THE DESIGN OF A WATER TUBE BOILER.

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In the following paper are given the calculations, etc., on which is based the design of a water tube boiler, to illustrate to undergraduate members, the methods employed in designing to set conditions.

Description of Conditions.

The boiler is to supply steam to a set of compound inverted surface condensing engines, cylinders, 8 in. and 16 in. in diameter, by 9 in. stroke; designed to give 200 i.h.p. at 500 revolutions per minute, with a steam pressure of 185 lbs. per square inch.

The machinery is to drive a boat—56 ft. in length, 9 ft. 9 in. beam, 4 ft. 7 in. moulded depth, displacement 17½ tons, on a draught of 3 ft., at an estimated speed of 14 knots per hour. The total weight of the machinery is not to exceed 175 cwts., of which it is estimated, the main engines, auxiliaries, condenser, tanks, shafting and propeller, will take 85 cwts., leaving 90 cwts for the boiler, complete with water, etc. The specification requires that the power and speed shall be obtained with a forced draught from a pressure in the stokehold of not more than that due to 3 in. of water.

The space available for the boiler room is 12 ft. long by 6 ft. between the bunkers, and of this length 5 ft. 9 in. at least is required for firing, leaving 6 ft. 3 in. as length available for the boiler. The height is not strictly limited, but is to be kept as low as possible, so as to obtain a low centre of gravity for the boiler.

Design of Boiler.

The following facts in connection with the design must be kept in view :—

(1) The boiler must be capable of absorbing as much as possible the heat generated.

(2) The circulation must be good.

(3) The parts of the boiler must be capable of examination and cleaning.

(4) Parts likely to be subjected to deterioration, must be capable of removal, with a minimum disturbance of other parts.

(5) The steam must be supplied as dry as possible.

Of these conditions.—

(1) May be satisfied by a boiler of the small tube type in which a number of tubes are so arranged, that the hot gases are brought into intimate contact with them.

(2) Can be arranged by connecting the steam and water chambers, by a tube or tubes, which are not exposed to the furnace gases.

(3) The drums must be of such size that they can be readily inspected internally.

(4) The tubes joining the water and steam drums and forming the heating surface, must be capable of being easily removed and replaced. This can be arranged for in several ways, such as by making the tubes of such form that they can be sprung into place; by using tubes approximately straight, and having small doors in the steam drum so arranged that the tubes may be drawn through them; or, say, by having the tubes curved and of such length that they can be drawn into the steam drum, and thence by means of the manhole.

This latter arrangement will do away with any joints in the steam drum, other than the manhole having to be broken to remove a tube. Also the tubes can be bent to such a curve that the entrance to circular drums can be kept nearly normal; this obviates the necessity of nearly flat tube plates in the water drums and awkward jointing of same, and allows of simple covers at the ends of the drums.

(5) By using a medium sized steam drum and allowing the the tubes to discharge below the water level, frothing will not be serious, and the steam can be obtained fairly dry, without the use of dish plates, by using internal pipes having slots in their upper sides.

In addition to the above, simplicity of construction and working, minimum number of joints, and simple connection of parts should be sought, even to the sacrifice of some efficiency, as the highest conditions of labour may not be always obtained.

This would lead to the adoption of circular shells for the steam and water drums, and tubes of simple curves with expanded ends, capable of being removed without disturbing the bunkers, or sides of the boiler; this being arranged for as previously intimated, by constructing so that they may be withdrawn into the steam drum.

Grate Surface and Heating Surface.

The first thing we require to know is the amount of heating surface and grate surface necessary to obtain the steam required, under the given conditions.

The data for obtaining these surfaces, can only be obtained by reference to the performances of existing boilers of a similar type, and considerable judgment is necessary in adopting values for calculation.

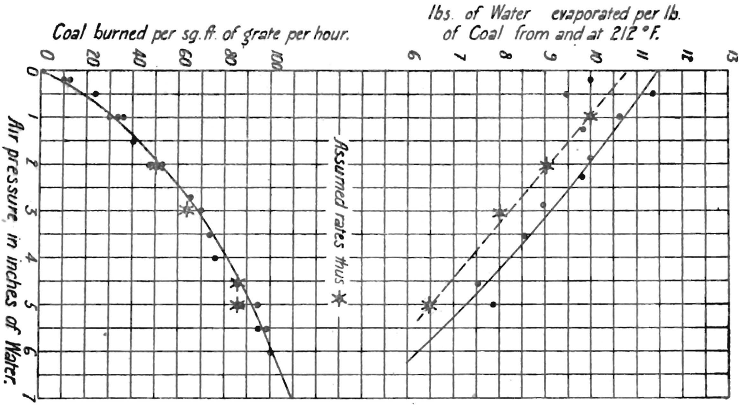

FIG. I.

In Fig. I. are shewn curves based on actual results giving the number of pounds of coal burnt to the square foot of grate per hour, and the evaporation per pound of coal, from and at 212° Fah., under various air pressures, from boilers of the small tube water tube type (Thorneycroft & Yarrow). In selecting values, it will be noticed that as the air pressure increases, the evaporation per pound of coal lessens, and also that the rate of burning per square foot of grate does not vary exactly as the air pressure. It may be assumed that these figures are obtained under the best conditions, and as it is desirable to obtain the contract power under ordinary condititions, lower efficiencies should be taken, than the curves indicate. If we assume the rate of firing to be as given, it would be better to take a lower value for the evaporation, and we should then in aiming at simplicity and compactness, assume a lower ratio of heating surface to grate surface than is often adopted in this class of boiler.

These ratios are commonly for the Throneycroft boiler (45 to 50) to 1. Yarrow (40 to 50) to 1; the higher values being generally used. If we take a ratio of 40 to 1 it is only necessary to fix the grate surface to get some idea of the size and shape of the boiler.

The I.H.P. that can be expected per square foot of grate, with different types of machinery may be taken as follows—air pressure about 2 in., being say, 56 lb. of coal per square foot of grate per hour.

	Compound.	Triple.	Quadruple.
Steam pressure, per square inch	180 lb.	220 lb.	250 lb.
I.H.P., per square foot of grate	18 to 22.	23 to 30.	28 to 32.

The points that might be selected for the various rates of working are shewn on the curve, and are as follows :—

Air pressure	3 in. of water.
Coal burned per square foot of grate per hour ..	.. 65 lb.
Evaporation per pound of coal from and at 212° F.	.. 8 lb.

Now the main engines are to deliver 200 i.h.p., and to obtain this, steam has to be supplied to various auxiliaries, viz., circulating pump, and fan engines. The consumption of steam may be reckoned in detail, or the whole consumption may be referred to the main engines. With a known type of machinery we could estimate in the latter way, assuming from 25 to 27 lbs. of steam per hour per i.h.p.

This would give $200 \times 27 = 5{,}400$ lbs. steam per hour.

In detail — Cyl. Area. Stroke. Cut-off. Strokes per hour. lbs. per cub. ft. of steam at 185.

$$\text{Main Engines } \frac{50{\cdot}26 \times 9 \times {\cdot}6 \times (1{,}000 \times 60) \times {\cdot}441}{1{,}728} = 4{,}100, \text{ say.}$$

Auxiliaries 2, say 3 in. dia. $\times$ $2\frac{1}{2}$ in. stroke (revs. 600 and 1,200)

Mean Stroke.

$$= \frac{7{\cdot}07 \times 2{\cdot}5 \times {\cdot}6 \times 1{,}800 \times 60 \times {\cdot}441 \times 2}{1{,}728} = 600, \text{ say.}$$

	4,700
Add 15% for wastes, etc.	700
Say 5,500 lb. of steam per hour	5,400

Now 8 lb. of water per pound of coal, from and at 212° F., are equivalent to $\frac{8}{1{\cdot}17} = 6{\cdot}8$ lb. of steam at 185 lb square inch from water at 100° F., per lb of coal.

Coal per square foot of grate per hour $= 65$

Steam per pound of coal per hour $= 6{\cdot}8$

$$\therefore \text{ grate area } = \frac{5{,}500}{65 \times 6{\cdot}8} = 12{\cdot}45 \text{ square foot.}$$

Say ... 12·5 ,, ,,

The heating surface required is therefore $12{\cdot}5 \times 40 = 500$ square foot.

This is equivalent to 11 lb. steam per square foot of heating surface, which is high for economical working, but as the air pressure is high, and weight of great importance, this could be accepted and economy at contract speed sacrificed. At lighter loads, when at ordinary steaming speeds, the evaporation per pound of coal would increase, the coal burned per square foot of grate would decrease, and the evaporation per square foot of heating surface would come down to normal.

For instance, at 11·5 knots per hour, about 135 i.h.p. would be required (this is about 2/3 full power), and this would take say, 26 lb. of steam per i.h.p. under more economical condition =

$\therefore$ Total per hour $= 135 \times 26 = 3{,}500$ lb. say. This would mean running with an air pressure of 1 in. only, burning 30 to 35 lb. of coal per square foot of grate, and evaporating about 10 lb. of water per pound of coal from and at 212° F., or,

$$\frac{10}{1{\cdot}17} \times 33 \times 12{\cdot}5 = 3{,}500 \text{ lb. about,}$$

and this gives an evaporation of 7 lb. of water per square foot of heating surface, which is about normal, being about 3·8 square foot of tube per i.h.p.

These points having been decided, the boiler can now be blocked out, to find out how the dimensions are likely to come within the conditions.

Assume the following in connection with the tubes :—

External diameter	...	$1\frac{1}{8}$ in. ; thickness, ·116 in.
Diameter at top	...	$1\frac{3}{16}$ in.
Diameter at bottom	...	1 in.

These will be spaced zig zag in the drums, say at 30° in the steam drum, longitudinal pitch $1\frac{9}{16}$ in., leaving $\frac{3}{8}$ in. between the top ends of the tubes.

These tubes have to stand a test pressure of 1,500 lb. per square inch, and with a thickness of 11 L.S.G., or, ·116 in., the stress is about 6,000 lb. per square inch.

With the tubes pitched as suggested, the distance between rows will be

$$\frac{1{\cdot}5625 \times \text{Cot } 30^\circ}{2} = 1\tfrac{11}{32} \text{ in.}$$

Say, $2\frac{11}{16}$ in. pitch circumferentially in steam drums.
$2\frac{1}{8}$ " " in water drums.

In choosing the sizes of the drums, they must be such that the tubes can be pitched in on the circumference. From a large number of examples of boilers of the small tube type, the ratio of the steam drum diameter to that of the water drums is about 2 to 1, sometimes running a little more,

Block out the boiler roughly and allow for the first trial :—

15 in from bottom of boiler to top of fire bars.
12 in. from topside of bars to C.L. of water drums.
2 ft. 6 in. + 1 ft. 3 in. from C.L. of water drums to C.L. of steam drum.

Making the steam drum 2 ft. 6 in. internal diameter, and the water drum, say, 1 ft. 3 in. internal diameter. Then allowing $2\frac{1}{2}$ in. for width of joint on water drum, and 1 in. clearance at sides of bunkers, the horizontal centres of the water drums will be

$$6 \text{ ft. less } (15 \text{ in.} + 2\tfrac{1}{2} \text{ in.} + 2\tfrac{1}{2} \text{ in.} + 1 \text{ in.} + 1 \text{ in.}) = 4 \text{ ft. } 2 \text{ in.}$$

Block these drums out, and then some idea can be obtained as to how the tubes can be packed. These can be sketched in, working on the assumed pitches, and keeping the curves tangent to a line of not more than 10° variation from the normal.

It is found that fourteen pitches can be got in, giving fourteen rows of tubes on each side, and also one row on each side for a set of wall tubes, which can be packed closer than usual, longitudinally, and will effectually strain the heat from the gases and act as a protection to the sides of the casing, enabling brickwork to be dispensed with. These can be $1\frac{1}{4}$ in. to $1\frac{5}{16}$ in. pitch, having their ends reduced to $\frac{7}{8}$ in. diameter.

The surface can now be reckoned out for one complete row, by assuming a mean length, or measuring the length of the tubes from the sketch.

Assume a mean length of 33 in., as the distance between outsides of drums is about 30 in.

$\therefore$ the surface of one complete row of tubes is

$$\frac{30 \times 33 \times 1{\cdot}125 \times 3{\cdot}14}{144} = 24{\cdot}2 \text{ square feet.}$$

$$\therefore \text{ total number of complete rows } = \frac{500}{24{\cdot}2}$$

= 21 say, ignoring the extra number of tubes in the row of wall tubes.

$\therefore$ distance between outside rows is

$$1{\cdot}\tfrac{9}{16} \times 20 = 31 \text{ in. about}$$
$$+ \tfrac{1}{2} \text{ pitch} = \tfrac{7}{8} \quad \text{say}$$

32 in., say.

Block this out on the longitudinal view of the boiler and sketch in the remainder of the boiler, roughly. There is then obtained a total length of

	inches.
Tubes, centres	32·0
1 in. each end from centre of tube to brickwork ...	2·0
Brickwork $2\frac{1}{2}$ in. each end	5·0
Dishing of drum, $5\frac{1}{4}$ in. each end	10·5
Allowance to get an easy bend on downcomer, of say, 6 in. diameter	12·0
Flange on downcomer...	5·5
Total	67 in.

This shows that the boiler can be lengthened by 8 in. without exceeding the space allowed. The tubes can therefore be lessened in length, and the drums made smaller in diameter, thus lessening the height of the boiler, and lowering the centre of gravity.

Assume reduced sizes, and block out boiler again.

Steam drums 2 ft. 3 in. diameter (internal).
Water drum 13 in. ,, ,,

Length of of tubes at closest part of drum can be about 2 ft., giving say 3 ft. 1 in. between the centre lines of drums.

C. to C. of water drums is

$$6 \text{ ft. less } (13 + 5 + 1\tfrac{1}{2}) = 4 \text{ ft. } 4\tfrac{1}{2} \text{ in.,}$$

sketch in the tubes, and ascertain the probable lengths.

Having got so far, the length of the boiler can be again checked as follows :—

Say thirteen tubes in a row of twenty-six for both sides, and two sets of wall tubes.

Then surface for one complete row of ordinary tubes taking a mean length of 27 in.

$$= \frac{26 \times 27 \times 1{\cdot}125 \times 3{\cdot}14}{144} = 19 \text{ square feet.}$$

This gives, say twenty-six rows required without the wall tubes, but something must be kept in hand, as it may be necessary or convenient to block out some of the tubes.

Length required $= 1\frac{9}{16}$ in. $\times\ 26\frac{1}{2} = 3$ ft. $5\frac{3}{8}$in. $\times \frac{1}{32}$.

Assuming thirty-two wall tubes, there are thirty-one spaces in, say 3 ft. $5\frac{1}{4}$ in. giving a pitch of $1\frac{5}{16}$ in. + $\frac{1}{64}$ in. leaving $\frac{3}{16}$ + $\frac{1}{64}$ in. between the tubes.

The total surface is

Twenty-six rows of ordinary tubes at 19 sq. ft.	=	494 sq. ft.
Sixty-four wall tubes, say 38 in. long ..	=	66 ,,
Total		560 ,,

This gives a sufficient margin to work on and would allow for a few tubes at each end, being blocked out, giving easier access to the ordinary tubes by means of the space between them and the wall tubes.

Therefore adopting these sizes, the length of boiler becomes

Tubes	3 ft. $5\frac{1}{2}$ in. say
C. of tubes to brickwork, 1 in. each end ..	2 ,,
Brickwork, $2\frac{1}{2}$ in. each end	5 ,,
Dishing of drums, 5 in. each end	10 ,,
Allowance for downcomer	1 ft. $5\frac{1}{2}$,,
Total	6 ft. 4 in.

which is near enough to the restricted dimensions and shews that the boiler as proposed can fulfil the conditions.

Checking size of fire grate

Length between brickwork $= 3$ ft. $5\frac{1}{2}$ in. + 1 in.

allowing that the bottom courses of brickwork are set in, so as to leave 1 in. air space.

$$\therefore \text{ width of grate} = \frac{12.5}{3{\cdot}54} = 3{\cdot}53.$$

Say 3 ft. $6\frac{1}{2}$ in.

this with $2\frac{1}{2}$ in. brickwork at sides of furnace, and will leave room for the boiler bearers in the boat, giving say $2\frac{1}{2}$ in. between bearer and plate, and allowing for a timber 5 in. wide.

Design of Steam Drum.

The tubes in perforating the plate leave a diminished section which can be allowed for as follows :—

Longitudinal pitch, $1\frac{9}{16}$ in.

Tubes, diameter, $1\frac{3}{16}$ in. at the steam drum end, requiring a hole, say $1\frac{3}{16}$ in. + $\frac{1}{32}$ in. and leaving $\frac{11}{32}$ in. between the holes.

The circumferential pitch is $2\frac{11}{16}$ in., which with zig zag tubes gives 1·552 in. diagonal pitch, leaving 1.552 − 1·21875 = ·333 in. between holes.

From this is obtained as the percentage of perforated as compared with solid plate $\frac{p - d}{p} \times 100$.

$$\frac{1{\cdot}5625 - 1{\cdot}21875}{1{\cdot}5625} \times 100 = 22\ \%.$$

$$\frac{1{\cdot}552 - 1{\cdot}21875}{1{\cdot}552} \times 100 = 21{\cdot}4\ \%.$$

the above is for the ordinary tubes.

For wall tubes

$$100 \times \frac{1{\cdot}3125 - {\cdot}90625}{1{\cdot}3125} = 31\ \%.$$

Internal diameter of steam drum	=	27 in.
Working pressure	=	185 lb. per square inch.
Tensile strength of steel ..	=	62,500 lb. ,, ,,

The factor of safety usually worked on for boilers of good workmanship is five, but as the shell in this case is exposed to the direct action of the hot gases, this would need to be increased. Lloyd's specify one-third increase for steam drums so exposed, and this gives a factor of 5 × 1·33 = 6·66. The thickness of plate required then becomes

$$t = \frac{p \times f \times r}{s \times \%}$$

p = working pressure lb. per square inch.

f = factor of safety.

r = radius of shell (internal).

s = tensile strength of material in pound per square inch.

$\%$ = $\%$ of plate as found above (least value).

$$t = \frac{185 \times 6{\cdot}66 \times 13{\cdot}5}{62500 \times 21{\cdot}4} = 1{\cdot}24$$ Say $1\frac{1}{4}$ in.

by Lloyd's rules for shells (steel)

$$t = \frac{w.p \times d}{c \times \%} + 2$$

where t = thickness in sixteenths of an inch.

$w.p.$ = working pressure lb. per square inch.

d = mean diameter.

c = 20 (less 1/3) or 13·33.

$\%$ = least $\%$ value for plate.

$$t = \frac{185 \times 28{\cdot}25}{13{\cdot}33 \times 21{\cdot}4} + 2 = 20{\cdot}3 \text{ (sixteenths)}$$

Say $1\frac{1}{4}$ in.

It is only the lower portion of the drum that is under these adverse conditions, and so to save weight the thickness of top portion could be lessened, making the joints with double riveted double butt straps, from which an efficiency of 70 % could be reckoned on.

Then by Lloyd's

$$t = \frac{w.p. \times d}{c \times \%} + 2 \therefore t = \frac{185 \times 27{\cdot}375}{20 \times 70} + 2 = 3{\cdot}6 + 2 = 5{\cdot}6 \text{ (sixteenths)}$$

Say $\frac{3}{8}$ in. thick.

In the above (c) is a constant of value (20) for the type of joint used, and the (%) is the least percentage of strength of joint for plate or rivets, as compared with the solid plate obtained as follows:—

$$\% \text{ plate } \frac{p - d}{p} \times 100$$

$$\% \text{ rivets } \frac{n \times a}{p \times t} \times 85 \times 1{\cdot}75$$

where
p = pitch.
d = diameter of rivet hole.
n = number of rivets in a pitch.
t = thickness of plate.
a = area of rivet, which is to be multiplied by 1·75 where the rivets are in double shear.

The rivet diameter can be taken as $t + \frac{1}{4}$ = diameter of rivet before driving, which is usual practice for this type of joint.

The pitch is first settled by trial so as to obtain something that will work in, but it must not exceed

$$(t \times c) + 1\tfrac{5}{8}$$

where c is a constant depending on the type of joint, having a value in this case of 3·5.

This gives the maximum pitch as

$$({\cdot}375 \text{ in.} \times 3{\cdot}5) + 1{\cdot}625 \text{ in.} = 2{\cdot}9 \text{ in.}$$

Taking a length on the drawing (after allowing for the circumferential lap joints and two short pitches) of 42 in., it is seen that $\frac{42}{2{\cdot}93} = 14{\cdot}3$ pitches, an odd number. The next pitch suitable would be $\frac{42}{16} = 2{\cdot}625$ which gives an even number, and a pitch below the maximum.

The joint then becomes

Rivet holes, $\frac{11}{16}$ in. diameter; pitch, $2\frac{5}{8}$ in.
Two butt straps ($\frac{2}{3}$ of plate) = $\frac{5}{16}$ say, minimum thickness.
Two rivets in a pitch.
% plate = 73·8 %
% rivets = 112 %

which is near enough to the percentage assumed.

Distance from centre of rivet to edge of plate (minimum)

$$1{\cdot}5d = 1\tfrac{1}{16} \text{ in.}$$

Distance between rows of rivets (mininum)

$$= \sqrt{\frac{(11p + 4d)(p + 4d)}{10}} = 1{\cdot}31$$

Say, $1\frac{3}{8}$ in.

Hence straps require to be 7 in. $\times \frac{5}{16}$ in.

See Plate I.

Dished Ends.—The dished ends under ordinary conditions would be strong enough, if made of the same thickness as the shell, and struck with a radius equal to diameter of shell. Owing however to the fact that openings have to be cut in the ends for manholes, mountings, etc., and that the plates must be made of a steel suitable for flanging, they must be designed to suit these conditions. Taking the back end and assuming a manhole 15 in. $\times$ 11 in., and also a 6 in. diameter pipe for the downcomer, and two $\frac{3}{4}$ in. bolt holes, there is out of a section of plate about 27 in. long (allowing for the radius at ends), 11 in. + 6 in. + $1\frac{1}{2}$ in. cut out. Of the 11 in. opening all would not be lost, as 1 in. all round could be assumed as flanged inwards to form the face of the joint for the door. This makes $14\frac{1}{2}$ cut out, being $\frac{14{\cdot}5}{27}$ or 54 % of the solid plate leaving 46 %). Now the radius of the end is 27 in. giving a portion of a sphere of 54 in. diameter.

Assuming a factor of five, on a flanging steel of 50,000 lb. per square inch, ultimate strength, the equation for strength of the sphere becomes, where f = safe stress per square inch.

$$\text{Area of 54 in. dia.} \times wp = f \times \pi \times 54 \text{ in. dia.} \times t \times \%$$

$$\therefore t = \frac{2{,}290 \times 185}{10{,}000 \times 3{\cdot}14 \times 54 \times {\cdot}46} = {\cdot}55 \text{ in. } \tfrac{9}{16} \text{ in.}$$

Say $\frac{5}{8}$ in. thick.

The front end would not be cut to such an extent as the back, having say at the least section 2 in. (for boss to guage glasscock) + $3\frac{1}{2}$ in. (for main stop valve) + $1\frac{1}{2}$ in. for bolt holes + $5\frac{1}{2}$ in. (for sight hole, which is made 6 in. $\times$ 9 in.) cut out, giving $\frac{12{\cdot}5}{27}$ or 46 % which leaves 54 % of solid plate.

This would give a thickness of .47 in.

Adopt $\frac{5}{8}$ in. at back, and $\frac{9}{16}$ in. at the front.

The size and the pitch of the rivets in the circumferential seams can be obtained as follows:—

Single riveted lap joint.

$d = 1{\cdot}2 \sqrt{t}$ before riveting.

$d = 1{\cdot}3 \sqrt{t}$ after ,,

$\therefore d = 1{\cdot}2 \sqrt{{\cdot}375} = \frac{3}{4}$ in. diameter.

$d = 1{\cdot}3 \sqrt{{\cdot}375} = {\cdot}8$ say, $\frac{13}{16}$ in. diameter.

pitch $= 1{\cdot}09 + d = 1{\cdot}89$ in.