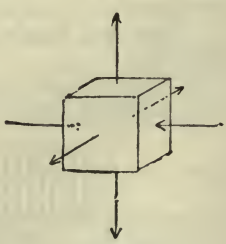


Fig. 131

about which anything definite is known, and that is that no material, however weak it may otherwise be, has been destroyed by fluid pressure, however great.

Solid tension (or negative fluid pressure) does not occur in practice, but the following two cases are an approach to this condition. If a solid sphere is heated to redness, and then plunged into cold water, the outer surface solidifies while the centre is still red hot. The external diameter will be somewhat larger than it would have been if cooled slowly; and when the centre has grown cold, a tension will be found there acting in every direction which may result in clinks in forgings, *i.e.* violent internal fractures. In large masses of steel this tension even comes into existence while the centre is still red hot, and cavities are formed, to prevent which ingots are never cast circular, but square or of polygonal shape, so that the sides may collapse. Of course it it impossible to estimate the stress which has produced these holes.

A somewhat similar stress is found at the point of contraction of test pieces (fig. 132) when of a circular section. The sample is being stretched in the direction of the arrows, *l*, *l*, and the lines of force, *s*, *s*, will adapt themselves to the fibres, at any rate at the circumference, and there their curved shape will produce radial tensions, as indicated by the looped arrows (fig. 133), so that at the centre of the smallest section there exists a tension in every direction.

This might explain why with mild steel, where there is considerable contraction of area, the fracture starts at the centre, the material being less able to withstand a solid tension than a simple one, as at the circumference of the fracture. A careful analysis of the distribution of stresses produced by the load at the instant of rupture will

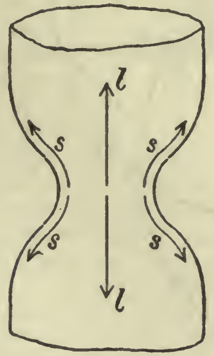


Fig. 132

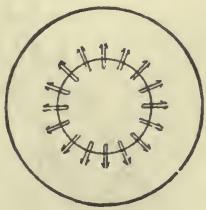


Fig. 133

perhaps enable one to obtain numerical values for different materials. Thus, it is not unusual for the contraction to exceed 50 %, and if the load at rupture was 80 % of the maximum, or, say, 24 tons per square in. instead of 30, the average stress in the reduced section must have been 48 tons; and when this is combined with the drum tension, due to the shape, which also exists there, it is not unreasonable to assume that the sample only gave way to a solid stress whose components amounted to from 96 to 144 tons. Hard cast steel will not resist these compound stresses without rupture, and therefore does not contract; and it might even be questioned whether for compound stresses this material is as strong as the milder qualities.

Recently Mr. J. J. Guest ('Phil. Mag.,' July 1900, p. 70) has investigated this subject by experimenting on steel, brass, and copper tubes. These were placed in a tensile testing machine and had attached to them other appliances, so that stresses by tension, torsion, or fluid pressure could be produced both singly and combined. Each tube was tested over and over again, but only up to its elastic limit. These and a few other tests suggest that a material breaks down, not exactly when a certain limit of stress is passed, but when the difference between any two of the three principal stresses, all acting at right angles to each other, exceeds a certain limit. Thus as there is no fluid pressure round an ordinary test piece, the tensile stress is also the difference between itself and the cross stresses which are nil. In wire drawing there is a circumferential pressure, equal to two very severe cross stresses and a longitudinal pull, and as the difference between a tension in one direction and a pressure in another is the sum of the two stresses, the material gives way when this sum reaches a certain limit. A shearing stress of the intensity σ is a com-

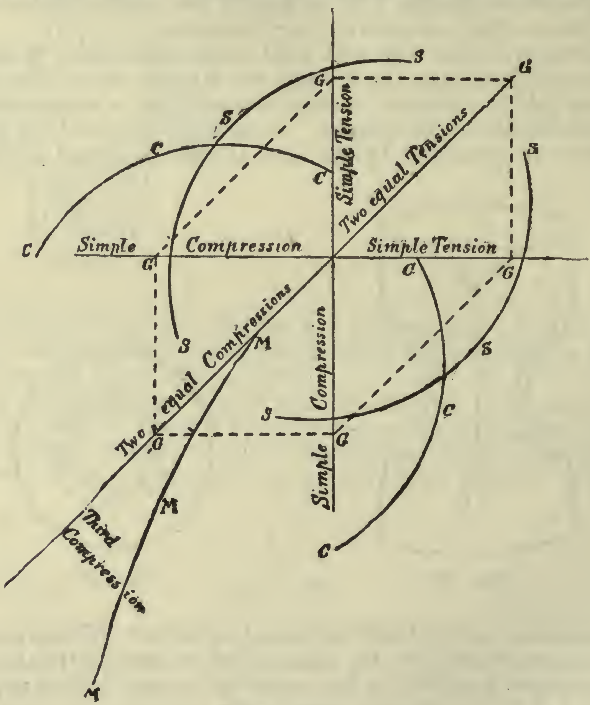


Fig 133*a*

pound of one tension stress, $+\sigma$, acting across one compression stress, $-\sigma$, and mild steel subjected to this compound stress break down when σ is about one half of the tensile limit of elasticity. This relationship is illustrated in fig. 133*a* by the dotted polygon marked G . . . G. One principal stress is marked off horizontally from the vertical line, and the other is marked off vertically from the horizontal line. If the two stresses intersect outside the polygon, the material breaks down. In fig. 133*a*, the lines SS represent characteristic boundary curves for mild steel, they do not differ materially from Guest law as represented by the polygon, but cast iron, marked CC, has a totally different curve; and marble, marked MM, differs from both. So that with our present limited knowledge about compound stresses it is safer not to believe in empyrical relations, but to assume that each material has a characteristic curve of its own. According to Guest's law cast iron should be weaker than mild steel both as regard tension, compression, shear drum tension and strangling stress, but if SS and CC correctly represent the properties of these two materials, and if two hollow spheres of identical dimension were made, one of cast iron and the other of steel, the former would be the weaker under internal pressure, resulting in drum tension, while the other would be weaker under external pressure, resulting in a strangling stress. The curve marked MM reproduces the results obtained by Th. v. Karman ('Deut. Ing.,' 1911, p. 1749). He subjected a cylinder of marble to a circumferential pressure, equivalent to two equal compressions which in fig. 133*a* meet on the diagonal, and in addition he loaded the cylinder axially with a third compression stress which is indicated by the curve MM. It will be seen that the higher this axial stress the higher may be the two other stresses. Other investigations into this matter are to be found as follows: E. L. Hancock and W. E. Williams ('Phil. Mag.,' 1906, v. 12, p. 418, etc., and 1908, v. 15, p. 81, etc.). Prof. Voigt ('Wied. An.,' 1894, v. 53, p. 43; 1899, v. 67, p. 452).

The above-mentioned results with mild steel are to a certain extent confirmed by some very interesting experiments on flat unstayed plates which were subjected to various pressures and very carefully gauged by Prof. C. Bach ('Deut. Ing.,' 1897, p. 1158, etc.).

These plates, five in number, were made of mild steel of 24·5 tons tenacity, with a limit of elasticity of 10·5 tons in tension and limit of plasticity of 15 tons. They ranged from $\frac{3}{8}$ to $\frac{3}{4}$ in. in thickness. In each plate the first permanent bulging occurred when the tension and compression stresses in the centres of the plates, as calculated by the author from the acquired curvatures, had reached from 7,000 to 8,000 lbs. (7,650 lbs. mean). This is almost exactly 30 % of the limit of elasticity of tension. Now these experiments do not show whether this giving way was due to V. p. 159, 'drum tension' on the outer surfaces of the plates, or to IV., 'strangling stresses' on the inner surfaces. Nor do these experiments show the drum tension due to the expansion of the shell. This shell plate was $\frac{7}{8}$ in. thick and 28 ins. diameter. Its enlargement would tend to increase the drum tensions (see p. 159) and reduce the strangling stresses, but this influence would hardly affect the stresses by more than 10 %. Should it be shown that this steel gave way at $7650 + 765 = 8415$ lbs. drum tension, then the result is in direct conflict with Mr. Guest's experiments, which show that with his harder steel the limit of elasticity for 'drum tension' may be in excess of that for simple tension. Possibly it will one day be shown that the plates gave way under a strangling stress of $7650-765=6885$ lbs.—*i.e.* at one-third of the elastic limit for tension, and that therefore a strangling stress is severer on mild steel than a shearing stress.

The nature of fractures of different materials indicates that they do not all behave in the same way under compound stresses. Thus mild steel tension fractures are always much inclined to the axis except near the middle, whereas for hard steel the fractured surfaces are normal to it and crystalline. Under torsion stresses mild steel fractures are normal to the axis, while hard steel fractures show a helical surface. Apparently mild steel yields with comparatively greater ease to shear than to tension, whereas hard steel is more easily torn than sheared. For this reason it is useless to speculate on the discrepancies between Platt and Hargraves and J. J. Guest's ratio of tensile and shearing strengths.

Further experiments dealing with this very interesting subject are urgently required.

Until experiments have been made to determine the permissible compound stresses, it would be useless to speculate as to their action, but enough has been said to show that we are as yet groping in the dark. It is even impossible to say whether a spherical boiler end may be made half as thick as the cylindrical shell; for although theory shows that the stress is only one-half, it also shows that there are two equal stresses, which have been called drum tension.

Fatigue.—Boiler plates, except perhaps in locomotives, are not subjected to innumerable rapid changes of stress called fatigue stresses, but as the subject has a direct bearing on the lasting properties of materials it requires a brief mention. It is believed, but experiments on this subject are still too few for drawing definite conclusions, that if a material be subjected to a large number of varying stresses, it will break down if the difference between the high and the low stresses exceeds a certain limit, no matter what the individual stresses may be. Wöhler was the first to make exhaustive experiments on this subject, but until recently no satisfactory law of fatigue had been established, and some of the early deductions are not reliable. The author has shown ('C. E.,' v. 188, p. 38, and 'M. E.,' 1911, III., p. 887) that the following relation exists. $S = L + C \sqrt[4]{1 : n}$ where L is the elastic limit of fatigue, C is a constant and n is the number of stress cycles (revolutions) of the intensity $\pm$ S which a material will stand before it breaks. The value of C varies considerably for different materials both absolutely and relatively to L. L also varies from say 12 to 16 for mild steels, to 20 and even 25 for hard steels. Hardening either cast steel or mild steel does not seem to affect this limit materially. The subject, although of extreme importance to engineers generally, is of no direct importance to boiler makers unless, as may be possible, fatigue tests should be able to give more reliable information about the quality of materials than do the present tests.

CHAPTER VII

MECHANICS

An understanding of stresses first became possible when the laws of elasticity began to get known, but since then the subject has had such fascination for mathematicians, that it is now difficult for men with little leisure to follow their investigations, even although they have been most admirably collected and edited by Mr. K. Pearson (J. Todhunter and K. Pearson). Unfortunately, too, these volumes do not give the results of researches in such a form as to make them of value to practical men. The matter is unquestionably a complicated one, as can be seen by the few formulæ in this chapter, most of which it has been found impracticable to work out in detail, although the necessary hints or references are given. One object of going into these matters is to place in the hands of engineers the means of analysing and criticising the various restrictive rules which from time to time are evolved; the other and more important object is to place in the hands of engineers the means of analysing experiences and experiments notably for ascertaining the stresses when the strains or deformations have been measured. Attention has therefore been paid to the relation between stresses and strains, and as a consequence Poisson's ratio has often to be mentioned. Most text-books content themselves with dealing with elastic deformations, but in boilers, where plates sometimes grow red hot and plastic, and also in all experiments where the metal ruptures, it is of importance to bear in mind that we are not dealing with a purely elastic material, and that factors of safety which are based on ultimate strength are not all of the same standard. For this reason some remarks will be made about plasticity and ultimate strengths.

Elasticity.—The elasticity of solids as revealed by extensometers attached to test pieces has already been dealt with. On boilers and other solid structures these instruments would also reveal deformation when pressures are applied, and the deformations would be measures of the stresses to which the structures are subjected, provided that the conditions are properly analysed, which, as will be seen from the few formulæ in this chapter, are but a small fraction of the formulæ to be found in text books on elasticity, is not always an easy matter. Unfortunately, too, the few formulæ on this subject which find their way into pocket books are generally deficient in so far as they ignore Poisson's ratio, which, as will be seen later on, affects the deformations of plates, and of structures subjected to compound stresses. The modulus of elasticity will here be denoted by the letter E expressed in pounds per square inch. $1:E$ is the elongation or shortening of a bar at a unit of length when subjected to a unit of stress.

Shearing Elasticity.—It has been explained in connection with fig. 125, that a shearing stress is a compound of tension stress and compression stress. The shearing elasticity is the angular displacement of the two diagonals as compared with the shearing stress producing it. As the cross contraction due to the tension stress has to be added to the contraction stress and *vice versâ*, it is evident that under this compound stress the material is more elastic and the modulus of shearing elasticity σ is smaller than E, the correct formula being $\sigma = \mathrm{E} \cdot \frac{1}{2\left(1 + \frac{1}{\mu}\right)} = \frac{30{,}000{,}000}{2 \cdot 6} = 11{,}600{,}000$ lbs. per sq. inch.

Experiments on this subject are to be found in the following publications: Bauschinger, 'Civil Ing.,' 1879, vol. 25, p. 81; E. H. Amagat, 'Comp. Rend.,' 1888-9, vol. 107-108; C. E. Stromeyer, 'Proceed.,' 1894, vol. 55, p. 377; J. Morrow, 'Phil. Mag.,' 1903, vol. 6; J. R. Benton, 'Physical Review,' 1901, vol. 12; P. Cardani, 'Physicalische Zeitschrift,' 1903, vol. 6; E. Grüneisen, 'Wied. Ann.,' 1908, vol. 25, p. 825.

The elastic constants for the more important metals and a few others are contained in the following table:—

Modulus of Material	Elasticity = E Tons Per Sq. Inch	Shearing Elasticity = E Tons Per Sq. Inch	Poisson's Ratio = $\frac{1}{\mu}$
Wrought Iron } from	13,000	4,500	0·31
Mild and Hard Steel } to	13,800	5,300	
Grey Cast Iron { from	4,500	—	—
Grey Cast Iron { to	8,000	—	—
Gun-metal, about	5,250	1,900	0·36
Brass, about	5,000	1,700	0·35
Copper, about	8,300	3,000	0·34
Nickel, about	13,000	5,000	0·31
Aluminum, about	4,600	1,700	0·34
Magnesium, about	2,600	1,000	—
Glass, about	4,500	—	—

Flexibility, or bending elasticity, is best reduced to E. As will shortly be explained, here also the co-efficient of cross contraction has an important influence.

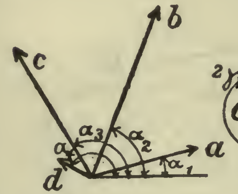
Plasticity.—The elastic limit and breakdown point have already been explained (p. 152).

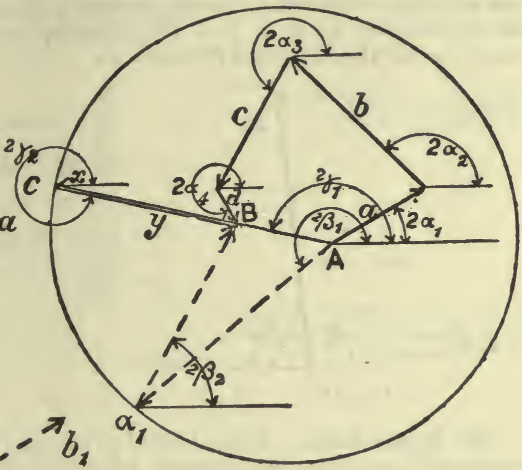
Resolution of Stresses.—Investigations in statics show that several forces acting through one point can always be replaced by a single resultant, and similarly several stresses can be replaced by resultants; but there will be three instead of one, unless they are all acting parallel to one plane, and in that case there will be two resultant stresses as against one resultant force. To resolve stresses which are irregularly distributed in space is a problem which need not be discussed in this chapter, particularly as it leads to rather complicated formulæ (Rankine, 'R. Soc. Edinburgh,' vol. xxvi. p. 715).

The Resolution of Stresses parallel to one plane is comparatively simple.

Let there be four stresses, *a*, *b*, *c*, *d* (fig. 134), acting in the directions α_1, α_2, α_3, α_4. Double each of these angles and construct the polygon A, *a*, *b*, *c*, *d*, B (fig. 135). The points A, B may then be looked upon as the two foci of an ellipse, the sum of the radii vectores A, α_1, B being equal to the sum of the stresses. By halving the angles $2\beta_1$ and $2\beta_2$, and drawing the stresses a_1 and b_1 from a point (fig. 136), a system of two resultants is obtained which would

FIG. 134





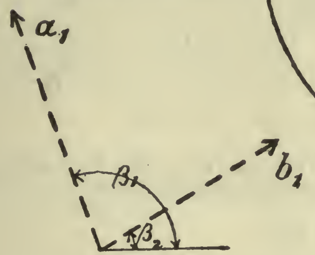


FIG. 136 FIG. 135

produce exactly the same strain as the four original stresses; but they are not necessarily at right angles to each other. In order to fulfil this condition, radii vectores ACB (fig. 135) will have to be drawn parallel to the major axis, and as $2\gamma_2 = 2\gamma_1 + 180°$ the resultants will be at right angles to each other, as shown in fig. 137. Their intensities are represented by the lengths AC $= x$ and CB $= y$.

The following is the algebraical solution of this question.

The notations are the same as in the figures.

As regards the angles we have—

$$\text{AB} \cdot \sin 2\gamma_1 = a \sin 2\alpha_1 + b \sin 2\alpha_2 + \&\text{c.}$$
$$\text{AB} \cdot \cos 2\gamma_1 = a \cos 2\alpha_1 + b \cos 2\alpha_2 + \&\text{c.}$$
$$\tan 2\gamma_1 = \frac{a \sin 2\alpha_1 + b \sin 2\alpha_2 + \&\text{c.}}{a \cos 2\alpha_1 + b \cos 2\alpha_2 + \&\text{c.}}$$

Divide $2\gamma_1$ by 2; this determines γ_1; and add 90°, which is the angle γ_2.

As regards the stresses we have—

$$x = \frac{(a + b + \&\text{c.})}{2} + \frac{\text{AB}}{2} = \frac{\Sigma(\text{S}) + \sqrt{(\Sigma\text{S} \cdot \sin 2\alpha)^2 + (\Sigma\text{S} \cdot \cos 2\alpha)^2}}{2}$$
$$y = \frac{(a + b + \&\text{c.})}{2} - \frac{\text{AB}}{2} = \frac{\Sigma(\text{S}) - \sqrt{(\Sigma\text{S} \cdot \sin 2\alpha)^2 + (\Sigma\text{S} \cdot \cos 2\alpha)^2}}{2}.$$

For a combination of two axial stress X($\alpha = 0$), Y($\alpha = 90°$) with one axial shearing stress σ($\alpha = 45°$ and 135°) we have:

$$\tan 2\gamma = 2\sigma : (\text{X} - \text{Y}) ; \; x = \tfrac{1}{2}(\text{X} + \text{Y}) + \sqrt{\tfrac{1}{4}(\text{X} + \text{Y})^2 + \sigma^2};$$
$$y = \tfrac{1}{2}(\text{X} + \text{Y}) - \sqrt{\tfrac{1}{4}(\text{X} + \text{Y})^2 + \sigma^2}$$

If some of the stresses are positive and the others negative, i.e. tension or compression, they should be placed in proper order in the polygon (fig. 135). But then instead of an ellipse, a hyperbola will be the boundary line. It can also be proved that there will only be one resultant to several shearing stresses. Thus, if in fig. 134 they are represented by a, b, c, d, the intensity of the resultant stress would be AB (fig. 135) and its angle γ_1.

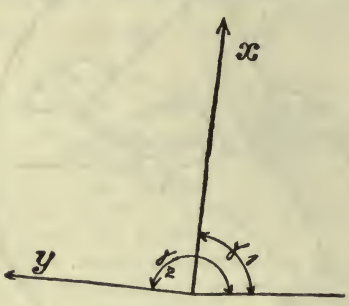


FIG. 137

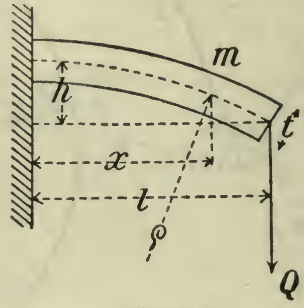


FIG. 138

The Elastic Beam.—When loaded at one end (fig. 138) the bending moment of a beam, at the distance x from its attachment, is

$$m = Q \cdot (l - x).$$

If the longitudinal stresses at this point vary from $-$ S at the bottom fibres to $+$ S at the top fibres, they combine to form a resisting moment which is equal to m, which for a rectangular section is :

$$m = \frac{S \cdot t^2 b}{6}.$$

Here b is the breadth of the beam, and t its thickness. For a round bar of the diameter d we have

$$m = \frac{\pi \cdot S \cdot d^3}{32}, \text{ or nearly } \frac{S \cdot d^3}{10}.$$

In flat plates b can be taken as unity, provided the pressures and forces are measured per unit of width.

The Radius of Curvature ρ can be determined from the following formula :

$$\frac{1}{\rho} = \frac{d^2 y}{dx^2} = \frac{m}{E \cdot I} = \frac{2 \cdot S}{t \cdot E}.$$

Here x and y are the co-ordinates of any point of the elastic line, E is the modulus of elasticity, and I is the moment of inertia of the section $\left(I = \frac{t^3}{12} \text{ for a flat beam 1 in. wide}\right)$. When $\frac{d^2 y}{dx^2}$ is constant we have $\frac{1}{\rho} = \frac{2 \cdot h}{l^2}$, and also $S = \frac{E \cdot t}{2 \cdot \rho} = E \frac{h \cdot t}{l^2}$.

Cross Curvature.—Besides bending lengthways, the edges of the beam tend to curl up (fig. 139); this is due to the cross contraction of the upper side caused by longitudinal tension, and the cross extension of the lower side caused by longitudinal compression. If the radius of this cross curvature be called ρ_1, then

$$\frac{1}{\rho_1} = \frac{1}{\rho} \cdot \frac{1}{\mu}$$

where $\frac{1}{\mu}$ is the co-efficient of cross contraction. In wide plates, which cannot curl crossways when bent, or in such bent plates as are constrained to remain flat crossways, it is evident that

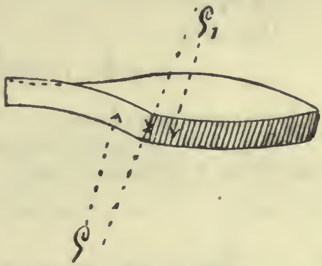


FIG. 139

there are internal or external forces at work reducing the cross curvature to a straight line, viz. $\rho_1 = \infty$, which again means that there are cross stresses C due to bending:

$$C = \frac{E \cdot t}{2 \cdot \rho_1} = \frac{E \cdot t}{2 \cdot \mu \cdot \rho}$$

Also that the longitudinal bending has been reduced, and ρ has been increased to ρ_2.

$$\frac{1}{\rho_2} = \frac{1}{\rho} + \frac{1}{\mu \cdot \rho_1}$$

The formulæ for a wide beam which remains straight axially are:

$$\frac{1}{\rho_2} = \frac{2}{t \cdot E} \cdot \left(S - \frac{C}{\mu}\right) = \frac{2 \cdot S}{t \cdot E} \cdot \left(1 - \frac{1}{\mu^2}\right)$$

$$S = \frac{E \cdot t}{2 \cdot \rho_2} \cdot \left(\frac{\mu^2}{\mu^2 - 1}\right) = E \cdot \frac{h \cdot t}{l^2} \cdot \left(\frac{\mu^2}{\mu^2 - 1}\right) = E \cdot \frac{h \cdot t}{l^2} \cdot 1{\cdot}1$$

$$C = \frac{S}{\mu} = E \cdot \frac{h \cdot t}{l^2} \cdot \left(\frac{\mu}{\mu^2 - 1}\right) = E \cdot \frac{h \cdot t}{l^2} \cdot 0 \cdot 33.$$

Compound Curvatures.—If the bent beam is not straight axially, then the stresses, as estimated from the curvatures, have to be combined as follows. Let the radii of the two curvatures be ρ_1 and ρ_2, then, always assuming $\frac{1}{\mu} = 0 \cdot 3$ we have:

$$S_1 = \frac{E \cdot t \cdot 1{\cdot}1}{2 \cdot \rho_1}, \text{ and } C_1 = \frac{E \cdot t \cdot 0{\cdot}33}{2 \cdot \rho_1}$$

$$C_2 = \frac{E \cdot t \cdot 1{\cdot}1}{2 \cdot \rho_2}, \text{ and } S_2 = \frac{E \cdot t \cdot 0{\cdot}33}{2 \cdot \rho_2}.$$

By addition we get:

$$S = S_1 + S_2 = E \cdot t \cdot 0{\cdot}55 \cdot \left(\frac{1}{\rho_1} + \frac{0{\cdot}3}{\rho_2}\right).$$

$$C = C_1 + C_2 = E \cdot t \cdot 0{\cdot}55 \cdot \left(\frac{1}{\rho_2} + \frac{0{\cdot}3}{\rho_1}\right).$$

If ρ_2 has the negative value $-\frac{\rho_1}{\mu} = -0.3 . \rho_1$, then $C = 0$ and $S = \frac{E . t}{2 . \rho_1}$, as already found.

If $\rho_2 = 0$,

$$S = E . t . 0{\cdot}55 . \frac{1}{\rho_1};\ C = E . t . 0{\cdot}165 . \frac{1}{\rho_1};$$

this has also been found.

If $\rho_2 = +\rho_1$, as for instance in a flat end plate, when elastically acquiring a spherical shape,

$$S = C = E . t . 0{\cdot}715 . \frac{1}{\rho_1}.$$

In this last case, where of course the two principal stresses are equal, a drum tension exists on the convex side of the plate, and a shrinking stress on the concave side (see p. 160); but, as the latter will most likely injure the material more seriously than a simple tension (see p. 163), it is very important that all measurements of curvature made for the determination of stresses should be taken both lengthways and crossways. Conversely, a case might occur where $\rho_1 = -\rho_2$; then $S = \frac{E . t}{2 . \rho} . \frac{\mu}{\mu + 1}$, showing that the stresses are smaller than usually supposed. Combined in this way the stresses constitute a shearing stress of a very peculiar nature, being directed in opposite directions on either side of the plate. This condition is found to exist along the pitch lines of screwed stays, near the centres of the pitches. The cross curvatures of beams naturally affect their deflections, and in this fact is to be sought the explanation why these curvatures do not agree with what (so-called) theory makes them out to be.

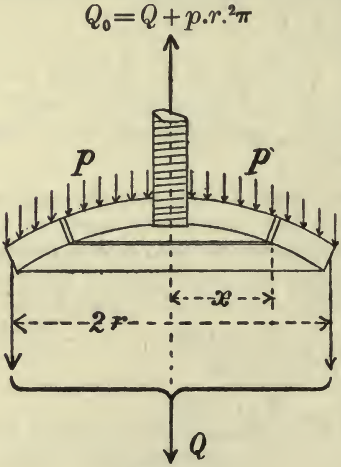


Fig. 140

Stresses in Flat Plates.—The formulæ which would have to be evolved in analysing these stresses are very complicated, even when dealing only with circular discs, and as such results are of very limited value for these investigations, and as analyses of square and rectangular plates leave out of account some important points, a short explanation as to the methods adopted is all that can be attempted here. Fig. 140 is a section through the centre of a circular disc of the diameter $2r$ and of the thickness t. It is loaded with a pressure p over its entire surface, and with a load Q spread over its circumference, and is supported by a central stay. Certain strains and stresses will be produced, which it is desired to determine.

Imagine the disc to be subdivided by a large number of radial lines; then one of the slices so formed could be represented by fig. 141. Its angle is β. It is probable that radial bending moments exist, and if the maximum radial tension stress of the upper fibres at the distance x is denoted by S the resisting moment will be

$$m_1 = \frac{S \cdot x \cdot \beta \cdot t^2}{6}.$$

If now the disc is subdivided into a number of concentric cylinders the thickness of whose walls is dx (figs. 142, 144), it will be found that under stress they have lost their original cylindrical shape, and have grown slightly conical. Let this angle be α. It is clear that

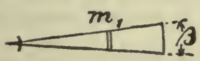


FIG. 141

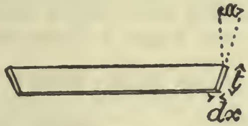


FIG. 142

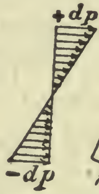


FIG. 143

FIG. 144

circumferential stresses exist in these rings, tension at the upper surface, and compression at the lower ones. Let the maximum value of the former be C. These stresses could also have been produced by an imaginary internal fluid pressure, ranging from $+ dp$ (fig. 143) at the upper surface to $-dp$ at the lower.

If now a short length, $\beta . x$, of the ring be examined, the tilting power of this imaginary fluid pressure would be found to be

$dm_2 = dp \cdot x\,\beta \cdot \frac{t^2}{6}$; but, as $- C \cdot dx = + dp \cdot x$, we have

$$-\frac{dm_2}{dx} = \frac{C \cdot \beta \cdot t^2}{6}.$$

The intensities of S and of C can, therefore, be expressed in terms of m_1 and m_2. The relation existing between these moments is determined by the deformations of the disc.

The differential equation for the elastic line is $\frac{d^2y}{dx^2} = \frac{m_1}{E \cdot I}$, as long as there are no cross stresses; but, as the circumferential tension C tends to shorten the radial fibres, the coefficient of cross contraction, $\frac{1}{\mu}$, must be introduced, and the equation of the elastic line is

$$\frac{d^2y}{dx^2} = \frac{2}{t \cdot E}\left(S - \frac{C}{\mu}\right).$$

α, the angle of the cone, is equal to $\frac{dy}{dx}$, which, by a similar reasoning to that of the previous case, is

$$\frac{dy}{dx} = \frac{2 \cdot x}{t \cdot E}\left(C - \frac{S}{\mu}\right).$$

Differentiating this, we get

$$\frac{d^2y}{dx^2} = \frac{2}{t \cdot \mathrm{E}}\left(\mathrm{C} - \frac{\mathrm{S}}{\mu} + x \cdot \frac{d\mathrm{C}}{dx} - \frac{x}{\mu} \cdot \frac{d\mathrm{S}}{dx}\right).$$

Combining this with the previous value for $\frac{d^2y}{dx^2}$, it disappears, and we get

$$\mathrm{O} = (\mathrm{C} - \mathrm{S})\left(1 + \frac{1}{\mu}\right) + x\left(\frac{d\mathrm{C}}{dx} - \frac{1}{\mu} \cdot \frac{d\mathrm{S}}{dx}\right).$$

This is the fundamental equation as regards the relation of the two stresses C and S. These have now to be compared with the external forces producing them.

For simplicity's sake, let the perforated plate (fig. 140) be suspended by a stay of the diameter d, and uniformly loaded only at its circumference by Q. Dealing once more with a narrow slice of the angle β, it will be found that at the point x there exists a vertical shearing force $q = \frac{\mathrm{Q} \cdot \beta}{2 \cdot \pi}$, which acting on the lever $x - \frac{d}{2}$ produces a moment $m_0 = \frac{\mathrm{Q} \cdot \beta}{2 \cdot \pi}\left(x - \frac{d}{2}\right)$, and therefore $\frac{dm_0}{dx}\frac{\mathrm{Q} \cdot \beta}{2 \cdot \pi}$

The moment due to the radial stresses is

$$m_1 = \frac{\beta \cdot t^2}{6} \cdot x \cdot \mathrm{S}, \text{ and } \frac{dm_1}{dx} = \frac{\mathrm{B} \cdot t^2}{6}\left(\mathrm{S} + x\,\frac{d\mathrm{S}}{dx}\right).$$

The elementary moment, dm_2, produced by the circumferential stresses C, has to balance $dm_0 + dm_1$.

$$-\,dm_2 = \frac{\beta \cdot t^2}{6} \cdot \mathrm{C} \cdot dx \,. = \left\{\frac{\beta \cdot t^2}{6}\left(\mathrm{S} + x\,\frac{d\mathrm{S}}{dx}\right) + \frac{\mathrm{Q} \cdot \beta}{2 \cdot \pi}\right\} dx,$$

from which it follows that

$$\mathrm{C} = \mathrm{S} + x \cdot \frac{d\mathrm{S}}{dx} + \frac{\mathrm{Q} \cdot 3}{\pi \cdot t^2}.$$

By differentiating and substituting the values of C and $\frac{d\mathrm{C}}{dx}$ in the previous equation an expression is obtained containing only S and x as variables. Integrating this twice, S is found, and C can then also be determined for any point, as well as the deflections and inclinations.

The same operation has to be carried out with a uniformly distributed pressure p, and also with an unperforated plate; but the resultant formulæ, as already mentioned, are too long to be reproduced here, and have not been practically applied. See Grashoff, 1866, pp. 248, 254, 263.

C. Bach ('Deut. Ing.,' 1897, vol. xli. p. 158) gives details of tests on flat plates, 28 ins. diameter, and from $\frac{3}{8}$ in. to $\frac{3}{4}$ in. thick. Permanent sets were noticed very early, but they did not exceed $\frac{1}{100}$ in. with the thin plates until a pressure of 100 lbs. had been reached: for the thick plates the pressure had to be 330 lbs., also ('Deut. Ing.,' 1908, vol. lii. p. 1876), two square plates 31·5 ins. between edge rows of rivets and one rectangular plate 31·5 × 15·8 ins. of respectively 0·315, 0·63, and 0·86 in. thickness. The first permanent sets were noticed with 15, 27, and 49 lbs. per square inch. The maximum pressures were 350, 300, and 400 lbs. None of the plates failed.

R. Wilson ('Enging.,' 1877, vol. xxiv. p. 239) gives the results of five experiments on unstayed flat and dished boiler ends, the diameter being 30 ins., and the thicknesses of end plates $\frac{3}{8}$ in. and $\frac{9}{16}$ in. They were either flat and riveted to angle irons, or flanged. The bursting pressures varied from 200 to 370 lbs. per square inch.

Elastic Beams.—A fair idea as to what happens in stayed plates may be arrived at by examining the so-called continuous beams, which, like railway rails, are supported at more than two points. The analysis is carried out as follows:—

Each span (fig. 145) should be examined separately. There will be the uniformly distributed pressure p, and the supporting forces

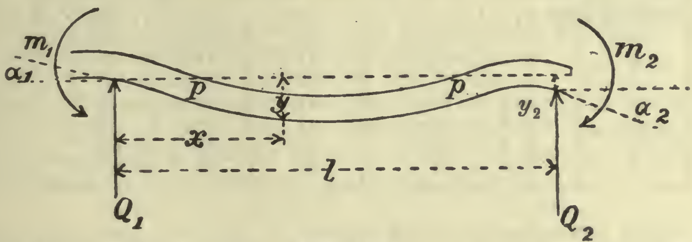


FIG. 145

$Q_1 + Q_2$. There will also be the two external moments m_1 and m_2, and the deflections, a_1 and a_2, and the depression y_2. Any of these values may be zero.

The differential equation for the elastic line is $\frac{1}{\rho} = \frac{d^2y}{dx^2} = m : \mathrm{EI}$, where I is the moment of inertia of the section of the beam, for a plate of the width 1 and the thickness t it is $\frac{1}{12}t^3$. Integrating the above equation between the limits x, $\frac{1}{2}l$, and l the values of the inclinations α and the deflections y are obtained.

Conditions $\alpha_1 =$	α	0	0	0
$m_2 =$	0	m	0	0
$Q_2 =$	0	0	Q	0
$p =$	0	0	0	p
Inclination α at x	α	$mx : \mathrm{EI}$	$-\frac{1}{2}Qx(l - \frac{1}{2}x) : \mathrm{EI}$	$\frac{1}{2}px(l^2 - lx + \frac{1}{3}x^2) : \mathrm{EI}$
,, α at $\frac{1}{2}l$	α	$\frac{1}{2}ml : \mathrm{EI}$	$-\frac{3}{8}Ql^2 : \mathrm{EI}$	$0{\cdot}146\,pl^3 : \mathrm{EI}$
,, α at l	α	$ml : \mathrm{EI}$	$-\frac{1}{2}Ql^2 : \mathrm{EI}$	$\frac{1}{6}pl^3 : \mathrm{EI}$
Deflection y at x	αx	$\frac{1}{2}mx^2 : \mathrm{EI}$	$-\frac{1}{2}Qx^2(l - \frac{1}{3}x) : \mathrm{EI}$	$\frac{1}{4}px^2(l^2 - \frac{2}{3}lx + \frac{1}{6}x^2) : \mathrm{EI}$
,, y at $\frac{1}{2}l$	$\frac{1}{2}\alpha l$	$\frac{1}{8}ml^2 : \mathrm{EI}$	$-0{\cdot}104\,Ql^3 : \mathrm{EI}$	$0{\cdot}0443\,pl^4 : \mathrm{EI}$
,, y at l	αl	$\frac{1}{2}ml^2 : \mathrm{EI}$	$-\frac{1}{3}Ql^3 : \mathrm{EI}$	$\frac{1}{8}pl^4 : \mathrm{EI}$

As an example determine in a continuous beam the bending moment $m = m_1 = m_2$ over the supports. For $y = l$ we have $\alpha = \alpha_1 = \alpha_2 = 0$ and $y = 0$.

$$\mathrm{EI}\alpha = 0 = ml - \tfrac{1}{2}Ql^2 + \tfrac{1}{6}pl^3;\ m = \tfrac{1}{2}Ql - \tfrac{1}{6}pl^2.$$
$$\mathrm{EI}y = 0 = \tfrac{1}{2}ml^2 - \tfrac{1}{3}Ql^3 + \tfrac{1}{8}pl^4;\ m = \tfrac{2}{3}Ql - \tfrac{1}{4}pl^2.$$

From these equations it follows that $Q = \frac{1}{2}pl$ and that $m = \frac{1}{12}pl^2$. The above formula for beams can be applied to flat plates by taking into account the tendency to curl up crossways (cross curvature) by replacing E by $E\frac{\mu^2}{\mu^2 - 1}$ or approximately by 1·1E. The formulæ will also be required for estimating deflection of steam pipes. They have to be combined with similar ones for curved beams, see later.

Girders.—Let L be the total length of the girder, while l and n are the pitch and number of stays; then $L = (n + 1) . l$ is double the distance of the end stays from the girder ends. If w is the distance apart of the girders, and p the pressure in the boiler, then the maximum bending moment in the girder is m in the following table :—

No. of Stays	Value of $\frac{4 . m}{p . w}$	No. of Stays	Value of $\frac{4 . m}{p . w}$
1	$1 . l . L$	9	$9 . l . L - 40 . l^2$
2	$2 . l . L - 2 . l^2$	10	$10 . l . L - 50 . l^2$
3	$3 . l . L - 4 . l^2$	11	$11 . l . L - 60 . l^2$
4	$4 . l . L - 8 . l^2$	12	$12 . l . L - 72 . l^2$
5	$5 . l . L - 12 . l^2$	13	$13 . l . L - 84 . l^2$
6	$6 . l . L - 18 . l^2$	14	$14 . l . L - 98 . l^2$
7	$7 . l . L - 24 . l^2$	15	$15 . l . L - 112 . l^2$
8	$8 . l . L - 32 . l^2$	16	$16 . l . L - 128 . l^2$

$S = \frac{6 . m}{h^2 . t} = \frac{3}{2} . \frac{p . w}{h^2 . t}\left(\frac{4 . m}{p . w}\right)$. Here h and t are respectively the height and thickness of the girder.

In the above formulæ the product $n . l$ is the length of the plate whose load is borne by the girder.

For even numbers of stays, $S = \frac{3}{2}\frac{pw}{h^2t}nl(L - \frac{1}{2}nl)$.

For odd numbers of stays, $S = \frac{3}{2}\frac{pw}{h^2t}nl\left(L - \frac{1}{2}nl + \frac{1}{2}\frac{l}{n}\right)$.

These formulæ harmonise with the Board of Trade rules if $L = (n + 1)l$, but generally L is somewhat longer, because the flanges of the plates permit of a greater distance than l between the foot of the girder and the first stay. The girder stresses which are customary with marine boilers are very much exceeded in locomotives. Girders of two $\frac{1}{2}$ plates, 4 ins. deep spanning fireboxes of 36 ins. between plates, and placed 4 ins. apart with 8 stays of 4 ins. pitch have been fitted to many locomotive boilers of 180 lbs. pressure and have been perfectly safe. The stress according to the above formula works out at 19·3 tons per sq. inch! On recalculating the stress on the assumption that the firebox plate, which was $\frac{3}{8}$ in. thick, with an inch space between it and the girder, formed part of this girder, the stress works out at 5·5 tons per sq. inch; but if this view be the correct one, the shearing forces between the plates and the girder should be taken into account. They amount to 15·3 tons for the outer stays and 4·3 tons for the next ones, provided, as has been assumed, that the plate and girders are rigidly secured to each other by the stays and ferrules. It cannot, however, in this case, be admitted that the stays, which were only $\frac{7}{8}$ in. diameter were equal to the above-mentioned loads. Probably they both bent and sheared a little, and thus increased the 5·5 tons bending stress by an unknown amount. If the end stays are made sufficiently strong to resist the shearing force in addition to the load due to the steam pressure, girders might unquestionably be made much lighter than at present. The hydraulic tests would overstrain these girders and reduce the elastic stresses to plastic ones of 12·9 tons (see p. 189).

The shearing force on the outer stay is found by multiplying the average stress in the plate between the outer stay and its neighbour into the sectional area of the plate ($w \times t_1$). The shearing force on the two outer stays is found in the same way, and the load on the second stay is found by difference. The average stress in the plate is equal to the maximum stress in the girder at one or the other of the stays multiplied by the ratio $v : x$ in fig. 145a.

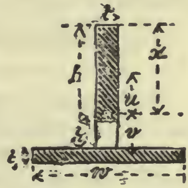


FIG. 145a

The moment of inertia of the rigid combination of a girder of the heights H and a plate of the width w as shown in figure is the sum of the moments of inertia of the two sections plus the product of the areas $\mathrm{H}t$ and wt_1 into the squares of the distances u and v of the respective centres of gravity of sections from the centre of gravity of the whole. The distance x is found with the help of the formula—

$$x(\mathrm{H}t + wt_1) = (\tfrac{1}{2}\mathrm{H}^2 t + (\mathrm{H} + h + \tfrac{1}{2}t_1)wt).$$

Then u and v are found by difference, and the moment of inertia is $\mathrm{I} = \frac{1}{12}(\mathrm{H}^3 t + t_1{}^3 w) + u^2\mathrm{H}t + v^2 wt_1$. The resisting moment is $\mathrm{SI} : x$, where S is the maximum stress in the top fibre of the girder.

Stayed Flat Plates.—The formulæ which were worked out for the continuous beam are not directly applicable to stayed flat plates; they only give average results for the whole width of such plates. They also do not take into account the cross bending. Some idea of the stresses in these plates will be gained by the following view. Let it be assumed that the boiler pressure is halved, and that one

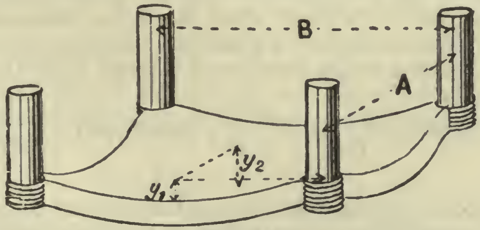


FIG. 146

half is active in producing bending along the pitch line B, fig. 146, while the other half produces bending along the pitch line A. The mean bending moment in either direction is then

$$m_{\mathrm{A}} = \frac{p \cdot \mathrm{B}}{2} \cdot \left(\frac{\mathrm{A}^2}{12} - \frac{\mathrm{A} \cdot x}{2} + \frac{x^2}{2}\right)$$

$$m_{\mathrm{B}} = \frac{p \cdot \mathrm{A}}{2} \cdot \left(\frac{\mathrm{B}^2}{12} - \frac{\mathrm{B} \cdot x}{2} + \frac{x^2}{2}\right).$$

On account of the easy elastic curvature of stayed plates along lines which pass half way between the stays, these bending moments are probably reduced to half of what they would be in a continuous beam, viz.:

$$m_A = \frac{p \,.\, B \,.\, A^2}{2 \,.\, 2 \,.\, 24}\,;\; m_B = \frac{p \,.\, A \,.\, B^2}{2 \,.\, 2 \,.\, 24}.$$

Now, according to Mr. Guest's experiments (p. 170), the presence of a cross tension stress does not affect the strength of the material, so that on the tension side of the plate we could allow two cross proof stresses of, say,

$$25{,}000 = 6 \,.\, \frac{M_A}{B \,.\, t^2} = \frac{p \,.\, B^2}{16 \,.\, t^2}\,;\text{ also } 25{,}000 = \frac{p \,.\, A^2}{16 \,.\, t^2}\,;$$

This is a very high result, and would indicate, as would be imagined, that stayed plates are not weakest in the spaces between the stays. If we take the values deduced from Professor C. Bach's experiments (p. 171) as being correct, we must admit that plastic bending due to double compression on one side will take place if the stress exceeds say 6,000 lbs. per square inch:

$$6{,}000 = \frac{p \,.\, A^2}{16 \,.\, t^2},\text{ or } \frac{p \,.\, B^2}{16 \,.\, t^2}.$$

This is also a very high value.

If now we look at the plate close to the stays, we may safely assume that, because of the concentration of forces at these points, the values of m at the supports as found above would not be reduced by half, but probably increased twofold.

$$m_A \,. = 2 \,.\, \frac{p \,.\, B \,.\, A^2}{2 \,.\, 12} = \frac{p \,.\, BA^2}{12}\text{ and } m_B = \frac{p \,.\, A \,.\, B^2}{12}.$$

As the plates are perforated, and as the screwed stays cannot be assumed to be part of the plate, there can be no radial stresses at the circumference of the holes, and we have only simple stresses, which, up to the proof pressure, might be allowed to reach 25,000 lbs. per square inch.

$$25{,}000 = \frac{6 \,.\, m_A}{B \,.\, t^2} = \frac{p \,.\, A^2}{2 \,.\, t^2}\,;\text{ also } 25{,}000 = \frac{p \,.\, B^2}{2 \,.\, t^2}.$$

This indicates that the stresses near the stays are about eight times as severe as between the stays, though, if the deductions drawn from Professor Bach's experiments are correct, the material, being more plastic under the conditions near the centres of the plates, would only be about 100 per cent. stronger than near the stays. If these views are correct the working pressure of a stayed flat plate would be $p = 25{,}000t^2 : A^2$, all dimensions being expressed in inches. If the thicknesses are expressed in sixteenths of an inch, then $p = 97{\cdot}5t^2 : A^2$, and the first indication of giving way would be at about double this pressure.

That even this result is not quite correct will be admitted when we consider the question of size of stay. Assuming that the screwed plate is not supported by screwed stays, but is resting on supports, then we may distinguish three cases. Firstly, the plate over the supports is not perforated. We have severe drum tension on the one side and severe strangling stress on the other. Secondly, if we drill a very small hole just over the support the compound stresses give way to single tension and compression stresses of double intensity, for this is a case very similar to that of a thick walled cylinder of infinite external diameter. Thirdly, if the holes are made large, the diameters of the supports being of course proportionately increased, these simple stresses will be much reduced. Therefore, the size of the stay is an important factor, all the more because of the shearing stress set up in the hole. This is fully borne out by experiments, and published rules have followed this up in so far as that they only allow a low working stress for screwed stays, and that they allow higher working pressures to plates supported by stays with nuts and washers. In such cases the working stress on the stays might also be increased, and, on the other hand, allowances might be made in the thickness of the plate when the screwed stays without washers are of very large diameter—as, for instance, when these stays are tubes.

Irregularly Stayed Plates.—By applying the formulæ of continuous beams to irregularly stayed plates—for instance, such as occur in boiler backs (fig. 147)—it can be shown that the stresses near

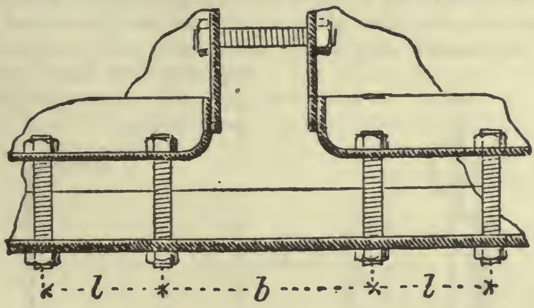


FIG. 147

the stays on either side of the pitch b are greater than the usually accepted formulæ would lead one to expect. The value of m_1 has to be increased from

$$m_1 = \frac{p \cdot l^2}{12} \text{ to } m_1 \frac{p \cdot b^3}{l \cdot 12}.$$

It is exceedingly difficult to arrive at a satisfactory view of the case illustrated in fig. 148, where the pitches A and B are not equal. The stresses set up parallel to the pitches A will certainly be affected by the pitch B, for the more this is reduced the less metal will remain between the stays, and in the extreme case when they nearly touch each other, i.e. when $B = d$, the stress S_1, due to the curvature at the line of holes, would be increased to $\frac{2 \cdot p \cdot l^3}{d \cdot t^2}$: and comparing this with the above value S, for such cases when A and B are equal, we find that $\frac{S_1}{S} = \frac{4}{3} \cdot \frac{A}{d}$.

On the other hand, when A and B are nearly equal, a slight reduction of B will tend towards a more uniform distribution of the stresses between the stays, thereby reducing the maximum values of S; and it is possible that an alteration of B may in the one case reduce, and in the other case increase, the maximum stresses.

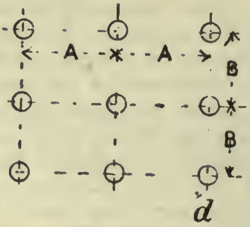


FIG. 148

Diagonal Pitches.—The Board of Trade has adopted the view that the maximum stresses are proportional to the product of A into B, but certain allowances are made for large variations (see Rules). Formerly 'Lloyd's Register' did not take into account the smaller of the two pitches, but since the first edition of this work was issued the view expressed in it, that the stresses in flat plates are probably proportional to the sum of the squares of the two pitches, has been accepted. This amounts to the same thing as measuring the diagonal pitch and doubling the accepted constants. For irregularly stayed plates the practice commonly followed on land boilers is to inscribe circles which will touch the rivet centres or stays: the diameters of these circles are taken to be the diagonals of squares. It should also be noted that on land considerably higher pressures than at sea are permitted on flat plates of the same dimensions. This is due to the experience that thick plates frequently crack, owing to panting; yet there is no indication that these practices lead to weak boilers.

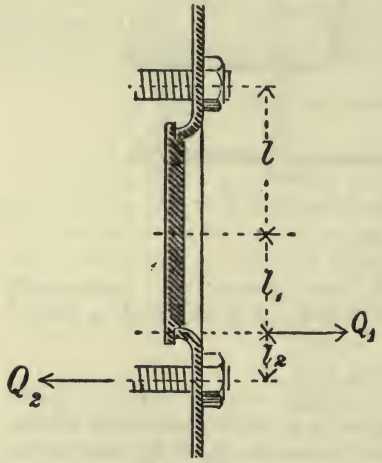


FIG. 149

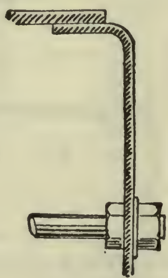


FIG. 150

Manhole Flanges.—A problem which presents itself with reference to the fitting of manholes in flat plates is the following (see fig. 149):—

Let $Q_1 = p \cdot l_1$, where p is the working pressure and l_1 is half the width of the manhole door; then the bending moment at the position of the stay is

$$m = p \cdot l_1 \cdot l_2 + \frac{p \cdot l_2^2}{2}$$

If experience has shown that for this particular thickness of plate a pitch of stays equal to L is permissible, then evidently the following equation must be true:—

$$\frac{p \cdot L^2}{12} = p \cdot \left(l_1 \cdot l_2 + \frac{l_2^2}{2} \right).$$

So that the maximum value for $l = l_1 + l_2$ is

$$l = l_1 \sqrt{1 + \frac{1}{6} \cdot \frac{L^2}{l_1^2}}.$$

There L is the pitch for stays in flat plates as found by accepted rules. The manhole doors must be estimated independently. Valuable experiments have been made as to their strengths by the German Admiralty, Danzig.

Effects of Boiler Deformations.—A little familiarity with the working of the equations for a and y of continuous beams (see p. 173) will soon lead to the conviction that the changes of form to be expected in boilers often produce stresses which far exceed any that may be due to the steam pressure alone. As a simple instance take the case of a long double-ended boiler, a steam-space stay placed too close to the shell plate (fig. 150). The stay's duty is to support the surrounding flat plate; but, on account of the longitudinal contraction of the shell, the stay will be in compression and actually assist the steam pressure in forcing out the plate, thereby seriously increasing the load on the adjoining stays, and increasing the bending stresses in the flange.

Screwed Stays near Flanges.—The reverse action is met with at the edges of the combustion-chamber backs. Evidently there is a pull in one direction at the flange, which is balanced by the pull on the stay (see fig. 151). This load Q_1 is proportional to the mean of the

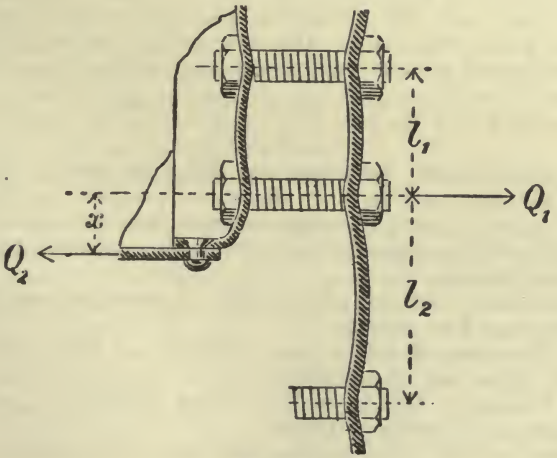


FIG. 151

pitches l_2 and l_1, and if the latter dimension is the maximum which practice has shown to be safe, then the distance x, measured from the centre of the flange, is found by the formula

$$x = -\frac{a}{2} + \sqrt{\frac{a^2}{4} + \frac{l_1^2}{6}},$$

where a is the width of the water-space between the two combustion chambers.

Stayed Concentric Rings.—It sometimes happens that a donkey boiler furnace has to be strengthened by radial stays (fig. 151a), and the question arises what is the safe working pressure of the stayed boiler, and what are the stresses in the plates and stays. Assume the pressure p in the annular space to consist of two pairs of partial ones of which one pair p_3, p_4 acts on the outer shell, and the other p_1, p_2 on the inner shell. The pressure p_1 reduces the diameter of the furnace shell, p_3 increases that of the outer one, the consequent separation of the two shells is Δl, where l is the length of the stays. The pressure p_2 acts on a rectangle of the inner shell supported by one stay, p_4 acts on a similar rectangle of the outer shell. The pull on the stay must, of course, be the same in both directions, and as the areas of the rectangles are proportional to the radii r_1 and r_2 of the two shells, we have $\frac{p_4}{p_2} = \frac{r_1}{r_2}$ where r_1 is the radius of the inner shell and r_2 that of the outer. Let h be the vertical pitch of the rows of stays, and n the number of stays in the circumference, then the pull on the stay is $p_2 h \frac{r_1 2\pi}{n}$. This pull elongates the stay, and this elongation must be exactly equal to the separation of the two shells. It is now an easy matter to determine the several partial pressures and the stresses which they produce.

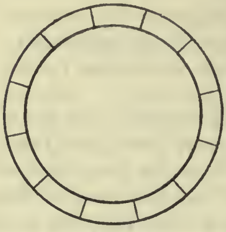


FIG. 151a

$$p_3 = p\left(\frac{r_2^2}{t_2} + \frac{r_1^2}{t_1}\right) : \left(\frac{r_2 r_1}{t_2} + \frac{r_1^2}{t_1} + \frac{r_1 mh}{a}(r_2 - r_1)\right)$$

and the other partial pressures are found as above. The plates between the stays are now in the same condition as flat plates subjected to the respective partial pressures, m is the angular pitch.

The following list contains most of the experiments that have been made on flat plates:—

W. Fairbairn, 1874, pp. 147 and 333. ½-in. copper and ⅜-in. iron plates, 5-in. and 4-in. pitches.

R. H. Thurston, 'Franklin Inst.,' 1872, iii. vol. lxiii. p. 93. $\frac{5}{16}$-in. iron plate, 8¾-in. and $9\frac{3}{16}$-in. pitch.

L. E. Fletcher, 'Rep. M. S. U. A.,' Oct. 1876, p. 31. Admiralty experiments on 'Thunderer' boiler.

W. Boyd, 'M. E.,' 1878, p. 223. ½-in. iron, $\frac{7}{16}$-in. steel plates, 9-in. pitch.

C. Back and R. Wilson (see p. 172). Unstayed plates.

D. Greig and Max Eyth, 'M. E.,' 1879, p. 282, ⅜-in. and $\frac{9}{16}$-in. plates, 4½-in pitch.

W. S. Hutton, 1889, p. 180.

Board of Trade experiments on mild steel, 'Parl. Rep.,' 1881 (c. 2897), 1885 (c. 4572).

German Admiralty, Danzig, 1884-92.

In addition to the above experiments, a few details have come to hand about three tests, but neither date nor place were obtainable nor any reason for keeping back this information. They are the first three tests in the following table which consist mainly of summaries of the Board of Trade and German experiments. They have been compiled on the following lines: the permanent elongations of the stay bolts were subtracted from the permanent sets of the centres of the squares or rectangles supported by these stays, and because the steps through which the test pressures advanced were very large, the intermediate pressures (limit pressures) under which the first indication of permanent set may be assumed to have taken place were estimated by interpolating the pressures for the smallest and second smallest average permanent sets of the centres of the plates. These estimates are detailed in the tables as well as the estimate of the constants C, to be used in the formula $WP = C . t^2 : P^2$ where P is the pitch of stays in inches and t the thickness of plate in sixteenths of an inch. C_1 is the constant based on the nominal thickness, C_2 is the constant based on the actual thickness measured at the stay holes, or where this was not ascertained, it is based on the recorded thicknesses of the test pieces cut from the plate. C_2 is also based on a uniform tenacity of 30 tons per sq. inch, it being assumed that the limit pressures for flat plates are proportional to the ultimate tenacities.

A. Plates with stay ends riveted over. The mean value of C_2 is 503. This value is probably 2°/₀ too high, because (see groups C and E) the test pieces seem as a rule to be about $\frac{1}{50}$ in. thinner than the plates from which they have been cut.

B. Plates with stays having single nuts. The mean value of C_2 is 439 or more, and is also probably 2°/₀ too high. The relatively higher results obtained in group A is undoubtedly due to the swelling of the stay ends due to riveting, which swellings would give additional support to the plates against deflections.

C. Plates with stays having double nuts. The mean value of C_2 is 522.

C and D. In the German experiments, G, H, the back end plates of the test boxes, were much thicker than the front end ones, but their actual thicknesses were not measured and their tenacity not determined, as with the other German plates their average tenacity is probably 27 tons per sq. inch. In each set of three experiments, I*a*, I*b*, I*c*, II*a*, etc., the back plates were not changed for new ones, and therefore the *b* and *c* tests in group D are for plates which have already been subjected to the maximum test pressures noted in groups C and D, after having been flattened out before retesting.

E. Plates with riveted washers and stays having double nuts. The average value of C_2 is 749, but this value is affected by the ratio of the thickness of the washer to that of the plate.

Value of Riveted Washers.—A convenient comparison between groups B, C, and E can be made by finding an average constant C_3

Steel Makers	Plate Thicknesses		Pitch of Stays	Diameters		Test Pressures and Mean Permanent Sets				Estimated Limit Pressures	Estimated Values of		Average Tenacity	Highest Test Pressure Attained	Remarks
	Nominal	Test Pieces		Of Stays	Of Nuts	First Set		Second Set							
A. *Stay Ends Riveted.*															
	in.	in.	in.	in.	in.	lbs.	in.	lbs.	in.	lbs.	C_1	C_2	tons.		
W. I.	$\frac{1}{4}$	—	8	$1\frac{1}{2}$	—	140	·02	180	0·56	138	550	—	—	275	Stay head flew off.
”	$\frac{3}{8}$	—	8	$1\frac{1}{2}$	—	225	·01	—	—	225[1]	400[1]	—	—	375	Stay threads stripped.
”	$\frac{1}{2}$	—	8	$1\frac{1}{2}$	—	400	·01	—	—	225[1]	225[1]	—	—	600	Stay threads stripped.
S.	$\frac{9}{32}$	·27	8	$1\frac{1}{2}$	—	200	·005	250	·025	188	599	635	30·6	350	{ Stay heads flew off. Plate cracked at 300 lbs.
W.	$\frac{5}{16}$	·29	11·31	$1\frac{1}{2}$	—	100	·028	125	·065	81	414	471	30·6	185	Stay heads flew off.
S.	$\frac{13}{32}$	·38	8	$1\frac{1}{2}$	—	325	·005	375	·025	312	486	560	29·5	495	Stay heads flew off at 475 lbs.
W.	$\frac{7}{16}$	·42	11·31	$1\frac{1}{2}$	—	150	·006	175	·170	122	320	373	29·6	280	Stay heads flew off.
S.	$\frac{1}{2}$	·51	8	$1\frac{1}{2}$	—	550	·005	600	·025	538	538	528	29·4	740	Stay heads flew off.
W.	$\frac{9}{16}$	·53	11·31	$1\frac{1}{2}$	—	275	·013	300	·030	228	360	453	27·8	425	Stay heads flew off.
B.	$\frac{1}{2}$	—	9	$1\frac{1}{2}$	—	390	—	—	—	390[1]	493[1]	—	—	550	Burst.
B.	$\frac{7}{16}$	—	9	$1\frac{3}{8}$	—	325	—	—	—	325[1]	538[1]	—	—	550	Burst.
B. *Stays Fitted with Single Nuts.*															
S.	$\frac{9}{32}$	·27	8	$1\frac{1}{2}$	2·25	200	·026	250	·040	150[2]	474[2]	504[2]	30·6	980	Plate cracked.
W.	$\frac{5}{16}$	·29	11·31	$1\frac{1}{2}$	2·25	100	·027	125	·037	75	373	425	30·6	450	Plate cracked.
S.	$\frac{13}{32}$	·38	8	$1\frac{1}{2}$	2·25	300	·010	350	·020	250[2]	380[2]	441[2]	29·5	1200	Stays tore.
A.	$\frac{13}{32}$	—	8	2	3·75	400	·010	500	·170	394	595	—	—	1100	Plate slipped over nuts.
W.	$\frac{7}{16}$	·42	11·31	$1\frac{3}{4}$	3·25	200	·022	250	·037	150[2]	392[2]	453[2]	29·6	700	Plate cracked.
A.	$\frac{15}{32}$	—	8	2	3·75	500	·023	600	·110	473	537	—	—	1550	Plate slipped over nuts.
S.	$\frac{1}{2}$	·51	8	$1\frac{1}{2}$	2·25	550	·01	600	·030	525	525	518	29·4	1300	Stays tore.
A.	$\frac{1}{2}$	—	8	2	3·75	650	·023	700	·060	619	619	—	—	2000	Plate slipped over nuts.
W.	$\frac{9}{16}$	·53	11·31	2	3·75	200	·015	250	·220	150[2]	238[2]	291[2]	27·8	980	Plate slipped over nuts.

[1] Or less; [2] Or more; W. I. Old experiments with wrought-iron plates. Plate and date of experiments not known; S. Steel plates by Steel Company of Scotland; W. Steel plates by the Weardale Co.; A. Steel-plated boxes tested at Messrs. D. Adamson's works; B. Boyd's experiments, 1878.

Mark on Box	Steel Makers	Thicknesses: Nominal	Thicknesses: Measured	Thicknesses: Test Pieces	Pitch of Stays	Diameter of Nut or Washer	Thickness of Riveted Washer	Test Pressures and Mean Permanent Sets: First Set		Second Set		Estimated Limit Pressures	Estimated Values of		Tenacity	Highest Test Pressure Attained	Remarks
C. *Stays Fitted with Double Nuts and Small Washers.*																	
		ins. or mm.	ins. or mm.	ins. or mm.	ins. or mm.	ins. or mm.	ins. or mm.	lbs.	ins. or mm.	lbs.	ins. or mm.	lbs.	C_1	C_2	tons	lbs.	
—	A.	$\frac{9}{16}$	—	—	$14\frac{3}{4}$	4·25	—	175	·025	200	·050	150[2]	380[2]	—	—	740	Plate cracked.
—	A.	$\frac{9}{16}$	—	—	$14\frac{3}{4}$	4·25	—	175	·020	200	·047	156	406	—	—	740	Plate cracked.
Ic	G. H.	12	12·3	12·0	300	95	4	213	·70	284	2·0	175	407	478	26·9	881	Plate cracked.
IIc	G. H.	15	14·5	14·3	350	104	4	213	·70	284	5·9	203	463	543	26·3	1064	
IIIc	G. H.	16	16·0	15·8	400	124	4	213	1·00	284	6·0	201	492	546	27·7	1107	Stays broke.
IIa	G. H.*b*	23	—	—	350	104	4	568	1·50	710	6·8	528	478	—	—	1078	
IIIa	G. H.*b*	26	—	—	400	124	5	497	·60	710	7·5	479	445	—	—	1136	
IVa	G. H.*b*	29	—	—	450	140	5	426	·70	568	5·0	403	365	—	—	1108	Stay broke.
Va	G. H.*b*	29	—	—	369	105	4	497	·40	668	2·4	483	294	—	—	923	Stay broke.
D. *Repeat Tests with Reflattened Back Plates which had been used with the Respective Front Plates.*																	
Ib	G. H.*b*	20	—	—	300	95	4	568	1·30	710	4·4	508	451	—	—	923	
IIb	G. H.*b*	23	—	—	350	104	4	568	2·10	710	6·8	505	458	—	—	796	
IIIb	G. H.*b*	26	—	—	400	124	5	426	·90	568	2·4	341	317	—	—	1050	Stay broke.
IVb	G. H.*b*	29	—	—	450	140	5	426	·50	568	6·6	414	392	—	—	1136	
Vb	G. H.*b*	29	—	—	369	105	4	568	·50	710	2·7	536	338	—	—	1008	Stay broke.
Ic	G. H.*b*	20	—	—	300	95	4	426	·30	568	2·2	404	354	—	—	881	
IIc	G. H.*b*	23	—	—	350	104	4	426	·40	568	3·6	408	369	—	—	1064	
IVc	G. H.*b*	29	—	—	450	140	5	426	1·00	568	6·4	400	377	—	—	994	
E. *Stays with Double Nuts. Large Washers Riveted to Plates.*																	
—	G.	$\frac{9}{16}$	—	—	14·5	9	$\frac{1}{2}$	300	·03	350	·063	255	662	—	—	860	Nut threads stripped.
—	G.	$\frac{9}{16}$	—	—	14·5	9	$\frac{1}{2}$	250	·01	300	·03	225	585	—	—	860	Nut threads stripped.
Ib	G. H.	10	9·4	9·4	300	165	12	284	·80	426	7·8	268	1005	1095	27·5	923	Plate cracked.
IIa	G. H.	15	14·6	14·3	350	192	14	284	·40	426	2·7	259	580	672	27·0	1078	
IIb	G. H.	12	11·3	11·2	350	192	14	213	·80	284	2·1	169	634	725	26·7	796	Plate cracked near edge.
IIIa	G. H.	16	16·3	16·1	400	220	16	248	·40	284	1·00	224	527	579	28·0	1136	
IIIb	G. H.	13	13·2	13·1	400	220	16	142	·20	213	·80	119	428	497	26·3	1050	Stay broke.
Va	G. H.	12	12·2	11·8	{345 / 392}	220	12	213	1·60	284	8·3	200	715	977	23·5	1108	Stay broke.
Vb	G. H.	10	9·2	—	{345 / 392}	220	12	142	1·10	213	3·8	113	696	—	26·2	1008	Stay broke and plate cracked at edge.

[2] Or more; A. Steel-plated boxes tested at Messrs. D. Adamson's Works; G. Steel boxes tested at the Glasgow Locomotive Works; G. H. Steel plates by Gute Hoffnung's Hütte; G. H.*b* are the back plates of the test boxes. Several were flattened and used again once or twice.

for groups B and C, in which the pitches are replaced by the diagonal pitches less the nut diameters, and then estimating similar diagonal pitches less effective washer diameters for group E. As the average value of C_3 for groups B and C is 585, the effective diameter D of riveted washers can be expressed by the formula

$$D_e = 0{\cdot}75 D_w t_w^2 : t_p^2$$

where D_w and t_w are the dimensions of the washer, and t_p the thickness of the plate. Let l be the diagonal pitch of the stays then the probable limit pressure would be

$$P = 585t^2 : (l - D_e)^2 = 585t^2 : (l - \tfrac{3}{4}D_w t_w^2 : t_p^2)^2.$$

This formula applies equally well to groups B, C, and E.

Application of Test Results.—Pressures calculated with the help of the constants found above are those at which the first indications of permanent set might be expected, and these pressures would be convenient limits for the hydraulic test pressures, a small allowance, say 25°/ₒ, being made on account of uncertainties about local thicknesses, tenacities, and other irregularities.

Safety of Stayed Plates is a subject on which these experiments throw little light, for amongst these forty-four tests there were only eight crackings at stay holes, but twenty-four cases of stay failures, which is a question of diameter and not of thickness. However, in those cases where the plates did crack, the ratios of the maximum pressures to limit pressures averages about four and one-half, so that if the working pressure is fixed at say half the limit pressure the factor of safety may be considered to be about nine.

Curved Beams.—An originally curved beam behaves very much in the same way as a straight one, except that the stresses are not distributed so uniformly. Let the line OP (fig. 152) be the neutral fibre, which does not alter its length while being bent. For any short length $\rho \,.\, d\alpha$ the elongation of the fibres above or below this line is proportional to their distances from it. But not so the strains, for, as the outer fibres are longer than the inner ones, so will the strains, and consequently also the stresses, be proportionately greater the nearer they are to the centre of curvature. Let ρ_o and ρ be the radii of curvature of the neutral fibre before and after bending, and let x be the distance of any fibre from this line; then, if E is the modulus of elasticity, it is easily shown that the stress is

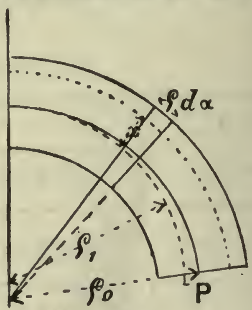


Fig. 152

$$S = E \,.\, \frac{x \cdot (\rho_o - \rho)}{(x + \rho_o) \cdot \rho}.$$

Then, on carrying out the necessary algebraical operations, n (figs. 153 and 154), the distance of the neutral fibre from the concave side of the rectangular beam, is found.

$$n = t\left(\frac{1}{\log \text{nat}\,(1+m)} - \frac{1}{m}\right)$$

The following are a few numerical values:

$$m = t : r = \infty \quad 10 \quad 5 \quad 1 \quad \tfrac{1}{2} \quad \tfrac{1}{10} \quad 0$$
$$n : t = 0 \quad \cdot317 \quad \cdot358 \quad \cdot443 \quad \cdot466 \quad \cdot492 \quad \cdot500$$

The maximum stress S exists on the concave side of the beam.

$$S = C \cdot 6 \cdot \frac{M}{bt^2}, \text{ where } M = \text{bending moment and}$$

$$C = \tfrac{1}{3} \cdot \frac{m[m - \log \text{nat}\ (1 + m)]}{(m + 2) \log \text{nat}\ (1 + m) - 1}.$$

The following are a few numerical values:

$$m = t : r = \infty \quad 10 \quad 5 \quad 1 \quad \tfrac{1}{2} \quad \tfrac{1}{10} \quad 0$$
$$C = \infty \quad 2\cdot889 \quad 2\cdot103 \quad 1\cdot287 \quad 1\cdot154 \quad 1\cdot033 \quad 1\cdot000$$

The practical conclusions to be drawn from these figures are that the bending stresses on the concave side of plates flanged to the usual radius, of, say, twice the thickness, are 15°/$_\circ$ greater than those in the adjoining flat plates; that when the inner radius is reduced to the thickness of the plate 29°/$_\circ$ has to be added to the stress; while when there is no round on the inside of the flange the stresses are infinitely great. The end of every crack can also be looked upon as the inner surface of a curved beam, and the lightest external force would extend

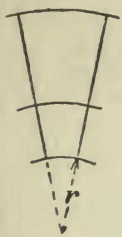


FIG. 153

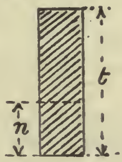


FIG. 154

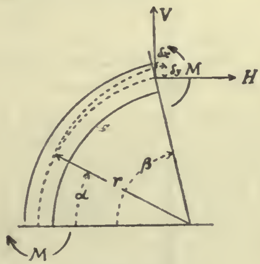


FIG. 155

it, unless there are other stresses beyond the crack holding the plate together. When they are absent, or when tension stresses exist in the plate, there is nothing to stop the crack from extending with the usual explosive rapidity. Similarly, the concave sides of cold bend samples are very weak, on account of their small radii, and, as the residual stresses in the samples all tend to produce tensions at this point, it is not surprising that the majority of such test pieces crack some time after they have been put aside, and without any apparent cause.

Elastic Deformations of Curved Beams.—As in the case of a straight beam let E be the modulus of elasticity, and I the moment of inertia of the section of the beam, the necessary allowance being made for the radius of curvature r (see above), Using polar co-ordinates we have the three expressions:

$$\frac{d\gamma}{ds} = \frac{d^2(\Delta r)}{r^2 da^2} = \frac{m}{\mathrm{E\,.\,I}}$$

$$\frac{d(\delta y)}{ds} = r(\cos a - \cos \beta)\frac{d^2(\Delta r)}{r^2 da^2} = \frac{m}{\mathrm{E\,.\,I}}r(\cos a - \cos \beta),$$

$$\frac{d(\delta x)}{ds} = r(\sin a - \sin \beta)\frac{d^2(\Delta r)}{r^2 da^2} = \frac{m}{\mathrm{E\,.\,I}}r(\sin a - \sin \beta).$$

Seeing that $\sin \beta$ is larger than $\sin a$, the last expression is negative.

In these expressions Δr is measured radially from the original centre line of the beam. γ is the angle included between the original and ultimate centre line of the beam at any angle a, and when $a = \beta$ the angle γ is also the tilt of the upper flange if the beam is a curved pipe. Δy is the rise of the end of the pipe, and Δx is the horizontal movement of the end of the pipe.

The moments acting on the beam at the point a are firstly $m = \mathrm{M}$, which is constant for any angle a, secondly $m = \mathrm{V}r(\cos a - \cos \beta)$, and thirdly $m = \mathrm{H}r(\sin a - \sin \beta)$. This last moment is of course negative. Introducing these values in the above equations and integrating after a from o to β, the expressions contained in the following table are obtained. The table also contains the values for the special case of a right-angled bend when $\beta = 90°$.

Moments	$\gamma \times \mathrm{EI}$	$\delta(y) \times \mathrm{EI}$	$\delta(x) \times \mathrm{EI}$
Deflections for any Angle β; $r\beta = l$, $r \cos \beta = m$, $r \sin \beta = n$			
M	$\mathrm{M}l$	$\mathrm{M}(rn - lm)$	$\mathrm{M}(r^2 - rm - ln)$
$\mathrm{H}n$	$\mathrm{H}(r^2 - rm - ln)$	$\mathrm{H}(-\frac{1}{2}r^3 + lmn + \frac{3}{2}r^2m)$	$\mathrm{H}(\frac{3}{2}r^2l - 2r^2n + \frac{3}{2}rmn - lm^2)$
$\mathrm{V}(r - m)$	$\mathrm{V}(rn - lm)$	$\mathrm{V}(\frac{1}{2}r^2l + rlm - \frac{3}{2}rmn)$	$\mathrm{V}(-\frac{1}{2}r^3 + lmn + \frac{3}{2}rm^2)$
Deflections for Right-Angled Bends; $\beta = 90°$			
M	$+ 1{\cdot}571\mathrm{M}r$	$+ \mathrm{M}r^2$	$- {\cdot}571\mathrm{M}r^2$
$\mathrm{H}r$	$- {\cdot}571\mathrm{H}r^2$	$- {\cdot}50\mathrm{H}r^3$	$+ {\cdot}356\mathrm{H}r^3$
$\mathrm{V}r$	$+ \mathrm{V}r^2$	$+ {\cdot}785\mathrm{V}r^3$	$- {\cdot}50\mathrm{V}r^3$

Experiments have been carried out by Prof. A. Bautlin ('Deutsch. Ing.,' 1910, v. 54, p. 43) on pipes and bars bent as shown in fig. 156. The dimensions and deformations are given in the following table. These experiments do not correctly represent the straining to which steam pipes are subjected, because the flanges of these bends were free to tilt, but Prof. Bautlin has compared the actual deformations with the calculated ones, and finds the ratios to be as follows: A : 4·89, B : 2·60, C : 0·94, D : 1·04, E : 0·97. He attributes the great elasticity of A and B: between two and five times as great as might be expected, to slight puckerings on the inner surfaces of these bent steel tubes, and other writers have tried to show that this greater elasticity may in part be accounted for by the change of shape of a circular tube when stressed.

	Horizontal Thrusts: Pounds						220	440	660	880	1,320	1,980	2,420
	Material	Dimensions of Bends					Deformations $\delta(l)$ of Half Loop						
		Diameters in.	Thickness in.	r_1 in.	r_2 in.	l in.	in.	in.	in.	in.	in.	in.	in.
A	Mild Steel	8·7 × 8·2	·26	33·1	22·0	37·4	—	—	·63	—	1·37	2·16	2·98
B	„ „	5·3 × 5·1	?	31·9	16·7	23·6	·69	1·42	—	—	—	—	—
C	Cast Iron	8·5 × 8·4	·78	34·0	22·0	47·5	—	·10	—	·22	·34	—	—
D	Steel Bars	3·15 square	—	19·7	11·8	21·7	—	—	·24	—	—	—	—
E	„ „	3·15 square	—	19·7	17·7	27·9	—	—	·33	—	—	—	—

End-Plate Flanges.—The agreement between experimental and calculated deflections of the square bars shows that the calculations might be applied to the determination of the deformations, and therefore also of the stresses of flanges of flat and dished boiler ends. That the matter is of importance will be evident from the following list of boiler explosions due to the grooving through of end-plate

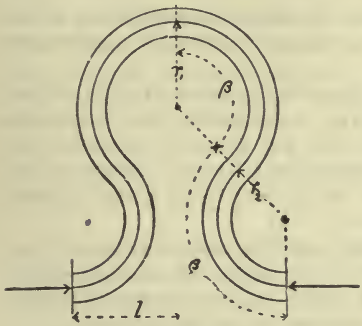


FIG. 156

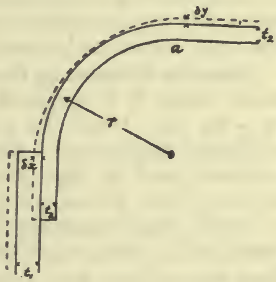


FIG. 157

flanges—Board of Trade Reports Nos. 148, 155, 162, 198, 226, 235, 290, 682, 967, 980, 1,155, 1,484, 1,699. Of these the last-named is very instructive, though the important point, the smallness of the radius of the flange, did not receive the attention it deserved. After this explosion quite a number of dished-end plates of boilers have been renewed. The above cases also include a few flat-ended boilers.

A general idea of the nature of the stresses can be obtained from fig. 157. Let t_1 be the thickness of the boiler shell plate of the diameter 2R, and let t_2 be the thickness of the end plate, with a flange of the radius r, then the upward force V (fig. 155) per 1 in. of circumference is $\frac{1}{2}P(R - r)$, or as r is small compared with R we may write $\frac{1}{2}PR$. This vertical force is the chief cause for the slight vertical displacement $\delta(y)$, but it also produces a slight horizontal displacement, which has to be subtracted from $\delta(R) = R^2P : t_1E$, then as $\beta = \frac{\pi}{2}$ we have according to the table

$$\delta(x) = \frac{R^2P}{t_1E} = (-\cdot571Mr^2 + \cdot356Hr^3 - \cdot50\tfrac{1}{2}PRr^3)\,\frac{I}{EI}.$$

Assuming that the flat plate does not bulge, which however is not strictly true, we have

$$\gamma = 0 = (1\cdot571Mr - \cdot571Hr^2 + \tfrac{1}{2}PRr^2)\frac{1}{EI}.$$

Combining these two equations and remembering that I the moment of inertia is $\frac{1}{12}t_2^3$, we find the moments M and Hr; Vr being given above.

$$M = PRr(0{\cdot}483 + 0{\cdot}202\left(\frac{t_2}{r}\right)^3\frac{r}{t_1}$$

$$Hr = PRr\left((0{\cdot}120 + 0{\cdot}146\left(\frac{t_2}{r}\right)^3\frac{r}{t_1}\right).$$

Near the shell plate the total bending moment is M + Vr − Hr, at a_1, it is M, and at some intermediate angle α it is M + V$r\cos\alpha$ − H$r(1 - \sin\alpha)$ (see fig. 156). It is a maximum, according to dimension, at about 45°, so that approximately the maximum bending moment is

$$P \,.\, Rr\left(0{\cdot}75 + 0{\cdot}10\left(\frac{t_2}{r}\right)^3\frac{r}{t_1}\right).$$

Any bulging of the front-end plate increases the coefficient of the last term very considerably. For stayed-end plates V is of course less than mentioned above.

Shearing Stresses in Beams.—No shearing stresses exist in a bent beam when it is exposed to only two external bending moments $m_1 = m_2$ (fig. 145). In every other case the shearing force in any particular section of a beam is exactly equal to the otherwise unbalanced load at that point. Thus, in fig. 145, p. 173, the upward shearing force at the point x is $Q_2 - p(l - x)$.

If this force were evenly distributed over the sectional area of the beam, it would be easy to calculate it; but there are certain conditions which have to be fulfilled, and which lead to somewhat complicated results, particularly if the section is an irregular one. In rectangular beams, and therefore also in flat plates, the case is simple.

Let σ represent the intensity of the shearing stress at a point situated at the height y above the neutral fibre op of a beam (fig. 159), while S is the longitudinal stress at that point; then it is easily proved that

$$\frac{d\sigma}{dy} = -\frac{dS}{dx}.$$ But in a rectangular beam

$$\frac{dS}{dx} = \frac{12}{t^3} \cdot y \cdot \frac{dm}{dx} = \frac{12}{t^3}yQ$$ and therefore $\frac{d\sigma}{dy} = C \,.\, y$, where C is a constant.

The maximum shearing stress at the neutral fibre is $2Q : 3t$.

If, then, the total shearing force in the section at the distance x is Σ we have

$$\sigma = \frac{3\Sigma}{2 \,.\, t}\left\{1 - 4\left(\frac{y}{t}\right)^2\right\}.$$[1]

[1] The axial shearing stresses on the threads of a screwed stay can be shown to be distributed in the same manner, and instead of making the thickness of the plate or nut half the diameter, it ought to be three-quarters.

Fig. 161 represents the distribution of the longitudinal stresses, and fig. 160 that of the shearing stresses.

As a shearing stress σ is a compound of a tension and compression stress S, acting at angles of 45° to the line of shear, as shown in fig. 125 (p. 168), it is easily proved that

$$\sigma = \pm S.$$

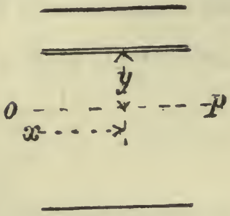


FIG. 159


FIG. 160


FIG. 161

In order, then, to resolve the various stresses to be met with in a beam into their respective right-angled resultants it is only necessary to apply the formulæ of p. 167; then

$$tg \, . \, 2 \, \gamma_1 = \frac{o+o+\sigma+\sigma}{-p+S+o+o} = \frac{2 \, . \, \sigma}{S-p}.$$

$$x = \frac{p+S}{2} + \sqrt{\frac{4 \, . \, \sigma^2 + (S-p)^2}{4}}, \quad y = \frac{p+S}{2} - \sqrt{\frac{4 \, . \, \sigma^2 + (S-p)^2}{4}}$$

Here p is the transverse pressure, but the signs of both S and p will have to be changed if they are compression stresses. The distribution of p is determined by σ.

It is now possible to construct curves showing the direction and intensities of the resultant stresses at any point of a beam (see figs. 184, 185, 186, p. 211).

Plastic Beams.—When flanging or bending plates, the elastic limit of their material is always passed, and the resultant deformation is a permanent one. In the chapter on 'Strength of Materials' a formula has been explained which enables one to calculate the stresses in the outside fibres of narrow plastic beams, when their curvature is known. Roughly, $S = \frac{4 \, . \, m}{t^2}$; so that S is only two-thirds of what it would be in an elastic beam, which means that a plastic beam is 50 % stronger than the elastic theory would make it.

Another very important difference between an elastic and a plastic beam is that the stresses of the former bear some relation to the deformation, while those of the latter are nearly, if not quite, independent of the same.

The work W_1 required to bend an elastic steel beam is proportional to the square of its final curvature, about

$$W_1 = \frac{280 \, . \, A \, . \, t^3}{\rho^2} \text{ foot tons.}$$

Here A is the superficial area of the plate, measured in square feet, and ρ the radius of curvature, measured in feet. By the time that the elastic limit is reached this amounts to about

$$W_1 = \frac{A \,.\, t}{450} \text{ foot tons,}$$

and is independent of the radius. In other words, the maximum energy which can be stored in a flat spring is proportional to its volume.

The work W_2 required for bending a plastic beam or plate is proportional to the curvature

$$W_2 = \frac{C \,.\, A \,.\, t^2}{\rho}.$$

The value of the constant C is about

6	for mild steel, cold.	5	for wrought iron, cold.
4	„ „ „ blue hot.	$3\frac{1}{2}$	„ „ „ blue hot.
0·3	„ „ „ red hot.	0·25	„ „ „ red hot.

Thus to bend a cold steel shell plate whose dimensions are 14 feet × 7 feet × 1 inch, and whose radius is 6·7 feet, a power of 88 foot tons would be required. To do it at a red heat requires only $4\frac{1}{2}$ foot tons.

The above values have been estimated from the following experiments and other observations. Dr. J. Kollmann (' Ver. Gew.,' 1880, vol. lix. p. 107, &c.) states that he found the limits of elasticity (? plasticity) to be as follows :—

Tables of Limits of Elasticity and Plasticity

Temperature, °F.	68°	1,380°	1,470°	1,560°	1,650°	1,750°
Iron . . . Tons	17·5	2·0	1·3	1·0	...	...
Mild steel . „	25·4	3·0	...	...	1·5	1·2

C. E. Stromeyer, ' C. E.,' 1886, vol. lxxxiv. p. 125, Nos. 4, 9, 10, 12, 18.

Temperatures, ° F. Colour of Fracture	68° ...	470° Straw	500° Blue
Hard steel . . . Tons	19·4	17·0	15·9
Mild „ . . . „	16·3	...	10·0

From this it will be seen that at a blue heat and at a red heat iron, as well as steel, grows more pliable in the ratio respectively of 3 to 2 and 20 to 1. The power required to permanently bend an iron and a steel plate 12 inches wide and 1 inch thick is, therefore, as follows :

Condition	Cold	Blue Hot	Red Hot
Iron plate . . Foot tons	60	40	3
Steel „ . . „	75	50	3

Cross Deformation of Plastic Beams.—When the elastic limit has been passed the value of $1/\mu$, the coefficient of cross contraction, increases from $\frac{1}{3}$ to $\frac{1}{2}$. For very narrow beams the ratio of ρ to ρ_1 (fig. 162) would then be as 1 to 2. In wide ones, such as shell plates, this deformation is of course impossible, and it is evident that a cross stress must have been set up, whose intensity is nearly equal to half the longitudinal one, or, say, + 12 tons per square inch, under the action of the bending rolls. But this condition also demands supplementary longitudinal stresses at the edges of the plate and compression stresses near its centre. They are all modified by the spring of the plate as it leaves the rolls; but it is quite clear that the residue is both of a severe and of a complicated nature. Stresses of a similarly complex nature also exist in small bent test samples; and it is interesting to note that when the steel is of an unsatisfactory quality it usually cracks at the centre of the width, showing that it was incapable of withstanding the combination called drum tension (see fig. 163 and p. 160) to be found there.

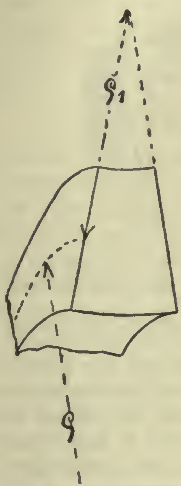


Fig. 162

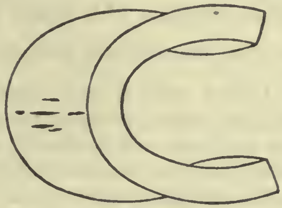


Fig. 163

The complicated nature of these stresses, particularly in wide beams, readily explains that the maximum stresses to be found in them when the plastic limit has once been passed differ very materially from those found by the formula $S = \frac{6 \,.\, m}{t^2}$.

Shearing Stresses in Plastic Beams.—The previously found formula, $\frac{d\sigma}{dy} = -\frac{dS}{dx}$, is independent of the elasticity of the material. If, therefore, the longitudinal stresses in a rectangular plastic beam are distributed as shown in fig. 164, then the shearing stresses must be distributed as in fig. 165. This illustration represents a case where the bending stresses have been produced by means of a light force acting on a long leverage, as, for instance, when shell plates are being passed through the bending rolls.

The case is very different when a strong force with a short leverage produces the same moment. Then the shearing stresses will be nearly uniformly distributed over the thickness (fig. 167), and the longitudinal stresses will have to accommodate themselves accordingly somewhat after the fashion shown by the curves in fig. 166. In the one case the beam gives way at the outside fibres, in the other at the neutral fibre. An intermediate stage is worthy of mention. Suppose that the plastic limit of tension and compression is 20 tons, while for shearing it is 15 tons; then both outside fibres of a beam would give way to longitudinal stresses, at the same time that the centre fibres commence to shear if these stresses are reacted on simultaneously. This happens in a square bar if the bending force is applied at a distance equal to $\frac{1}{3}$ of its thickness, and doubtless most rivets behave in this way.

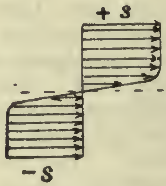


FIG. 164

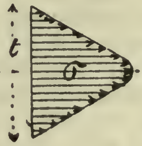


FIG. 165

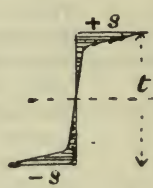


FIG. 166

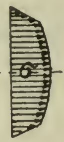


FIG. 167

Stresses in Cylindrical Shells.—The circumferential stress in a cylindrical shell of the mean diameter D is found by the well-known formula $S = \frac{p \cdot D}{2 \cdot t}$, where p is the internal pressure and t the thickness of the shell plate. If the ends of the shell plates are joined together by riveted longitudinal seams, the percentage of their strength will have to be taken into account. If the boiler shell is built up of several strakes, and the longitudinal seams break joint, it has been argued that the stresses are proportionally reduced, and that they should be calculated as follows: $S = \frac{p \cdot D \cdot l}{2 \cdot t \cdot \Sigma(\Delta l)}$. Here l is the length of the boiler, and $\Sigma(\Delta l)$ is the sum of the widths of the plates, including the flanged end plates, from which the rivet holes of only one longitudinal seam have been subtracted. As $\Sigma(\Delta l)$ is often greater than l, this view would lead to the conclusion that a riveted shell is stronger than a solid one. Clearly, this argument is only true as regards the mean stresses, and that it leads to valueless results is proved by the fact that the same reasoning applied to a beam would lead to a wrong conclusion, for its mean stress is just nothing, being $+ S$ at the top and $- S$ at the bottom.

A point where variation of stresses in shell plates may be expected is at A (fig. 168), in the solid plate near the end of a longitudinal seam. Let it be assumed that this seam is more elastic than the plate, i.e. that whereas a stress of one ton would stretch the latter $\frac{1}{13000}$ of its length, the same stress would cause the joint to spring or stretch, say, four times as much. Then, as these two parts are firmly connected by the circumferential seam, the solid plate at A would be subjected to a four times greater stress than the joint. A solitary and perhaps not very reliable experiment on this subject showed that the stress was actually eight times greater, and similar results were obtained near the single butt-strap joints on ship's sides (C. E. Stromeyer, 'N.A.,' 1886, vol. xxvii. p. 34). The remedy which readily suggests itself is to make the longitudinal seams more substantial—say, of the double butt-strap type. Now, however, this part may be more rigid than the plate, and will have to bear a proportionately heavier load, and being perforated is less capable of sustaining it. A more correct principle would be to make the longitudinal joints exactly as elastic as the solid plate. A few remarks on the elasticity of a riveted joint will be found further on (p. 218), when discussing their theories, but a true solution can be obtained only by careful experiments.

The order in which seams naturally range themselves as regards rigidity is—1st. Single butt-strap joint. 2nd. Lap joint. 3rd. Butt joint with one wide and one narrow strap. 4th. Butt joint with two equally wide straps.

Of course the number of rows of rivets, their pitch and diameter, and the thickness of the cover plates, will affect the results.

The Longitudinal Contraction of Cylindrical Shells is $\frac{S \cdot l}{E \cdot \mu}$, where l is the length and S the circumferential stress. In a boiler of 17 ft. length this will amount to about 0·025 in. at the ordinary working pressure. If no stays are fitted to take up the longitudinal stress, which is one half of the above, it will be found that the elongation, $\Delta l = \frac{S}{E} \cdot l\left(\frac{1}{2} - \frac{1}{\mu}\right) = \frac{p \cdot D}{E \cdot T} \cdot l\left(\frac{1}{2} - \frac{1}{\mu}\right)$. This can be utilised for the determination of $1/\mu$, the coefficient of cross contraction. By riveting longitudinal strips on a boiler shell, the longitudinal contraction is reduced, at least locally, and this is exactly what all riveted longitudinal seams do. Consequently, longitudinal compression stresses are set up in them, which must produce tension stresses in some other part of the shell, probably along either side of the seam. A longitudinal compression stress c will also be set up at A (fig. 168), which, combining with the circumferential stress s, constitutes a shearing stress, and that, as has been shown by experiments (see fig. 123, p. 157), is far more injurious to iron and steel than a simple tension. Therefore, whether elastic lap joints or the more solid butt-strapped joints are used, very severe stresses will be found in the adjoining plates.

Longitudinal contraction takes place in cylindrical furnaces exposed to external pressure, and the two deformations, amounting to large fractions of an inch, cause severe stress and grooving in end plates. These can be obviated by using corrugated flues for which the longitudinal elasticities have been determined as per following table (p. 194). (See M.S.U.A., 1905, p. 13).

Influence of End Plates on the Stresses in Boiler Shells.—In fig. 169 let the line CD represent the shape of half the length l of a boiler shell of the diameter D, while subjected to an internal pressure p_1. The shell plate will acquire this curvature if secured to the end plate at C, as shown; if unsecured it would have remained cylindrical and have occupied the position at the line AB. Under that condition the increase of half the diameter is $y_1 = \frac{p_1 \cdot D^2}{4 \cdot E \cdot t}$, where t is the thickness of the plate and E the modulus of elasticity. The stress will be $S_1 = \frac{p_1 \cdot D}{2 \cdot t}$; but, as will be seen in the diagram, if the ends are held down there is no circumferential straining of the shell at C, while at D it is reduced in the ratio of $y_1 - y_0$ to y_1. The object of this investigation is to ascertain what this value is.

A little reflection will show that the circumferential stress S at any point x or z is proportional to the distance of that point of the shell

Experiments on the Longitudinal Elasticity of Cylindrical Flues

Type of Cylindrical Flue	Material		Dimensions				Contraction per Single Corrugation Loaded on Circumference			Load per Inch of Circumference when Limit was Reached		Authority
	Tenacity	Elongation	Int. Diam.	Corrugations		Thickness—nominal	1 Ton per inch		Limit	Actual	½-in. Plate	
				Depth	Pitch		Actual	½-in. Plate	½-in. Plate			
	Tons	%	In.	In.	In.	In.	In.	In.	In.	Tons	Tons	
Adamson's Flange . .	21·9	29·5	37·4	...	32·00	0·550	0·0138	0·0184	0·0184	1·212	1·00	v. Knaudt VI.
Fox's Corrugation . .	23·6	25·0	37·6	1·73	5·94	0·393	0·0265	0·0129	0·0185	0·887	1·43	„ I.
„ „ . .	29·5	28·5	37·5	2·25	5·94	0·460	0·0235	0·0183	0·0218	1·010	1·19	„ II.
„ „ . .	21·2	30·0	37·5	1·79	5·94	0·472	0·0184	0·0155	0·0188	1·084	1·21	„ III.
„ „ . .	22·2	29·5	37·2	1·92	5·94	0·535	0·0126	0·0155	0·0177	1·309	1·14	„ IV.
„ „ . .	24·4	26·0	39·1	0·98	5·94	0·393	0·0061	0·0029	0·0076	1·600	2·59	„ V.
„ „ . .	...	...	29·9	1·58	6·00	0·386	0·0242	0·0111	0·0176	0·950	1·59	Prof. Bach.
Morrison's Corrugation .	...	...	29·6	1·58	7·85	0·372	0·0282	0·0112	0·0221	1·200	2·17	„
Leeds Forge (Bulb) . .	...	...	38·25	2·25	8·00	0·440	0·0531	0·0362	0·0331	0·708	0·91	Leeds Forge Co.
Purves (Ribbed) . .	27·2	26·0	37·2	...	9·00	0·472	0·0027	0·0022	0·0116	4·220	4·73	v. Knaudt VII.
Holmes' (Single Bulbs) .	...	...	29·3	3·46	24·00	0·528	0·0783	0·0920	0·0700	0·850	0·76	Prof. Bach.
„ „ .	...	...	29·3	3·46	24·00	0·400	0·2110	0·1080	0·0640	0·380	0·59	„
Cylindrical Tube 6 in. long (estimated) . . .	...	...	...	...	...	...	...	0·0009	...	...	2·500	...

from its original position. $S = S_1 \cdot \frac{y_1 - y}{y_1} = \frac{p_1 \cdot D}{2 \cdot t} \cdot \frac{(y_1 - y)}{y_1}$. But as this stress is capable of balancing only part of the internal pressure, there remains $p = p_1\ y/y_1$, which has to be transmitted to the end plate,

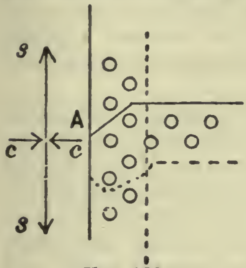


FIG. 168

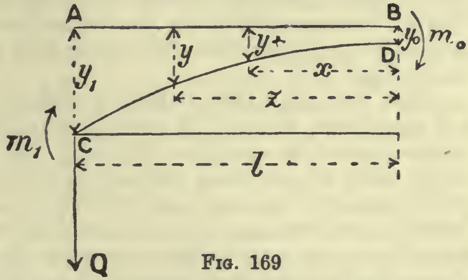


FIG. 169

and in doing so longitudinal bending stresses are set up in the shell, which have to be ascertained. This can be done by examining a long strip of the boiler shell—say, 1 in. wide—loaded irregularly with a pressure p, and supported at its ends by loads Q. A bending moment m_1 will also be found there. Evidently the load Q is proportional to the area ABDC.

$Q = \frac{p_1}{y_1} \cdot \int_0^l y\ dz$. The bending moment at the distance x is

$$m = -m_1 + Q\,(l - x) - \frac{p_1}{y_1} \cdot \int_x^l y \cdot (z - x) dz,$$

which may be written $\frac{E \cdot t^3}{12} \cdot \frac{d^2y}{dx^2} = m =$

$$-m_1 + \frac{p_1}{y_1}\left[l \int_0^l y \cdot dz - x \int_0^x y \cdot dz - \int_x^l y \cdot z \cdot dz \right]$$

This leads to the equation $y = \frac{1}{2}y_0\,(e^{ax} + e^{-ax}) \cos ax + A(e^{ax} - e^{-ax}) \sin ax$.

By integration and differentiation the values of $\frac{1}{2}y_0$, A, and a can be found, and numerical values obtained, when the conditions of the external forces are known. Thus, when very thick shell plates are attached to thin end plates, or when these have a very weak and well-rounded flange, $m_1 = 0$ or nearly so. In the latter of these two cases y_1 is also reduced, on account of the spring of the flange, which is proportional to Q, and this value has therefore also to be reduced, and the relief afforded to the shell plate is small. The calculations are too complicated to be reproduced here. The problem has been discussed on somewhat different lines by Dr. F. Grashoff, 1878, p. 316, and by J. T. Nicolson, 'N.-E. C. I.,' 1891, vol. vii. p. 205. Adapting some of his results to a boiler of 15 ft. diameter, with 1-in. shell plates at 100 lbs. pressure, we find the circumferential stress in the centre of the length to be as follows:

Length between end plates (feet)	∞	15	10	5	2	1
Circumferential stress at centre (tons)	3·35	3·35	3·34	3·15	2·96	·70

But this is only true if the end plates are rigid while $m_1 = 0$. In practice this is never the case, and a very considerable deformation must take place in the end plates. The back end, which is practically a flat plate, will be strained uniformly. If made exceptionally thin, it would have to expand as much as the shell, but, not being able to contract crossways, the consequent drum tension z would be $pD : 2t(\mu - 1)$, or about $50°/_\circ$ more than the circumferential tension in the shell.

With the front plate, which is perforated by furnaces, manholes, and tube holes, it would at first sight appear as if locally the stresses might grow to be excessive, but as in the remaining narrow flanged strips there are no cross stresses, the remaining surface ones cannot exceed those in the shell plate. The weakest points are to be found between the furnace front holes; but, as no fractures take place there, it is but reasonable to suppose that the stresses in the end plates, due to their attachment to the shell, are small, the roundness of the flange providing the necessary springiness.

Stresses near the Dome Holes.—A problem which is often met with in boiler designing is the efficient staying of the corners of two

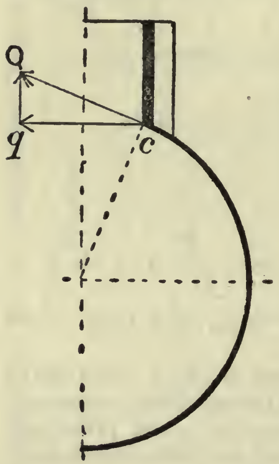


Fig. 170

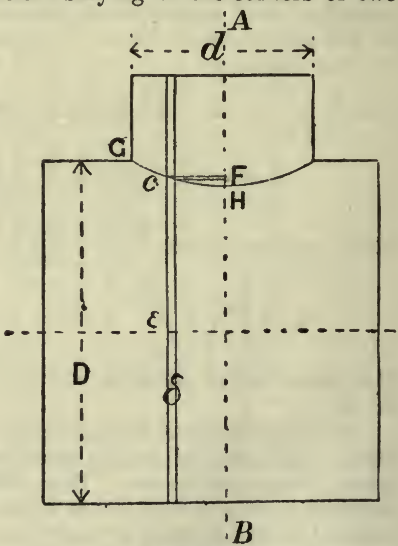


Fig. 171

intersecting cylindrical shells (fig. 171). Fig. 170 is a section through the line $c\epsilon\delta$.

Assuming that the larger cylinder is cut up into numerous rings of the thickness δ, and that internal pressure is p, then if the ring under consideration were severed at the point C it would have to be retained in its position by a pull $Q = \frac{D}{2} \cdot \delta \cdot p$.

Now the vertical component of this pull could be transmitted to the straight side of the cylinder d, but a stay would have to be fitted to take up the horizontal component q. A little consideration will show that it bears the same relation to Q as the distance $c\epsilon$ bears to half the diameter D. The same arguments show that the horizontal component of the narrow strip cF is also proportional to its projected area. This shows that the greater pull is found at G and the smaller one at H. The sides of the holes are also severely stressed, see p. 216.

Stresses in Cylindrical Flues.—The same formula as for cylindrical shells may be applied here, but, when experimenting, buckles will show themselves long before even the limit of elasticity for compression has been reached, and the pressure cannot then be increased, because the flue grows weaker and weaker the more it alters its shape. It is important to know at what pressure this change takes place. Let the elliptical dotted line (fig. 172) be the shape of the originally cylindrical flue, and let the black line be the curve of thrusts. This is a nearly circular line, and can be found by graphic construction, as is done in the case of arches, &c. The intensity of the thrust is $\frac{D}{2}p$, where p is the external pressure. To find the bending moment at any position (say, at the angle a) it is only necessary to multiply this thrust into the distance h between the two lines. At the major and minor axes the moments are maxima $m_1 = -m_2 = \frac{D \cdot p \cdot \delta}{2 \cdot 2}$. For any other point $m = m_1 \cos 2a$. This equation is practically identical with that obtained for a straight strut of the length $D\pi/4$ with loose ends and loaded with the above force. If accidentally bent, such a strut would straighten itself while supporting its load K, provided it does not exceed $K = \frac{16 \cdot E \cdot I}{D^2}$. Substituting the value for K, and also for $I = \frac{t^3}{12}$, the moment of inertia, we have $\frac{D}{2}p = \frac{16 \cdot E \cdot t^3}{12 \cdot D^2}$, from which it follows that p is limited:

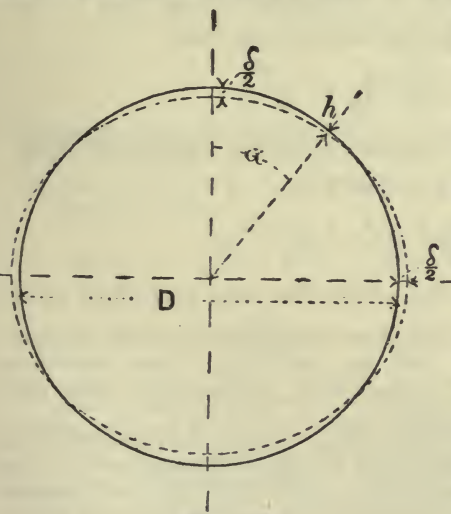


Fig. 172

$$p \leqq \frac{8}{3} \cdot E \cdot \frac{t^3}{D^3} = 8 \cdot 10^7 \cdot \frac{t^3}{D^3} \text{ lbs. per sq. in.}$$

As will be seen, p in this formula depends only on the modulus of elasticity, and not on the working strength of the material; or, in other words, the strength of a flue to resist a collapsing or buckling pressure depends on its rigidity, whereas its strength to resist a crushing pressure depends on the strength of the material. The two values are equal when

$$S = \frac{4}{3} Et^2 : D^2.$$

Here S is the plastic limit of the material. This formula expresses the conditions under which a flue would give way simultaneously both by crushing and collapsing. Assuming that $E \div S = 1200$, we have

$$\frac{t}{D} = \sqrt{\frac{3}{4800}} = \frac{1}{40}.$$

This means, that as long as the ratio of t to D is greater than $\frac{1}{40}$, a perfectly cylindrical tube, unsupported at its ends, will give way by crushing and not by collapsing. Boiler tubes are not supported at their ends, which is the same thing as if they were of infinite length; yet, on account of their relative great thickness, it is not wrong to estimate their strength by the formula $p = 2tS : D$ for cylindrical furnaces, the ratio $t : D$ is always less than 0·025.

Corrugated Flues.—By corrugating a flue, I, the moment of inertia of one inch of its section is increased from $\frac{t^3}{12}$ to $\frac{t^3}{12} + \frac{t \cdot h^2}{8}$, where h is the depth of the corrugations, and we have

$$p = 32 \cdot \frac{E}{D^3}\left(\frac{t^3}{12} + \frac{t \cdot h^2}{8}\right).$$

Let $h = 1\frac{1}{2}$ in., then $\frac{t^3}{12}$ can be neglected, and we find that the crushing and collapsing pressures are equal when

$$p = \frac{r \cdot t \cdot S}{D} = \frac{4 \cdot t \cdot h^2 \cdot E}{D^3};$$

that is, when $\frac{1 \cdot S}{2 \cdot E} = \frac{h^2}{D^2}$. This shows that even very slight corrugations will make ordinary boiler flues independent of their end supports (see Dr. F. Grashoff, 1866, pp. 232, 235). These deductions suggest that the higher the pressures on furnaces, the wider may the stiffening rings be placed apart.

Influence of Rings and Furnace Ends.—The previous formulæ take no account of the strengthening effect of rings or furnace ends. One way of dealing with this question is to regard the shell of a furnace as if it were a wide flat column, supported at its ends by two solid guides (fig. 173).

Let L be one-eighth of the furnace circumference, whose diameter is D, and let l equal half its unsupported length, while q is the circumferential thrust per inch of length.

Then, if we take a narrow vertical strip (fig. 173A), the bending moment would be $q \cdot y$. This would produce a curvature

$$\frac{1}{\rho} = \frac{d^2y}{dx^2} = \frac{q \cdot y}{E \cdot I}.$$

If, on the other hand, we divide the shell into horizontal strips, they will be bent as shown in fig. 174. The external forces producing this deflection are due to certain horizontal shearing forces, which when

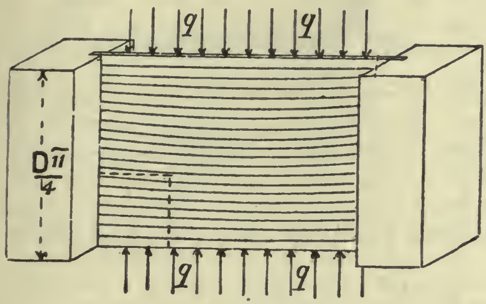


FIG. 173

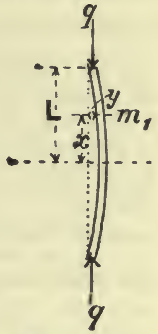


FIG. 173A

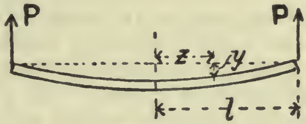


FIG. 174

summed up can be represented by an imaginary horizontal pressure p, which is balanced by P at either end. The slight extra relief due to the action of force q on the radius ρ may be neglected (fig. 175).

We then have

$$m_1 = \mathrm{E} \,.\, \mathrm{I} \frac{d^2y}{dx_1{}^2} = q \cdot y - \int_{x_1}^{\mathrm{L}} p\,(x - x_1)\, dx + (\mathrm{L} - x_1) \int_{x_1}^{\mathrm{L}} p \,.\, dx,$$

$$m_2 = \mathrm{E} \,.\, \mathrm{I} \frac{d^2y}{dz_1{}^2} = (l - z_1) \int_{z_1}^{l} p \,.\, dz - \int_{z_1}^{l} p\,(z - z_1)\, dz.$$

It is evident that to work out these formulæ would require more space than can be spared, and as the result, like the previous ones, is only applicable to perfectly circular furnaces, it will not be carried further. In practice the problem is made more complicated by the back end being irregularly supported. This is also shown in fig. 173, which represents the case of a furnace whose saddle is indicated by a vertical dotted line and is supported by the tube plate, while the bottom extends the whole length as far as the combustion chamber back plate.

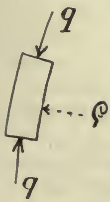


FIG. 175

Ribbed and Flanged Furnaces.—Another way of looking at the problem, and one which is intimately mixed up with the above, is the relief afforded to the circumferential stresses by the end plates or stiffening rings. Thus in fig. 176 the line AB represents the original position of the cylindrical part of the flue, whose section is shown above. CD is its position when the diameter has decreased under pressure if rings are fitted to the ends, while EF would be its position if no rings were fitted. The circumferential stresses are, of course, proportional to the compressions δ_1 and δ_o, showing that the relief afforded at the centre is $\frac{y_o}{\delta_1 + y_1}$. In this fig. 176 y_o and y_1 have the same meaning as in fig. 169, p 195, and could, if required, be calculated by the formulæ to be found there. When doing this it must not be

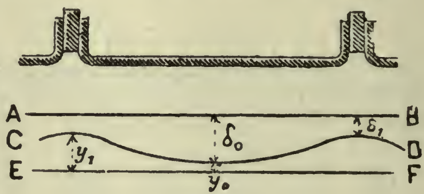


FIG. 176

forgotten that $\frac{dy}{dx} = 0$ both when $x = 0$ and when $x = l$, and further that approximately

$$y_1 = \frac{p_1 \cdot D^2}{4 \cdot E \cdot t} - \frac{D \cdot Q}{2 \cdot E \cdot a},$$

where a is the excess sectional area of the strengthening ring over and above that of the cylindrical part, and Q has the same value as on p. 195.

Oval Furnaces.—The same reasoning which led (fig. 172) to a limiting pressure for a perfectly cylindrical furnace has to be modified if the furnace was originally elliptical. Let $2 \cdot \Delta$ be the difference between the major and minor axes before straining, and let $2 \cdot \delta$ be the difference after the external pressure has been applied; then it can easily be shown that the furnace still remains elliptical, and that

$$\delta = \Delta \cdot \frac{16 \cdot E \cdot I}{16 \cdot E \cdot I - K \cdot D^2} = \Delta \cdot \frac{32 \cdot E \cdot I}{32 \cdot E \cdot I - p \cdot D^3}.$$

Example: Let $\Delta = \frac{1}{2}$ in., $t = \frac{1}{2}$ in., D = 40 in., p = 100 lbs., E = 30,000,000 pounds; then

$$\frac{\delta}{\Delta} = \frac{1}{1 - 0{\cdot}64} = 2{\cdot}78$$

$$\delta = 1{\cdot}4 \text{ ins.}$$

This is of course only true for long furnaces where the supporting influence of the ends can be neglected. In a corrugated flue whose moment of inertia of section is about ten times greater than the above half-inch plate, the ratio would be

$$\frac{\delta}{\Delta} = \frac{1}{1 - 0{\cdot}064} = 1{\cdot}068.$$

This shows how enormously rigid these furnaces are, even although they, like long plain ones, have no effective end support.

When the furnaces are short, or when strengthening rings are fitted, the problem grows to be a very complicated one, every deformation producing secondary strains and stresses. The approximate formulæ which the author has worked out are far too complicated to be of practical value. It may, however, be pointed out that the amount of deflection is a fair measure of the stresses. Thus, in the above example of a plain furnace, where $\frac{\delta}{\Delta} = 2{\cdot}78$ for an infinite length, it is probable that for a length of 6 ft. the elastic deformation $\delta - \Delta$ would be reduced from $1{\cdot}78 . \Delta$ to about $0{\cdot}9 . \Delta$, and that for shorter lengths this deformation would diminish inversely as the square of the length, being only $0{\cdot}1 . \Delta$ for a 2-ft. length.

In a truly cylindrical furnace of the above dimensions the circumferential compression stress is 4,000 lbs. If the length is infinite, and $\delta = 2{\cdot}78 . \Delta$, then the additional stress due to bending is 33,360 lbs., making a total of 37,360 lbs.

In a corrugated furnace of equal ellipticity the total circumferential stress would be 16,816 lbs. If the plain furnace is made only 6 ft. long its circumferential bending stresses would be reduced approximately by one half, making a total stress of, say, 25,000 lbs., and for a 2 ft. length the total stress would hardly exceed 6,000 lbs.

As pointed out on p. 28 fierce fires tend to make furnaces oval, while scale, greasy deposits, &c., raise the mean temperature of the plates so much that the limit of elasticity is lowered (see p. 190), but even then, although the plates do give way a little, the flanged rings, if fitted, give ample support to the cylindrical portions of the furnaces, showing that in long furnaces their circumferential strengths are of importance, whereas short furnaces may almost be treated as flat plates supported between beams—in this case strengthening rings.

Shearing Stresses in Furnaces.—Above the furnace bars the temperature of the furnace plates is higher than that below the bars. This is due partly to the fire on one side and to scale or grease on the other. A heated furnace top is therefore longer than its bottom, resulting in a slight general bend, which is in itself of little importance; but at the boundary of heat and cold very severe shearing stresses are set up, particularly at the furnace ends. Suppose that the mean temperature of the furnace plate above the bars is 20° F. higher than below the bars, and that there is a zone of 2 in. alongside the firebars where the change of temperature is a gradual one. Then, if the furnace is 6 ft. long, giving a span of 3 ft. on either side of the grate centre, the forward and back furnace top would grow $\frac{1}{250}$ in. longer than the bottom: the shearing angle produced at the front end would therefore be $\frac{1}{2} . \frac{1}{250}$; and, as the modulus of shearing elasticity is 11,600,000 (see p. 166), the shearing stress would be 23,200 lbs. per square inch. In a longer furnace, with a fiercer fire, with cold water fed into the boiler bottom, and scale on the furnace above the bars, these shearing stresses may grow to be very serious. They probably account for the leakages at the furnace fronts and back ends just below the bars.

Longitudinal Stresses in Furnaces.—As in beams and in shell plates, so there will be cross stresses in furnaces, and it is of importance, particularly when testing them, to be able to ascertain their intensities. Thus, in the case of the above-mentioned deformed oval furnace there will be longitudinal tension and compression stresses amounting to about 4 tons per square inch, all tending to make the ends of the furnace more oval than its centre.

Perfectly circular furnaces tend to elongate under pressure, just as boiler shells contract. When testing a furnace in a shell to which it is firmly riveted at the ends, longitudinal stresses are set up. It will be found that the furnace is then subjected to a longitudinal compression stress, which is about 30 per cent. of its circumferential stress, so that under these conditions, which are not necessarily reproduced in practice, the plain furnace plates are subjected to a shrinking stress, while this is not the case with corrugated ones. Plain and also ribbed furnaces might therefore be expected to do better in a boiler than in the test apparatus.

This view is supported by the fact that when experimentally testing furnaces with welded seams they usually give way along a line exactly opposite the riveted seam of the shell, being the point where the longitudinal stresses would be greatest.

Experiments on Cylindrical Furnaces.—The earliest experiments on the influence of length on the strength of furnaces were made for the Manchester Steam Users' Association, under the direction of Sir William Fairbairn, who published them in 1856. With the exception

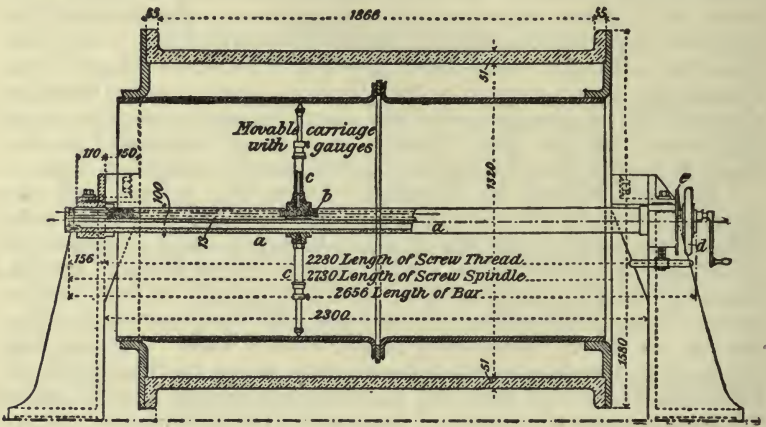


Fig. 177

of his Nos. 7, 8, 10 and 11, which differ among themselves by about 35 %, his results agree generally with the view that the strength of a flue varies inversely as its length. But a careful examination of his publications shows that he assumed the 150 sq. ft. of tinned sheet iron, of which he made the tubes, to be of one thickness, whereas it is more than probable that differences of at least $\frac{1}{100}$ in. must have existed, and this represents 20 per cent. of the thickness, or 35 per cent. of the square of the thickness. It is, further, but reasonable to suppose that the longer tubes would contain more weak parts than the short ones, and further they would be more likely to be more oval than the short ones, and would therefore be the first to collapse. For these reasons the author does not attach the same importance to these experiments as some of the mathematicians who have evolved elaborate empirical formulæ from them. Fairbairn further made a few experiments on full-sized furnaces, but, as has been pointed out, in one case either the furnace or the boiler diameter must be wrongly stated. Further experiments were made by R. V. J. Knight, 'C. E.,' 1878, vol. li. p. 133; by the Washington Navy Yard, 1874, published by W. H. Shock, 1880, p. 113; by the Board of Trade; by Carman, 'University Illinois Bul.,' 1906, vol. 3; by Stewart, 'Am. M. E.,' 1906, p. 730, and lastly, some very elaborate experiments by the German Admiralty at their works at Danzig. The dimensions are given in millimetres. The total number of readings taken with each furnace frequently amounted to more than five hundred, and are a most valuable contribution to the practical part of the science of elasticity. The gauging instrument, fig. 177, consists of two stout brackets carrying a boring bar with a screw in a slot, and on this bar is a carriage with movable gauges, which is brought into contact with the furnace plate for measuring purposes.

Hydraulic Tests of Plain and Flanged Cylindrical Furnaces

Experimenter or Works	Lengths		Diameters		Thickness		Collapsing		Material	Tenacity	Joint		Remarks
	Total	Between Rings	Max. External	Diff.	At bulge	Mean	Pressure	Stress			Long.	Ring[1]	
	ins.	ins.	ins.	in.		ins.	lbs.	tons		tons			
Fairbairn	60	...	14⅝	...	...	⅛	125	3·3	I	...	Lap.	L	
„	37	...	9	...	...	·14	262	3·7	I	...	Lap.	...	
„	37	...	9	...	...	·14	378	5·4	I	...	Butt	...	
„	276	...	7·87	...	...	·157	110	1·2	I	...	Lap.	L	
„	61	...	18¾	...	...	¼	420	7·0	I	...	Lap.	L	
„	360	...	33½	...	...	·34	99	2·2	I	...	Lap.	L	
„	420	...	42	...	...	⅜	97	2·7	I	...	Lap.	L	
„	300	...	42	...	...	⅜	127	3·2	I	...	Lap.	L	
Knight	48	24	36	...	...	¼	235	7·5	I	...	Weld	A	
„	48	..	36	...	...	¼	217	7·0	I	...	Weld	...	
„	48	24	36	...	...	⅜	468	10·0	I	...	Weld	A	
„	48	...	36	...	...	⅜	390	8·3	I	...	Weld	...	
Washington Navy Yard	71	35½	54	½	...	¼ *b*	105	5·0	I	...	Butt	B	
„	71	35½	54	½	...	¼	120	5·8	I	...	Butt	B	
„	71	35	54	...	...	¼	133	6·4	I	...	Butt	A	
„	71	36	54	...	...	¼	130	6·3	I	...	Butt	A	
Board of Trade	38½	...	44⅜	...	...	11/32	200	5·8	I	...	Lap.	...	Donkey boiler
„	46	...	43	...	...	⅜	180	4·6	I	...	Lap.	...	Donkey boiler
„	86	...	38	...	...	⅜	187·5	4·2	I	...	Weld	...	
„	86	...	38	...	...	½	450	7·6	I	...	Weld	A	
„	108	...	37	...	...	17/32	260	4·0	I	...	Lap.	L	Old boiler
Dalzell Works	94	...	32·94	...	...	·42	370	6·5	S	28·1	Weld	...	Compressive strength of this material was under 18 tons
„ „	94	46¼	32·94	...	...	·42	600	10·5	S	28·1	Weld	A	„
„ „	94	46¼	32·94	...	...	·42	600	10·5	S	28·1	Weld	A	No collapse
Hall, Russell & Co.	77⅜	18¼	43	9/64	...	½	700	13·5	S	...	Weld	A	
„	77⅜	19	43	9/64	...	15/32	740	15·2	S	...	Weld	A	No collapse
J. Howden & Co.	84¼	23	43·09	...	...	·546	840	14·7	S	...	Weld	A	
„	84¼	23	43·09	...	...	·546	760	13·3	S	...	Weld	A	
„	84¼	23	43·09	...	...	·546	840	14·7	S	...	Weld	A	
„	84¼	23	43·09	...	...	·546	835	14·6	S	...	Weld	A	

[1] A stands for Adamson's ring. The lengths between rings are variously measured from centre to centre—this is the case in the German Admiralty experiments—or between ends of rounds of flange. If the end lengths of furnaces are not flanged, the lengths are measured to centres of rivet holes.
L stands for circumferential lap joints; they add a little to the furnace strength.
B stands for circumferential butt strap joints. They also add to the furnace strength.

As far as practicable, all these experiments mentioned above are collected together in the following table; those on plain furnaces will also be found in 'N. A.,' 1899, vol. li. p. 142, M.S.U.A. 1899. That paper also contains a Table II. of 119 cases of accidentally collapsed furnaces, about which sufficient details were known to calculate the collapsing stresses; of these, however, 16 furnaces were oval and 22 doubtful. That paper also contains Table IV. of

30 cases of furnaces which, although oval, had worked safely for years.

The formula deduced from these experiments is

$$S \text{ tons per sq. inch} = 3 + \frac{169{,}000\ T^2}{13{,}000\ T^2 + L^2},$$

where L and T are the furnace lengths and plate thicknesses in inches. Stewart obtains the following empirical formula from his experiments

$$S = 2{\cdot}75ET^2 : (1 - \mu^2)D^2.$$

Hydraulic Tests of Plain and Flanged Cylindrical Furnaces made by the German Admiralty, Danzig

Number	Lengths		Diameter		Thickness		Collapsing		Material	Tenacity	Joint or Seam		Remarks
	Total, Rivet to Rivet	Ring to Ring or Rivet	External Maximum	Difference	Mean or Nominal	At Bulge	Pressure	Stress			Longitudinal	Circumferential[1]	
	ins.	ins.	ins.	ins.	ins.	ins.	lbs.	tons		tons			
5	79	...	40·4	·25	·45	...	under 455	8·8	I	23·8	Butt	...	Collapsed at say 440 lbs.
13	78½	42½	40·7	·07	·59	·575	997	15·7	I	23·0	„	A	Collapsed
13	78½	36	40·7	·07	·59	...	997	15·4	I	23·0	„	A	Uninjured
1	78	41½	40·0	·10	·33	·323	under 342	9·1	I	21·7	„	A	Collapsed near butt
1	78	36½	40·1	·20	·33	...	say 330	8·9	I	21·7	„	A & H	Uninjured
6	78	41½	40·3	·17	·45	·440	641	13·1	I	26·3	„	A	Collapsed near butt
6	78	36½	40·2	·07	·45	...	641	12·8	I	26·3	„	A & H	Uninjured
10	78½	30	38·3	·24	·51	...	812	13·6	I	23·7	Weld	A	„
10	78½	28	38·3	·13	·51	·510	812	13·6	I	23·7	„	A	Collapsed
10	78½	20½	38·4	·17	·51	·496	783	13·5	I	23·7	„	A	„ tenacity at bulge 18·4 tons
12	84	27	38·6	·06	·59	·606	1,138	16·2	I	24·9	Butt	A	Collapsed at reduced neck
12	84	30¼	38·6	·06	·59	...	1,138	16·6	I	24·9	„	A	Uninjured
12	84	26¾	38·7	·16	·59	...	1,138	16·6	I	24·9	„	A	„
2	78	28½	40·0	·04	·30	·295	427	12·9	I	23·9	„	A	Collapsed
2	78	26½	40·0	·25	·30	...	427	12·7	I	23·9	„	A & H	Uninjured
2	78	23	40·0	·08	·30	...	427	12·7	I	23·9	„	„	„
8	78	28¼	40·4	·15	·43	·423	641	13·7	I	24·4	„	A	Collapsed
8	78	26	40·4	·20	·43	...	641	13·4	I	24·4	„	A & H	Uninjured
8	78	23¾	40·4	·14	·43	...	641	13·4	I	24·4	„	„	„
3	78	41½	40·0	·04	·32	·307	427	12·4	I	26·6	„	„	Collapsed from end to hoop
3	78	36½	40·1	·22	·32	...	427	11·9	I	26·6	„	„	Uninjured
7	77½	41½	40·3	·28	·43	·420	under 641	13·5	I	23·9	„	„	Collapsed from end to hoop
7	77½	36	40·3	·63	·43	..	say 630	13·2	I	23·9	„	„	Uninjured
4	78	28½	40·3	·24	·31	...	455	15·2	I	22·8	„	„	„
4	78	26½	40·2	·24	·31	·307	427	12·5	I	22·8	„	„	Collapsed. The hoop rivets tore at 455 lbs.
4	78	23	40·2	·08	·31	...	455	13·2	I	22·8	„	„	Uninjured
9	78	28¼	40·4	·11	·45	...	783	15·7	I	21·9	„	„	„
9	78	26	40·4	·10	·45	·443	783	15·7	I	21·9	„	„	Collapsed. The hoop rivets tore later at 455 lbs.
9	78	23¾	40·4	·12	·45	...	783	15·7	I	21·9	„	„	Uninjured
11	78½	30	38·5	·12	·49	·472	712	13·0	S	24·1	Weld	A	Collapsed
11	78½	28	38·5	·06	·49	·468	683	12·5	S	24·1	„	A	„
11	78½	20½	38·5	...	·49	...	769	13·5	S	24·1	„	A	Uninjured

[1] A stands for Adamson's ring. See footnote of previous table.
H stands for angle-iron hoop. The furnace lengths are measured from ring to ring or end seam, but not to hoop. If the hoops are considered to give support, then in each line marked with H the lengths in the third column will have to be halved.

Hydraulic Tests of Corrugated Furnaces (Fox's). Nos. 1–7, made at Leeds Forge. Nos. 14–17 made by German Admiralty, Danzig

Number	Length: Total	Length: Of Flat	Diameter: Extreme Mean	Diameter: Difference	Thickness[1]: Nominal or Mean	Thickness[1]: At Bulge	Collapsing: Pressure	Collapsing: Stress[1]	Tenacity	Remarks. Position of Bulgings
	ins.	ins.	ins.	ins.	ins.	ins.	lbs.	tons	tons	
1	81	...	$35\frac{7}{8}$	...	·52	...	900	13·9	22·7	No details
2	76	$5\frac{7}{8}$	$34\frac{1}{8}$	$\frac{9}{32}$	·349	...	830	18·1	29·0	On flat
3	76	6	$34\frac{1}{4}$	$\frac{5}{32}$	·378	...	800	16·2	29·2	On flat. Thickness 0·331 ins.
4	79	6	$33\frac{3}{8}$	$\frac{4}{32}$	·452	·419	1,130	18·8	29·3	Second corrugation
5	80	$6\frac{9}{16}$	34·5	$\frac{4}{32}$	·468	...	1,090	17·9	29·0	On flat
6	$77\frac{1}{2}$	$6\frac{5}{16}$	33·6	$\frac{7}{32}$	·574	·551	1,400	18·3	28·5	Eleventh corrugation
7	78	$6\frac{1}{4}$	34·9	$\frac{8}{32}$	·575	·542	1,410	19·1	29·4	Front corrugation
14	90	$3\frac{1}{2}$	40·2	$\frac{4}{32}$	·492	·362	925	16·9	23·7	Total length after standing 15 mins.
15	79	$11\frac{1}{2}$	40·2	$\frac{2}{32}$	·559	·462	996	16·0	25·9	On flat. External diameter 38·5 ins.
16	79	4	42·6	$\frac{2}{32}$	·642	·522	1,066	15·7	24·2	Near centre after standing 3 mins.
17	$80\frac{1}{2}$	$6\frac{1}{2}$	48·3	$\frac{1}{30}$	·652	·540	996	16·5	23·6	Near centre after standing 2 mins.

Hydraulic Tests of Various Corrugated Furnaces

Number	Length: Total	Length: Of Flat	Diameter: External Mean	Diameter: Difference	Thickness[1]: Mean or Nominal	Thickness[1]: At Bulge	Collapsing: Pressure	Collapsing: Stress[1]	Tenacity	Remarks
	ins.	ins.	ins.	ins.	ins.	ins.	lbs.	tons	tons	
Farnley Spirally Corrugated Furnaces, 2 ins. deep, 6 ins. pitch										
1	$77\frac{3}{4}$	...	39·2	...	·559	...	835	13·1	...	
2	$78\frac{3}{4}$	...	39·1	...	·548	...	850	13·8	...	
3	$78\frac{3}{4}$	...	38·9	...	·442	...	670	13·2	...	
4	$78\frac{3}{4}$	...	39·4	...	·446	...	570	11·2	...	
5	$78\frac{1}{16}$	...	39·3	...	·354	...	515	12·8	...	
Holmes' Corrugated Furnaces										
6	84	...	35·4	$\frac{4}{32}$	515	...	950	14·6	27·3	One corrugation and adjacent two plain parts collapsed
7	84	...	35·6	$\frac{12}{32}$	·452	...	750	13·2	27·5	One plain part collapsed
8	84	...	35·5	$\frac{6}{32}$	·557	...	920	13·1	26·8	,, ,, ,,
Morrison's Suspension Furnace										
9	$77\frac{1}{2}$	$4\frac{3}{4}$	34·1	$\frac{11}{32}$	·325	...	795	18·6	27·6	Collapsed at plain part and at weld
10	$79\frac{1}{2}$	$4\frac{1}{2}$	34·3	$\frac{3}{32}$	·373	...	900	18·5	27·3	,, ,, second corrugation
11	$77\frac{1}{2}$	$4\frac{1}{2}$	34·4	$\frac{12}{64}$	·470	...	1,100	18·0	26·9	,, along weld
12	$78\frac{3}{4}$	$4\frac{3}{4}$	34·7	$\frac{11}{32}$	·452	...	1,050	18·0	26·6	,, ,, ,,
13	$79\frac{1}{2}$	$4\frac{3}{4}$	34·7	$\frac{3}{32}$	·530	...	1,300	19·0	27·5	,, first to third corrugations
14	79	5	34·1	$\frac{7}{32}$	·567	...	1,340	18·1	27·1	,, at plain part

[1] Note for last two tables. In order to obtain the true stresses in the material the total section of the furnace plate should be measured. This is done in the German Admiralty experiments, where the thickness of the furnace ends is multiplied into the furnace length. In the English experiments the thicknesses of the corrugated plates are taken; they should have been multiplied into the total lengths of the corrugations, and not into the furnace lengths. On an average, the corrugated plates are 11 per cent. thinner than the ends of the furnaces; all the English stresses should therefore be reduced about $12\frac{1}{2}$ per cent.

Hydraulic Tests of Purves' Ribbed Furnaces

Number	Length: Total	Length: Of Flat	Diameter: Over Plain Part	Diameter: Difference	Thickness: Of Plain Part	Thickness: At Bulge	Collapsing: Pressure	Collapsing: Stress	Tenacity	Remarks
	ins.	ins.	ins.	ins.	ins.	ins.	lbs.	tons	tons	
Tests made at Sir John Brown & Co.'s Works, Sheffield										
1	$78\frac{3}{8}$	$13\frac{13}{16}$	38·6	$\frac{7}{32}$	·568	...	740	11·2	27·2	Collapsed on flat
2	$78\frac{3}{8}$	$13\frac{13}{16}$	38·7	$\frac{11}{32}$	·546	...	760	12·1	26·6	" "
3	$79\frac{1}{2}$	$7\frac{1}{2}$	38·6	$\frac{8}{32}$	·295	...	650	19·0	25·9	" between first and second rib
4	$79\frac{1}{2}$	$7\frac{1}{2}$	38·6	$\frac{6}{32}$	·311	...	635	17·6	27·9	" " sixth and seventh rib
5	$79\frac{1}{2}$	$7\frac{1}{2}$	38·8	$\frac{15}{32}$	·509	...	800	13·6	26·1	" end to end
6	$79\frac{1}{2}$	$7\frac{1}{2}$	38·9	$\frac{8}{32}$	·559	...	800	12·4	27·9	" " "
7	$79\frac{1}{2}$	$7\frac{1}{2}$	38·8	$\frac{10}{32}$	·611	...	875	12·4	27·6	" " "
8	$79\frac{1}{2}$	$7\frac{1}{2}$	38·8	$\frac{11}{32}$	·613	...	873	12·3	28·3	" " "
9	...	$9\frac{1}{4}$	38·8	...	·307	...	675	19·0	28·8	
10	..	$9\frac{3}{8}$	38·7	...	·362	..	700	19·0	28·0	
11	...	$9\frac{3}{8}$	38·7	...	·461	...	870	16·3	27·3	
12	...	$9\frac{3}{8}$	38·7	...	·466	...	950	17·6	29·3	
13	...	$9\frac{1}{2}$	38·6	...	·585	...	1,065	15·7	28·7	
14	...	$9\frac{11}{16}$	38·6	...	·578	...	1,145	17·1	27·4	
15	...	...	38·7	...	·522	...	1,020	16·9	...	
Test made at German Admiralty Works, Danzig										
18	79	$9\frac{1}{2}$	38·0	$\frac{4}{32}$	...	·524	855	13·8	26·5	Collapsed end to end

Curved Plates.—With oval furnaces, in which there is only a slight variation of curvature, the bending stress due to m_1 (p. 197) would be $3 . \delta : t$ times the mean circumferential stress. What, it may be asked, will be the effect if the curvature changes still more rapidly, viz. from, say, 24 ins. to a straight line, as often happens in combus-

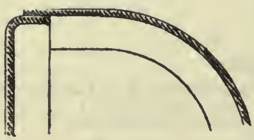


Fig. 178

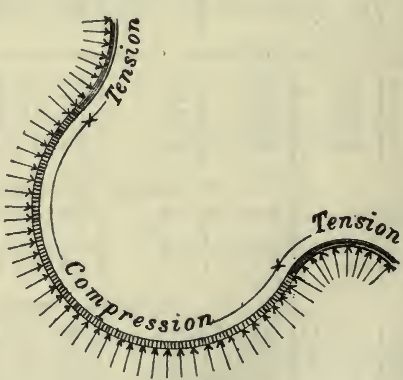


Fig. 179

tion chamber tops (fig. 178)? or if it changes from one side to the other, as in the case of combustion chamber bottoms (fig. 179)? When the pressure is on the convex side, the circumferential stresses are all compression, while when it acts on the concave side they are tension. This is shown in fig. 179, which represents part of the bottom of a combustion chamber with two furnaces.

As sketched, the conditions are more unfavourable than when building an arched bridge without abutments and without any support, for the ends are in this case actually being pulled at. The stresses which are necessarily set up are of the most complicated nature. Thus there will be circumferential shearing stresses, which increase very seriously towards the forward and back ends of these plates, particularly along the line of change of flexure. Horizontal seams should not be fitted here. These shearing stresses are also very severe on the circumferential seams of the furnace back ends and of the combustion chamber backs; and the extra circumferential compression stresses which are thereby set up in the furnace saddles may have been the cause of occasional mishaps. The staying of these parts to the boiler shell will necessarily afford local relief, but very often leads to excessive bending strains in the screwed stay near the water level.

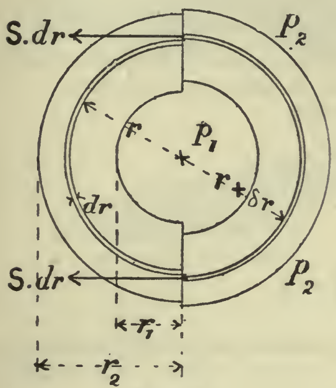


Fig. 180

Thick Cylindrical Shells.—This subject would be out of place here were it not that some of the mathematical deductions will assist in estimating the distribution of stresses in riveted joints.

Let the left- and right-hand side of fig. 180 represent half the tube, respectively with and without the internal and external pressures p_1 and p_2. Then, dealing with the thin-walled ring dr of the radius r, we find that if p is the radial pressure at this distance from the centre, and if S is the circumferential stress, — S being circumferential tension, we have

$\mathrm{S}\,.\,dr = (p + dp)\,(r + dr) - p\,.\,r.$ Neglecting $dp\,.\,dr$, we have

$$(1) \qquad \mathrm{S} - p = r\,.\,\frac{dp}{dr}.$$

The radius of the ring has now decreased

$$(2) \qquad \delta r = -\,r\,.\left(\frac{\mathrm{S}}{\mathrm{E}} - \frac{p}{\mu\,.\,\mathrm{E}}\right).$$

The thickness dr of the ring under consideration has decreased by

$$(3) \qquad \delta\,(dr) = d - dr\left(\frac{p}{\mathrm{E}} - \frac{\mathrm{S}}{\mu\,.\,\mathrm{E}}\right).$$

If there are no longitudinal stresses in the cylinder, then its length increases

$$(4) \qquad \delta l = +\,\frac{l\,.\,(\mathrm{S} + p)}{\mu\,.\,\mathrm{E}}.$$

Differentiating (2) we have

$$\frac{d\,(\delta r)}{dr} = -\,\frac{\mathrm{S}}{\mathrm{E}} + \frac{p}{\mu\,\mathrm{E}} - \frac{r}{\mathrm{E}}\left(\frac{d\mathrm{S}}{dr} - \frac{dp}{\mu\,.\,dr}\right).$$

But $d(\delta r) = \delta(dr)$, and combining this with (3)

$$(S - p)\left(1 + \frac{1}{\mu}\right) + r\left(\frac{dS}{dr} - \frac{dp}{\mu . dr}\right) = o.$$

Substituting the values of $S - p$ from (1) we find

$$(5) \qquad \frac{dp}{dr} + \frac{dS}{dr} = o.$$

It follows that $p + S$ is constant for all values of r, which shows that δl (4) is also constant, and need not be taken into account.

Differentiating (1) leads to the equation

$$(6) \qquad \frac{dS}{dr} - 2\frac{dp}{dr} - r\frac{d^2p}{dr^2} = o,$$

and combining this with (5) we get

$$(7) \qquad 3\frac{dp}{dr} + r\frac{d^2p}{dr^2} = o.$$

Integrating this and introducing the various constants leads to the following equations:

$$(8) \qquad p_1 + S_1 = p + S = 2\frac{(p^2r_2^2 - p_1r_1^2)}{r_2^2 - r_1^2};$$

$$S_1 = \frac{2 . p_2 . r_2^2 - p_1 . (r_1^2 - r_2^2)}{r_2^2 - r_1^2};$$

$S_1 - p_1 = 2\frac{p_2r_1^2 - p_1r_2^2}{r_2^2 - r_1^2}$. According to Guest's law this difference of stress which is the internal pressure added to the circumferential stress should not exceed a certain limit. If $r_2 = \propto$, $S_1 = -p_1$; the combination of S_1 and p_1 is a pure shearing stress of the intensity $\sigma_1 = p_1 = -S_1$. At the radius r we have

$$(9) \qquad p = \frac{(p_1 - p_2)r_1^2r_2^2}{r^2(r_2^2 - r_1^2)} + \frac{p_2r_2^2 - p_1r_1^2}{r_2^2 - r_1^2}$$

$$(10) \qquad S = -\frac{(p_1 - p_2)\,r_1^2r_2^2}{r^2\,(r_2^2 - r_1^2)} + \frac{p_2r_2^2 - p_1r_1^2}{r_2^2 - r_1^2}.$$

From this it will be seen that $p + S = o$ when $p_2r_2^2 = p_1r_1^2$. In that case too the metal is exposed to uniformly distributed spiral shearing stresses, σ, as shown in fig. 181: its effects are often seen around punched holes.

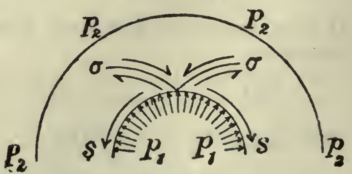
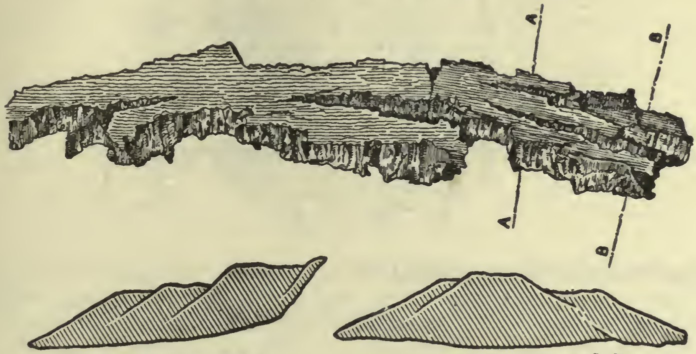


Fig. 181

This condition is also approached, at least near the internal circumference, when r_2 is very large. For practical illustrations see 'C. E.,' 1893, vol. cxi. p. 212, and fig. 182, which represents a piece of a steel shell burst by a high explosive. All the fractured surfaces are sheared and inclined at an angle of 45°.

Other conclusions to be drawn are—

1st. The maximum stresses, either tension or compression, are always found at the internal circumference.



Enlarged Section at A.A Enlarged Section at B.B

Fig. 182

2nd. If the internal pressure $p_1 = o$, then the maximum circumferential compression stress is

$$S_1 = + \frac{p_2 \, . \, 2 \, . \, r_2^2}{r_2^2 - r_1^2}.$$

An approximately correct formula for cylindrical tubes would therefore be

$$S_1 = p_2 \frac{D + t}{2 \, . \, t},$$

where D is the external diameter and t the thickness of plate.

3rd. If the external pressure $p_2 = o$, then the maximum circumferential tension stress is

$$S_1 = - p_1 \frac{r_2^2 + r_1^2}{r_2^2 - r_1^2} \text{ and } S_1 - p_1 = - 2p_1 \frac{r_2^2}{r_2^2 - r_1^2}.$$

For boiler shell plates an approximately correct formula is therefore

$$S_1 = - p_1 \frac{D_1 + t}{2 \, . \, t} \text{ and } S_1 - p_1 = - p_1 \frac{D_1 + 2t}{2t},$$

where D_1 is the internal and $D_1 + t$ the mean, and $D_1 + 2t$ the external diameter of the shell. In both these cases, therefore, the maximum simple stress has to be estimated as if the pressure were acting on an imaginary tube whose diameter is a little larger than the actual one, and $S_1 - p_1$ has to be estimated as if the internal pressure were acting on the external diameter.

These corrections are too unimportant to be taken into serious count in practice, except, as will shortly be explained, in the case of rivet holes.

The deformations of thick-walled tubes are found by inserting the values of S_1 and p_1 in equation (2):

$$\frac{\delta r_1}{r_1} = \frac{p_1}{E}\left\{\frac{r_2^2 + r_1^2}{r_2^2 - r_1^2} + \frac{1}{\mu}\right\} - \frac{2 \cdot p_2 \cdot r_2^2}{E \cdot (r_2^2 - r_1^2)}.$$

$$\frac{\delta r_2}{r_2} = \frac{2 \cdot p_1 r_1^2}{E \cdot (r_2^2 - r_1^2)} - \frac{p_2}{E}\left\{\frac{r_2^2 + r_1^2}{r_2^2 - r_1^2} - \frac{1}{\mu}\right\}.$$

An approximate formula for boiler shells would be

$$\frac{\delta r_2}{r_2} = \frac{p_1}{E} \cdot \frac{2 \cdot r_1 - t}{2t}.$$

and for boiler furnaces it would be

$$\frac{\delta r_1}{r_1} = \frac{p_2}{E} \cdot \frac{2 \cdot r_2 + t}{2t},$$

Plastic Tubes.—As regards the distribution of stresses in thick-walled cylinders when the plastic limit has been reached, little can be learnt until the plastic properties of steel subjected to compound stresses are better known. However, as the value of $\frac{1}{\mu}$ changes when this limit is passed from about $\frac{1}{3}$ to $\frac{1}{2}$, it is evident that it ought to be introduced into the various formulæ as a function of r, and then $\frac{\delta l}{l}$ is not any more a constant value, nor is $S + p$; and what complicates matters still more is that longitudinal stresses are set up which increase with increasing length of tube. The stresses in a gun barrel which is on the point of bursting are therefore not at all as simple as is usually assumed. A thorough analysis of the shape of the swelling of the material round a carefully drifted hole (fig. 183), similar to that carried out in the case of torsion tests, could perhaps be made to throw light on this subject.

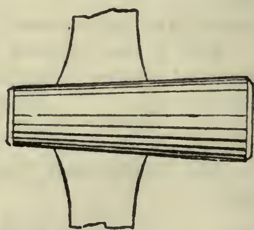


Fig. 183

Riveted Joints.—At first sight it would appear that there is no simpler problem than to find the stresses in a riveted joint. Given the sectional area of the perforated plate, and the sectional areas of the rivets, the average stresses ought to be easily calculated. But these are only the average stresses. To find the positions and intensities of the maximum stresses is far more complicated, but worth examining, not only on account of the importance of the subject, but also because the difficulty of dealing with it more in detail will make it clear how little is yet known about other problems of boiler mechanics which, even at first sight, strike one as more complicated than this one.

Deformation of Rivets.—Fig. 184 represents part of a butt-strapped joint. A force Q is being transmitted from the central (shell) plate to the two butt straps and in doing so the bearing pressures p_1 and p_2

come into existence. They tend to give the rivet a slight bend, and this deformation will cause an irregularity in their distribution, the

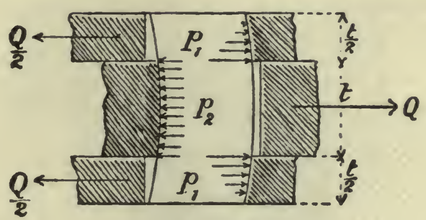


FIG. 184

pressure being proportional to the deformation. This is evidently a similar case to the one which presented itself when dealing with the influence of end plates on the distribution of stresses in cylindrical shells, and which, as was there shown, leads to the most complicated formulæ. Here it will be assumed that as long as the elastic limit has not been reached, p_2 is uniformly distributed over the thickness t of the shell plate, and p_1 is distributed over the thickness of the butt straps in the shape of a triangle. Then in the above case $p_1 = 2p_2 = \frac{2 \cdot Q}{t \cdot d}$.

Stress in Rivets.—These pressures produce bending and shearing stresses which, when reduced to right-angled resultants (see p. 167), act in directions which are indicated in figs. 185 and 186. In both the intensities are indicated by the shading of the various zones, and if placed

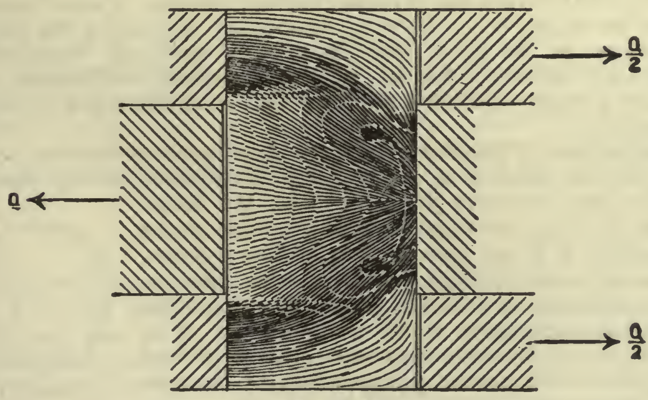


FIG. 185

over each other it would be found that for any particular point the two sets of stresses cross each other at right angles. Fig. 185 shows only compression stresses, and fig. 186 shows tension stresses at the left edge, which are gradually reduced and change into compression stresses at the right-hand centre. The stresses and their angles have been calculated on the assumption that the rivet diameter is equal to the thickness of the shell plate and twice as thick as the butt straps, and that the pressure is distributed as in fig. 184, but rounded off at the corners.

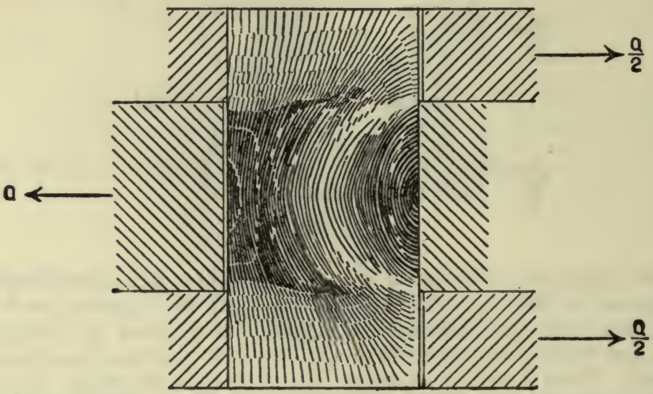


FIG. 186

Then the mean stress along the line of shear, which is the mean of the tension and compression stresses, is 1·13 ton per ton of mean bearing pressure, and the maximum stress in this line is 1·55 ton. On the upper part of the left edge there are some severe compression stresses, the maximum being 2·33 tons; and in the lower right-hand corner shrinking stresses are met with, consisting of two right-angled stresses of 1·2 and 1·45 ton per ton of mean bearing pressure. These diagrams and values might be used for the purpose of calculating the elastic deformation of the rivet, and the bearing pressures would then have to be modified; but as no very important deductions will be drawn from these results, the more correct curves have not been determined.

Distribution of Bearing Pressure.—Another correction has to be introduced on account of the irregular distribution of the bearing pressure over the rivet diameter. Let the dotted line (fig. 187) represent the imaginary outline of the rivet section if it had been free to shift its position. But having come in contact with the circumference of the hole, which is shown by a black line, a pressure is called into existence, which may be taken to be proportional to the distance between the black and dotted lines. This normal pressure between the rivet and the plate is shown in fig. 188. $p = p_o \cos \alpha$. Here p_o is the maximum bearing pressure. The sum of $p \cos \alpha$ for the diameter d of the rivet is $\frac{\pi}{4} \cdot p_o \cdot d \cdot t$. This is equal to the load Q on the rivet, and the maximum bearing pressure at the centre line is

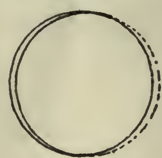


FIG. 187

$$p_o = \frac{4 \cdot Q}{\pi \cdot d \cdot t} = 1{\cdot}275 \frac{Q}{d \cdot t}.$$

In the butt straps, as has been explained, this pressure will perhaps be double as much, depending on the flexure of the rivet and compression of the rivet hole.

Fig. 188 shows that $d_1 = d \cos \alpha$, and that if the rivet were divided into numerous axial laminæ of equal thickness, the thrust on each one

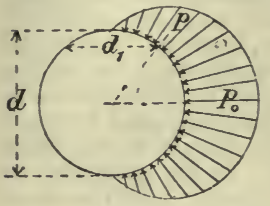


FIG. 188

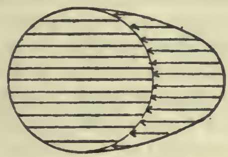


FIG. 189

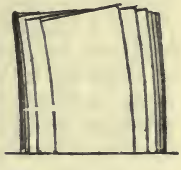


FIG. 190

would be proportional to its own width, and if not connected amongst each other they would be deflected by different amounts (see figs. 189 and 190).

As these independent motions cannot take place there must exist a series of cross stresses in the rivets, which prevent the laminæ from slipping. It is also probable that in adjusting themselves to their surrounding conditions the outer laminæ do not offer as much resistance as the inner ones, and therefore the normal pressure (fig. 188) is more likely to be distributed according to the formula $p = p_o \cos \alpha^3$, which, if true, would lead to the conclusion that the **maximum bearing pressure** to be found between the rivet and its hole is

$$p_o = \frac{3 \,.\, Q}{2 \,.\, d \,.\, t},$$

which is 50 % greater than the mean.

The Shearing Stress in a Rivet increases from nothing at either of its ends and at the centre of its length to a maximum at the plane of shear. In this plane it is greatest at the centre of the rivet, falling off in a parabolic curve towards the circumference. The maximum shearing stress in the centre is therefore about double the mean, viz.

$$\sigma = \frac{8 \,.\, Q}{\pi \,.\, d^2} = 2{\cdot}54 \frac{Q}{d^2},$$

which coefficient is nearly the same as $1{\cdot}55 \times 1{\cdot}5$ as found above.

Shearing and Bending Stresses.—If the leverage with which a load acts on a beam is equal to one-third of its thickness, then the limits of elasticity, both for tension compression and shear, are reached simultaneously (see p. 192). In a lap joint this condition most likely exists. In butt-strapped joints it will depend upon the thicknesses, and on the distribution of the bearing pressure, whether this is so or not. If the strap is equal to the thickness of the rivet, and the pressure distributed as in fig. 184, or if the thickness of the strap is $\frac{2}{3}$ of the rivet diameter, and p_o is uniformly distributed over the length, then the above condition exists. In other cases the rivet gives way first, either by bending alone or by shearing alone. Of course these remarks only apply up to the elastic limit, and even here they are seriously modified by the various cross stresses which it has only been possible to hint at.

Plastic Rivets.—As soon as the stresses increase to such an extent that either at one point or another the elastic, or even the plastic, limit has been reached, then the conditions are altogether changed. All the pressures are more evenly distributed. The shearing stress is similarly affected, and is probably uniformly distributed, as shown in fig. 167, p. 192. The ultimate strength of a rivet may, therefore, be estimated by those formulæ which give the mean stresses, while its working strength up to the elastic limit has to be found as explained above. It is, therefore, very misleading to assume that the elastic limit and ultimate strength of a rivet stand in the same relation as the elastic limit and ultimate strength of a simple bar. In the latter case the ratio is about as one to two, in the former it is more nearly as one to five. A nominal factor of safety of five leads to a construction in which the limit of elasticity of a rivet is just reached with the ordinary working pressure.

When the dimensions of a rivet have been so arranged that the limit of elasticity of the metal is reached simultaneously at various parts, which is of course the most advantageous condition, then it does not at all follow that, on increasing the load, rupture will also take place simultaneously at all these points. In fact, there can be no doubt but that the bending moments increase at a relatively greater rate than the load, while the axial stresses due to them increase irregularly. While this is going on, the shearing stresses distribute themselves more uniformly and sink into relative insignificance, so that a rivet designed on the best principles for working conditions would rupture primarily through bending, and at a lower load than another which is so designed that all the ultimate stresses are reached simultaneously. Of course in this case the rivet would not be an equally efficient one under working conditions. That unexplained actions of this sort do exist is proved by nearly all experiments on this subject, for it will be found that it is not always the part which is apparently most strained that gives way. In fact, the stronger part—either the rivet or the plate—seems as if it were always endowed with extra strength. Thus a joint may tear through the plate when its stress is, say, 28 tons, while that in the rivets is 14 tons. But by reducing the sectional area of the latter it is possible to construct a joint which will give way at, say, 20 tons, while the plate remains intact, though in this case it has been strained up to 32 tons or more.

Rivet Holes.—Circumferentially the bearing pressure in a rivet hole is of course distributed exactly in the same way as on the rivet. The stresses which are thereby produced in the surrounding material might be determined with considerable accuracy by a careful and exhaustive analysis; but this cannot be attempted here, and it will only be possible to recapitulate some of the views on this subject, and which are intended to be approximately correct.

Firstly, the stresses between the rivet holes are supposed to be uniformly distributed. This is evidently incorrect.

Secondly, the shell plates, or the straps, are supposed to be built up of a number of tapes or links, each one being of a sufficient section for carrying the load which is placed on it (figs. 191, 192). The depths of these links over the rivets must be made 50 % greater than their widths. Although this view rather begs the question, it seems

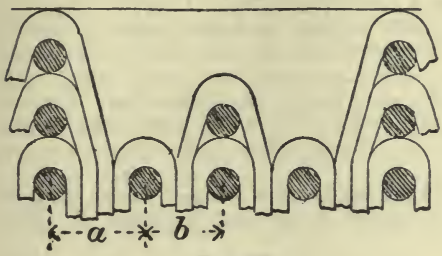


Fig. 191

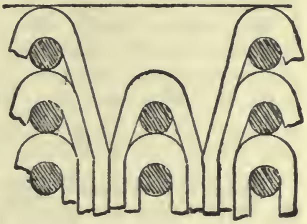


Fig. 192

to give results which are well supported by practice, and if oftener applied would lead to modifications in some of the complicated riveted joints. Thus in fig. 191 it will be noticed that the inner row of rivets is irregularly pitched. This is necessary, as otherwise the tapes from the outer row would have to be split in two and spread out at an angle.

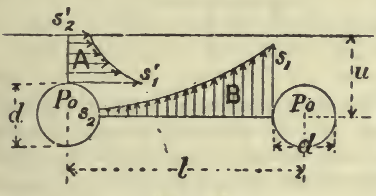


Fig. 193

Thirdly, in order to estimate the amount of metal required between the rivet hole and the edge of the plate, this part is looked upon as part of a continuous beam.

Fourthly, the circumferential shearing stresses are used as a basis.

Fifthly, a more thorough method is to deal with the metal surrounding the rivet as if it were part of a thick-walled cylinder.

Distribution of Stresses near Rivet Holes.—In fig. 193 the curves A and B represent the distribution of the circumferential stresses at these two points, due to the internal pressure p_o, which in this case is supposed to be uniformly distributed. In the formula (8, p. 208) for the curve A we have $r_1 = \frac{d}{2}$, $r_2 = u$, so that

$$S'_1 = -p_o \frac{4u^2 + d^2}{4u^2 - d^2}.$$

And for the curve B we have $r_1 = \frac{d}{2}$, $r_2 = l - \frac{d}{2}$.

$$S_1 = -p_o \cdot \frac{4 . l^2 - 4 . l . d + 2 . d^2}{4 . l^2 - 4 . l . d}, \qquad S_2 = -p_o \frac{2 . d^2}{4 . l^2 - 4 l . d}.$$

Of course, as the adjoining rivet also produces stresses, S_2 has to be added to S_1, and we have

$$S_q = S_2 + S_1 = -p_o \cdot \frac{l^2 - l . d + d^2}{l^2 - l . d}.$$

The following tables contain some numerical values of the ratios of $S_1{}^1 : -p_o$, and of $S_o : -p_o$, for different values of d, u, l:—

$u : d =$	·75	1	1·25	1·5	1·75	2
$S_1{}^1 : -p_o =$	2·6	1·67	1·38	1·25	1·18	1·13

$l : d =$	3	3·5	4	4·5	5	6	7	8
$S_o : -p_o =$	1·17	1·11	1·08	1·06	1·05	1·03	1·02	1·02

From this it would appear that the tension stresses at the sides of the rivet holes are slightly in excess of the rivet pressure p_o. Approximately

$$S_o = -p_o \cdot (1 + \frac{d}{5 \cdot l}).$$

Similarly, the stress $S^1{}_1$ may be expressed with sufficient accuracy by the formula

$$S^1{}_1 = 1{\cdot}25 \cdot p_o \sqrt{\frac{2 \cdot d}{2 \cdot u - d}}$$

As has already been shown, these circumferential tension stresses combine with the radial pressures p_o to form shearing stresses, acting along spiral lines (fig. 182, p. 209); and, as mild steel gives way more readily under this stress than under either simple tension or simple compression, it is only natural that when the plate, and not the rivet, gives way, the line of fracture should start where shown in fig. 194; also compare fig. 181, p. 208.

Fig. 194

It has been explained on p. 193 that, on account of the axial contraction of boiler shells, when subjected to internal pressure, the longitudinal seams are subjected to a compression stress. At the edges of the plates this will amount to about 30 % of the circumferential tension, and has to be subtracted from $S^1{}_1$, but it increases the spiral shearing stresses.

The above remarks apply only to the outer rows of rivets. In the second and third rows the conditions are far more complicated; but if, as is usual, the rivets are placed closer together, the stresses will be more uniform, and therefore relatively less severe.

Prof. E. G. Coker and Mr. W. A. Scoble ('N. A.,' 1913) have utilised the phenomena associated with the polarisation of light for analysing stresses in transparent objects. Manufacturers of high class lenses and prisms examine the slabs of glass from which they intend to make their objects in polarised light, in order to detect strains due to too rapid annealing. The colours which appear are due to angular displacements of the planes of polarisation. These angles are nearly proportional to the differences of the principal stresses to which the object is subjected, and to its thickness, and it is therefore an easy matter to estimate these differences of stresses $(x - y)$ by a comparison of colours. For mild steel, which nearly conforms to Guest's law, and breaks down when a certain difference of stress is reached, these simple observations would suffice, but if the individual stresses are to be estimated it is necessary to determine their sums $(x + y)$. Professors Coker and Scoble determine these sums by measuring the local contractions of thickness, which is about one-third of the elongations which are being produced in the direction of the stresses. Then x is half the sum of the two observations, and y is half their difference. These experiments can be carried out on sheets of glass, but it is a troublesome material on account of its liability to break, and therefore very transparent xylonite has been used. Their latest experiments were undertaken with a view to determining the stresses in riveted plates, which stresses have not been determined mathematically, though a formula has been constructed for stress in a plate with a hole in it which is subjected to a single stress. By their optical method they found that the maximum stress very near the hole was 2·37 times the mean stress of the plate, whereas the estimated stress for an infinitely wide plate is 2·38 times the mean. The theoretical maximum stress at the edge of the hole is three times the mean. This mean stress includes the hole which was one-quarter the width of the plate, so that the mean stress on the remaining 75°/ₒ of the plate is 1·33 of the above, and the maximum stress at the edge of the hole is theoretically 2·25 times as great. Similar tests were carried out by E. Preuss ('Deut. Ing.,' 1912, vol. 56, p. 1780) by measuring the elongations near holes which had been drilled into test pieces of various widths, and the following results were obtained. They also apply to large holes in shells.

Ratio of Width to Hole Diameter . .	8	4	2·4	1·71
Ratio of Maximum to Mean Stress . .	2·35	2·34	2·13	2·24

These results are in fair agreement with the above, and confirm the claim that this optical method of measuring stresses is reasonably correct. Messrs. Coker and Scoble therefore applied it to xylonite plates which were pulled by metal studs (glass studs would have been better) and measured stresses for widths of four diameters and three diameters which indicate that the edges of the holes are stressed about four times the average, including the holes, or about three times the mean stress in the remaining metal. According to the above formula based on the theory of thick-walled cylinders, the ratio of maximum to mean stress for $l = 4d$ is 3·24.

Plasticity of the Solid Plate.—When the stresses have grown so intense that the plastic limit has been reached they will rearrange themselves; but instead of being distributed more uniformly, it is probable that they will become more local, the maximum stresses being found round the circumferences of the holes. Testing a set of joints to destruction cannot, therefore, give a correct idea as to the proportions which should exist between the thickness of the plates, the rivet diameters, and the holes.

Distribution of Stresses amongst several Rows of Rivets.—In fig. 195 the slanting position of the rivets is intended to indicate that a treble riveted lap-joint is being strained. If, as is generally supposed, each rivet bears the same load, then every one will be inclined to the same angle; but then the stretch, and therefore also the stress of the plate *b* would be equal to that of *d*, and the same for *c* and *e*. But it is obvious that, whereas the full stress is to be found

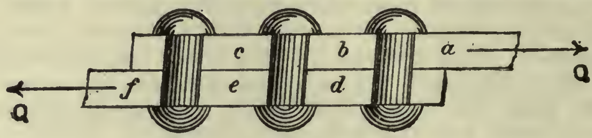


FIG. 195

at *a*, two-thirds will be found at *b* or at *e*, and only one-third at *c* and *d*, because in either case each rivet has only transmitted one-third of the load. These two distances have, therefore, only stretched half as much as *b* and *e*, and therefore the central rivet will not, as has been assumed, be inclined as much as the other two, and the shearing stress to which it is subjected will be less than the average, while for the two outer ones it will be more.

There are obvious reasons why it is easier to calculate the distribution of stresses in a butt joint than in a lap joint; but even this is a most laborious operation, and the results are not quite reliable. The following estimate was obtained with a joint whose dimensions were taken from practice, and which was intended to represent part of a double butt-strapped joint, having a percentage of 85·3. (See fig. 196.)

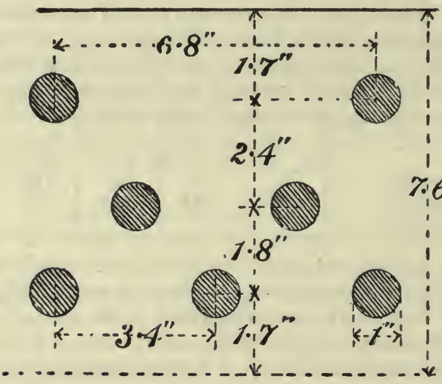


FIG. 196

Assuming the circumferential stress on the shell plate to be 5·05 tons per inch, each rivet is supposed to be subjected to a mean stress of 5 tons per sq. in., and the mean stress in the plate between the rivets would be 5·92 tons. However, it was found that the mean shearing stress in the outer rivets is 5·10 tons, in the intermediate ones 4·20 tons, and 5·75 in the inner ones. This joint, therefore, is only equal to 74·1 % of the solid plate, which is a little less than might be obtained by using the far more practical double-riveted joint, having rivets 1¼ in. diam. and a pitch of 4·9 ins.

This should be a sufficient justification for the practice of not crediting a joint with a greater strength than 75 % of the solid plate.

J. T. Milton ('N. A.,' 1885, vol. xxvi. p. 204) has recorded experiments on this subject which show that, even for the ultimate strength of a joint, considerable differences exist as regards the stresses in the various rows of rivets, and no doubt experiments on the elaborate joints to be found in bridges would show that their high percentage is imaginary.

Experiments on Riveted Joints.—A very exhaustive list of experiments on riveted joints will be found, W. C. Unwin, 'M. E.,' 1881, p. 303. These and a few subsequent ones are contained in the following table:—

W. Fairbairn	1850	'Phil. Transactions,' vol. ii. p. 677.
E. Clark	"	'Britannia and Conway Bridges,' vol. i. ch. iv.
D. K. Clark	1858	'Recent Practice.'
J. Grantham	1860	'N. A.,' vol. i. p. 57
D. Kirkaldy	1862	. . .
W. Fairbairn	1864	'Soc. Arts,' vol. xiii. p. 20.
N. Barnaby	1865–6	'Eng. Scot.,' vol. ix. p. 153.
J. Price	1869–70	" vol. xiii. p. 47.
J. Cochrane	1870	'C. E.,' vol. xxx. p. 265.
W. Waller	1871	'N. Eng. I.,' vol. xx. p. 117.
W. R. Browne	1872	'M. E.,' p. 53.
J. G. Wright	"	" pp. 77, 89.
Report	"	'N. Eng. I.,' vol. xxi. p. 67.
D. Kirkaldy	"	'Experiments on Riveted Joints.'
Sir W. Fairbairn	1873	'R. Soc. Edinburgh.'
B. B. Stoney	1875	'R. Ir. Ac.,' vol. xxv. p. 151.
J. Barba	"	Ch. iii.
J. Riley	1876	'N. A.,' vol. xvii. p. 135.
R. B. Longridge	1877	'Engr.,' vol. xliii. p. 125.
R. V. Knight	"	'C. E.,' vol. li. p. 131.
W. Boyd	1878	'M. E.,' p. 217.
B. Martell	"	'N. A.,' vol. xix. p. 12.
R. V. Knight	"	'C. E.,' vol. liv. p. 161.
Dr. H. Zimmermann	"	'Zeit. Bauk.,' vol. i. p. 530.
N. Barnaby	1879	'I. and S. I.,' p. 238.
D. Greig and M. Eyth	"	'M. E.,' p. 268.
R. B. Longridge	"	'On the Strength of Riveted Joints.'
R. V. J. Knight	1881	'M. E.,' p. 720.
Prof. A. B. W. Kennedy	"	" pp. 205, 232, 717.
" " "	1882	" pp. 143, 242.
W. Parker	"	'C. E.,' vol. lxix. p. 50.
C. H. Moberley	"	" vol. lxix. p. 337.
R. B. Longridge	1884	" vol. lxxx. p. 154.
Prof. A. B. W. Kennedy	1885	'M. E.,' pp. 198, 249.
J. G. Wildish	"	'N. A.,' vol. xxvi. p. 179.
J. T. Milton	"	" vol. xxvi. p. 204.
Prof. A. B. W Kennedy	"	'M. E.,' p. 538.
G. W. Manuel	1889	'N. A.,' vol. xxx. p. 292.

The above contain chiefly those experiments in which actual joints were torn. Related subjects are sometimes to be found in these papers, but it will be more convenient to look for them under the following headings:—

Drilled and Punched Plates.

H. Sharp	1868	'N. A.,' vol. ix. p. 10.
J. Cochrane	1872	'M. E.,' p. 97.
J. Riley	1876	'N. A.,' vol. xvii. p. 135.
B. Walker	,,	'M. E.,' p. 97.
A. C. Kirk	1877	'N. A.,' vol. xviii. p. 303.
B. Martell	1878	,, vol. xix. p. 1.
W. Parker	,,	,, vol. xix. p. 172.
W. Boyd	,,	'M. E.,' p. 222.
J. G. Muir	,,	,, p. 285.
N. Barnaby	1879	'I. and S. I.,' p. 45.
,,	1881	'M. E.,' p. 313.
E. Richards	1882	'I. and S. I.,' p. 43.
T. Wrightson	,,	,, p. 49.
W. Parker	,,	'C. E.,' vol. lxix. p. 50.
W. Beck-Gerhard	1884	'Gorni J.,' p. 347.
J. G. Wildish	1885	'N. A.,' vol. xxvi. p. 193.
P. D. Bennett	1886	'M. E.,' pp. 27, 44.
L. Tetmayer	,,	'C. E.,' vol. lxxxv. p. 421.
— Rudellof	1889	'Mitt. Berlin,' p. 97.

Shearing and Torsion.

D. Greig and Max Eyth	1879	'M. E.,' p. 268.
— Brock	1880	'N. A.,' vol. xxi. pp. 191, 204.
D. Greig and Max Eyth	1881	'M. E.,' p. 313.
E. Richards	1882	'I. and S. I.,' p. 11.
V. Appleby	1883	'C. E.,' vol. lxxiv. p. 268.
J. G. Wildish	1885	'N. A.,' vol. xxvi. p. 198.
Prof. W. A. B. Kennedy	,,	'M. E.,' p. 249.
J. Platt and R. F. Hayward	1887	'C. E.,' vol. xc. p. 382.
C. H. Carus-Wilson	1890	'Proceedings,' vol. xlvii. p. 363.

Friction of Joints.

D. Greig and Max Eyth	1879	'M. E.,' p. 268.
Clark Kaven and Lavelley	1881	,, p. 327.
,, ,, ,,	1897	'Deut. Ing.,' p. 739.

Diagonal Joints.

J. G. Wright	1872	'M. E.,' pp. 79, 90.
W. H. Shock	1874	P. 198.
'Enging.'	1887	'Numerous letters,' vols. xliii., xliv. 'Experiments,' vol. xliii. pp. 380, 428.

Theories about Riveted Joints.

J. H. Latham	1858	P. 1.
T. Baldwin	1866	'Soc. Eng.,' p. 150.
W. C. Unwin	1868	. . . .
C. Reilly	1870	'C. E.,' vol. xxix. p. 454.
W. R. Browne	1872	'M. E.,' p. 53.
,, ,,	,,	'Engr.,' vol. xxxiv. p. 362.
Report	,,	'N. Eng. I.,' vol. xxi. p. 67.
W. R. Browne	1873	. . . .
H. MacColl	1874–5	'Eng. Soc.,' vol. xviii. p. 111.
L. E. Fletcher	1876	'M. E.,' p. 64.
D. K. Clark	1878	Rules.
D. Adamson	,,	'I. and S. I.,' p. 392.
R. H. Twedell	1881	'M. E.,' p. 293.
G. Clouzel	,,	,, p. 167.
W. C. Unwin	,,	,, pp. 313, 332.

W. S. Hall	1885	.	'M. E.,' p. 231.
J. A. Rowe	1884–5	.	'N. E. C. I.,' p. 73.
J. T. Milton	1885	.	'N. A.,' vol. xxvi. p. 204.

Factor of Safety.—Having dealt with the peculiar behaviour of materials under various conditions, and also with some problems connected with the mechanics of a boiler, it is necessary to add a few remarks on the term factor of safety, which from an engineering point of view might be called the measure of our ignorance: it was adopted at a time when practically nothing was known about the strength of materials, the distribution of stresses, and their influences; but even now the ultimate strength of a structure is taken as a standard, and it is assumed that if the loads which produced rupture were reduced to, say, one-fifth, then the stresses would be similarly reduced; but that this is not the case has been repeatedly explained in the last few pages. For instance:—

1st. In a tensile test piece the ultimate stress is a compound, resembling a negative fluid pressure, and this stands in no relation to the axial tension which exists in the sample at only one-fifth the ultimate load. (See fig. 132, p. 161.)

2nd. The working stresses in a narrow beam are relatively 33 % severer than the ultimate stresses. A nominal factor of safety of 5 is in this case actually only $3\frac{1}{3}$. (See p. 192.)

3rd. The working and ultimate torsional strength of mild steel stand in a similar relation, and in this case the nominal and actual factors of safety stand in the relation of 5 to $3\frac{3}{4}$. (See fig. 123, p. 157.)

4th. The shearing stresses in rivets under working conditions and during rupture have just been explained, and in this case the actual factor of safety is very much smaller than the nominal. (See p. 157.)

As will be seen, a great deal depends upon whether the formulæ by which the stresses have been calculated are sufficiently comprehensive. If they are, and if the material is reliable, and the working conditions thoroughly well known, then there ought to be no reason for not reducing the factor of safety, for, from what has just been said, it is evident that some parts of a boiler are even now being worked with an actual factor of safety which is somewhat smaller than the real one, and there is no reason why other parts, where the nominal and actual factors are nearly equal, should not have the benefit, and might perhaps be made somewhat weaker than they are now.

It is also evident that when analysing the facts connected with the bursting, either experimental or accidental, of a boiler, it is very important to take into account the various deformations, for whereas elasticity of a structure is rightly looked upon as an extra guarantee of strength, plasticity throws the stresses into parts of a structure where they are least expected. To be thoroughly valuable, experiments on large structures should not only aim at obtaining information as to the weakest parts at the instant of rupture, but very careful measurements should be taken to determine the elastic deformations before that pressure is reached. (See p. 302.)

Barking Boiler Explosion.—Explosions amongst marine boilers are almost unknown: the case of one which exploded at Barking, January 6, 1899 (Board of Trade Report, No. 1173), due to excessive pressure, is therefore of special interest, and gives some indications as to permissible working stresses.

The boiler was built of iron in 1878 by Messrs. R. and W. Hawthorn, Leslie & Co. *Dimensions*: Diameter 10 ft., length 9 ft. 10 ins.; one dome; two furnaces 34 ins. diameter; two separate combustion chambers. *Thicknesses*: Shell $\frac{3}{4}$ in., joints welded; furnaces $\frac{7}{16}$ in. (single butt straps), reduced by corrosion to $\frac{5}{16}$ in. along firebars; combustion chambers $\frac{7}{16}$ in., reduced generally to $\frac{3}{8}$ in., except at bottom of back, where thicknesses of $\frac{5}{16}$ in. were general; these parts had also been patched. Screwed stays $1\frac{1}{4}$ in., riveted over ends except 21 in the back, which were new and nutted. Marginal stays $1\frac{3}{8}$ in. diameter, nutted. One screwed stay had a cover patch (see fig. 71). Pitches 8 ins. square, and also 8 ins. × 12 ins. Front end plates, top $\frac{11}{16}$ in., apparently not reduced. Stays $2\frac{1}{8}$ in. diameter, with plus threaded ends welded on; diameter reduced to about $1\frac{7}{8}$ in. Washers 6 in. diameter, $\frac{1}{2}$ in. thick; pitch 14 ins. square, and also 15 ins. × 12 ins. Tube plate $\frac{11}{16}$ in.; pitches of stay tubes $13\frac{1}{2}$ ins. × 9 ins. Back plate $\frac{5}{8}$ in. Maximum pitch of stays 8 ins. × 12 ins.

The accident was due to two successive pressure gauges being out of order, whereby the man who was setting the safety valves screwed them down to 240 lbs., at which pressure the boiler exploded.

The shell tore along the top, through the dome hole, at $8\frac{1}{2}$ tons per square inch, but not through any weld; it is, however, more than probable, there being two reports, that the first to rupture were the welds of the steam space stays, say at 7·6 tons per square inch; then the end plates tore, and then the shell. The furnaces do not appear to have collapsed. The stress in these plates was 4·2 tons at the thickest uncorroded parts, and 5·8 tons near the firebars. The tube plate does not appear to have bulged. Using the formula $\text{W.P.} = C\frac{t^2}{p^2}$, where the thickness is expressed in sixteenths of an inch, we get $C = 265$. The steam space plates were bulged, but it is not certain that this happened before the explosion. In this case we get $C = 450$. The combustion-chamber plates do not appear to have bulged, nor were many stays drawn out. Here we have $C = 310$ for the $\frac{7}{16}$ in. plates, and $C = 620$ for the $\frac{5}{16}$ in. plates. Many of the screwed stays were wasted.

The experience gained in this case is directly applicable to boiler strengths, except as regards the furnaces, for in this case the fires were only very light at the time of the explosion, and therefore not likely to produce stresses and deformations previously discussed. The questions to be decided are, therefore: what extra allowance can be made for the use of steel instead of welded iron, and what factor of safety is desirable? For here, of course, as the boiler was under working conditions—i.e. high pressure and high temperature—our uncertainty as to what happens when we calculate boiler strengths with the help of experimental data of cold steel falls away.

A very instructive case as regards strengths of boiler girders is the explosion (Board of Trade Report, 1900, No. 1254) of a stationary loco-boiler. The girders were 74 ins. long, 7 ins. deep, $1\frac{1}{8}$ in. thick, placed 5 ins. apart, and each one carrying 14 stays, pitched $4\frac{3}{4}$ ins., and supported, but at the wrong points, by slings. It was assumed that the explosion took place at a pressure of 140 lbs., but the boiler had recently been tested to 200 lbs. At this pressure, neglecting the influence of the slings, the stress was 33 tons per square inch, or if, as appears probable. the girders gave way when hot at a pressure of 140 lbs., then the stress under these conditions was 23 tons per square inch.

List of some Boiler Tests and Explosions.

R. H. Thurston, 1872, 'Frankl. Inst.,' iii. vol. lxiii.

p 89. Tested boiler to 82 lbs. cold, then burst at 90 lbs. hot.

p. 93. Stayed flat plates. Tested cold to 138 lbs., then burst at 165 lbs. hot.

p. 95. Tested box boiler to 60 lbs. cold. Vertical braces gave way; repaired these and tested hot. Reports were heard at 50 lbs. pressure; boiler burst at $53\frac{1}{2}$ lbs. pressure.

p. 99. Tested boiler to 200 lbs. at 100° F. Several braces broke under 115 lbs. steam pressure.

p. 99. U.S.A. steamer 'Algonquin' tested cold to 150 lbs. Some braces broke under 100 lbs. steam pressure.

R. H. Thurston, 1887, 'Explosions.' Tested boiler to 300 lbs. steam; burst at 235 lbs. on opening valve.

P. Carmichael, 'Eng. Scot.,' 1869–70, vol. xiii.; also 1878–9, vol. xxii. Experimental bursting of a boiler.

L. E. Fletcher, 1876, 'M. E.,' p. 59. Experimental bursting of a boiler (cold).

W. Siemens, 'M. E.,' 1878.

D. Greig and Max Eyth, 'M. E.,' 1879. Experimental bursting of three boiler shells.

W. Parker, 'N. A.,' 1881, vol. xxii. p. 12. 'Livadia's' boiler burst under cold water test above working pressure.

Ibid., 'N. A.,' 1885, vol. xxvi. p. 253. Shell boiler burst under hydraulic test.

Ibid., 'N. A.,' 1889, vol. xxx. p. 290. Experimental bursting of a boiler (cold).

J. Scott, 'N. A.,' 1889, vol. xxx. p. 285. Experimental bursting of a Navy boiler.

Parliamentary Committee on Boiler Explosions, 1817.

Report on boiler explosions. Rep. Comm. Parl., 1849.

Parliamentary Committee on Boiler Explosions, 1871. Parliamentary Reports No. 186, vol. lxvi. p. 43, No. 378, vol. lxvi. p. 85; 1877, No. 361, vol. lxviii. p. 373.

Martens, 'Rep. U.S.A.,' various dates.

'Board of Trade Reports on Boiler Explosions.' Amongst these latter the following are of interest:—

1861, part iv. Locomotive boiler tested to 196 lbs. at 162° F.; burst seven months later under 120 lbs.

'Acrefair,' Dec. 10, 1880. Boiler locally weakened to 61 lbs. permissible working pressure; burst at 32 lbs.

No. 228. Drying cylinder burst. Factor of safety about 20·3.

No. 229. Lancashire boiler shell burst. Factor of safety was about 2·8.

No. 237. Locomotive boiler burst.

No. 243. Boiler shell burst through manhole at 100 lbs., after having been recently tested to 150 lbs. steam.

No. 249. Boiler shell exploded. Factor of safety 2·75.

No. 252. Boiler shell exploded. Factor of safety 11.

No. 265. Boiler shell exploded at 83 lbs. It had recently been tested cold to 92 lbs.

No. 314. Boiler shell exploded. Factor of safety 3·5.

No. 346. Furnace collapsed at 50 lbs. It had been tested to 95 lbs. only three days previously.

Other interesting cases are—Explosion on the steamer 'Mülheim No. 5,' 1866; explosion on the steamer 'Parana,' 1869; explosion on the steamer

'America,' 1871; explosion of a locomotive boiler, 'Engineering,' 1890, vol. l. p. 332.

T. W. Trail ('C. E.,' 1884, vol. li. p. 31) mentions the bursting of two boilers whose factors of safety were 3·9.

L. E. Fletcher ('C. E.,' 1884, vol. lxxx. pp. 136 and 139) mentions several curious explosions.

J. J. Platt, 'M. E.,' 1878, p. 260. A steel fire-box plate cracked after being in use five months.

MacFarlane Gray ('N. A.,' 1877, vol. xviii. p. 326) mentions that a boiler intended for a working pressure of 30 lbs. was tested to 400, repaired, and then worked all right.

J. A. Rowe, 1884, p. 7. A boiler burst at 30 lbs. working pressure after a cold water test to 59 lbs.

See list, p. 136.

CHAPTER VIII

BOILER CONSTRUCTION

IN the following pages the various workshop practices of boiler construction will be dealt with as briefly as the subject will allow. The question of cost cannot, of course, be entered upon, but occasional reference will be made to the time required for the various operations; this necessarily varies in different works, depending not only on the perfection of the machinery, but also on the skill and energy of the men. More attention has been paid to the different practices and tools for obtaining the same object, and occasionally methods have been mentioned which are practically obsolete or not in use, but which may have done good service or must be looked upon as warning examples. Naturally, every modern device has not been discussed, but it is hoped that none except unimportant ones have been neglected.

The order in which boiler construction will be taken is to deal firstly with the various operations to be performed on the boiler shell, then with the internal parts, and then with the boiler as a whole.

The plates as they arrive from the rolling mills are never exactly of the specified sizes, being usually about $\frac{1}{2}$ in. larger in each dimension. The thickness is kept within reasonable limits by the condition that any excess weight beyond, say, 5% margin will not be paid for; but difficulties are sometimes occasioned by stipulations that certain weights shall not be exceeded which, when compared with the plate thicknesses, are too low. The weight of 1 sq. foot of iron is equal to about 40 lbs. per inch of thickness, and 41 lbs. for steel.

The edges of plates are nearly always thinner than the centres. This is almost unavoidable, for as soon as the rolls are set so that the conditions are reversed the plates are seriously puckered. It is not possible to give a full explanation without entering into unnecessary details, but there is no difficulty in understanding that if during the last pass through the rolls the thickness of a plate is reduced $\frac{1}{32}$ in., while its thickness is $\frac{1}{2}$ in. at the edges and $\frac{7}{16}$ at the centre, then these parts will stretch relatively $\frac{1}{16}$ and $\frac{1}{14}$ of their length, and the centre must pucker. Should the rolls have been made quite parallel, then as soon as they get heated their centres swell and produce the above result. If they are made slightly hollow, the plates which are rolled first will be thin at the edges, and perhaps frilled; but the greatest trouble is experienced when the widths of the plates to be rolled vary much, for an enormous pressure (about 500 to 1,000 tons, has to be exerted, and the spring in the rolls is very appreciable.

Shearing Operations.—In a boiler shop large shearing machines are not required, but certain plates, such as those for the boiler ends and for the combustion chambers, are often ordered with a comparatively large margin, on account of flanging, and this excess is then most easily removed by shearing. A small but powerful machine, capable of cutting steel plates up to $1\frac{1}{4}$ in. thick, will be found a very handy tool. The levers and handles for working it should not project in front; otherwise they may come in contact with such of the plates whose flanges have to be sheared while standing on end. The arrangements for replacing the shearing blades should be such that this can be done quickly, so that, as occasion arises, curved or cornered shears may be substituted for straight ones.

The question as to how much material ought to be planed off sheared edges in order to remove the injurious effect is, and probably will remain, an unsettled one. Some engineers look on shearing and punching as being perfectly harmless; others insist on subsequently planing away at least $\frac{1}{4}$ in. Probably the experiences which led to these diverging views are due to the use of various qualities of material. Remarks on this subject will be found in the chapter on 'Strength of Materials,' and the conclusions arrived at there are that good material is not injured, while bad material grows brittle, and the more so the thicker the plates are. Evidence will also be found there in support of the view that the brittleness caused by shearing gradually extends into the plates. Under any circumstances it is well to guard against possible failings of this sort by insisting on a cold bending test of samples with sheared edges, or of samples with punched holes. In this latter case a standard punch and bolster should be determined upon, as their relative diameters influence the results. If a sample which has been punched or sheared bends well when cold, and particularly if it does so after having been put aside for a week or more, then there will be little fear that the material can satisfactorily withstand this and the less severe workshop treatment.

Punching Operations.—The effects of punching are so very similar to those produced by shearing that nothing need here be said about them, except, perhaps, that it would be better if no punching machines were used in a boiler shop, so that all holes could only be drilled. However that is rarely, if ever, the case, and therefore it is as well to fit as strong and accurate a machine as can be obtained, and to insist on carefulness in setting the bolster. If carelessly placed the punched hole will be a very irregular one (see fig. 197). This is particularly the case if the guides of the punching press slide are a loose fit, for then it will happen that one hole is quite fair, while the next has a slanting form, or even a worse one (as in fig. 198), due to the punch having started penetrating while its edge was actually overlapping the circumference of the die. The breaking of punches under such conditions is not to be wondered at.

The clearance, or rather the difference, between the diameters of the punch and of the die hole is usually about 10 to 15%. This agrees fairly well with the following published recommendations:—

W. H. Shock (1880, p. 166) gives 15 to 20%. J. H. Wicksteed

('M. E.,' 1878, p. 244) recommends one-sixth of the thickness of plate. This amounts to 11% where the diameters of the holes are half as large again as the thickness of the plate, and 17 % when they are equal.

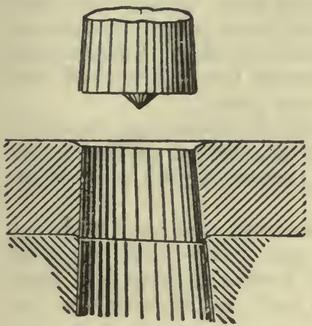


Fig. 197

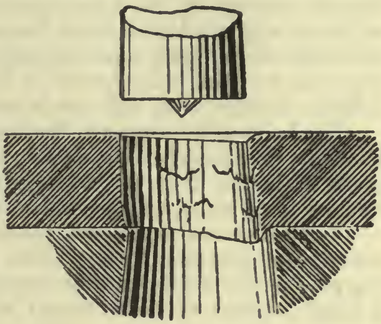


Fig. 198

Spiral and Slanting Punches have been used not only to lessen the injury to the plate, but also to reduce the pressure on the machine. W. Barr (1880, p. 92) mentions that Kennedy's spiral punch (fig. 199) requires only one-third the power of an ordinary one. Stern ('M. E.,' 1878, p. 239) gives the following information:—

—	Diam. of Hole	Punching Force	Tenacity of Punched Plate
Ordinary punch . . .	$\frac{7}{8}$ in.	33 to 35 tons	26 tons
Spiral " . . .	$\frac{7}{8}$ "	22 " 25 "	$28\frac{1}{2}$ "

L. Hill ('M. E.,' 1878, p. 244) mentions slanting punches (figs. 200, 201).

It is only natural that, in comparison with ordinary punches, less force, though possibly just as much power, is required with spiral or slanting punches, because with them only a small part of the circumference is doing work, just in the same way as less force is

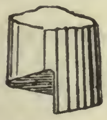


Fig. 199

Fig. 200

Fig. 201

required in a shearing machine if the cutting blade is set at a steep angle. When large holes have to be punched, there ought to be no hesitation in substituting one of the above punches for the ordinary flat one. Fig. 200 seems to be the best suited for hand holes, &c., not only because of its being symmetrical, but because the greater part of the edge is shaped like a good cutting tool. The reverse or V shape would in this respect be the worst possible. The spiral punch leaves a mark in the circumference, which in large holes may be an inconvenience.

A favourite plan in some works is to punch the holes in all those plates which for some reason, such as flanging, have to be annealed, but, on account of the blindness of some of these holes when fitted together, a great deal of hand labour has to be expended on them in the way of chipping and rimering, so that if added together the expenses for such holes would be found greater than if they had been drilled; besides, they will not be satisfactory jobs.

The holes of the seams in the furnaces are also sometimes punched; but here, particularly with the circumferential seams at the front ends, there is danger that the drawing of the furnace mouth, to meet the flanged front plate, will produce cracks.

Various seams in the flat plates of the front and back ends are sometimes punched, even when they are $\frac{7}{8}$ in. thick; but the warping of the plates and the certainty of having to chip and rimer a large number of holes ought to be a sufficient objection to this practice.

The lower edges of the back tube plates are almost invariably punched, leaving them quite ragged. In some works they are left in this condition, as it is useless to caulk this edge. For appearance sake, most boiler-makers chip it. Under any circumstances it would be best to leave a good bevel, so as to prevent steam lodging there and causing the saddle seam to heat. By using a square or rectangular punch such edges would be left in a better condition for chipping.

Punching is also resorted to for producing holes for guiding the trepanning tools with which the tube plates are bored. This is a bad practice, and leads to irregularities amounting to $\frac{1}{8}$ in. in the various diameters of the finished holes, and even affects their roundness.

A less objectionable though still unsatisfactory use of the punch is the practice of perforating those points of the back end plates and of the combustion chamber plates where the screwed stays are to be fitted. When placed in position drills and then taps are passed through them, which remove all brittleness, even if the plates have not been annealed after punching. (For injury done by punching see p. 220.)

Illustrations of various types of shearing and punching machines will be found in the following publications:—

'Engineering,' vol. xxxix. p. 219, 'Hydraulic Shearing Machine'; vol. xlii. p. 221, 'Portable Pneumatic Punching Machine'; vol. xliv. p. 16, 'Shearing Machine'; vol. l. p. 688, 'Punching and Shearing Machine'; vol. l. pp. 177–179, 243, 247, 494, 519, 'Punching Machines.'

Planing Operations.—Illustrations of plate-edge planing machines will be found in 'Engineering,' vol. xxxvi. p. 384; vol. xlix. pp. 245, 252; vol. l. pp. 536, 625. All the shell plates have to be planed at their edges. They are bolted down on the planing machine (fig. 202), and a tool in the slide rest R_1 travels along its edge, at a speed of about 10 to 18 ft. per minute, mean speed 12 ft. with a feed of $\frac{1}{32}$ in. for 1-in. plates, and proportionately more or less for thicker or thinner plates, removing the superfluous material at the rate of 5 to 6 cubic inches per minute. To this has to be added the time for setting the plate. The total time required to set and plane four shell plates measuring 20 ft. × $5\frac{1}{2}$ ft. × $1\frac{1}{8}$ in., equal to 102 ft. running, amounted to about 12 hours, from which it is clear that the proper setting of the plates, which had to be repeated eight times, takes up very much more time than that required for the removal of the material. The machine used had two sets of frames, placed at right angles (see fig. 202).

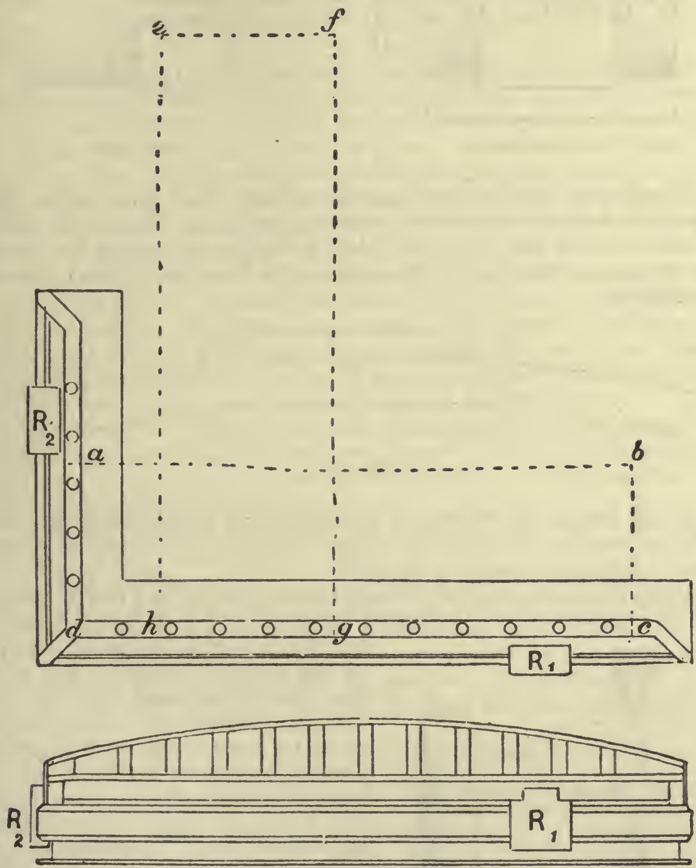


FIG. 202

With the old-fashioned machines, in which the plate has to be reset for each edge, the time required would have been twice as long, and much more space would have been required, for the plate, which is shown by the dotted lines *a, b, c, d,* would have to be turned round to the position *e, f, g, h.*

A little saving in time is effected if the planing tool be so fitted that it can be turned round, and cut both during the forward and backward travel (see fig. 203). But when the edges have to be bevelled the tool must have two cutting faces (see fig. 204).

The amount of bevel of edges to be caulked varies from nothing to 1 : 3 (see p. 300), but for the butts of shell plates it should be about 1 : 30; otherwise they will not close properly (see fig. 205).

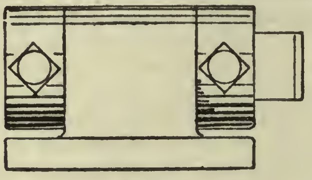


Fig. 203

Fig. 204

Should the planing machine be shorter than the plates, their edges will have to be planed in two operations. The usual plan is to withdraw the tool gradually when it reaches the end of the stroke, but in some works part of the material at this point is first removed by chipping.

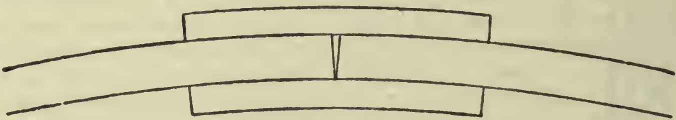


Fig. 205

In old machines, where the holding-down frame is secured at its ends, as shown in fig. 206, only definite lengths of plates can be planed, unless a bracket is bolted to the bed.

Large plates are generally ordered with a margin of $\frac{1}{4}$ inch all round, and this is occasionally exceeded by another $\frac{1}{4}$ inch when shearing. All this material has to be removed on the planing machine, and great care has to be taken that the final dimensions are correct. They should never be marked off with anything else but a steel or iron rule, for the differences of expansion of wood, iron, and brass are very appreciable on large dimensions. The working drawings should always contain the widths and lengths of the plates, though the latter dimension is generally omitted. To find the exact circumferential length of a shell plate multiply the *mean* diameter of the particular strake by 3·1416.[1]

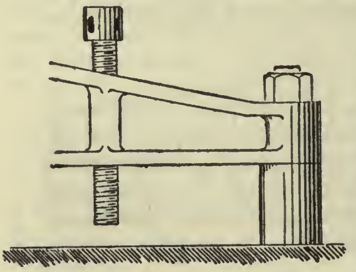


Fig. 206

This length has to be divided by two, three, or four, according to the number of joints in the shell. The length of the adjoining strake

[1] A rough approximation to this value is $\frac{22}{7}$; where very great accuracy is desired the fraction $\frac{355}{113}$ may be used.

may be found in the same way, but should be checked by adding or subtracting 6·283 times the mean thickness of the two plates as measured at their overlapping edges. It is necessary to be accurate on this point, otherwise the plates will not butt properly. In lap-jointed boilers one width of the lap has to be added to the length of each plate.

The furnace and combustion-chamber side plates are also planed on these machines, and their dimensions are either marked on the drawings or have to be measured from them. The lengths should be found by measuring the circumferences of the flanged plates and furnaces when fitted together.

The flat edges of the front and back end plates are also planed in these machines, but only after having been flanged and fitted together.

Drilling Operations.—In all first-class works every hole in a boiler is drilled, and generally drilled in place. A considerable amount of ingenuity has therefore been expended in designing machines which will do this work with speed and accuracy, and which can be adapted to various uses. Naturally a great many different patterns are in use, as will be seen from the following list:—

W. S. Hall, 'M. E.,' 1878, p. 565. Drilling machinery. He mentions Hutchinson's, Welch's, Buckton and Wicksteed's, and Buckton's multiple drilling machines; Adamson's, Dickinson's, Jordan's, and Kennedy's drilling machines; Hall's portable, Brown's and Thorn's steam drilling machines; McKay's equilibrium drill and other furnace and manhole boring tools, and also Shaw's flexible shaft.

W. Arrol ('M. E.,' 1887, p. 312) describes the drilling machinery used in the construction of the Forth Bridge. This paper is well worth studying, though not intended for boiler work. In 'Engineering' will be found sketches of the following:—

Ordinary drilling machines, vol. xxxiii. p. 348; vol. xxxv. p. 99; vol. xlvi. p. 327; vol. xlviii. p. 191.

Radial drilling machines, vol. xxxiii. p. 134; vol. xxxviii. p. 388; vol. xxxix. pp. 57, 360; vol. xl. p. 246; vol. xli. p. 29; vol. xliii. p. 269; vol. xliv. p. 42; vol. xlvi. p. 468; vol. xlviii. p. 180; vol. xlix. p. 248; vol. l. p. 184; vol. lxxxv. p. 46; vol. lxxxvi. p. 540; vol. lxxxviii. p. 393.

Multiple drilling machines, vol. xxxiv. p. 373; vol. xliii. p. 69; vol. xliv. pp. 150, 289, 292; vol. xlv. p. 451; vol. xlvi. p. 90; vol. xlix. pp. 250, 251.

Shell and back end drilling machines, vol. xxxiii. p. 586; vol. xl. p. 424; vol. xli. p. 620; vol. xlii. p. 420; vol. xlviii. p. 501; vol. lv. p. 318; vol. lviii. p. 780; vol. lxiv. p. 751.

Portable drilling machines (for furnaces), vol. xxxii. p. 162; vol. xxxix. p. 570; vol. xlii. p. 637; (for shells) vol. xl. p. 320; (hydraulic) vol. xliii. pp. 130, 131; vol. xliv. p. 295; vol. xlvii. p. 644; (hand) vol. xlv. p. 185. Ring-shaped shell drilling machine (Forth Bridge), vol. xxxix. p. 57; vol. xlix. p. 248.

Combined multiple and radial drill, vol. xlii. p. 613.

Drilling and tapping machine, vol. l. p. 184; xciv. p. 82.

Drill grinders, vol. xl. p. 320; vol. xliii. p. 101; vol. xliv. p. 6; vol. l. p. 674.

These numerous references make it needless to discuss the distinguishing features of the various types, and attention will only be drawn to a few points.

The Ordinary Drilling Machine is used for little else but small or occasional jobs. Its table can be raised or lowered, and can generally be turned round a vertical axis. The feeding is done by hand. In shipyards, where these machines are used for countersinking holes, the feeding arrangement is a balance lever.

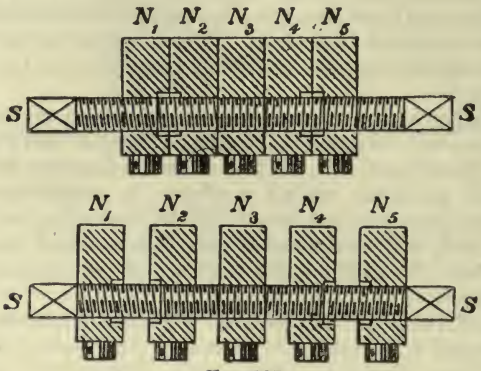


Fig. 207

Radial Drilling Machines are made in various forms. Sometimes they are fixed against walls, and are driven by belting from above, or they stand alone and are driven from underground. In that case they are generally made so that their arm can sweep through 360°. In some cases the table is adjustable, in others the arm can be raised or

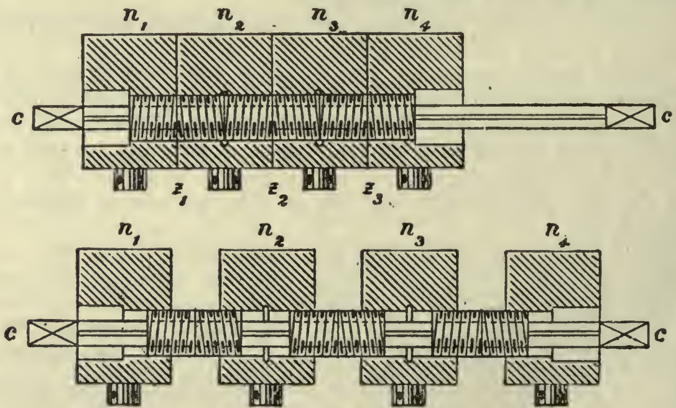


Fig. 208

lowered. Arrangements for doing this by power are a great convenience, and all moving parts should be balanced. It is well to make the balance weights extra heavy, so as to reduce the slack of the drill to a minimum.

Multiple Drilling Machines are passing out of favour in boiler shops, because their chief occupation, drilling shell plates before bending, is gone. They are generally arranged to slide along a frame, and each spindle can be shifted and worked independently of the others, or they can be set to the proper pitch and moved along together. In some machines the spindles can also be moved at right angles to the frame, so that a double row of holes can be drilled without resetting the plate. In these machines the various spindles cannot of course be brought close together, which causes a slight waste of time at the two ends of a seam, but is otherwise an advantage. In another type of machine the spindles are placed very close together. Figs. 207, 208 show two arrangements for adjusting the pitch in these cases. A turn of the screw S (fig. 207) will separate the nuts N_1, N_2, N_4, N_5 from the centre one N_3, and they are then clamped. Or a turn of the spindle C (fig. 208) will turn the hollow right- and left-handed screws z_1, z_2, z_3, which connect the four nuts n_1, n_2, n_3, n_4, and these are then clamped. The drill spindles are attached to these nuts and move with them. Should one of the drills break, or be put out of use, all the others would of course have to be stopped for a time.

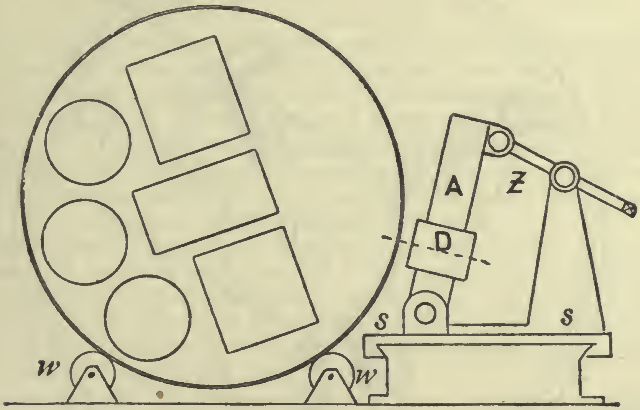


FIG. 209

Shell Plate Drilling Machines usually consist of two vertical columns, of which one or both are movable. One, two, or even three drill spindles are attached to a slide, which can be moved up or down these columns into any required position. If shells are to be drilled they are placed on a turntable between the two columns (sometimes there are three or four). Much time is wasted over the longitudinal seams, for usually only one can be attacked at a time, and in this respect it is a great advantage if each column can be moved both in a radial and in a circumferential direction, even if only through a limited range. Then the angular adjustment need not be done by the turntable.

Another shell-drilling machine is shown in fig. 209, and more fully illustrated, 'Enging.,' 1912, vol. xciv. p. 51. The boiler shell rests on four wheels, W, W. A slide rest S is movable along the bed plate, and carries an arm A, which can be set to any angle by means of the screw z. It carries the drilling spindle D, which can be moved up or down. It is clear that the drills—for there are usually two—can, within certain limits, be set to any required angle or position. When all the holes in a given span are drilled, the boiler is turned round the necessary angle and a fresh start made.

Boiler Back End Drilling Machines usually consist of two columns, both movable along a strong bed plate; the drill spindles can, as in one of the previous cases, be raised or lowered to any convenient height, so that every part of the back end plate of a boiler could be drilled if placed in the proper position. These machines are sometimes fitted with the necessary gear for tapping holes and screwing in the screw stays, and in some cases circular saws can be attached, with which to cut off the projecting ends.

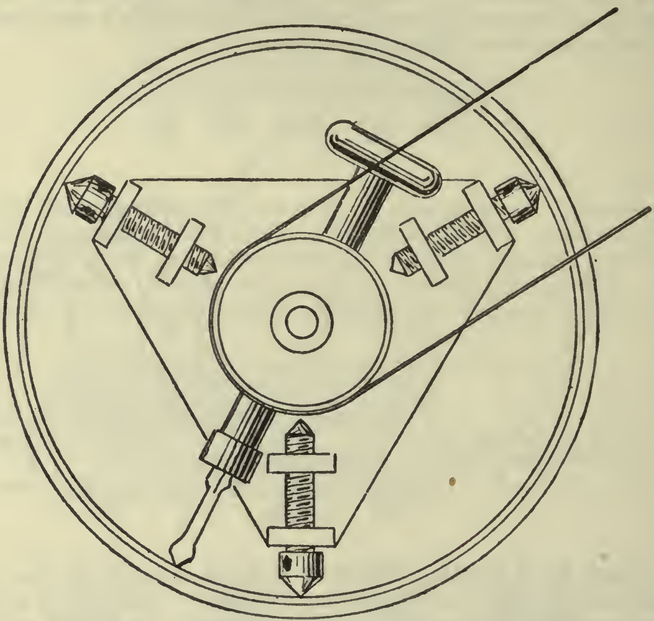


Fig. 210

A very convenient machine for drilling furnaces in place is shown (fig. 210). A frame with three or four set screws is firmly screwed into the mouth of the furnace; a light drilling machine is attached to the centre of this frame in such a manner that it can be clamped in any radial position. The driving wheel projects beyond the mouth of the furnace, and is driven by a belt or gut rope. Of late, electric motors have been applied to this purpose. A somewhat similar machine, but fixed, is sometimes used for drilling the furnace fronts out of place after the furnaces have been fitted and the holes marked off.

Any one of the more powerful drilling machines can be used for boring the tube plate holes. A small hole is first drilled or punched in the centre. The guiding spindle S (fig. 211) of a special tool holder, which is secured to the drilling machine, is inserted into one of them, and the cutting of the circumference of the large hole is then done by the parting tool T while it revolves. P is the tube plate which is to be bored. Unless the spindle S is a very good fit in its guide hole, the tube holes are very apt to be irregular both as regards shape and size, and as the setting of the tool T is a somewhat tedious job, it is not to be wondered at that every time it has to be done the diameter of the hole changes. This can easily be demonstrated by measuring several holes, which will be found to differ sometimes by as much as $\frac{1}{16}$ in. even in the same plate. No wonder, then, that tubes which are somewhat less in diameter than the smallest holes should split while being expanded into the larger ones. In some machines S is replaced by a sliding centre which rests in a large centre punch mark and guides the cutter. In some works the tube holes are rimered out after boring, which is an excellent practice. Less commendable is the system of tapping the stay tube holes before the plates are riveted up; some of the threads are sure to be nearly cut away.

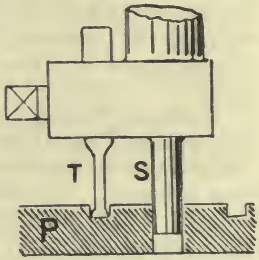


FIG. 211

Boring Tools.—Other contrivances related to drilling machines are furnace and manhole boring tools. If intended only for the former of these purposes, they are practically nothing but very powerful drilling machines; only, instead of a drill, an arm with a slide rest is attached to the vertical spindle. A parting tool is secured to this rest, as shown in fig. 212, and the furnace front plate bolted in the desired position on the bed plate and bored out. Most of these machines are now arranged in such a manner that the vertical spindle can be made to travel horizontally backwards and forwards, whereby the hole cut into the plate will be an elliptic one, as required for manholes.

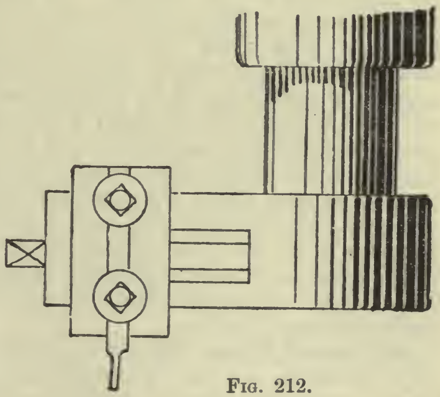


FIG. 212.

As regards the drills themselves something may be learnt from W. S. Hall's paper ('M. E.,' 1878, p. 565), as well as from W. F. Smith's (ibid. 1883, p. 56), who discusses the cutting angles of tools and drills and speeds. The oxy-acetylene blowpipe is now used for cutting out large holes.

Time required for Drilling.—To drill a hole 1 in. deep per minute, and to spend about half a minute for setting, seems to be a fair allowance for holes of from $\frac{3}{4}$ in. to $1\frac{1}{4}$ in. diameter. W. F. Smith gives a circumferential drill speed of 20 ft. per minute with $\frac{1}{100}$ in. feed, while J. H. Wicksteed, in the discussion which followed, recommends 40 ft. and $\frac{1}{200}$ in., so that as regards speed of feed the result is the same in both cases. For 1-in. holes this means a drilling speed of about $1\frac{1}{2}$ in. per minute, which is a very high value.

Dempster Smith, 'M. E.,' 1909, vol. i. p. 315 deals very fully with the question of speed, feed, thrust, and power. He had the advantage of exceeding the speeds which were customary before high-speed steels were invented. He looks on 60 ft. per minute as a safe circumferential speed for drills, and finds that the drilling power for a given speed and feed increases as $d^{0\cdot8}$, and the horse-power per cubic inch of metal removed is proportional to $d^{0\cdot2}\ t^{0\cdot3}$ where d is the diameter of drill and t the feed per minute both measured in inches, which means that this power is almost independent of the diameter and the feed, as it increases very slightly when either are increased. For steel the feed is about $d^{\frac{1}{3}} : 100$ ins. per revolution, and as these are about $230 : d$, the feed per minute is about $2\cdot3d^{\frac{2}{3}}$ ins. per minute. Thus a 1-in. drill should pass through 1 in. of metal in 24 seconds, and the times for $\frac{1}{2}$-in. and $1\frac{1}{2}$-in. drills are respectively 38 and 18 seconds. Of course these details refer to the materials used in these experiments.

Hand drilling is much slower, being at the rate of about 5 to 6 ins. depth of 1-in. holes per hour, and about 3 minutes for setting. Smaller holes can be drilled more quickly, and, generally speaking, the labour is proportional to the weight of metal removed. The limiting conditions for automatic feed are the strength of the drill and the clearance for the borings. In both respects the twist drill is superior to the ordinary one, but very few boiler shops have retained it in use. It is stated that after a time these drills wear away near their ends, and grow taper, and get jammed in the holes and break. Another and probably more powerful reason is that sufficient care is not taken to run them perfectly true, nor to guide them as required, nor to sharpen them correctly; and if all or nearly all the work is thrown on one cutting edge, it cannot be expected that the result will be a satisfactory one. The irregular action which takes place with ordinary drills if one cutting edge is longer than the other is sometimes made use of to produce taper holes. The deeper the drill penetrates when in this condition, the larger grows the diameter of the hole. This is of course impossible with twist drills, as they are guided by the hole they make.

Marking off Holes.—Should it be necessary to drill shell plates before bending, it is very important to make accurate templets both for the inside and outside strakes, comprising about thirty or forty holes, and the exact positions of several holes should be carefully calculated as a check on the templets. In works where this plan is adopted the necessary appliances for drilling in place are doubtless wanting, and then the holes in the end plate flanges and in the longitudinal seams will have to be drilled by hand, the outer machine-drilled holes acting as guides. The latter should never be drilled before bending the plates, as they are thereby seriously weakened at points where strength is of the utmost importance.

Great care should always be taken that these holes cover each other perfectly, for if blind, the sectional area of the rivet in the plane of shear is reduced, or if this is put right by chipping and rimering, the section of the plate round the hole is reduced.

Another practice is to punch or even to drill the holes of a smaller diameter than required, and to drill away the superfluous material when fitted together; but if the view be true that the injury to the material extends slowly for a considerable distance, then this practice ought to be condemned.

The marking off of back plates for boilers and combustion chambers can be done from the drawings, but often the combustion chamber plates after being drilled are laid on the boiler plates for marking off. In either case the pitch in the boiler plate should be slightly greater than the other, and also higher, so as to allow of the stays being on the slant.

Bending Operations.—Since the failure of the steel shell plate of the steam yacht 'Livadia' there exists a very justifiable dread of bending such plates while hot, but as long as this operation is not carried out at a blue heat the plates ought to suffer no permanent injury, and where the rolls are not sufficiently strong to bend cold plates they will have to be heated. It must not be forgotten that the 'Livadia' case is not the only one in which the shell plates cracked, and that several instances are known where this happened with plates that had been bent cold.

A very strong objection against bending shell plates while hot is the necessity of being possessed of a very long heating furnace, the extra time required for warming and then for cooling the plates, and the difficulty of obtaining a uniform temperature, which leads to irregular curvatures. These, and not the supposed injury done to the shell, are probably the most potent reasons which have induced manufacturers to adopt the plan of bending plates cold.

The Bending Rolls have necessarily to be proportionately stronger, and the following few notes will be a guide in the matter.

The resistance to bending beyond the limit of elasticity is independent of the curvature, and is approximately equal to $4 . t^2 . b$ for iron and $5 . t^2 . b$ for steel. Here t is the thickness and b the breadth of the plate, measured in inches. If the plate is heated to redness the coefficients 4 and 5 are reduced to about $\frac{1}{2}$ (see p. 190).

As the strength of the bending rolls is proportional to the cube of their diameters, and inversely proportional to the square of their length, and as the bending moment exerted by the rolls on the plate is, in the case of three rolls, of which the two smaller ones nearly touch (see fig. 215), proportional to the diameter, we find that $D^2 = c . L . t$ (see p. 169).

Here D is the diameter and L the length of the rolls, and c a constant which is equal to about $4\frac{1}{2}$ for the lower or outer ones of a system of three rolls, and the upper or inner one should be 25% larger than these, as it supports double the load.

For red-hot plates the constants may be reduced to 1·5, which would allow of the rolls being reduced to about half the above diameters.

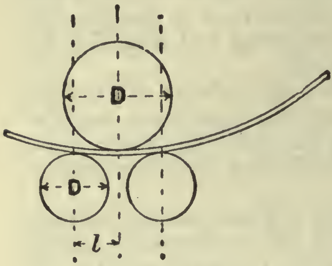


Fig. 215

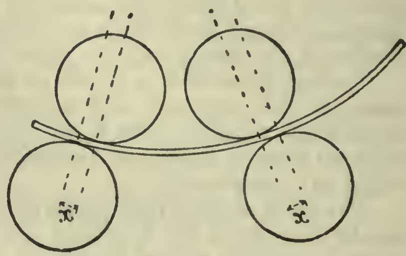


Fig. 217

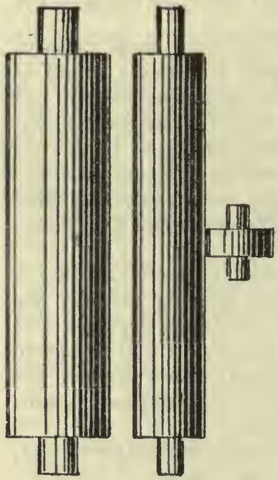


Fig. 216

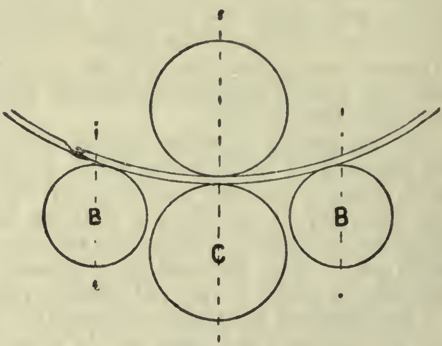


Fig. 218

The smaller the diameters are made, the shorter will be the unbent end pieces of plate, and, as this is a very desirable object, various devices are in use for attaining it.

The two outer (or lower) rolls are sometimes supported by anti-friction rollers, as shown in fig. 216. Or instead of three rolls four are used (fig. 217). The leverage x of the bending forces of the rolls can be very much reduced by this means, but on account of the greater pressure the diameters have to be proportionately increased.

Another plan is to have the four rolls arranged as shown in fig. 218, but the advantages are not apparent, and it would even seem that by removing the central roll C and bringing the two rolls B closer together

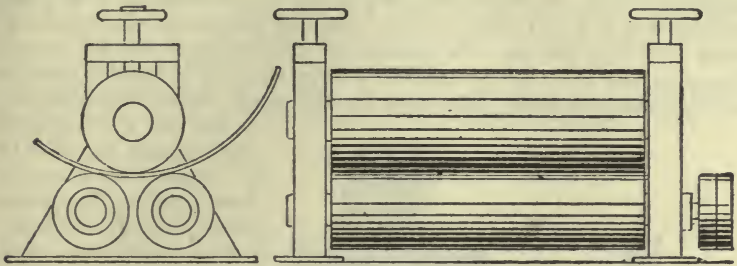


FIG. 219

the bending could be done better. In some machines the upper roll can be moved horizontally, but this also demands that the diameters should be large.

The bending rolls may be placed either horizontally (fig. 219) or vertically (fig. 220). The latter plan is certainly the most convenient, and is being generally adopted. The upper framework is shown in plan, and is so arranged that the inner roll can be lifted out, in order that shell or furnace plates may be rolled in one piece. Of the horizontal rolls it is usually only the two lower ones which are driven, while with the vertical rolls all three turn together. They thereby acquire a better grip of the plate, but even in that case it is advantageous to cut a few grooves into the driving rolls, as they materially assist in dragging in the plate.

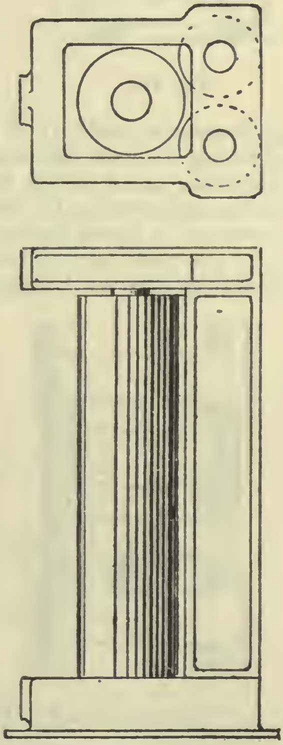


FIG. 220

Sketches of various types of bending rolls will be found in the following volumes of 'Engineering':—Horizontal plate bending rolls: vol. xxxiii. p. 134; vol. xlix. p. 529; vol. l. pp. 327, 480, 688. Vertical rolls: vol. xxxii. p. 135; vol. xliv. p. 258. Plate straightening machines with five rolls: vol. xl. pp. 9, 321, 619; vol. xlv. p. 135; vol. l. pp. 276, 606. Bending presses: vol. xliii. p. 491; vol. xlix. p. 245; vol. lviii. p. 477.

In some works the plates are passed several times through the bending rolls while these are being gradually screwed closer together. When possible, and particularly if the plates are hot, the curving should be carried out in one pass, for, independently of the disadvantages of punishing the material repeatedly, it will be found that less force is required for a single bending than for several, if that is necessary. This, however, is only possible if the machine is in good working order, and if full reliance may be placed on the marks to which the rolls are set. On account of the spring of the rolls some allowance has to be made, according as to whether very wide or very narrow plates are being bent, and for the same reason the influence of the extra stiffness of thick plates has also to be taken into account. After bending, the plates uncurl slightly, but absolute accuracy need not be aimed at.

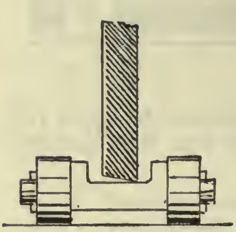


Fig. 221

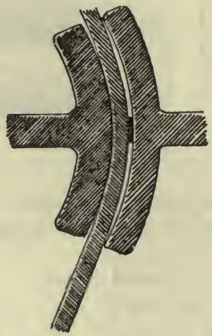


Fig. 222

With vertical rolls the shop floor should be square to their axis, and instead of using round iron rolling rods to support the plates, small carriages (fig. 221) will be found to follow the curvatures of the plate more smoothly, and not give rise to jerky motions. The speed at which the rolls are worked is about 18 ft. per minute, but it takes altogether about thirty minutes to bend one piece of shell plate. Before commencing the bending, a circular chalk line, to judge of the curvature, is drawn on the floor, passing through the roll space; but it is also necessary to have curved templets, with which the upper edge is gauged, as the wear on the roller bearing is not an equal one.

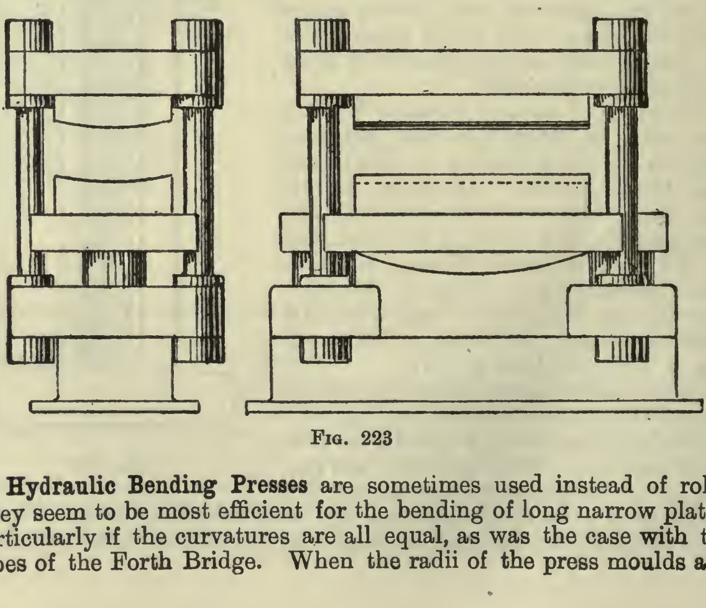


Fig. 223

Hydraulic Bending Presses are sometimes used instead of rolls. They seem to be most efficient for the bending of long narrow plates, particularly if the curvatures are all equal, as was the case with the tubes of the Forth Bridge. When the radii of the press moulds and the shell differ materially, liners have to be interposed, as in fig. 222, producing a rounded polygonal, instead of a perfectly circular shape. This is the case even, though to a much less extent, when no liners are used, for the press is never powerful enough to force the plate into absolutely close contact with both moulds.

Templets have to be applied to the plates while being pressed step by step, otherwise irregularities are sure to occur. But no

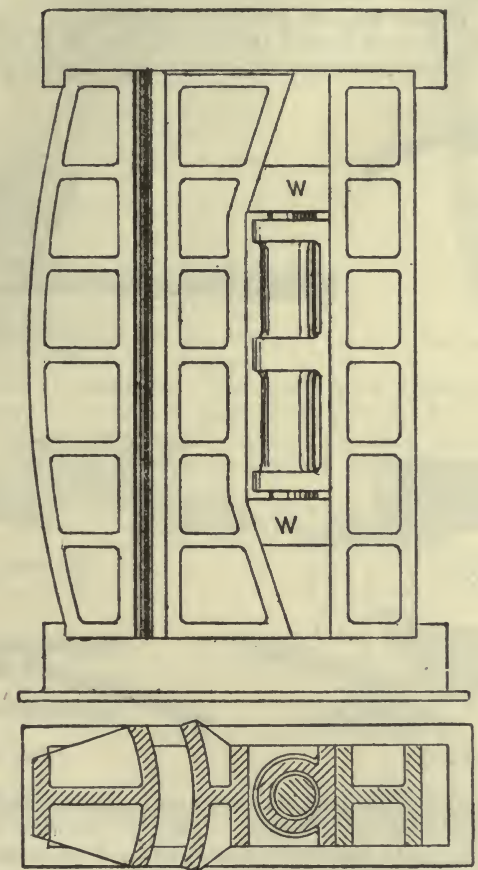


Fig. 224

amount of care can do away with the irregular distribution of the stresses in the plates, and this plan cannot therefore be looked upon as a good one for furnaces, because with them the stresses are compressive.

A few sketches of some hydraulic bending presses are shown in figs. 223, 224. In the first of these the two rams act directly on the press frames, while in the other motion is imparted by means of wedges, W. In both cases it is necessary to let the two plungers move together; this is easily done by working the two force pumps from one shaft, and by having an accumulator with two rams instead of one.

Shell Plate Ends.—From previous remarks it will have been gathered that one of the chief difficulties to contend with is the bending of the ends of the plates in such a manner that the general curvature is uniform, no matter whether the joint is to be butt-strapped, welded, or lap-jointed as in fig. 225.

There are various ways of producing these forms. In some works the end of the plate is heated before bending; it is then laid on a slanting anvil block (fig. 226), with or without curvature, and hammered


Fig. 225

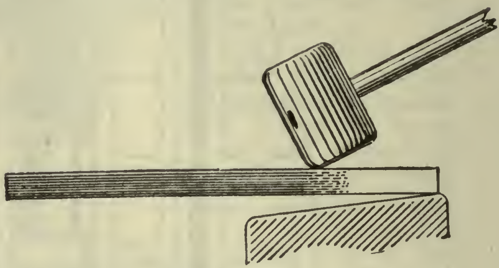


Fig. 226

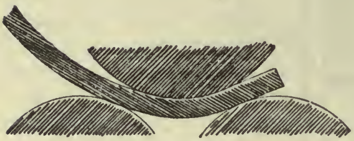


Fig. 227

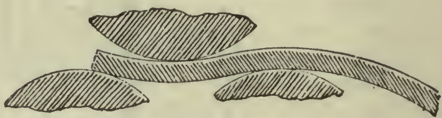


Fig. 228

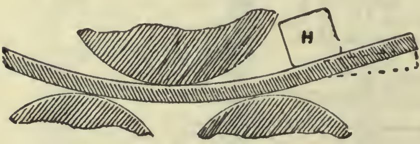


Fig. 229

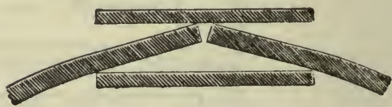


Fig. 230

with mallets. It is then heated over its entire length, and put through the rolls, or it is bent cold.

Another plan is to give the top or inner roll an extra screw-down when the end of the plate has been reached (fig. 227). In order to get the correct shape on the other end of a lap-jointed plate it has to be taken out of the rolls and reversed, as in fig. 228.

This plan is unsatisfactory unless carried out on hot plates, and even then it is clear that the joining surfaces are not flat but irregularly curved.

A better shape is obtained by the following plan, but it is attended with great danger, because the bending may accidentally be carried out at a blue heat, producing either fracture or brittleness. The plate is fixed as shown (fig. 229), and a heater placed at H, and when locally warmed the bending is done by hammering. The ends of plates to be butted are sometimes left flat, and are drawn together by the butt straps (fig. 230). Barbarous though this method is, the finished shape is a fairly true one, because the strength of the two butt straps is about equal to that of the shell plate, and the resultant curvature is the same as for the rest of the boiler. With thick plates this plan gives too much trouble while the riveting is going on, and under any circumstances it is not a satisfactory one. In some works the plates are ordered extra long and the ends cut off after bending and used as butt straps.

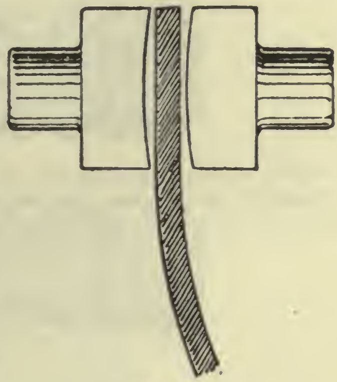


FIG. 231

The most satisfactory results are undoubtedly obtained by bending the ends of the plates with suitable moulds in a strong riveting machine, either before or after bolting the plates together. The moulds (fig. 231) should be about 9 or 12 ins. long, and gently rounded at their ends. The bending or pressing is done cold. The same moulds can also be used for curving the butt straps.

Ends of Riveted Seams.—In lap-jointed shell plates the corners have to be tapered off previous to bending (figs. 232, 233). This is

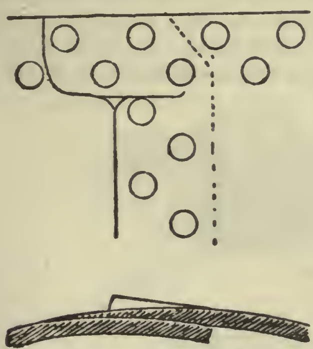


FIG. 232

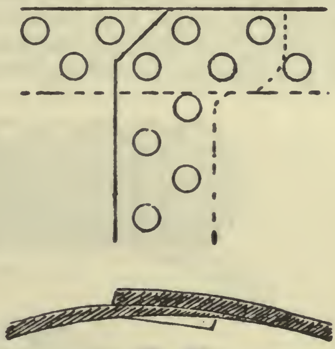


FIG. 233

FIG. 234

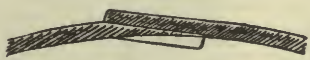


FIG. 235

usually done by heating them and drawing them out under a steam hammer or by hand. On account of the heaviness of the plates the latter plan is most convenient. Fin-shaped tools should be used, at least for the heavy work, and it is always well to re-heat the surroundings of such corners when finished to prevent cracking. Very satisfactory results are obtained by chipping or planing the corners. (See figs. 234, 235, and p. 249.)

In the case of butt-strapped joints the arrangements are various. They are sometimes left square and butted at A A (figs. 236, 237)

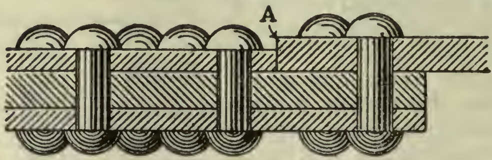


Fig. 236

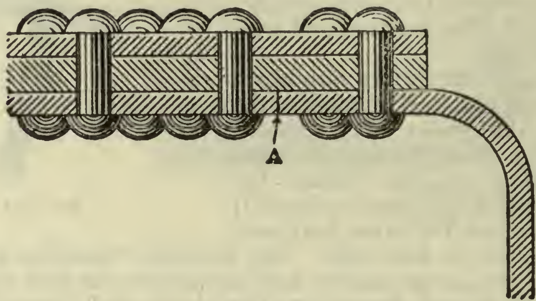


Fig. 237

against the flanged end plates or the adjoining strakes, in which case it is very necessary to be careful that the lengths of the butt straps are correct, and also that they are correctly fitted. Another plan, but one in which the advantages obtained are doubtful, is to bevel the

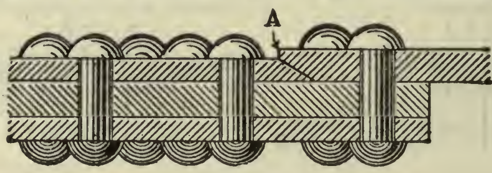


Fig. 238

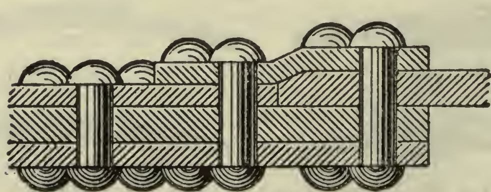


Fig. 239

butt straps (fig. 238). They are more difficult to make, to fit, and to caulk than the previous ones.

In both these cases, but especially in the first, it is customary to fit another cover plate, as shown in fig. 239.

Another plan—probably the most efficient—is to draw out or plane the end of the outer butt strap and chip away part of the shell, as shown in fig. 240. Instead of chipping these parts they may with advantage

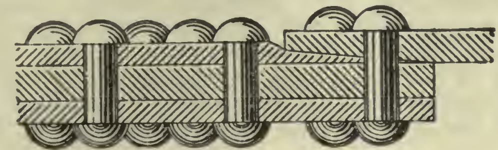


Fig. 240

be planed before bending, and the butt straps might also be planed to shape. Fig. 241 shows another, but not a satisfactory, arrangement. Sometimes the butt ends are welded, or the end rivet in a butt seam may with advantage be replaced by a screwed stud.

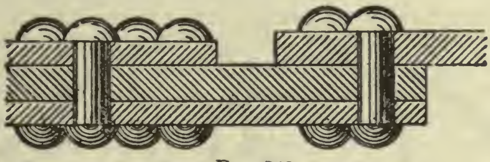


Fig. 241

The Fitting together of the Shell Plates is comparatively simple, they being easily held together by temporary bolts and straps. With butt-strapped joints it is an advantage to be able to draw the butts tight together, and it is best done as shown in fig. 242, by bolting brackets to the circumferential seams and drawing these together by means of strong bolts, but as these butts can never be watertight fits it is not necessary to go to much trouble about them. The end plates, either front or

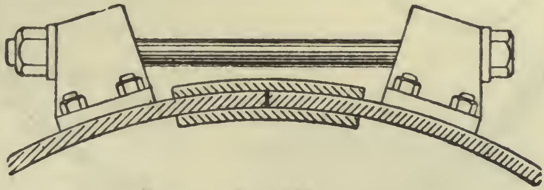


Fig. 242

back, are made in two or three pieces, unless they are small enough to be made in one. The lower part of the end plate can now easily be secured in its correct position by bolts, starting at the bottom; this draws the shell up, and the upper part of the end plate can now be inserted and bolted to the top of the shell. Where the end plates are made of three pieces, the centre one is generally fitted last, but if it is desired to be prepared for having to renew the furnaces, it is the lower plate which should be fitted last.

Riveting.—The older types of power riveters were actuated by steam, while all newer ones are worked by hydraulic pressure.

Formerly, too, it was thought necessary that the pressure should be applied suddenly, in imitation of the blows of a hammer; but it has been found that better results are obtained if the pressure is steady, provided, of course, that it is sufficiently intense. Machines have also been constructed which will form heads at both ends, so that, instead of rivets, pieces of round bars might be used. The results do not appear to have been very satisfactory, except that the heads thus formed required little or no caulking. Practically the same result can be obtained with an ordinary machine by using pan-

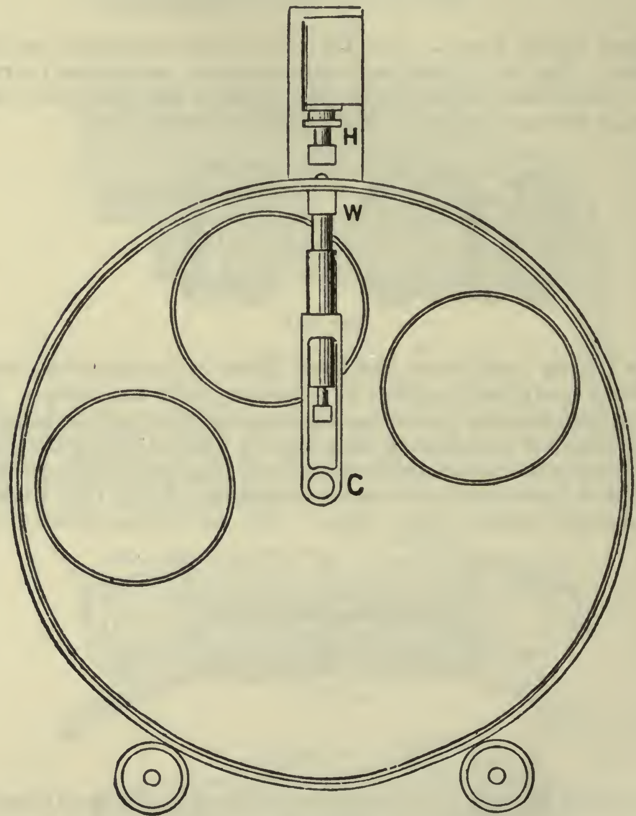


Fig. 243

headed rivets and a spherical die. The deformation which the head experiences assists in closing it up to the plate. Under any circumstances the rivets should be inserted from the inside of the boiler. Rivets should always be slightly larger than required so that they will more than fill their holes; their heads will then have short fins. If these are absent it is probable that the rivets do not fill their holes.

Riveting Machines.—Good illustrations of several types of riveters will be found in the following numbers of 'Engineering':—Vol. xxxiii.

p. 199; vol. xxxv. p. 462; vol. xxxvi. p. 492; vol. xl. p. 317; vol. xlii. p. 80; vol. xlix. p. 259. Portable riveting machines: vol. xxxix. pp. 471, 474; vol. xliv. p. 289 (vol. xliv. p. 299, pneumatic); vol. xlix. p. 256. Hydraulic riveter with plate-closing arrangements, vol. xliii. p. 531. Hydraulic riveter for using rods instead of rivets, vol. xlix. p. 533. Portable riveter for the circumferential seams of boilers, vol. xliii. pp. 490, 491, vol. lxii. p. 422. Electric: vol. lxxxvii. p. 592

The riveting of the last circumferential seams can be performed by the last-mentioned tool when both end plates, having flanges turned inward, have been fitted in place. The front tube plate is made in three pieces, of which the centre one is not fitted till later; the boiler is laid on its back end, and the furnaces, which are not yet riveted to the fronts, are dropped to the bottom, and can, as occasion requires, be shifted to make room for the riveter. This consists of two long arms, of which one passes through the opening of the tube plate to the circumference; the other reaches the circumference from the outside, and contains the hydraulic cylinder, &c.

Another machine (fig. 243), designed for the same object, is worked by steam. To the centre (C) of the end plate is bolted a long arm with a powerful spring and a heavy holding-up weight (W), which presses against the rivet in the circumference of the end plate flange; a strong lever (not shown) is fitted, with which this weight can be pulled back and shifted to the next hole, where the rivet has previously been inserted from inside. The lever is now released, whereby the rivet is pressed home; a small steam hammer (H) is then made to strike the other end of the rivet, shaping it as required.

In works where either of these tools is used—where, therefore, both end plates can be riveted by machinery—the shell seams might have been riveted with the help of comparatively light machines, similar to those used in the construction of the Forth Bridge.

The seam of a flanged out shell plate shown in fig. 244 allows the riveting to be done by machinery, but it is a heavy design, and the seam cannot be caulked on the inside unless a caulking ring is fitted.

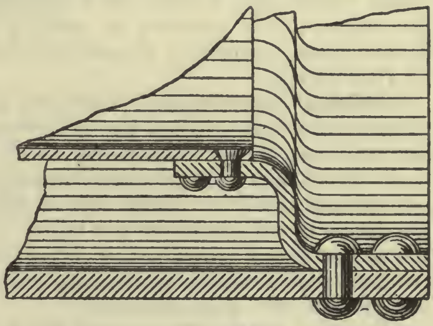


FIG. 244

Riveting Pressure.—Although machine riveting seems a simple matter, there are a few points which, if not attended to, will give trouble. The pressure used should not be too great, otherwise the plate may crack along the pitch line, either at once or when the boiler is in use. A pressure of 100 tons per square inch of section of rivet hole seems to be ample, while anything above 150 tons seems to be dangerous. Allowing for the size of the rivet head, and assuming the red-hot rivet to be as fluid under pressure as water, this pressure would give rise to a tension of 50 tons per square inch in the metal surrounding the hole. Even allowing 50 % for friction and other causes, the stress is still an excessive one.

The distance of the metal from rivet hole to edge of plate must be sufficient to prevent bulging. On p. 215 it has been shown that, as regards strength of joint, it is necessary that the above dimension should be at least equal to $\frac{3}{4}\frac{p-n\,.\,d}{N.}$, where p is the pitch of rivets in the outer row, N the total number of rivets in one pitch, and n the number of rivets in the innermost row. This leads to the generally adopted practice of making this margin equal to the rivet diameter. The shearing stress which would be set up in this part of the seam, while riveting with a pressure of 150 tons per square inch of rivet hole, would be about 20 tons, which is also a dangerous amount, and the edges of nearly all machine-riveted butt straps will be found bulged.

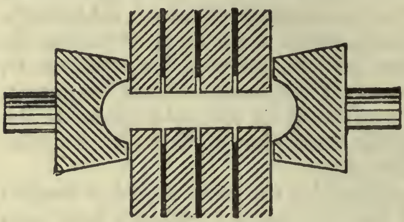


Fig. 245

It must not be forgotten that, as at present constructed, hydraulic riveters exert a greater pressure than their nominal one, for on opening the valve to the cylinder the accumlator weight falls, being arrested only when the rivet is struck. Of course the acquired energy of the drop makes itself suddenly felt as a very serious, but at present not measured, blow, exceeding by many tons the nominal pressure.

It is very important to have the various plates screwed close together; otherwise, and particularly if the heads are small or countersunk and the rivets hot, plastic metal will force its way between the plates, as shown in fig. 245. Some riveting machines are so arranged that they can exert a pressure on the surrounding plates before and during the time that the rivet is being pressed. But whether this machine be used or not, the plates should always be well bolted together; if loose the rivets may be too short and not fill their holes.

Bolting Rivet Seams.—The bolts may remain in place till all the alternate holes are riveted up, or they can be gradually removed while the holes are being successively filled. The latter plan seems to be both slower and more unsatisfactory than the first one, for it tends to stretch and shift the plates, and is adopted for this purpose in shipbuilding when the butts do not meet. It is claimed that this plan has the advantage of warming the plate round the succeeding hole by the previous rivet, thereby preventing cracks; but such a danger ought not to exist with good material.

It will be found that the work can be done both better and quicker if the rivet holes are not filled up in succession, for, in order to do this, the bolts near the riveting machine have to be removed, and that, of course, is inconvenient. If the rivets are put slightly into the holes, and at once riveted up, the boiler shell has to be shifted backwards and forwards over a distance of about 2 ft. before the next rivet can be inserted. If, on the other hand, three or four rivets are placed simultaneously into adjoining holes, the speed of riveting can certainly be increased, but that means that the second rivet is being pressed, while the first one is still hot and pliable, and it will almost certainly be stretched a little by the spring of the plate, so that some of the beneficial influences of the powerful hydraulic pressure are being wasted. On the other hand, if the pressure is kept up till the first rivet is cold there is no advantage in having three others waiting all this time in

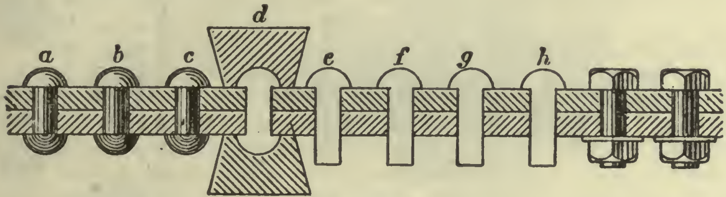


FIG. 246

adjoining holes. This practice is illustrated in fig. 246. The rivet *d* is being closed, and *e*, *f*, *g*, *h* are waiting, while *a*, *b*, *c* are finished. In fig. 247 the rivet *b* is being subjected to the hydraulic pressure; *a* has just been pressed, but cannot well be affected by the new operation on account of a strong bolt filling the intermediate hole. The rivet *c* has just been placed in its hole, and can readily be acted upon when the pressure on *b* has lasted long enough. When every alternate hole

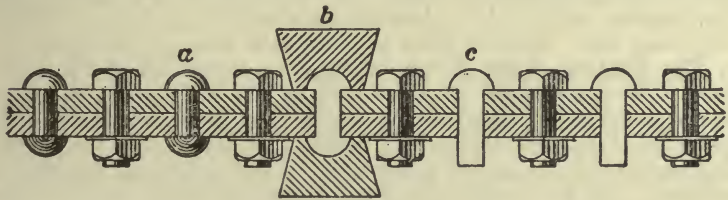


FIG. 247

has been filled, all the bolts are removed together, and their holes are then riveted. After the bolts have been inserted, and before riveting is commenced, hydraulic pressure is applied all round the seam, and if this should slacken some of the bolts they are tightened up.

Irregularly-shaped Rivet Head.—Anybody watching the process of riveting must be struck with the primitive means used for guiding the rivet holes to the dies. An overhead crane, which might be more usefully employed, is carrying the boiler, which is constantly swinging about. Crowbars are stuck into some of the rivet holes, and the men tug at these till the rivet, which has been inserted, is in the right position. As the weight of the boiler shell is often nearer 20 tons than 10 tons, it is natural that there must be much pulling and shoving till the right point is approximately reached. But even then the angular position of the shell is not correct, and the centre line of the riveter does not coincide with the axis of the rivet hole. Naturally, when the pressure is now applied, a few small but violent oscillations take place, and the chances are very much against the rivet head having been formed centrally round its shank. This is illustrated in fig. 248. Of course, externally there is no indication of this state of affairs, except perhaps that the dies leave semicircular marks round the heads, as shown in fig. 249. It ought not to be a difficult matter to devise appliances

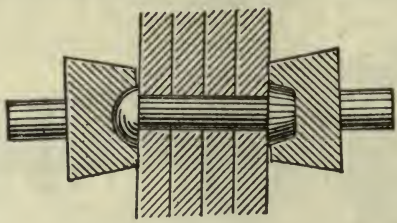


FIG. 248

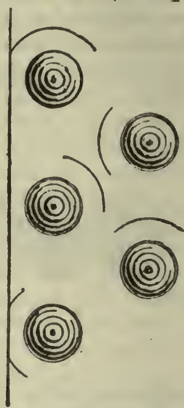


FIG. 249

which would turn the boiler shell to its right position with more certainty than is at present possible.

Time required for Riveting.—When the pressure has been applied it should not be taken off again in less than about one minute (and the longer it remains the better); otherwise the rivet will not have cooled sufficiently, and the remaining spring in the plates will stretch it and reduce its diameter. The men do not generally care to keep the pressure on for so long, because the dies, &c., get hot and soft and soon wear out. It takes about 15 minutes to close up 10 rivets.

The Heating of the Rivets is done in a small reverberatory furnace. Sometimes gas or oil is used as fuel. They should not be raised to too high a temperature, or else they grow too plastic under hydraulic pressure and spread out between the plates (fig. 245, p. 249). Nor should they be heated too quickly, or else only their outsides are softened. On the other hand, it is not good to spend too much time over the heating, for a long exposure, particularly in an oxidising flame, reduces the strength of the material of the outer surface, and it is just this which is subjected to the severest stresses when under working condition. Fifteen to twenty minutes for heating is the general practice.

Unlike iron rivets, steel ones cannot be burnt without showing it when in place.

The influence of heating rivets to a red or a white heat is discussed by M. Considère, 'An. Pont. Ch.,' 1885, 6th ser. vol. ix. p. 574, and 1886, 6th ser. vol. xi. p. 5.

Rivets are made about $\frac{1}{16}$ in. less in diameter than the holes they are intended to fill. The length, L, of their shanks has, therefore, to be made greater than T, the combined thickness of the plates. $L=1{\cdot}5\ D + T\ (1+\frac{1}{8} . D)$, where D is the diameter of the rivet hole. The weights of rivets may be calculated by the formula

$$Q . \text{cwt.} = 2 . n . \frac{D^2}{1000} . (L + 1{\cdot}6\ D),$$

where n is the number of rivets.

Internal Parts of Boilers.—Nearly all internal plates, as well as the end plates, have to be flanged. The only operations which precede this one are the bending and riveting or welding of the furnaces, the thinning or drawing out of some corners, and the drilling or punching of a few holes for temporarily securing the plates. Welding will be touched upon later, and the other preliminary operations require no remark, except, perhaps, that the drawing out of the corners may be done either by a steam hammer or by hand. The latter plan takes longer, but seems to produce a better job. (See p. 245.)

Before discussing the various flanging operations it is well to have a clear idea as to the risks attending them.

Dangers attending Flanging Operations.—Whenever a piece of iron or steel is being heated there is, of course, a danger of burning it. This should never happen while flanging, for during this operation such a heat ought not to be approached. A more real trouble is the wasting away which takes place, particularly with iron, which loses a considerable part of its thickness each time that it is re-heated. There is the further danger of reducing the thickness of the plate by drawing it out, and also the difficulty of producing the correct shape of flanges, and of preserving it, while adjoining parts are being heated.

More serious than any of these troubles is the risk of cracking the plates. There seem to be three ways of doing this :—1st, by using red-short iron or steel; 2nd, by not annealing the flanged plates, which then retain strains that may ultimately lead to ruptures; 3rd, the working of iron or steel at a blue heat. The plates which are thus injured will either break at once or, being now in a brittle condition, will crack later on.

On all these subjects interesting information will be found in the chapter on 'Strength of Materials.'

The Danger of not Annealing flanged plates has been demonstrated over and over again in boiler yards by plates cracking. Few of the manufacturers care to give details of their experience, especially as the steel makers readily substitute new plates for the spoilt ones, and do not care to have such cases noised abroad. Those which have been published will be found in the chapter on 'Strength of Materials,' but numerous others are continually occurring. Flanged tube plates (figs. 250, 253, 256), flanged end plates (fig. 255), furnace fronts (fig. 251), and with ships the huckster plates, boss plates (fig. 254), and garboard strakes, sometimes give trouble by cracking.

Stresses Due to Flanging.—Here it will not be out of place to draw attention to the fact, which is thoroughly supported by experiments, that iron and steel are affected in their quality by severe stresses, and particularly that, by subjecting test pieces to a slowly increasing but severe pull, their ultimate strength can be raised, while their ductility is very much reduced. The most important point is that the limit of elasticity rises under this stress, until it actually exceeds the original strength of the material. In other words, taking a test bar whose limit of elasticity is 10 tons, and whose ultimate strength is 30 tons, with 20% elongation, it is possible, by slowly increasing the load, to raise both the limit of elasticity and the ultimate strength to about 35 tons, while the elongation which takes place during the final loading is reduced to a very few per cent., possibly to nothing. When this point is reached, the test piece gives way completely, without any additional weights being added. Now this gradual increasing of stresses is reproduced while flanging a plate. The shape does not matter much, except, perhaps, as regards the intensity of the stresses which are set up. In forming a tube plate, first one side is flanged, except at the corners, then the next one in the same way, and then the third. When this is finished the corners are once more heated and flanged. Now it is clear that, if the operation commences at A (fig. 252), the heat will have little effect at first; but even while B is being flanged and A is cooling, strong tension stresses would be produced; these are very much increased by the time that C is finished. Each one of these parts has probably had a chance of cooling slowly from about 1,000° F., and if the plate were now put aside, and

Fig. 250

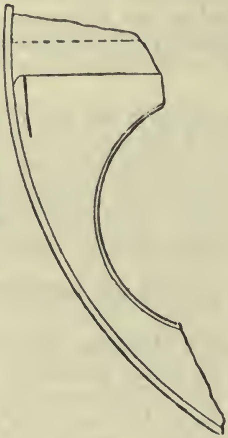


Fig. 251

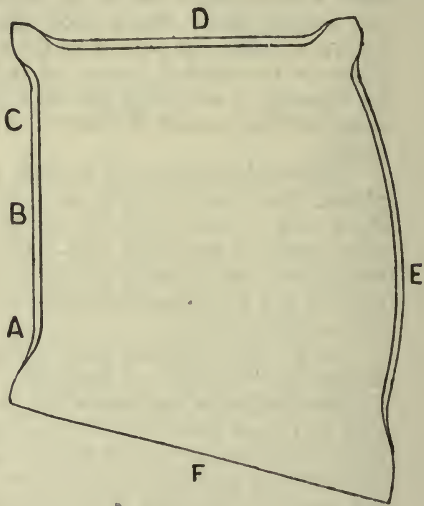


Fig. 252

measurements could be taken, it would be found that these lengths of the plate had not shortened as much as would be due to their change of temperature. If a steel bar, 1 in. in area, is rigidly secured at its ends, and cooled from 212° F. to 32° F., it would then exert a pull of about 13 tons, so that it is not unreasonable to imagine that the cooling of the flange from a red heat would produce a stress of at least 20 tons. But a tension stress at one edge of a plate must produce a compression stress at the centre and a tension at the circumference. In other words, the plate yields a little, and possibly, instead of finding a stress of 20 tons at only one edge, we find a fairly uniform stress of 10 tons all round the plate, and a compression stress in the centre. While the second side is being flanged new stresses are set up all round; then, when the third side goes through this manipulation, the stresses will be once more increased. To give a correct idea of the distribution of the stresses would be impossible, but for the purpose of illustration it may be assumed that the flange ABC is now subjected to 30 tons tension, the flange D to 20 tons, and E to 10 tons. The unflanged parts, F, would be subjected to 30 tons, and the centre of the plate is in compression. The next operation consists in flanging the corners, and there is no doubt that heating them relieves the other stresses at the circumference a little, but on cooling they will be more intense than before. Even now, if the cooling were carried out quickly, there might be no danger, because one or the other part would gently elongate; but, as is usually the case where a failure has subsequently occurred, the plate has been put aside for the night, and next morning the flat part had cracked (fig. 253). Slowly the breaking stress was reached, and then, as in the case of test pieces, the material gave way completely. In a case mentioned by the late Dr. Kirk (fig. 253), 'N. A.,' 1882, vol. xxiii. p. 131, the crack extended about 18 ins. into the plate, and measured ½ in. open at the edge. It is not at all certain that, by cracking, the plate was relieved of all its strains; but even assuming this, and assuming also that the stresses were uniformly distributed over the total circumference of about 14 ft., the opening of the crack to ½ in. would show that in this particular case the circumferential stress amounted to $\frac{1}{2} \cdot \frac{13,000}{168}$, or nearly 40 tons per square in., showing that the above estimate of 30 tons was not too high. Other cases are illustrated in figs. 250, 251, 253, 254, 255, 256. It has not yet been possible to obtain tensile tests of plates which have not failed, but which might be expected to do so. They would certainly throw a strong light on the subject. Test pieces should be sawn (not sheared) out of the plate all along its circumference. An accurate measurement of the limit of elasticity would give conclusive information as to the intensity of the stress which that particular part had been resisting; but the greatest care would have to be taken not to bend the sample (p. 152).

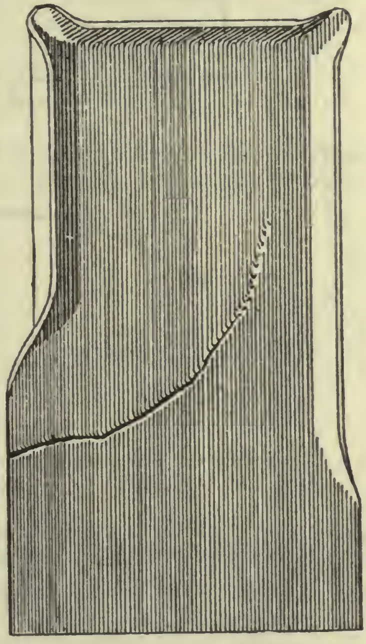


Fig. 253

Before leaving this subject it may be as well to explain why cracks of this sort extend so far beyond the overstrained part. The contraction of area at the edges of the crack at its inner end, and even beyond this point, is a sure sign of the ductility of the metal, at least in the centre of the plate. A careful study of the subject will show that at the instant when a fracture takes place the two separating surfaces are travelling away from each other at a velocity equal to that of sound. For iron and steel it is 17,500 feet per second. The rate at which the crack mentioned above was extending towards the centre of the plate must have been = 1,260,000 feet, or 550 miles, per second, and it is

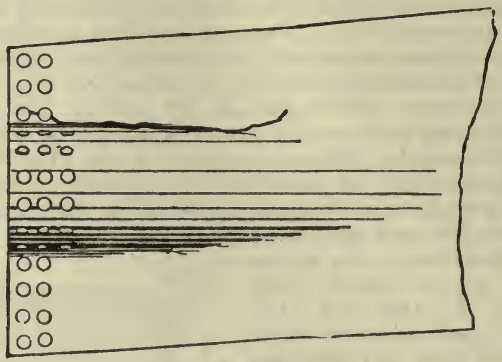


FIG. 254

therefore not surprising that it had overshot its mark. Instances could be mentioned where the crack had extended right across the

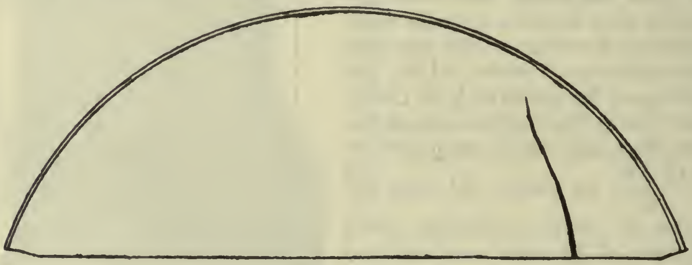


FIG. 255

plate, and in one case a plate actually broke in two, one piece knocking down a man. If the above estimate of 40 tons stress per square

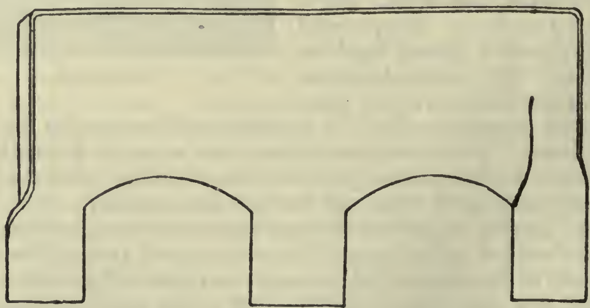


FIG. 256

inch is a correct one, then the amount of energy which was relieved by the plate cracking was equal to lifting it bodily 16 feet into the air.

The loudness of the report when these cracks occur, and the destructive energy which some steel armour plates have displayed when cracking spontaneously, are proof that these estimates are not excessive. The existence of such mischievous powers in the interior of boiler plates should not be tolerated, and there ought really to be no objection to the annealing of plates which have been flanged. When this cannot be done at once, the centre of the plate should be heated to redness immediately after flanging, and before the edges have lost their heat. The compression stresses in the centre of the plate are thereby partly removed. If the plates have been very much buckled during flanging, they should not be flattened except during the annealing process, for it is chiefly the very flat plates which crack.

Working Plates when Partly Cooled.—The other danger to which allusion has been made is the working of steel and iron at a blue heat. Experiments on this subject will be found in the chapter on 'Strength of Materials.' Here it is only necessary to draw attention to workshop practices. The ease with which such failures can be attributed either

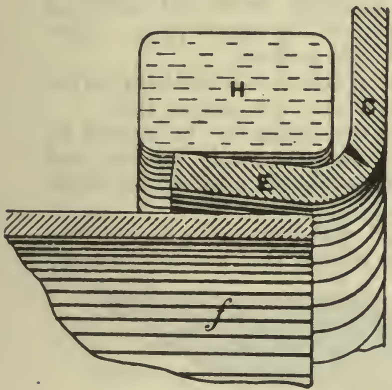


FIG. 257

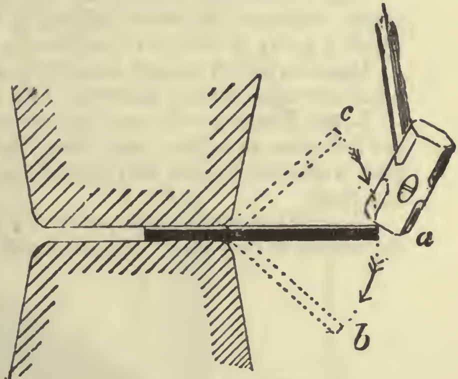


FIG. 258

to redshortness, coldshortness, or generally unsuitable material, and the absence of any chemical or mechanical test, make it difficult to be sure, in any particular case, what has been the cause of the breakage; but about the following one there can be little doubt.

The flange E (fig. 257) of part of a furnace front plate was so much out of shape that the furnace, *f*, could not be drawn up sufficiently to make a good job, and it had to be knocked in a little; but being rather thick, a substantial heater, H, was first applied, as shown. A few weeks later, while the boiler was being tested by hydraulic pressure, a loud report was heard, and though no leakage took place, an internal crack, *c*, was ultimately discovered, extending round one-quarter of the flange, yet penetrating only to within one-sixteenth of the outside surface. Too little is as yet known about this subject, though every boiler-maker should be made aware of it, which can easily be done by letting him make the following bends:—

Strips of mild steel 6 ins. long, $1\frac{1}{2}$ in. wide, and about $\frac{3}{8}$ in. thick, should be treated as follows:—

No. 1 should be placed under a steam hammer (fig. 258), allowing about 3 ins. to project as far as *a*. This end should then be bent down to the angle *b* by striking it with a sledge hammer. It should then be reversed to the position *c*, and again bent down, and the operation continued till breakage takes place. If good, the material will stand 20 half-bends, either with sheared edges, annealed, or hardened.

No. 2 should be placed between two heaters, H, H (fig. 259), and kept there till its edge turns straw colour to violet. It should then be taken to the steam hammer and bent as before. Two instead of 20 bends will now suffice to break it.

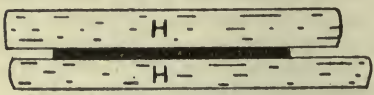


Fig. 259

No. 3 should be treated like No. 2, but the bending should only be carried on till there is the first indication of a crack. The sample should then be put aside for a day to cool slowly, when it can readily be broken with a hand hammer or by throwing it on an anvil.

No. 4 should be heated like No. 2, and then drawn out under the steam hammer till its thickness is reduced by about $\frac{1}{16}$ in. After waiting a day it will have become as brittle as glass.

Cases in which the influence of working flanges at a blue heat may at least be suspected are shown in figs. 250, 251. Compare p. 150.

Local Heating.—It may be possible that plates can be injured by the influence of a blue heat even while they are partly red-hot, and that is another reason why they should always be annealed after working them.

Fig. 260 shows a plate partly flanged, and also locally heated near the centre of one edge. Evidently this red-hot part is surrounded by a zone of which one part is blue hot.

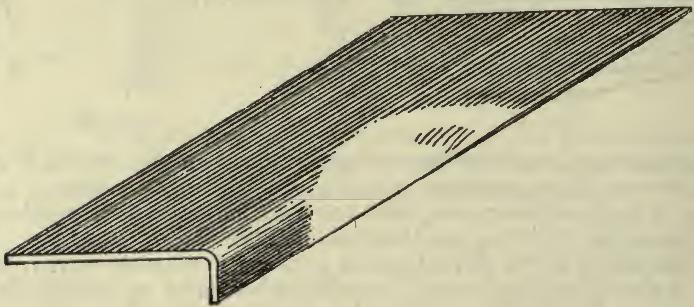


Fig. 260

There is also a strong probability that one part of this zone will retain its temperature for a long while, for although the whole plate is radiating heat, the colder parts do not lose it as quickly as the hot ones, and are also being warmed by them, so that at some point the gain and the loss must be equal during a considerable period. If the plate is being hammered while this point of permanent temperature is just blue hot, very serious injury may be done to the material. It is therefore not as improbable as it might otherwise seem to find that such a plate contains as many brittle zones as there have been heats applied to it. The danger of these local defects is accentuated by the adjoining parts being tough and ductile, and such a plate might be compared to one made up of glass ribs and sheets of lead. There is naturally much difficulty in reproducing these conditions experimentally, but the danger exists all the same, and should be guarded against by careful annealing.

Cases have occurred where unflanged plates whose corners had been drawn out and then laid aside cracked overnight. It is difficult to imagine that the quality had nothing to do with this, though the tests were good, but stresses had evidently been set up in a similar manner to those explained on p. 253.

Hydraulic Flanging.—Formerly presses similar to that shown at fig. 223, p. 240, were used for flanging, but the necessity of requiring

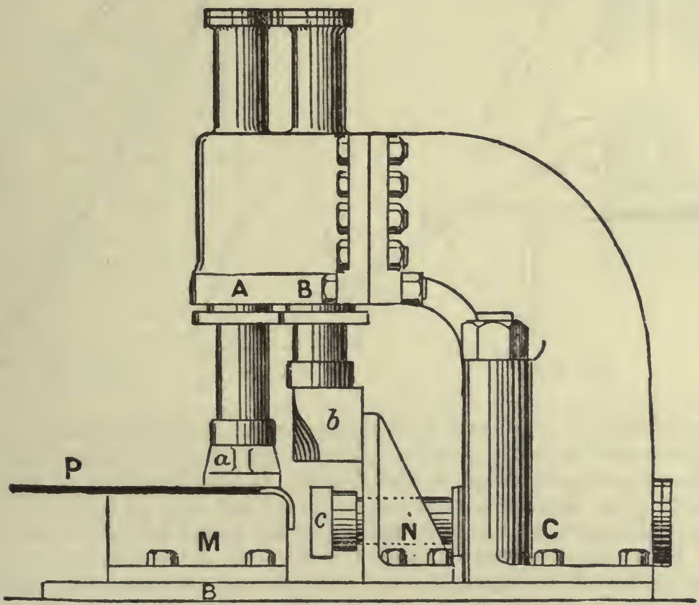


Fig. 261

numerous moulds and various other inconveniences have led to the very extensive adoption of Tweddell's flanging press (fig. 261). See also 'Enging.,' 1912, vol. xciv. p. 83. It consists of three hydraulic cylinders, A, B, C, to whose rams various moulds or head-pieces can be keyed, *a*, *b*, *c*. The whole is supported on a strong bed-plate, B, to which the mould block M and the angle frame N (for guiding *b*) can be bolted. The press is shown in the act of flanging a boiler end plate, P, which is firmly held down by *a*, while the moulding iron *b* is forced down the side and bends the plate. After each stroke *a* is lifted so as to allow the plate P to be moved a little, and then *b* descends again. When the whole length of one heat has been dealt with in this way, the flange will be very irregular and frilled, as shown in fig. 262. These irregularities are removed by forcing the ram *c* against the circumference of the flange. The head-piece *a* should be of ample size, so as not to injure the plate, but not too large, otherwise the plate remains perfectly flat after flanging, in which condition it is more liable to crack than if somewhat warped or buckled. The plate should be hottest at the edge, for if that part is left dark red, the thickness of the plate will be reduced at the bend, and the puckers cannot be easily removed. Fig. 263 shows an end plate partly flanged. One part, *a*, is bent down, the other part, *b*, is still straight, and the metal between *a* and *b* must have stretched

Fig. 262

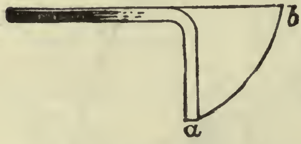


Fig. 263

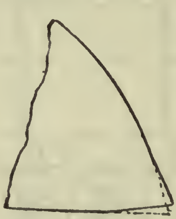


Fig. 264

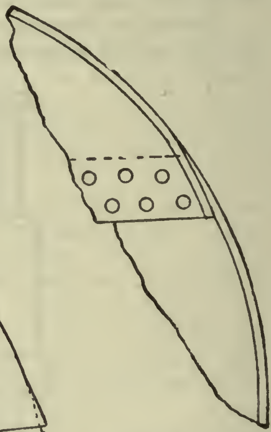


Fig. 265

considerably. Besides this the circumference of the plate at *b* is greater than that of the flange *a*, so that when finished there must be a considerable compression stress in the flange. It has the effect of slightly bending the plate edgeways, as can be noticed by the curving of any straight line which has been scribed on the plates before flanging.

Short lengths, varying from 4 to 6 feet of the circumference (and even 8 feet of thick plates), are heated at one time and the flanging completed before the next length is taken. The heating takes about twenty minutes, and the flanging is at the rate of about 1 foot in 2 minutes. The extreme ends of the flanges are never of the shape finally required, and have to be dealt with subsequently by hand.

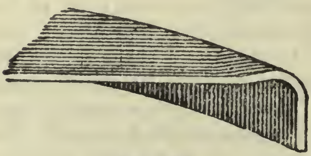


Fig. 266

Fig. 264 shows how the corners of the plates are cut previous to flanging. The outside plate (see fig. 265) is drawn out at the corner, and when subsequently flanged has a very irregular appearance, as shown in fig. 280, p. 265. Fig. 266 shows how the corners rise up. This is due to the above-mentioned compression of the material. Before allowing the flange to cool, the corners of the inside flange have to be knocked in with a few blows of a hammer, and the corners of the outside flange knocked out; otherwise these parts will have to be re-heated before fitting together, which would be a great waste of time.

Flanging with a Steam Hammer is done when no other means are available. The plate is placed at an angle on rollers (fig. 267). H is the moulded hammer-head; M is the lower mould, keyed to the anvil block. A tie rod or plate is also bolted to it for holding the pivot of the plate P, which has been firmly bolted to the centre. The flanges occasionally get torn by this rough treatment, and must then be welded.

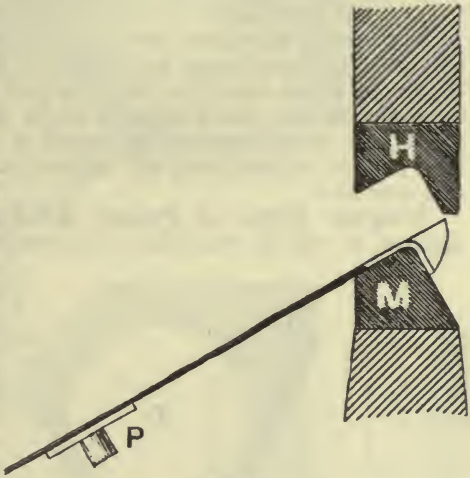


FIG. 267

Hand Flanging. — It goes without saying that up to certain thicknesses flanging can be done by hand. It is usual to dispense with pivots and stops, which are necessary with machine flanging, and to be guided as regards shape by deep centre punch marks on the plate. For end plates it is always better to use a pivot; the circumference can then be made more truly circular.

The size to which plates have to be sheared for flanging depends, of course, on the depth of the flange, but also on the skill of the operator. With some a very much greater margin must be left for irregular work than with others. The following is a customary rule, and produces flanges with an average of 1 in. for waste:—

To the external size of the flanged plate add the depth of the flange, measured inside.

For end plates of boilers this is the usual custom. The furnace holes are, however, cut 2 ins. smaller in diameter than would be required by this rule, while for machine-pressed dome ends the extra margin for the flanges is reduced to about 75 % of the above, because they draw out considerably.

Tube plates and combustion chamber back plates have to be ordered with a margin which is 1 in. in excess of the depth of the flange.

An extra allowance of 1 in. has to be added at the corners of plates to be flanged by a steam hammer. (See fig. 264.)

Flanging Furnace Holes.—Another operation which is carried out under the press is the flanging of the holes in the furnace front plates. Having been bored out to the correct diameter, as indicated above, the circumference of each hole is heated and flanged separately. This is also done under a Tweddell's flanging press. A strong cast-iron ring mould is bolted to the bed plate, and the two plungers *a* and *b* (fig. 261) are secured to a strong cast-iron die. The cocks and valves of the press are then altered, so that the plungers work in unison, and when the furnace front plate has been placed in its proper position on the ring mould, the die is forced through it, producing the desired flange The power required seems to be at the rate of about 5 tons per foot of circumference with a $\frac{3}{4}$-in. plate and with a sufficient depth for one row of rivets, and double this power for inch plates or for treble-riveted flanges. After being heated it takes about 15 minutes to carry out the flanging.

The furnace front plate should be firmly held in its proper position, because a tendency exists for the die to drag down only one side of the flange, and in doing this the plate gets moved. If the ring mould is strong enough, the plunger *c* might secure the plate. The die should also be as taper as the length of the stroke will allow; less force is then required, and the tendency to shift the plate is reduced to a minimum.

Irregular Shapes of Furnace Holes.—Simple as the operation appears, it will be found that the holes produced in this way do

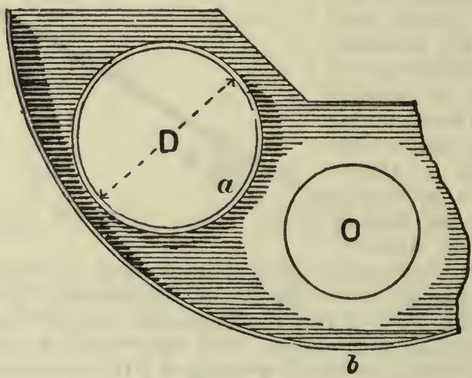


Fig. 268

not always turn out perfectly circular, unless certain precautions have been taken. One of these is not to heat the plate further than is absolutely necessary, so as not to warm, and thereby weaken, those flanges which have already been finished. Thus, if the finished flange at *a* (fig. 268) were to be heated while the circumference of the hole O is over the furnace, a slight contraction would take place at this point during flanging, and the diameter D would be reduced. Similarly, a contraction would take place at *b* if the outside flange were heated at this point. This contraction is the necessary consequence of the stretching of every part of the circumference of O, just as the closing-up action, while flanging the outer circumference of end plates, tends to elongate the adjoining parts.

These deformations can be partly prevented by placing a stout iron ring (fig. 269), cut as shown, inside the finished flange and tightly wedging it into position, but only after the adjoining hole has been heated, and just before it is placed under the press. In some works all three or four furnace holes are flanged in one operation, and sometimes also the outside flange is done at the same time. This of course requires a very powerful press, similar to the one shown at fig. 223, p. 240. The great inconvenience of this method is the large number of moulds and dies required; but the results are highly satisfactory, the only serious trouble being the drawing away of the

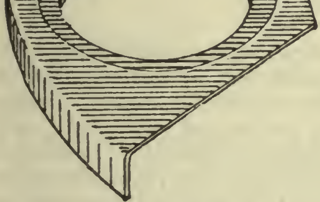



FIG. 269

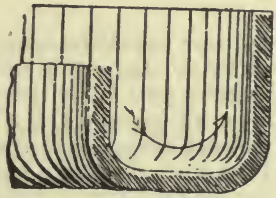


FIG. 270

metal from the weaker or lower flange towards the stronger one (see fig. 270), which very often leads to the outer flange being higher and the inner one being lower than either should be. Fig. 271 is a section through a furnace-hole flange, and shows the deformation to be expected.

The actual flanging operation is easier with iron than with steel, because this metal is softer, and because it can be worked at a higher

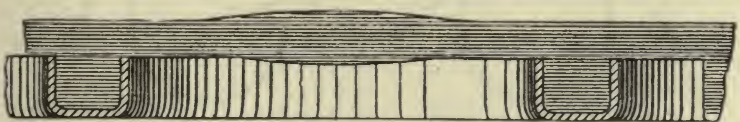


FIG. 271

temperature, and fewer heats are necessary. On the other hand, only the very best qualities of iron can pass through this operation without showing cracks or other defects, and there is also much waste of thickness, due to the burning of the surface. That steel possesses great advantages is proved by the fact that hardly any other material is now used for this purpose.

Annealing.—It might have been better to postpone the necessary remarks about annealing until all flanging operations have been discussed, but, as there is no more difficult piece to deal with than a large furnace front, the subject has been introduced here.

The heating furnace is of the ordinary reverberatory type (fig. 272). It is usual to employ coal for fuel, but gas is also used, and it has been affirmed that it causes plates to grow brittle. The flame travels right across the furnace, and gets drawn down at the front. If the furnace is very large, the firing is done at one side, and the downcast is placed at the other. Doors should be fitted so that the plate can be watched,

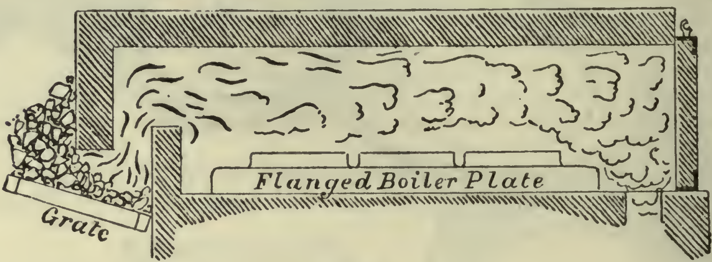


Fig. 272

and care should be taken to keep the temperature comparatively low, partly in order not to burn the iron or steel, but particularly not to heat one part of the plate before another; otherwise serious distortions will occur. Large plates are taken out, turned end for end, and replaced before finally allowing them to cool. J. H. Brinell, 'I. and S. I.,' 1886, vol. i. p. 365, has shown how best to anneal. The plates should be heated quickly to a good heat, cooled rapidly at first and then slowly. Long heating at a low temperature is bad, and can be detected with the microscope.

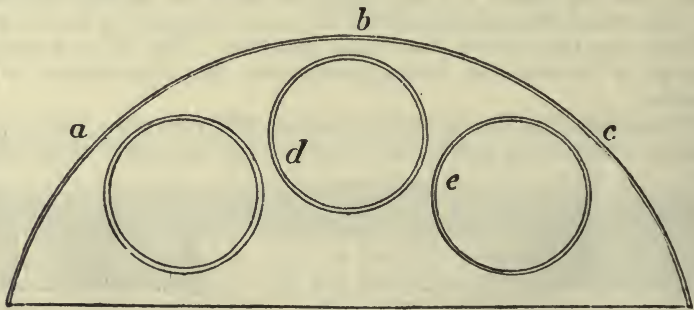


Fig. 273

Deformations Produced by Annealing.—Even with the most uniform heating it will be found that the strains which have been produced while flanging will make themselves felt in the most annoying manner. As already mentioned, the tilting up of the outside flange shortens it, so that it is in compression, and the adjoining part of the plate will be in tension. In a furnace front plate this is counteracted by the stretching of the furnace-hole flanges, but only partially; for, on heating such a plate, it draws as indicated in fig. 273. The metal at a, b, and c elongates, and the outer flange comes in, sometimes as much as ½ in.

The metal shortens at d and e, so that in one direction the diameters increase and in the other they decrease, their difference occasionally amounting to as much as 1 in. Partial relief is given by annealing the furnace front plate after the circumference has been flanged, and once more when the holes are finished.

To flange the furnace holes oval, which is sometimes done, does not give good results, partly because the changes of form cannot be previously estimated, but more particularly because the outside flange gets drawn in during annealing, and then, although the furnace holes may be circular, the outside flange is locally flattened.

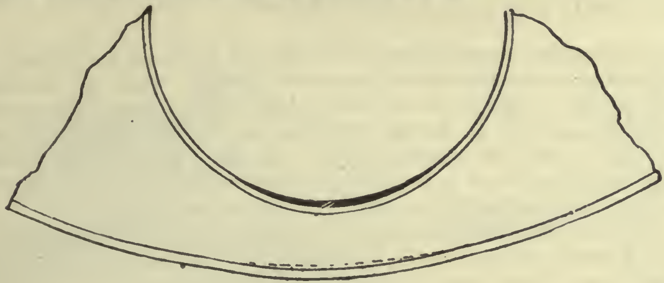


FIG. 274

All these distortions depend not only on the shapes and sizes of the flanges, but also on the nature and number of the heat at which the various parts were bent. By making the outer flange and that of the furnace hole perfectly circular, then reheating both locally, and forcing them out at this point by means of a liner shown in black in fig. 274, and then annealing the plate, the flanges return from the position by the black line to their original shape, as shown by a dotted line. If they are still oval these parts are set by hand during the process of annealing, the plate being drawn out of the furnace for this purpose and partly flattened, and then replaced and reheated, and if necessary reset. Where great accuracy is aimed at, solid segments of circles (fig. 275) are wedged into the furnace holes before annealing; they have to be made rather strong, but for all that waste away if used too often.

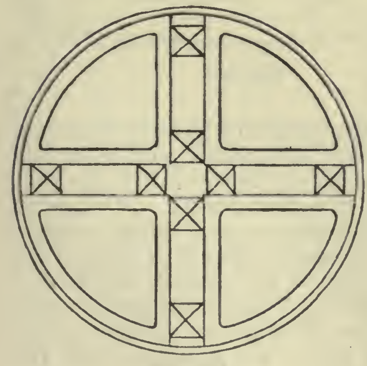


FIG. 275

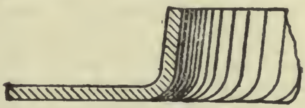


FIG. 276

A careful examination of machine-flanged holes, after they have been annealed, will show that they have closed in a little at their edge (fig. 276). This, like the other deformations, is due to the stresses set up by the flanging operations. It is needless to say that all other flanged plates are distorted more or less by annealing. End plates cannot, therefore, be fitted together before this is done.

The plates are flattened when they leave the annealing furnace; this is very necessary, especially along the edges, where the riveted seams come. The operation is not difficult, but care must be taken

Fig. 277

that the necessary hammering of the flanges does not spoil them. It is necessary to test the circumference by templet while the plates are still hot, and to gauge the furnace holes. Occasionally the circumferential flange will be found set up as in fig. 277. This happens particularly near the furnace holes. In such cases heavy double-handed hammers have to be used to knock it down again. Portable steam hammers are also used for this and other purposes.

Plates in which the flanges of the edges and of furnace holes are turned towards opposite sides are troublesome objects. Not only is it difficult to get them level, but at the points where the two flanges are nearest each other a disagreeable tendency exists for them to tilt over (fig. 278), and also to bend down bodily, as shown in fig. 279. Considerable experience is necessary to get these parts into shape.

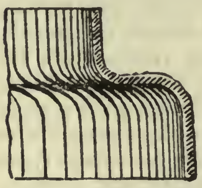


Fig. 278

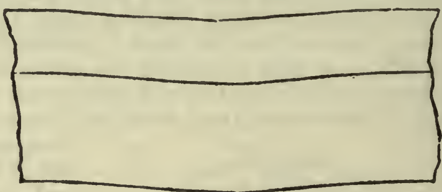


Fig. 279

Plates should remain in an annealing furnace for at least two hours but it takes a few more hours before they have cooled after being laid on the shop floor. A lesson in annealing may be learnt from the plan adopted for optical glass, which leads to perfect results. It is heated to the required temperature, and the furnace is then cooled quickly, but only through a small range of temperature, and it is only when the whole of the glass has adapted itself to the reduced heat that it is lowered once more. This is repeated till ordinary temperatures are reached. If the temperature is reduced steadily the surfaces would necessarily be slightly colder than the inside, and straining could not be prevented.

Final Flangings.—Having been thoroughly annealed the various parts of the end plates are now clamped together, care being taken that the circumference is a true circle, and of the right diameter, sc as to fit the shell. A few holes are drilled through the various cross-seams, and these are then screwed together by well-fitting bolts. Girders, or, for small-sized plates, angle irons, are also bolted to the plates, to keep them flat while suspended, and then one corner joint after another is heated and brought to its correct shape by hand hammering. This is rather a troublesome operation, and generally requires two heats, but if the corners are carefully prepared much time is saved. When, finally, the two plates are in close contact at the corners templets are applied, to test whether the curvature is correct. These parts should project slightly, as they get drawn in on cooling.

The shaping of these corners will be best understood by referring to fig. 280, which shows the irregular shape of the corner plates as they

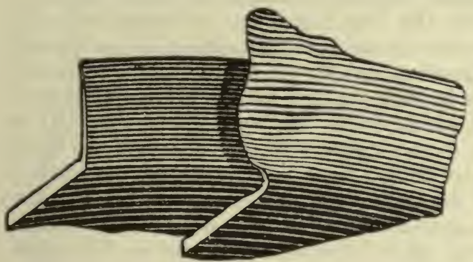


Fig. 280

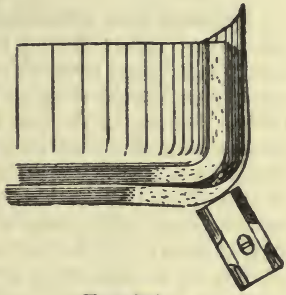


Fig. 281

have left the press. The corner is now heated, and the outer plate struck, as shown in fig. 281, starting at the bottom, so that the humps (which are shown in fig. 280) of both the outside and inside plate are driven in. Gradually the upper parts of the flanges are reached by the hammer. In the meantime, especially for light plates, heavy hammers or weights have been held inside the flange, in order to make the blows more effective and to bring the plates well together. At a later stage the inside is struck to drive that flange out and to stretch the outer one a little. All this hammering is performed with the end plates, suspended by a chain, so that the heated corner is quite accessible. The final operation is performed on a slab of iron (fig. 282). If there are two horizontal seams in the back end plates, it is usual to rivet up one of them, either before or immediately after the above operation. If these corners are welded (fig. 283) none of the seams may be riveted up first; the shaping of the corners is done during and immediately after welding. After this operation it is well, and even necessary, to reheat the neighbourhood of these parts.

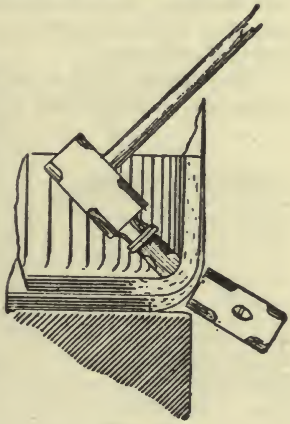


Fig. 282

This has to do service instead of annealing. These welded corners contract about $\frac{1}{8}$ in., and it is necessary to make the proper allowances. It would be well to leave these corners full and trim them cold. The stay holes, unless they have been punched before annealing, or if it is intended to bore them in place, are now drilled, and the plates can now be fitted into the shells, drilled, and riveted up, as already explained. In some works the straight edges of all flat plates are bent so as to make bevelled joints; but this is unnecessary, costly, and dangerous, for heaters have often to be used, and the plates may thus be made brittle.

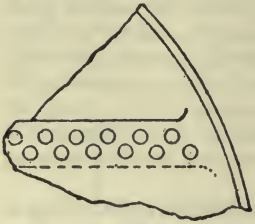


FIG. 283

Riveting End Plate Seams.—The cross seams are often riveted by machinery before fitting into the boiler, and as, particularly in the front plate, some rivet holes are countersunk, the plates do not expand equally during the riveting process, and all the trouble taken to ensure perfect fits at the flanges is in vain, in fact the difference of expansion is sometimes so great as to crack the welded corners. It is therefore advisable to bolt these seams well together and to start riveting from both ends (see p. 283). Even now, however, the corners are not likely to fit close to the shell, so that in some works after the greater part of the flange to shell riveting has been done, these corners are heated to redness by small blast furnaces, then hammered to close them up to the shell and riveted. It is not an ideal proceeding, but being sanctioned by Lloyd's Register it is not likely to be given up. The subsequent cracking of one of these heated corners indicates that if other means for making these corners tight can be shown to be satisfactory they should be insisted on.

Flanging Furnace Saddles.—Undoubtedly, one of the most difficult flanging operations is the shaping of furnace saddles. Recent attempts to do the work by Tweddell's flanging press are said to be highly satisfactory.

Two objects have to be kept in view, viz. not to burn or otherwise injure the material, and not to reduce its thickness. Suppose that a

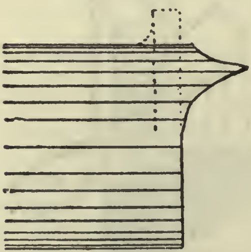


FIG. 284

FIG. 285

FIG. 286

FIG. 287

cylindrical shell (fig. 284) is marked with a number of equidistant longitudinal lines, and that the end of this cylinder is flanged like a furnace saddle, as shown in dotted lines, the straight ones can be made to take up any of the positions shown in the end views (figs. 285, 286, 287). In the first case the lines radiate from the furnace centre, and it is clear that the metal will have been much stretched, and consequently much thinned, at the corner. This happens if the corner is made very hot during flanging, and is also flanged first In the second case the lines which extend up to the corners are parallel, showing that thus far the metal has not been stretched or thinned. But that is not the case from the corner to the crown, where the lines spread out very much. This can only happen if the centre has been heated and flanged first, and kept hot while flanging the corners. Fig. 287

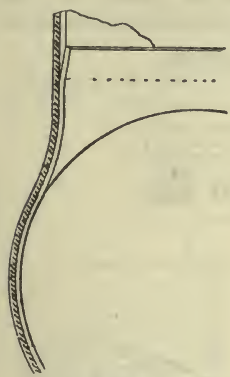


Fig. 288

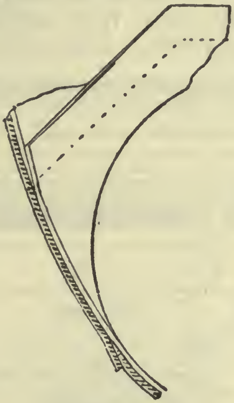


Fig. 289

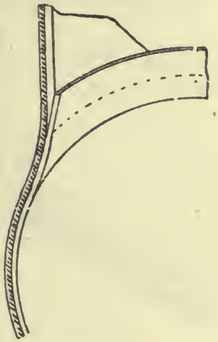


Fig. 290

shows how the lines should have distributed themselves. The available means for attaining this end are: firstly, not to heat the whole width of the flange, but only the curved part, leaving the edges of a dull red heat; secondly, to do the flanging gradually, and not to finish one corner outright, but to partly flange first one corner, then the other, and so on; thirdly, by leaving an extra amount of metal on those parts of the flanges which are most easily drawn thin, and then if their edges are not heated too much they might even thicken some parts near the curve by stumping them up.

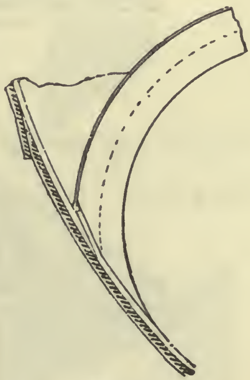


Fig. 291

The most difficult furnaces to flange are those with high saddles, as shown in figs. 288 and 289. As nothing is gained by making them of this shape, it is better to keep the flanges as low as possible, as shown in figs. 290 and 291. The deep flanges usually require from fifteen to twenty heats, while it is asserted that some smiths can produce the low flanged saddles in five or seven heats, but in that case the curvatures must be kept large. Each heat requires about one hour.

The Flanging Operation is carried out as follows:—The anvil mould M is shaped as shown in section (fig. 292). It is part of a segment of a circle, and is bolted to a strong bed plate, B (fig. 293), to which another light frame, F, and a stop, S, have been secured. The furnace is heated at the point where it is to be flanged, and laid on the mould, and the flanging is then done with mallets.

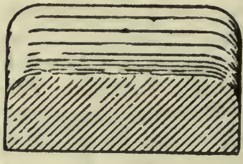


Fig. 292

Care has to be taken while doing this. If the edge of the plate were to be struck first, as shown in fig. 294, it would chiefly bend and stretch the edge (see fig. 295); it is therefore usual to strike at first nearer the anvil block (fig. 296). This blow produces little effect at the edge, which therefore remains thick. Further blows produce curves, as shown in figs. 297, 298, 299.

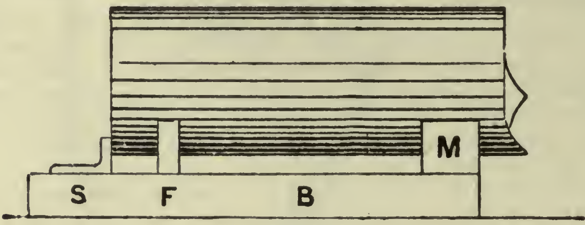


Fig. 293

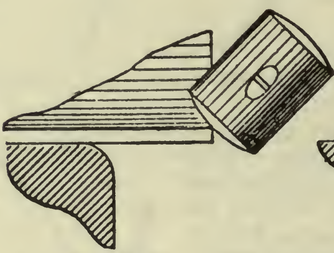


Fig. 294

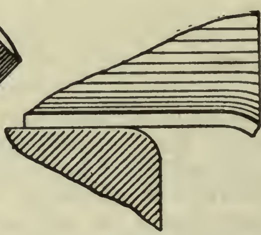


Fig. 295

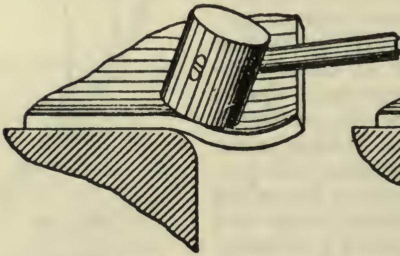


Fig. 296

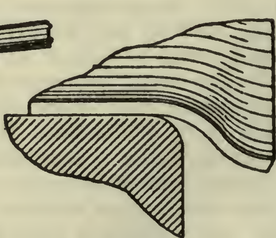


Fig. 297

By starting from the inside more pressure is exerted there, and the tendency to contracting the furnace at the saddle is reduced. It has already been pointed out that the stretching of one part of a plate sets up compression stresses in other parts, which show themselves by a

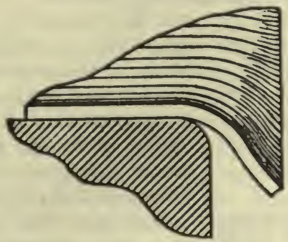


FIG. 298

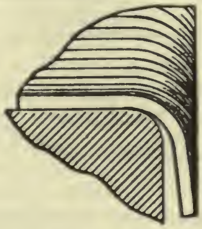


FIG. 299

swelling up (see fig. 300); but this is easily removed with the help of a facing iron, as shown in fig. 301. The effect of a hard tool is to

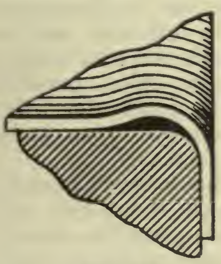


FIG. 300

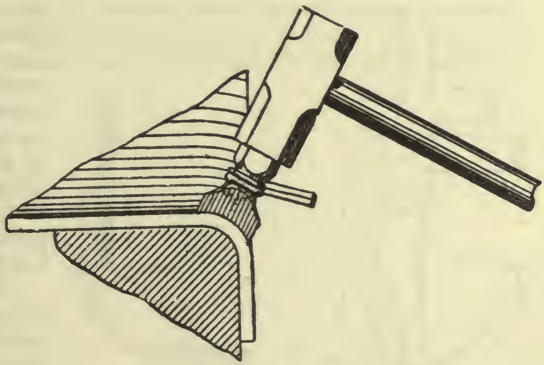


FIG. 301

stretch the metal locally, and that naturally increases the diameter of the furnace at that part.

This bulging was once a very noticeable feature in some patent furnaces. When allowed to remain, such faults have repeatedly been mistaken for indications of weakness or partial collapses, and much unnecessary anxiety or expense might have been saved if they had been removed at first. Very little flanging is required at the corner, for the corners help to draw this part out.

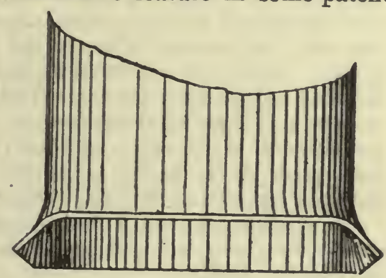


FIG. 302

When all this work has been done the furnace will have the appearance shown in fig. 302. To finish it one of the corners is heated at a time, and placed on an anvil mould, as shown in fig. 303. It is then hammered to the correct shape. If the preliminary flanging has not been carried out carefully, troubles may be expected, not only at the corners, but also with the round parts of the furnace, which easily lose their shape.

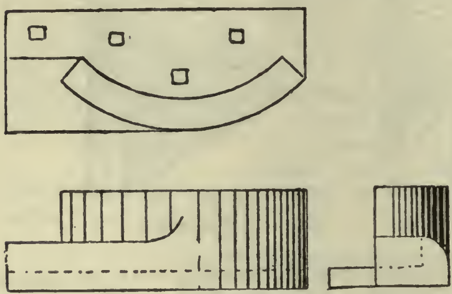


Fig. 303

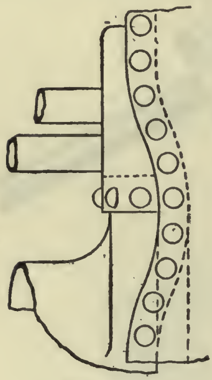


Fig. 304

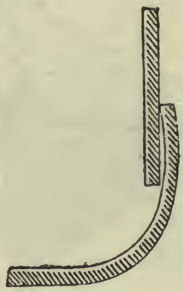


Fig. 305

These anvil moulds (fig. 303) may be dispensed with if the curvatures of the corners are sufficiently large, but this demands a slight deviation from the usual design; either the furnaces have to be flanged round the tube plate, or the top corners of the saddle flanges have to be left long, and the radius of the tube-plate flange increased locally, while the combustion-chamber plate is cut away, as shown in fig. 304. This entails chipping and is expensive. A cheaper and equally efficient plan is to plane the combustion-chamber sides parallel, and to make the tube-plate flange deeper at its lower end.

The furnaces should now be thoroughly annealed, gauged for roundness, and corrected before cooling, and, above all, the vertical part of each saddle flange should be made perfectly flat, so that it fits the flat part of the tube plate without having to be reheated. Where the flues are not supplied ready flanged by the steel works it is best to fit them and the tube plates together before annealing, and to heat and work the corners when bolted together, as is done with the end plates. If the two are fitted together after this process, it is best, but difficult, only to heat the corners of the tube plate, because it is easier to anneal it than an entire furnace. If these various precautions are not adopted it will often be found that the furnace saddle and the tube plate fit as badly as shown in fig. 305. Such work can be detected by cutting out occasional rivets.

In some works this fitting is done cold, but then the use of heaters cannot be prevented, and brittleness, due to working at a blue heat, may be the result.

In some works the furnace saddle flange is left very deep, particularly at the corners, as shown in figs. 288, 289 (p. 267). In others it is kept as small as possible (figs. 290, 291). The latter plan, particularly if the curvatures are not too sharp, is by far the best; not only is it easier to do the flanging, reducing the number of heats from over twenty to less than ten, but, on account of the gentler treatment of the material, its liability to crack, after the boiler has been put in use, is so very much reduced that no fear need be entertained on this point.

That these troubles are not alone due to unequal expansion of parts of the boiler, is proved by the fact that it is the corners of the central furnaces which generally crack, while those nearest the shell, where the strains are certainly most severe, do not suffer so often. But recent numerous failures cannot be said to have fixed the blame on the quality of the material, and the only alternative explanation is that the steel has been injured during flanging, either by burning it, by heating it too often or over fires giving off noxious vapours, by insufficient annealing, by manipulating the corners at a blue heat, or by overheating and straining when in use.

Flanging Tube Plates.—The preceding remarks make it unnecessary to add much about the flanging of tube and combustion chamber plates. As previously mentioned (p. 253), it is usual to flange the straight edges first, and to leave all the corners to the last. This practice need not be adhered to when doing the work under a press. In order to obtain the correct shapes, machine flanging must be done with suitable moulds, and then it is also of importance to shear pieces off the corners, not so much for the purpose of saving labour when trimming the edges, as to prevent the drawing out of the material. Careful measurements will show that the thickness of the metal is distributed irregularly, as shown in fig. 306. By keeping the edges of the plates hotter than the curved part this thinning action is reduced to a minimum.

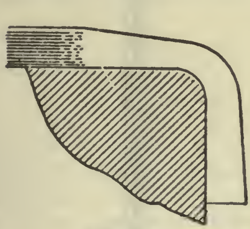


Fig. 306

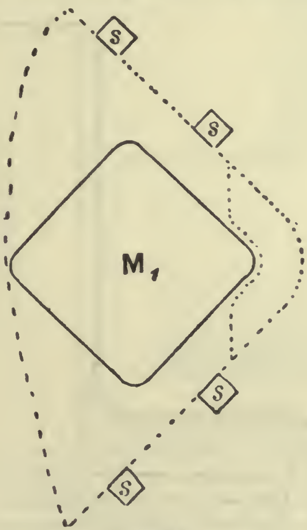


Fig. 307

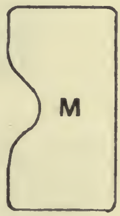


Fig. 308

If the edges have been flanged before the corners, the flanging mould M (fig. 308) should have a good taper, so as to press the finished flanges firmly against the mould block M_1 (fig. 307); for if M is left nearly square there will be a strong tendency to draw the plate away from M_1. This is particularly the case when flanging the corners first, as shown in fig. 307; and then strong stops, S S, S S, have to be fitted. After this operation the plate should have the appearance shown by fig. 309, and the edges can then be flanged with great accuracy.

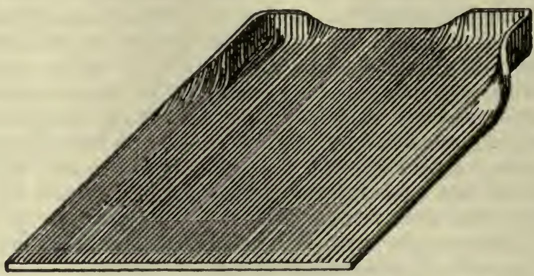


Fig. 309

Hand Flanging.—When flanging by hand stops are sometimes used, though in some works it is deemed sufficient to make a few chalk marks on the flanging mould, and some smiths content themselves with marking the line of flange with deep centre punch marks.

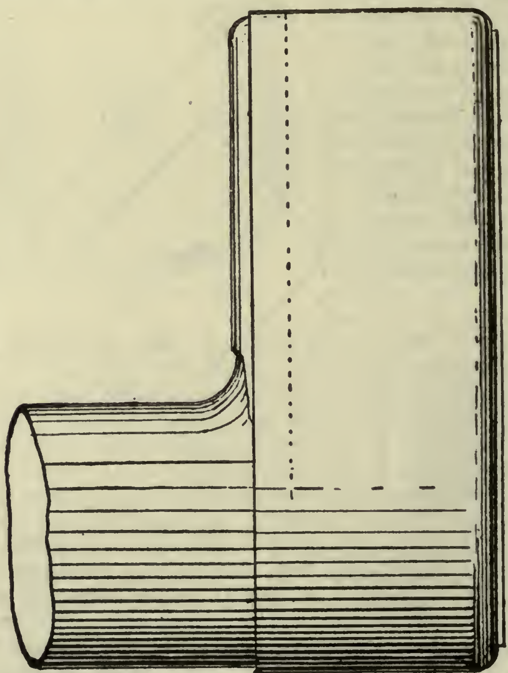


Fig. 310

The combustion chamber sides are seldom flanged. Now and then designs are met with where the back edge is turned in (fig. 310).

There is no advantage in this plan, and the flanging operation is a more difficult one than usual.

Another arrangement is to leave the back tube plate and combustion chamber plate flat, and to flange out the back and front edges of the top and side plates (figs. 311, 312). Unless very much rounded at the corners the flanges grow thin, or tear, and must be welded. Boilers with these combustion chambers are said to be difficult to clean, and objections have been raised against the caulking liners (fig. 312), but they do not seem to give trouble. Some combustion chambers have rounded backs (fig. 313). These must, of course, be bent in the rolls before flanging. Some combustion chamber tops are flanged to meet the girder plates (see fig. 375, p. 288). Instead of the furnaces, the lower ends of the back tube plates may be flanged (fig. 314) like the

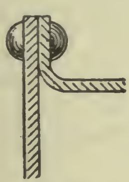


Fig. 311

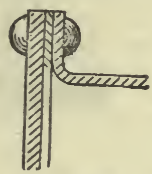


Fig. 312

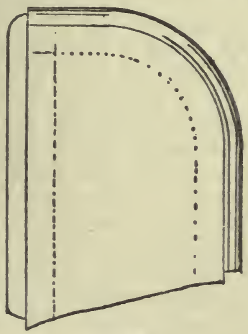


Fig. 313

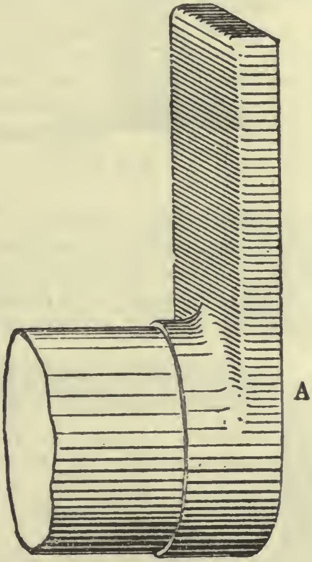


Fig. 314

forward outer plate of the fire-box of a locomotive. It is better not to let the tube plate extend much below the centre line of the furnace, because it is next to impossible to make the two a good fit. Besides, an extra seam across the combustion chamber saves a seam round the furnace bottom. If the tube plate is made to end at A, the top part can be fitted quite close, and there will be less chance of leakage, a danger to which this arrangement is specially liable.

Reference has occasionally been made to the practice of flanging the furnace saddles round the tube plate corners—*i.e.* placing the back tube plate on the fire side of the furnace flange (fig. 315). It has the advantage over the ordinary style (fig. 316) of allowing both seams to be caulked efficiently; nor can steam-bubbles lodge under the landing, and it permits of the radius of the tube plate flange being made smaller than that of the furnace flange. It is asserted that the flame impinging on the caulked edge will do harm, but no trouble has ever been noticed at this point. In the one case the side corners will have to be shaped as shown in end view (figs. 288–291, p. 267); in the other, as in fig. 67, p. 37. In the one case the tube plate corners, in the other

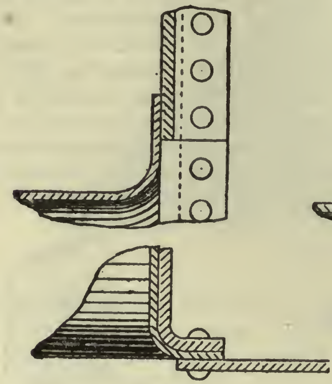


Fig. 315

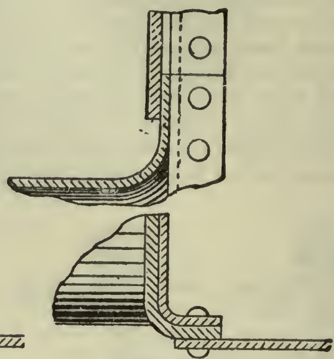


Fig. 316

the furnace saddle corners, will have to be drawn out or chipped taper. As these corners have often given trouble by leaking, some works have adopted the plan of welding them. They ought to be annealed.

Before concluding the remarks on this subject it is necessary to mention an isolated practice of flanging the shell plates, instead of the boiler end plate (fig. 317). There does not seem to be any advantage, except that the upper end plates of boilers need not be annealed. The shell plate being in tension, any injury caused to these parts by the flanging operation would be doubly dangerous, as it cannot be removed by annealing.

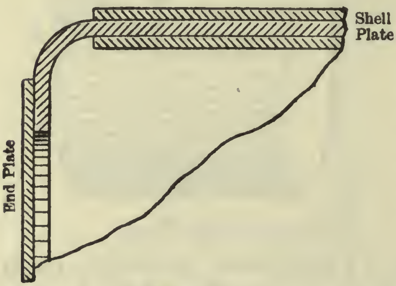


Fig. 317

The work is carried out as follows: The ends of the longitudinal seams are welded, the seams riveted, and the edge of the shell heated and flanged in a Tweddell's flanging press, which, to suit the requirements of the case, has to be placed on its back.

Another interesting subject is cold flanging. The results of Messrs. Easton and Anderson's experience will be found in the 'Journal of the Iron and Steel Institute' for 1882, p. 528. They produced annular discs of the shape shown in fig. 318 with plates $\frac{5}{16}$ in. thick, but the results cannot be said to have been satisfactory, for out of sixty plates, Landore SS quality, nine cracked their inner flanges, and one its outer flange; five were not annealed, and of these three cracked. Out of fourteen which had been twice annealed two cracked. The forty-one remaining were annealed only once, but some of them as long as forty-eight hours and in ashes. Five of these cracked, although one-half of the lot were flanged slowly, requiring about three to four minutes instead of a quarter of a minute.

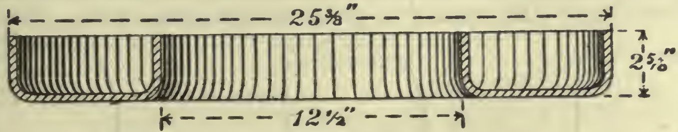


FIG. 318

No mention is made as to whether any of the plates cracked later on, though it is to be expected, for nearly all bent test pieces crack at their inner radius some time after leaving the press or hammer.

Fitting Together of Plates.—Having completed the flanging and annealing, the plates are fitted together and secured to each other by means of a few bolts. Whenever possible, the various seams are closed up cold by hammering. Troublesome parts are warmed by heaters, but, as they often reach a blue heat, it would be better to heat them properly. On this point practices differ. Some works prefer heating only one plate, bolting it to the other, and hammering it, so as to fit the cold one. This plan requires very great care in the flanging of the plate which remains cold, as it cannot be made to alter its shape, and the danger exists that it will be made blue hot. Other works heat both plates while bolted together; then, of course, it is easy to correct any slight defects of form in both. Hardly any works re-anneal these pieces, which may account for some cracks. Less risk would be run if the corners at least were reheated.

Particular care should be taken with the flanged corners of the furnaces where they meet the back tube plate and the combustion chamber sides. Not only will injudicious treatment increase the liability of these parts to crack after the boiler has been put in use, but also, on account of the impossibility or difficulty of caulking this seam at both edges, there is a greater chance of the water forcing its way through here than through any other seam of the boiler (see fig. 413, p. 296). Bad plating very generally leads to troublesome leakages.

Fitting Combustion Chamber Plates.—The fitting together of the various internal parts is done as follows: The tube plates are bolted to the furnaces and hammered up close, the rivet holes drilled, and the riveting carried out at once or postponed till the other parts have been prepared. The combustion chamber back plate is then bolted to the furnace and tube plate by means of strips of iron. The sides, top and bottom plates are then successively fitted. The order in which this is done depends on the position of the seams. Representations of the various plans will be found in figs. 319–327. If the whole of the riveting is to be done by hydraulic power, the arrangement shown in fig. 325 must be adopted, the top plate being riveted before the sides are put in position; and these again are riveted before the bottom plate is secured. But this can only be riveted if the depth of the riveter is sufficiently long to reach from the furnace mouth to the back plate. If the head of the machine is a clumsy one, the flanges will have to be made sufficiently deep so that the rivets can be reached. By removing the back plate after all the side plates have been fitted it is possible to rivet at least the tube plate flange by machinery. If the riveting of all the combustion chamber seams is to be done by hand, it is immaterial in which order the plates are put on, and preference will naturally be given to arrangements like those in

Fig. 319

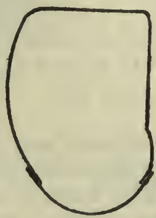


Fig. 320

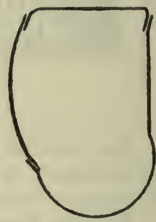


Fig. 321

figs. 319, 320. When the two plates are of equal thickness the seams can be placed higher up, as in fig. 321. Should the lengths of the plates be found too great, or, what is more likely, should the back combustion chamber plate not be shaped exactly like the back tube plate and

Fig. 322

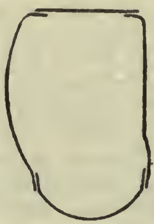


Fig. 323

Fig. 324

furnace, which would lead to trouble in fitting the circumferential ones in one or two lengths, it may be safer to use three plates, as in figs. 321, 322, 323, or even four plates (figs. 324, 325, 326, 327). The arrangements shown in figs. 319, 320, 321, 322, 324, 325 require that

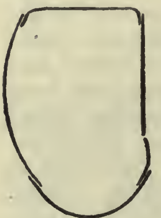


Fig. 325

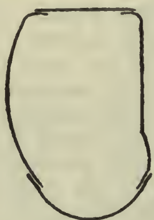


Fig. 326

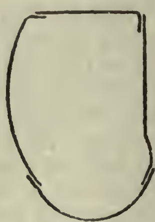


Fig. 327

one or the other of the plates should be bent at two points, which, for fitting, is more troublesome than if each plate has got to be bent at only one point, as in fig. 327. The arrangement shown in fig. 322, and on a large scale in fig. 328, doubles the bending operations. The latter figure also shows how difficult it would be to caulk the internal edge of such a seam if placed higher up in the curve, and also that it should never be placed near a stay. (See also fig. 329.) The corners of all these plates ought to be drawn out as shown in figs. 232 and 233 (p. 243). The various seams should also be slightly bevelled, and it is as well to do this before fitting them together.

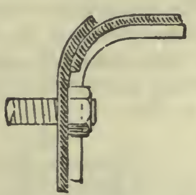


FIG. 328

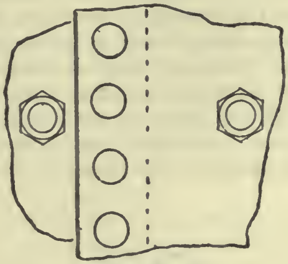


FIG. 329

The combustion chamber bottoms and sides are bent cold to their various curves by passing them through the bending rolls, but the corners, particularly where they have to be drawn out, are heated. It is difficult to obtain quite the correct shape at once, and the final setting has to be done in place. Heaters are frequently used, but whether it is that a blue heat is never reached, or whether the work done to the plates at this temperature is not sufficiently severe, it certainly does not as yet seem to have led to any failures in the shop or subsequently.

Drilling Combustion Chamber Plates.—When the various plates have been properly fitted together, their seams are drilled in place—generally by hand, but also by machinery. In some works all the outer plates are first removed and drilled by machinery, or even punched, and then refitted, and the inner plates drilled. Sometimes the outer plates are perforated before fitting. If punched, there is always the danger that the plates may crack while bending them, or if punched after bending they are liable to warp, and part of the fitting work has to be done over again. Another plan is to drill all the flanges before fitting, to mark off the holes from the inside on the outer plates, and to drill these by machinery. In any case, all holes in the flanges will have to be countersunk on the inside, for it is difficult to caulk any other heads when so near to a corner. It is also customary, but not necessary, to countersink all rivet holes on the fire side of the various lap joints; but it does not appear that projecting heads burn off. Wherever there is sufficient space for riveting, or when this is done by machinery, the rivets should be inserted from the water side of the plates.

The angle of the countersink varies from 15° to 45°. The smaller the angle, the smaller is the power of the rivet to draw the plates together while it is cooling. It vanishes altogether when the apex of the cone lies beyond the flat base of the head, or if both ends are countersunk this limit is reached when both cones touch each other at their apexes.

Riveting Combustion Chambers.—The remarks made while discussing the riveting of shell plates, &c., apply to a certain extent to this case. The very greatest care should be taken to ensure perfect contact of the plates at the saddle and its corners, for this is almost the hottest part of the boiler, and any air-spaces are sure, sooner or later, to lead to troubles which rapidly extend. On account of the difficulty of flanging the furnace ends, these parts are not always as flat as the tube plate to which they are connected, and carelessness in riveting easily leads to the condition of things illustrated in fig. 305, p. 270. Careless work at these seams is readily exposed by removing a rivet or two. In corners the rivets may be arranged as in fig. 330 or fig. 331, the latter of the two plans being the more generally adopted. In some works the corner rivets are replaced by screws beaded over at either end. For remarks on caulking see p. 293.

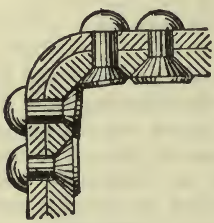


Fig. 330

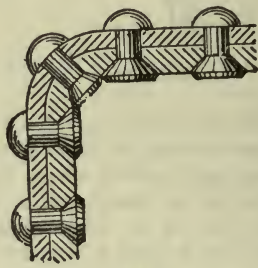


Fig. 331

Planing Edges of Flanges.—All the edges of the plates ought first to have been machined. With the circumferential plates this is or can be done before bending, and in that case they can be bevelled; the flanges, unless they are trimmed with milling tools, are left with square edges. In some works they are subsequently chipped, while in others they are only chipped, and, if little material has to be removed, this is done after the seams have been riveted. These remarks apply as well to the flanged end plates as to those of the internal parts. If much material has to be removed, this can be done by the shearing or the punching machine, usually before the plates are fitted; or the superfluous parts are removed by a cross-cut chisel. The furnace saddle seams are often shaped by slotting machines or band saws.

Planing machine lathes with very large face plates, or turntables, are also used for this work. The cutting tools or knives are made very wide, sometimes more than 12 ins., and it is then possible to deal with all sorts of irregular-shaped flanges. Of course, as there is always only one thickness of plate, only part of the width of the tool is used at a time, and the feed is a pretty heavy one. The cutting speed is about 10 ft. per minute, and the depth of a cut about $\frac{1}{32}$ in. Circular saws, which cut off a solid piece of the flange, or milling cutters, have also been tried for removing the superfluous material, but are not generally used. They travel at the rate of about 2 ft. per hour. See 'Enging.,' 1909, vol. lxxxviii. p. 212. The furnace front plate flanges can be machined by the same machine (fig. 212, p. 235) which cuts the holes, but a tool with a very wide cutting edge replaces the parting tool shown there.

Fitting Internal Parts into Shell.—The furnaces and combustion chambers having been riveted together, each one has to be fitted into the furnace front plate. If all the furnaces lead into one combustion chamber, their back ends should not be riveted up without the furnace front plate being in position, otherwise the most serious inconveniences will be encountered when trying to put the two together.

Where each combustion chamber has a separate furnace, these are fitted into the flanged front plate and their holes drilled and riveted, while the stays from one combustion chamber side to the other are screwed into place. Then, when this work has been completed the whole of the combustion chambers, furnaces, and furnace front plate are rigidly connected, and may be lifted into position in the boiler shell, to which the back plate has already been riveted.

If the front tube plate is to be placed inside of the furnace front plate, a little simple manœuvring may be necessary.

In some works the screwing together of the combustion chambers and the riveting of the furnace front seams are carried out inside the boiler shell after the furnace front plate has been riveted up. No advantage is gained, and on account of the confined space the work will be both slow and bad. Besides, if one of the furnaces should crack while its front end is being expanded, all the work just mentioned would have to be done over again.

Fitting Furnaces.—On account of the difficulty of making a furnace front plate with, say, three or four holes, slip easily on to as many

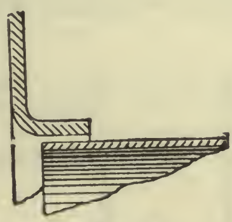


Fig. 332

Fig. 333

furnaces, these are generally made at least $\frac{1}{4}$ in. smaller in diameter than their respective holes. Sometimes there will be even $\frac{1}{2}$ in. of difference. Then, if the flange is left square, as shown in fig. 332, the expanded furnace mouth will only bear at the front edge, as shown in fig. 333. It is, therefore, customary to leave the flange slightly conical, as shown in fig. 334. In order to get a good fit, as in fig. 335, the diameter of the front edge of the furnace will occasionally have to be increased even as much as $\frac{1}{2}$ in. It is dangerous, and in cold weather impossible, to do this expanding without heaters; and when they are used there is the further danger of overheating and of making the plates permanently brittle by hammering them when blue hot, and cracks at these seams are not unknown.

The work is carried out as follows: The furnaces are secured as centrally as possible in their respective holes, and all the holes drilled in place either by hand or by the machine (fig. 210, p. 234). Numerous bolts are then inserted, heaters applied, and the bolts screwed tighter, while the front seam is being hammered, until the plates are in perfect contact. The troubles which these seams sometimes give have led a few engineers to make them treble-riveted, but an apparently safer and more convenient plan would be to make the flanges conical in the

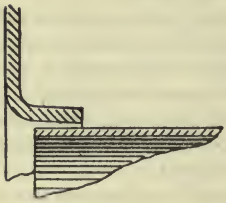


Fig. 334

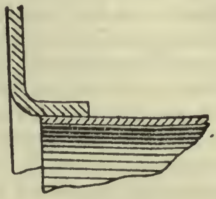


Fig. 335

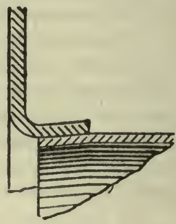


Fig. 336

other direction (fig. 336), and either turn the front end of the furnace or give it a slight bevel, as shown. Instead of heaters, portable coke furnaces are sometimes used, which heat the seams almost to redness, and they can be screwed close without hammering.

The Riveting of Furnace Front Seams is not quite so simple as would at first sight appear, particularly if the water-spaces are made narrow, and there is always a very strong inclination to do this in order to gain space. With this object in view the flanges of the furnace holes are sometimes turned the other way to those of the circumferential one (fig. 244, p. 247 ; fig. 278, p. 264), or the furnace is made taper

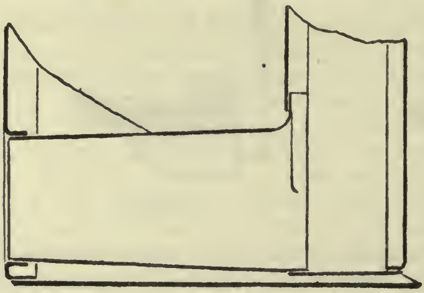


Fig. 337

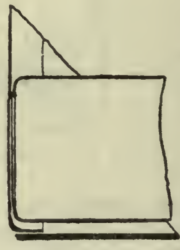


Fig. 338

(fig. 337) or its front end contracted a few inches, or the furnace is fitted at an angle. Carried to extremes this principle will be recognised in the arrangement shown in fig. 338, which would permit of the furnaces touching each other. When, in addition, the longitudinal seams both of the furnaces and the shell accidentally come together (see fig. 339), the boiler bottom is practically cut off from the upper part, and the consequent want of circulation may make itself seriously felt. Figs. 340, 341, 342 show an arrangement which also permits of the furnaces being placed very near each other and near the shell, except the central one, because here both flanges are turned in the same direction. (See also figs. 84–87, p. 42.)

In cases where two flanges are so close together that a rivet can neither be introduced nor subsequently caulked, the holes have to be carefully drilled, tapped, and countersunk, and accurately-fitting screws, with conical heads, inserted. The ends of the screws are

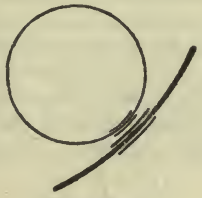


Fig. 339

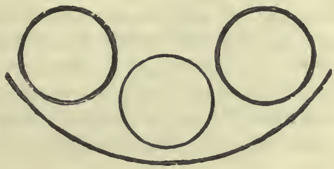


Fig. 340

beaded over, and the heads caulked. An arrangement in which all this difficulty is overcome, and in which the front plate holes need not be flanged, is shown in fig. 343. The furnace front flanges will overlap each other. (See also fig. 41, p. 21.)

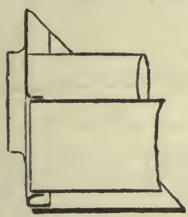


Fig. 341

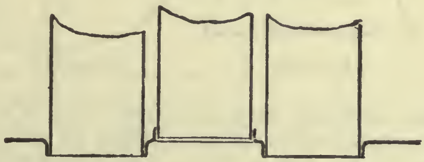


Fig. 342

Furnace Saddles.—The various designs for securing the furnaces to the combustion chambers are shown in the following sketches Fig. 344 is the most common, except, perhaps, when the back end seam is single riveted all the way round, instead of being double

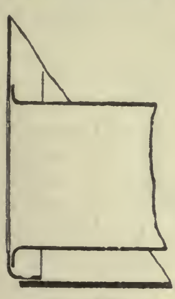


Fig. 343

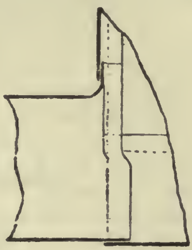


Fig. 344

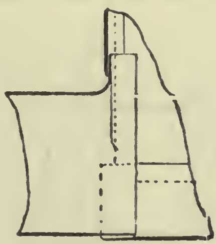


Fig. 345

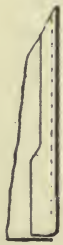


Fig. 346

riveted from below the line of fire bars, as shown. In through combustion chambers of double-ended boilers this seam is sometimes double riveted above and treble riveted below (see fig. 345). The increased width of the lower flange is sometimes met with in the back plate of combustion chambers (see fig. 346). In both the above cases the tube plate is on the water side of the saddle plate, and it is affirmed that unless the edge of the plate is well bevelled, and the rivet heads countersunk on the water side, steam-bubbles will lodge there. It is difficult to caulk this seam on the inside because of the furnace, and on the outside because of the tubes. The curvature of the tube plate side flanges is a very sharp one, in order to get the tubes as near the edge as possible, and when the furnace saddle flange is inside of this tube plate this curvature is often so sharp as to be more like a corner, and therefore very liable to crack (see fig. 43, p. 21). One way out of this difficulty is to increase the radius of curvature of the tube plate side flanges where they meet the saddle corners. This part

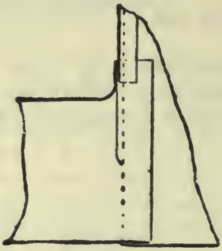


Fig. 347

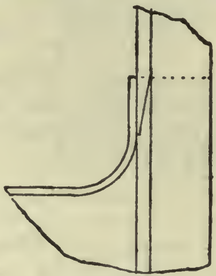


Fig. 348

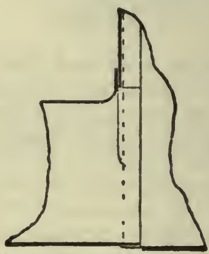


Fig. 349

must therefore be made deeper, and it is necessary to cut away the combustion chamber side plates, as shown either in black or dotted lines in fig. 304, p. 270. Another arrangement is shown in fig. 347 (see fig. 60, p. 35). The saddle being flanged over the tube plate, it can be made with a gentler curve than in figs. 344 and 345. The saddle seam is sometimes double riveted, and believing that the large amount of metal at this point would lead to burning, some engineers plane or chip this seam (see fig. 348). When it is necessary to fit very thick plates to the combustion chamber bottoms, they are sometimes planed thin at the seam before bending, as shown in fig. 349, but no object seems to have been gained by adopting this plan.

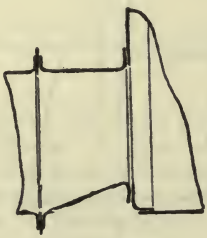


Fig. 350

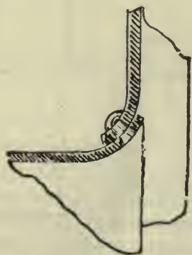


Fig. 351

It is objected that the saddle seams of the last three constructions expose a caulked edge to the impinging action of the flame, but in practice they answer well. It is certainly far easier to caulk both edges of this seam, and there is no projection under which steam-bubbles could find a lodgment. A similar seam is shown in fig. 350, which is adopted with ribbed furnaces and with those fitted with several Adamson's rings. The tube plate hole is not flanged, and can be carried to the very bottom of the combustion chamber.

The arrangement shown in fig. 351 is adopted when there is little spare space between the furnace top and the tubes, but as the work of flanging is doubled, and the caulking difficult, it cannot be recommended.

A somewhat more objectionable arrangement is shown in fig. 352, the tube plate being flanged to meet the furnace, and the seam being

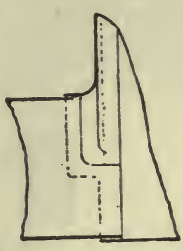


Fig. 352

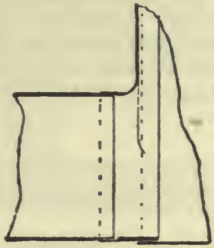


Fig. 353

exposed both to the radiant heat of the incandescent fuel and to the convection of the hot gases and flame.

The very greatest care has to be taken to secure metallic contact of the plates, and the flanged tube plate is sometimes bored to fit the furnace which has been turned (see fig. 353). In order to minimise the chance of leakage, and to reduce the labour of caulking, it is a good plan to remove the scale of all plates by pickling them in a 1 per cent. solution of hydrochloric acid, or to sponge the seams with sal-ammoniac. Other remarks about riveting these seams will be found on p. 34.

The back end seams are sometimes welded (see figs. 424, 425, p. 299), and an excellent job can be made of them, but in case of collapse they are apt to crack.

Riveting Saddle Seams.—While being riveted, seams invariably stretch, more particularly if the holes are countersunk, and generally the two plates do not stretch equally, and the result in saddle seams is either that the plate with the outer flange grows longer than the inner one, whereby the previously well-fitted corner opens, or else the inner plate stretches more than the outer one, causing the flange of the latter to crack. Therefore, under no circumstances should riveting be commenced at one side and carried across; it is also not well to commence in the centre and work to both sides. The best plan is to commence at both sides and work towards the middle, alternately riveting one side and then the other. If the reheating of corners without subsequent annealing were not objectionable it would be well if the saddle corners could be closed up hot after riveting (see p. 266).

The Screwed Stays between the combustion chamber backs or sides and the back end or shell of the boiler are sometimes fitted before the circumferential seam of the front plate is riveted, sometimes afterwards. This being done by hand, all the holes will have been previously drilled or punched, and the plates annealed; they are tapped when in position, and the stays screwed and beaded over, or caulked and nutted. If the tapping and screwing of stays are done by machinery, the drilling is also done by the same machine (see p. 233). The sawing off of the ends should not be necessary, for accurate measurements of the lengths could have been taken, and it is certainly cheaper to saw off the correct lengths in a small machine than in one of these very expensive ones. It takes about 10 minutes to tap a pair of holes and to screw in the stay, and another 20 minutes to caulk and to screw up the nuts. When done by hand, the whole operation takes about one hour.

When screwed in by hand, these stays are often made with a square head (fig. 354). The labour of cutting off the ends would be

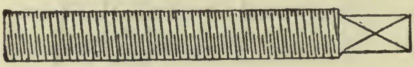


Fig. 354

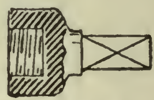


Fig. 355

saved by the use of a closed nut with a shank (figs. 355, 356). Its depth should not exceed the thickness of a stay nut. Stay holes are always tapped when the plates have been placed in their final position, and immediately before the stays are fitted; otherwise it would be impossible to enter them. It does not seem to matter much that the threads of the stays and of the tap are not exactly alike. This difference is due both to the tap and the stays altering their length respectively during hardening and during screwing.

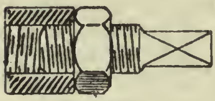


Fig. 356

The stays should be a tight fit, particularly at their inner end. This is not always the case, even in some of the best boiler-shops, and often leads to leakage with the least overheating. As the nutting of loose stays is difficult, they are at once scragged with a caulking tool, and inspection does not reveal the slackness.

The Pitch of Threads of the stays varies in different shops, the natural tendency being to make the pitch as fine as possible, because the effective diameter is measured from the bottom of the thread; but it takes longer to tap and screw them. There is little harm in fine threads when nuts are screwed on the ends, but a coarse and deep thread is necessary when the ends are only beaded over.

Cases in which explosions or mishaps have been traced to the use of fine threads are to be found in 'Engineering,' 1887, vol. xliii. p. 396, and 1890, vol. l. p. 85.

Should the plate bulge, as it often does when hot, its inner (water) side will leave the screw threads entirely (fig. 357), and only the outer edge will hang on to the stay by the small riveted head and by a very few threads (fig. 357); whereas with a coarse thread, as in fig. 358, it would require a serious amount of bulging before any of the threads are quite clear of each other.

The threads of the taps and stays are never quite the same, which, when fine ones are used, may cause them to strip.

Stay Nuts ought not to be thicker, or only a very little thicker, than the plates which they have to support: not because of the danger of

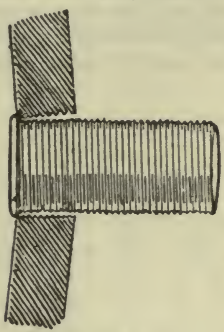


FIG. 357

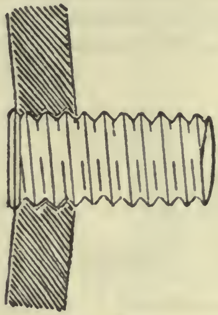


FIG. 358

burning them, but for fear of stripping the thread in the plate. This is illustrated in fig. 359, where, if the full power which such a nut could stand were applied, the threads in the plate would certainly strip. The drawing office habit of showing the stays horizontal instead of normal to the combustion chamber plate, and the carelessness with which the holes are drilled, placing the stays at an angle, and making the nuts bear on one side only, add to their power of doing harm. The taper washers which are then fitted often make matters worse by turning round, the thick part being found where the thin part ought to be. To prevent this, the bearing side of the nut should be faced. The

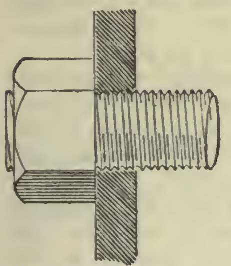


FIG. 359

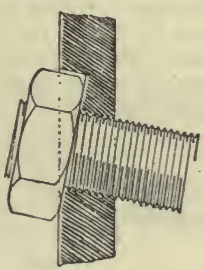


FIG. 360

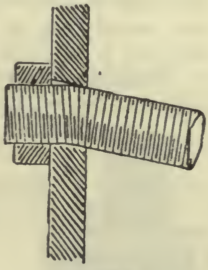


FIG. 361

thick external plates may be recessed, as shown in fig. 360. This is easily done with a rose disc of the size of the nut: it is slipped over the stay, the nut screwed down on it, and it is then turned with a spanner, its outer edge being formed square or hexagonal. In many works the stays are bent by striking long steel nuts temporarily screwed on with a hammer (fig. 361). Thick stays are not likely to bend; their fit in the plate will be spoilt and their threads damaged.

One often finds red-lead cement, and also flat washers, under the nuts in the combustion chambers. Neither are wanted, doing more harm than good. Nobody would contend that the cement could stop any leakage, the circumference of the stay having been caulked ; it can, therefore, only act as a non-conducting layer between the boiler plate and the nut, causing the latter to burn when exposed to the flame. The interposition of a washer doubles this danger. The cement is also sure to get into the threads between the nut and the stay, where it acts as an efficient non-conductor and prevents the nut being cooled by the stay. Fig. 362 shows stay bolts which are used by some builders.

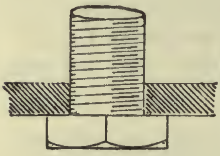


Fig. 362

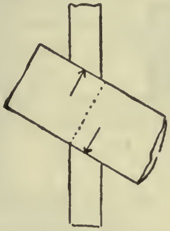


Fig. 363

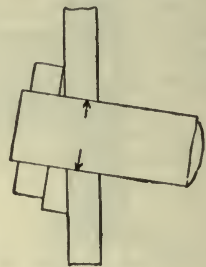


Fig. 364

An ideally perfect stay should allow of the nut in the combustion chamber being brought into as absolute a metallic contact with both stay and plate as is possible : it will thereby be prevented from burning, and will give the most solid and efficient support to the perforated plate. Taper washers ought therefore to be dispensed with as much as possible on the combustion chamber ends of the stays, and in no case should the angle be so large as shown in fig. 363 ; in fact, the angle should not exceed $\frac{1}{3}\,\frac{t}{d}$, where t is the thickness of plate and d the effective diameter of the stay. With a $\frac{1}{2}$-in. plate and a $1\frac{1}{8}$-in. stay this would give an angle of $\frac{1}{6}$, or 10° (see fig. 364). Near flanged seams it is difficult to fit nuts ; then riveting should be allowed.

Caulking Screwed Stays.—Screwed stays are caulked with an ordinary tool before the nuts are fitted (fig. 365). Locomotive engineers, who do not seem to like nutted stays, have used hollow bars and made them tight by drifting. Mr. Yarrow has also adopted this plan, but on account of the impossibility of getting at the outside ends, he makes the hole larger at the inner end than at the other, and uses both drifts from the fire side ('N. A.,' 1891, vol. xxxii. p. 102, plate 22). The drift used should be threaded so that by screwing on a nut it can be drawn out again.

Fig. 365

Girders.—These are fitted to combustion chamber tops and other flat parts in the inside of boilers, transmitting the pressure from these parts to others which are better adapted to support it. If the girders end near stays, these will have to be made of a larger sectional area, in order to support the extra load which is thrown upon them.

Most girders are made of two plates, one on either side of the stays which they have to support (figs. 366, 367, 368, 369, 370). These are connected to each other by rivets and distance pieces, and caps are fitted under the nuts of the stays.

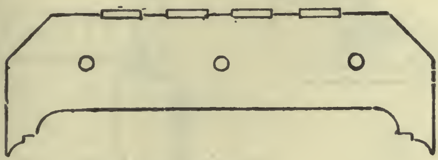


Fig. 366

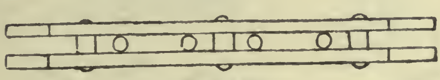


Fig. 367

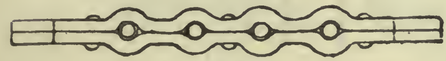


Fig. 368

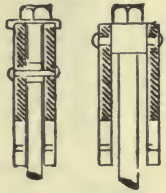


Fig. 369. Fig. 370

Instead of riveting the plates together they are sometimes welded at their ends. Cast-iron feet at the ends are also used.

Sometimes they are forged or slotted out of the solid (fig. 371) or are made of cast steel (fig. 372).

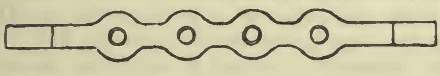


Fig. 371

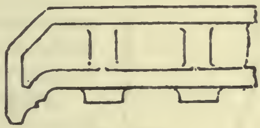


Fig. 372

In some works the girders are arranged alternately with an odd and an even number of stays, the pitch being measured diagonally. The saving in weight, however, is too slight to balance other inconveniences.

Angle Iron and Web Girders.—Instead of girders with stays, angle irons (fig. 373), and even beams (fig. 374), are sometimes riveted

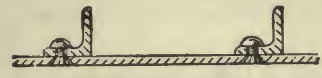


Fig. 373

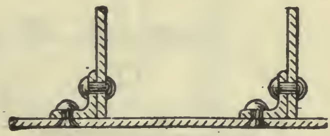


Fig. 374

to the combustion chamber tops, but this is a bad practice, as the plates generally crack under the doubled parts, in the same way as they do under palm stays (see fig. 37, p. 20).

Another plan, and one which is coming into more general use, is to flange the combustion chamber plates, and rivet them to vertical webs (figs. 375 and 376). The ends of these vertical plates have to be forged with projecting feet (fig. 384, p. 289), which tuck in under the

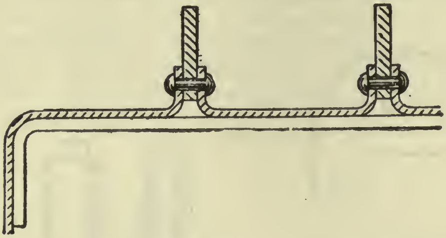


FIG. 375

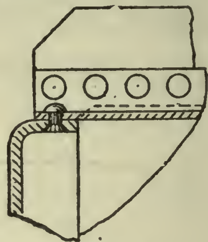


FIG. 376

FIG. 377

combustion chamber top plate flanges. It is not difficult either to rivet these seams or to caulk them from the fire side.

FIG. 378

Some other shapes of girders are mentioned by D. S. Smart ('C. E.,' 1884, vol. lxxx. p. 132).

Fig. 377 shows a corrugated combustion chamber top which is occasionally fitted; its strength can be calculated with the help of a formula on p. 200. Occasionally the tops are rounded, as in fig. 378.

Suspended Girders.—Instead of allowing the girders to rest entirely on the corners of the combustion chambers, they can be suspended by stays to the boiler shells (figs. 379, 380), or their ends need not rest on the tube plate at all; but in either case the end stays should be kept far away from the flanges, in order that the expansion of the shell diameter, which amounts to about $\frac{1}{16}$ in.,

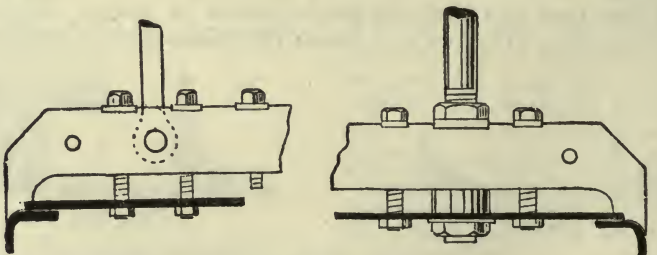


FIG 379

should not open their seams. The top ends of the suspending rods are attached to double angle irons, riveted to the shell. The top ends are flattened out as in fig. 381, so that they may be secured by several rivets, but generally they end in an eye and a bolt, with a split key passing through their projecting end (fig. 382). Sometimes these stays are tapped both into the combustion chamber and shell plate.

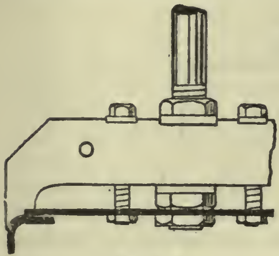


Fig. 380

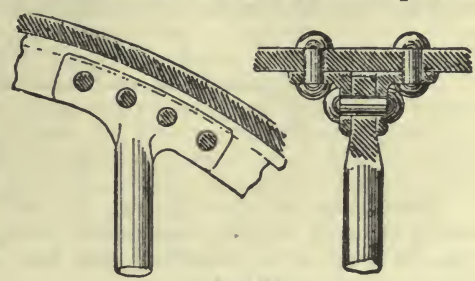


Fig. 381

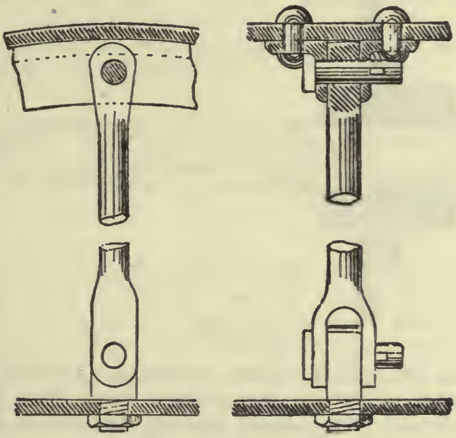


Fig. 382

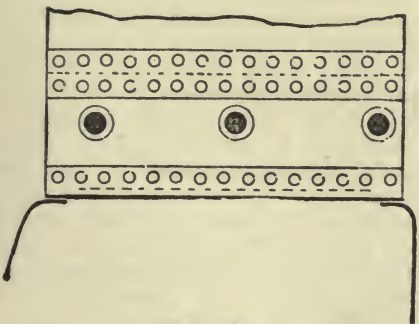


Fig. 383

Fig. 384

Vertical Stays.—It also happens that girders are dispensed with entirely, and every one of the combustion chamber top stays is carried up to the shell. This arrangement is carried out as shown in fig. 382.

The reason for fitting these stays to the shell is, in most cases, to relieve the tube plate of its load. But this relief may be carried to excess, and may lead to the stretching of tube plate holes if the stays are fixed near the flanges. (See p. 34.)

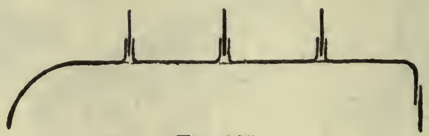


FIG. 385

Plate Stays are shown in figs. 383, 385; they are carried to the shell plate (figs. 383, 385) and riveted to angle irons. Fig. 384 shows how the two ends of the vertical plate have to be stumped up so as to fill the roundings at the end.

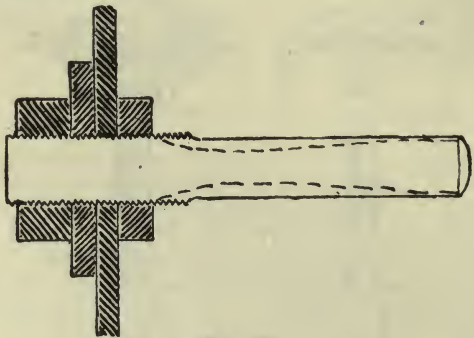


FIG. 386

These plates are usually fitted crossways (fig. 383). Holes are provided, through which the steam-space stays have to pass. For very deep combustion chambers, the top has to be supported by

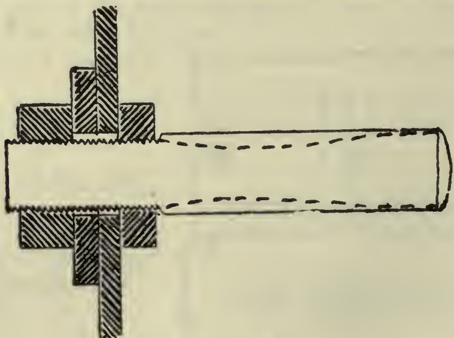


FIG. 387

several webs, but their number depends on the thickness of the plates; see also s.s. 'Iberia,' 'Enging.,' vol. lvi. p. 207.

The Steam-Space Stays are sometimes tapped into the plates (fig. 386), and sometimes fitted with nuts on either side of each plate (fig. 387). Both arrangements have their disadvantages. In the one case the stays have to be swelled at one or both of their ends and threaded, then screwed into the plates, caulked both inside and out, and nutted. After being in use for some time the diameter at the front end will be seriously reduced by corrosion, as indicated by the dotted lines (fig. 386), and the stays must be renewed. This is not the case with those stays which have been threaded without swelling, for the exposed part of the stay may be very seriously reduced before making it weaker than the screwed part; in this case the thread at the front ends should not extend beyond the inside nut. The objection previously urged against fine threads does not apply to nutted stays (see p. 285).

The trouble with these stays is that they leak, for it is only grummets and washers that can be used to prevent this. The hollow space is sometimes filled with sheet iron and caulked, but better results seem to be obtained with asbestos packing. The outside washers for these stays are very often riveted to the plate, giving it a better support. In confined spaces it may be necessary to fit the stays in two lengths. This is shown in figs. 388 and 389.

Fig. 388

Fig. 389

Boiler Tubes.—One of the last things to be done to a boiler is to fit the tubes. This is a simple matter: the tubes arrive cut to the right lengths and probably also annealed at their ends; they are passed through the two tube plates, and their ends are expanded. Formerly this was done by means of conical drifts, and some people still advocate their use, but the common practice is to use expanders. These consist of several—usually three—small rollers, partly projecting out of the circumference of an iron case. A taper mandril is place in the centre and driven in while being turned round, thereby causing the rollers and their case to revolve, at the same time exerting a pressure on the tube, which expands and gets firmly bedded against the metal of the tube plate. Plain tubes can be placed in position and expanded at the rate of about six per hour.

The past disasters with the tubes of Navy boilers, leading as they did to the appointment of a special committee of inquiry, are a sufficient proof that the mode of securing them is not a perfect one. Undoubtedly the severe conditions of Navy trials search out any defects which may exist, and when leakage has once commenced it seems impossible to stop it again until the tubes are re-expanded. By some it is affirmed that the trouble is caused by overheating the tube plate (see p. 28). Others believe that the chilling effect of inrushes of cold air, or that structural peculiarities, are to blame. Thus each tube has a slight twist, due to the expander having been worked in one direction only. On heating the tubes, they may untwist slightly. This motion will be particularly injurious if the tube plate holes are not perfectly circular. The taper shape of the tube expander which is used to force a parallel tube into a parallel hole may also be a cause.

Of the various attempted remedies, none have yet given permanently satisfactory results, and for the merchant service, where practically no troubles are experienced, no deviation from the present course would be advisable. It consists in fitting a sufficient number of stay tubes.

Stay Tubes.—These are fitted before the others, in order to be able to screw up the nuts. If the tubes are left parallel, the back end requires to be screwed sufficiently long so that the inside nut of the front end can be inserted, as shown in fig. 390. The back end is

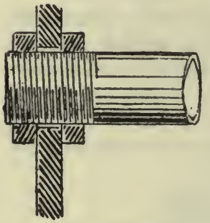


Fig. 390

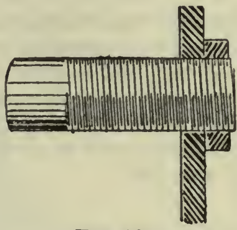


Fig. 391

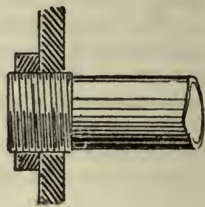


Fig. 392

beaded over, as in fig. 393, or nutted, as in figs 391, 394. Nowadays nearly all stay tubes are swelled at their front end (fig. 392), and both plates are tapped together. Fig. 395 shows a section of a tool which is very convenient for screwing or unscrewing stay tubes. Two half-round grooves, deeper than shown, are cut into the sides of a spindle which is just large enough to enter the tube, and two short lengths of steel of a lenticular section are placed into these recesses and held there by any convenient means. Their outer edges, being roughened,

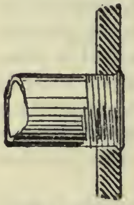


Fig. 393

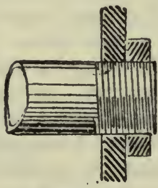


Fig. 394

Fig. 395

grip the inside of the tube whichever way they are turned. Both tube ends are expanded and also caulked, and either beaded over or fitted with nuts. The other tubes are then placed in position and expanded. Stay tubes are insisted on in the merchant service, while the locomotives and American steamers do without them, but all the ends are beaded. Exhaustive experiments on the holding power of tubes are mentioned by W. H. Shock, 1880, p. 217. As regards taper-ended tubes experiments were carried out by Martens ('Mitt., Berlin,' 1887, vol. v. p. 65). None of these results were obtained at steaming temperatures.

The time required to tap the tube plate, fit a stay tube, and bead or nut it, is about one hour and a half. The taps are hollow, and can be adjusted on a long spindle to suit any length of tubes; these are threaded in a lathe.

Caulking.—In order to explain what takes place during the duration of a blow, when the hammer, the caulking tool, and the plate are in contact, it will be necessary to divide the plate into layers of, say, $\frac{1}{100}$ in. in thickness, as shown in fig. 396. The velocity of the hammer and caulking tool is imparted to the first layer, and quickly transmitted to the next, and further on. The pressure which is

FIG. 396

required to transmit this velocity from the caulking tool to the first layer, and then from one layer to another, is at first in excess of the elastic limit, and produces a permanent deformation, as shown. But this pressure reacts on the caulking tool, causing it to rebound after only a few layers have acquired the high velocity; but, their mass being small as compared with that of the hammer, their pressure on the further layers is not sufficiently great to flatten and spread them out. The permanent effect will, therefore, be that which is illustrated in fig. 396. A little reflection will show that, the lighter the hammer, the sooner it rebounds, and the fewer the layers acted upon, and the

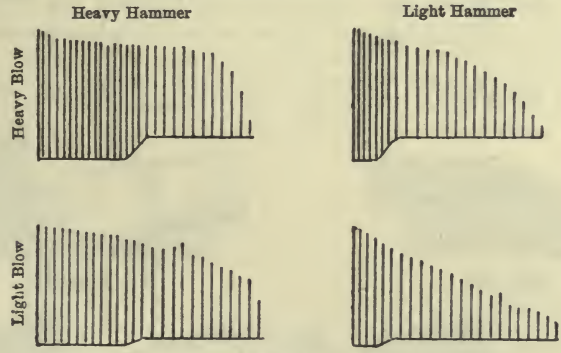


FIG. 397

lighter the blow, the slower the imparted velocity, and the smaller the deformations. Thus, if twenty-five layers equal to $\frac{1}{4}$ in. are deformed by the blow of a 7-lb. hammer on a 2-lb. caulking tool, only half that number, equal to $\frac{1}{8}$ in., would be spread out when using a hammer weighing $3\frac{1}{2}$ lbs. and a 1-lb. tool. By increasing the velocity, the force between the caulking tool and plate would be increased, and the swelling would be greater, but the distance to which this swelling extends would not be altered—at least, not materially. In an exaggerated form the effects would, therefore, be as shown in fig. 397. For this reason heavy hammers are first used, and then light ones.

The next thing to be considered is the shape of the tool and the shape of the edge of the plate. In fig. 398 the edge of the plate is square, and the effect of the caulking tool is seen under it (fig. 399). The metal has simply been swelled up.

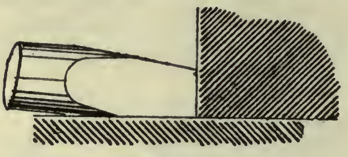


Fig. 398

Fig. 400

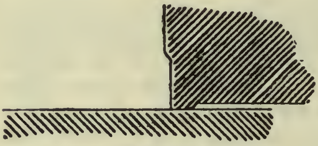


Fig. 399

Fig. 401

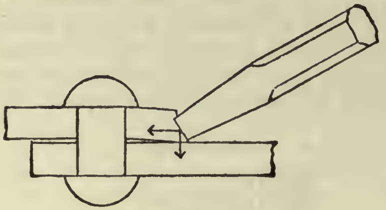


Fig. 402

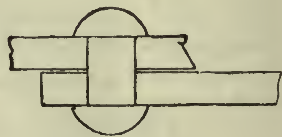


Fig. 403

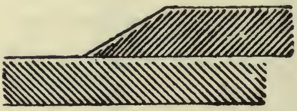


Fig. 404

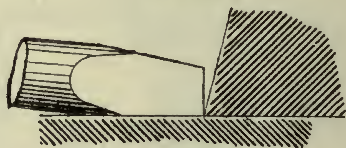


Fig. 405

Fig. 406

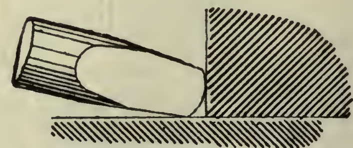


Fig. 407

In fig. 400 the edge of the plate is bevelled, and the caulking tool is placed firmly against it. The result of the blow is seen above (fig. 401). Not only is the edge of the plate swelled up, as in the previous case, but the lower plate is scraped up, forming a small ridge; and, thirdly, the blow being directed downward, both plates are depressed; but as there is more spring in the outer plate (lap), it will not suffer as much permanent deflection as the lower one, and the result will be that the edge remains slightly, but permanently, open. This view is further illustrated in figs. 402, 403. An exaggerated seam of this sort is shown in fig. 404, and it is quite clear that no amount of hammering on the slanting surface would caulk the joint effectively. The maximum angle met with in practice is 1 : 3.

A better result would be obtained by placing the tool as shown in fig. 405, but, on account of the thinness of the edge, the caulking would not be very deep (fig. 406). Another method is shown in figs.

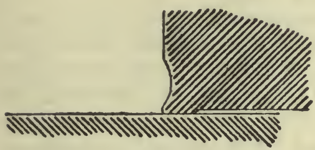


FIG. 408

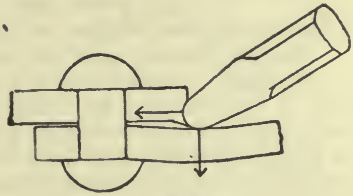


FIG. 409

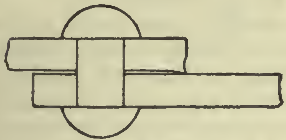


FIG. 410

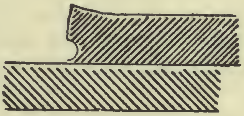


FIG. 411

407 and 408, and strongly recommended by locomotive engineers. The effect of a blow in this case is almost the very reverse of the previous one; for while the edge of the metal is being swelled, the plate under it is being struck down, and, as the hammer rebounds, the lower plate springs back and presses firmly against the caulked edge (see figs. 409, 410). A tool of this shape will cause the swelling to extend

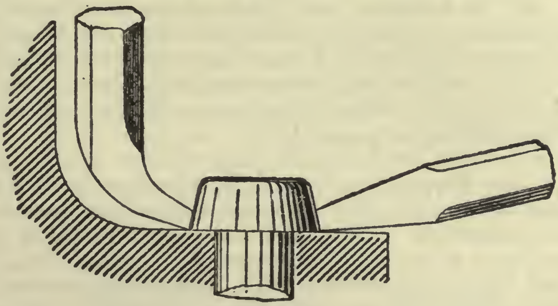


FIG. 412

farther into the plate, and thereby give a larger bearing surface. Care should be taken not to make this tool too small, otherwise it will act like a wedge, and press up the outer surface (fig. 411). Special tools have to be used for inside corners and for some rivets; they are generally shaped as shown in fig. 412. It is painful to handle them, and their work is never very satisfactory. The jar of the blow on all caulking tools is greatly reduced, if their ends are grooved, by striking them on a coarse file in a red-hot condition.

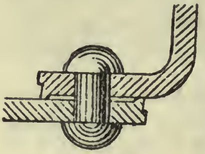


Fig. 413

There is no need for caulking the inside edges of seams if these are well bedded; but when the plates are not close, as shown in fig. 413, the inside edge should at least be fullered to prevent a rocking of the plates.

Most of the above remarks are also applicable to pneumatic caulking, but it must not be forgotten that as it is not limited like human power, much damage can be done on badly plated seams.

Electric Welding of riveted seams has been successfully carried out.

Pickling.—Riveted seams are said to be tighter if all black oxide scale is first removed from their surfaces, and the Admiralty practice with regard to tube ends is to grind them bright, so that they shall be in metallic contact with the tube plate. The pickling fluid which is used for removing the black scale consists of 1 per cent. of hydrochloric acid in water. Sulphuric acid, more than any other, has the effect of making steel brittle, particularly the hard qualities (see p. 146).

Welding Operations.—Reference has already been made to the fact that many of the seams of the internal parts can be welded, thereby saving the labour of flanging, drilling, riveting, and caulking. At one time efforts were made to weld iron shell plates, but the results were not encouraging, many seams having subsequently to be covered with straps. The introduction of steel, and the difficulty of welding it, stopped all progress. For a time it was only the furnaces which were welded; for, as these are not subjected to circumferential tension strains, the Board of Trade and 'Lloyd's Register' raised no objection. It is, however, well known that even now great difficulty is experienced in keeping the welded seams of some patent flues intact during the process of manufacture.

No doubt can be felt that better results are now obtained than formerly: they are doubtless due to improvements in the production of the milder qualities of steel (20 to 25 tons), which can be made almost absolutely free from sulphur and phosphorus, two of the most injurious impurities. The influence of various chemicals does not seem to be accurately known, but what information could be collected on this subject will be found on pp. 133-135.

The most reliable test for ascertaining whether steel is weldable is to cut off a strip about 18 ins. long, heat its centre, bend it and weld it, bend back the two ends (fig. 415), and then pull the sample asunder in the testing machine. The welded surfaces will probably be smooth and bright, except at the edges, where small patches of metal have left one side and stuck to the other. The larger these patches, and the greater their number, the more weldable is the steel. This test is not applicable to iron, as this metal tears through at the corners. To test it welded joints are re-heated and bent; if good, the weld should not open (see fig. 414). The ordinary tensile test is of little use when applied to welded samples, for, provided the joint is sufficiently taper, the strength will appear satisfactory. A far better test is to trepan small rings out of a welded plate, tap them, and tear them asunder by means of two screw plugs (fig. 416). Very serious defects have occasionally been exposed by this means.

Having obtained a good material, several precautions have to be taken to ensure a good weld. The heating should be carried to the right point. This knowledge can only be gained by practice. Steel, unlike iron, should not be heated to such an extent as to cause sparking.

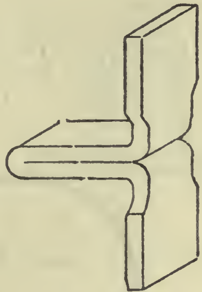


FIG. 414

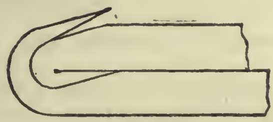


FIG. 415

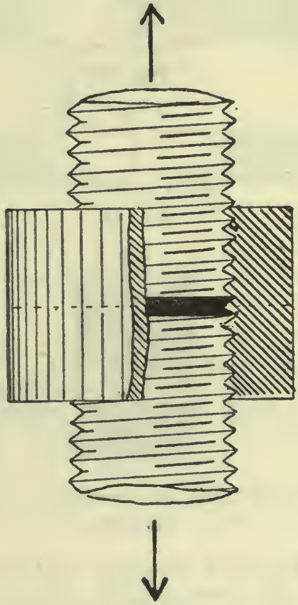


FIG. 416

Both sides of the plate should be exposed to the fire. If this is impossible the joint must be left wide open, to allow the flame to pass to the other side, where a cap of firebrick is placed. Having completed a short length of weld, and while heating the next, care should be taken to let the escaping flame pass over the recently welded part so as to keep it hot, otherwise the resultant irregular contraction will produce cracks.

The anvil on which the welding takes place should be as solid as possible. Any appreciable amount of spring affects the quality of the seam.

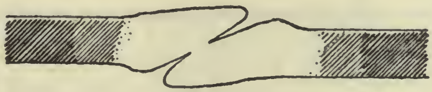


FIG. 417

Fluxes are used for iron, but seldom for steel. The edges of the plates, whether of steel or iron, are slightly tapered, as shown (fig. 417), care being taken to keep the surfaces convex, so that the centres touch first and the slag thereby gets driven out. Or the edges are formed into V's (fig. 418). They are first pressed firmly together, and then hammered. A very convenient method of welding steel plates together is to insert a separately-heated good weldable iron bar into the seam. Where practicable this piece is shaped like a double-headed rail (fig. 419).

FIG. 418

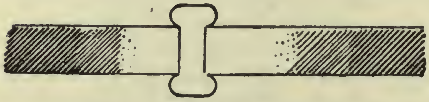


FIG. 419

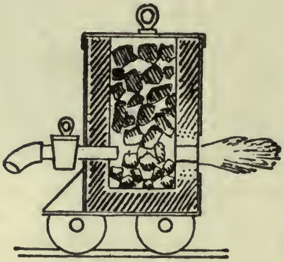


FIG. 420

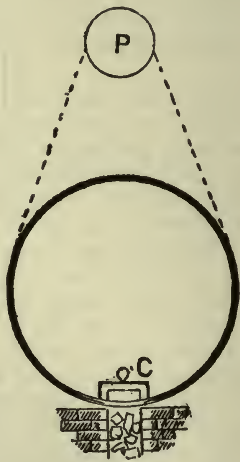


FIG. 421

Fires for Welding.—The heating of the plates is usually done over coke fires, using a blast. Much time is saved if these furnaces are filled and replenished with red-hot coke taken out of an adjoining furnace. The air blast for welding ought not to pass through more than about 18 ins. of coke, otherwise the combustion is not perfect, and the flame not hot enough. (See W. van Folten, 'Stahl und Eisen,' 1893, p. 26.)

Fig. 420 shows a movable furnace delivering its flame in a horizontal direction. It can only be used with plates placed vertically, and then it is usual to place one furnace on each side.

Longitudinal seams in furnaces are very conveniently heated as shown in fig. 421. As the flame strikes only one side of the plate, it

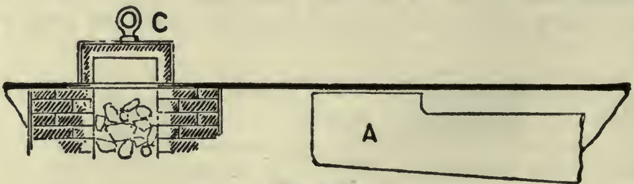


FIG. 422

is necessary to prevent radiation by the little firebrick cover C. It is also necessary to keep the seam wide open at this point, by bending back its edges, so that the flame can pass through it; otherwise only one side of the plate will weld. When ready, the furnace plate is lifted and turned round by means of the pulley P. This can be dispensed with by placing the furnace inside of the flue, as shown in fig. 422. In this case, when the seam has been sufficiently heated it is placed over the projecting anvil A, and welded.

Where gas is used for heating purposes, the arrangement shown in fig. 423 is a very convenient one, the ignited mixture of gas and air

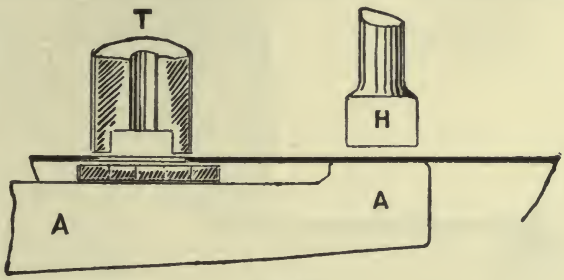


Fig. 423

passing down the tube T. A is the anvil, and H the steam hammer. Rollers are now very largely used instead of hammers. It is dangerous to use fans for supplying the air to the gas, as their action is not so reliable as that of any of the positive blowers, and causes explosions.

When the tube plate and furnace are to be welded together, this is usually done at the corner (see fig. 424). No flanging is then necessary.

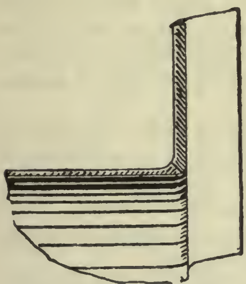


Fig. 424

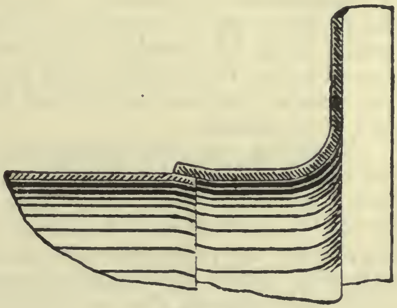


Fig. 425

But another plan may be mentioned, according to which the tube plate is first flanged so as to form a saddle, and this part is then welded to the furnace (fig. 425). This method is particularly convenient for use with patent flues, which, with one exception, are weakest near the combustion chamber end. With the above plan the thick metal of the tube plate supports the flue.

Although not carried out to any great extent, all the seams of the furnaces and combustion chambers can be welded, and the finished article will then have the appearance shown in fig. 426.

For remarks about electric welding and oxy-acetylene welding, see 'Steam Pipes'.

Hydraulic Test.—Although not forming part of the construction, the final hydraulic test is the concluding operation before the boiler is put into the ship. It is usual to have a preliminary test the day before, when any defective caulking can be made good, but with good work

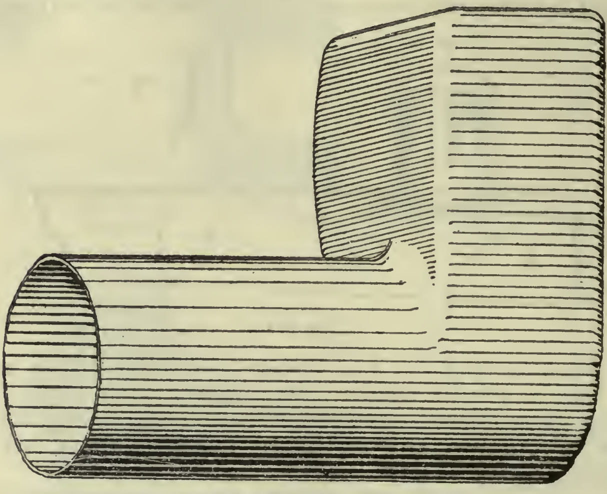


Fig. 426

this should not be necessary. The generally adopted plan is to raise the pressure step by step, and at once caulk any defect which shows itself. If the full pressure is put on at once, the leakages may be so excessive that they cannot all be put right. During this preliminary test the stay nuts are not screwed on, so that the stays may if necessary be recaulked.

Much has been said and written against testing boilers to double the working pressure, but, in spite of assertions to the contrary, riveted seams, and even welds, which were found to be perfectly tight with the cold test, commenced to leak at half that pressure when hot. This is probably due to the difference in the conditions of testing, and not, as is often stated, to the previous excessive proof stress.

It is also argued that an hydraulic test not exceeding the working pressure will detect defective material and flaws. This is not borne out by experience (see p. 223).

In one sense, the hydraulic test can therefore be looked upon as a guarantee of good material, although its chief object is to detect bad workmanship. This will show itself by leakages.

The amount of water which has to be pressed into a boiler amounts to about $\frac{1}{3}$ per cent. of its gross volume, unless all the air has not been removed, or unless the boiler is weak, in which case more water has to be pumped into it.

Boiler Deformations.—Observations as to the deformations are also made, but it cannot be said that they have been of much assistance in detecting local weaknesses; but they undoubtedly offer a means of studying the actual stresses in boilers, and a few remarks on the subject will not be out of place. That they are very much larger than most engineers expected was shown during the discussion on J. T. Milton's paper ('N. A.,' 1893, vol. xxxiv. p. 157) on this subject.

The stretching of the boiler circumference can be roughly measured by coiling a wire $1\frac{1}{2}$ times round it (fig. 427) and weighting the two

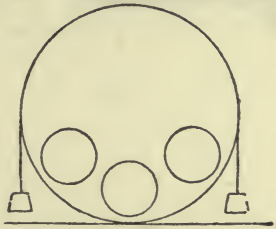


FIG. 427

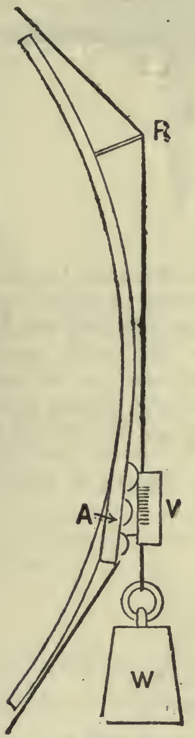


FIG. 428

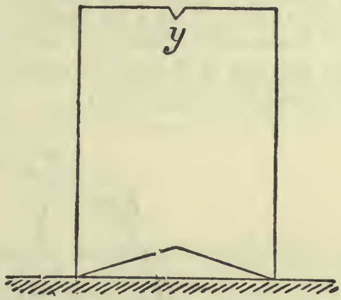


FIG. 429

ends, and then marking the two wires at the top of the boiler before and during the test. The stretch ought to be about $\frac{1}{1000}$ of the circumference, or, say, $\frac{3}{8}$ in. for an 11-ft. boiler, and $\frac{1}{2}$ in. for a 13-ft. boiler. It will be found that the reading is less, and also, on relieving the pressure, that the original marks do not coincide. This is not permanent set, but is due to the friction between the wire and the shell. More accurate results can be obtained if one end of the wire is bolted to the boiler and small rocking frames interposed between the shell and the wire at other points (see fig. 428). The wire is secured by means of a small bolt to the rivet-head A, and led from one rocking frame to another till it reaches R, and the weight W is then hung to the loose end, while a scale is secured partly to the bolt-head A and partly to the wire at V. The rocking frames (fig. 429) are made of thin steel plates, sharpened at the bottom. The wire passes over the notch *y*. By fixing several wires round one boiler the straining of the various parts of the shell and the influence of the end plates and seams can easily be ascertained (see p. 195).

Most of the other parts of the boiler can best be measured by rods or battens having a micrometer screw or other contrivance attached

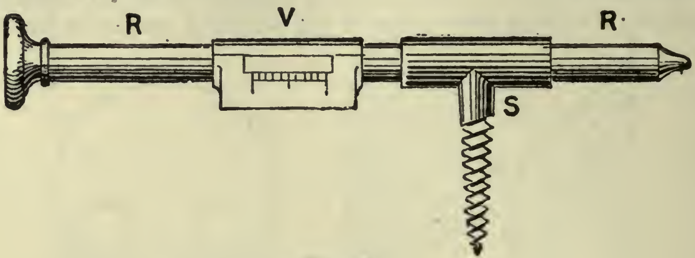


Fig. 430

to their ends. Fig. 430 shows a convenient instrument for this purpose: S is one of a large number of small sleeves, to which ordinary wood screws have been brazed, and by which means they are secured to the wooden rod or batten; R R is a steel rod which easily fits the sleeve S, and is accurately graduated, preferably in millimetres; V is another sleeve, slit open on one side, and containing a vernier. The two should be a sliding fit. Measurements of the various points of a boiler before and after testing are taken by consecutively inserting the rod R R with its vernier into all the sleeves S and noting the readings (see p. 169).

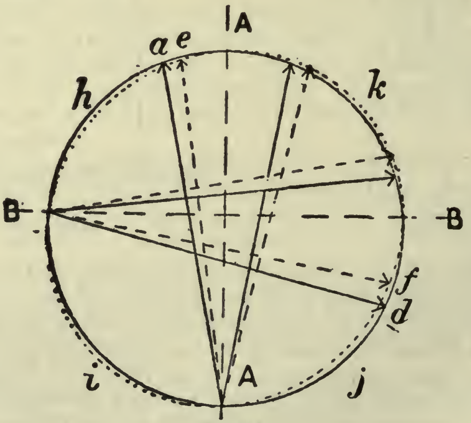


Fig. 431

For themeasurement of furnace deflections, ordinary trammels are more convenient, as, by their means, it is easier to detect the major and minor axes of deformation.

For instance, if measurements are taken at the axes A A and B B (fig. 431), with a deformation as shown by the dotted lines, then the final readings would differ but little from the original ones. However, if trammels are used, they will show that the diameters had changed their angle, and the trammel resting at A would touch *e* instead of *a*, while the one resting at B would touch *f* instead of *d*. This would lead to further measurements being taken at *jh* and *ik*, and would show that a serious deformation was taking place there. Should one

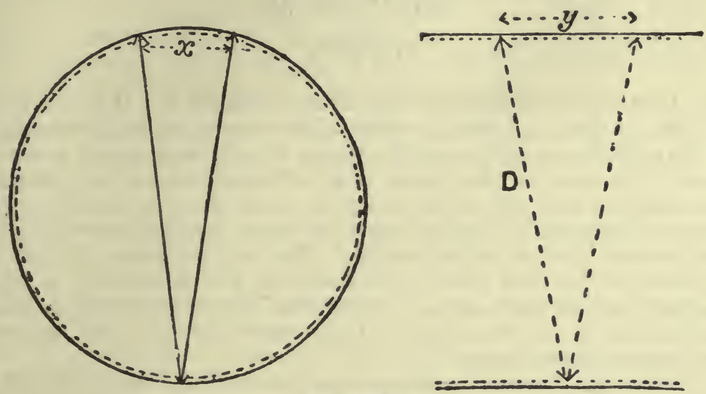


FIG. 432

of the diameters be so much reduced that the trammel cannot be got into position, its points of touching must be marked *along* the axis of the flue.

In order to reduce these measurements to absolute readings, employ the following formula : Δ_1 and Δ_2 are the difference between the diameter of the furnace and the length, D, of the trammel; x and y are shown in fig. 432.

$$\Delta_1 = \frac{y^2}{8D}. \qquad \Delta_2 = \frac{x^2}{8D}.$$

If the original reading was x and the subsequent one y, then the deformation of the furnace was $\Delta_1 + \Delta_2 = \frac{x^2 + y^2}{8D}$.

Let $D = 40$ ins., $x = 4$ ins., $y = 2$ ins., then

$$\Delta_1 + \Delta_2 = \frac{16 + 4}{8 \times 40} = \tfrac{1}{16} \text{ in.}$$

If the first measurement of x was 6 ins., and the second one 4 ins., then the deformation was also $\frac{1}{16}$ in., viz. $\frac{36 - 16}{8 \times 40}$.

Having tested the boiler, it ought to be examined internally, after which it is usual to cement the lower seams of the shell, to cut the various openings for steam pipes, &c., and to fit the boiler on board.

On account of the numerous recent cracks in furnace saddles, it has been suggested that these parts should be severely hammered after the test.

CHAPTER IX

DESIGN

The Proportions of Heating and Grate Surfaces and sectional areas of tubes, funnel, and other parts of boilers vary considerably according to the experiences of the manufacturers or shipowners, and it would be rash to attempt to harmonise these divergent views; all that can be attempted is to reduce the problem to the simplest elements, and these, it is hoped, will indicate how to make comparisons and draw conclusions from authentic records. The two conflicting factors are economy on the one hand, and maximum performance for a given weight of boiler on the other. In the one case large heating surfaces are necessary, in the other a high temperature of the escaping products is unavoidable.

Assuming the case of a boiler supplied with feed water of 100° F. and worked at a pressure of 150 lbs. (water temperature 315° F.), and assuming that the fuel is capable of evaporating 15 lbs. of water at and from 212° F., equal to 12·3 lbs. under the above conditions, and assuming that 15 lbs. of steam are required for each I.H.P., then the following results may be expected:

Pounds of water evaporated per square foot of heating surface per hour	2 to 3	5	9
The evaporating efficiency of the boiler will be approximately	70 %	65 %	60 %
The uptake temperatures will be about	500°	700°	900°
Weight of water in lbs. evaporated per lb. of fuel	8·6	8·0	7·4
Weight of fuel per hour per I.H.P.	1·75	1·88	2·03
Square feet of heating surface per I.H.P.	5 to 7½	3	1⅔

With the help of the above-mentioned funnel temperatures it is possible to estimate the force of its draught when the height is known.

Funnel Draught Measured in Inches, Water.

Mean Temperature of Waste Gases	Mean Height of Funnel above Fire Bars				
	20 ft.	40 ft.	60 ft.	80 ft.	100 ft.
400° F.	·14	·28	·42	·56	·70
600° F.	·17	·34	·51	·68	·85
800° F.	·19	·38	·57	·76	·95
1,000° F.	·20	·41	·61	·82	1·02

Any desired pressure or suction can of course be obtained by mechanical means (forced draught). If the specific resistance due to fuel and other obstructions were known, it would be easy to estimate Q, the weight of air which passes through 1 square foot of grate per hour: $Q = 730\sqrt{\frac{h}{1+r}}$, where h is the draught pressure in inches of water, and r the specific resistance to the air passage; but as this latter value can only be guessed at (see p. 102), and as the weight of coal consumed is not strictly proportional to the weight of air which passes through the grate, the matter is simplified by comparing the coal consumption and funnel draught direct.

Approximate Coal Consumption in lbs. per Square Foot of Grate.

Draught pressure, inches (water)		$\frac{1}{4}$	$\frac{1}{2}$	$\frac{3}{4}$	1	2	3	4
Pounds of coal burnt per hour per sq. ft. of grate	from	15	20	25	30	40	50	60
	to	20	25	30	40	50	75	80

The following case will illustrate the use to which these tables can be put. The relations between the various surfaces are to be determined for a boiler whose funnel height is 80 feet, and which is to give economical results. The uptake temperature should therefore not exceed 500° F., and the draught suction will be about ·6 in.; the consumption per square foot of grate will be about 20 lbs. per hour; the indicated horse-power will be about 11 per square foot of grate, and the heating surface will have to be about 55 times as large as the grate area.

If economy is of secondary consideration, then the uptake temperature will have to be about 1,000°, the draught will be ·85 in., the consumption 30 lbs. per square foot of grate, and the horse-power about 14, which is not much more than in the above case; but the heating surface will be very much less, viz. 23 times as large as the grate.

Had the funnel been shorter, the consumption per square foot of grate would have been less, and the ratio of the heating surface would also have been reduced, whereas with forced draught it would have to be increased.

The absence of any reliable data on the relationship between funnel height, grate area, and heating surface makes it difficult to check the above values by actual performances, and the above tables and calculations should, therefore, only be looked upon as an indication how deductions could be drawn from really reliable information. The necessity for doing this has occasionally made itself felt in boilers which were to be exceptionally economical. In one case the lowness of the funnel and its temperature reduced the boiler performance so seriously that it was necessary to reduce the heating surface by blocking up a large number of tubes; and it is evident that if these boilers had originally been properly designed, they could have been made of very much smaller dimensions.

Funnel Dimensions.—A natural desire to reduce both the height and diameter of a funnel, so as to offer little resistance to the speed of the ship, will, if carried too far, reduce the draught, and with it the boiler performance. Many people hold the view that a short funnel with a large diameter is as efficient as a tall one with a reduced sectional area; but this can only be true in cases where the resistance to the motion of the products of combustion is greatest in the funnel. In well-designed ones this is far from being the case, and then, within certain limits, the draught is not affected by the diameter. The one limit is determined by the velocity of the waste gases, which should not exceed 25 ft. per second under natural draught. The other limit is more difficult to fix, but it is quite certain that if a funnel is made too large in section, cold air will rush down from above and interfere with the up current. This happens with those funnels which occasionally draw well, and at other times badly. It is well known that factory chimneys parallel outside and conical inside, which were the fashion some years ago, were serious offenders in this respect, from which it is reasonable to conclude that not only the size and proportions, but also the shapes of funnels, affect the results.

Dimensions of Grates and Furnaces.—As a general rule, and for ordinary work, the stoking of furnaces whose bars are longer than 5 ft. cannot be done economically. When their lengths exceed 6 ft. most of the extra coal burnt is simply wasted. This is particularly the case with forced draught, and some engineers advocate that under such conditions the grate should not be longer than 4 ft.

Stoking is seriously interfered with if the furnace diameters are small, and probably the cooling influence of the plates close to the fire retards combustion, whereby unconsumed gases are permitted to escape. So that, unless grates are very short, furnaces should not be made less than 3 ft. diameter, and where flaming coals are used they should be still larger.

Builders prefer small furnaces, because with them more grate surface and more heating surface can be got into a boiler of a given diameter; but steam users should see that this does not lead to their being supplied with an inferior boiler.

Tubes.—The furnace and tube lengths are practically identical, and vary from 4 ft. for short double-ended boilers to 9 ft. for single-ended boilers with forced draught. In the latter case the tube diameters are about $2\frac{1}{2}$ ins., while for natural draught, where a small internal sectional area would offer too much resistance, the length is usually about 24 diameters.

Tube Surfaces, and Space Occupied.—The following table gives some of the dimensions of boiler tubes. The thickness of the metal is assumed to be ·15 in., or about No. 9 wire gauge. The external and internal heating surfaces are given per foot of length. The internal sectional area of a single tube is also added. The last two lines of the table contain the amount of end space required by one tube when it is surrounded either by 1 in. or $1\frac{1}{4}$ in. of water space (see dotted lines, fig. 433).

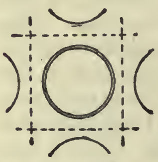


Fig. 433

In one and the same boiler the number of tubes for each furnace sometimes differs considerably. This is a bad design, for there are either too few for the one or too many for the other. The tube surface amounts to from 75 to 85 % of the total heating surface.

External diameter, inches	$2\frac{1}{4}$	$2\frac{1}{2}$	$2\frac{3}{4}$	3	$3\frac{1}{4}$	$3\frac{1}{2}$	$3\frac{3}{4}$
Sq.ft. of external surface per ft.	·59	·65	·72	·79	·85	·92	·98
„ internal „ „	·51	·57	·64	·71	·77	·84	·90
„ „ sectional area	·021	·029	·033	·040	·048	·056	·065
Square inches of end space per tube.							
Water spaces = 1 in.	10·5	12·2	14·0	16·0	18·1	20·2	22·6
„ = $1\frac{1}{2}$ in.	12·2	14·0	16·0	18·1	20·2	22·6	25·0

Boiler Diameter.—Having decided how many square feet of tube heating surface the boiler should contain, what diameter and length the tubes should be, and what widths of water space are to be left between the tubes, the above table will enable one to estimate how many square inches of tube plate area will be taken up by the tubes. These areas are represented by A and B (fig. 434), and the various letters indicate the other water and steam spaces, viz.:

- *a* is the distance between two furnace diameters.
- *b* is the distance between the furnace and the shell.
- *c* is the distance from the wing furnace to the lower row of tubes.
- *d* is the furnace diameter.
- *e* is the distance from the centre furnaces to the lower row of tubes.
- *f* is the distance from the shell to the wing tubes.
- *h* is the same dimension measured in the steam space.
- *g* is the distance between the nests of tubes.

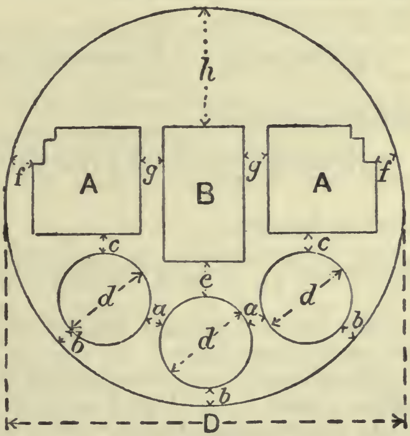


FIG. 434

All these dimensions are to be taken, not from centre to centre, nor from the circumferences, but from the squares surrounding the tubes, as shown in fig. 435.

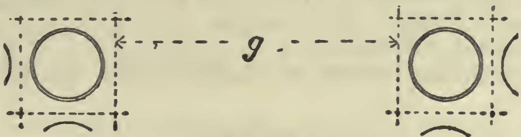


FIG. 435

Let Σ (T) represent the sum of the end space required for the tubes. In fig. 434 this would be 2A + B. Let D be the boiler diameter, then, $\Sigma\ (T) = 0{\cdot}45(D-d-K)^2$.

The value of K is found by multiplying the various water and steam spaces by the numbers contained in the following table, and then adding them together :

Number of Furnaces in Boiler . .	Two	Three	Four
Multiply a by	·20	·70	·45
" b "	·60	·60	·45
" c "	...	·20	·70
" the sum of e by . . .	·30	·25	·35
" " " g " . . .	·20	·25	·35
" f by	·40	·40	·50

The above formula will be found to agree fairly well with the general practice as long as h is equal to one-third of the boiler diameter ($\frac{1}{3}$ D). Where this relation does not exist, subtract from D, $\frac{3}{4}$ in., 1 in., and $1\frac{1}{8}$ in., for two-, three-, and four-furnaced boilers respectively, for every additional inch of steam space, so that if written out *in extenso* the above formula would be—

For two furnaces :

$$\Sigma(\mathrm{T})=0{\cdot}45\left(\frac{5}{4}\mathrm{D}-\frac{3}{4}h-d-{\cdot}20[a+3b+2f+\frac{3}{2}\Sigma(c)+\Sigma(g)]\right)^2.$$

For three furnaces :

$$\Sigma(\mathrm{T})=0{\cdot}45\left(\frac{5}{4}\mathrm{D}-h-d-{\cdot}7a+{\cdot}6b+{\cdot}2c+{\cdot}4f+{\cdot}25[\Sigma(e)+\Sigma(g)]\right)^2.$$

For four furnaces :

$$\Sigma(\mathrm{T})=0{\cdot}45\left(\frac{11}{8}\mathrm{D}-\frac{9}{8}h-d-{\cdot}45(a+b)+{\cdot}35[2c+\Sigma(e)+\Sigma(g)]+{\cdot}5f\right)^2.$$

The sign Σ ( ) means that the letter in the bracket is the sum of the several dimensions.

These formulæ can be simplified by adopting the following average values : a and b vary from 3 ins. to 8 ins., and are usually 5 ins.

In two-furnaced boilers c varies from 3 ins. to 11 ins., but it is generally made $7\frac{1}{2}$ ins. In other boilers it varies from 8 to 12 ins., the mean being 10 ins.

e is sometimes as much as 15 ins. Generally both c and e are made 10 ins.

f and g both vary from 8 ins. to 12 ins. The usual practice is to make both about 10 ins. ; this would give $10\frac{1}{2}$ ins. of clear water space at the wings, and 11 ins. at the centres.

The mean value of h is $\frac{1}{3}$ D.

Substituting these average values, the formulæ are reduced to the following :—

For two furnaces and two combustion chambers—

$$\Sigma(\mathrm{T}) = 0{\cdot}45\,(\mathrm{D}-d-12\tfrac{1}{4})^2.$$

For three furnaces and three combustion chambers—

$$\Sigma(\mathrm{T}) = 0{\cdot}45(0{\cdot}92\mathrm{D} - d - 22\tfrac{1}{2})^2.$$

For four furnaces and two combustion chambers—

$$\Sigma(T) = 0{\cdot}45\,(D - d - 24\tfrac{3}{4})^2.$$

For four furnaces and three combustion chambers—

$$\Sigma(T) = 0{\cdot}45\,(D - d - 28\tfrac{1}{4})^2.$$

For the purpose of ascertaining the diameter it will be more convenient to alter these formulæ as follows:—

$$D = \frac{3}{2}\sqrt{\Sigma(T)} + d + K.$$

The Length of the Boiler is fixed by the length of the tubes or furnace, by the depth of the combustion chambers, and by the water space at their backs.

The latter should not be made less than 5 ins., but cases are met with where they are reduced to 3 ins. They should be made wider at the top than at the bottom, the usual angle being $\frac{1}{2}$ in. per foot of depth.

The Combustion Chambers, measured horizontally, should be made as deep as possible: 28 ins. and 36 ins. seem to be the smallest limits for single- or double-ended boilers respectively. Generally this depth is about 12 ins. greater than half the furnace diameter for single-ended boilers, while for double-ended ones with through combustion chambers it is made about 24 ins. deeper than half the furnace diameter.

The following are the relations usually existing between various boiler dimensions:—

Boilers with Two Furnaces.—The ratio of boiler diameter to furnace diameter is generally as 10 to 3, but sometimes 10 % more or less. Custom is equally divided between leading the two furnaces into two combustion chambers or into one. In the latter case the central water space between the tubes is sometimes dispensed with, but generally a few rows of tubes are left out along this line. The diameters of these boilers range from 8 ft. to 14 ft., and the lengths from 8 ft. to 10 ft. for natural draught, and up to 12 ft. for forced draught. The shortest double-ended boilers are 12 ft. long, and the longest 18 ft.

Boilers with Three Furnaces.—The ratio of boiler diameter to furnace diameter is generally as 4 to 1, but sometimes 7 % more or less. Generally three combustion chambers are fitted, but sometimes only one, and in that case the tubes are mostly divided into two groups with a water space in the centre. In rare cases there are no water spaces except at the sides. The diameters of these boilers range from 11 to 16 ft., and the lengths from 9 to 11 ft. for natural, and 12 ft. for forced draught. Double-ended boilers are sometimes made 20 ft. long.

Boilers with Four Furnaces.—The ratio of boiler diameter to furnace diameter is generally 5 to 1, but sometimes 5 % more or less. The combustion chambers are usually so arranged that the two central furnaces are led into one, and the two wing ones into separate chambers, so that there are one large and two small ones. Sometimes there are only two combustion chambers, and very rarely there are four, or only a single one. The diameters range from 13 to 17 ft., and the lengths are the same as for three-furnaced boilers. Double-ended ones are rarely built.

Boiler Performances.—The following are a few rough rules for estimating the heating surface and power of a boiler under natural draught:—

D = boiler diameter in feet.
L = boiler length in feet.
P = working pressure in pounds per square inch.
HS = heating surface in square feet.
Q = boiler weight in tons (no funnel, &c.).
W = water weight in tons.
C = coefficients.

IHP = indicated horse-power = from $\frac{1}{6}$ to $\frac{1}{2}$ HS and up to 1 HS with forced draughts. With water tube boilers these coefficients have to be reduced about 25 %.

$HS = C_1 \cdot D^2 \cdot L$
$C_1 = 0{\cdot}9$ for single-ended boilers, natural draught.
$C_1 = 0{\cdot}95$ to $1{\cdot}0$ for single-ended boilers, forced draught.
$C_1 = 1{\cdot}0$ for double-ended boilers, natural draught.
$C_1 = 1{\cdot}05$ for double-ended boilers, forced draught.
$Q = C_2 (D^2 \cdot L + HS)(P + 60)$.
$1 : C_2 = 27{,}000$ for single-ended boilers to Lloyd's Rules.
$1 : C_2 = 25{,}700$ for single-ended boilers to Board of Trade Rules.
$1 : C_2 = 30{,}000$ for double-ended boilers to Lloyd's Rules.
$1 : C_2 = 28{,}600$ for double-ended boilers to Board of Trade Rules.
$W = 0{\cdot}01\ D^2L$.

In 1898 on the Clyde and before the recent rise in prices, the cost of single-ended boilers working at 150 pounds and designed to Lloyd's Rules, not including funnel, &c., was

$$\text{Cost in } £ = 0{\cdot}35\ D^2 \cdot L + 2\,(D^2 \cdot L)^{2/3}.$$

The first term represents the cost of material and such charges as are nearly proportional to weight, and therefore to pressure; the second term represents such charges—as riveting, tapping, screwing, caulking —as are nearly proportional to the boiler surfaces.

All the above coefficients vary in different works, as they depend on local practices and appliances.

The following are a few published details of boiler weights. Bertin & Robertson, p. 355, 'Weights of Water-tube Boilers'; also 'Norman,' 'Enging.,' vol. lviii. p. 701; 'Kensington,' 'Enging.,' vol. lviii. p. 199; U.S. 'Minneapolis,' 'Enging.,' vol. lx. p. 600.

Two lists of boiler performances have been published by F. Marshall, 'M. E.,' 1881, p. 449, and 1891, p. 337.

The following is a list of published drawings of marine boilers:—

F. Colyer, 1886. Three-furnaced oval boiler, by Maudslay, 12 ft. 4 ins. diam., 14 ft. 1 in. diam., 9 ft. 11 ins. long; three-furnaced cylindrical boiler, 17 ft. 4 ins. diam., 9 ft. 1 in. long.

Schwarz Flemming, 1873. Forty-nine sketches of boilers.

C. Busley, 1883. Boilers in vessels of German Navy.

B. N. Bartol, 1851. Sketches and surfaces of American boilers.

J. T. Winton, 1883. Various types of boilers.

N. P. Burgh, 1873, contains drawings of about 20 boilers and sketches of 90 types of patented boilers from 1852–71.

N. Foley, 1891. Plates 6, 7, 8, give full detailed drawings of the three following boilers:—

Single-ended, 2 furnaces, 39 ins. internal diam., 1,099 sq. ft. heating surface. Outside dimensions, 12 ft. × 9 ft. 8 ins. Weight, 25 tons. Steam pressure, 150 lbs.

Single-ended, 3 furnaces, 35½ ins. internal diam., 1,334 sq. ft. heating surface. Outside dimensions, 12 ft. 6 ins. × 9 ft. 8 ins. Weight, 27 tons. Steam pressure, 150 lbs.

Double-ended, 4 furnaces, 34 ins. internal diam., 1,843 sq. ft. heating surface. Outside dimensions, 10 ft. 6 ins. × 16 ft. 6 ins. Weight, 34 tons. Steam pressure, 160 lbs. (See also p. 318.)

Other types of boilers will be found in the following lists:—

Dry Back Type.—'Enging.,' vol. lxiii. p. 474.

Navy Type.—'Surprise' and 'Alacrity,' Palmers', 'Enging.,' vol. xl. pp. 447, 450; 'Melbourne,' Simmons & Co., 'Engr.,' vol. lx. p. 392; 'Yorktown,' 'Enging.,' vol. li. p. 493; 'Bergen,' 'Enging.,' vol. xlix. p. 191; 'Mouche,' Belliss & Co., 'Enging.,' vol. xxxix. p. 81; 'Enging.,' vol. lv. p. 694.

Locomotive Type.—Schichau, 'Enging.,' vol. xxxiv. p. 579; Yarrow, 'Enging.,' vol. xlii. p. 179; 'Sunderland,' Doxford, 'Enging.,' vol. xlix. p. 30 (for burning petroleum); Hick, Hargreaves, 'Enging.,' vol. xlix. p. 528; 'Barham' and 'Bellona,' Hawthorn, Leslie & Co., 'Enging.,' vol. l. p. 705; 'Phlegeton,' Soc. Ann. Claparede, 'Engr.,' vol. lx. p. 277; Sectional locomotive boiler for transport, Sandycroft Foundry, 'Enging.,' vol. xxxviii. p. 261; N. Foley, 1891, pl. x. and xi.; 'Enging.,' vol. lv. p. 695; H.M.S. 'Hazard,' 'Enging.,' vol. lviii. p. 421.

Water Tube Boilers.—J. F. Spencer, 'M. E.,' 1859, p. 264; Z. Colburn, ibid., 1864, p. 61; Laybourne, ibid., 1871, p. 263; D. Joy, 'I. and S. I.,' 1874, p. 220; Perkins boiler, Maw, ciii.; Adams and Co., 'Enging.,' vol. xxxiv. p. 251; J. F. Flanery, 'C. E.,' 1878, vol. liv. p. 123; J. T. Thornycroft, 'Enging.,' vol. xxxv. p. 463, vol. xliv. p. 105, vol. xlvii. p. 402; 'N. A.,' 1889, vol. xxx. p. 271; 'C. E.,' 1890, vol. xcix. p. 41; Ward's Patent, 'Enging.,' vol. xlvii. p. 322; Yarrow, 'Enging.,' vol. li. p. 79. All recent vessels of the French Navy are having tubular boilers fitted (Belleville, Lagrafel-D'Allest), J. T. Milton, 'N. A.,' 1893, vol. xxxv. Danish cruiser 'Geiser' (Thornycroft), 'Enging.,' vol. liv. p. 789; 'Daring' (Thornycroft), 'Enging.,' vol. lvi. p. 667; 'Algerie' (Babcox and Wilcocks), 'Enging.,' vol. lviii. p. 433; Stirling Boiler, 'Enging.,' vol. lix. p. 825; Stirling type, 'Enging.,' vol. lix. p. 825; H.M.S. 'Terrible' (Belleville details), 'Enging.,' vol. lix. p. 822; Du Temple, 'Enging.,' vol. lx. p. 57; Normand, 'Enging.,' vol. lx. p. 75; H.M.S. 'Surly,' 'Enging.,' vol. lx. p. 630; Niclausse, 'Enging.,' vol. lx. p. 91; Thornycroft, 'Enging.,' vol. lx. p. 269; Ligaudy, 'Enging.,' vol. lx. p. 541; Serpollet, 'Enging.,' vol. lx. p. 501; Haythorn, 'Enging.,' vol. lx. p. 680; Russian 'Kherson' (Belleville), 'Enging.,' vol. lxii. p. 799; H.M S. 'Spanker' (Du Temple), 'Enging.,' vol. lxiii. p. 777; H.M.S. 'Pelorus' (Normand), 'Enging.,' vol. lxiii. p. 385; 'Laos' (Belleville), 'Enging.,' *List of Published Boiler Drawings*

Vessel's Name	Builder	Where Published[1]	Dimensions: Diameter	Dimensions: Length	Dimensions: Product L × D²	Furnaces: No.	Furnaces: Type	Furnaces: Internal Diameter	Working Pressure	Per Boiler: Grate Surface	Per Boiler: Heating Surface	Per Boiler: Indicated Horse-Power	—
DOUBLE-ENDED BOILERS													
			ft. ins.	ft. ins.				ins.	lbs.	sq. ft.	sq. ft.		
'Kaiser Wilhelm II.'	Vulcan	'Enging.,' vol. lxxvi. p. 245	19 5	20 10	7870	8	Mor.	47½	213	201	6889	...	High-water level
'Kaiser W. d. Grosse'	Vulcan	" vol. lxv. p. 649	16 11	20 6	5880	8	Purves	47	175	201	6469	...	90 tons
'Osterley'	London & Glasgow	" vol. lxxxvii. p. 863	16 6	20 2	3320	8	Fox	39	215	156	6200	2760	Forced draught
'Kensington'	J. & G. Thomson	" vol. lviii. p. 199	15 9	21 5	5310	8	Purves	39	200	153	4870	3320	Funnel 84 ft.
U.S. 'Olympia'	...	" vol. lv. p. 611	15 3	21 3	4990	8	Fox	39	160	165	5670	...	...
U.S. 'Minneapolis'	Cramp & Son	" vol. lx. p. 600	15 9	20 0	4960	8	Fox	40	...	188	6140	3100	...
'Hanoverian'	Doxford	Maw, cxix.; 'Enging.,' vol. xxxv. p. 338	15 6	20 0	4700	6	Flanged	46	...	126	4015	...	...
H.M.S. 'Edgar,' 'Hawk'	Fairfield Co.	'Enging.,' liii. p. 11, lv. p. 700	16 0	18 0	4600	8	Fox	38	155	...	5027	...	...
H.M.S. 'Gibraltar'	...	" vol. lv. p. 901	16 0	18 0	4600	8	Purves	43½	155	174	5770	2825	...
'Sylvania'	London & Glasgow	" vol. lx. p. 539	15 0	20 3	4570	6	Mor.	83½	180	110	4805	2677	...
Ophir'	Napier	" vol. lii. p. 586	15 1	19 1	4340	6	Purves	42	160	126	4015	...	Funnel 88 ft.
Scot'	Denny	" vol. lii. p. 39	15 3	18 0	4240	6	Fox	45	170	139	3827	1943	Funnel 98 ft.
Norman'	Harland & Wolff	" vol. lviii. p. 699	15 3	17 6	4100	6	Mor.	44½	180	120	4230	1840	...
'Teucer,' 'Orestes,' &c.	Scott	Maw, clxii.; 'Enging.,' vol. xxxvi. p. 544	12 0 × 14 2	23 9	4040	6	Fox	40	...	...	...	...	Holt type
'Parisian'	Napier	Maw, v.; 'Enging.,' vol. xxxii. p. 276	15 0	17 9	4030	6	Plain	45	75	136	3794	1505	Funnel 72·5 ft.
'Infanta Maria Theresa'	Soc. Las Astilluos	'Enging.,' vol. lvii. p. 805	15 3	13 3	3800	8	Purves	40	...	169	5067	2745	...
'Mexican'	Clark	Maw, xxxiii.; 'Enging.,' vol. xxxiii. p. 51	12 8 × 16 6	17 6	3660	6	Fox	40	90	...	3333	...	...
'Iberia'	D. Rollo	" vol. xvi. p. 207	14 6	17 0	3580	6	Fox	40	180	...	...	1125	...
'Lackawanna'	D. & J. Dunlop	" vol. lvii. p. 197	14 6	16 9	3540	6	Purves	40	160	115	3575	...	...
'Delaware'	D. J. Dunlop & Co.	" vol. xvii. p. 197	14 6	16 9	3520	6	Purves	40	160	115	3575	1343	...
'Kaiser W. II.'	Vulcan	" vol. l. p. 188	13 1½	19 1¼	3290	4	Fox	47	157	98	3000	1110	...
'Oroya'	Barrow	'Engr.,' vol. lxiii. p. 290	13 6	18 0	3280	6	...	35	160	104	2940	...	...
'Arizona'	Elder	Maw, xcv.	13 6	18 0	3280	6	Bowl Rings	39	90	...	...	...	...
'Kathleen Mavourneen'	Jack	'Enging.,' vol. xli. p. 271	14 3	16 1	3270	6	Plain	40	...	110	3314	...	Funnel 62 ft.
H.M.S. 'Sybille'	...	" vol. lv. p. 694	13 0	18 5	3100	6	Purves	40	...	...	...	...	...
'Indra'	Fawcett Preston	" vol. li. p. 525	13 6	16 6	3010	6	Fox	38	150	...	3082	876	...
'Golconda' (*ex* 'Nulli Secunda')	Doxford	" vol. xlii. p. 543	12 9	17 9	2890	4	Fox	44	150	82	2844	...	Funnel 76 ft.
'Assyrian Monarch'	Earles	Maw, cli.	12 3	18 6	2780	6	Fox	45	...	...	...	...	...
'Grecian'	Doxford	" cxviii.	11 0 × 13 6	18 6	2750	6	Plain	34	...	...	...	...	Funnel 61 ft.
——	German	'Engr.,' vol. lxiv. p. 516	12 6	17 5	2740	4	Fox	43	143	...	...	...	...
'Tartar'	Richardson	" vol. lxix. p. 420	13 0	14 9	2490	4	...	42	...	80	2613	...	Funnel 69ft. × 9½
H.M.S. 'Barracouta'	Palmers	" vol. xlix. p. 476	10 9	17 6	2020	4	Fox	37	...	...	...	...	...

SINGLE-ENDED BOILERS

'Kaiser W. d. Grosse'	Vulcan	'Enging.,' vol. lxv. p. 649	16 11	11 6	3280	4	Purves	47	175	100	3235	...	...
'Corsican'	Barclay Curle	" vol. lxxxv. p. 111	16 6	12 0	3270	4	Beigtn.	41	190	77	3143	1360	Forced draught
'Osterley'	London & Glasgow	" vol. lxxxvii. p. 863	16 6	11 6	3130	4	Fox	39	215	78	3100	1380	...
'Africa'	Australian Lloyds	" vol. lxxviii. p. 374	15 6	12 2	2940	3	Purves	44½	200	61	2563	...	...
'Kensington'	J. & G. Thomson	" vol. lviii. p. 199	15 9	11 3	2800	4	Purves	39	200	77	2435	1663	Funnel 84 ft.
'Isle of Dursey'	Wallsend	" vol. xxxvii. p. 188	16 0	10 8	2730	3	Plain	34	150	42	1650	600	...
H.M.S. 'Renown'	Maudslay, Sons & Field	" vol. lx. p. 79	16 3	10 3	2720	4	Mor.	42½	155	102	3105	1500	...
U.S. 'Olympia'	...	" vol. lv. p. 611	15 8	10 11½	2550	4	Fox	39	...	82	2830	...	...
H.M.S. 'Crescent'	...	" vol. lv. p. 695	16 1	9 10½	2550	4	Fox	44	...	106	3098	...	...
'Infanta Maria Theresa'	Soc. Las Astilluos	" vol. lvii. p. 805	15 3	10 6	2450	4	Purves	40	150	84	2825	...	...
'Ophir'	Napier	" vol. lii. p. 587	15 1	10 8	2430	3	Purves	42	...	63	2170	700	Funnel 88 ft.
'Norman'	Harland & Wolff	" vol. lviii. p. 699	15 3	10 3	2390	3	Mor.	43¾	180	60	2118	917	...
'County of Salop'	Barrow	" vol. xxxviii. p. 567	14 6	11 0	2310	3	Fox	40	...	60	1872	557	...
H.M.S. 'Monarch'	...	" vol. lv. p. 694	15 3	9 10	2290	4	Fox	41	...	80	2310	...	...
H.M.S. 'Royal Oak'	Laird Bros.	" vol. lvii. p. 515	15 4	9 4	2200	4	Fox	38	155	89	2520	1446	...
'Main'	Palmers	'Engr.,' vol. lxix. p. 330	14 8	10 0	2150	3	...	42	135	69	2350	1120	...
H.M.S. 'Royal Sovereign'	...	'Enging.,' vol. lv. p. 695	15 0	9 4	2100	4	Fox	38½	...	90	2500	...	...
I.R.N. 'Sinope'	Napier	" vol. l. p. 81	14 7	9 10	2090	3	Plain	48	125	76	1870	917	Funnel 80 ft.
'Lynx,' 'Antelope,' &c.	Laird	" vol. xlix. p. 699	14 3	10 2	2060	3	Purves	40	150	59	1900	850	...
—	Tod McGregor	Maw, viii.	12 6 ×15 8	10 3	2010	3	Plain	39	...	...	...	...	...
'Hunstanton,' 'Godiva,' &c.	Westgarth English	'Enging.,' vol. xxxvii. p. 473	13 9	10 3	1950	3	Plain	40	80	...	1798	700	...
Italian 'Sicilia'	Ansaldo & Co.	" vol. lxi. p. 344	14 3	9 7	1950	4	Plain	38	100	...	...	...	...
'City of San Francisco'	Roach	Maw, clvi.	13 0	10 6	1780	3	Plain	39	...	...	...	...	...
'Inchdune'	Central Marine	'Enging.,' vol. lxxi. p. 72	13 0	10 6	1780	3	Purves	34½	267	...	...	800	Superheater
'Princess Elizabeth,' &c.	Elder	" vol. xxxi. p. 219	13 7½	9 1	1690	3	Plain	54	70	91	1826	886	Furn'ce necks contracted
H.M.S. 'Sybilla'	...	" vol. lv. p. 694	13 0	9 6	1600	3	Purves	40	...	...	...	...	...
'Ozar'	Wallsend	Maw, cxxxiii. 'Enging.,' vol. xxxv. p. 420	12 6	10 0	1590	3	Flanged	39	80	58·5	1427	...	...
'Royal Prince'	Dickinson	" vol. xli. p. 613	12 6	10 0	1590	3	Fox	34	150	36	1350	409	Funnel 52 ft.
'The Earl'	Finch	" vol. xliv. p. 403	12 6	10 0	1590	3	...	36	100	54	...	530	Funnel 24 ft.
'County of York'	Barrow	" vol. xliii. p. 297	12 0	10 4	1500	3	Fox	33	164	50	1346	980	...
H.M.S. 'Malacca'	J. Penn & Son	" vol. lvii. p. 463	9 6	14 0	1400	5	Plain	30	60	...	...	...	Navy Type
'Cairndhu'	Fleming & Ferguson	" vol. lviii. p. 387	11 8	10 0	1370	2	Plain	39	80	39	1134	375	...
'Dredger'	Hawks Crawshaw	" vol. xxxv. p. 247	10 6 ×11 9	10 6	1300	2	Plain	40	80	40	1150	300	...
'Westmoreland'	Rollo	" vol. xlii. p. 71	12 0	9 0	1290	3	Fox	32	150	34	1200	320	Funnel 44 ft.
'Normandy'	Elder	" vol. xxxix. p. 283	11 10	8 3	1160	2	Fox	45	110	49	1176	630	...
'Arabian'	Rankine & Blackmore	" vol. xxxviii. p. 83	10 6	9 5	1040	2	Fox	36	140	33	920	340	...
Yt. 'Speedy'	Ramage & Ferguson	" vol. lxii. p. 241	10 9	9 0	1040	2	Purves	40	180	...	...	400	...
'Schelde II.'	L. Smit & Son	" vol. lx. p. 662	9 10	9 10	950	2	...	...	150	...	850	...	...
'Churchill'	Hall Russell	" vol. xxxvii. p. 2	9 6	10 0	900	2	...	...	100	...	850	300	Funnel 32 ft.
'Grace Darling'	Fleming & Ferguson	'Engr.,' vol. lxv. p. 237	10 0	9 0	900	2	Purves	35	...	27	752	360	...
'Infanta Maria Theresa'	Soc. Las Astilluos	'Enging.,' vol. lviii. p. 805	7 9	8 10½	530	2	Purves	29	150	25	458	...	...
'Gladiator'	Ramage & Ferguson	" vol. xliii. p. 105	8 6	7 9	560	1	Plain	...	150	15	500	...	...
—	Hodge	" vol. xxxvi. p. 4	4 9	6 9	166	1	Plain	24	...	...	...	...	...

[1] Maw, with Roman numerals, stands for W. H. Maw's *Recent Practice in Marine Engineering*, London, 1883 · Mor. for *Morison's corrugated furnaces*.

vol. lxvii. p. 544; Maxim, 'Enging.,' vol. lviii. p. 195; Niclausse, 'N.A.,' 1896, vol. xxxvii. p. 119. Bertin and Robertson.

General descriptions of water-tube boilers: 'Marine Boilers,' Bertin and Robertson; Water-tube Boilers, J. T. Milton, 'N. A.,' 1893, vol. xxxv. pp. 86, 295; Water-tube Boilers, J. K. Robinson, U. S. Navy, Am. Soc., 'N. E.,' 'C. E.,' 1895; also 'Enging.,' vol. lx. p. 554. Circulation in Water-tube Boilers: Prof. Watkinson, 'N. A.,' 1896, vol. xxxvii. p. 267; also 'Enging.,' vol. lxi. p. 437; J. A. Normand, 'N. A.,' 1896, vol. xxxvii. p. 109; C. C. P. Fitzgerald, 'N. A.,' 1897, vol. xxxviii. p. 165; A. F. Yarrow, 'N. A.,' 1898, vol. xl. p. 114; A. F. Yarrow, 'N. A.,' 1899, vol. xli. p. 333. Very instructive information about boilers in general will be found in the Admiralty Report, 1892.

In the following list the vessel's name and reference only are given. The drawings show the positions of boilers, pipes, &c., as fitted on board :—

W. H. Maw, 1883: 'Gallia,' plate xv.; 'Hohenzollern,' xxi.; H.M.S. 'Grappler,' 'Wrangler,' 'Wasp,' 'Banterer,' 'Espoir,' xlvii. and xlviii.; 'Servia,' lxiv.; H.M.S. 'Rover,' lxxi.; 'Arizona,' xcii.; 'Grecian,' cxviii.; 'Assyrian Monarch,' cxlvi. and cxlvii.; 'Czar,' cxxxi.; H.M.S. 'Conqueror,' cxliii.; 'Normandie,' clxviii. 'Engr.': 'Chicago,' vol. lxi. p. 86; 'Oroya,' vol. lxiii. p. 234; 'Grace Darling,' vol. lxv. p. 236; 'Elbe,' vol. lxiv. p. 522. 'Enging.': 'Princess Elizabeth,' 'Princess Marie,' vol. xxxi. p. 119; 'Servia,' vol. xxxiii. p. 247; 'Satellite,' 'Conqueror,' vol. xxxv. pp. 266, 267; 'Czar,' vol. xxxv. p. 364; 'Churchill,' vol. xxxvii. p. 2; 'Normandie,' vol. xxxvii. p. 65; 'Godiva,' 'Stokesley,' and 'Hunstanton,' vol. xxxvii. p. 380; 'County of Salop,' vol. xxxviii. p. 516; H.M.S. 'Boadicea,' 'Bacchante,' vol. xl. pp. 325, 328; 'Alacrity' and 'Surprise,' vol. xl. p. 589; 'Kathleen Mavourneen,' vol. xli. p. 221; H.M.S. 'Mersey' and 'Rodney,' vol. xli. p. 449; 'Royal Prince,' vol. xli. p. 588; 'Westmoreland,' vol. xlii. p. 70; 'Gladiator,' vol. xliii. p. 104; 'County of York,' vol. xliii. p. 246; 'The Earl,' vol. xliv. p. 405; H.M.S. 'Orlando,' vol. xliv. p. 492; 'Elbe,' vol. xliv. pp. 656, 661; 'Islander,' vol. xlvi. p. 304; H.M.S. 'Barracouta,' vol. xlix. p. 467; 'Sunderland' (torpedo boat), vol. xlix. p. 32; 'Kaiser Wilhelm II.,' vol. l. p. 126; H.M.S. 'Barham' and 'Barracouta,' vol. l. p. 628; 'Normannia,' vol. l. p. 252; 'City of Vienna' (forced draught), vol. li. p. 398; 'Indra,' vol. li. p. 525; 'Scot,' vol. lii. p. 10; 'Ophir,' vol. lii. pp. 535, 591; H.M.S. 'Edgar,' vol. liii. p. 74; 'Infanta Maria Theresa,' 'Enging.,' vol. lvii. p. 805; 'Norman,' 'Enging.,' vol. lviii. p. 699; Russian 'Kherson' (Belleville), 'Enging.,' vol. lxii. p. 799; H.M.S. 'Powerful' (Belleville), 'Enging.,' vol. lxii. p. 403; 'Kaiser Wilhelm der Grosse,' 'Enging.,' vol. lxiv. p. 364; 'Cristobal Colon,' (Water-tube), 'Enging.,' vol. lxiv. p. 206; 'Engr.,' vol. lxxx. pp. 228, 523, 543 (pipes); 'Corsican,' 'Enging.,' vol. lxxxv. p. 46; H.M.S. 'Lord Nelson,' 'Enging.,' vol. lxxxvi. p. 421; 'Orsova,' 'Enging.,' vol. lxxxvii. p. 715: 'Nippore Yusen Kaisha,' 'Enging.,' vol. lxxxix. p. 210; 'Olympic,' 'Enging.,' vol. xc. p. 692; 'Recalde,' vol. xciii. p. 288.

The Scantlings of Boilers are in most cases determined either according to Lloyd's or the Board of Trade rules, to which a special chapter is devoted. Here it will not be out of place to indicate the methods to be employed for finding the best proportions, either as regards efficiency or weight. It is, of course, impossible to make any comparisons as regards cost, because the practices and available tools of different shops are not known; nor will any attempt be made to deal with the subject exhaustively, and only two simple comparisons will be carried out. In the one case it will be shown which form of joint and which percentage of riveting gives the lightest boiler shell, but only if built strictly according to Lloyd's rules. In the other case the lightest means of staying flat boiler plates will be discussed, and a few remarks will be added, showing how to proceed when dealing with such rules.

A very complete collection of rules and regulations by various Governments for boiler construction, &c., will be found in Delauny-Belville, 1886.

Riveted Joints.—Lap joints (fig. 436) and double butt-strap joints (fig. 437) are those most commonly used in boilers, though occasion-

Fig. 436

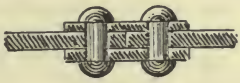


Fig. 437

ally Rowe's joint (fig. 438) is met with; it has been designed with the object of keeping the pitch of the outer row of rivets small, so

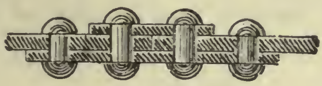


Fig. 438

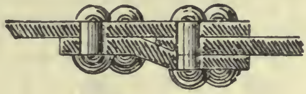


Fig. 439

as to increase the water-tightness. A similar object is evident in fig. 439. Single butt-strapped joints (fig. 440) are only used on furnaces

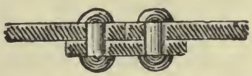


Fig. 440

Fig. 441

or to cover defective welds. Fig. 441 is a design intended to save weight.

When there are more than two rows of rivets in a lap joint, or more than three in a double butt-strapped one, it is desirable to arrange the rivets in such a manner that the percentage of strength of joint is the same for every row of rivets. It will be found that this is only the case in joints designed similarly to those shown in figs. 443 to 447. In these cases the quotient obtained by dividing the number of rivets in two adjoining rows is a constant value. If there are relatively fewer rivets in the inner row than in the outer one—as, for instance, in fig. 442—the percentage of strength along that line is too high; while if there had been relatively more rivets, it would have been too low. Let the quotient of the number of rivets in one row, when divided by the number in the adjoining inner row, be denoted by n; then B, the value of percentage of the joint—all rows having equal values—is found in the following table. A lower percentage than that given in this table should not be used, because the inner rows would be weaker than the outer ones, which gives a bad seam for caulking.

Table of Percentages of Riveted Joints.

m	$1\frac{1}{4}$	$1\frac{1}{3}$	$1\frac{1}{2}$	$1\frac{2}{3}$	$1\frac{3}{4}$	2	$2\frac{1}{4}$	$2\frac{1}{3}$
Lap joint, 3 rows .	...	52·6	63·6[1]	71·0	73·7	80·0	84·2	...
„ „ 4 „ .	53·0	60·8	71·4[2]	78·1	80·5	85·7	...	...
„ „ 5 „ .	60·2	68·2	78·4[3]	84·7	86·5	...	...	...
„ „ 6 „ .	65·6	73·8	82·6	87·9	...	...	...	...
„ „ 7 „ .	70·5	77·9	86·6	...	...	...	...	...
„ „ 8 „ .	74·2	81·2	...	...	...	...	...	...
Butt „ 2 „ .	...	...	55·5	63·9	67·3[4]	75·0[5]	80·2	87·1
„ „ 3 „ .	...	57·8	70·4	78·4	81·4[6]	87·5[7]	...	...
„ „ 3 „ .	$1\frac{1}{2}$ in each inner row } 66·7			2 in each inner row } 83·33[8]			...	

[1] See fig. 455. [2] See fig. 446. [3] See fig. 447.
[4] This is the value for Rowe's joint, having two rows of rivets on each side of butt with equal numbers of rivets in each (fig. 438). [5] See fig. 443.
[6] This is the value for Rowe's joint, having three rows of rivets on each side of butt, there being 1, 1, and $1\frac{2}{3}$ rivets respectively in the 1st, 2nd, and 3rd rows (fig. 448). [7] See fig. 444. [8] See fig. 442.

The riveted joint most commonly used is shown in fig. 442. It has been explained (see p. 222) that the distance from the edge of the

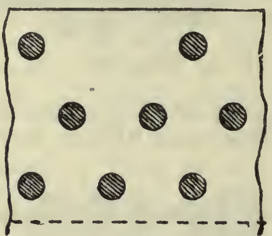


Fig. 442

Fig. 443

plate to the circumference of the rivet hole should not be less than its diameter. As regards the distances of the various rows of rivets from each other, a fairly uniform practice seems to exist of making the angular position of the rivets equal to about 50° to 55°, and in some works even 60° is customary. Some experiments (see p. 155) show that, except for an angle of about 60°, diagonal joints are no stronger than longitudinal ones. At this angle the longitudinal and diagonal pitches are equal. For smaller angles the percentage of the plate might be ascertained by measuring the zigzag line of all the outer rivets, subtracting all their diameters, and then dividing by the zigzag length. Few joints would then show the high percentage now claimed for them.

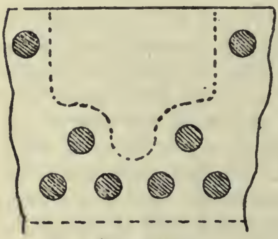


Fig. 444

It has also been shown (see p. 218) that if the rows of rivets are placed very wide apart, the stresses are not uniformly distributed amongst them. This tends to lower the value of an ordinary joint, but the unequal straining would be somewhat reduced by cutting away the metal along the dotted line (fig. 444). In the absence of any conclusive experiments or investigations, it is safest to adhere to the general practice mentioned above. In double-ended boilers it will often be found that the percentage of plate remaining between the holes of the screwed stays is less than that between the rivets of the longitudinal seams. The stays will then have to be pitched diagonally. But some allowance might be made for the absence of bearing pressure (see p. 215).

Lightest Joints.—For the purpose of comparing the weight of various joints, and also the amount of metal removed by drilling, which is a sort of measure of the labour expended in the shops, the above drawings of riveted joints have been made; in every case the holes are placed at an angle of 60°. The dimensions are in accordance with Lloyd's Boiler Rule for a steel plate 1 in. thick, the rivet diameters not being less than 1 in., and the percentage of the plate and of the rivet sections being equal. The estimates are contained in the following table:—

Comparison of various Riveted Joints.

Type of Joint	Percentage	Rivets			W.P. of a 12-ft. Boiler	Weight per 1 ft. of Shell			Boiler Shell Weight / W.P. Length
		No. per Pitch	Diam.	Pitch		3 Joints including Rivet Heads	Shell Plate and 3 Joints	Weight of Rivets	
			Inch	Inch	lbs.	lbs.	lbs.	lbs.	
Lap . fig. 445 .	70	3½	1	3·33	126	93	1643	39	13·0
,, . ,, 446 .	77	5	1	4·35	142	114	1664	43	11·8
,, . ,, 447 .	82·9	7¼	1	5·84	150	174	1723	46	11·4
Butt . ,, (1) .	74·5	2½	1	3·92	145	146	1696	37	11·7
,, . ,, 443 .	77·8	3	1	4·50	151	155	1705	40	11·3
,, . ,, 442 .	85·4	5	1	6·85	166	241	1791	42	10·8
,, . ,, 444 .	89·1	7	1	9·16	173	306	1856	48	10·7
,, . ,, 448 .	81·4	3¾	1·04	5·60	152	259	1809	39	11·9

[1] Like part of fig. 445.

The last column contains the quotients obtained by dividing the total weight of 12 ins. of length of the boiler shell by the respective working pressures, as found by the rules; these are convenient measures of the efficiencies of the various joints. It will be seen that the quadruple lap joint (fig. 446), though simpler, is almost as good as the double-riveted butt joint (fig. 443), and that the common butt-strap joint (fig. 442) is nearly equal to the one shown in fig. 444. This order is slightly changed if the circumference of the shell is made up of only one or two instead of three plates. Of course, in works where the machines are incapable of

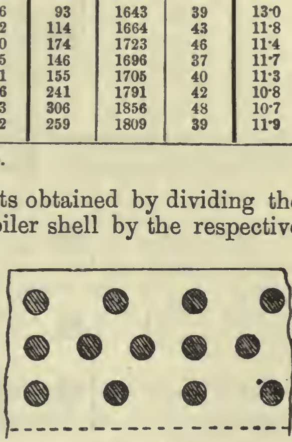


Fig. 445

dealing with plates of more than a certain thickness it may be necessary to use joints of the very highest percentage, and then the one sketched in fig. 444 is undoubtedly the best (see p. 218).

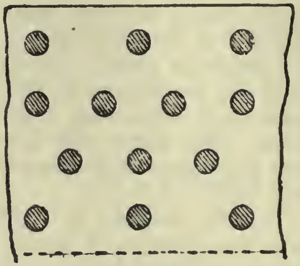


Fig. 446

Best Arrangement of Staying Flat Plates.—A comparison similar to the above can be carried out for flat plates. Here it cannot be a

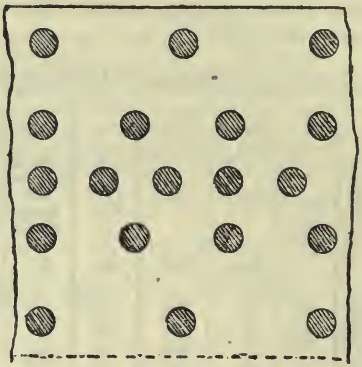


Fig. 447

question of variation of pitch, for it is quite clear that the total weight of stays is a constant quantity, while the weight of the flat plates

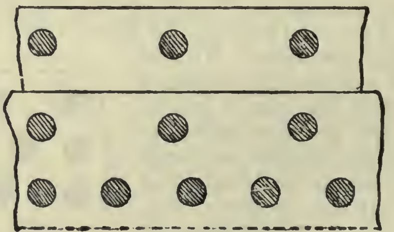


Fig. 448

varies as the thickness, or inversely as the pitch of the stays, and by placing these very close together the plates could be made quite thin.

In the following table the pitch of the stays in the combustion chamber plates is 10 ins. each way, the working pressure is 100 lbs., and the diameter of the screwed stay $1\frac{1}{2}$ in.

Table of Total Weight of Flat Combustion Chamber Plates

—		Plate Thickness	Total Weight of 100 sq. ins., including Nuts, &c.
		Inch	lbs.
By Lloyd's Register Boiler Rules	stay ends riveted .	·625	18·1
	„ „ nutted .	·562	16·7
By Board of Trade Boiler Rules	„ „ riveted .	·730	21·1
	„ „ nutted .	·580	17·2

In the following table the area of steam-space plating supported by each stay of $2\frac{1}{4}$ ins. diameter is 300 square ins.

Table of Weight of Flat Steam-space Plates

Conditions of Staying Flat Plates in Steam Space	Constants	Plate Thickness	Weight of 300 sq. ins. Plate	Weight of 300 sq. ins. Do. and Nuts, &c.
		Inch	lbs.	lbs.
Lloyd's Boiler Rules.				
Double nuts and no washers	175	·82	70·0	77·5
„ „ washers ($\frac{1}{3}$ P)	185	·80	68·0	78·5
„ „ „ riveted ($\frac{2}{5}$ P × $\frac{1}{2}$ T) .	200	·77	65·4	84·7
„ „ „ „ ($\frac{2}{3}$ P × T) .	220	·73	62·4	95·3
„ „ strip-riveted ($\frac{2}{3}$ P × T) . .	220 / 240	·71	61·0	112·8
„ „ doubling plate ($\frac{2}{3}$ T) . .	175	·61 / ·41	87·0	98·1
Board of Trade Rules.				
Stay ends nutted	112·5	·97	82·7	84·1
Double nuts and washers ($3d \times \frac{2}{3}$ T) . .	125	·91	78·0	91·7
„ „ „ riveted ($\frac{2}{3}$ P × T) .	187·5	·73	62·6	95·6
„ „ strips „ „ .	200	·71	60·5	111·9

Here, again, it will be seen that the mode of attachment which permits of the use of the thinnest plate does not lead to the lightest construction.

Doubling strips and plates are sometimes arranged as shown in fig. 449, the adjoining plates overlapping each other. In such cases the corners should be cut away, so as to reduce the length of the taper.

In the case illustrated in fig. 450, the plate has evidently been weakened by the seam. Doubling plates are also fitted to the front tube plate between the nests of tubes, and to the back plates of the boilers wherever the stays are too wide apart, and here, too, the laps are sometimes made very wide. Of course, the number or the sectional area of the adjoining stays or stay tubes must be increased to bear the extra load. Angle irons are also fitted, but usually in such a manner that they give no support.

These few suggestions about the design of certain parts of boilers, so that they shall be as light as possible, can readily be extended to

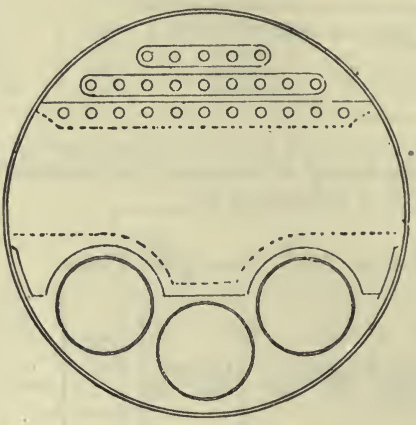


Fig. 449

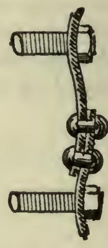


Fig. 450

other parts, and also to the problem of cheapness and despatch; but when the main principles have been settled, care must still be exercised in carrying them out, for other circumstances may come into play. This occurs in the following simple design of a steel boiler in accordance with Lloyd's Register Boiler Rules. It is intended to illustrate how the various rules can be applied, and the various references to the tables will show how to use them advantageously.

In order to shorten the explanations, practically no notice has been taken of mechanical difficulties, which are discussed elsewhere, and the design is necessarily imperfect in these respects.

The External Diameter of the shell is not to exceed 13 ft. = 156 ins., the heating surface is to be 2,000 square ft., and the grate surface 50 square ft., but the length of the grate is to be limited to $5\frac{1}{4}$ ft. All the vertical water spaces are to be 10 ins. wide, and the steam-space stays are not to be placed closer together than 16 ins. The water space round each tube is to be 1 in., round the furnaces 4 ins., and above them $7\frac{1}{2}$ ins. Working pressure, 160 lbs.

The Furnace Diameter is found by dividing 50 square ft. by $5\frac{1}{4}$ ft., which gives 114 ins. of furnace fronts. Two furnaces would be too large, but three of 38 ins. internal and about $39\frac{1}{4}$ ins. external diameter will be convenient (see pp. 100, 309, and table, p. 354).

The Tube End Spaces have to be estimated, as explained on p. 307. The internal diameter of the shell is about 154 ins., and the water spaces, as explained on p. 307, are $9\frac{1}{2}$ ins. at the wings, 9 ins. between the nest of tubes, and 7 ins. above the furnaces. The steam space has been made one-third of the boiler diameter.

Water Spaces (p. 313)	*a*	*b*	*c*	*e*	*f*	*g*	Total
Water spaces, inches Multiples	4 ·7	4 ·6	7 ·2	7 ·25	9·5 ·4	9 2 × ·25	— —
Products	2·8	2·4	1·4	1·7	3·8	4·5	16·6

Then $\Sigma(T) = \cdot 45\,(154 - 39{\cdot}25 - 16{\cdot}6)^2 = 4{,}330$ square ins. This is the available tube end area.

Tubes.—If the tubes are made $3\frac{1}{4}$ ins. diameter, then each one will occupy an end space of $(4\frac{1}{4})^2 = 18{\cdot}1$ square ins. Dividing this into 4,330 gives 240 as the necessary number of tubes. The tube surface will be about 80 per cent. of the total—say, 1,600 square ft.—and therefore each tube must have a surface of 6·7 square ft., which necessitates that its length should be 7·9 ft. (see table, p. 313). If 3-in. or $3\frac{1}{2}$-in. tubes had been decided upon, these numbers would have been respectively 272 and 7·5 ft. and 215 and 8·1 ft. These 240 tubes should be arranged in bundles of as nearly as possible equal numbers (78), and if possible in such a manner that the number of vertical and horizontal rows are all odd numbers, for then the staying is very much simplified.

The circular line A B (fig. 451) should be drawn; it marks off the boundary for the centres of the extreme tubes. Its radius is $66\frac{5}{8}$ ins. Measurement or calculation will show that after deducting the central water spaces there remain 121 ins. for the horizontal spacing of the tubes, which, as they are placed $4\frac{1}{4}$ ins. apart, number 28. The lines C D and E F, 7 ins. above the furnace crowns, indicate the lower boundary for the tube spaces, while N M is the upper limit. The height between these lines is $34\frac{1}{4}$ at the wings and 52 ins. at the centre, equal to about 8 and 12 tube pitches. The tube nests can now be arranged as shown in fig. 451, viz. $8 \times 10 + 12 \times 7 + 8 \times 10 = 244$ (odd numbers of rows being impossible). Of these, 6 fall away at the four corners, leaving 3 more than required. This agreement is sufficiently close for all practical purposes, but before proceeding it is advisable to estimate the

Heating Surface in the Combustion Chambers, so as to be sure that the total of 2,000 square ft. is reached. The clear depth, according to p. 309, is 31 ins., and the heating surface other than that of the tubes will be about 380 square ft., which, added to $238 \times 6{\cdot}8 = 1{,}620$, is exactly 2,000 square ft. Had this result not been obtained, then the tube lengths would have had to be altered a little. The boiler length is 11 ft. 3 ins.

The Lengths of the Shell Plates can now be fixed, and the riveted joints arranged.

Longitudinal Joints and Shell Plates.—As explained in connection with the tables on p. 352, the numerals there given should be so selected that they are larger than the numbers of rivets in one pitch. In this case we will first choose 5 rivets, then the numeral selection is 4·98 and the percentage $85\frac{1}{3}$. With this percentage and Lloyd's formula for shell plates with steel of 27 tons we get 15·8 sixteenths of an inch and a pitch of 6·74 ins.; we naturally choose 1 in. plates and $6\frac{3}{4}$ in. pitch of rivets. On checking this it will be found to be in excess of the required pressure, and also that the $22\frac{1}{2}$ feet of riveted seams will require 334 rivets.

If we decide on 3 rivets we find that 78 per cent. joint leads exactly to a plate of $1\frac{1}{16}$ in. and that the numeral against this percentage is 3·03. Dividing by 3 and multiplying by $1\frac{1}{16}$, the size of rivets is found to be $1\frac{1}{16}$ and thus according to the first column the pitch is 4·83 ins., so that 280 rivets will be required in a length of $22\frac{1}{2}$ feet. The saving in the cost of riveting due to using 54 less (their total weight however is the same) will not balance the increased cost of the extra sixteenth of thickness in the shell plate. We can therefore decide for the high percentage joint and thinner plate.

The Thickness of the Front Tube Plate is determined either by the distances at M (fig. 451) across the wide water spaces, by the distance at N from the tubes to the shell, or by the mean pitch of the stay tubes. The latter dimension is the mean of the mean horizontal and of the vertical pitches, and is most easily determined by counting the number of tubes that form the four sides of a figure whose corners are occupied by stay tubes, then multiplying this sum by one-quarter of the tube pitches. In this design the tubes are placed $4\frac{1}{4}$ ins. apart, and the mean pitch can vary from $8\frac{1}{2}$ to $9\frac{9}{16}$, $10\frac{5}{8}$, $11\frac{11}{16}$, or $12\frac{3}{4}$ ins., requiring (see table, p. 360) that the plates should be either $\frac{5}{8}$, $\frac{11}{16}$, $\frac{3}{4}$, $\frac{13}{16}$, or $\frac{7}{8}$ in. thick. The plates across the $13\frac{1}{4}$ in. wide water spaces would have to be either $\frac{15}{16}$ or $\frac{7}{8}$ in. thick according to the method of attaching the stay tubes. In this design the latter thickness will be used, and the stay tubes have therefore to be arranged as shown in fig. 451, those at the circumference being nutted.

If $\frac{5}{8}$-in. plates had been adopted it would have been necessary to fit about 25 extra stay tubes, as well as two doubling plates, to the front plate, as shown in dotted lines, whereby the cost would have been raised and no weight saved.

The Back Tube Plate need only be $\frac{13}{16}$ in. thick, as the mean pitch of the stay tubes does not exceed $11\frac{11}{16}$ ins.

Stay Tubes.—The constant for estimating the thickness of the front tube plate near the water line is 140, giving a pitch of 13·1 ins. with the $\frac{7}{8}$ in. plate. The top stay tubes will therefore have to support half this height and $1\frac{1}{2}$ pitch, equal to $6\frac{3}{8}$ in. below their centre line; but the area thus found has to be diminished by the section of four tubes: $(6{\cdot}55 + 6{\cdot}37) \times 8\frac{1}{2} - 4 \times 3\frac{1}{4}^{\circ} = 77$ square ins. The corner stays at M have each to support $(6{\cdot}55 + 4{\cdot}25) \times (4{\cdot}25 + 6{\cdot}75) - 2\frac{1}{4} \times 3\frac{1}{4}^{\circ} = 100$ square ins., and those in the nests of the tubes 87 square ins. The respective loads on each will be 12,300, 16,000, and 13,900 lbs., which, according to the table on p. 358, require that the minimum thicknesses of the stay tubes shall be $\frac{1}{4}$ in. and 9 threads per in., $\frac{5}{16}$ in. and 9 threads, and $\frac{1}{4}$ in. and 12 threads. The latter being too fine a screw, all the tubes will be made $\frac{5}{16}$ in. thick with 9 threads per inch.

End Plates in Steam Space.—One of the conditions of this design is, that the steam-space stays must be pitched at least 16 ins. apart. As previously mentioned (p. 319), the lightest construction is obtained if they are secured by double nuts. The constant being 175, 1-in. end plates will be required (see table, p. 358). But a trial shows that this leads to an inconvenient distribution of stays. Of the various alternatives which suggest themselves, a plate $1\frac{1}{8}$ in. thick will be used, provided that the flanging plant is sufficiently powerful to deal with it.

The maximum permissible pitch of stays is now 18·8 ins., and allowing $1\frac{1}{8}$ in. for the thickness of flange, and 2 ins. for its radius, the maximum distance from the shell at which the stays may be placed is 22 ins. This boundary line is marked G H (fig. 451). The lower boundary line, G I, is then drawn 15 ins. above the centres of the top row of tubes. This height is found by taking the mean of the pitches found for a $\frac{7}{8}$-in. plate with 140 constant and $1\frac{1}{8}$-in. plate with 175 constant. The position of the stays can now be marked off, care being taken that none of them are stationed over a water space, and all of them must fall outside of the line I G H.

The Sectional Areas of the Steam-space Stays are calculated by drawing the line B J, which is the lower limit of the area supported by the shell: it is found by adding half the permissible pitch of 18·8 ins. to the flange and its radius. B K is the upper limit of the area supported by the stay tubes, and, as already shown, it amounts to 6·55 ins. A few vertical and horizontal lines are now drawn to mark off the areas supported by the various stays, and it is then found that the load on the upper left-hand one is about 22,000 lbs., and on all the others about 53,000 lbs. According to the table on p. 363, these loads would require 2-in. and 3-in. stays with 6 threads per inch, because the next smaller sizes, $1\frac{7}{8}$ in. with 11 and $2\frac{3}{4}$ ins. with 12 threads, are not advisable.

The Combustion Chamber Side and Top Plates can now be drawn as shown in thick black lines. The centre of the wing radius may be placed at O, in order not to have an awkward corner at L, but then a stay will have to be fitted at A, which is inconvenient, especially if a manhole is fitted at the front end. The corners at M are struck with a radius of $3\frac{1}{2}$ ins., which allows 1 in. for the inner curve of the flange and $\frac{5}{8}$ in. for its thickness. For the position of seams see figs. 321–7, p. 276.

Combustion Chamber Backs.—The centres of the furnaces, of the steam-space stays, and of the combustion chamber corners are transferred from fig. 451 to 452. The double boundary lines *a b c d* and *e f g l h* are then drawn. The outer one should be 3 ins. within the outer plates, so that the stay nuts, which are about 3 ins. over cants, will not have to rest on the curved part of the flange, which is assumed to be 1 inch. The inner boundary should be determined by theoretical considerations (see p. 179), but as the numerical results are unpractical, the line has been drawn $1\frac{1}{2}$ in. within the outer one, or $4\frac{1}{2}$ ins. from the side and top plates, so that the distance measured from the commencement of the flange to the edge of the nut is about $1\frac{1}{2}$ in. All the stays at the circumference should fall within these two lines, as shown in fig. 452.

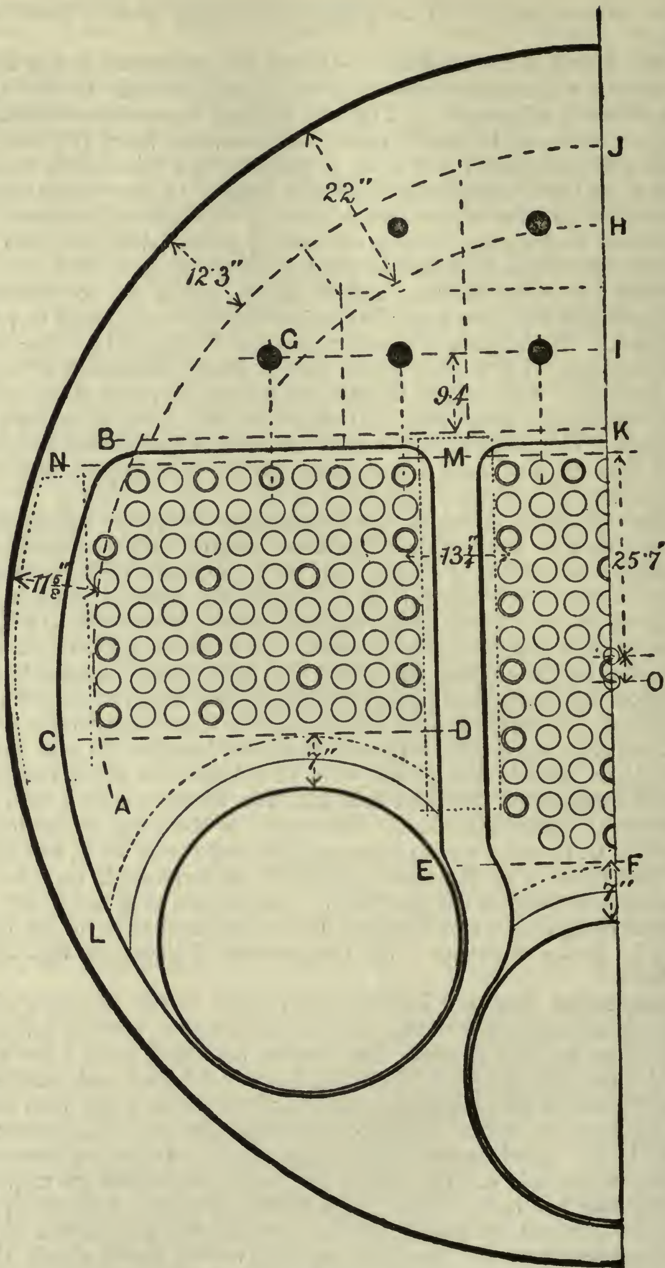


Fig. 451

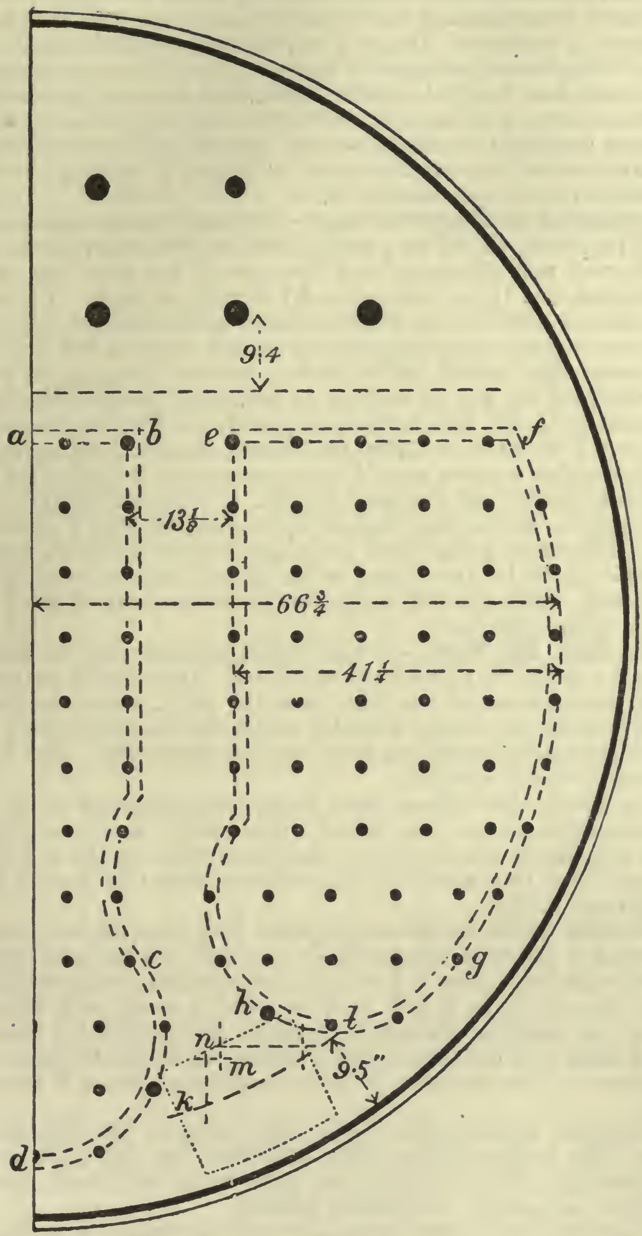


FIG. 452

The maximum and minimum distances for the horizontal pitches of the screwed stays are $23\frac{1}{2}$ to $26\frac{1}{2}$ ins. for the centre and $38\frac{3}{4}$ to $41\frac{3}{4}$ ins. for the wing chambers. Using $\frac{9}{16}$ in. plates with nutted stays, the pitch is $8\frac{1}{4}$. They can be arranged as shown. It will be noticed that the lower rows have had to be shifted $4\frac{1}{8}$ in., and that the uppermost and lowermost stays just touch the inner boundary line. The combustion chamber backstays should be normal to these plates and slightly out of normal to the boiler back plates. It is only a drawing office convenience to show them horizontal.

Sectional Area of Screwed Stays.—The maximum permissible pitch for $\frac{9}{16}$ in. plates, viz. $8\frac{1}{4}$ ins., having been adopted, the table on p. 355 can be used with advantage, and the sizes of the stays are at once determined, viz. $1\frac{1}{2}$ in. diameter and 7 threads per inch. The size of the screwed stays of the circumferences are determined separately. The steam-space stays support the plating down to 9·4 ins. below the lowest ones, so that the top row of screwed stays has to support the remaining 6·1 ins. as well as $4\frac{1}{8}$ ins. lower down. Multiplying this sum by $8\frac{1}{4}$ ins. (the horizontal pitch) and 160 lbs., the load on each stay is found to be 13,500 lbs., which, according to the table on p. 363, requires a screw stay $1\frac{5}{8}$ in. diameter with 8 threads per inch.

The stays at *b* and *e* support $10\frac{1}{4}$ ins. $\times$ ($6\frac{3}{4}$ ins. $+$ $4\frac{1}{8}$ ins.) $\times$ 160 $=$ 17,800 lbs., and require stays of $1\frac{3}{4}$ in. diameter with 8 threads per inch. The others below these two support loads of $10\frac{7}{8}$ ins. $\times$ $8\frac{1}{4}$ ins. $\times$ 160 $=$ 14,360 lbs., and have to be $1\frac{5}{8}$ in. diameter, with at least 11 threads. This size will be adopted for all the stays at the circumference except *b* and *e*.

The Back End Plate.—It will be seen that the pitch of the stays across *b e* cannot be reduced below $13\frac{1}{8}$ ins. According to the table on p. 358 the thickness of the plate must be $\frac{15}{16}$ in., unless the stays at the edge of the combustion chamber are placed at an angle (see p. 285), or unless doubling strips are fitted between these stays. But neither alternative will be adopted.

The permissible distance from the upper screw stays to the lower steam-space stays has to be found. As drawn it measures $15\frac{1}{2}$ ins., and a calculation shows that, for the above thickness ($1\frac{1}{8}$ and $\frac{15}{16}$ in.) it might have been made $16\frac{1}{8}$ ins., showing that this part is strong enough (see p. 323).

Doubling Plates at Bottom of Back End.—By drawing the line *k l*, which is the width supported by the shell, and amounts to $9\frac{1}{2}$ ins., and also the boundaries *k m n* and *n l* of the areas supported by the lower screwed stays, it will be found that a small area is unsupported; a doubling plate has therefore to be fitted, as shown in dotted lines, and the stays near its corners at *k* and at *h* have to be increased to $1\frac{3}{4}$ in. diameter, in order to take their share of this extra load.

There are various other details—the furnaces (pp. 309, 354), the girders, the combustion chamber side stays (pp. 29, 277), and the manholes (pp. 32, 178, 343)—but they have either been discussed elsewhere or present no difficult features.

When they have been calculated, nothing remains to be done but to complete the drawing by inserting the various seams and flanges, and then making out a list of the full sizes of the plates (pp. 225, 258), the lengths of the stay bars, and the weights of the various sized rivets (p. 250), ready for ordering. All the dimensions should be carefully checked, and if the boiler is to be built under the inspection of any society, the working pressure for each part should be calculated from the *invoiced dimensions* and with the help of the latest edition of their rules, and preferably without the use of any tables. If required, a tracing should also be submitted for approval, so as to obviate all risks of having to make alterations.

The two following tables, which contain the effective diameters and the effective sectional areas of screwed stays, may be of use when other stresses than those of the tables, pp. 362, 363, are to be adopted.

Effective Diameters of Screwed Stays.

Outside Diameters	Whitworth Screws		Number of Screw Threads per Inch						
			6	7	8	9	10	11	12
	Number of Threads	Effective Diameters at Bottom of Threads							
Inches		Inches	Inches	Inches	Inches	Inches	Inches	Inches	Inches
$\frac{3}{4}$	10	·62	...	...	...	...	·62	·63	·64
$\frac{7}{8}$	9	·73	...	...	...	·73	·75	·76	·76
1	8	·84	...	...	·84	·86	·87	·88	·89
$1\frac{1}{8}$	7	·94	...	·94	·96	·98	1·00	1·01	1·02
$1\frac{1}{4}$	7	1·07	...	1·07	1·09	1·11	1·12	1·13	1·14
$1\frac{3}{8}$	6	1·16	1·16	1·19	1·21	1·23	1·25	1·26	1·27
$1\frac{1}{2}$	6	1·29	1·29	1·32	1·34	1·36	1·37	1·38	1·39
$1\frac{5}{8}$	5	1·37	1·41	1·44	1·46	1·48	1·50	1·51	1·52
$1\frac{3}{4}$	5	1·49	1·54	1·57	1·59	1·61	1·62	1·63	1·64
$1\frac{7}{8}$	$4\frac{1}{2}$	1·59	1·66	1·69	1·71	1·73	1·75	1·76	1·77
2	$4\frac{1}{2}$	1·72	1·79	1·82	1·84	1·86	1·87	1·88	1·89
$2\frac{1}{8}$	$4\frac{1}{2}$	1·84	1·91	1·94	1·96	1·98	2·00	2 01	2·02
$2\frac{1}{4}$	4	1·93	2·04	2·07	2·09	2·11	2·12	2·13	2·14
$2\frac{3}{8}$	4	2·05	2·16	2·19	2·21	2·23	2·25	2·26	2·27
$2\frac{1}{2}$	4	2·18	2·29	2·32	2·34	2·36	2·37	2·38	2·39
$2\frac{5}{8}$	4	2·30	2·41	2·44	2·46	2·48	2·50	2·51	2·52
$2\frac{3}{4}$	4	2·43	2·54	2·57	2·59	2·61	2·62	2·63	2·64
$2\frac{7}{8}$	$3\frac{1}{2}$	2·51	2 66	2·69	2·71	2·73	2·75	2·76	2·77
3	$3\frac{1}{2}$	2·63	2·79	2·82	2·84	2·86	2·87	2·88	2·89
$3\frac{1}{8}$	...	...	2·91	2·94	2·96	2·98	3·00	3·01	3·02
$3\frac{1}{4}$	$3\frac{1}{4}$	2·86	3·04	3·07	3·09	3·11	3·12	3·13	3·14
$3\frac{3}{8}$	...	...	3·16	3·19	3·21	3·23	3·25	3·26	3·27
$3\frac{1}{2}$	$3\frac{1}{4}$	3·11	3·29	3·32	3·34	3·36	3·37	3·38	3·39
$3\frac{5}{8}$	...	...	3·41	3·44	3·46	3·48	3·50	3·51	3·52
$3\frac{3}{4}$	3	3·32	3·54	3·57	3·59	3·61	3·62	3·63	3·64
$3\frac{7}{8}$	...	...	3·66	3·69	3·71	3·73	3·75	3·76	3·77
4	3	3·57	3·79	3·82	3·84	3·86	3·87	3·88	3·89

Effective Sectional Areas of Stays.

Outside Diameters	Sectional Areas	Whitworth Screws		Number of Screw Threads per Inch						
				6	7	8	9	10	11	12
		Number of Threads	Effective Sectional Areas							
Inches	Sq. Ins.		Sq. Ins.	Sq. Ins.	Sq. Ins.	Sq. Ins.	Sq. Ins.	Sq. Ins.	Sq. Ins.	Sq. Ins.
¾	·44	10	·30	...	...	...	...	·30	·31	·32
⅞	·60	9	·42	...	...	...	·42	·44	·45	·46
1	·79	8	·55	...	...	·55	·58	·60	·61	·63
1⅛	·99	7	·70	...	·70	·73	·76	·78	·80	·81
1¼	1·23	7	·89	...	·89	·93	·96	·99	1·01	1·03
1⅜	1·48	6	1·06	1·06	1·12	1·16	1·19	1·22	1·24	1·26
1½	1·77	6	1·30	1·30	1·36	1·41	1·45	1·48	1·50	1·52
1⅝	2·07	5	1·47	1·56	1·63	1·69	1·73	1·76	1·79	1·81
1¾	2·40	5	1·75	1·85	1·93	1·99	2·03	2·07	2·10	2·12
1⅞	2·76	4½	1·99	2·17	2·25	2·31	2·36	2·40	2·43	2·45
2	3·14	4½	2·31	2·51	2·59	2·66	2·71	2·75	2·79	2·81
2⅛	3·55	4½	2·66	2·87	2·96	3·03	3·09	3·13	3·17	3·20
2¼	3·98	4	2·93	3·26	3·36	3·43	3·49	3·54	3·58	3·61
2⅜	4·43	4	3·32	3·67	3·77	3·85	3·91	3·96	4·01	4·04
2½	4·91	4	3·73	4·11	4·22	4·30	4·37	4·42	4·46	4·50
2⅝	5·41	4	4·17	4·57	4·68	4·77	4·84	4·90	4·94	4·98
2¾	5·94	4	4·64	5·05	5·18	5·27	5·34	5·40	5·45	5·49
2⅞	6·49	3½	4·94	5·56	5·69	5·79	5·87	5·93	5·98	6·02
3	7·07	3½	5·45	6·10	6·23	6·33	6·41	6·48	6·53	6·57
3⅛	7·67	...	...	6·66	6·80	6·90	6·99	7·06	7·11	7·15
3¼	8·29	3¼	6·41	7·24	7·39	7·50	7·59	7·65	7·71	7·76
3⅜	8·95	...	...	7·85	8·00	8·12	8·21	8·28	8·34	8·39
3½	9·62	3¼	7·58	8·48	8·64	8·76	8·85	8·93	8·99	9·04
3⅝	10·32	...	...	9·14	9·31	9·43	9·53	9·60	9·67	9·72
3¾	11·04	3	8·67	9·82	9·99	10·12	10·22	10·30	10·37	10·42
3⅞	11·79	...	...	10·53	10·71	10·84	10·94	11·03	11·15	11·15
4	12·56	3	10·03	11·26	11·44	11·58	11·69	11·77	11·85	11·90

CHAPTER X

STEAM PIPES

STEAM pipes are, by the Boiler Explosions Acts, classed as boilers, and now that the majority of pipes are being made of steel, they have so much in common with boilers that they deserve to be mentioned here.

The function of steam pipes is to convey steam from boilers to engines with as little loss as possible either of pressure due to friction or of volume due to radiation. Steam pipes should be safe, which means that the material should be good and the thicknesses ample for the intended pressure, but they should not be so rigid or heavy as to cause fracture, and the general arrangement should be such that water will not lodge in them and cause water hammers and explosions.

The friction of steam flowing in pipes has not been the subject of many experimental researches, the most reliable deductions having still to be drawn from experiments with air and water, allowance being made for differences of density. In short pipes the chief resistance is due to the imparting of a certain velocity to the steam, and is equal to a height of fluid of the steam density $h = v^2 : 2g$, where h is the pressure per square foot divided by the density in cubic feet. The velocity in this formula is intermediate between that at the centre and at the circumference of the pipe. Let Q be the weight of steam in pounds per hour, and a the sectional area of the pipe in square inches, then the velocity v in feet per second is $C \times Q \times \text{T.abs.} : a \times \text{P.abs.}$, where C is 0·121 for uniformly distributed velocity and about 0·100 in practice. The loss of pressure in pounds per square inch would therefore be about

$$\Delta P = Q^2 \times \text{T.abs.} : a^2 \times P \times 540{,}000.$$

Here a is the smallest sectional area of the pipe or any of its valves. When there are several restrictions in a pipe, ΔP increases to a certain extent, allowances having to be made for the several changes from high to low velocities and *vice versâ*, a reducing coefficient of 0·6 being applied at each change. It is claimed for certain steam sluice valves that if the approaches to and departures from a comparatively small hole are properly shaped, then the loss due to a change from slow to high velocity is regained by the change from high to low. This is a subject which turbine builders are now studying with the greatest attention.

In addition to the above-mentioned loss there is the surface friction of the pipe. If the velocity of the steam were the same all over the section of the pipe, the friction would be proportional to the square of the velocity and to the total internal surface of the pipe. In practice the case is not so simple, and at the point where the steam enters the pipe, or immediately behind a valve or other obstruction, the steam is thoroughly churned up and the surface velocity may be equal to or in excess of the mean velocity, but at some distance from this point there will be a great difference between the central and the surface velocities, so that the friction per square foot of surface decreases with the length of the pipe. This decrease is not the same under all conditions but is affected by the average velocity; for if this is low the difference between the centre and surface velocity will be great, whereas if it is high the difference, due to whirling will be slight, and thus the length influence diminishes with increasing velocity. These remarks apply to steady flows of steam; for intermittent flows, as is mostly the case in pipes on steamers, the flow is a pulsating one, and the behaviour of the steam will be as follows. As soon as the slide valve is open there will be a violent rush forward of the core of the steam; this rush, on account of the viscosity of steam, will affect the circumferential layers of steam, which will attain a maximum velocity at the instant when the slide valve is shut, and together with the core steam, will be dammed up and the pressure increased near the valve. This should cause a slight flow of the core away from the valve. It will thus be seen that under these conditions the steam pipe is practically reduced in sectional area by the circumferential layers of fairly stationary steam, and it is not improbable that the enlarging of a diameter of a pipe merely increases the thickness of the inert layers, and may be of less advantage than is generally imagined.

As already mentioned, experiments on this subject are scarce, and practical experiences are likely to be misleading unless account be taken of the velocity of pressure waves in steam in relation to the effective lengths of pipes. The average velocity of pressure waves in saturated steam seems to be about 1,400 ft. per second, no matter what the pressure. If the engine is running 300 revolutions a minute or 10 strokes a second, the steam pressure pulsation waves in the steam pipe will have lengths of 140 ft., so that if in one ship the steam pipe is 70 ft. long there will be temporary reductions of pressure at the slide valve while opening, due to the reflected suction waves which were created by previous openings of the other steam part. If, however, the pipe were to be made 100 ft. long, the pressure wave due to the damming up of steam after the closing of a port, would travel to the boiler and be back in time to add a little pressure to the inflowing steam. A comparison of two such results would therefore lead to the erroneous conclusion that there is less friction in a long pipe than in a short one.

The following table contains the names of those who have experimented on this subject. Fuller details are to be found in 'Deut. Ing.,' 1908, vol. 52, p. 82. The same volume, p. 481, and a few subsequent numbers, contain Mr. C. Eberle's experiments and deductions, amongst which may be mentioned that with superheated steam there is a marked drop of temperature from the centre to the circumference of a pipe amounting to as much as 70° F. in a 3-inch pipe. With the exception of Mr. Eberle's experiments on steam all the rest deal only with the friction of air.

Fritzsche has analysed previous results and compared them with his own experiments, arriving at the following expression—

$$\Delta p = p \, . \, \Psi \, . \, l \, . \, v^2 : d \, . \, T_c,$$

where p and Δp are the pressure and loss of pressure, l and d are the length and diameter in metres and millimetres, T_c is the absolute temperature, and v is the velocity in metres per second, while Ψ is a coefficient for which a table has to be consulted; it is about 0·020 for customary pressures and dimensions and 0·000,08 for feet and inches.

In practice a velocity of about 8,000 ft. per minute is customary except with short cut-off in the engine when the mean velocity may be 10,000 ft. For turbine supply the mean velocity is about 7,000 ft.

Experiments on Air Friction in Pipes.

Date	Name	Pipe			Air or Gas	
		Material	Diameters	Lengths	Velocities	Pressures
			ins.	ft.	ft. per sec.	Atmos.
1819	Gerard	C. and W. Iron	0·6-3·2	23-2,050	290-1,180	1
,,	G. G. Schmidt	Glass	0·9	0·5-3·3	?	?
1823	Koch	Brass	0·25-0·55	82-260	490-1,870	1
1829	d'Aubuisson	Tin plates	1·0-4·0	30-1270	49 390	1
1837	Buff	Lead	3·3-4·2	2·6-35	98-490	1
1845	Pecqueur	Lead	0·4-1·2	13-222	780-3,330	2·5-4·5
1856	Weissbach	Glass, Brass	0·4-0·6	5·5-10·0	590-3,540	1
,,	,,	Zinc	1·0	33	590-1,770	1
1857	Rittinger	Riveted W. Iron	8·2	70-200	300	1
1861	Blochmann	Drawn W. Iron	0·6-1·0	?	4 83	1
1863	Arson	C. Iron, Tin plate	3·2-2·0	up to 1,000	up to 237	1
,,	,,	C. Iron	2·0	up to 1,000	up to 237	1
1878	Stockalper	W. Iron	6·0	1,700	93-225	3·7-5·6
,,	,,	W. and C. Iron	8·0	15,000	93-225	3·7-5·6
1879	Devillez	C. Iron	2·8-5·0	560-3,200	40-590	1·8-5·6
1884	Althans	C. Iron	14·3	30	69-153	1
1890	Meissner	Zinc	10·4-20·0	92-650	27-116	1
,,	Riedler	C. Iron	12·0	11,000-54,300	53-170	6·3-8·0
1892	Lorenz	C. Iron	4·0	98	155-185	6·7-6·8
,,	Ledoun	W. Iron	1·8-4·0	97	147-1,630	1·4-5·7
1897	Flieger	Smooth-bored	0·2	0·6	1,570-5,200	1-2·3
,,	Zeuner	Smooth brass	0·2	13	1,180-4,130	0·75
1904	Rietschel	Copper	0·9-4·0	28-78	59-295	1
,,	,,	Zinc	2·0-4·0	28-78	59-295	1
,,	Brabbée	Riveted W. Iron	12·0-31·5	980-2,250	59-333	1
,,	Fritzsche	W. Iron	1·0-1·5	51	49-114	0·2-11·1
1908	Eberle	W. Iron	2·75-5·9	87	137-1,445	3-10

Radiation Losses of steam pipes.—Seeing that the loss of pressure due to friction diminishes inversely as the square of the diameter of the pipe, and that the loss of volume due to cooling and condensation increases with the diameter of the pipe, there must be a certain diameter for any length of pipe for which the sum of these two losses is a minimum. To ascertain this best diameter Chr. Eberle ('Deut. Ing.,' 1908, vol. 52, p. 481, etc.) carried out elaborate experiments on these two losses, both with saturated and with superheated steam. Unfortunately he could draw no definite conclusions as to the best diameters because his results are too complicated, but some of them are extremely interesting and deserve mentioning. Thus he finds that water-catchers collect more water when the steam is stationary than when it is flowing, which would be in contradiction to Mr. Jordan's experiments (p. 118), were it not that this can be attributed to inefficiency on the part of the water-catchers. On making the necessary allowances, Mr. Eberle could detect no velocity influence as regards radiation losses, or rather it was masked by the temperature effect, all the high velocity tests having been carried out with much hotter steam that the low velocity tests. One deduction to be drawn from these tests by interpolation is that at about 340° C. the heat loss for stationary superheated steam would probably be 3·3 B.T.U. per sq. foot of wrought iron surface per hour for 1° C. difference of temperature, and practically the same value was found for saturated steam at the same temperature, so that this loss is independent of the condition of the steam.

Safety of Steam Pipes.—As long as steam pipes were made of copper, a metal which is easily fashioned into bends, the designing of steam-pipe arrangements presented no serious difficulty; now however that mild steel is being used, the severe stresses which it may set up must be guarded against, for although it can be bent much further than copper without permanent set, it is nevertheless much stronger. It is therefore necessary to enquire carefully into the causes which have up to the present led to failures of steam pipes on steamers.

Water-Hammer.—A very complete analysis of about 100 cases of steam-pipe explosions due to water-hammer has been carried out by the author (M.S.U.A., 1908). These include the following Board of Trade reports on explosions on steamers: Nos. 1398; 589, 929, 1437; 926; 468, 580, 1056, 1109, 1295, 1515, 1582, 1738; 572, 1048, 1537, 1747; 512, 933, 986, 1264. Of these the last three reports do not contain sufficient information to permit of the true causes of the explosions being discovered.

Water Pockets.—Nine other cases, No. 468 to 572, were due to

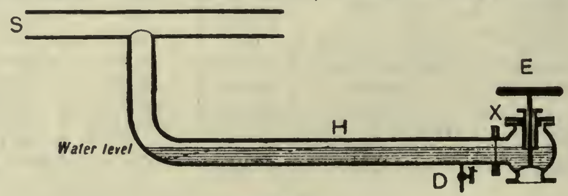


Fig 453.

the sudden admission of steam, S, into ranges of pipes containing water and shown diagramatically in fig. 453. The effect of this admission of steam was that a most rapid condensation took place, and the resulting steam blast along the surface of the water raised waves, as shown in dotted lines at H. Instantly the steam space from H to E would be converted into a vacuum, the wave would be shot towards E and the valve would explode at X. Such explosions could not occur either if there were no water in the pipe, or if the valves or pipe joints had leaked and had admitted air, which, on admitting steam, would form an effective cushion. To guard against this class of mishaps, which, as already stated, form the majority of marine-pipe explosions, it might be well to provide steam pipes with valves which would admit air whenever there is a vacuum. Most automatic steam traps are arranged to admit air when cold, but they do not appear to be used on steamers.

Steam-Pipe Drain Cocks.—The above-mentioned explosions would, of course, not have occurred if the drain cock D had been opened some considerable time before steam was admitted, but the opening of drain cocks with steam in the pipes is not unattended with danger, as will be seen from the following further cases, Nos. 1398, 589, 929, see fig. 454. The original water-level in the steam pipe is marked in the diagram. Steam was admitted at S, while the drain cock at D was open; the water-level gradually sank till the condition sketched

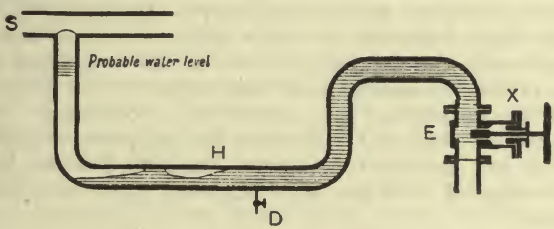


FIG. 454

in fig. 453 was attained; a wave and a vacuum were formed and an explosion occurred at H. In these cases the valve E was shut, but the same results would have occurred if it had been opened instead of the drain D. In Nos. 1437 and 926 the conditions were the same as the above, but the pipe was too strong at H and did not burst there; the pressure wave travelled to the valve E, and, due to reflection, its intensity was doubled locally and led to a fracture at X.

Submerged Valves.—Explosions Nos. 1048 and 1747 were due to the opening of submerged valves, like E in fig. 453, through which steam entered, the water-level probably standing higher than shown. If or when the steam pressure under this valve was greater than that above, it would lift this valve, and the steam coming in contact with cold water would immediately be condensed, the water would bang against the valve and do the damage. It is practically impossible to prevent this class of explosions if engineers omit to drain such pipes. Case No. 1537 is probably unique. The port and starboard boiler were connected by a cross pipe; the ship had a list to starboard, and as the port valve was open, steam from that boiler had condensed and was resting on the starboard valve. Then the vessel took a list to port, and while the cold water was flowing to port the rush of steam raised a wave which shattered the starboard valve.

Air in Pipes.—No. 512 is a case where an explosion seems to have occurred in spite of the presence of air in the pipe. Imagine in fig. 454 the lower portion of the pipe to be full of water with steam pressure at S, and air under compression from E to the down bend; then on suddenly opening the valve E this air would rapidly escape, and the following plug of water would close the valve or choke it and smash its chest.

Accidents Ashore.—Ashore, where steam pipes are very long and arrangements are sometimes very complicated, explosions which are almost inexplicable occur. They have all been dealt with in the Manchester Steam Users' Association Memorandum for 1908, and the following are here mentioned as they may occur on steamers. Nos. 1360, explosion of a downward sloping pipe; in 1622 and 1678 the pressure wave travelled a long way through a superheater and caused the explosion due to the reflection of the pressure wave. No. 342 was due to the starting of the engine; Nos. 1571, 1465, 1658, 1669, 1646 are complicated pipe arrangements which may occur on steamers. In No. 1283 an isolating valve seems to have been partly to blame for the explosion. In No. 1125 the opening of one submerged valve resulted in an overflow of water into the main steam pipe, and the explosion of another as yet closed valve. In Nos. 1162 and 1274 the rush of steam from one or more boilers seems to have stowed or dammed water into unused portions of the range, and a subsequent change of flow of steam caused this water to flow and to bring comparatively cold portions into contact with hot steam. Amongst over 100 explosions enquired into there was only one wrought-iron pipe, No. 1636, and one mild-steel pipe, No. 1360, but this latter one was of basic material. All the other exploded pipes were either cast iron or copper.

Steam-Pipe Materials.—It would thus seem that wrought iron or mild steel is a very suitable material for withstanding the severe water-hammer blows to which steam pipes are sometimes subjected, and since the use of these materials has been sanctioned on steamers, steam pipes are often made of them. Nevertheless, even with these materials accidents can occur, and every precaution should be adopted to obviate them. These precautions are, that the boiler stop valve should whenever possible be placed at a higher level than the engine shop valve, so that any water in the pipe will drain towards the engine. This condition is rarely fulfilled on steamers. There should not be any pockets in these steam pipes in which water could lodge, and if these pockets cannot be avoided, they should be well drained. Nevertheless, drain cocks ought not to be opened when any of the boiler shop valves are open. The safe plan is to admit a little steam into the pipe, close the shop valve and then drain the pipe, repeating the operation until all the water has run out or until the pipe has drawn in air. The fitting of vacuum valves or automatic steam traps can do no harm, and may prevent an occasional explosion.

Fatigue of Steam Pipes.—The second important cause of steam-pipe explosion is want of elasticity to meet the relative movements of boilers and engines, which cause has been somewhat intensified by the use of steel pipes, and the natural tendency to dispense with costly expansion bends. Another danger associated with the rigidity of steel steam pipes is that a much greater strain is now thrown on shop valves than was possible when copper pipes were in use, and as a safeguard against this danger it would be well to use cast-steel valve cases as is now the usual practice ashore. Even then, however, there remains the danger of the pipes breaking close to their flanges, or their drawing out of the flanges if these are not properly designed. Therefore, before dealing with expansion arrangements for pipes their manufacture will be briefly reviewed.

Lap-Welded Steel Pipes are made by bending a plate having taper edges to the required cylindrical shape, heating it as a whole to a welding heat, and passing it through rolls which lap-weld it over a bullet-shaped plug. It is usual to repeat the heating and rolling so as to ensure a perfect weld. With this process there is danger of burning the steel unless the greatest care be taken to maintain the correct temperature over the entire length of the pipe; which means of course that pipes by certain makers are likely to be more uniformly reliable than those by others, who are less painstaking in regulating the heats of their furnaces. By this process the welding is done by roller pressure on a bullet-shaped thorn, the lap being at the top when the pipe passes into the rolls. It may however happen that one or the other lap slips or stretches a little, and then the joint forms a helical line on the pipe, which means that, instead of being subjected to a good vertical rolling pressure over its whole length, the lap-weld receives hardly any pressure where, during the rolling process, it was at the side of the pipe. Pipes having a spiral weld should therefore be unhesitatingly rejected. On account of the ease with which the mild qualities of basic steel can be welded there is a great temptation to use this material, but as it is liable to failures which as yet can only be explained as being due to the unreliability of this material, acid steel should be used where absolute reliability is aimed at.

Pipes of 12 ins. diameter and above are welded on short lengths of a few inches at a time, the heating being done as shown in fig. 423, p. 299. There seems to be very little reduction of thickness, if care be taken to have a reducing flame, and plates as thin as $\frac{1}{8}$ in. can be effectively lap-welded by this process.

Solid Drawn Pipes, as the name implies, are seamless ones, which means that, as they have not to be welded, comparatively hard steels may be used. There are said to be over 100 different processes of manufacture, the chief ones being as follows: Steel ingots are cast with a core hole and then drawn out. Until recently great difficulty was experienced in obtaining solid castings, the air which escaped from the core producing blow holes. Another process is to cast solid steel ingots, bore and draw them out. Steel is also cast into moulds which revolve rapidly; after a time a thick and sound shell forms, and the core is then run out and the ingot drawn out. Amongst the various methods of drawing out may be mentioned a comparatively recent invention in which the axes of two or more rollers are set at slight angles to the axis of the pipe; they revolve and roll round a bullet-shaped thorn over which the hot cylindrical ingot is slipped. The rollers reduce the metal to the required thickness, and at the same time drag it along the thorn. In those pipes the fibre is longitudinal. In the Maunesmann process these rollers are very long, and as they are necessarily taper their circumferential velocities vary, which leads to the hollowing of solid bars or the thinning of the cylindrical ingots. In these pipes the fibre is circumferential.

Bends in Steel Pipes.—Steel pipes have to be most carefully filled with dry sand which has to be firmly rammed down inch by inch; they are then slowly heated to redness, sufficient time being allowed for the core of the sand to become hot, and the pipes are then bent by means of halyards, windlasses, and ropes, or by turn-tables. The thinning of the metal at the outside of bends, and the further bending of parts which have already acquired the desired curvature is prevented by spraying these parts with cold water. As soon as the bending is completed and the spraying discontinued the heat in the sand in the pipes anneals them. Short bends are made by sawing a number of taper slots into them, bending them and electrically welding the closed-up slots.

Copper Pipes are usually brazed, two sheets of copper having been hammered to the desired shape and lap-jointed. Some pipes are solid drawn from hollow ingots, others are made by an electric process, copper being deposited on a revolving mandril over which agate rollers are constantly travelling in order to harden the metal; these and the solid drawn copper pipes can be bent cold after being filled with resin.

Steam Pipe Flanges.—The flanges for copper pipes are made of gun metal to which they are brazed. The fact that several copper pipes have worked satisfactorily for years and have then failed shortly after rebrazing suggests that occasionally injudicious heating during the brazing operations may do harm. Another form of flange is shown in fig. 455, the ends of the two pipe ends being flanged out and clamped together by means of two rings which have to be slipped over the pipes before flanging. Two cases of failures of these joints at the roots of the flanges are reported, No. 1921 (sheet copper), No. 1069 (electro deposited copper), and as the flanging was done cold the failures were not due to bad heat treatment (unless the flanges were annealed), but to severe straining.

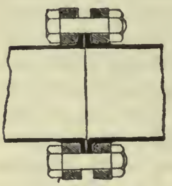


Fig. 455

Various modifications of this form of flange are in use on the continent for steel pipes, whose ends are generally thickened before flanging. In one modification each ring consists of four segments which overlap each other. Possibly this is a makeshift for a break-down, for these segments can be placed round the pipes after flanging.

Steel pipes of small diameter are often screwed into their flanges, see fig. 456, as is customary with gas and water pipes, but this is a dangerous practice if the pipes are subjected to severe bending strains as is the usual case on steamers, for these combinations are extremely weak at the flanges. It must be remembered that the bore of the pipe may vary, whereas the diameter of the cut threads is made to a standard gauge; if then by chance the pipe diameter should be $\frac{1}{8}$ in.

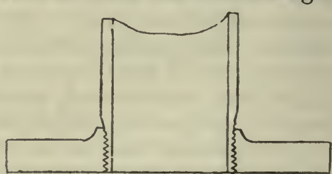


Fig. 456

larger than intended, the remaining metal at the threaded part is reduced by $\frac{1}{16}$ in. With pipes of large diameter there is the additional danger that, in order to ensure tightness much force has to be used for screwing the pipe into the flange, and if any caulking has to be done the circumferential compression stress in the pipe should tend to pucker the comparatively thin metal. At any rate large pipes are rarely fitted with screwed-on flanges.

When large pipes were first made of steel, their flanges were riveted on. This is unquestionably still the most reliable system, but it is clumsy, expensive, and the rivet heads offer some resistance to the flow of steam.

More recently the flanges have been welded on, but a few failures in which the pipes drew out of their flanges had shaken confidence in this method, which is however being restored since two improvements have been introduced; firstly, the outer edges of the flanged holes are recessed, fig. 457, so that the pipe end is bended over in addition to being welded, and a better weld than formerly is secured by making the flange hole smaller than the pipe, reducing the diameter of the end of the pipe before insertion in the hole, and enlarging both pipe and flange during the welding process, whereby a severer welding pressure than formerly is attained.

FIG. 457

Electric Welding if properly carried out is unquestionably a very reliable method of attaching flanges to pipes, but it lends itself to wash welding or surface welding. For instance, the flange shown in fig. 457, but without the recess, could be welded where the fillet touches the pipe and where the pipe appears on the flat of the flange, but such a seam would soon open out. The more effective method of electric welding is shown in fig. 458. The flange F has no fillet but is cut taper along the line *ab*, and placed close to the end of the pipe P, and the V-shaped hollow between flange and pipe is then slowly filled up with best Swedish iron, shown as layers in the figure, by means of the electric arc. Cheap filling material should not be used because it may result in the cracking of a very costly seam, and care should be taken to hold the anode as far as possible away from the molten steel, in order that the carbon particles which are carried by the electric current may have time to burn instead of being transferred to the steel and making it hard. The weld should be lightly hammered while hot.

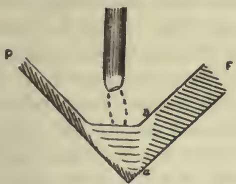


FIG. 458

Acetylene Welding of pipes is very similar to electric welding, except that in order to maintain the seam at a high temperature two flames should be used, one near *b*, which does the actual welding, and one at *a* for heating purposes only. The iron surfaces should be very clean, otherwise the added Swedish iron will not adhere to them.

Pipe Failures due to defects and to movements. Amongst about 100 reported failures of marine pipes due to movements or vibrations of engines and boilers, there were forty failures of brazed copper pipes, thirty-nine failures of solid drawn copper pipes, and fifteen of electro-deposited pipes. There was only one failure of a steel pipe, which was due to its having been screwed into its flange. There were amongst the brazed, solid drawn, and electro-deposited pipes respectively 4, 5, and 3 longitudinal fractures. They all occurred on bends and along what was the neutral fibre during the bending process, which seems to indicate that the severe shearing stresses to which these parts of the pipes were subjected had permanently injurious effects.

The remaining eighty-two failures were circumferential cracks generally close to the flanges, and they are nearly all attributable to straining and vibrations. In some cases the fractures occurred while the engines were racing, but there are also a few cases where pipes broke close to the flanges a few days after repair or renewal. Thus in the cases reported by the Board of Trade, Nos. 767 and 1355 broke respectively in two and three days after renewal ; Nos. 1160 and 1616 in six days, and Nos. 945, 1111, 1268, 1290, 1795, 1852, and 1904 broke in about one month after renewal.

Annealed Copper Pipes.—Some of the last-mentioned failures may have been due to injudicious heating. Thus Nos. 1922 and 1926 had been in use respectively for ten and six years without trouble, but failed respectively in three and two months after annealing. Other failures after annealing are : two cases after four years, two after two years, two after one year, one after eight months, one after seven months, two after six months, two after five months, one after four months, one after three months, one after fifteen days, or fifteen failures occurred on an average fourteen months after annealing, or rather after unsuitable annealing.

Some of the above-mentioned failures are almost certainly attributable to unsuitable copper, the old parts of the pipes breaking after new ends had been brazed on.

Deformations.—In the majority of cases the very large differences of movement between engines and boilers has been the cause of failures, and as this cause will have as injurious an effect on steel pipes as it has had on copper pipes, it is desirable that engine-builders should make careful observations of these relative movements and allow for them. Report No. 1296 mentions a horizontal movement of $\frac{15}{32}$ in. between the boiler and engine valves on a length of 8 ft. The pipe which was 5 ins. diameter had a very short bend and fractured seven months after being fitted, and again one month after being repaired (see p. 186 for stresses due to bending).

Expansion Bends.—The most common form of expansion bend is shown in fig. 459, but figs. 460 and 461 are possible modifications, the

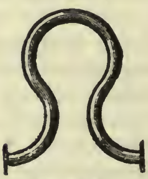


Fig. 459

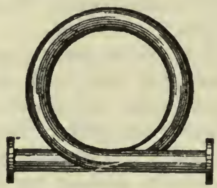


Fig. 460

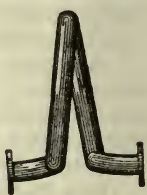


Fig. 461

later form being the most elastic. Assuming these pipes to be 6 ins. in diameter and the height 4 ft., the relative displacement of flanges in figs. 459 and 460 may be $\pm \frac{3}{4}$ in. for steel and $\pm \frac{3}{8}$ in. for copper. With bends as shown in fig. 461 these amounts may be doubled. The thickness of the metal of the pipes does not affect the permissible movement, which is strictly proportional to the ratio of the dimensions of the loops to the diameters of the pipes. Additional details will be found on p. 187. In some cases where expansion bends had to be provided for long lengths of pipes of large diameter, these were severed and attached to boxes, the gap being bridged by a bundle of small diameter tubes.

Expansion Joints.—Occasionally expansion glands are fitted, but seeing that about ten out of about 100 failed pipes were fitted with these glands they do not seem to have been of much use. The report numbers are 480*, 1011, 1113, 1172*, 1185*, 1187, 1207, 1214, 1666*, 1852*. Those marked with * deal with curved pipes, for which expansion glands are of course useless, because the steam pressure acting on the sectional area of the pipe, and with a leverage equal to the bend, would produce a larger movement, if unchecked, than could be taken up by the gland. Even with straight pipes the thrust due to the steam pressure is very severe. Thus with a 6-in. pipe and a pressure of 200 lbs. per sq. in. the thrust is nearly 3 tons. An equilibrium expansion gland has been devised, but does not seem to have been much used, in which the pipe, whose end is closed and whose circumference is perforated with slits over a short length, passes through two glands situated at the ends of a box to which the continuation of the pipe is attached. Swivel joints are not common either ashore or on steamers. They have been used in the Main Railway Station, Munich, see fig. 462. The most effective way of applying them would be to have two swivels at the ends of a bend, for then every movement would find relief. For marine purposes a more reliable arrangement than the spring Z would be necessary to keep the socket in place when the ship vibrates, the most convenient arrangement being no doubt to provide the end of the pipe with ribs, F, supporting a central sphere, C, against which a set screw, C,S, would have to press.

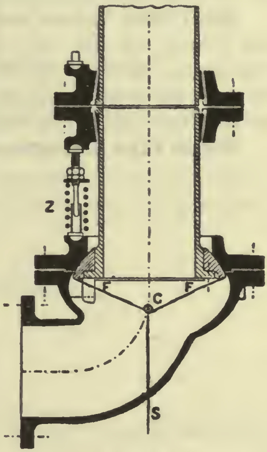


Fig. 462

Steam-Pipe Supports.—The more elastic a steam pipe is made, the more is it liable to be thrown out of line and otherwise strained when the ship rolls, when the boilers move or the engines vibrate. Seeing that some of these movements are independent of the ship's structure, they would produce very severe strains in the pipes if these were secured to the ship. The general tendency therefore is to dispense with hangers as much as possible, and to use them only where the pipes are very long. Considerable support would be afforded to the pipes without apparently any injurious effect if their weights were balanced as in fig. 463. The method does not appear to have been tried, and should be carefully considered before adoption. Spring supports should not be used on steam pipes; if long and elastic they must be secured against longitudinal movements either at one of their ends or near their middles. Here also springs should not be used as buffers, for they have caused breakages. In these cases (ashore) the rush of steam set up longitudinal vibrations which were in no way damped by two end springs; finally one of the elbow joints (cast iron on steel pipes) was fractured.

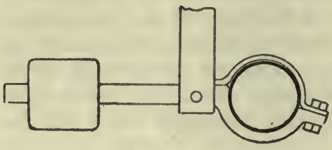


Fig. 463

Steel Stop Valves and Bends.—Marine engineers, although they are giving up the use of copper, will doubtless continue to prefer expansion bends to glands, but they should not overlook the fact that as steel pipes are stronger than copper ones, they throw heavy strains on cast-iron valves or bends.

Steam Pipe Arrangements, see p. 314.

CHAPTER XI

LLOYD'S REGISTER BOILER RULES

EXTRACTS FROM RULES PUBLISHED IN 1907–8.

Rules for Determining the Working Pressure to be allowed in New Boilers

Cylindrical Shells of Iron Boilers.—The strength of circular shells of iron boilers to be calculated from the strength of the longitudinal joints by the following formula:—

$$\frac{C \times T \times B}{D} = \text{working pressure},$$

where C = coefficient as per following table,
T = thickness of plate in inches,
D = mean diameter of shell in inches,
B = percentage of strength of joint found as follows—the least percentage to be taken.

For plate at joint $B = \dfrac{p - d}{p} \times 100$;

for rivets at joint $B = \dfrac{n \times a}{p \times T} \times 100$; with iron rivets in iron plates with punched holes,

$B = \dfrac{n \times a}{p \times T} \times 90$ with iron rivets in iron plates with drilled holes.

(In case of rivets being in double shear, $1{\cdot}75a$ is to be used instead of a),

where p = pitch of rivets,
d = diameter of rivets,
a = sectional area of rivets.
n = number of rows of rivets.

MEM.—In any case where the strength of the longitudinal joint is satisfactorily shown by experiment to be greater than given by this formula, the actual strength may be taken in the calculation.

Table of Coefficients.—Iron Boilers

Description of Longitudinal Joint	For Plates ½-in. thick and under	For Plates ¾-in. and above ½-in.	For Plates above ¾-in. thick
Lap joint, punched holes . . .	155	165	170
„ „ drilled „ . . .	170	180	190
Double butt-strap joint, punched holes	170	180	190
„ „ „ , drilled „	180	190	200

NOTE.—The inside butt strap to be at least ¾ of the strength of the longitudinal joint.

Cylindrical Shells of Steel Boilers.—The strength of cylindrical shells of steel boilers is to be calculated from the following formula :—

$$\frac{\mathsf{C} \times (\mathsf{T} - 2) \times \mathsf{B}}{\mathsf{D}} = \text{working pressure in lbs. per square inch,}$$

where D = mean diameter of shell in inches,

T = thickness of plate in sixteenths of an inch,

C = 22 when the longitudinal seams are fitted with double butt straps of equal width,

C = 21·25 when they are fitted with double butt straps of unequal width, only covering on one side the reduced section of plate at the outer lines of rivets (fig. 438, p. 321),

C = 20·5 when the longitudinal seams are lap joints.

If the minimum tensile strength of shell plates is other than 28 tons per square inch, these values of C should be correspondingly modified.

B = the least percentage of strength of longitudinal joint, found as follows:

For plate at joint $\mathsf{B} = \dfrac{p - d}{p} \times 100$;

for rivets at joint $\mathsf{B} = \dfrac{n \times a}{p \times t} \times 85$ where steel rivets are used,

$\mathsf{B} = \dfrac{n \times a}{p \times t} \times 70$ where iron rivets are used,

where p = pitch of rivets in inches,

t = thickness of plate in inches,

d = diameter of rivet holes in inches,

n = number of rivets used per pitch in the longitudinal joint,

a = sectional area of rivet in square inches.

In case of rivets in double shear $1{\cdot}75a$ is to be used instead of a.

NOTE.—The inside butt strap to be at least $\frac{3}{4}$ of the strength of the longitudinal joint.

NOTE.—For the shell plates of superheaters or steam chests enclosed in the uptakes, or exposed to the direct action of the flame, the coefficients should be $\frac{2}{3}$ of those given above.

Proper deductions are to be made for openings in shell.

All manholes in circular shells to be stiffened with compensating rings.

The shell plates under domes in boilers so fitted to be stayed from the top of the dome or otherwise stiffened.

Stays.—The strength of stays supporting flat surfaces is to be calculated from the smallest part of the stay or fastening, and the strain upon them is not to exceed the following limits, namely:—

Iron Stays.—For stays not exceeding $1\frac{1}{2}$ in. smallest diameter, and for all stays which are welded, 6,000 lbs. per sq. in.; for unwelded stays above $1\frac{1}{2}$ in. smallest diameter, 7,500 lbs. per sq. in.

Steel Stays.—For screw stays not exceeding $1\frac{1}{2}$ in. smallest diameter, 8,000 lbs. per sq. in.; for screw stays above $1\frac{1}{2}$ in. smallest diameter, 9,000 lbs. per sq. inch. For other stays not exceeding $1\frac{1}{2}$ in. smallest diameter, 9,000 lbs. per sq. in., and for stays exceeding $1\frac{1}{2}$ in. smallest diameter, 10,400 lbs. per sq. in. No steel stays are to be welded.

Stay Tubes.—The stress is not to exceed 7,500 lbs per sq. in.

Flat Plates.—The strength of flat plates supported by stays to be taken from the following formula:—

$$\frac{C \times T^2}{P^2} = \text{working pressure in lbs. per square inch,}$$

where T = thickness of plate in sixteenths of an inch,

P^2 = square of pitch in inches; if the pitch in the rows is not equal to that between the rows, then the mean of the squares of the two pitches is to be taken,

C = 90 for iron or steel plates $\frac{7}{16}$ thick and under, fitted with screw stays with riveted heads,

C = 100 for iron or steel plates above $\frac{7}{16}$ thick, fitted with screw stays with riveted heads,

C = 110 for iron or steel plates $\frac{7}{16}$ thick and under, fitted with stays and nuts,

C = 120 for iron plates above $\frac{7}{16}$ thick, and for steel plates above $\frac{7}{16}$ and under $\frac{9}{16}$ thick, fitted with screw stays and nuts,

C = 135 for steel plates $\frac{9}{16}$ thick and above, fitted with screw stays and nuts,

C = 140 for iron plates fitted with stays with double nuts,

C = 150 for iron plates fitted with stays, with double nuts and washers outside the plates, of at least $\frac{1}{3}$ of the pitch in diameter and $\frac{1}{2}$ the thickness of the plates,

C = 160 for iron plates fitted with stays, with double nuts and washers riveted to the outside of the plates, of at least $\frac{2}{5}$ of the pitch in diameter and $\frac{1}{2}$ the thickness of the plates,

C = 175 for iron plates fitted with stays, with double nuts and washers riveted to the outside of the plates, when the washers are at least $\frac{2}{3}$ of the pitch in diameter and of the same thickness as the plates.

For iron plates fitted with stays, with double nuts and doubling strips riveted to the outside of the plates, of the same thickness as the plates, and of a width equal to $\frac{2}{3}$ the distance between the rows of stays, C may be taken as 175, if P is taken to be the distance between the rows, and 190 when P is taken to be the pitch between the stays in the rows.

For steel plates, other than those for combustion chambers, the values of C may be increased as follows :—

$C =$ 140	increased to	175,
150	„	185,
160	„	200,
175	„	220,
190	„	240.

If flat plates are strengthened with doubling plates securely riveted to them, having a thickness of not less than $\frac{2}{3}$ of that of the plates, the strength to be taken from

$$\frac{C \times (T + t/2)^2}{P^2} = \text{working pressure in lbs. per square inch;}$$

where $t =$ thickness of doubling plates in sixteenths, and C, T and P are as above.

NOTE.—In the case of front plates of boilers in the steam space these numbers should be reduced 20 %, unless the plates are guarded from the direct action of the heat.

For steel tube plates in the nest of tubes the strength to be taken from

$$\frac{140 \times T^2}{P^2} = \text{working pressure in lbs. per square inch:}$$

where $T =$ the thickness of the plates in sixteenths of an inch,
$P =$ the *mean* pitch of stay tubes from centre to centre.

For the wide water spaces between the nests of tubes the strength to be taken from

$$\frac{C \times T^2}{P^2} = \text{working pressure in lbs. per square inch;}$$

where $P =$ the horizontal distance from centre to centre of the bounding rows of tubes, and
$C =$ 120 where the stay tubes are pitched with two plain tubes between them, and are not fitted with nuts outside the plates,
$C =$ 130 if they are fitted with nuts outside the plates,
$C =$ 140 if each alternate tube is a stay tube not fitted with nuts,
$C =$ 150 if they are fitted with nuts outside the plates,
$C =$ 160 if every tube in these rows is a stay tube, and not fitted with nuts,
$C =$ 170 if every tube in these rows is a stay tube, and each alternate stay tube is fitted with nuts outside the plates.

The thickness of tube plates of combustion chambers in cases where the pressure on the top of the chambers is borne by these plates is not to be less than that given by the following rule :—

$$T = \frac{P \times W \times D}{1750 \times (D - d)};$$

where P = working pressure in lbs. per square inch,
W = width of combustion chamber between plates in inches,
D = horizontal pitch of tubes in inches,
d = inside diameter of plain tubes in inches,
T = thickness of tube plates in sixteenths of an inch.

Girders.—The strength of girders supporting the tops of combustion chambers and other flat surfaces to be taken from the following formula :—

$$\frac{C \times d^2 \times T}{(L - P) \times D \times L} = \text{working pressure in lbs. per square inch;}$$

where L = width between tube plates, or tube plate and back plate of chamber,
P = pitch of stays in girders,
D = distance from centre to centre of girders,
d = depth of girder at centre,
T = thickness of girder at centre. All these dimensions to be taken in inches.

Wrought Iron.

$$C = \begin{cases} 6{,}000 \text{ if there is one stay to each girder,} \\ 9{,}000 \text{ if there are two or three stays to each girder,} \\ 10{,}000 \text{ if there are four or five stays to each girder,} \\ 10{,}500 \text{ if there are six or seven stays to each girder,} \\ 10{,}800 \text{ if there are eight stays or above to each girder.} \end{cases}$$

Wrought Steel.

$$C = \begin{cases} 7{,}110 \text{ if there is one stay to each girder,} \\ 10{,}660 \text{ if there are two or three stays to each girder,} \\ 11{,}850 \text{ if there are four or five stays to each girder,} \\ 12{,}440 \text{ if there are six or seven stays to each girder,} \\ 12{,}800 \text{ if there are eight stays or above to each girder.} \end{cases}$$

If the minimum tensile strength of girder plates is other than 28 tons per square inch, these values of C shall be correspondingly modified.

Circular Furnaces.—The strength of plain furnaces to resist collapsing to be calculated as follows :—

Where the length of the plain cylindrical part of the furnace exceeds 120 times the thickness of the plate, the working pressure is to be calculated by the following formula :—

$$\frac{1{,}075{,}200 \times T^2}{L \times D} = \text{working pressure in lbs. per square inch;}$$

where the length of the plain cylindrical part of the furnace is less than 120 times the thickness of the plate, the working pressure is to be calculated by the following formula :—

$$\frac{50 \times (300\,T - L)}{D} = \text{working pressure in lbs. per square inch;}$$

where D = outside diameter of furnace in inches,
T = thickness of plate in inches,
L = length of plain cylindrical part in inches, measured from the centres of the rivets connecting the furnaces to the flanges of the end and tube plates, or from the commencement of the curvature of the flanges of the furnace where it is flanged or fitted with Adamson's rings.

In the furnaces referred to below the formulæ given are applicable if the steel used has a tensile strength of not less than 26 nor more than 30 tons per square inch. If the material of furnaces has a less tensile strength than 26 tons per square inch, then for each ton per square inch which the minimum tensile strength falls below 26, the coefficient is to be correspondingly decreased by $\frac{1}{26}$ part.

The strength of corrugated furnaces made on Fox's, Morison's, Deighton's, or Beardmore's plan to be calculated from

$$\frac{1{,}259 \times (T-2)}{D} = \text{working pressure in lbs. per square inch.}$$

The strength of spirally corrugated furnaces is to be calculated from the following formula :—

$$\frac{912 \times (T-2)}{D} = \text{working pressure in lbs. per square inch;}$$

where T = thickness of plate in sixteenths of an inch,
and D = outside diameter of corrugated furnaces in inches.

The strength of Improved Purves's furnaces with ribs 9 inches apart, and of Brown's Cambered furnaces with ribs either 8 inches or 9 inches apart, to be calculated from the following formula :—

$$\frac{1{,}160 \times (T-2)}{D} = \text{working pressure in lbs. per square inch;}$$

where T = thickness of plate in sixteenths of an inch,
and D = smallest outside diameter of furnaces in inches.

The strength of the Leeds Forge bulb furnace is to be calculated from the following formula :—

$$\frac{1{,}259 \times (T-2)}{D} = \text{working pressure in lbs. per square inch;}$$

where T = thickness of plate in sixteenths of an inch,
and D = smallest outside diameter in inches.

The strength of Holmes' Patent Furnaces, in which the corrugations are not more than 16 inches apart from centre to centre, and not less than 2 ins. high, to be calculated from the following formula :—

$$\text{Working pressure in lbs. per square inch} = \frac{945 \times (T-2)}{D}$$

where T = thickness of plain portions of furnace in sixteenths of an inch,
D = outside diameter of plain parts of the furnace in inches.

Donkey Boilers.—The iron used in the construction of the fire boxes, uptakes, and water tubes of donkey boilers shall be of good quality, and to the satisfaction of the surveyors, who may in any cases where they deem it advisable apply the following tests:—

Thickness of Plates	To Bend Cold through an Angle of	
	With the Grain	Across the Grain
$\frac{5}{16}$	80°	45°
$\frac{6}{16}$	70°	35°
$\frac{7}{16}$	55°	25°
$\frac{8}{16}$	40°	20°

The material to stand bending *hot* to an angle of 90°, over a radius not greater than $1\frac{1}{2}$ time the thickness of the plates.

General Remarks about Boilers under Construction.—The surveyors will be guided in fixing the working pressure by the tables and formulæ annexed.

Any novelty in the construction of the machinery or boilers to be reported to the Committee.

The boilers, together with the machinery, to be inspected at different stages of construction.

All the holes in steel boilers should be drilled, but if they be punched the plates are to be afterwards annealed.

All plates that are dished or flanged, or in any way heated in the fire for working, except those that are subjected to a compressive stress only, are to be annealed after the operations are completed.

No steel stays are to be welded.

Unless otherwise specified, the rules for the construction of iron boilers will apply equally to boilers made of steel.

The boilers to be tested by hydraulic pressure, in the presence of the engineer-surveyor, to twice the working pressure, and carefully gauged while under test.

Two safety valves to be fitted to each boiler and loaded to the working pressure in the presence of the surveyor. In the case of boilers of greater working pressure than 60 lbs. per sq. in., the safety valves may be loaded to 5 lbs. above the working pressure. If common valves are used, their combined areas to be at least half a square inch to each square foot of grate surface. If improved valves are used, they are to be tested under steam in the presence of the surveyor; the accumulation in no case to exceed 10 % of the working pressure.

An approved safety valve also to be fitted to the superheater.

In winch boilers one safety valve will be allowed, provided its area be not less than half a square inch per square foot of grate surface.

Each valve to be arranged so that no extra load can be added when steam is up, and to be fitted with easing gear which must lift the valve itself. All safety-valve spindles to extend through the covers and to be fitted with sockets and cross handles, allowing them to be lifted and turned round in their seats, and their efficiency tested at any time.

Stop valves to be fitted so that each boiler can be worked separately.

Each boiler to be fitted with a separate steam-gauge, to accurately indicate the pressure.

Each boiler to be fitted with a blow-off cock independent of that on the vessel's outside plating.

The machinery and boilers are to be securely fixed to the vessel to the satisfaction of the surveyor.

SUMMARY.

The preceding rules are summarised in the following short table, in which the method has been carried out of indicating all such dimensions as are measured in inches by capitals, such as are measured in sixteenths of an inch by small letters, and all coefficients, &c., by black letters:—

C and **C′** = coefficients.

W P = permissible working pressure.

B and **B′** = percentage of joint respectively of plate and of rivets.

N = number of rivets included within one pitch of external row.

T and t = thicknesses of plates measured respectively in inches and in sixteenths of an inch.

P = pitches in inches of rivets or stays in flat plates, or tubes in tube plates.

P_c = distance apart of girders or cross pitch of stays in inches.

D = mean diameter of shells and diameter of furnaces in inches, measured as follows: for all plain furnaces, or made with ribs (Purves's), with flanges (Adamson's rings), or for Holmes's furnaces, the outside diameter of the plain cylindrical part is to be taken, and the thickness of the plates measured at these parts. For Fox's and Morison's corrugated furnaces the extreme external diameter is to be taken.

D and D_1 = effective external and internal diameters of plain or stay tubes, and effective diameter of rivets, or of stays, in inches.

L or L′ = length of plain cylindrical parts of furnaces measured respectively in inches or feet.

L = length of girders measured in inches = internal distance between tube and back plates.

H = depth of girders measured in inches.

A = sectional area of stays or stay tubes, or of rivets, in square inches.

Σ **A** = sum of areas of holes in tube plate in square inches.

Riveted Joints. (See appended Tables.)

Percentage of plate $\mathbf{B} = \frac{P - D}{P} . 100$

Percentage of rivets $\mathbf{B}' = \mathbf{C} . \mathbf{N} . \frac{A}{P . T}$, or $\mathbf{C}' . \mathbf{N} . \frac{D^2}{P . T}$.

TABLE OF COEFFICIENTS					
Materials of		Double Butt Straps		Lap Joints	
Plate	Rivet	C	C′	C	C′
Iron, punched	Iron	175·0	137·4	100	78·5
„ drilled	„	157·5	123·8	90	70·7
Steel „	„	122·5	96·3	70	55·0
„ „	Steel	148·7	116·8	85	66·7

Iron Boiler Shells. $\mathbf{W\,P = C\,.\,(B \text{ or } B')\,.\,\frac{T}{D}}$, or $\mathbf{C'\,.\,(B \text{ or } B')\,.\,\frac{t}{D}}$.

TABLE OF COEFFICIENTS												
Joint	Double Butt Straps						Lap joints					
Plates	C			C′			C			C′		
Thickness of Plates	½ in.	¾ in.	above	½ in.	¾ in.	above	½ in.	¾ in.	above	½ in.	¾ in.	above
Punched	170	180	190	10·61	11·24	11·87	155	165	170	9·68	10·31	10·61
Drilled	180	190	200	11·24	11·87	12·50	170	180	190	10·61	11·24	11·87

Steel Boiler Shells. $\mathbf{W\,P = C\,.\,(B \text{ or } B')\,.\,\frac{t-2}{D}}$.

TABLE OF COEFICIENTS C							
Tenacity of Steel, Tons	26	27	28	29	30	31	32
Lap joints	19·03	19·76	20·50	21·23	21·96	22·69	23·42
Butt straps of unequal widths	19·73	20·49	21·25	22·01	22·76	23·52	24·28
„ of equal widths	20·43	21·21	22·00	22·78	23·57	24·35	25·14

Furnaces. (See appended Tables.)

Plain furnaces when L exceeds 120 T:

$$\mathbf{W\,P} = \frac{1{,}075\,,200\,.\,T^2}{D\,.\,L} = \frac{4{,}200\,.\,t^2}{D\,.\,L} = \frac{350\,.\,t^2}{D\,.\,L'}.$$

Plain furnaces when L is less than 120 T:

$$\mathbf{W\,P} = \frac{50\,.\,(300\,T - L)}{D} = \frac{600}{D}\,.\,(1{\cdot}5625\,.\,t - L').$$

Patent and other Furnaces. $\mathbf{C'}\,.\,\frac{t-2}{D}$.

For measurements of D see p. 342.

TABLE OF COEFFICIENTS FOR STEEL OF 26 TONS AND MORE	C′
Corrugated flue, various types	1,259
Purves's ribbed flue	1,160
Farnley's spirally corrugated flue	912
Holmes's flue	945

STAYED FLAT PLATES. If pitches are equal $\mathbf{WP} = \frac{C \cdot t^2}{P^2}$

(See appended Tables.)

If pitches are unequal $\mathbf{WP} = \frac{2 \cdot C \cdot t^2}{P^2 + P_c^2} = \frac{2 \cdot C \cdot t^2}{\Delta_2}$, where Δ is the diameter of the greatest inscribed circle.

TABLE OF COEFFICIENTS—C	Iron	Steel
Stay ends riveted plates up to $\frac{7}{16}$ inch	90	90
„ „ „ above $\frac{7}{16}$ inch	100	100
„ nutted „ up to $\frac{7}{16}$ inch	110	110
„ „ „ above $\frac{7}{16}$ inch	120	120
„ „ „ $\frac{9}{16}$ inch thick and above	120	135
Double nuts	140	175
„ and washers ($\frac{1}{3}$ P × $\frac{1}{2}$ T)	150	185
„ and riveted washers ($\frac{2}{5}$ P × $\frac{1}{2}$ T)	160	200
„ „ ($\frac{2}{3}$ P × T)	175	220
„ and doubling strips ($\frac{2}{3}$ P × T) lengthways	190	240
Tube plate	140	140
Tube plate between nests of tubes	Beaded	Nutted
When there are two plain tubes between stays	120	130
„ is one plain tube between stays	140	150
„ every tube is a stay tube	160	170
Doubling plates $\mathbf{W\,P.} = C\,\frac{(2t + t')^2}{2 \cdot P^2}$, { t' is the thickness of the doubling plate in sixteenths of an inch		

$$\text{Tube plates, } \mathbf{WP} = \frac{1{,}600 \cdot (P - D'_1) \cdot t}{L \cdot P}.$$

STAYS. (See appended Tables.)

$$\text{STAYS.} \quad \mathbf{WP} = C \cdot \frac{A}{P \cdot P_c} \text{ or } C' \cdot \frac{D^2}{P \cdot P_c}$$

$$\text{Stay tubes.} \quad \mathbf{WP} = C \cdot \frac{A}{P \cdot P_c - \Sigma A}, \text{ or } C' \cdot \frac{D^2 - D_1}{P \cdot P_c - \Sigma A}.$$

TABLE OF COEFFICIENTS	C		C′	
	Iron	Steel	Iron	Steel
Screwed stays up to $1\frac{1}{2}$ inch effective diameter	6,000	8,000	4,712	6,283
„ above $1\frac{1}{2}$ inch effective diameter	7,500	9,000	5,891	7,068
Stay tubes	7,500	7,500	5,890	5,890

$$\text{GIRDERS.} \quad \mathbf{WP} = C\,\frac{H^2\,T}{L \cdot (L - P) \cdot P_e}$$

TABLE OF COEFFICIENTS	C	
Number of Stays per Girder	Iron	Steel
One	6,000	7,110
Two or three	9,000	10,660
Four or five	10,000	11,850
Six or seven	10,500	12,440
Eight or more	10,800	12,800

TABLES.

When used in the drawing office, it is strongly recommended that those parts of the following tables which are inapplicable to the particular works should be obliterated.

Table for Finding the Diameters of Rivets and of Pitches in Riveted Joints.

A short table will be found on p. 323 which shows the smallest permissible percentage for any form of joint, the condition being that when that particular percentage is adopted the strength of the joint shall be exactly the same for each row of rivets. If this percentage is exceeded, the inner rows will be stronger than the outer ones. If the joint is made of a smaller percentage, the inner rows of rivets will be the weakest and must be calculated separately.

Having decided on a particular percentage, the best dimensions of the joint can be ascertained from the following table, which contains the values of $\mathbf{N} . \mathrm{D} \div \mathrm{T}$ = number of rivets × diameter of rivets ÷ thickness of shell plate. Having found the number in the table against the desired percentage, it is only necessary to multiply it by the thickness of the plate, and then, dividing by the number of rivets within one pitch, their diameters are found. The percentage and therefore also the numeral should be so chosen that the latter is larger than the number of rivets in one pitch.

Example.—Butt-strap joint 1-inch steel plates, steel rivets, three rows, with two rivets in each inner row. Then the smallest percentage to be adopted for this joint is 83·33. Interpolating the values in the following table, it will be found that the value of $\mathbf{N} . \mathrm{D} \div \mathrm{T}$ is 4·28. Dividing this by 5, the number of rivets, then their diameters must be ·856. In the first column of the same table will be found the value of pitch ÷ rivet diameter = 6. Therefore the pitch is 6 × ·856 = 5·136 inches.

If it is desired to make the rivet diameter equal to the thickness of the plate, then a value must be selected where $\mathbf{N} . \mathrm{D} \div \mathrm{T} = 5$, i.e. $85\frac{5}{7}$ per cent.; then the pitch would have to be 6·98 inches. (See p. 347.)

Pitch ÷ Rivet Diameter	Percentage of Plate	LAP JOINT				BUTT STRAPS			
		Punched Iron Plates Iron Rivets	Drilled Iron Plates Iron Rivets	Steel Plates Steel Rivets	Steel Plates Iron Rivets	Punched Iron Plates Iron Rivets	Drilled Iron Plates Iron Rivets	Steel Plates Steel Rivets	Steel Plates Iron Rivets
		Constants							
		100	90	85	70	$\frac{7}{4}$·100	$\frac{7}{4}$·90	$\frac{7}{4}$·85	$\frac{7}{4}$·70
2·50	60	...	2·12	2·25	2·73	...	...	...	...
2·56	61	...	2·21	2·34	2·84	...	...	...	...
2·63	62	2·08	2·31	2·44	2·97	...	...	...	...
2·70	63	2·17	2·41	2·55	3·10	...	...	...	...
2·78	64	2·26	2·52	2·66	3·23	...	...	...	...
2·86	65	2·36	2·63	2·78	3·38	...	...	...	...
2·94	66	2·47	2·75	2·91	3·53	...	...	...	2·02
3·03	67	2·59	2·87	3·04	3·69	...	...	...	2·11
3·12	68	2·71	3·01	3·18	3·87	...	...	...	2·21
3·23	69	2·83	3·15	3·33	4·05	...	...	...	2·31
3·33	70	2·97	3·30	3·49	4·24	...	...	...	2·42
3·45	71	3·12	3·46	3·66	4·45	...	...	2·10	2·54
3·57	72	3·27	3·64	3·85	4·68	...	2·08	2·20	2·67
3·70	73	3·44	3·82	4·05	4·92	...	2·19	2·31	2·81
3·85	74	3·62	4·03	4·26	5·18	2·07	2·31	2·44	2·96
4·00	75	3·82	4·24	4·49	5·46	2·18	2·43	2·57	3·12
4·08	75½	3·92	4·36	4·61	5·61	2·24	2·49	2·64	3·20
4·17	76	4·03	4·48	4·74	5·76	2·30	2·56	2·71	3·29
4·26	76½	4·14	4·61	4·87	5·92	2·37	2·63	2·79	3·38
4·34	77	4·26	4·74	5·01	6·09	2·44	2·71	2·87	3·48
4·44	77½	4·38	4·87	5·16	6·27	2·51	2·78	2·95	3·58
4·55	78	4·51	5·02	5·31	6·45	2·58	2·86	3·03	3·68
4·65	78½	4·65	5·17	5·47	6·64	2·66	2·95	3·12	3·79
4·76	79	4·79	5·32	5·64	6·84	2·74	3·04	3·22	3·91
4·88	79½	4·94	5·49	5·81	7·05	2·82	3·13	3·32	4·03
5·00	80	5·09	5·66	5·99	7·28	2·91	3·23	3·42	4·16
5·13	80½	5·26	5·84	6·18	7·51	3·00	3·33	3·53	4·29
5·26	81	5·43	6·03	6·39	7·75	3·10	3·44	3·65	4·43
5·40	81½	5·61	6·23	6·60	8·01	3·21	3·56	3·77	4·58
5·56	82	5·80	6·44	6·82	8·29	3·32	3·68	3·90	4·73
5·71	82½	6·00	6·67	7·06	8·57	3·43	3·81	4·04	4·90
5·88	83	6·22	6·91	7·31	8·88	3·55	3·94	4·18	5·07
6·06	83½	6·44	7·16	7·58	9·20	3·68	4·08	4·33	5·26
6·25	84	6·68	7·43	7·86	9·54	3·82	4·25	4·49	5·46
6·45	84½	6·94	7·71	8·17	9·92	3·97	4·41	4·67	5·67
6·67	85	7·22	8·02	8·49	10·31	4·12	4·58	4·85	5·89
6·82	85⅓	7·41	8·23	8·72	...	4·23	4·70	4·98	6·05
6·98	85⅔	7·61	8·46	8·95	...	4·35	4·83	5·12	6·21
7·14	86	7·82	8·69	9·20	...	4·47	4·97	5·26	6·38
7·32	86⅓	8·04	8·94	9·46	...	4·60	5·11	5·41	6·57
7·50	86⅔	8·28	9·20	9·74	...	4·73	5·25	5·56	6·76
7·69	87	8·52	9·47	10·02	...	4·87	5·41	5·73	6·96
7·89	87⅓	8·78	9·75	...	...	5·02	5·57	5·90	7·17
8·10	87⅔	9·05	10·06	...	...	5·17	5·75	6·08	7·39
8·33	88	9·34	...	...	...	5·33	5·93	6·28	7·62
8·57	88⅓	9·64	...	...	...	5·51	6·12	6·48	7·87
8·82	88⅔	9·96	...	...	...	5·69	6·32	6·70	8·13
9·09	89	10·30	...	...	...	5·89	6·54	6·93	8·41
9·37	89⅓	...	...	...	...	6·09	6·77	7·17	8·70
9·68	89⅔	...	...	...	...	6·31	7·01	7·43	9·02
0·00	90	...	...	...	...	6·55	7·28	7·70	9·36

In many of the high-percentage joints it will be found that unless the diameters of the rivets are less than the thickness of the plates the most efficient proportions cannot be adopted.

The following tables contain the internal diameters of furnaces, according to Lloyd's Register Boiler Rules.

INTERNAL DIAMETER OF CIRCULAR FURNACES IN INCHES.

Thickness. Inches	Plain or Flanged Furnaces $350\frac{t^2}{D.L'}$ or $\frac{50(300T-L)}{D}$												Patent Furnaces			
	Lengths of Furnaces												Tenacity of Steel			Holmes
													Above 26 T		Under 26 T	
													Corr.	Purves	Corr.	
	9 ft.	8 ft.	7 ft.	6 ft.	5 ft.	4 ft.	3 ft.	30 in.	24 in.	18 in.	12 in.	9 in.	$C.(t-2)\div D$			
60 LBS. WORKING PRESSURE.																
$\frac{3}{8}$	...	25·5	29·2	34·2	41·2	55·0	63·0	68·0	73·0	78·0	83·0	85·5	79·9	76·6	62·5	62·2
$\frac{13}{32}$	26·6	30·0	34·4	40·3	48·5	...	...	...	...	...	...	...	...	...	...	...
$\frac{7}{16}$	30·9	34·9	40·0	46·8	56·3	...	...	...	...	...	...	...	...	...	...	...
$\frac{15}{32}$	35·5	40·1	45·9	53·7	...	...	...	...	...	...	...	...	...	...	...	...
$\frac{1}{2}$	40·5	45·7	52·3	...	...	...	...	...	...	...	...	...	...	...	...	...
$\frac{17}{32}$	45·8	51·6	...	...	...	...	...	...	...	...	...	...	...	...	...	...
$\frac{9}{16}$	51·4	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
80 LBS. WORKING PRESSURE.																
$\frac{3}{8}$	...	...	...	25·5	30·7	41·1	47·0	50·8	54·5	58·3	62·0	63·9	58·9	57·2	46·0	46·5
$\frac{13}{32}$	...	...	25·6	30·0	36·2	45·3	52·8	...	...	...	...	...	...	...	52·2	52·3
$\frac{7}{16}$	...	25·9	29·7	34·8	42·0	51·1	...	...	...	...	...	...	...	...	...	...
$\frac{15}{32}$	26·3	29·8	34·2	40·1	45·9	...	...	...	...	...	...	...	...	...	...	...
$\frac{1}{2}$	30·1	34·0	39·0	45·7	49·0	...	...	...	...	...	...	...	...	...	...	...
$\frac{17}{32}$	34·1	38·4	44·1	51·6	52·1	...	...	...	...	...	...	...	...	...	...	...
$\frac{9}{16}$	38·3	43·2	49·5	...	...	...	...	...	...	...	...	...	...	...	...	...
$\frac{19}{32}$	42·7	48·2	55·2	...	...	...	...	...	...	...	...	...	...	...	...	...
$\frac{5}{8}$	47·3	53·4	...	...	...	...	...	...	...	...	...	...	...	...	...	...
$\frac{21}{32}$	52·3	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
100 LBS. WORKING PRESSURE.																
$\frac{3}{8}$	...	...	...	...	...	32·7	37·5	40·5	43·5	46·5	49·5	51·0	46·4	45·6	36·0	37·0
$\frac{13}{32}$	...	...	...	...	28·7	36·1	42·1	45·1	48·1	51·1	...	...	52·7	51·4	41·0	41·7
$\frac{7}{16}$	...	...	...	27·7	33·4	40·7	46·7	49·7	52·7	...	...	...	...	...	46·0	46·4
$\frac{15}{32}$	...	...	27·2	31·9	36·6	45·3	51·3	54·3	...	...	...	...	...	...	51·0	51·0
$\frac{1}{2}$	...	27·0	31·0	36·3	44·0	50·0	...	...	...	...	...	...	...	...	...	...
$\frac{17}{32}$	27·0	30·5	35·1	41·0	48·6	...	...	...	...	...	...	...	...	...	...	...
$\frac{9}{16}$	30·4	34·3	39·4	46·0	53·2	...	...	...	...	...	...	...	...	...	...	...
$\frac{19}{32}$	33·9	38·3	43·9	51·4	...	...	...	...	...	...	...	...	...	...	...	...
$\frac{5}{8}$	37·6	42·5	48·7	...	...	...	...	...	...	...	...	...	...	...	...	...
$\frac{21}{32}$	41·6	46·8	53·6	...	...	...	...	...	...	...	...	...	...	...	...	...
$\frac{11}{16}$	45·7	51·5	...	...	...	...	...	...	...	...	...	...	...	...	...	...
$\frac{23}{32}$	50·0	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
120 LBS. WORKING PRESSURE.																
$\frac{3}{8}$	...	...	...	...	...	27·1	31·1	33·6	36·1	38·6	41·1	42·3	38·0	37·9	29·3	30·7
$\frac{13}{32}$	...	...	...	...	...	29·9	34·9	37·4	39·9	42·4	44·9	46·2	43·2	42·7	33·5	34·6
$\frac{7}{16}$	...	...	...	...	27·7	33·8	38·8	41·3	43·8	46·3	48·8	50·0	48·5	47·5	37·7	38·5
$\frac{15}{32}$	...	...	...	26·4	30·3	37·6	42·6	45·1	47·6	50·1	52·6	...	53·7	52·2	41·8	42·4
$\frac{1}{2}$	...	...	25·7	30·1	36·5	41·5	46·5	49·0	51·5	...	...	...	...	...	46·0	46·2
$\frac{17}{32}$	...	25·3	29·0	34·0	40·0	45·3	50·3	52·8	...	...	...	...	...	...	50·2	50·1

INTERNAL DIAMETER OF CIRCULAR FURNACES IN INCHES—*continued.*

Thickness. Inches	Plain or Flanged Furnaces $350\frac{t^2}{D.L}$, or $\frac{50(300T-L)}{D}$												Patent Furnaces			
	Lengths of Furnaces												Tenacity of Steel			Holmes
													Abov 26 T		Under 26 T	
													Corr.	Purves	Corr.	
	9 ft.	8 ft.	7 ft.	6 ft.	5 ft.	4 ft.	3 ft.	30 in.	24 in.	18 in.	12 in.	9 in.	$C.(t-2)\div D$			
	120 LBS. WORKING PRESSURE—*Continued.*															
$\frac{9}{16}$	25·1	28·4	32·6	38·1	44·1	49·1	...	...	...	...	...	...	...	...	...	...
$\frac{19}{32}$	28·1	31·7	36·4	42·6	48·0	53·0	...	...	...	...	...	...	...	...	...	...
$\frac{5}{8}$	31·2	35·2	40·4	46·8	51·8	...	...	...	...	...	...	...	...	...	...	...
$\frac{21}{32}$	34·4	38·9	44·6	50·7	...	...	...	...	...	...	...	...	...	...	...	...
$\frac{11}{16}$	37·8	42·7	49·0	...	...	...	...	...	...	...	...	...	...	...	...	...
$\frac{23}{32}$	41·4	46·7	53·4	...	...	...	...	...	...	...	...	...	...	...	...	...
$\frac{3}{4}$	45·2	51·0	...	...	...	...	...	...	...	...	...	...	...	...	...	...
$\frac{25}{32}$	49·0	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
	140 LBS. WORKING PRESSURE.															
$\frac{3}{8}$	...	...	...	...	...	...	25·1	28·7	30·8	33·0	35·1	36·2	32·0	32·4	24·6	26·2
$\frac{13}{32}$	...	...	...	...	...	25·5	29·8	32·0	34·1	36·2	38·4	39·5	36·5	36·5	28·1	29·6
$\frac{7}{16}$	...	...	...	...	...	28·8	33·1	35·2	37·4	39·5	41·0	42·7	41·0	40·6	31·7	32·9
$\frac{15}{32}$	...	...	...	...	25·6	32·1	36·4	38·5	40·7	42·8	45·0	46·0	45·5	44·6	35·3	36·2
$\frac{1}{2}$	...	...	...	25·7	31·1	35·4	39·7	41·8	44·0	46·1	48·2	49·0	50·0	48·7	38·9	39·5
$\frac{17}{32}$	...	...	24·7	28·9	34·4	38·7	43·0	45·1	47·2	49·4	51·5	52·6	54·4	52·8	42·4	42·8
$\frac{9}{16}$	...	...	27·8	32·5	37·7	42·0	46·2	48·4	50·5	...	...	...	...	...	46·0	46·1
$\frac{19}{32}$	...	27·0	31·0	36·4	41·0	45·2	49·6	51·7	...	...	...	...	...	...	49·6	49·4
$\frac{5}{8}$	26·6	30·0	34·4	40·0	44·2	48·5	52·8	...	...	...	...	...	...	...	53·1	53·7
$\frac{21}{32}$	29·3	33·1	38·0	43·2	47·5	51·8	...	...	...	...	...	...	...	...	...	...
$\frac{11}{16}$	32·2	36·4	41·8	46·5	50·8	...	...	...	...	...	...	...	...	...	...	...
$\frac{23}{32}$	35·3	39·8	45·5	49·8	...	...	...	...	...	...	...	...	...	...	...	...
$\frac{3}{4}$	38·5	43·5	48·8	53·1	...	...	...	...	...	...	...	...	...	...	...	...
$\frac{25}{32}$	44·0	47·2	52·1	...	...	...	...	...	...	...	...	...	...	...	...	...
	160 LBS. WORKING PRESSURE.															
$\frac{3}{8}$	...	...	...	...	...	...	..	25·0	26·9	28·7	30·6	31·5	27·5	28·2	...	...
$\frac{13}{32}$	...	...	...	...	...	...	26·0	27·9	29·7	31·6	33·5	34·4	31·4	31·8	24·1	25·8
$\frac{7}{16}$	...	...	...	...	...	25·1	28·8	30·7	32·6	34·4	35·7	37·3	35·3	35·4	27·2	28·7
$\frac{15}{32}$	...	...	...	...	...	28·0	31·7	33·7	36·5	37·3	39·2	40·1	39·3	38·9	30·4	31·5
$\frac{1}{2}$	...	...	...	...	27·1	30·8	34·6	36·5	38·3	40·2	42·1	43·0	43·2	42·5	33·5	34·4
$\frac{17}{32}$	...	...	...	25·2	29·9	33·7	37·4	39·3	41·2	43·1	44·9	45·9	47·1	46·1	36·6	37·3
$\frac{9}{16}$	...	...	...	28·2	32·8	36·6	40·3	42·2	44·1	45·9	47·8	48·7	51·1	49·6	39·7	40·2
$\frac{19}{32}$	...	...	27·0	31·7	35·7	39·4	43·2	45·1	46·9	48·8	50·2	51·6	55·0	53·2	42·9	43·1
$\frac{5}{8}$	...	26·1	30·0	34·8	38·5	42·3	46·0	47·9	49·8	51·7	...	...	...	...	46·0	46·0
$\frac{21}{32}$	25·5	28·8	33·1	37·7	41·4	45·2	48·9	50·8	52·7	...	...	...	...	...	49·1	48·9
$\frac{11}{16}$	28·0	31·7	36·4	40·5	44·3	48·0	51·8	...	...	...	...	...	...	...	52·2	51·8
$\frac{23}{32}$	30·7	34·7	39·6	43·4	47·1	50·9	...	...	...	...	...	...	...	...	...	...
$\frac{3}{4}$	33·5	37·8	43·5	46·3	50·0	...	...	...	...	...	...	...	...	...	...	...
$\frac{25}{32}$	36·4	41·1	45·4	49·1	...	...	...	...	...	...	...	...	...	...	...	...

INTERNAL DIAMETER OF CIRCULAR FURNACES IN INCHES—*continued.*

Thickness. Inches	Plain or Flanged Furnaces $850\frac{t^2}{D.L'}$ or $\frac{50(300\,T-L)}{D}$												Patent Furnaces			
													Tenacity of Steel			Holmes
													Above 26 T		Under 26 T	
	Lengths of Furnaces												Corr.	Purves	Corr.	
	9 ft.	8 ft.	7 ft.	6 ft.	5 ft.	4 ft.	3 ft.	30 in.	24 in.	18 in.	12 in.	9 in.	C . (t−2)÷D			
	180 LBS. WORKING PRESSURE.															
3/8	...	...	...	...	...	...	...	...	...	...	27.1	28.0	24.0	25.0	...	...
13/32	...	...	...	...	...	...	...	...	26.3	28.0	29.7	30.5	27.5	28.2	...	...
7/16	...	...	...	...	...	...	25.5	27.2	28.9	30.5	31.6	33.0	31.0	31.3	...	25.4
15/32	...	...	...	...	...	...	28.1	29.7	31.4	33.1	34.7	35.6	34.5	34.5	26.6	27.9
1/2	...	...	...	...	...	27.3	30.6	32.3	34.0	35.6	37.3	38.1	38.0	37.7	29.3	30.5
17/32	...	...	...	...	26.5	29.8	33.2	34.8	36.5	38.2	39.8	40.7	41.5	40.8	32.1	33.1
9/16	...	...	...	25.0	29.0	32.4	35.7	37.4	39.0	40.7	42.4	43.2	45.0	44.0	34.9	35.7
19/32	...	...	...	28.0	31.6	34.7	38.3	39.9	41.6	43.2	44.9	45.7	48.5	47.1	37.7	38.2
5/8	...	...	26.5	30.8	34.1	37.5	40.8	42.5	44.1	45.8	47.5	48.2	51.9	50.3	40.4	40.8
21/32	...	25.5	29.3	33.3	36.7	40.0	43.3	45.0	46.7	48.3	50.0	50.8	...	...	43.2	43.4
11/16	...	28.2	32.3	35.9	39.2	42.5	45.9	47.5	49.2	50.9	...	...	...	...	46.0	45.9
23/32	27.1	30.6	35.1	38.4	41.7	45.1	48.4	50.1	51.7	...	...	...	...	...	48.8	48.5
3/4	29.7	34.5	37.6	41.0	44.3	47.6	51.0	...	...	...	...	...	...	...	51.6	51.0
25/32	32.1	36.4	40.2	43.5	46.8	50.2	...	...	...	...	...	...	...	...	...	...
	200 LBS. WORKING PRESSURE.															
3/8	...	...	...	...	...	...	...	...	...	...	...	25.1	...	...	...	...
13/32	...	...	...	...	...	...	...	...	...	25.1	26.6	27.4	...	25.3	...	...
7/16	...	...	...	...	...	...	...	...	25.9	27.4	28.4	29.6	27.5	28.1	...	...
15/32	...	...	...	...	...	...	25.2	26.7	28.2	29.7	31.2	31.9	30.6	31.0	...	25.0
1/2	...	...	...	...	...	...	27.5	29.0	30.5	32.0	33.5	34.2	33.8	33.8	26.0	27.3
17/32	...	...	...	...	...	26.7	29.7	31.2	32.7	34.2	35.7	36.5	36.9	36.6	28.5	29.6
9/16	...	...	...	...	26.0	29.0	32.0	33.5	35.0	36.5	38.0	38.8	40.1	39.5	31.0	31.9
19/32	...	...	...	...	28.3	31.3	34.3	35.8	37.3	38.8	39.3	41.0	43.2	42.3	33.5	34.3
5/8	...	...	...	27.6	30.6	33.6	36.6	38.1	39.6	41.1	42.6	43.3	46.4	45.1	36.0	36.5
21/32	...	...	26.1	29.9	32.9	35.9	38.9	40.4	41.9	43.4	44.9	45.6	49.5	48.0	38.5	38.7
11/16	...	25.1	28.8	32.1	35.1	38.1	41.1	43.6	44.1	45.6	47.1	47.9	52.6	50.8	41.0	41.0
23/32	...	27.4	31.4	34.4	37.4	40.4	43.4	44.9	46.4	47.9	48.4	50.1	...	...	43.5	43.3
3/4	26.5	30.0	33.7	36.7	39.7	42.7	45.7	47.2	48.7	50.2	51.7	...	...	...	46.0	45.6
25/32	28.8	32.6	36.0	39.0	42.0	45.0	48.0	49.5	51.0	...	...	...	...	...	48.5	47.9

Tables for Finding Maximum Pitch of Stays.

The following two sets of tables contain the maximum permissible pitches for various working pressures, thicknesses, and methods of attachments, both for stays and stay tubes. When the cross pitches are not equal, the square root of the mean of their squares should be taken. If the stays or stay tubes are pitched irregularly the square root of half of the square of the diameter of the largest inscribed circle should be taken, for the pitch, or multiply it by $\sqrt{0{\cdot}5} = 0{\cdot}707$.

Maximum Pitch in Inches of Stayed Flat Plates.

Plate Thickness	STAYED IRON PLATES							STAYED STEEL PLATES						
	Stay Ends Riveted	Stays Nutted	Stays with Double Nuts	Do. & Washers, $\frac{1}{3}$P × $\frac{1}{2}$T	Do. Washers, Riveted. $\frac{2}{3}$P × $\frac{1}{2}$T	Do. Do., $\frac{2}{3}$P × T, or across Doubling Strips	Doubling Strips Lengthways	Stay Ends Riveted	Stays Nutted	Stays with Double Nuts	Do. & Washers, $\frac{1}{3}$P × $\frac{1}{2}$T	Do. Riveted, $\frac{2}{3}$P × $\frac{1}{2}$T	Do. Do., $\frac{2}{3}$P × T, or across Doubling Strips	Doubling Strips Lengthways
Constants	90 100	110 120	140	150	160	175	190	90 100	110 120 135	175	185	200	220	240
							60 lbs. Working Pressure.							
Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
$\frac{3}{8}$	7·34	8·12	9·16	9·48	9·79	10·2	10·6	7·34	8·12	10·2	10·5	10·9	11·4	12·0
$\frac{7}{16}$	8·57	9·47	10·7	11·0	11·4	11·9	12·4	8·57	9·47	11·9	12·2	12·7	13·4	14·0
$\frac{1}{2}$	10·2	11·3	12·2	12·6	13·0	13·6	14·2	10·2	11·3	13·6	14·0	14·6	15·3	16·0
$\frac{9}{16}$	11·5	12·7	13·7	14·2	14·6	15·3	16·0	11·5	13·5	15·3	15·8	16·4	17·2	18·0
$\frac{5}{8}$	12·8	14·1	15·2	15·8	16·3	17·0	17·7	12·8	15·0	17·0	17·5	18·2	19·1	20·0
$\frac{11}{16}$	14·1	15·5	16·8	17·3	17·9	18·7	19·5	14·1	16·5	18·7	19·3	20·1	21·1	...
$\frac{3}{4}$	15·4	16·9	18·3	18·9	19·5	20·4	21·2	15·4	18·0	20·4	21·1	...	...	...
$\frac{13}{16}$	16·7	18·3	19·8	20·6	21·2	...	...	16·7	19·5	...	...	...	...	...
$\frac{7}{8}$	18·0	19·7	21·3	...	...	...	...	18·0	21·0	...	...	...	...	...
$\frac{15}{16}$	19·3	21·1	...	...	...	...	...	19·3	...	...	...	...	...	...
1	20·6	...	...	...	...	...	...	20·6	...	...	...	...	...	...
							80 lbs. Working Pressure.							
$\frac{3}{8}$	6·36	7·03	7·93	8·21	8·48	8·87	9·24	6·36	7·03	8·87	9·12	9·48	9·95	10·3
$\frac{7}{16}$	7·42	8·20	9·26	9·58	9·89	10·3	10·7	7·42	8·21	10·3	10·6	11·0	11·6	12·1
$\frac{1}{2}$	8·94	10·79	10·6	10·9	11·3	11·8	12·3	8·94	9·89	11·8	12·1	12·6	13·2	13·8
$\frac{9}{16}$	10·0	11·0	11·9	12·3	12·7	13·3	13·8	10·0	11·6	13·3	13·6	14·2	14·9	15·5
$\frac{5}{8}$	11·1	12·2	13·2	13·6	14·1	14·7	15·4	11·1	12·9	14·7	15·2	15·8	16·5	17·3
$\frac{11}{16}$	12·2	13·4	14·5	15·0	15·5	16·2	16·9	12·2	14·2	16·2	16·7	17·3	18·2	19·0
$\frac{3}{4}$	13·4	14·6	15·8	16·4	16·9	17·7	18·4	13·4	15·5	17·7	18·2	18·9	19·9	20·8
$\frac{13}{16}$	14·5	15·9	17·1	17·8	18·3	19·2	20·0	14·5	16·8	19·2	19·7	20·5	21·6	...
$\frac{7}{8}$	15·6	17·1	18·5	19·1	19·7	20·6	...	15·6	18·1	20·7	21·3	...	...	...
$\frac{15}{16}$	16·7	18·3	19·8	20·5	21·2	...	...	16·7	19·4	...	...	...	...	...
1	17·8	19·5	21·2	...	...	...	...	17·8	20·7	...	...	...	...	...
$1\frac{1}{16}$	19·0	20·8	...	...	...	...	...	19·0	...	...	...	...	...	...

	STAYED IRON PLATES							STAYED STEEL PLATES						
Plate Thickness	Stay Ends Riveted	Stays Nutted	Stays with Double Nuts	Do. & Washers, $\frac{2}{3}$P×$\frac{1}{2}$T	Do. Washers, Riveted, $\frac{2}{3}$P×$\frac{1}{2}$T	Do. Do., $\frac{2}{3}$P×T, or across Doubling Strips	Doubling Strips Lengthways	Stay Ends Riveted	Stays Nutted	Stays with Double Nuts	Do. & Washers, $\frac{2}{3}$P×$\frac{1}{2}$T	Do. Riveted, $\frac{2}{3}$P×$\frac{1}{2}$T	Do. Do., $\frac{2}{3}$P×T, or across Doubling Strips	Doubling Strips Lengthways
Constants	90 100	110 120	140	150	160	175	190	90 100	110 120 135	175	185	200	220	240
							100 lbs. Working Pressure.							
Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
$\frac{3}{8}$	5·79	6·29	7·09	7·34	7·58	7·93	8·27	5·79	6·29	7·93	8·16	8·48	8·89	9·29
$\frac{7}{16}$	6·64	7·34	8·28	8·57	8·85	9·26	9·64	6·64	7·34	9·26	9·52	9·89	10·3	10·8
$\frac{1}{2}$	8·00	8·76	9·46	9·79	10·1	10·6	11·0	8·00	8·76	10·6	10·8	11·3	11·8	12·3
$\frac{9}{16}$	9·00	9·85	10·6	11·0	11·3	11·9	12·4	9·00	10·4	11·9	12·2	12·7	13·3	13·9
$\frac{5}{8}$	10·0	10·9	11·8	12·2	12·6	13·2	13·7	10·0	11·6	13·2	13·6	14·1	14·8	15·4
$\frac{11}{16}$	11·0	12·0	13·0	13·4	13·9	14·5	15·1	11·0	12·7	14·5	14·9	15·5	16·3	17·0
$\frac{3}{4}$	12·0	13·1	14·1	14·6	15·1	15·8	16·5	12·0	13·9	15·8	16·3	16·9	17·7	18·5
$\frac{13}{16}$	13·0	14·2	15·3	15·9	16·4	17·1	17·9	13·0	15·1	17·1	17·6	18·3	19·2	20·1
$\frac{7}{8}$	14·0	15·3	16·5	17·1	17·7	18·5	19·2	14·0	16·2	18·5	19·0	19·7	20·7	...
$\frac{15}{16}$	15·0	16·4	17·7	18·3	18·9	19·8	20·6	15·0	17·4	19·8	20·4	21·1	...	..
1	16·0	17·5	18·9	19·5	20·2	21·1	...	16·0	18·5	21·1	...	...	...	...
$1\frac{1}{16}$	17·0	18·6	20·1	20·8	...	...	...	17·0	19·7	...	...	...	...	...
$1\frac{1}{8}$	18·0	19·7	...	...	...	...	...	18·0	20·9	...	...	...	...	...
							120 lbs. Working Pressure.							
$\frac{3}{8}$	5·19	5·74	6·48	6·70	6·92	7·24	7·55	5·19	5·74	7·24	7·44	7·74	8·12	8·48
$\frac{7}{16}$	6·06	6·70	7·56	7·82	8·07	8·45	8·80	6·06	6·70	8·45	8·69	9·03	9·47	9·89
$\frac{1}{2}$	7·30	8·00	8·64	8·94	9·23	9·66	10·0	7·30	8·00	9·66	9·93	10·3	10·8	11·3
$\frac{9}{16}$	8·21	9·00	9·72	10·0	10·3	10·8	11·3	8·21	9·54	10·8	11·1	11·6	12·1	12·7
$\frac{5}{8}$	9·12	10·0	10·8	11·1	11·5	12·0	12·5	9·12	10·6	12·0	12·4	12·9	13·5	14·1
$\frac{11}{16}$	10·0	11·0	11·8	12·2	12·7	13·2	13·8	10·0	11·6	13·2	13·6	14·2	14·8	15·5
$\frac{3}{4}$	10·9	12·0	12·9	13·4	13·8	14·4	15·1	10·9	12·7	14·4	14·8	15·4	16·2	16·9
$\frac{13}{16}$	11·8	13·0	14·0	14·5	15·0	15·7	16·3	11·8	13·7	15·7	16·1	16·7	17·6	18·3
$\frac{7}{8}$	12·7	14·0	15·1	15·6	16·1	16·9	17·6	12·7	14·8	16·9	17·4	18·0	18·9	19·7
$\frac{15}{16}$	13·6	15·0	16·2	16·7	17·3	18·1	18·8	13·6	15·9	18·1	18·6	19·3	20·3	21·1
1	14·6	16·0	17·2	17·8	18·4	19·3	20·0	14·6	16·9	19·3	19·8	20·6	...	...
$1\frac{1}{16}$	15·5	17·0	18·3	19·0	19·6	20·5	...	15·5	18·0	20·5	21·0	...	...	...
$1\frac{1}{8}$	16·4	18·0	19·4	20·1	20·8	...	...	16·4	19·0	...	...	...	...	...
							140 lbs. Working Pressure.							
$\frac{3}{8}$	4·81	5·31	6·00	6·21	6·41	6·70	7·02	4·81	5·31	6·70	6·89	7·17	7·51	7·85
$\frac{7}{16}$	5·61	6·20	7·00	7·24	7·48	7·82	8·19	5·61	6·20	7·82	8·04	8·36	8·76	9·16
$\frac{1}{2}$	6·76	7·40	8·00	8·28	8·55	8·94	9·36	6·76	7·40	8·94	9·19	9·56	10·0	10·5
$\frac{9}{16}$	7·60	8·32	9·00	9·31	9·62	10·0	10·5	7·60	8·83	10·0	10·3	10·7	11·2	11·8
$\frac{5}{8}$	8·45	9·25	10·0	10·3	10·6	11·1	11·7	8·45	9·81	11·1	11·4	11·9	12·5	13·1
$\frac{11}{16}$	9·29	10·2	11·0	11·3	11·7	12·2	12·8	9·29	10·8	12·2	12·6	13·1	13·7	14·4
$\frac{3}{4}$	10·1	11·1	12·0	12·4	12·8	13·4	14·0	10·1	11·7	13·4	13·8	14·3	15·0	15·7
$\frac{13}{16}$	10·9	12·0	13·0	13·4	13·8	14·5	15·2	10·9	12·7	14·5	14·9	15·5	16·2	17·0
$\frac{7}{8}$	11·8	12·9	14·0	14·4	14·9	15·6	16·3	11·8	13·7	15·6	16·0	16·7	17·5	18·3
$\frac{15}{16}$	12·6	13·8	15·0	15·5	16·0	16·7	17·5	12·6	14·7	16·7	17·2	17·9	18·7	19·6
1	13·5	14·8	16·0	16·5	17·1	17·8	18·7	13·5	15·7	17·8	18·3	19·1	20·0	20·9
$1\frac{1}{16}$	14·3	15·7	17·0	17·5	18·1	19·0	19·9	14·3	16·6	19·0	19·5	20·3	...	...
$1\frac{1}{8}$	15·2	16·6	18·0	18·6	19·2	20·1	21·1	15·2	17·6	20·1	20·6	...	...	...

	STAYED IRON PLATES							STAYED STEEL PLATES						
Plate Thickness	Stay Ends Riveted	Stays Nutted	Stays with Double Nuts	Do. & Washers, $\frac{1}{3}$ P × $\frac{1}{2}$ T	Do. Washers, Riveted, $\frac{2}{3}$ P × $\frac{1}{2}$ T	Do. Do., $\frac{2}{3}$ P × T, or across Doubling Strips	Doubling Strips Lengthways	Stay Ends Riveted	Stays Nutted	Stays with Double Nuts	Do. & Washers, $\frac{1}{3}$ P × $\frac{1}{2}$ T	Do. Riveted, $\frac{2}{3}$ P × $\frac{1}{2}$ T	Do. Do., $\frac{2}{3}$ P × T, or across Doubling Strips	Doubling Strips Lengthways
Constants	90 100	110 120	140	150	160	175	190	90 100	110 120 135	175	185	200	220	240
							160 lbs. Working Pressure.							
Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
3/8	4·50	4·97	5·61	5·80	6·00	6·27	6·53	4·50	4·97	6·27	6·45	6·70	7·03	7·34
7/16	5·25	5·80	6·54	6·77	7·00	7·31	7·62	5·25	5·80	7·31	7·52	7·82	8·20	8·57
1/2	6·32	6·93	7·48	7·74	8·00	8·36	8·71	6·32	6·93	8·36	8·60	8·94	9·38	9·80
9/16	7·11	7·79	8·41	8·71	9·00	9·41	9·80	7·11	8·26	9·41	9·67	10·0	10·5	11·0
5/8	7·90	8·66	9·35	9·68	10·0	10·4	10·8	7·90	9·18	10·4	10·7	11·1	11·7	12·2
11/16	8·69	9·53	10·2	10·6	11·0	11·5	11·9	8·69	10·1	11·5	11·8	12·2	12·9	13·4
3/4	9·48	10·3	11·2	11·6	12·0	12·5	13·0	9·48	11·0	12·5	12·9	13·4	14·0	14·6
13/16	10·2	11·2	12·1	12·5	13·0	13·5	14·1	10·2	11·9	13·5	13·9	14·5	15·2	15·9
7/8	11·0	12·1	13·0	13·5	14·0	14·6	15·2	11·0	12·8	14·6	15·0	15·6	16·4	17·1
15/16	11·8	12·9	14·0	14·5	15·0	15·6	16·3	11·8	13·7	15·6	16·1	16·7	17·5	18·3
1	12·6	13·8	14·9	15·4	16·0	16·7	17·4	12·6	14·7	16·7	17·2	17·8	18·7	19·5
1 1/16	13·4	14·7	15·9	16·4	17·0	17·7	18·5	13·4	15·6	17·7	18·2	19·0	19·9	20·8
1 1/8	14·2	15·5	16·8	17·4	18·0	18·8	19·6	14·2	16·5	18·8	19·3	20·1	21·0	...
							180 lbs. Working Pressure.							
3/8	4·24	4·69	5·29	5·47	5·65	5·91	6·16	4·24	4·69	5·91	6·08	6·32	6·63	6·92
7/16	4·94	5·47	6·17	6·39	6·59	6·90	7·19	4·94	5·47	6·90	7·09	7·37	7·74	8·08
1/2	5·96	6·53	7·05	7·30	7·54	7·88	8·22	5·96	6·53	7·88	8·11	8·43	8·84	9·23
9/16	6·70	7·34	7·93	8·21	8·48	8·87	9·24	6·70	7·79	8·87	9·12	9·48	9·94	10·3
5/8	7·45	8·16	8·81	9·12	9·42	9·86	10·2	7·45	8·66	9·86	10·1	10·5	11·0	11·5
11/16	8·19	8·98	9·70	10·0	10·3	10·8	11·3	8·19	9·52	10·8	11·1	11·5	12·1	12·7
3/4	8·94	9·79	10·5	10·9	11·3	11·8	12·3	8·94	10·3	11·8	12·1	12·6	13·2	13·8
13/16	9·68	10·6	11·4	11·8	12·2	12·8	13·3	9·68	11·2	12·8	13·1	13·7	14·3	15·0
7/8	10·4	11·4	12·3	12·7	13·1	13·8	14·3	10·3	12·1	13·8	14·1	14·7	15·4	16·1
15/16	11·1	12·2	13·2	13·6	14·1	14·7	15·4	11·1	12·9	14·7	15·2	15·8	16·5	17·3
1	11·9	13·0	14·1	14·6	15·0	15·7	16·4	11·9	13·8	15·7	16·2	16·8	17·6	18·4
1 1/16	12·6	13·8	14·9	15·5	16·0	16·7	17·4	12·6	14·7	16·7	17·2	17·9	18·7	19·6
1 1/8	13·4	14·6	15·8	16·4	16·9	17·7	18·4	13·4	15·5	17·7	18·2	18·9	19·8	20·7
							200 lbs. Working Pressure.							
3/8	4·02	4·44	5·02	5·19	5·36	5·61	5·84	4·02	4·44	5·61	5·77	6·00	6·29	6·57
7/16	4·69	5·19	5·85	6·06	6·26	6·24	6·82	4·69	5·19	6·54	6·73	7·00	7·34	7·66
1/2	5·65	6·19	6·69	6·92	7·15	7·48	7·79	5·65	6·19	7·48	7·69	8·00	8·39	8·76
9/16	6·36	6·97	7·53	7·79	8·05	8·41	8·77	6·36	7·39	8·41	8·65	9·00	9·43	9·85
5/8	7·07	7·74	8·36	8·66	8·94	9·35	9·74	7·07	8·21	9·35	9·61	10·0	10·4	10·9
11/16	7·77	8·52	9·20	9·52	9·83	10·2	10·7	7·77	9·03	10·2	10·5	11·0	11·5	12·0
3/4	8·48	9·29	10·0	10·3	10·7	11·2	11·6	8·48	9·85	11·2	11·5	12·0	12·5	13·1
13/16	9·19	10·0	10·8	11·2	11·6	12·1	12·6	9·19	10·6	12·1	12·5	13·0	13·6	14·2
7/8	9·89	10·8	11·7	12·1	12·5	13·0	13·6	9·89	11·5	13·0	13·4	14·0	14·6	15·3
15/16	10·6	11·6	12·5	12·9	13·4	14·0	14·6	10·6	12·3	14·0	14·4	15·0	15·7	16·4
1	11·3	12·3	13·3	13·8	14·3	14·9	15·5	11·3	13·1	14·9	15·3	16·0	16·7	17·5
1 1/16	12·0	13·1	14·2	14·7	15·2	15·9	16·5	12·0	13·9	15·9	16·3	17·0	17·8	18·6
1 1/8	12·7	13·9	15·0	15·5	16·1	16·8	17·5	12·7	14·7	16·8	17·3	18·0	19·8	19·7

Pitch in Inches of Tube Plate Stays, Centre to Centre.

Plate Thickness	Mean Pitch in Nest of Tubes	Wide Water Spaces						Mean Pitch in Nest of Tubes	Wide Water Spaces					
		Stay Ends Beaded			Stay Ends Nutted				Stay Ends Beaded			Stay Ends Nutted		
		Number of Plain Tubes between Stays							Number of Plain Tubes between Stays					
		0	1	2	0	1	2		0	1	2	0	1	2
Constants	140	160	140	120	170	150	130	140	160	140	120	170	150	130
	60 lbs. Working Pressure.							100 lbs. Working Pressure.						
$\frac{3}{8}$	9·16	9·80	9·16	8·48	10·1	9·49	8·83	7·10	7·59	7·10	6·57	7·82	7·35	6·84
$\frac{7}{16}$	10·7	11·4	10·7	9·90	11·8	11·1	10·3	8·28	8·85	8·28	7·67	9·13	8·57	7·98
$\frac{1}{2}$	12·2	13·1	12·2	11·3	13·5	12·6	11·8	9·47	10·1	9·47	8·76	10·4	9·80	9·12
$\frac{9}{16}$	13·7	14·7	13·7	12·7	15·1	14·2	13·2	10·6	11·4	10·6	9·86	11·7	11·0	10·3
$\frac{5}{8}$	15·3	16·3	15·3	14·1	16·8	15·8	14·7	11·8	12·6	11·8	11·0	13·0	12·2	11·4
$\frac{11}{16}$	16·8	18·0	16·8	15·6	18·5	17·4	16·2	13·0	13·9	13·0	12·0	14·3	13·5	12·5
$\frac{3}{4}$	18·3	19·6	18·3	17·0	20·2	19·0	17·7	14·2	15·2	14·2	13·1	15·6	14·7	13·7
$\frac{13}{16}$	19·9	21·2	19·9	18·4	...	20·6	19·1	15·4	16·4	15·4	14·2	16·9	15·9	14·8
$\frac{7}{8}$	21·4	...	21·4	19·8	...	...	20·6	16·6	17·7	16·6	15·3	18·3	17·1	16·0
$\frac{15}{16}$	...	...	...	21·1	...	...	...	17·7	19·0	17·7	16·4	19·6	19·4	17·1
1	...	...	...	...	...	...	...	18·9	20·2	18·9	17·5	20·9	19·6	18·2
$1\frac{1}{16}$	...	...	...	...	...	...	...	20·1	...	20·1	18·6	...	20·8	19·4
$1\frac{1}{8}$	...	...	...	...	...	...	...	...	...	...	19·7	...	...	20·5
	80 lbs. Working Pressure.							120 lbs. Working Pressure.						
$\frac{3}{8}$	7·94	8·49	7·94	7·35	8·75	8·22	7·65	6·48	6·93	6·48	6·00	7·14	6·71	6·24
$\frac{7}{16}$	9·26	9·90	9·26	8·57	10·2	9·58	8·92	7·56	8·07	7·56	7·00	8·33	7·83	7·29
$\frac{1}{2}$	10·7	11·3	10·7	9·79	11·7	11·0	10·2	8·64	9·24	8·64	8·00	9·52	8·94	8·33
$\frac{9}{16}$	11·9	12·7	11·9	11·0	13·1	12·3	11·5	9·72	10·4	9·72	9·00	10·7	10·1	9·37
$\frac{5}{8}$	13·2	14·1	13·2	12·2	14·6	13·7	12·7	10·8	11·5	10·8	10·0	11·9	11·2	10·4
$\frac{11}{16}$	14·6	15·6	14·6	13·5	16·0	15·1	14·0	11·9	12·7	11·9	11·0	13·1	12·3	11·4
$\frac{3}{4}$	15·9	17·0	15·9	14·7	17·5	16·4	15·3	13·0	13·9	13·0	12·0	14·3	13·4	12·5
$\frac{13}{16}$	17·2	18·4	17·2	15·9	18·9	17·8	16·6	14·0	15·0	14·0	13·0	15·5	14·5	13·5
$\frac{7}{8}$	18·5	19·8	18·5	17·1	20·4	19·2	17·8	15·1	16·2	15·1	14·0	16·7	15·7	14·6
$\frac{15}{16}$	19·8	21·2	19·8	18·4	...	20·5	19·1	16·2	17·3	16·2	15·0	17·9	16·8	15·6
1	21·2	...	21·2	19·6	...	...	20·4	17·3	18·5	17·3	16·0	19·0	17·9	16·7
$1\frac{1}{16}$	...	...	...	20·8	...	...	...	18·4	19·6	18·4	17·0	20·2	19·0	17·7
$1\frac{1}{8}$	...	...	...	...	...	...	...	19·4	20·8	19·4	18·0	...	20·1	18·7

Pitches in Inches of Tube Plate Stays, Centre to Centre.

Plate Thickness	Mean Pitch in Nest of Tubes	Wide Water Spaces: Stay Ends Beaded — Number of Plain Tubes between Stays: 0	1	2	Stay Ends Nutted: 0	1	2
Constants	140	160	140	120	170	150	130
140 lbs. Working Pressure.							
3/8	6·00	6·41	6·00	5·55	6·61	6·21	5·82
7/16	7·00	7·48	7·00	6·48	7·71	7·25	6·74
1/2	8·00	8·56	8·00	7·41	8·81	8·28	7·71
9/16	9·00	9·62	9·00	8·32	9·91	9·32	8·67
5/8	10·0	10·7	10·0	9·26	11·0	10·4	9·64
11/16	11·0	11·8	11·0	10·2	12·1	11·4	10·6
3/4	12·0	12·8	12·0	11·1	13·2	12·4	11·6
13/16	13·0	13·9	13·0	12·0	14·3	13·5	12·5
7/8	14·0	15·0	14·0	13·0	15·4	14·5	13·5
15/16	15·0	16·0	15·0	13·9	16·5	15·5	14·5
1	16·0	17·1	16·0	14·8	17·6	16·6	15·4
1 1/16	17·0	18·2	17·0	15·7	18·7	17·6	16·4
1 1/8	18·0	19·3	18·0	16·7	19·8	18·6	17·3
160 lbs. Working Pressure.							
3/8	5·61	6·00	5·61	5·20	6·18	5·81	5·41
7/16	6·55	7·00	6·55	6·07	7·22	6·78	6·31
1/2	7·48	8·00	7·48	6·93	8·25	7·74	7·21
9/16	8·42	9·00	8·42	7·80	9·28	8·71	8·11
5/8	9·35	10·0	9·35	8·67	10·3	9·68	9·01
11/16	10·3	11·0	10·3	9·53	11·3	10·6	9·92
3/4	11·2	12·0	11·2	10·4	12·4	11·6	10·8
13/16	12·2	13·0	12·2	11·3	13·4	12·6	11·7
7/8	13·1	14·0	13·1	12·1	14·4	13·6	12·6
15/16	14·0	15·0	14·0	13·0	15·5	14·4	13·5
1	15·0	16·0	15·0	13·9	16·5	15·5	14·4
1 1/16	15·9	17·0	15·9	14·7	17·5	16·5	15·3
1 1/8	16·8	18·0	16·8	15·6	18·6	17·4	16·2

Plate Thickness	Mean Pitch in Nest of Tubes	Wide Water Spaces: Stay Ends Beaded — Number of Plain Tubes between Stays: 0	1	2	Stay Ends Nutted: 0	1	2
Constants	140	160	140	120	170	150	130
180 lbs. Working Pressure.							
3/8	5·29	5·66	5·29	4·90	5·83	5·48	5·10
7/16	6·17	6·60	6·17	5·72	6·80	6·39	5·95
1/2	7·05	7·54	7·05	6·53	7·77	7·30	6·80
9/16	7·94	8·48	7·94	7·35	8·75	8·22	7·65
5/8	8·82	9·43	8·82	8·16	9·72	9·13	8·50
11/16	9·70	10·4	9·70	8·98	10·7	10·0	9·35
3/4	10·6	11·3	10·6	9·78	11·7	11·0	10·2
13/16	11·5	12·3	11·5	10·6	12·6	11·9	11·0
7/8	12·3	13·2	12·3	11·4	13·6	12·8	11·9
15/16	13·2	14·1	13·2	12·2	14·6	13·7	12·7
1	14·1	15·1	14·1	13·1	15·5	14·6	13·6
1 1/16	15·0	16·0	15·0	13·9	16·5	15·5	14·4
1 1/8	15·9	17·0	15·9	14·7	17·5	16·4	15·3
200 lbs. Working Pressure.							
3/8	5·02	5·37	5·02	4·65	5·53	5·20	4·84
7/16	5·86	6·26	5·86	5·42	6·45	6·06	5·64
1/2	6·69	7·15	6·69	6·20	7·39	6·93	6·45
9/16	7·53	8·05	7·93	6·97	8·31	7·80	7·26
5/8	8·37	8·94	8·37	7·75	9·22	8·66	8·06
11/16	9·20	9·84	9·20	8·52	10·1	9·52	8·87
3/4	10·0	10·7	10·0	9·29	11·1	10·4	9·67
13/16	10·9	11·6	10·9	10·1	12·0	11·3	10·5
7/8	11·7	12·5	11·7	10·8	12·9	12·1	11·3
15/16	12·5	13·4	12·5	11·6	13·8	13·0	12·1
1	13·4	14·3	13·4	12·4	14·7	13·9	12·9
1 1/16	14·2	15·2	14·2	13·2	15·7	14·7	13·7
1 1/8	15·1	16·1	15·1	13·9	16·6	15·6	14·5

Table of Relative Sizes of Screwed Stays and Flat Plates.

A simple relation exists between the thickness of a plate and its stays, and when the maximum pitch has been determined by the previous tables, the sizes, i.e. external diameters and pitch of threads, can be found direct by consulting the following table. Should the given sizes not be convenient, then either the diameter or the number of threads must be increased. Smaller numbers than those given in this table may not be adopted. The process can also be reversed.

Thickness of Plates in Inches		⅜	7/16	½	9/16	⅝	11/16	¾	13/16	⅞	15/16	1	1 1/16	1⅛
Method of Attaching Stays	Plate Con.	**IRON PLATES AND IRON STAYS**												
Stay ends riveted	90 / 100	1 8	1⅛ 9	1⅜ 7	1½ 7	1⅝ 8	...	...	...	...	...	...	...	...
Stays nutted .	110 / 120	1⅛ 7	1¼ 7	1½ 6	1⅝ 7	1⅝ 11	...	...	...	...	...	...	...	...
Double nuts .	140	1¼ 7	1⅜ 8	1½ 11	1⅝ 11	1¾ 6	1⅞ 7	2 8	2⅛ 10	2⅜ 6	2½ 7	2⅝ 8	2¾ 10	3 6
Do. and washers (⅓P×⅓T) .	150	1¼ 7	1⅜ 10	1⅝ 7	1⅝ 11	1¾ 9	1⅞ 10	2⅛ 6	2¼ 8	2⅜ 9	2½ 12	2¾ 7	2⅞ 8	3 10
Do. do. riveted (⅔P×⅓T) . .	160	1¼ 9	1½ 6	1⅝ 9	1⅝ 11	1⅞ 6	2 7	2⅛ 9	2¼ 12	2½ 7	2⅝ 9	2¾ 11	3 7	3⅛ 8
Do. do. (⅔P×T) .	175	1¼ 12	1½ 8	1⅝ 11	1¾ 6	1⅞ 7	2 10	2¼ 6	2⅜ 8	2½ 11	2¾ 7	2⅞ 9	3 12	3¼ 7
		STEEL PLATES AND STEEL STAYS												
Stay ends riveted	90 / 100	⅞ 9	1 8	1⅛ 11	1¼ 12	1⅜ 12	1½ 12	1⅝ 11	...	...	...	...	...	...
Stays nutted .	110 / 120 / 135	1 8	1⅛ 7	1¼ 9	1½ 7	1⅝ 8	1⅝ 12	1⅞ 6	...	...	...	...	...	...
Double nuts . .	175	1⅛ 11	1⅜ 7	1½ 8	1⅝ 11	1¾ 8	1⅞ 9	2 12	2¼ 7	2⅜ 8	2½ 9	2⅝ 12	2⅞ 7	3 8
Do. and washers (⅓P×⅓T). .	185	1¼ 7	1⅜ 8	1½ 10	1⅝ 11	1¾ 10	2 6	2⅛ 7	2¼ 9	2⅜ 12	2⅝ 7	2¾ 8	2⅞ 10	3⅛ 6
Do. do. riveted (⅔P×⅓T) .	200	1¼ 8	1⅜ 10	1⅝ 8	1⅝ 12	1⅞ 7	2 9	2⅛ 12	2⅜ 7	2½ 9	2¾ 6	2⅞ 7	3 10	3¼ 6
Do. do. (⅔P×T)	220	1¼ 10	1½ 7	1⅝ 10	1¾ 8	1⅞ 12	2⅛ 7	2¼ 10	2½ 7	2⅝ 9	2¾ 12	3 7	3⅛ 10	3⅜ 7
		STEEL PLATES AND IRON STAYS												
Stay ends riveted	90 / 100	1 8	1⅛ 9	1⅜ 7	1½ 7	1⅝ 8	1⅝ 11	1¾ 7	...	...	...	...	...	...
Stays nutted .	110 / 120 / 135	1⅛ 7	1¼ 7	1½ 6	1⅝ 11	1⅝ 12	1⅞ 7	2 7	...	...	...	...	...	...
Double nuts . .	175	1⅜ 6	1½ 9	1⅝ 11	1¾ 7	1⅞ 9	2 12	2¼ 7	2⅜ 10	2⅝ 6	2¾ 8	2⅞ 11	3⅛ 7	3¼ 9
Do. and washers (⅓P×⅓T) .	185	1⅜ 7	1½ 12	1⅝ 11	1¾ 9	2 6	2⅛ 8	2¼ 11	2½ 7	2⅝ 9	2⅞ 6	3 8	3⅛ 12	3⅜ 7
Do. do. riveted (⅔P×⅓T) .	200	1⅜ 10	1⅝ 7	1⅝ 11	1⅞ 6	2⅛ 8	2¼ 6	2⅜ 8	2⅝ 6	2¾ 8	2⅞ 12	3⅛ 8	3¼ 11	3½ 7
Do. do. (⅔P×T) .	220	1½ 7	1⅝ 11	1¾ 7	1⅞ 10	2⅛ 7	2¼ 11	2½ 7	2⅝ 12	2⅞ 8	3⅛ 8	3¼ 9	3½ 6	3⅝ 9

Note.—The first of the numbers are the external diameters of the stays, the second numbers (black) are the number of threads per inch.

The following three tables contain respectively the maximum permissible load for screwed iron or steel stays, and for stay tubes, both for Whitworth screws and for finer pitches. They will be found convenient where the pitching of the stays is unequal:—

PERMISSIBLE WORKING LOADS ON **Iron** STAYS WITH STRESSES OF 6,000 AND 7,500 LBS. PER SQUARE INCH. FOR DIAMETERS AND SECTIONS SEE PAGES 327 and 328.

Outside Diameters	Working Loads for Plus Threads	Whitworth Threads		Number of Threads per Inch						
				6	7	8	9	10	11	12
		Number of Threads	Working Loads of Screwed Stays							
Inches	Lbs.		Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
¾	2,650	10	1,823	...	...	...	...	1,823	1,893	1,950
⅞	3,608	9	2,530	...	...	...	2,530	2,630	2,713	2,782
1	4,712	8	3,325	...	...	3,325	3,467	3,583	3,680	3,760
1⅛	5,964	7	4,183	...	4,183	4,388	4,552	4,684	4,795	4,887
1¼	7,363	7	5,367	...	5,367	5,599	5,783	5,932	6,057	6,160
1⅜	8,909	6	6,359	6,359	6,697	6,956	7,162	7,328	7,466	7,580
1½	10,602	6	7,802	7,802	8,176	8,461	8,688	8,870	9,022	9,148
1⅝	15,550	5	8,831	9,391	9,802	10,114	10,361	10,561	13,407	13,579
1¾	18,040	5	10,518	13,910	14,467	14,892	15,227	15,497	15,721	15,907
1⅞	20,710	4½	14,903	16,265	16,867	17,325	17,687	17,977	18,219	18,419
2	23,562	4½	17,338	18,804	19,449	19,943	20,331	20,643	20,901	21,115
2⅛	26,599	4½	19,952	21,527	22,219	22,744	23,159	23,491	23,767	23,995
2¼	29,820	4	21,942	24,435	25,171	25,731	26,170	26,524	26,818	27,059
2⅜	33,225	4	24,876	27,526	28,308	28,900	29,366	29,740	30,052	30,307
2½	36,813	4	27,888	30,802	31,629	32,253	32,747	33,142	33,469	33,740
2⅝	40,591	4	31,295	34,260	35,133	35,792	36,310	36,727	37,072	37,357
2¾	44,547	4	34,783	37,903	38,816	39,514	40,060	40,496	40,859	41,158
2⅞	48,689	3½	37,088	41,732	42,694	43,420	43,992	44,450	44,828	45,141
3	53,014	3½	40,874	45,744	46,751	47,511	48,107	48,586	48,984	49,309
3⅛	57,524	...	...	49,940	50,991	51,785	52,407	52,907	53,322	53,662
3¼	62,229	3¼	48,047	54,319	55,416	56,243	56,893	57,413	57,846	58,201
3⅜	67,097	...	...	58,883	60,025	60,885	61,561	62,103	62,551	62,920
3½	72,157	3¼	56,827	63,631	64,817	65,711	66,413	66,978	67,442	67,825
3⅝	77,401	...	...	68,563	69,791	70,721	71,452	72,037	72,519	72,915
3¾	82,830	3	65,059	73,680	74,946	75,916	76,670	77,277	77,777	78,189
3⅞	88,446	...	...	78,979	80,293	81,300	82,074	82,702	83,220	83,645
	94,145	3	75,207	84,465	85,822	86,857	87,661	88,312	88,846	89,282

PERMISSIBLE WORKING LOADS ON **Steel** STAYS WITH STRESSES OF 8,000, 9,000, AND 10,000 LBS. PER SQUARE INCH. FOR DIAMETERS AND SECTIONS SEE PAGES 327 and 328.

Outside Diameters	Working Loads for Plus Threads	Whitworth Threads: Number of Threads	Whitworth Threads	Number of Threads per Inch: 6	7	8	9	10	11	12
			Working Loads of Screwed Stays							
Inches	Lbs.		Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
$\frac{3}{4}$	3,976	10	2,431	...	...	...	...	2,431	2,523	2,600
$\frac{7}{8}$	5,412	9	3,374	...	...	...	3,374	3,506	3,617	3,709
1	7,069	8	4,433	...	...	4,433	4,623	4,778	4,907	5,014
$1\frac{1}{8}$	8,946	7	5,578	...	5,578	5,851	6,069	6,245	6,393	6,515
$1\frac{1}{4}$	11,045	7	7,156	...	7,516	7,465	7,711	7,910	8,074	8,213
$1\frac{3}{8}$	13,365	6	8,479	8,479	8,931	9,275	9,549	9,770	9,954	10,107
$1\frac{1}{2}$	17,671	6	10,402	10,402	10,902	11,282	11,584	11,827	12,030	12,198
$1\frac{5}{8}$	20,740	5	11,775	12,522	13,069	13,486	13,814	14,081	16,089	16,295
$1\frac{3}{4}$	24,053	5	14,024	16,692	17,361	17,870	18,273	18,596	18,866	19,089
$1\frac{7}{8}$	27,612	$4\frac{1}{2}$	17,884	19,518	20,241	20,790	21,225	21,573	21,863	22,103
2	31,416	$4\frac{1}{2}$	20,805	22,565	23,341	23,932	24,397	24,771	25,081	25,338
$2\frac{1}{8}$	35,466	$4\frac{1}{2}$	23,942	25,833	26,663	27,293	27,790	28,190	28,520	28,795
$2\frac{1}{4}$	39,761	4	26,330	29,322	30,206	30,874	31,405	31,829	32,180	32,471
$2\frac{3}{8}$	44,301	4	29,850	33,032	33,970	34,680	35,239	35,689	36,062	36,369
$2\frac{1}{2}$	49,087	4	33,590	36,962	37,955	38,704	39,295	39,770	40,163	40,488
$2\frac{5}{8}$	54,119	4	37,560	41,112	42,160	42,951	43,573	44,073	44,487	44,828
$2\frac{3}{4}$	59,396	4	41,739	45,485	46,586	47,417	48,067	48,595	49,031	49,389
$2\frac{7}{8}$	64,918	$3\frac{1}{2}$	44,506	50,079	51,233	52,105	52,780	53,340	53,794	54,169
3	70,686	$3\frac{1}{2}$	49,049	54,893	56,101	57,003	57,729	58,303	58,780	59,171
$3\frac{1}{8}$	76,700	...	...	59,928	61,190	62,143	62,889	63,490	63,987	64,395
$3\frac{1}{4}$	82,958	$3\frac{1}{4}$	57,657	65,182	66,498	67,492	68,271	68,896	69,415	69,842
$3\frac{3}{8}$	89,462	...	...	70,660	72,030	73,062	73,874	74,524	75,061	75,505
$3\frac{1}{2}$	96,211	$3\frac{1}{4}$	68,192	76,353	77,781	78,855	79,700	80,374	80,931	81,391
$3\frac{5}{8}$	103,206	...	...	82,276	83,755	84,865	85,742	86,444	87,023	87,493
$3\frac{3}{4}$	110,447	3	78,171	88,416	89,948	91,098	92,006	92,732	93,333	93,824
$3\frac{7}{8}$	117,932	...	...	94,774	96,364	97,550	98,490	99,243	99,864	100,374
4	125,664	3	90,260	101,354	102,997	104,229	105,201	105,975	106,614	107,145

Stay Tubes, Sectional Areas, and Working Loads.

Number of Threads per Inch		8	9	10	11	12	Plus Thrds.	8	9	10	11	12	Plus Threads
External Diameter	Thickness of Metal	Sectional Area of Metal						Permissible Working Load (7,500 lbs.)					
Ins.	In.	Sq. In.	Sq. In.	Sq. In.	Sq. In.	Sq. In.	Sq. In.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
2	¼	·89	·94	·98	1·02	1·05	1·37	6,690	7,080	7,380	7,650	7,870	10,310
	5/16	1·17	1·22	1·26	1·30	1·33	1·65	8,800	9,190	9,520	9,770	9,990	12,420
	⅜	1·43	1·48	1·52	1·56	1·59	1·91	10,740	11,130	11,430	11,790	11,920	14,360
2¼	¼	1·02	1·08	1·13	1·17	1·20	1·57	7,690	8,130	8,480	8,790	9,040	11,780
	5/16	1·35	1·41	1·46	1·50	1·53	1·94	10,180	10,620	10,970	11,270	11,520	14,260
	⅜	1·66	1·72	1·77	1·81	1·84	2·20	12,480	12,920	13,270	13,570	13,820	16,570
2½	¼	1·16	1·22	1·27	1·32	1·35	1·76	8,680	9,180	9,570	9,910	10,190	13,230
	5/16	1·54	1·60	1·65	1·70	1·74	2·14	11,540	12,040	12,430	12,770	13,050	16,100
	⅜	1·89	1·96	2·01	2·05	2·09	2·50	14,210	14,710	15,100	15,440	15,720	18,770
2¾	¼	1·29	1·36	1·42	1·47	1·51	1·96	9,690	10,230	10,670	11,040	11,340	14,730
	5/16	1·72	1·79	1·85	1·90	1·94	2·39	12,920	13,470	13,890	14,270	14,580	17,940
	⅜	2·12	2·20	2·25	2·30	2·34	2·79	15,850	16,500	16,820	17,300	17,610	20,990
3	¼	1·42	1·50	1·56	1·62	1·66	2·15	10,695	11,295	11,760	12,170	12,510	16,190
	5/16	1·90	1·98	2·04	2·10	2·14	2·63	14,280	14,880	15,360	15,760	16,080	19,780
	⅜	2·35	2·43	2·50	2·55	2·60	3·09	17,690	18,290	18,760	19,220	19,510	23,190
	7/16	2·78	2·86	2·93	2·98	3·03	3·52	20,910	21,510	21,990	22,390	22,710	26,410
3¼	¼	1·56	1·64	1·71	1·77	1·82	2·35	11,700	12,360	12,870	13,310	13,650	17,670
	5/16	2·08	2·17	2·24	2·30	2·35	2·88	15,650	16,310	16,820	17,260	17,610	21,630
	⅜	2·59	2·67	2·74	2·80	2·85	3·38	19,420	20,080	20,590	21,030	21,400	25,400
	7/16	3·06	3·15	3·22	3·28	3·33	3·86	23,020	23,670	24,180	24,630	24,990	29,000
3½	¼	1·69	1·78	1·86	1·92	1·97	2·55	12,690	13,400	13,960	14,430	14,830	19,140
	5/16	2·27	2·36	2·43	2·50	2·55	3·12	17,020	17,730	18,290	18,760	19,160	23,460
	⅜	2·82	2·91	2·99	3·05	3·10	3·68	21,170	21,870	22,440	22,910	23,280	27,390
	7/16	3·35	3·44	3·51	3·58	3·63	4·20	25,120	25,830	26,390	26,860	27,230	31,560
3¾	¼	1·82	1·92	2·00	2·07	2·13	2·74	13,690	14,440	15,960	15,570	15,060	20,610
	5/16	2·45	2·55	2·63	2·70	2·75	3·37	18,390	19,150	19,750	20,260	20,660	25,310
	⅜	3·05	3·15	3·23	3·30	3·36	3·97	22,890	23,660	24,260	24,770	25,200	29,820
	7/16	3·63	3·73	3·81	3·88	3·93	4·55	27,220	27,990	28,590	29,100	29,520	34,140

CHAPTER XII

BOARD OF TRADE BOILER RULES

EXTRACTS FROM THE 'INSTRUCTIONS AS TO THE SURVEY OF PASSENGER STEAM SHIPS.' PUBLISHED BY THE BOARD OF TRADE, 1913.

(*Paragraphs referring to Surveyor's duties and to old boilers, and the section referring exclusively to iron boilers have been omitted.*)

Boilers and Superheaters.

101. Fixing of Working Pressure; and Examination of Tracings.—The Surveyor should fix the working pressure for boilers by a series of calculations of the strength of the various parts, taking into consideration the workmanship and material. The Board of Trade have arranged to receive, for examination by their Surveyors, plans and particulars of boilers before the commencement of manufacture, by these means hoping to prevent questions arising after the boilers are finished and on board. This practice has been found to work well in saving the time of the Surveyors, and in preventing expense, inconvenience, and delay to owners.

Tracings of boilers may be received for examination upon payment of the usual fee, and the Surveyors may proceed as far as witnessing the hydraulic test and making the subsequent internal examination before any further instalment of the passenger certificate survey fee is paid. Engineers and boil-makers should be advised of this arrangement. . . .

The Surveyor cannot declare a boiler to be safe unless he is fully informed as to its construction, material, and workmanship. He should, therefore, be very careful how he ventures to give a declaration for a boiler that he is not called upon to survey until after it is completed, and fixed in the ship.

105. Liquid Fuel.—A declaration should not be granted for a vessel in which liquid fuel is to be used without previously submitting the case for the Board's consideration. In all cases in which it is intended that such fuel should be used on board passenger vessels, a full description of the fuel and of the apparatus for supplying it to the boilers, and of the method and place proposed for stowing it should be given. Complete drawings should be supplied by the owners, so as to enable the Board to decide whether the arrangements are such as can be considered reasonably safe to be passed by the Surveyor. (For the regulations regarding motor boats, *see* Part V.)

108. Makers of Steel.—When the steel is not to be made by any of the following makers, the case must receive the special consideration of the Board, and this should be specially noted by the Surveyors.

Messrs. W. Beardmore & Co.	For plates, angles, stay and rivet bars.
,, J. Brown & Co.	
,, Cammel, Laird & Co.	
,, The Clydebridge Steel Co. ...	
,, D. Colville and Sons	
,, The Consett Iron Co., Ltd. ...	
,, The Glasgow Iron and Steel Co., Ltd.	
,, Guest, Keen and Nettlefolds, Ltd.	
,, The Leeds Forge Co.	
,, Palmer's Shipbuilding and Iron Co., Ltd.	
,, The South Durham Steel and Iron Co., Ltd.	
,, John Spencer and Sons, Ltd. ...	
,, The Steel Co. of Scotland ...	
,, The Weardale Steel, Coal and Coke Co., Ltd.	
,, Stewarts and Lloyds, Ltd. ...	
,, The Pargate Iron and Steel Co.	
,, J. Dunlop & Co....	For plates.
,, Baldwins, Ltd.	For stay and rivet bars.
,, The Lanarkshire Steel Co., Ltd.	
,, Waverley Iron & Steel Co. ...	
,, Dorman, Long & Co., Ltd. ...	For stay and rivet bars, angles and flats.
,, The Wigan Coal and Iron Co., Ltd.	
,, Cargo Fleet Iron and Steel Co.	For bars of angle, channel, and H-sections.

109. Process of Manufacture: Annealing Shell Plates.—All steel intended for use in the construction of boilers and for forgings should be made by the open-hearth process. Boiler plates should be of acid quality; but the other portions of boilers, and forgings, may be made of either acid or basic steel. In the case of castings, the steel may be made by any process which has been approved by the Board of Trade.

It is very desirable that all plates (especially those of great thickness) intended for the shells of boilers should be annealed, but it is important that the process should be carefully effected, the plates being heated singly to a suitable temperature, in a properly constructed furnace, and allowed to cool separately and uniformally out of the furnace.

110. Selection and Treatment of Test-Pieces.—All the test-pieces required should be selected by the Surveyor, and, except where otherwise specified in these regulations, the tests should be made in his presence at the place of manufacture, and before the despatch of the material.

Test specimens may be cut from the rolled material either lengthwise or crosswise, but cross check tests of boiler shell plates and butt straps should be made occasionly, the specimens being stamped by the Surveyor before they are cut from the plates.

If any material is annealed or otherwise heat-treated, the test-pieces should be similarly and simultaneously treated with the material before they are tested. The specimens should not be further heated, excepting those of temper-bending tests which should be heated to a blood red and quenched in water at a temperature not exceeding 80° Fahr.

As regards forgings, the test-pieces should be taken from a part of the forgings of sectional dimensions not less than those of the body of the forging, and they should be machined to size without further forging down.

Test-pieces should not be cut off forgings or castings until they have been stamped by the Surveyor after the annealing has been completed.

When a number of articles are cut from one plate, bar or forging, the number of tests required should be the same as that required from the original piece, provided the articles have not been further heated or forged, and can be indentified as having formed part of the original piece.

When a number of small forgings are made from the same ingot, or a number of small castings from the same charge of steel, the full number of tests specified hereafter need not be made; tensile and bending tests at the rate of one of each for every four articles will, as a rule, in such cases be sufficient.

116. General Instructions.—The following instructions regarding boiler material refer to steel of ordinary mild quality. Where high tensile steel is used, the requirements specified by the Board in each case should be adhered to.

Plates.

117. Number and Nature of Tests.—A tensile and a bending test should be taken from each plate, as rolled; but, when the weight of the plate exceeds two and a half tons, a tensile and a bending test should be taken from each end. Bending tests only, however, need be made from plates for which a greater stress than is allowed for iron is not desired.

The plates for man-hole doors, and for compensating rings around the openings for doors, should be tested in the usual manner.

118. Tensile Strength and Elongation.—The tensile strength of plates not intended to be worked in the fire or exposed to flame, for which special limits have not been sanctioned, should be between 27 and 32 tons per square inch: that of other plates, from 26 to 30 tons per square inch. The elongation should not be less than 20 per cent. in a length of 8 inches for material $\frac{3}{8}$ inch in thickness and upwards which is required to have a tensile strength of 27 to 32 tons per square inch, and not less than 23 per cent. if the tensile strength is required to be between 26 and 30 tons per square inch. For material under $\frac{3}{8}$ inch in thickness, the elongation may be reduced; but, for each eighth of an inch of diminution in thickness, the reduction should not be more than 3 per cent. below the elongations mentioned.

119. **Bend Tests.**—Bending test-pieces should withstand being bent, without fracture, until the sides are parallel at a distance apart of not more than three times the thickness of the specimen. The bending tests of plates not intended to be worked in the fire or exposed to flame may be made with strips in the same condition as the plates: those from other plates should be made with strips which have been tempered.

120. **Witnessing of Tests by Surveyor.**—It is very desirable that the Surveyor should witness the whole of these tests; but, in the case of plates made from steel manufactured by any of the makers whose names are given in Section 108, he need only select and witness tests from one in four of the plates of each thickness, unless the weight of the plate is over two and a half tons, or special limits of strength, or, in the case of shell plates, a minimum tensile strength exceeding 27 tons is required, in which cases the Surveyor should see the tests made from all the plates.

Angle, Rivet, and Stay Bars.

121. **Number and Nature of Tests.**—One tensile test should be made from each 15, or part of 15, bars rolled of each section or diameter from the same charge, but not less than two tensile tests should be made, unless the total number of bars rolled from the same charge is 8, or less than 8, and the bars are of the same section or diameter, when one test will suffice. For round bars $1\frac{3}{4}$ inches in diameter, and under, the numbers 50 and 20 should be substituted for 15 and 8 respectively, as determining the number of tests necessary.

A cold and a temper bend should be made from stay bars in the same proportion as that in which tensile tests are required; and a cold or temper bend should be made from each angle or tee bar rolled. No bending tests need be made from rivet bars.

122. **Tensile Strength and Elongation of Stays, Angles, and Tee Bars.**—The tensile strength of longitudinal stays, angles, and tee bars should be between 27 and 32 tons per square inch, with an elongation of not less than 20 per cent. measured on the appropriate standard test-piece (A or B). For bars for combustion-chamber stays, the tensile strength should be between 26 and 32 tons per square inch, with an elongation of not less than 23 per cent. measured on the standard test-piece. When, however, stay bars are tested on a gauge length of four times the diameter (test-piece F), the elongations should be 24 per cent. and 28 per cent. respectively.

For tee or angle bars under $\frac{3}{8}$ inch in thickness, the elongation may be 3 per cent. below that specified for plates.

123. **Bend Tests.**—Bending test-pieces should withstand being bent, without fracture, until the sides are parallel at a distance apart of not more than three times the thickness or diameter of the specimen.

124. **Rivet Bars.**—The tensile strength of rivet bars should be be-

tween 26 and 30 tons per square inch, with an elongation of not less than 25 per cent. measured on the standard test-piece B, or 30 per cent. if measured on test-piece F.

Rivets.

125. **Nature of Tests.**—A few rivets of each size should be selected by the Surveyor from the bulk, and should be subjected to the following tests:—

(*a*) The rivet shanks to be bent cold and hammered until the two parts of the shank touch, without fracture on the outside of the bend.

(*b*) The rivet heads to be flattened, while hot, until their diameter is two and a half times the diameter of the shank, without cracking at the edges.

A few check tensile tests of shell rivets should also be made when the Surveyor considers it necessary. The elongation should, when practicable, be taken in a length of two and a half times the diameter of the prepared part; the tensile strength should be from 27 to 32 tons per square inch and the contraction of area about 60 per cent.

Tubes.

126. **Number and Nature of Tests.**—

(a) *Solid-drawn Steel Steam Pipes, Boiler Tubes, etc., subject to Internal Pressure.*

The makers should take a few samples from each batch of tubes and test them for tensile strength and elongation. A bending test should also be made by them from the scrap end of each tube drawn.

Tensile and bending tests should also be made in the Surveyor's presence from specimens selected by him in the following proportion, from the tubes made from *each charge*:—

Tubes up to and including 3 inches in diameter: 1 in 40 or part thereof.

Tubes above 3 inches up to and including 4 inches in diameter: 1 in 20 or part thereof.

Tubes above 4 inches up to and including 5 inches in diameter: 1 in 10 or part thereof.

Tubes above 5 inches up to and including 7 inches in diameter: 1 in 6 or part thereof.

Tubes above 7 inches in diameter: 1 in 4 or part therefore.

The tensile strength should range between 23 and 30 tons per square inch, and the elongation should not be less than 20 per cent. in a length of 8 inches, or 18 per cent. if the thickness of the tubes is less than $\frac{1}{4}$ of an inch.

All the tubes should be tested by the makers to a suitable hydraulic pressure, and the tests of at least 25 per cent. of them should be witnessed by the Surveyor. The tests of all steam pipes should, however, be witnessed by the Surveyor on completion of the pipes, that is after they have been bent to shape and the flanges have been secured in position (*see* Section 174).

(b) *Solid-drawn Steel Tubes subject to External Pressure.*

If no allowance over that given for iron tubes is required, a few bending tests should be made from the scrap ends of the stay tubes, but special tests need not be made from the ordinary tubes if the Surveyor finds the general quality of the material satisfactory and he is satisfied.

If allowance over iron is required, tensile and bending tests should be witnessed by the Surveyor in the proportions given for solid-drawn steel steam pipes. The tensile strength should range between 23 and 30 tons per square inch, and the elongation should be at least that required for similar solid-drawn steam pipes.

All the tubes should be tested by the makers to a suitable hydraulic pressure, but the tests need not be witnessed by the Surveyor if he is satisfied that the tubes have been duly tested by the makers.

(c) *Steel Lap-Welded Tubes subject to External Pressure.*

(i.) *Steel Tubes for which no Allowance over Iron is required.*—A few bending tests should be made from the scrap ends of the stay tubes or the strips from which they are made, but special tests need not be made from the ordinary tubes if the general nature of the material has been found satisfactory and the Surveyor is satisfied.

(ii.) *Steel Stay Tubes for which Allowance over Iron is required.*—Tensile and bending tests should be made from 25 per cent. of the *strips* from which the tubes are made. The tensile strength should range between 23 and 30 tons per square inch, and the elongation should be at least 20 per cent. in a length of 8 inches when the strips are tested in their normal condition.

All the tubes should be tested by the makers to a suitable hydraulic pressure, but the tests need not be witnessed by the Surveyor if he is satisfied that the tubes have been duly tested by the makers.

127. General Requirements for all Tubes.—The hydraulic test should not, in any case, be less than three times the working pressure, and it should not exceed four times the pressure given by the rule :—

$$\frac{6000 \times \text{thickness in inches}}{\text{inside diameter in inches}} = \text{pressure,}$$

in the case of lap-welded tubes, or five times that pressure in the case of solid-drawn steel tubes.

All the tests mentioned should be made in the Surveyor's presence, except where otherwise stated, and such means as may be necessary should be taken to satisfy the Surveyor that the specimens he may have to test have been cut from the tubes they represent.

If any of the aforesaid tubes are made in long lengths and passed by the Surveyor in that condition, the number of tests required may be calculated on the number of tubes as made, notwithstanding that they may afterwards be cut up into shorter lengths.

Steel tubes should be made of open-hearth acid steel, unless material of other quality has been specially approved for the purpose. Solid-drawn tubes of a thickness exceeding $\frac{1}{4}$ inch should be finished by the hot-drawn process, unless cold-drawing has been specially sanctioned, and all cold-drawn tubes should afterwards be efficiently annealed.

Forgings.

128. **Number and Nature of Tests.**—At least one tensile and one bend test should be taken from each forging; but, if the weight exceeds three tons, a tensile and a bending test should be taken from each end.

129. **Tensile Strength and Elongation.**—The tensile strength of steel forgings should not exceed 40 tons per square inch; and the elongation measured on the appropriate standard test-piece C, D, or E, should not be less than 17 per cent. for 40-ton steel; and in no case may the sum of the tensile strength and the corresponding elongation be less than 57.

130. **Bend Tests.**—The bending test-pieces should withstand being bent without fracture through an angle of 180°, the internal radius of the bend being not greater than that specified below:—

Maximum specified tensile strength of forging.	Internal radius of test-piece after bending.
	inch.
Up to 32 tons per square inch	$\frac{1}{4}$
Above 32 tons and up to 36 tons per square inch	$\frac{3}{8}$
Above 36 tons and up to 40 tons per square inch	$\frac{5}{8}$

131. **General Instructions.**—The forgings should be made from sound ingots, and not more than the lower two-thirds of the ingot may be utilised for forging. The sectional area of the body of the forging may not exceed one-fifth of the sectional area of the original ingot; and no part of the forging should have more than two-thirds of the sectional area of the ignot. All ignot steel forgings should, after completion, be thoroughly annealed at a uniform temperature; and, if any subsequent heating is done, the forging should, if required by the Surveyor, be again annealed.

Castings.

132. **Number and Nature of Tests.**—No tests need be made from unimportant steel castings or from steel castings which are used for articles usually made of cast-iron, if the scantlings are not materially reduced below what would be required if cast-iron were used. All other steel castings should be tested as follows:—

At least one tensile and one bending test should be made from the castings from each charge; and, where a casting is made from more than one charge, at least four tensile and four bending tests should be made from pieces cast as far apart as possible on the casting, and as near the top and the bottom respectively as practicable.

Where more than one casting is made from one charge, at least one tensile and one bending test should be made from the castings run from one common pouring head; but separate tests should be made from each casting or set of castings run from each separate pouring head. Small castings may, however, be dealt with in accordance with the provisions of Section 110.

133. Tensile Strength and Elongation.—The tensile strength may range from 26 to 40 tons per square inch, with an elongation, measured on the standard test-piece C, D, or E, of not less than 15 per cent. If, however, the castings are to be used for the more important pieces of machinery, such as pistons, etc., or for articles usually made of wrought material, the elongation should not be less than 20 per cent. where the corresponding tensile strength is between 26 and 35 tons per square inch.

134. Bend Tests.—The bending test-pieces should withstand being bent, without fracture, through an angle of 60° if the tensile strength is between 35 and 40 tons per square inch, and, in the case of other castings, through an angle of 90°: but, if they are required to be of the superior quality previously referred to, the angle should not be less than 120°.

The internal radius of the bend in each case should not be greater than one inch.

135. Annealing.—All steel castings should be thoroughly annealed at a uniform temperature, and should be allowed to cool down prior to removal from the annealing furnace; and, if subsequently heated, with the Surveyor's approval, should again be similarly annealed, if required by the Surveyor.

Steel Boilers.

136. Thickness of Plates: Drilling, Welding and Annealing.—The thickness of plates, other than tube strips, used in the construction of boilers should not be less than $\frac{5}{16}$ inch.

It is expected that the rivet holes will be drilled, and not punched. Plates that are drilled in place should be taken apart and the burr taken off, and the holes slightly countersunk from the outside.

Butt straps should be cut from plates, and not from bars.

Steel plates which have been welded should not be passed if subject to a tensile stress, and those welded and subject to a compressive stress should be efficiently annealed.

Local heating of the plates should be avoided, as many plates have failed from having been so treated.

All plates that have been flanged or locally heated, and all stays and stay tubes which have been locally heated, should be carefully annealed after being so treated.

137. Cylindrical Boiler Shells.—The Board of Trade consider that boilers well constructed, well designed, and made of good material should be allowed an advantage in the matter of working pressure over boilers inferior in any of the above respects, as, unless this is done, the superior boiler is placed at a disadvantage, and good workmanship and material will be discouraged. They have therefore caused the following rules to be prepared:—

When the cylindrical shells of boilers are made of material which has been duly tested and approved, with all the rivet holes drilled in place and all the seams fitted with double butt straps, each of at least five-eighths the thickness of the plates they cover, and all the seams at least double riveted with rivets having an allowance of not more than 87·5 per cent. over the single shear, provided that the boilers have been open to inspection during the whole period of construction, then 4·5 may be used as the factor of safety, the minimum actual tensile strength of the plates being used in calculating the working pressure.

When the above conditions are not complied with, the additions in the following scale should be made to the factor of safety, according to the circumstances of each case :—

Longitudinal seams.	A†	·15	To be added if all the holes are fair and good, but drilled out of place after bending.
	B†	·3	To be added if all the holes are fair and good, but drilled before bending.
	C	·2	To be added if double butt straps are not fitted, and the seams are lapped and double riveted.
	D	·1	To be added if double butt straps are not fitted, and the seams are lapped and treble riveted.
	E	·3	To be added if only single butt straps are fitted, and the seams are double riveted.
	F	·15	To be added if only single butt straps are fitted, and the seams are treble riveted.
	G	1·0	To be added if any joint is single riveted.
	H*	·4	To be added if there are two or more belts of plates, and the seams are not properly crossed.
Circumferential seams.	I†	·1	To be added if the holes are fair and good, but drilled out of place after bending.
	J†	·15	To be added if the holes are fair and good, but drilled before bending.
	K	·1	To be added if the seams are fitted with single butt straps, and are double riveted.
	L	·2	To be added if the seams are fitted with single butt straps, and are single riveted.
	M	·1	To be added if the seams are fitted with double butt straps, and are single riveted.
	N	·1	To be added if the seams are lapped and double riveted.
	O	·2	To be added if the seams are lapped and single riveted.
	P‡	·3	To be added if the boiler is of such a length as to fire from both ends, or is of unusual length, as in the case of flue boilers, and the seams are fitted as described opposite K, M, or N, but, if the seams are as described opposite L or O, P·4 should be added—in lieu of the addition designated by K, M, N, L or O, as the case may be.

* The factor may be increased still further if the workmanship is such as, in the Surveyor's judgment, to render such increase necessary.

† If the holes are to be rimered or bored out in place, the case should be submitted to the Board as to the reduction or omission of A, B, I, and J, as heretofore.

‡ If the middle circumferential seams are double strapped and double riveted, or lapped and treble riveted, and the calculated strength is not less than 65 per cent. of the solid plate, no addition to the factor need be made in respect of the length of the boiler.

The joints referred to in the table as circumferential seams do not include the joints between the shell and end plates, which it is expected will be at least double riveted when the thickness of the shell exceeds $\frac{1}{2}$ inch.

When surveying boilers that have not been open to inspection during construction, the case should be submitted to the Board as to the factor to be used.

The strength of ordinary joints is found by the following formulæ:—

$$\frac{100\,(d - d)}{p} = \left\{\begin{array}{l}\text{per-centage of strength of plate at joint as com-}\\ \text{pared with the solid plate; and}\end{array}\right.$$

$$\frac{100 \times S_2 \times a \times n \times C}{S_1 \times p \times T} = \left\{\begin{array}{l}\text{per-centage of strength of rivets as}\\ \text{compared with the solid plate;}\end{array}\right.$$

where p = pitch of rivets, in inches;
d = diameter of rivets, in inches;
a = area of one rivet, in square inches;
n = number of rows of rivets;
T = thickness of plate, in inches;
C = 1 for rivets in single shear;
C = 1·875 for rivets in double shear;
S_1 = minimum tensile strength of plates, in tons per square inch; and
S_2 = shearing strength of rivets, in tons per square inch, which is taken to be 23.

If the percentage strength of the rivets in the longitudinal seams is found by calculation to be less than the calculated percentage strength of the plate, the working pressure of the shell should be calculated in the following manner from each percentage:—

$$\frac{S_1 \times 2240 \times {}^{\circ}\!/_{\circ} \text{ strength of joint} \times 2\,T}{D \times \text{factor of safety} \times 100} = \text{pressure};$$

D being the inside diameter of the boiler, in inches.

When using the percentage strength of the plate, 4·5 plus the additions suitable for the method of construction should be used as the nominal factor of safety; but, when using the percentage strength of the rivets, 4·5 should be used as the factor of safety. The smaller of the two pressures so found is the pressure to be allowed per square inch on the safety valves. (See the formulæ as given in detail in Section 138.)

Maximum Pitches for Riveted Joints.

T = thickness of plate in inches;
p = maximum pitch of rivets in inches, provided it does not exceed $10\frac{1}{2}$ inches; and
C = constant applicable from the following table:—

Number of Rivets in one Pitch.	Constants for Lap Joints.	Constants for Double Butt Strap Joints.
1	1·31	1·75
2	2·62	3·50
3	3·47	4·63
4	4·14	5·52
5	—	6·00

$$(C \times T) + 1\tfrac{5}{8} = p.$$

When the work is first class, such pitches may be adopted so far as safety is concerned, yet, in some cases, it may be well not to adopt the greatest pitch found by the formula. The maximum pitch should *not*, however, exceed $10\frac{1}{2}$ inches with the thickest plates for boiler shells. If in any case the pitch is found to exceed that arrived at by the foregoing formula, for the particular description of joint and thickness of plate, such pitches should *not* be passed, but all such cases should be reported.

139. Openings in Shells ; Doors, etc.—The openings in the shells of cylindrical boilers should have their shorter axes placed longitudinally.

Compensating rings of at least the same effective sectional area as the plates cut out, and not less in thickness than the plates to which they are attached, should be fitted around all man-holes and openings, or the surrounding portion of the plates otherwise efficiently stiffened.

It is very desirable that the compensating rings around openings in flat surfaces be made of **L**- or **T**-bars. When a ring is not fitted around such an opening and the plate is flanged for compensation, the total depth, D, of the flange should not be less than that given by the following equation :—

$$D = \sqrt{\text{width of opening} \times \text{thickness of plate.}}$$

Cast-iron doors should not be passed, and in all cases in which the Surveyors find that cast-iron is employed in the construction of boilers in such a manner as to be subjected to the pressure of steam or water, they should report the circumstances to the Board of Trade.

The neutral part of the boiler shells under steam domes should be efficiently stiffened and stayed, as serious accidents have arisen from the want of such precautions.

In connection with openings in the ends of steam receivers, etc., the Surveyors should bear in mind that, although the usual strengthening plate may be fitted around such openings, the end is still weaker than if no opening had been made, and that the fact of there being a plate or the flange of a neck-piece riveted around the opening may not remove the necessity for having the end suitably stayed.

140. Hemispherical Ends.—Hemispherical ends subject to internal pressure may be allowed double the pressure that is suitable for a cylinder of the same diameter and thickness.

141. Dished Ends.—Ends of steam receivers which are dished and flanged in a hydraulic press, the whole end being operated upon at each heat, may be passed without stays, provided that the radius of the end is not greater than the diameter of the shell to which it is attached and does not exceed 4 feet, that the outer radius of the flange at the root is not less than 3 inches, that the end is made of mild steel of the usual quality, and that it is efficiently annealed after the completion of the setting and flanging; but the maximum pressure allowed without stays should not exceed that found by the following formula:—

$$\frac{90000 \times T \times h}{D^2} = \text{working pressure};$$

where T = thickness of end, in inches;
D = diameter of shell, inside, in inches;
h = distance, in inches, from the centre of the end on the inner side to a straight line passing through the points where the curve of the end due to the inner radius, and produced, would intersect the sides of the cylindrical shell, produced if necessary, at the diameter D, and

$$= R - \sqrt{R^2 - \frac{D^2}{4}}, \text{ in which } R = \begin{cases}\text{inner radius of end, in} \\ \text{inches.}\end{cases}$$

If there is a man-hole in the end, T should exceed that found by the formula by $\frac{1}{8}$ inch.

The above instructions do not apply to dished ends of vertical donkey boilers, which are subject to the thrust of the uptake in addition to the pressure of steam.

142. Stays for Dished Ends.—Stays, properly distributed, should be fitted to dished ends which are not of the thickness required for flat ends, or which do not comply with the requirements stated in Section 141; but, if the ends are sufficient for the pressure needed, when considered as portions of spheres, the stays, if of solid steel, may have a nominal stress of 18,000 lbs. per square inch of net section. If dished ends are not equal to the pressure needed when considered as portions of spheres, they should be stayed as flat surfaces.

143. Stays for Flat Surfaces.—Solid steel stays for supporting flat surfaces may be allowed a working stress of 9,000 lbs. per square inch of net section, provided the tensile strength and elongation are as stated in Section 122. Steel stays which have been welded should not be passed. (This does not apply to stay tubes which are welded longitudinally.)

When the threads of longitudinal stays are finer than six per inch, the depth of the external nuts should be at least $1\frac{1}{4}$ times the diameter of the stay.

The areas of diagonal stays are found in the following way: Find the area of a direct stay needed to support the surface, multiply this area by the length of the diagonal stay, and divide the product by the length of a line drawn at right angles to the surface supported to the end of the diagonal stay; the quotient will be the area of the diagonal stay required.

When gusset stays are used, their area should be in excess of that found in the above way.

144. Stay Tubes.—Stay tubes made of steel which has been tested and has been found to comply with the higher requirements stated in Section 126, may be allowed a stress not exceeding 7,500 lbs. per square inch of net section, provided that their net thickness is in no case less than $\frac{1}{4}$ inch.

145. Ordinary Tubes.—Ordinary smoke tubes, if made of steel for which no special allowance is required, should be of a thickness at least equal to that found by the formula given in Section 156 for similar tubes of iron.

146. Girders for Flat Surfaces.—When the tops of combustion-boxes or other parts of a boiler are supported by stays held by solid girders of rectangular section, the following formula should be used for finding the working pressure to be allowed for the girders, assuming that they are not subject to a temperature greater than the ordinary heat of steam, and that, in the case of combustion-boxes, the ends are properly bedded to the edges of the tube-plate and the back plate of the combustion-box :—

$$\frac{C \times d^2 \times T}{(W - P)\, D \times L} = \text{working pressure ;}$$

where W = width of combustion-box, in inches ;
P = pitch of supporting stays, in inches ;
D = distance between the girders from centre to centre, in inches ;
L = length of girder, in feet ;
d = depth of girder, in inches ;
T = thickness of girder, in inches ;
N = number of supporting stays ;

$$C = \frac{N \times 1320^*}{N + 1} \text{ when the number of stays is odd ; and}$$

$$C = \frac{(N + 1)\, 1320^*}{N + 2} \text{ when the number of stays is even.}$$

The working pressure for the supporting stays and for the plate between them should be determined by the rules for ordinary stays and plates.

147. Flat Surfaces.—The pressure on plates forming flat surfaces is found by the following formula :—

$$\frac{C\ (or\ c) \times (T + 1)^2}{S - 6} = \text{working pressure ;}$$

where T = thickness of the plate, in sixteenths of an inch ;
S = surface supported, in sqare inches ; and
C = constant for steel } according to the following
c = *constant for iron* } circumstances :—

C = 240, c = *192* { when the plates are not exposed to the impact of heat or flame, and the stays are fitted with nuts on both sides of the plates, and doubling strips, not less in width than two-thirds the pitch of the stays and of the thickness of the plates, are securely riveted to the outside of the plates they cover ;

* *1,200 for iron.*

$C = 210$ $c = 168$	when the plates are not exposed to the impact of heat or flame, and the stays are fitted with nuts on both sides of the plates, and with washers, not less in diameter than two-thirds the pitch of the stays and of the same thickness as the plates, securely riveted to the outside of the plates they cover ;
$C = 165$ $c = 132$	when the plates are not exposed to the impact of heat or flame, and the stays are fitted with nuts on both sides of the plates, and with washers outside the plates, at least three times the diameter of the stay and two-thirds the thickness of the plates they cover ;
$C = 150$ $c = 120$	when the plates are not exposed to the impact of heat or flame, and the stays are fitted with nuts on both sides of the plates ;
$C = 112{\cdot}5$ $c = 90$	when tube-plates are not exposed to the direct impact of heat or flame, and the stays are fitted with nuts ;
$C = 77$ $c = 70$	when tube-plates are not exposed to the direct impact of heat or flame, and the stay tubes are screwed into the plates and expanded ;
$C = 77$ $c = 70$	when the plates are not exposed to the impact of heat or flame, and the stays are screwed into the plates and riveted over ;
$C = 75$ $c = 60$	when the plates are not exposed to the impact of heat, with steam in contact with the plates, and the stays fitted with nuts and washers, the latter being at least three times the diameter of the stay and two-thirds the thickness of the plates they cover ;
$C = 67{\cdot}5$ $c = 54$	when the plates are exposed to the impact of heat, with steam in contact with the plates, and the stays fitted with nuts only ;
$C = 100$ $c = 80$	when the plates are exposed to the impact of heat or flame, with water in contact with the plates, and the stays screwed into the plates and fitted with nuts ;
$C = 66$ $c = 60$	when the plates are exposed to the impact of heat or flame, with water in contact with the plates, and the stays screwed into the plates and having the ends riveted over to form substantial heads ; and
$C = 39.6$ $c = 36$	when the plates are exposed to the impact of heat, with steam in contact with the plates, with the stays screwed into the plates and having the ends riveted over to form substantial heads.

When the riveted ends of screwed stays are much worn, or when the nuts are burned, the constants should be reduced, but the Surveyor must act according to the circumstances that present themselves at the time of the survey, and it is expected that, in cases where the riveted ends of screwed stays in the combustion-boxes and furnaces are found in this state, it will be often necessary to reduce the constant 66 to about 40.

When the plates are not exposed to the impact of heat or flame, and doubling plates covering the whole of the flat surfaces are riveted to the plates, the working pressure may be found by the following formula :—

$$\frac{C\ (or\ c)\ (T + 1)^2 + C\ (or\ c)\ (T_1 + 1)^2}{S - 6} = \text{working pressure};$$

where T = thickness of the plate, in sixteenths of an inch ;
T_1 = thickness of the doubling plate, in sixteenths of an inch ;
S = surface supported, in square inches ; and
C (*or c*) = constant applicable to the case, as previously given.

When doubling plates do not cover the whole of the flat surfaces, but are fitted between the rows of supporting stays, the strength allowed for them should be two-thirds only of that which would be allowed for similar doubling plates extending beyond and embracing the supporting stays.

In calculating the working pressure of the portion of tube-plates between the boxes of tubes, the value of S in the above formula should be found as follows :—

$$\frac{D^2 + d^2}{2} = S;$$

where D = horizontal pitch of the stay tubes, in inches ;
and d = vertical do. do. do.

The pitches should be measured from centre to centre of the stay tubes, and no deduction should be made for any tubes in the contained surface.

In the body of tube-plates, the value of S may be found in the ordinary way, and the area of the tubes in the space bounded by the stay tubes may be deducted.

In cases where plates are stiffened by **T**- or **L**-bars, and a greater pressure is required for the plates than is allowed by the use of the foregoing constants, the case should be submitted for the consideration of the Board of Trade.

When dealing with end plates in the vicinity of horizontal furnaces such as are now generally used in boilers, the Surveyors need not require stays to be fitted if the flat portion of the plate is sufficient according to the rules for flat surfaces ; but, when the furnaces and combustion-box stays are not efficient as through stays, as in the case of certain tug-boat boilers and of other marine boilers in which furnaces having coned back ends are fitted, stays are required, and in the latter case it will often be found that three through stays, distributed around each of the lower man-holes between the furnaces, are necessary in order to afford the desired strength. The Surveyor should, however, submit the particulars of such cases for consideration when in doubt as to how to deal with them.

When a circular flat end is bolted or riveted to an outside ring or flange of a cylindrical shell, S in the formula may be taken as the area of the square inscribed in the circle passing through the centres of the bolts or rivets securing the end, provided the ring or flange is of sufficient thickness.

148. Compressive Stress on Tube-Plates.—The Surveyors should not in any case allow a greater compressive stress on the tube-plates than 14,000* lbs. per square inch, which is that used in the following formula :—

$$\frac{(D - d)\,T \times 28000\dagger}{W \times D.} = \text{working pressure};$$

where D = least horizontal distance between centres of tubes, in inches ;
d = inside diameter of ordinary tubes, in inches ;
T = thickness of tube-plate, in inches ; and
W = width of combustion-box, in inches, between the tube-plate and back of fire-box, or distance between the combustion-box tube-plates when the boiler is double-ended and the box common to the furnaces at both ends.

149. Circular Furnaces.—Circular furnaces with the longitudinal joints welded, or made with single butt straps double riveted, or double butt straps single riveted :—

$$\frac{99000\ddagger \times \text{the square of the thickness of the plate, in inches.}}{(\text{length, in feet} + 1) \times \text{diameter, in inches.}}$$

= working pressure per square inch, provided it does *not* exceed that found by the following formula :—

$$\frac{9900\,\S \times \text{thickness, in inches}}{\text{diameter, in inches}} = \begin{cases}\text{working pressure per square} \\ \text{inch.}\end{cases}$$

The second formula limits the crushing stress on the material to 4,950 ‖ lbs. per square inch.

The length is to be measured between the rings, if the furnace is made with rings.

If the longitudinal joints instead of being butted are lap-jointed in the ordinary way and double riveted, then 82,500 should be used instead of 99,000 ; but, where the lap is bevelled and so made as to give the flues the form of a *true* circle, then 88,000 may be used.

When the workmanship is not of the best quality, the constants given above, for the former rule, should be reduced, according to circumstances, and the judgment of the Surveyor. Some of the conditions of best workmanship are that the rivet holes shall be drilled after the bending is done and when the plates are in place, and the plates afterwards taken apart, the burr on the holes taken off, and the holes slightly countersunk from the outside.**

* *11,000 for iron.* † *22,000 for iron.* ‡ *90,000 for iron.*
§ *9,000 for iron.* ‖ *4,500 for iron.*

** In the case of upright fire-boxes of donkey or similar boilers, 10 per cent. should be deducted from the constant given above, applicable to the respective classes of work, and the constant for the crushing formula should be reduced by a similar amount, becoming 8,910 (*or 8,100*).

150. Furnaces Corrugated, or Ribbed and Grooved.—Machine made furnaces of the Fox, Morison or Deighton corrugated types, manufactured by The Leeds Forge Company, John Brown & Co., Sheffield, The Deighton Patent Flue and Tube Co., Leeds, and Wm. Beardmore & Co., Glasgow; of the Purves ribbed and grooved type, or Brown's cambered type, manufactured by John Brown & Co., Sheffield; and of the Fox or Morison corrugated type manufactured by Thomas Piggott & Co., Birmingham, provided they are practically true circles, and the plates are not less than $\frac{5}{16}$ inch thick, may be allowed the working pressure found by the following formula :—

$$\frac{C \times T}{D} = \text{working pressure};$$

where $C = 14{,}000$; *

T = thickness, in inches, measured at the bottom of the corrugation or camber; and

D = outside diameter, in inches, measured at the bottom of the corrugations or cambers when the furnace is of the corrugated or cambered type, or over the plain parts when it is of the ribbed and grooved description.

In the Fox furnace, the pitch of the corrugations should not exceed 6 inches; and in the Morison furnace and the Deighton furnace, the pitch should not exceed 8 inches. In these kinds of furnaces, the depth from top of corrugations outside to bottom of corrugations inside should not be less than 2 inches.

The ribs of ribbed and grooved furnaces should not be less than $1\frac{5}{16}$ inches above the plain parts, the depths of the grooves not more than $\frac{3}{4}$ inch, and the length between the centres of the ribs not over 9 inches. In Brown's cambered furnace, the thickness of metal at the centre of the ribs should be at least $\frac{3}{16}$ inch greater than the thickness at the bottom of the camber, the tops of the ribs should be curved to a radius of $1\frac{3}{8}$ inches, and the grooves beneath the ribs to a radius of $\frac{45}{64}$ inch, the height of the ribs above the bottom of the camber should not be less than $1\frac{5}{8}$ inches, and the pitch of the ribs should not be more than 9 inches.

Machine made furnaces of the bulb type, manufactured by The Leeds Forge Company, may be allowed the working pressure found by the following formula, provided that they are practically true circles, that the pitch of the bulbs does not exceed 8 inches, that the depth from the top of the bulbs to the plain parts at the centre of the pitch is not less than $2\frac{1}{4}$ inches, that the plates are not less than $\frac{5}{16}$ inch thick, and the plain parts between the bulbs are fairly uniform in thickness :—

$$\frac{15000 \times T}{D} = \text{working pressure};$$

where T = thickness of the plain parts between the bulbs, in inches; and

D = outside diameter at the middle of the plain parts between the bulbs, in inches.

* This constant only applies to the furnaces when made by the firms named. The Surveyors should report full particulars of any case in which the owners or builders of a passenger steamship propose to use furnaces of any of these types if made by other makers.

In each of these kinds of furnaces, the plain parts at the back ends should be so made that the length, measured from the waterside of the back tube-plate to the centre of the back end corrugation or rib, does not exceed 9 inches. The plain parts at the front ends should also be so made that the length of the flat, measured from the centre of the rivets by which the furnace is secured to the front end plate to the centre of the first corrugation or rib, does not exceed 9 inches. When the plain parts at the back ends are made conical, and the flange by which the attachment is made to the back tube-plate is continuous, a length of $10\frac{1}{2}$ inches may be allowed between the waterside of the back tube-plate and the centre of the first corrugation or rib. When this method of construction is adopted, the vertical section through the neck-piece should be kept as circular as is practicable, the set-up at the bottom should not exceed 8 inches measured over the plates, and in no case should the vertical axis exceed the horizontal one by more than $14\frac{1}{2}$ per cent. The plates at the ends should not be unduly thinned in the flanging.

If the furnace is riveted in two or more lengths, the case should be submitted for consideration.

151. Furnaces made up of Flanged Rings.—When horizontal furnaces of ordinary diameter are constructed of a series of rings welded longitudinally, and the ends of each ring are flanged and riveted to the next, with a caulking ring intervening, and so forming the furnace, the working pressure is found by the following formula, provided that the length of the rings, in inches, measured over the flanges, is not greater than (120 T − 12), and the flanging is performed at one heat by a suitable flanging machine, and provided also that the conditions which are printed after the formula are complied with :—

$$\frac{9900 \times T}{3 \times D}\left(5 - \frac{l + 12}{60 \times T}\right) = \text{working pressure};$$

where T = thickness of plate, in inches.
l = length of rings, in inches, and measured over the flanges ;
D = outside diameter of furnace, in inches.

The radius of the flanges on the fire side should be about $1\frac{1}{2}$ inches. The depth of the flanges from the fire side should be three times the diameter of the rivet plus $1\frac{1}{2}$ inches, and the thickness of the flanges should be as near the thickness of the body of the plate as practicable. The distance from the edge of the rivet holes to the edge of the flange should not be less than the diameter of the rivet, and the diameter of the rivets at least $\frac{3}{8}$ inch greater than the thickness of the plate. The depth of the ring between the flanges should not be less than three times the diameter of the rivets, the fire edge of the rings should be at about the termination of the curve of flange, and the thickness not less than half the thickness of the furnace plate. It is very desirable that these rings should be turned.

The holes in the flanges and rings should be drilled in place, if practicable ; but, if not drilled in place, they should be drilled smaller than the size required, and afterwards, when in place, rimered out until the holes are quite fair ; the holes should be slightly tapered, and the heads of the rivets should be of moderate size.

After all the welding, flanging, and heating is completed, each ring should be efficiently annealed in one operation.

When the flanges of the back ends of the furnaces are not continuous, and the lower parts of the back rings are supported by substantial T-bars securely riveted to the plates, the first constant used in the formula for these rings should not exceed 8,910.

The formula may also be used for determining the pressure which may be allowed on the bottoms of combustion-chambers to which furnaces, having continuous flanges at the back ends, are fitted. In this case, l may be taken as the length over plates; and no alteration of any of the constants need be made unless the full thickness of the plate is not maintained to the bottom row of stays at each side, when 40 should be substituted for 60.

IRON BOILERS.

152. Cylindrical Boiler Shells.—In calculating the pressure for which the cylindrical shells of iron boilers are suitable, the tensile strength of the plates may be taken as equal to 47,000 lbs. per square inch with the grain, and 40,000 lbs. across the grain; and the shearing strength, per square inch, of the rivets as equal to the tensile strength, per square inch, of the plates; and, if the boilers have been open to inspection during the whole period of construction, and the workmanship, material and design are of the highest class, then 5 may be used as the factor of safety, the additions specified in the rules for steel boilers being made according to the other circumstances stated. If, however, the iron is tested and the elongation, measured in a length of 10 inches, is not less than 14 per cent. with, and 8 per cent. across, the grain, and the Surveyors are otherwise satisfied as to the quality of the plates and rivets, 4·5 may be used as the factor of safety instead of 5, in which case the minimum actual tensile strength of the plates should be used in calculating the working pressure, which is to be determined by the rule given on page .

Butt straps should be of as good quality as the shell plates, and those for the longitudinal seams should be cut across the fibre.

153. Stays for Dished Ends.—Solid iron stays for supporting dished ends which are found to be equal to the pressure needed, when considered as portions of spheres, may have a nominal stress of 14,000 lbs. per square inch of net section, but the stress should not exceed 10,000 lbs. when the stays have been welded.

154. Stays for Flat Surfaces.—In the case of new boilers, the Surveyor may allow a stress not exceeding 7,000 lbs. per square inch of net section on solid iron screwed stays supporting flat surfaces, but the stress should not exceed 5,000 lbs. when the stays have been welded.

It has been brought to the notice of the Board of Trade that iron screwed stays in the combustion-chambers of high pressure marine boilers, when of suitable quality, have been found to resist the cross-breaking stresses, to which they are subjected in working, better than similar stays made of steel, and a stress of 9,000 lbs. per square inch may be allowed on such stays, if solid and unwelded, provided the bars from which they are made have been tested in the Surveyor's presence, and found to have a tensile strength of not less than 21½ tons per square inch and an elongation, measured in a length of 8 inches, of not less than 27 per cent.

Iron bars which are to be used for combustion-chamber stays intended to be subjected to a stress of more than 7,000 lbs. per square inch should be tested for tensile strength in the same way as similar steel bars, but in the following proportions: one bar in 20 when the diameter of the bars does not exceed 1 inch; one bar in 12 when not over 1½ inches; and one bar in 8 when the diameter exceeds 1½ inches. If, however, the bars are rolled at the same time and in lengths not exceeding 20 feet, half these numbers of tests may be accepted, provided the results are uniform and satisfactory, and the Surveyor is satisfied. Two satisfactory re-tests should be made for each failure in any particular batch. All the bars, as rolled, should be stamped with consecutive numbers, and any pieces into which a bar may be cut should be stamped with the original number of the bar.

In all cases in which iron bars are to be used for the making of screwed stays for which a stress of 9,000 lbs. per square inch of net section is claimed, the Surveyor should, as far as possible, satisfy himself that the bars he may select for testing purposes fairly represent in quality the parcel of bars from which they are taken; and further, he should see that the material is what it purports to be, and, whenever the tensile strength exceeds 24 tons per square inch, special precautions should be taken, by careful examination of the fracture, and other ready means, to guard against the substitution of very mild steel, or of iron mixed with steel scrap, in place of iron.

In this connection, the Surveyors are informed that mild steel is sometimes improperly termed "ingot iron," and must in all cases, as regards tests and treatment while being worked into boilers, be dealt with in accordance with the Board's instructions with respect to mild steel for boilers.

Iron.

155. Stay Tubes.—A stress of 6,000 lbs. may be allowed on the net section of iron stay tubes, provided that the net thickness is in no case less than ¼ inch.

156. Ordinary Tubes.—The thickness of ordinary smoke tubes made of iron should not be less than that found by the following formula:—

$$T = \frac{B \times D}{9000} + \cdot 085,$$

$$\text{or } B = \frac{9000\,(T - \cdot 085)}{D};$$

where T = thickness of tube, in inches;
D = outside diameter of tube, in inches; and
B = working pressure.

157. Girders, Flat Surfaces, Tube-Plates, and Furnaces.—When dealing with iron girders, flat surfaces, tube plates, and plain furnaces, the constants in the rules for steel boilers should be altered as indicated in italics.

158. General Instructions.—In other respects, the instructions, generally, regarding steel boilers apply equally to iron boilers, excepting those in which reference is made to furnaces which are not of the plain type and to the welding and heat treatment of the material.

160. Iron Cylindrical Superheaters.—The strength of the joints of cylindrical superheaters and the factor of safety are found in the same manner as in the case of cylindrical boilers and steam receivers, but, instead of using 47,000 lbs. as the tensile strength of iron, 30,000 lbs. is adopted, unless the heat or flame impinges at, or nearly at, right angles to the plate, when 22,400 lbs. is substituted.

When a superheater is constructed with a tube subject to external pressure, the working pressure should be ascertained by the rules given for circular furnaces, but the constants should be reduced as 30 to 47.

In all cases the internal steam pipes should be so fitted that the steam in flowing to them will pass over all the plates which have steam in contact with them, and are exposed to the impact of heat or flame.

161. Steel for Superheaters, etc.—The tubes for superheaters of the type in which there is a coil, or series of small tubes, subject to internal pressure, should be made of solid-drawn steel ; but, excepting for such parts, the use of steel for the heating surfaces of superheaters should be discouraged. This applies to the unshielded uptakes of all boilers, including ordinary vertical donkey boilers ; and, whenever it is proposed to use steel for this purpose, the particulars should be submitted to the Board of Trade for consideration.

The ends of superheaters, and the flat ends, etc., of all boilers, as far as the steam space extends, should be fitted with shield or baffle plates where exposed to the hot gases in the uptake.

STEAM PIPES, etc.

170. Copper Pipes.—The working pressure of well-made copper pipes, when the joints are brazed, is found by the following formula :—

$$\frac{6000 \times (T - \frac{1}{16})}{D} = \text{working pressure};$$

where T = thickness, in inches ; and

D = inside diameter, in inches.

When the pipes are solid-drawn and not over 10 inches in diameter, substitute in the foregoing formula $\frac{1}{32}$ for $\frac{1}{16}$.

In any case where the Board of Trade authorise a trial to be made of electro-deposited pipes, the Surveyors should continue, as at present, to see that the conditions on which such trial is authorised are fulfilled. All cases in which it comes to the Surveyors' knowledge that electro-deposited pipes have failed, either in the copper-smiths' works or in actual working, should be fully reported.

171. Wrought-Iron and Steel Pipes.—The internal pressure on lap-welded wrought-iron pipes made of good material, and on lap-welded or solid-drawn steel pipes, may be determined by the following formula, provided that the minimum thickness is not less than $\frac{1}{4}$ inch, and the workmanship, hydraulic test, etc., are satisfactory :—

$$\frac{6000 \times T}{D} = \text{working pressure};$$

where T = thickness, in inches ; and
D = inside diameter, in inches.

Steel pipes which have welded longitudinal seams, and are subject to internal pressure, should in each case be provided with a strap fitted externally and riveted over the weld. All steel pipes should be efficiently annealed after being heated locally for welding or bending them to shape.

The flanges of wrought-iron and of steel pipes should be made of solid wrought material of ductile quality.

Pipes for superheated steam should be made of wrought-iron or steel, and not of copper.

172. Draining Steam Pipes.—Efficient means should be provided for draining all steam pipes. Boiler stop-valves cannot be regarded as suitable for this purpose. All drain cocks or valves should be accessible, and so placed as to render it practicable to drain the water from any portion of the steam pipes or chests in connection therewith. Drain pipes should be fitted to drain cocks or valves when the latter are in such a position that the water or steam discharged from them would be likely to cause personal injury. It is desirable that the drains should be automatic in their action.

173. Expansion Joints.—The parts of a socket expansion joint subject to rubbing action should be made of brass or of other metal which will not rust.

In all cases in which such a joint is fitted to a bent steam pipe, the Surveyor should require a fixed gland and bolts, or other efficient means, to be provided to prevent the end of the pipe being forced out of the socket. This regulation should be complied with in all cases of bent pipes fitted with socket expansion joints, and it is also desirable that fixed glands and bolts should be fitted to the expansion joints of *straight* steam pipes, as cases have occurred, particularly with small straight pipes, in which the ends have been forced out of the sockets.

A socket expansion joint on a bent pipe is not a desirable arrangement, and, when adopted, the pipe should be anchored or provided with a strut at the bend, to relieve it of any undue bending stresses which might otherwise be produced by the internal pressure on a surface of the pipe equal to the area due to its bore.

174. Examination and Testing of Steam Pipes.—Surveyors should pay particular attention to the examination and testing of steam pipes, and a record of the tests should be kept in the office boiler book.

All new copper steam pipes should be tested by hydraulic pressure to not less than twice and not more than two and one-half times the working pressure. The higher test should be that usually employed. When, however, special considerations arise, the case should be fully submitted and instructions obtained before the Surveyor proceeds with the hydraulic test.

Wrought-iron lap-welded, and steel steam pipes should be tested by hydraulic pressure when new, with the flanges secured in place, to at least three times the working pressure, but a higher test pressure need not be objected to provided that it does not exceed four times the pressure found by the rule in Section 171, or five times that pressure in the case of solid-drawn steel pipes.

As regards old pipes, the Surveyor may, at any time he thinks it necessary, before he gives a declaration, require them to be tested by hydraulic pressure to satisfy himself as to any doubtful part, or parts, and he may also require the removal of any of such pipes in order that their interior may be examined and their actual thickness and condition ascertained, but they should be tested periodically, to not less than twice the maximum working pressure to which they are subject, as follows :—

Main Steam Pipes.

Note.—**Main steam pipes include the main range and its branches from the various boilers and those to the propelling machinery, also all steam pipes joining two or more boilers together.**

(*a*) All copper bars having brazed longitudinal seams, whether forming a complete range or only part of a range of pipes, should, with the exception of those referred to in paragraph (*c*), be examined and tested, with the lagging removed, at least once in about every four years.

(*b*) Iron, steel, or solid-drawn copper pipes should, when the diameter exceeds 3 inches, be stripped and tested by hydraulic pressure at least once in six years.

(*c*) In the case of pipes with a diameter of 3 inches or less the Surveyor may use his own discretion as to the removal of the lagging for more than a few inches near each flange when the hydraulic test is applied.

Auxiliary Steam Pipes having an Internal Diameter Exceeding 6 inches.

(*a*) Copper pipes having brazed longitudinal seams should be stripped and tested by hydraulic pressure at least once in four years.

(*b*) Iron, steel, or solid-drawn copper pipes should be stripped and tested by hydraulic pressure at least once in six years.

Auxiliary Steam Pipes having an Internal Diameter Exceeding 3 inches and not Exceeding 6 inches.

(*a*) Copper pipes having brazed longitudinal seams should be stripped for not less than 2 inches at each flange and tested by hydraulic pressure at least once in every four years.

(*b*) Iron, steel, or solid-drawn copper pipes should be stripped for not less than 2 inches at each flange and tested by hydralic pressure at least once in six years.

General.

In all cases where the pipes are not wholly stripped, the hydraulic test pressure should remain on the pipes for such time as the Surveyor considers necessary, but in no case for less than 20 consecutive minutes. Any length from which leakage is observed at other places than the flanges should be stripped, repaired, and re-tested.

The foregoing instructions apply to all steam pipes, the bursting of which would probably cause loss of life or serious injury, but it is not expected that the Surveyors will insist on the testing of small pipes, from which the free outflow of steam would cause no danger or inconvenience, and which would not easily burst in any circumstances.

When a vessel is surveyed for a passenger certificate after transference from a foreign flag, or for the first time, all the steam pipes should be tested as indicated in these instructions.

It has been brought to the notice of the Board that severe corrosion has, in a number of cases, been found on the outer surface of copper steam pipes which have been covered with non-conducting material composed chiefly of asbestos combined with a binding material, more particularly where the pipes have been wetted by sea water. The Surveyors should therefore pay special attention to the examination and condition of copper steam pipes which may be covered with asbestos and other porous lagging, especially when the pipes are exposed to the action of sea water, and, where they find any evidence of the existence of corrosion, they should take special steps to satisfy themselves that the pipes are of sufficient thickness and otherwise safe for the pressure at which they are worked. It is most desirable that copper pipes which would be liable to be occasionally wetted by sea water should be efficiently protected by a waterproof covering over the lagging, which should, preferably, be free from acid and otherwise non-corrosive.

Safety-Valves.

175. Examination of Safety-Valves.—The Surveyor, in his examination of the machinery and boilers, is particularly to direct his attention to the safety-valves, and, whenever he considers it necessary, he is to satisfy himself as to the pressure on the boiler by actual trial.

The Surveyor is to fix the limits of the weight to be placed on the safety-valves, and the responsibility of issuing a declaration before he is fully satisfied on the point is very grave. The law places on the Surveyor the responsibility of "declaring" that the boilers are in his judgment sufficient with the weights he states.

176. Surveyor to see Valves Weighted.—When the Surveyor has determined the working pressure, he is to see the safety-valves weighted accordingly, and the weights or springs fixed in such a manner as to preclude the possibility of them shifting or in any way increasing the pressure. The limit of the weight on the valves is to be inserted in the declaration, and should it at any time come to a Surveyor's knowledge that the weights have been shifted, or the loading of the valves has been otherwise altered, or that the valves have been in any way interfered with, so as to increase the pressure without the sanction of the Board of Trade, he is at once to report the facts to the Board.

If any person places an undue weight on the safety-valve of *any* steamship, or in the case of steamships surveyed under the Act, increases such weight beyond the limits fixed by the Engineer Surveyor, he shall, in addition to any other liability he may incur by so doing, be liable for each offence to a fine not exceeding one hundred pounds.

177. Provision as regards Safety-Valves.—Cases have come under the notice of the Board in which there were pipes between the boilers and the safety-valve chests. Such arrangement is not in accordance with the Act, which distinctly provides that the safety-valves shall be upon the boilers.

The Surveyors are instructed that in all *new boilers*, and in boilers in which *alterations can be easily made*, the valve chest should be placed directly on the boiler; and the neck, or part between the chest and the flange which is bolted on to the boiler, should be as short as possible and be cast in one with the chest.

The Surveyor should note that it is not intended by this instruction that vessels with old boilers which have been previously passed with such an arrangement should be detained for the alterations to be carried out.

Of course in any case in which the Surveyor is of opinion that it is positively dangerous to have a length of pipe between the boilers and the safety-valve chest, it is his duty at once to insist on the requisite alterations being made before granting a declaration.

179. Area of Safety-valves.—The locked-up valves, *i.e.*, those out of the control of the engineer when steam is up, should have an area not smaller, and a pressure not greater, than those which are not locked-up, if any such valves are fitted.

When natural draught is used, the area per square foot of fire-grate surface of the locked-up safety-valves should not be less than that given in the following table opposite the boiler pressure intended, but in no case should the valves be less than 2 inches in diameter. This applies to new vessels and to vessels which have not previously received a passenger certificate.

When, however, the valves are of the common description, and are made in accordance with the table, it will be necessary either to fit them with springs having great elasticity, or to provide other means to keep the accumulation within moderate limits.

Safety-Valve Areas
(Natural Draught)

Boiler Pressure	Area of Valve per Square Foot of Fire Grate	Boiler Pressure	Area of Valve per Square Foot of Fire Grate	Boiler Pressure	Area of Valve per Square Foot of Fire Grate
15	1·250	23	·986	31	·815
16	1·209	24	·961	32	·797
17	1·171	25	·937	33	·781
18	1·136	26	·914	34	·765
19	1·102	27	·892	35	·750
20	1·071	28	·872	36	·735
21	1·041	29	·852	37	·721
22	1·013	30	·833	38	·707

Safety-Valve Areas.—cont.

Boiler Pressure	Area of Valve per Square Foot of Fire Grate	Boiler Pressure	Area of Valve per Square Foot of Fire Grate	Boiler Pressure	Area of Valve per Square Foot of Fire Grate
39	·694	93	·347	147	·231
40	·681	94	·344	148	·230
41	·669	95	·340	149	·228
42	·657	96	·337	150	·227
43	·646	97	·334	151	·225
44	·635	98	·331	152	·224
45	·625	99	·328	153	·223
46	·614	100	·326	154	·221
47	·604	101	·323	155	·220
48	·595	102	·320	156	·219
49	·585	103	·317	157	·218
50	·576	104	·315	158	·216
51	·568	105	·312	159	·215
52	·559	106	·309	160	·214
53	·551	107	·307	161	·213
54	·543	108	·304	162	·211
55	·535	109	·302	163	·210
56	·528	110	·300	164	·209
57	·520	111	·297	165	·208
58	·513	112	·295	166	·207
59	·506	113	·292	167	·206
60	·500	114	·290	168	·204
61	·493	115	·288	169	·203
62	·487	116	·286	170	·202
63	·480	117	·284	171	·201
64	·474	118	·281	172	·200
65	·468	119	·279	173	·199
66	·462	120	·277	174	·198
67	·457	121	·275	175	·197
68	·451	122	·273	176	·196
69	·446	123	·271	177	·195
70	·441	124	·269	178	·194
71	·436	125	·267	179	·193
72	·431	126	·265	180	·192
73	·426	127	·264	181	·191
74	·421	128	·262	182	·190
75	·416	129	·260	183	·189
76	·412	130	·258	184	·188
77	·407	131	·256	185	·187
78	·403	132	·255	186	·186
79	·398	133	·253	187	·185
80	·394	134	·251	188	·184
81	·390	135	·250	189	·183
82	·386	136	·248	190	·182
83	·382	137	·246	191	·181
84	·378	138	·245	192	·181
85	·375	139	·243	193	·180
86	·371	140	·241	194	·179
87	·367	141	·240	195	·178
88	·364	142	·238	196	·177
89	·360	143	·237	197	·176
90	·357	144	·235	198	·176
91	·353	145	·234	199	·175
92	·350	146	·232	200	·174

When forced draught is used, the area of the safety-valves should not be less than that found by the following formula :—

$$\frac{A \times \left\{ \begin{array}{l}\text{estimated consumption of coal per square} \\ \quad\text{foot of grate in lbs. per hour}\end{array}\right.}{20} = \begin{array}{c}\text{area of valves} \\ \text{required;}\end{array}$$

where A = area of valves as found from the table.

When the pressure exceeds 180 lbs. per square inch, the accumulation of pressure at the steam test will probably be exceptionally high, unless the area of the branch leading from the valve chest is in excess of the area of the valves, and the area of the main waste steam pipe is correspondingly in excess of the gross area of the valves.

When ascertaining the fire-grate area, the length of the grate should be measured from the inner edge of the dead-plate to the front of the bridge, and the width from side to side of the furnace on the top of the bars at the middle of their length.

181. Spring-loaded Safety-Valves.—The Surveyor need raise no question as to the sufficiency of spring-loaded valves, if the results of the steam trial for accumulation of pressure are satisfactory, and if the following conditions are complied with :—

(1) That at least two valves are fitted to each boiler.
(2) That the valves are of the size prescribed by Section 179.
(3) That the springs and valves are so cased-in and locked-up that they cannot be tampered with.
(4) That provision is made to prevent the valves flying off in case of the springs breaking.
(5) That screw lifting-gear is provided to ease all the valves, as required by Section 180.
(6) That the size of the steel of which the springs are made is in accordance with that found by the following formula :—

$$\sqrt[3]{\frac{s \times D}{c}} = d;$$

where s = the load on the spring, in lbs. ;
D = the diameter of the spring (from centre to centre of wire), in inches ;
d = the diameter, or side of square, of the wire, in inches ;
c = 8,000 for round steel ; and
c = 11,000 for square steel.

(7) That the springs are protected from the steam and impurities issuing from the valves.
(8) That, when valves are loaded by direct springs, the compressing screws abut against metal stops or washers, when the loads sanctioned by the Surveyor are on the valves.
(9) That the springs have a sufficient number of coils to allow a compression under the working load of at least one quarter the diameter of the valve.

The number of coils required for a given compression, or the compression due to the load, is given, approximately, by the following formula :—

$$N = \frac{K \times C \times d^4}{s \times D^3}, \text{ or}$$

$$K = \frac{s \times D^3 \times N}{C \times d^4};$$

where N = number of free coils in spring;
K = compression, in inches;
d = diameter of steel, or side of square, *in sixteenths of an inch;*
C = 22 for round, and 30 for square steel; and
s and D have the same meanings as before.

The steel of safety-valve springs should not as a rule be less than $\frac{1}{4}$ inch in diameter or side of square.

182. Spring-loaded Valves to be Tested under Steam.—In no case is the Surveyor to give a declaration for spring-loaded valves, unless he has examined them and is acquainted with the details of their construction, and unless he has tried them under full steam, and full firing, for at least 15 minutes with the feed-water shut off and stop-valve closed, and is fully satisfied with the result of the test. In special cases, when the valves are of novel design or where forced draught is used, the results of the test under full steam should be reported on the form Surveys 21, but, if the Surveyor is fully satisfied with them, he need not delay the granting of the declaration for the vessel pending the approval of the Board of Trade. If, however, the accumulation of pressure exceeds 10 per cent. of the loaded pressure, he should withhold his declaration and report the case to the Board, submitting a sketch, if necessary, and stating the strength pressure and the working pressure of the boilers.

When witnessing safety-valve tests for accumulation of pressure, the Surveyors are to use the pressure gauges supplied by the Board of Trade for the purpose. No steam gauge should be used without having a syphon filled with water between it and the boiler, and care should be taken to see that the gauge pipe and syphon are clear before attaching the gauge.

183. Plans of New Designs or of Alterations in Details to be submitted.—In the case of safety-valves of which the principle and details have already been passed by the Board of Trade, the Surveyor need not require plans to be submitted so long as the details are unaltered, as to which he must fully satisfy himself; but, in any new arrangement of valves, or in any case in which any detail of approved valves is altered, he should, before assuming the responsibility of passing them, report particulars, with a drawing to scale, to the Board of Trade. He can make this drawing himself from the actual parts of the valves fitted, but, in order to save time and to faciliate the survey, the owners or makers may prefer to send in tracings of their own, before the valves are placed on the boiler. If they do this the survey can be more readily made, and delay and expense may be saved to owners, as the Surveyor will not then have to spend his time, and delay the ship, in preparing drawings and comparing them with the valves.

The tracings of new safety-valve designs should, where practicable, be transmitted to the Board of Trade for consideration before the construction of the valves is commenced.

The Surveyors should arrange with manufacturers to supply the designs of safety-valves which they intend to make. An easy method of facilitating this matter is for the manufacturer to leave in the local Surveyor's office an approved plan or plans of his valve or valves, and then afterwards to inform the Surveyors that the valves fitted are according to drawing A, B, or C, as the case may be. By this means, when once a design has been agreed upon, and is adhered to, all subsequent questions and delays will be prevented.

185. **Stop-Valves.**—No boiler or steam chamber should be so constructed, fitted, or arranged that the escape of steam from it through the safety-valves required by the Act of Parliament can be wholly, or partially, intercepted by the action of another valve.

Combined stop- and safety-valve chests are undesirable mountings, and in any case in which it is desired to adopt them, full particulars should be submitted to the Board for consideration.

A stop-valve should always be fitted between the boiler and the steam pipe, and, where two or more boilers are connected with a steam receiver or superheater, between each boiler and the superheater or steam receiver.

The necks of stop-valve chests and other boiler mountings should be as short as practicable; and the chests should be tested when new to double the working pressure.

186. **Water-Gauges, Test Cocks, Steam-Gauges, etc.**—Each boiler should be fitted with a glass water-gauge, at least three test cocks, and a steam-gauge. Boilers that are fired from both ends, and those of unusual width, should have a glass water-gauge and three test cocks at each end or side, as the case may be. An additional glass water-gauge may, however, be substituted for three test cocks. When a steamship has more than one boiler, each boiler should be treated as a separate one, and have all the requisite fittings.

When the water-gauge cocks are not attached directly to the shell of the boiler, but to a stand-pipe or column, cocks should, as a general rule, be fitted between the boiler and the stand-pipes, etc., and may be placed either on the boiler or at the stand-pipe. Such cocks need not, however, be insisted on in cases where the columns, stand-pipes, etc., are of moderate length and of suitable strength, provided that the diameter of the bore at any part is not less than 3 inches. The Surveyors should not pass valves between the boiler and the stand-pipe.

If the column, stand-pipes, etc., are of less diameter than 3 inches, and the pipes are bolted to the boiler without the intervention of cocks, the arrangement need not be objected to, if otherwise satisfactory, provided that there is no difficulty in keeping the passages at the ends clear and in ascertaining that they are so. To do this it will be necessary that the passage in the part of the column between the top and bottom gauge-glass cocks be cut off or closed, which may be done permanently, or by the interposition of a cock at that part. The latter is a convenient and desirable arrangement even when cocks are fitted on the boiler.

In the case of high pressure boilers, it is desirable that the cocks in connection with the water-gauges should be fitted with handles which can be expeditiously manipulated from a convenient position.

It is desirable in all cases that test cocks should be fitted directly to the skin of the boiler, and when the water-gauge is attached to a column, the opening through which is stopped or can be cut off, the test cocks *must* be fitted directly to the skin of the boiler.

The Surveyors should satisfy themselves by actual examination whether the glass water-gauges of the boilers of the vessels they survey are clear, and also whether they are fitted with automatic valves or fittings, as the existence of such fittings cannot always be ascertained by external examination. In all cases where automatic gauges are fitted, full particulars thereof should be submitted for consideration and approval before the gauges are passed.

189. **Circulating Pipes.**—When the pipes are so arranged that the water in the boilers can be circulated by means of the donkey pump, similar precautions should be observed; and a cock fitted with a spanner and guard, the handle of which will stand above the level of the platform, should be fitted to the circulating pipes, preferably near the pump.

190. **Non-return Valves to Pipes.**—A non-return valve having a screw spindle not attached, by which the valve may be set down on its seat when necessary, should be fitted to all pipes which are so led or placed that water could, unless such non-return valves were fitted, run from the boiler or the sea into the bilge, either by accidentally or intentionally leaving a cock or valve open; the only exception to this requirement has reference to the fireman's ash cock, which must have a cock or valve on the ship's side and be above the stoke-hold plates.

191. **Cast-Iron Stand-Pipes, Cocks, etc.**—Cast-iron stand-pipes or cocks intended for the passage through them of hot brine should not be passed.

Surveyors should also discourage the use of cast-iron chocks and saddles for boilers, and particular attention should be paid to the chocking of boilers, more especially when they are fired athwartships.

192. **Exhaust Pipe of Donkey Engine not led through Ship's Side.**—The exhaust pipe for the donkey engine should not be led through the ship's side; it should be led on deck or into the main waste steam pipe, and in all cases it should have a drain-cock on it.

SUMMARY

The preceding rules are summarised in the following short table, in which the method has been carried out of indicating all such dimensions as are measured in inches by capitals, such as are measured in sixteenths of an inch by small letters, and all coefficients, &c., by black letters.

C and **C′** = coefficients.
W P = permissible working pressure.
B and **B′** = percentage of joint, respectively of plate and of rivets.
N = number of rivets included within one pitch of external row.
F = factor of safety.
T and t = thickness of plate, measured respectively in inches and in sixteenths.
P and P_c = pitches of rivets, or of stays, or of stay tubes.
D = internal diameter of shells, or of plain tubes; effective diameters of rivets or of stays, or external diameter of furnaces in inches, measured at the plain cylindrical part or at the bottom of the corrugations.
L′ and L″ = length of plain cylindrical parts of furnaces, depth of combination chamber, or length of girders, measured respectively in feet and in inches.
H = depth of girders measured in inches.
A = sectional area of stays or of rivets in square inches.

Percentage of plate: $\mathbf{B} = (P - D) : 100\,P$.

Percentage of rivets:—

Double Butt Straps: $\mathbf{B}' = 187{\cdot}5\,\mathbf{N}\,.\,\mathbf{A} \div P\,.\,T = 147\,\mathbf{N}\,.\,D^2 \div P\,.\,T.$

Lap Joints: $\mathbf{B}' = 100\,\mathbf{NA} \div PT = 78{\cdot}5\,\mathbf{N}\,.\,D^2 \div P\,.\,T.$

The coefficients for determining the working pressure of the shell depend on the material and on conditions of workmanship, the ultimate strength of the material being divided by the sum of 4·5 and one or both of the following additions :—

Table of Additions to the Factors of Safety

Type of Joint	Longitudinal Seams				Circumferential Seams				
	Double Butt Straps	Lap Joint			Double Butt Straps		Lap Joint		
Riveting	Any Sort	Single	Double	Treble	Single[1]	Double	Single[1]	Double[1]	Treble
Holes drilled before bending .	0·3	1·3	0·6	0·4	0·25	0·15	0·35	0·25	0·15
,, ,, after ,, .	0·15	1·15	0·45	0·25	0·2	0·10	0·3	0·2	0·1
,, ,, in place ,, .	0·0	1·0	0·3	0·1	0·1	0·0	0·2	0·1	0·0

For good but not tested iron add ·5.
If there are no circumferential seams add ·4.
[1] In these three cases ·3 has to be added for double-ended boilers, but only for the central seam.

$$\textbf{Iron Boiler Shells.} \quad \mathbf{WP} = \frac{\mathbf{C}}{\mathbf{F}} \cdot \frac{(\mathbf{B} \text{ or } \mathbf{B}')}{50} \cdot \frac{\mathrm{T}}{\mathrm{D}}.$$

The value of **C** is

40,000 for wrought iron (across the grain), not tested.
47,000 ,, ,, ,, (with the grain) ,,

If tested, **C** is the lowest ultimate tensile strength in lbs. per sq. inch. For superheaters use 30,000, or if in contact with flame use 22,400.

Steel Boiler Shells.—

$$\mathbf{WP} = 44{\cdot}8\,\mathbf{S}\,.\,\mathrm{T}\,.\,\mathbf{B} \div \mathrm{F}\,.\,\mathrm{D}, \text{ or } = 229\ \mathrm{T}\,.\,\mathbf{B}' \div \mathrm{D}.$$

S in tons per sq. inch is the minimum ascertained tensile strength of the shell plates.

FURNACES. (See appended Tables.)

Plain Furnaces. $$\text{WP} = \frac{C\,.\,T^2\,.}{D\,.\,(L'+1)}, \text{ or } \frac{C'\,.\,t^2}{D\,.\,(L'+1)}.$$

TABLE OF VALUES OF C AND C′

Material of Furnace	C			C′		
	Iron		Steel	Iron		Steel
Longitudinal Seams of Furnace	Drilled	Punched	Drilled	Drilled	Punch'd	Drilled
Single butt straps double-riveted, or double butt straps single-riveted .	90,000	85,000	99,000	351·6	332·0	386·7
Single butt straps single-riveted, or lap-bevelled and double-riveted . . .	80,000	75,000	88,000	312·5	293·0	343·7
Lap not bevelled, double-riveted	75,000	70,000	82,500	293·0	273·4	322·2
Lap bevelled, single-riveted .	70,000	65,000	77,000	273·4	255·9	300·8
Lap not bevelled, single-riveted	65,000	60,000	71,500	253·9	234·4	279·3
Longitudinal seams welded .	90,000		99,000	351·6		386·7

In no case should the working pressure exceed the values found by the following formulæ :—

$$\text{WP} = \frac{9{,}000\ T}{D} = \frac{555\ t}{D} \text{ for iron.}$$

$$\text{WP} = \frac{9{,}900\ T}{D} = \frac{617\ t}{D} \text{ for steel.}$$

For short lengths of flanged furnaces use the following formulæ :—

$$\left.\begin{aligned} \text{WP} &= \frac{9{,}900}{D}\,.\left(\frac{5}{3}\,.\,T - \frac{L''}{180} - \frac{1}{15}\right) \\ &= \frac{619}{D}\,.\left(\frac{5}{3}\,.\,t - L''\,. - 1{\cdot}067\right) \end{aligned}\right\} \text{ for steel plates.}$$

These formulæ can only be used as long as $L'' \leqq 120\ T - 12$.

The working pressures for various patent furnaces are found by the following formula :—

Fox's corrugated steel furnaces
Purves's ribbed „ „ } $\text{WP} = 14{,}000\,.\,\frac{T}{D} = 875\,.\,\frac{t}{D}$.

Leeds Forge Bulb „ „ $\text{WP} = 15{,}000\,.\,\frac{T}{D} = 937{\cdot}5\ \frac{t}{D}$.

FLAT PLATE. (See appended Tables.)

$$W\,P = \frac{C\,.\,(t+1)^2}{(P\,.\,P_c - 6)}.$$

If the pitches P and P_c are equal, then

$$P = \sqrt{6 + \frac{C\,.\,(t+1)^2}{W\,P}}.$$

For unequal pitches (see T. W. Trail, 1890, pp. 67, 199) apparently

$$W\,P = \frac{2\,C\,.\,(t+1)^2}{p^2 - 12},$$

where p = diagonal pitch or diameter of largest inscribed circle

TABLE OF VALUES OF C.

Material of Plate	Iron				Steel			
Position of Stay and its Mode of Attachment	Front Tube Plate	Combust. Chamber	In Up take	External Plate	Front Tube Plate	Combust. Chamber	In Up-take	External Plate
Stay ends riveted or expanded .	70	60	36	70	77	66	39·6	77
Stay screwed into plate and nutted	90	80	54	...	112·5	100	67·5	...
Stay double nutted	...	...	.	120	...	...	...	150
,, ,, washer $3\,D \times \frac{2}{3}\,T$	...	...	60	132	...	...	75	165
,, ,, ,, $\frac{2}{3}\,P \times T$.	...	...	...	168	...	...	...	210
,, ,, bar $\frac{2}{3}\,P \times T$.	...	...	...	192	...	...	...	240

INNER TUBE PLATES.

$$W\,P = \frac{C\,.\,(P - D)\,T}{L''\,.\,P} = \frac{C'\,.\,(P - D)\,t}{L''\,.\,P}.$$

TABLE OF VALUES OF C AND C′.

Material	Iron	Steel
C	22,000	28,000
C′	1,375	1,750

STAYS. (See appended Tables.)

$$WP = C\frac{A}{P.P_e}, \text{ or } C'\frac{D^2}{P.P_e}.$$

TABLE OF VALUES OF C AND C′.

Material of Stays	C	C′
Welded or heated iron stays	5,000	3,925
Iron stays	7,000	5,498
Steel "	9,000	7,069
Iron stay tubes if ¼ inch thick net	6,000	...
Steel " " if ¼ inch thick net	7,500	...

GIRDERS. $$WP = \frac{C \,.\, H^2 T}{L'' . (L'' - P) . P_e}.$$

TABLE OF VALUES OF C.

Number of Stays in Girders	Material of Girders	
	Iron	Steel
One stay	6,000	6,600
Two or three stays	9,000	9,900
Four or five "	10,000	11,000

TABLES.

When used in the drawing office, it is strongly recommended that those parts of the following tables which are inapplicable to the particular works should be obliterated.

TABLE FOR FINDING THE DIAMETERS OF RIVETS AND OF PITCH IN RIVETED JOINTS.

A short table will be found on p. 316 which shows the smallest permissible percentage for any particular joint; the conditions are fully explained and illustrated.

The following table contains the values of $\mathbf{N} \,.\, D \div T$ = number of rivets × diameter of rivets ÷ thickness of steel plate. Having found the number in the table corresponding to the desired percentage, it has to be divided by the number of rivets and multiplied by the thickness of the plate. The result is the diameter of the rivet hole. This has to be multiplied by the corresponding value in the first column of this table, viz. pitch ÷ rivet diameter, and the product is the pitch of the rivet. For examples see pp. 322, 351.

This is only true if the factor of safety is 4·5. For any other factor $\mathbf{F}$ the value $\mathbf{N} \,.\, D \div T$ has to be multiplied by $\frac{4{\cdot}5}{\mathbf{F}}$.

The pitch is limited by the formula $P = 1\frac{5}{8} + \mathbf{C} \times T$, and may not exceed 10″. (See p. 374.)

MINIMUM VALUES OF $N \cdot D \div T = \dfrac{\text{Number} \times \text{Diameter of Rivets}}{\text{Thickness of Plate}}$.

Pitch ÷ Rivet Diameter	Percentage of Plate	LAP JOINT			BUTT STRAPS		
		Iron Plates Iron Rivets	Steel Plates Steel Rivets	Steel Plates Iron Rivets	Iron Plates Iron Rivets	Steel Plates Steel Rivets	Steel Plates Iron Rivets
		Constants					
		100	$\frac{23}{28}\cdot100$	$\frac{8}{13}\cdot100$	$\frac{7}{4}\cdot100$	$\frac{7}{4}\cdot\frac{23}{28}\cdot100$	$\frac{7}{4}\cdot\frac{8}{13}\cdot100$
2·50	60	...	2·32	3·10	...	...	...
2·56	61	...	2·42	3·24	...	...	...
2·63	62	2·08	2·53	3·38	...	...	...
2·70	63	2·17	2·64	3·52	...	...	2·01
2·78	64	2·26	2·76	3·68	...	...	2·10
2·86	65	2·36	2·88	3·84	...	...	2·19
2·94	66	2·47	3·01	4·02	...	...	2·29
3·03	67	2·59	3·15	4·20	...	...	2·40
3·12	68	2·71	3·29	4·40	...	...	2·51
3·22	69	2·83	3·45	4·61	...	...	2·63
3·33	70	2·97	3·62	4·83	...	2·07	2·76
3·45	71	3·12	3·79	5·07	...	2·17	2·89
3·57	72	3·27	3·99	5·32	...	2·28	3·04
3·70	73	3·44	4·19	5·59	...	2·39	3·20
3·85	74	3·62	4·41	5·89	2·07	2·52	3·37
4·00	75	3·82	4·65	6·21	2·18	2·66	3·55
4·08	75½	3·92	4·78	6·38	2·24	2·73	3·64
4·17	76	4·03	4·91	6·55	2·30	2·80	3·74
4·26	76½	4·14	5·05	6·74	2·37	2·88	3·85
4·35	77	4·26	5·19	6·93	2·44	2·97	3·96
4·45	77½	4·38	5·34	7·13	2·51	3·05	4·07
4·55	78	4·51	5·49	7·34	2·58	3·14	4·19
4·65	78½	4·65	5·66	7·55	2·66	3·23	4·32
4·76	79	4·79	5·83	7·74	2·74	3·33	4·45
4·88	79½	4·94	6·01	8·02	2·82	3·43	4·58
5·00	80	5·09	6·20	8·28	2·91	3·54	4·73
5·13	80½	5·26	6·40	8·54	3·00	3·66	4·88
5·26	81	5·43	6·61	8·82	3·10	3·78	5·04
5·40	81½	5·61	6·83	9·11	3·21	3·90	5·21
5·55	82	5·80	7·06	9·43	3·32	4·03	5·39
5·71	82½	6·00	7·31	9·75	3·43	4·17	5·57
5·88	83	6·22	7·57	10·10	3·55	4·32	5·77
6·06	83½	6·44	7·84	...	3·68	4·48	5·98
6·25	84	6·68	8·14	...	3·82	4·65	6·21
6·45	84½	6·94	8·45	...	3·97	4·83	6·45
6·67	85	7·22	8·78	...	4·12	5·02	6·70
6·82	85⅓	7·41	9·02	...	4·23	5·15	6·88
6·98	85⅔	7·61	9·26	...	4·35	5·29	7·07
7·14	86	7·82	9·52	...	4·47	5·44	7·26
7·32	86⅓	8·04	9·79	...	4·60	5·60	7·47
7·50	86⅔	8·28	10·07	...	4·73	5·76	7·68
7·69	87	8·52	...	...	4·87	5·93	7·91
7·89	87⅓	8·78	...	...	5·02	6·11	8·15
8·10	87⅔	9·05	...	...	5·17	6·30	8·40
8·33	88	9·34	...	...	5·34	6·50	8·67
8·57	88⅓	9·64	...	...	5·51	6·71	8·95
8·82	88⅔	9·96	...	...	5·69	6·93	9·25
9·09	89	10·30	...	...	5·89	7·17	9·57
9·37	89⅓	...	...	...	6·09	7·42	9·90
9·68	89⅔	...	...	...	6·31	7·69	10·26
10·00	90	...	...	...	6·55	7·97	...

The following tables contain the **internal** diameters of furnaces, calculated according to the Board of Trade Rules :—

INTERNAL DIAMETERS OF CIRCULAR FURNACES IN INCHES.

TYPE	PLAIN FLUES (Iron)						PLAIN FLUES (Steel)						ADAMSON'S RINGS			FOX & PURV.
Formula	$90{,}000 \cdot \frac{T^2}{D.(L'+1)}$					$9{,}000 \cdot \frac{T}{D}$	$99{,}000 \cdot \frac{T^2}{D.(L'+1)}$					$9{,}900 \cdot \frac{T}{D}$	$\frac{619}{D} \cdot (\frac{1}{2}l - L''\cdot089 - 1\cdot07)$			$14{,}000 \cdot \frac{T}{D}$
L. or C.	9 Ft.	8 Ft.	7 Ft.	6 Ft.	5 Ft.		9 Ft.	8 Ft.	7 Ft.	6 Ft.	5 Ft.		36 In.	24 In.	18 In.	
Plate Thickness. Inches																
60 lbs. Working Pressure																
$\frac{3}{8}$	...	...	25·6	29·4	34·4	55·5	...	...	28·2	32·4	37·9	61·0	59·1	68·2	73·7	86·7
$\frac{13}{32}$	...	26·7	30·1	34·5	40·4	...	26·4	29·4	33·2	38·1	44·6	...	...	...	...	...
$\frac{7}{16}$	27·8	31·0	35·0	40·1	47·0	...	30·7	34·2	38·6	44·2	51·7	...	...	...	...	...
$\frac{15}{32}$	32·0	35·7	40·2	46·1	54·0	...	35·3	39·3	44·4	50·8	...	...	...	...	...	...
$\frac{1}{2}$	36·5	40·6	45·9	52·5	...	...	40·2	44·8	50·6	...	...	...	...	...	..	...
$\frac{17}{32}$	41·3	46·0	51·8	...	...	...	45·5	50·7	...	...	...	...	...	...	...	...
$\frac{9}{16}$	46·3	51·6	...	...	...	...	51·1	...	...	...	...	...	...	...	...	...
$\frac{19}{32}$	51·7	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
$\frac{5}{8}$	...	...	...	...	...	...	...	...	...	...	...	...	...	...	..	...
80 lbs. Working Pressure																
$\frac{3}{8}$	...	...	...	...	25·6	41·5	...	...	...	...	28·2	45·7	44·1	51·0	55·1	64·9
$\frac{13}{32}$	...	...	..	25·7	30·1	44·9	...	...	...	28·4	33·2	49·5	50·4	...	...	...
$\frac{7}{16}$	...	...	26·0	29·9	35·0	48·4	...	...	28·7	32·9	38·6	53·3	...	...	...	...
$\frac{15}{32}$	...	26·4	30·0	34·4	40·3	51·8	26·1	29·1	33·0	37·9	44·4	...	...	...	...	...
$\frac{1}{2}$	27·1	30·2	34·2	39·2	45·9	...	29·9	33·4	37·7	43·2	50·6	...	...	...	...	...
$\frac{17}{32}$	30·7	34·2	38·6	44·3	51·9	...	33·8	37·7	42·6	48·8	...	...	...	...	...	...
$\frac{9}{16}$	34·6	38·4	43·4	49·7	...	...	38·0	42·4	47·8	54·8	...	...	...	...	...	...
$\frac{19}{32}$	38·5	42·9	48·4	55·5	...	...	42·4	47·3	53·3	...	...	...	...	...	...	...
$\frac{5}{8}$	42·7	47·5	53·7	...	...	...	47·1	50·9	...	...	...	..	...	...	...	...
100 lbs. Working Pressure																
$\frac{3}{8}$	...	...	...	...	...	33·0	...	...	...	...	...	36·4	35·1	40·6	43·9	51·7
$\frac{13}{32}$	...	...	...	...	...	35·8	...	...	...	...	26·3	39·4	40·1	45·6	48·9	...
$\frac{7}{16}$	...	...	...	...	27·8	38·5	...	...	...	26·2	30·7	42·4	45·1	50·6	53·9	...
$\frac{15}{32}$	...	...	...	27·3	32·0	41·3	...	...	26·2	30·1	35·5	45·4	50·2	...	...	...
$\frac{1}{2}$	...	24·0	27·1	31·1	36·5	44·0	...	26·5	29·9	34·3	40·2	48·5	...	...	...	...
$\frac{17}{32}$	24·2	27·2	30·7	35·2	41·3	46·7	26·8	30·0	33·8	38·8	45·5	51·5	...	...	...	...
$\frac{9}{16}$	27·4	30·5	34·5	39·6	46·3	49·5	30·2	33·7	38·0	43·6	51·1	...	...	...	...	...
$\frac{19}{32}$	30·5	34·1	38·4	44·1	51·7	52·3	33·7	37·6	42·4	48·7	...	...	...	...	...	...
$\frac{5}{8}$	33·9	36·7	42·7	49·7	...	...	38·4	41·5	47·1	54·5	...	...	...	...	...	...
120 lbs. Working Pressure																
$\frac{3}{8}$	...	...	...	...	...	...	...	...	...	...	...	30·2	29·0	33·6	36·3	43·0
$\frac{13}{32}$	...	...	...	...	...	29·7	...	...	...	...	...	32·7	33·2	37·8	40·5	46·6
$\frac{7}{16}$	...	...	...	...	...	31·9	...	...	...	...	...	35·2	37·4	42·0	44·7	50·2
$\frac{15}{32}$	...	...	...	...	26·5	34·2	...	...	...	...	29·2	37·7	41·6	46·2	48·9	...
$\frac{1}{2}$	...	...	...	25·8	30·2	36·5	...	...	...	28·5	33·4	40·2	45·8	50·3	53·1	...
$\frac{17}{32}$	...	...	25·4	29·2	34·2	38·8	...	...	28·0	32·2	37·7	42·7	50·0	...	...	...
$\frac{9}{16}$	...	25·2	28·5	32·8	38·3	41·1	...	27·9	31·5	36·1	42·3	45·3	...	...	...	...
$\frac{19}{32}$	25·2	28·2	31·9	36·6	42·9	43·3	27·9	31·1	35·2	42·0	47·3	47·8	...	...	...	...
$\frac{5}{8}$	28·0	31·4	35·4	41·1	47·6	45·6	31·0	34·5	39·0	45·3	52·5	50·3	...	...	...	...

Internal Diameters of Circular Furnaces in Inches—*cont.*

TYPE	PLAIN FLUES (Iron)						PLAIN FLUES (Steel)						ADAMSON'S RINGS			FOX & PURV.
Formula	$90{,}000 . \frac{T^2}{D.(L'+1)}$					$\frac{T}{D}$	$99{,}000 . \frac{T^2}{D.(L'+1)}$					$\frac{T}{D}$	$\frac{619}{D}$ ([illegible]t $- L''\cdot089 - 1\cdot07)$			$C . \frac{T}{D}$
L. or C.	9 Ft.	8 Ft.	7 Ft.	6 Ft.	5 Ft.	9,000	9 Ft.	8 Ft.	7 Ft.	6 Ft.	5 Ft.	9,900	36 In.	24 In.	18 In.	14,000
Plate Thickness. Inches.																
140 lbs. Working Pressure																
$\frac{3}{8}$	...	...	...	...	...	...	...	...	...	...	...	...	...	28·9	31·2	36·7
$\frac{13}{32}$	...	...	...	...	...	...	...	...	...	...	...	27·9	28·4	32·4	34·7	39·8
$\frac{7}{16}$	...	...	...	...	...	...	...	...	...	...	...	30·0	32·0	35·9	38·2	42·9
$\frac{15}{32}$	...	...	...	...	...	29·2	...	...	...	...	...	32·2	35·5	39·5	41·8	45·9
$\frac{1}{2}$	...	...	...	...	...	31·1	...	...	...	...	28·5	34·3	39·1	43·1	45·4	49·0
$\frac{17}{32}$	...	...	...	...	29·2	33·1	...	...	...	27·4	32·2	36·5	42·7	46·6	48·9	52·1
$\frac{9}{16}$	...	...	24·3	27·9	32·8	35·1	...	...	26·8	30·8	36·1	38·7	46·2	50·1	52·5	...
$\frac{19}{32}$	...	...	27·1	31·2	36·7	37·0	...	...	30·0	34·4	40·4	40·8	49·8	...	...	...
$\frac{5}{8}$	...	25·8	30·1	35·1	39·0	39·0	26·4	28·5	33·3	38·7	44·8	42·9	53·4	...	...	...
160 lbs. Working Pressure																
$\frac{3}{8}$	...	...	...	...	...	...	...	...	...	...	...	...	...	...	27·2	32·1
$\frac{13}{32}$	...	...	...	...	...	...	...	...	...	...	...	...	...	28·2	30·2	34·7
$\frac{7}{16}$	...	...	...	...	...	...	...	...	...	...	...	...	27·9	31·3	33·3	37·4
$\frac{15}{32}$	...	...	...	...	...	...	...	...	...	...	...	...	31·0	34·4	36·4	40·1
$\frac{1}{2}$	...	...	...	...	...	...	...	...	...	...	...	29·9	34·1	37·5	39·6	42·7
$\frac{17}{32}$	...	...	...	...	25·5	28·8	...	...	...	...	28·0	31·9	37·2	40·6	42·7	45·4
$\frac{9}{16}$	...	...	...	...	28·6	30·5		...	...	26·8	31·5	33·7	40·3	43·7	45·8	48·1
$\frac{19}{32}$	...	...	...	27·1	31·9	32·2	...	...	25·1	30·0	35·2	35·5	43·4	46·8	49·9	50·8
$\frac{5}{8}$	...	...	26·2	30·6	33·9	33·9	24·3	25·5	28·9	33·7	39·0	37·4	46·5	49·9	52·0	...
180 lbs. Working Pressure																
$\frac{3}{8}$	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	28·4
$\frac{13}{32}$	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	30·8
$\frac{7}{16}$	...	...	...	...	...	...	...	...	...	...	...	...	...	27·7	29·6	33·1
$\frac{15}{32}$	...	...	...	...	...	...	...	...	...	...	...	...	27·4	30·5	32·3	35·5
$\frac{1}{2}$	...	...	...	...	...	...	...	...	...	...	...	...	30·2	33·3	35·1	37·9
$\frac{17}{32}$	...	...	...	...	...	...	...	...	...	...	...	...	33·0	36·0	37·8	40·2
$\frac{9}{16}$	...	...	...	...	...	...	...	...	...	...	27·9	29·8	35·7	38·8	40·6	42·6
$\frac{19}{32}$	...	...	...	...	28·2	28·5	...	...	...	27·5	31·1	31·5	38·5	41·5	43·4	45·0
$\frac{5}{8}$	...	...	...	27·0	30·0	30·0	...	...	25·6	29·8	34·6	33·1	41·2	44·3	46·1	47·4
200 lbs. Working Pressure																
$\frac{3}{8}$	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	25·5
$\frac{13}{32}$	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	27·6
$\frac{7}{16}$	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	29·7
$\frac{15}{32}$	...	...	...	...	...	...	...	...	...	...	...	...	...	...	29·0	31·9
$\frac{1}{2}$	...	...	...	...	...	...	...	...	...	...	...	...	...	29·9	31·5	34·0
$\frac{17}{32}$	...	...	...	...	...	...	...	...	...	...	...	...	29·5	32·3	34·0	36·1
$\frac{9}{16}$	...	...	...	...	...	...	...	...	...	...	...	...	32·0	34·8	36·5	38·2
$\frac{19}{32}$	...	...	...	...	...	...	...	...	...	...	27·9	...	34·5	37·3	38·9	40·4
$\frac{5}{8}$	...	...	...	...	27·0	27·0	...	...	...	26·7	31·0	29·7	36·9	39·8	41·4	42·5

Permissible Working Loads of Untested Iron Stays with a Stress of 7,000 lbs. per Square Inch. For Diameter and Sections, see pp. 327, 328.

Outside Diameters	Working Loads for Plus Threads	Whitworth Threads: Number of Threads	Whitworth Threads: Working Loads	Number of Threads per Inch[1]: 6	7	8	9	10	11	12
			Working Loads of Screwed Stays at 7,000 lbs. per Square Inch							
Inches	Lbs.		Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
¾	3,093	10	2,127	...	...	...	...	2,127	2,208	2,275
⅞	4,209	9	2,952	...	...	...	2,952	3,068	3,165	3,245
1	5,498	8	3,879	...	...	3,879	4,045	4,180	4,293	4,387
1⅛	6,958	7	4,880	...	4,880	5,120	5,310	5,465	5,594	5,701
1¼	8,590	7	6,262	...	6,262	6,532	6,747	6,921	7,065	7,186
1⅜	10,394	6	7,419	7,419	7,813	8,116	8,355	8,549	8,710	8,844
1½	12,370	6	9,102	9,102	9,539	9,871	10,136	10,349	10,526	10,673
1⅝	14,517	5	10,303	10,956	11,435	11,800	12,086	12,321	12,514	12,674
1¾	16,837	5	12,271	12,983	13,503	13,899	14,212	14,464	14,673	14,847
1⅞	19,330	4½	13,910	15,181	15,743	16,170	16,508	16,779	17,004	17,191
2	21,991	4½	16,182	17,550	18,154	18,614	18,976	19,266	19,508	19,707
2⅛	24,826	4½	18,623	20,092	20,738	21,228	21,616	21,925	22,182	22,396
2¼	27,833	4	20,479	22,806	23,493	24,016	24,426	24,756	25,028	25,255
2⅜	31,011	4	23,217	25,698	26,421	26,973	27,408	27,758	28,048	28,287
2½	34,362	4	26,127	28,748	29,520	30,103	30,564	30,932	31,238	31,491
2⅝	37,884	4	29,210	31,976	32,791	33,406	33,888	34,279	34,601	34,866
2¾	41,578	4	32,464	35,377	36,233	36,880	37,361	37,796	38,135	38,413
2⅞	45,443	3½	34,616	38,950	39,848	40,526	41,059	41,487	41,840	42,131
3	49,480	3½	38,149	42,694	43,634	44,344	44,900	45,347	45,718	46,022
3⅛	53,690	...	...	46,611	47,592	48,334	48,914	49,380	49,768	50,085
3¼	58,071	3¼	44,844	50,697	51,719	52,494	53,100	53,586	53,990	54,321
3⅜	62,624	...	...	54,958	56,023	56,826	57,457	57,963	58,481	58,726
3½	67,347	3¼	53,038	59,389	60,496	61,332	61,986	62,513	62,946	63,304
3⅝	72,249	...	...	63,993	65,143	66,006	66,688	67,234	67,684	68,060
3¾	77,315	3	60,722	68,768	69,960	70,854	71,560	72,125	72,592	72,986
3⅞	82,556	...	...	73,713	74,950	75,874	76,605	77,189	77,672	78,069
4	87,969	3	70,192	78,834	80,111	81,067	81,823	82,425	82,924	83,340

[1] Nuts to be 1¼ diameters deep.

PERMISSIBLE WORKING LOADS OF Steel STAYS AND OF IRON STAYS OF 21·5 TONS TENACITY WITH A STRESS OF 9,000 LBS. PER SQUARE INCH. FOR DIAMETERS AND SECTIONS, SEE PP. 327, 328.

Outside Diameters	Working Loads for Plus Threads	Whitworth Threads		Number of Threads per Inch [1]						
				6	7	8	9	10	11	12
		Number of Threads	Working Loads of Screwed Stays at 9,000 lbs. per Square Inch							
Inches	Lbs.		Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
3/4	3,976	10	2,740	...	...	...	...	2,734	2,839	2,925
7/8	5,412	9	3,795	...	...	...	3,795	3,944	4,069	4,172
1	7,069	8	4,988	...	...	4,988	5,201	5,375	5,520	5,641
1 1/8	8,946	7	6,275	...	6,275	6,581	6,827	7,026	7,192	7,330
1 1/4	11,045	7	8,051	...	8,051	8,398	8,675	8,898	9,085	9,240
1 3/8	13,365	6	9,539	9,539	10,047	10,435	10,742	10,992	11,199	11,371
1 1/2	15,904	6	11,703	11,703	12,264	12,692	13,032	13,306	13,533	13,723
1 5/8	18,662	5	13,247	14,087	14,702	15,171	15,541	15,841	16,089	16,295
1 3/4	21,644	5	15,777	16,692	17,361	17,870	18,273	18,596	18,866	19,089
1 7/8	24,845	4 1/2	17,884	19,518	20,241	20,790	21,225	21,573	21,863	22,103
2	28,275	4 1/2	20,805	22,568	23,341	23,932	24,397	24,772	25,081	25,338
2 1/8	31,916	4 1/2	23,942	25,833	26,663	27,293	27,790	28,190	28,520	28,795
2 1/4	35,782	4	26,330	29,322	30,206	30,874	31,405	31,829	32,180	32,471
2 3/8	39,871	4	29,850	33,032	33,970	34,680	35,239	35,689	36,006	36,369
2 1/2	44,177	4	33,590	36,962	37,955	38,704	39,295	39,770	40,163	40,488
2 5/8	48,707	4	37,560	41,112	42,160	42,951	43,573	44,073	44,487	44,828
2 3/4	53,457	4	41,739	45,485	46,586	47,417	48,067	48,595	49,031	49,389
2 7/8	58,427	3 1/2	44,506	50,079	51,233	52,105	52,780	53,340	53,794	54,169
3	63,617	3 1/2	49,049	54,893	56,101	57,003	57,729	58,303	58,781	59,171
3 1/8	69,029	...	53,801	59,928	61,190	62,143	62,889	63,490	63,987	64,395
3 1/4	74,660	3 1/4	57,657	65,182	66,498	67,492	68,271	68,896	69,415	69,842
3 3/8	80,516	...	62,814	70,660	72,030	73,062	73,874	74,524	75,061	75,505
3 1/2	85,689	3 1/4	68,192	76,353	77,791	78,855	79,700	80,374	80,930	81,391
3 5/8	92,883	...	73,793	82,276	83,755	84,865	85,742	86,444	87,023	87,493
3 3/4	99,398	3	78,171	88,416	89,948	91,098	92,006	92,732	93,333	93,824
3 7/8	106,134	...	84,056	94,774	96,364	97,550	98,490	99,243	99,864	100,374
4	113,090	3	90,260	101,354	102,997	104,229	105,201	105,975	106,614	107,145

[1] Nuts to be 1 1/4 diameters deep.

INDEX

Abbreviations: B. T. = Board of Trade. L. R. = Lloyd's Register.

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