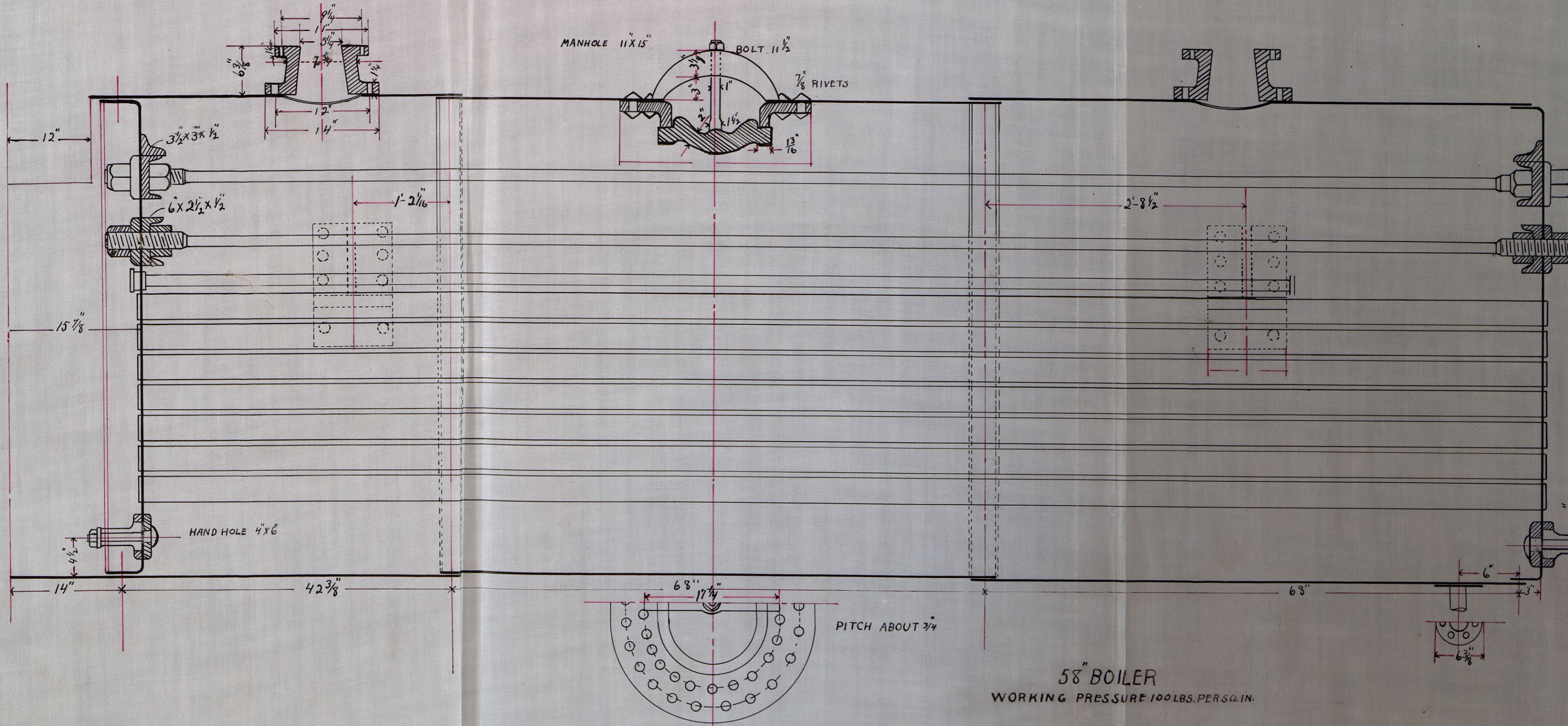


DESIGN OF A 75 H. P. A. S. M. E
STANDARD MULTITUBULAR BOILER.

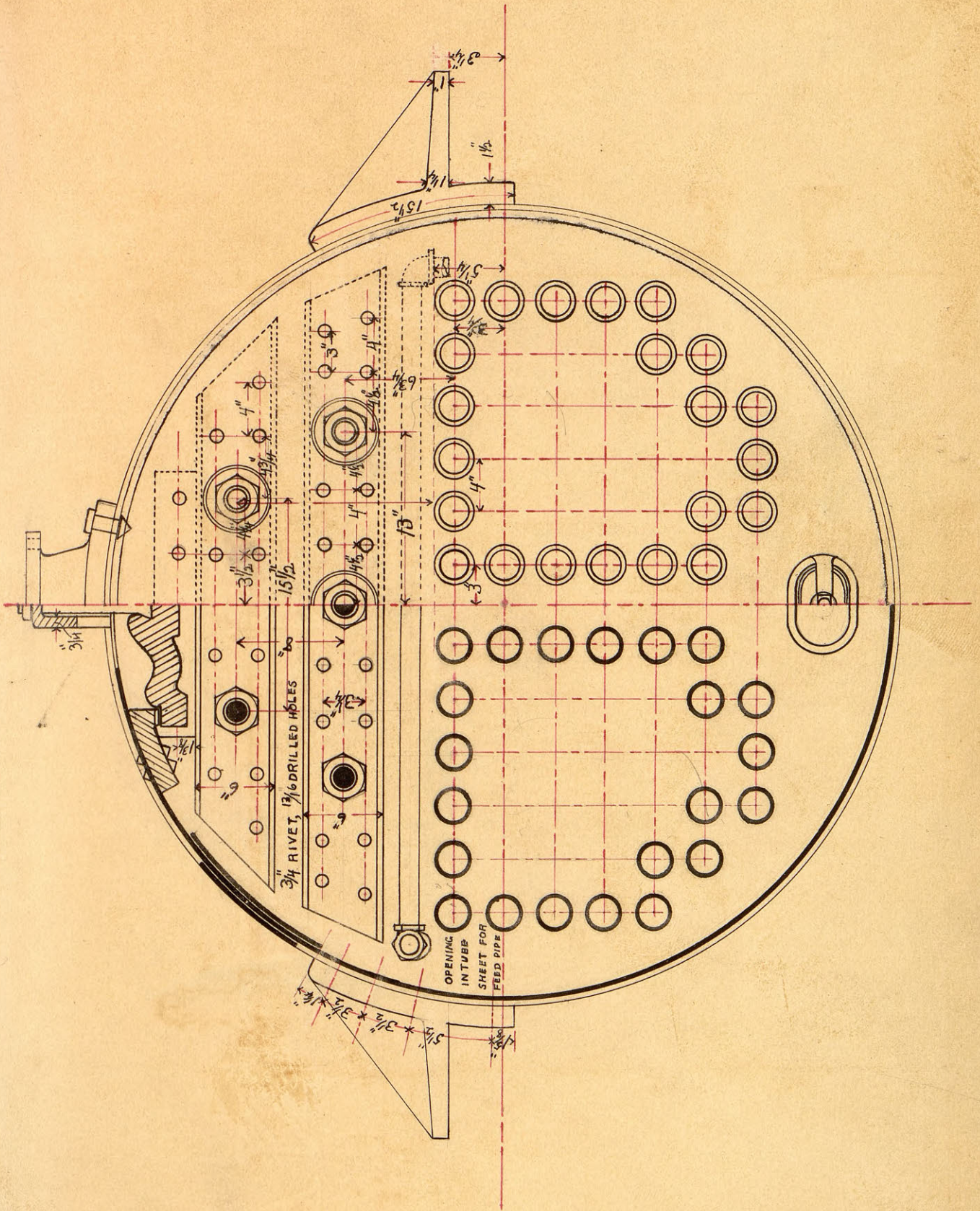
BY

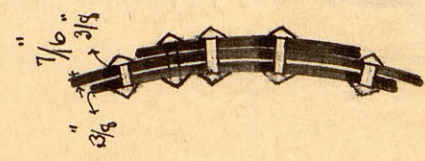
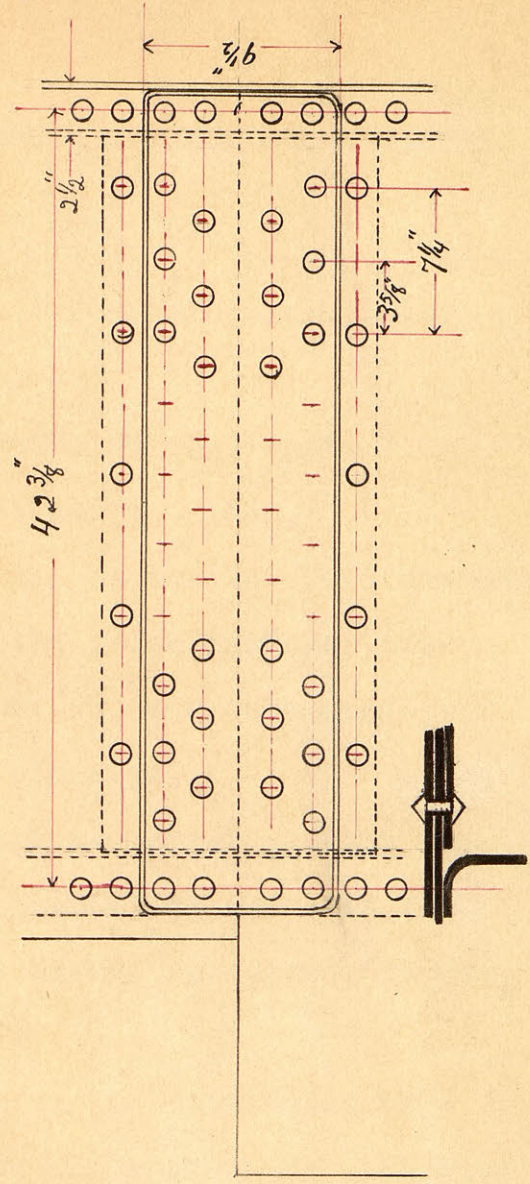
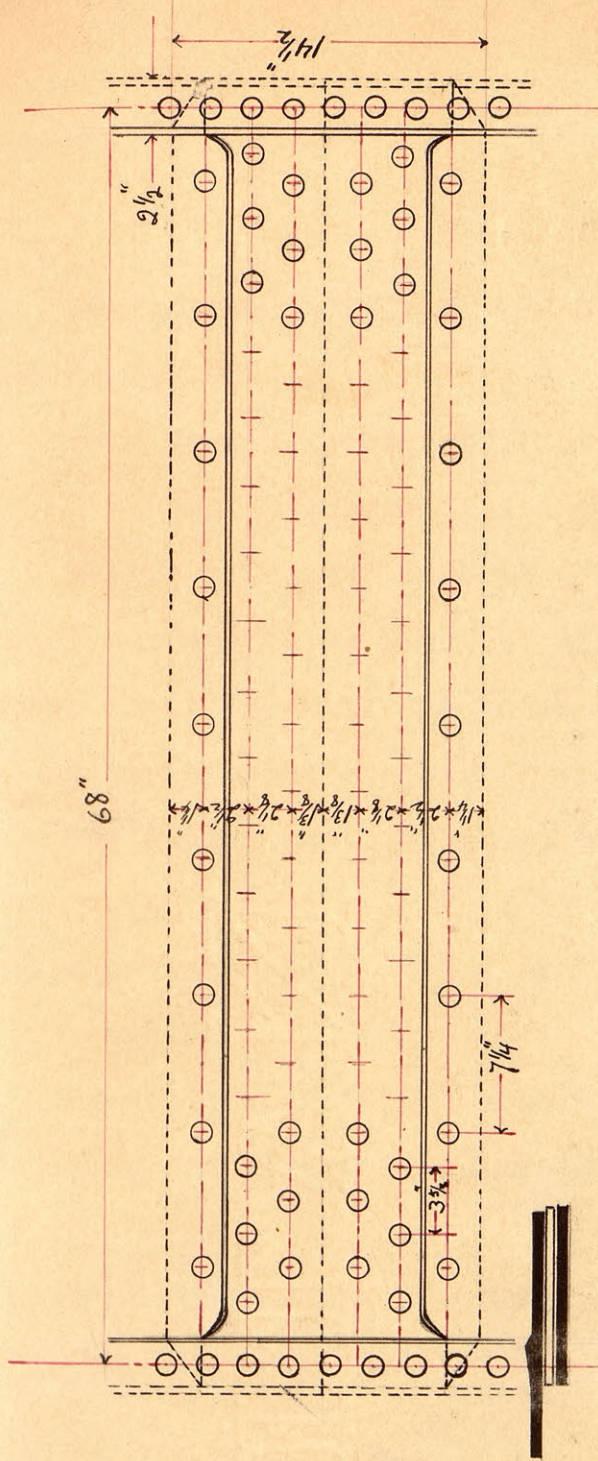
H. P. RICHARDS,

'02.



58" BOILER
 WORKING PRESSURE 100 LBS. PER SQ. IN.





PROBLEM FOR THESIS.

Design for a 75 H. P. (A. S. M. E standard) horizontal multitubular boiler.

Working pressure is 100 pounds per sq. inch.

Test pressure is 150 pounds per sq. inch.

Length of tubes, 15 feet which is not to exceed 60 diameter the heating surface to be about 37 times the grate surface; the steam space about one-half the water space; and the tube area between one-seventh and one-eighth the grate area.

The boiler is to be supplied with two steam outlets for five inch pipes, a hand hole four by six inches (4" x 6") at the bottom of each tube sheet, and a man hole eleven by fifteen inches (11" x 15") inside.

Provisions are also to be made for the attachment of a one and one-fourth (1-1/4") inch feed pipe and a two (2") inch blow off pipe.

The reach of the riveting machine is six feet, six inches (6' - 6").

GRATE AREA.

The A. S. M. E standard requires that 34.5 pounds of water per hour, shall be evaporated from and at 212°F for each H. P.

$$\log 75 = 1.875061$$

$$\log 34.5 = \frac{1.537819}{3.412880} = 0.450615 \text{ pounds}$$

$$\text{anti log } 3.412880 = 2587.5 \text{ pounds}$$

This gives the number of pounds of water evaporated per hour, from and at 212°F, for a 75 H. P. boiler.

With an equivalent evaporation of 9 pounds of water per pound of coal; the coal burned per hour will be -

$$\log 2587.5 = 3.41288$$

$$\text{co log } 9 = \frac{\bar{1}.04576}{2.45864}$$

$$\text{Anti log } 2.45864 = 287.5 \text{ pounds.}$$

With a rate of combustion of 12 pounds of coal per square foot of grate surface, the grate area must be:

$$\log 287.5 = 2.45864$$

$$\text{Colog } 12 = \frac{\bar{2}.92082}{1.37946}$$

$$\text{Anti log } 1.37946 = 23.958 \text{ square feet.}$$

TUBES.

In the problem, the length of the tubes is 15 feet which is not to exceed 60 diameters. This gives for the external diameter of the tubes 3 inches.

For a 3 inch tube, the external area is 7.07 square inches, and 6.08 square inches for the internal area.

The internal circumference is 8.74 inches, and the external circumference is 9.42 inches.

Let the area thru the tubes be chosen as $\frac{1}{7.5}$ of the grate area, equal to

$$\log 23.95 = 1.37946$$

$$\log 144 = 2.15836$$

$$\text{colog } 7.5 = \frac{\bar{1}.12494}{2.66276}$$

$$\text{Anti-log } 2.66276 = 460 \text{ square inches}$$

Since the area thru one tube is 6.08 square inches, there will be required:

$$\log 460 = 2.66276$$

$$\text{colog } 6.08 = \frac{\bar{1}.21610}{1.87886}$$

Anti-log 1.87886 = 75.66 tube
therefore use 76 tubes.

STEAM SPACE.

Taking .85 cu. ft. of steam per H. P. we have, for the steam space,

$$.85 \times 75 = 63.75 \text{ cu. ft.}$$

DIAMETER OF BOILER.

The steam space is to be $1/2$ the water space. The steam space, water space, and the space occupied by the tubes makes up the total contents of the boiler. Now there are 76 tubes, 3 inches in diameter and 15 feet long. The area of the external transverse section has been found to be 7.07 square inches, the space occupied by the tubes is consequently:

$$\log 76 = 1.88081$$

$$\log 7.07 = 0.84942$$

$$\log 15 = 1.17609$$

$$\text{colog } 144 = \frac{\bar{3}.84164}{1.74796}$$

$$\text{Antilog } 1.74796 = 55.95 \text{ cubic feet.}$$

The total space in the boiler is

$$55.95 + 63.75 + 127.5 = 247.2 \text{ cubic feet.}$$

where

$$55.95 \text{ cubic feet} = \text{tube space}$$

$$63.75 \quad " \quad " = \text{steam} \quad "$$

$$127.5 \quad " \quad " = \text{water} \quad "$$

The cylinder is 15 feet long, so its transverse area is:

$$\log 247.2 = 2.39305$$

$$\text{colog } 15 = \frac{\bar{2}.82391}{1.21696}$$

$$\text{Anti-log } 1.21696 = 16.48 \text{ square feet.}$$

This corresponds to a diameter of 54.96 inches, but I took 58 inches as the diameter of the boiler.

HEATING SURFACE.

The heating surface of a cylindrical tubular boiler consists of all the shell below the supports at the side wall, all the inside of the tubes, and part of the rear tube plate. Usually half of the cylindrical part of the shell is heating surface. In this case the heating surface exclusive of the tube plate, will amount to:

SHELL:

$$\log .5 = \bar{1}.69897$$

$$\log 3.1416 = 0.49715$$

$$\log 58 = 1.76343$$

$$\log 15 = 1.17609$$

$$\text{colog } 12 = \frac{\bar{2}.92082}{2.05646}$$

$$\text{Anti-log } 2.05646 = 113.87 \text{ square feet.}$$

TUBES.

$$\log 76 = 1.88081$$

$$\log 8.74 = 0.94151$$

$$\log 15 = 1.17609$$

$$\text{colog } 12 = \frac{\bar{2}.92082}{2.91923}$$

$$\text{Anti-log } 2.91923 = 830.29 \text{ square feet.}$$

The total heating surface exclusive of the tube plate is:

$$830.29 + 113.87 = 944.16 \text{ square feet.}$$

The heating surface is to be about 37 times the grate surface. I find that the relation between the heating surface and grate surface is:

$$944.16 : 23.96 :: x : 1$$

$$x = 39.4$$

This gives the heating surface 39.4 times the grate surface.

WATER LEVEL.

It is now necessary to determine the position of the water level to see if there will be sufficient free-water surface and sufficient distance from the water level, to the shell above it.

Since the whole boiler is cylindrical, the area of the head of the boiler exposed to steam and to water will have the same ratio as that of the steam space to the water space. Consequently the area of the head above the water-level must be $1/3$ of the total area of the head less the combined areas of the tubes.

The area of a circle that has a diameter of 58 inches is 2642 inches.

The area of 76 tubes, each having an external cross section of 7.07 will be:

$$76 \times 7.07 = 537.32 \text{ square inches.}$$

The area of the head exposed to steam is;

$$2642.08 - 537.32 = 2104.76$$

$$1/3 \times 2104.76 = 701.58 \text{ square inches.}$$

The height of a segment of a 58 inch circle, which has the area of 701.58 square inches.

The height is:

$$\begin{aligned} \log 701.58 &= 2.84608 \\ \text{colog } 58 &= \bar{2}.23657 \\ \text{colog } 58 &= \frac{\bar{2}.23657}{1.31922} \\ \text{Anti-log } \bar{1}.31922 &= .2088 \end{aligned}$$

The ratio of the height to the diameter is .311. The height of the segment is,

$$58 \times .311 = 18.038 \text{ inches}$$

The water level will be;

$$29 - 18.038 = 10.96 \text{ inches above the centre of the boiler.}$$

THICKNESS OF SHELL.

Assuming 5 as a factor of safety; and the method of determining the thickness of the shell is:

$$t = \frac{p r}{s} ; \text{ where}$$

t = thickness of shell
 p = pounds per square inch
 r = radius of boiler in inches
 s = stress per square inch on the metal.

$$\begin{array}{r} \log 100 = 2.00000 \\ \log 29 = 1.46240 \\ \log 5 = 0.69897 \\ \text{colog } 55000 = \frac{5.25964}{1.42101} \end{array}$$

Anti-log $\bar{1}.42101 = .26 = 6/16$ inches nearly, and adding $1/16$ inch for corrosion gives $7/16$ inch for the thickness of the shell.

HAND HOLES.

The hand holes are to be 4 by 6 inches.

The area of the hand hole is .785 b d, which is;

$$\begin{array}{r} \log .785 = \bar{1}.89487 \\ \log 4 = 0.60206 \\ \log 6 = 0.77815 \\ \hline 1.27508 \end{array}$$

Anti-log 1.27508 = 18.84 square inches

The pressure that comes on this area will be;

$$\begin{array}{r} \log 18.84 = 1.27508 \\ \log 150 = 2.17609 \\ \hline 3.45117 \end{array}$$

Anti-log 3.45117 = 2826 pounds.

Treating this as a beam 6 inches long and 4 inches wide; gives for the value of M:

$$\frac{M}{S} = \frac{I}{C} \quad M = \frac{S I}{C}$$

M = bending moment
 S = working stress
 I = moment of inertia
 C = distance the most strained fibre is from neutral axis.

$$\begin{aligned} \log 2826 &= 3.45117 \\ \log 6 &= 0.77815 \\ \text{colog } 8 &= \frac{1.09691}{3.32623} \end{aligned}$$

$$\text{Anti-log } 3.32623 = 2119 \text{ pounds}$$

Assuming 5 as a factor of safety, gives for d ,

$$\frac{I}{C} = \frac{6 b d^3}{6}, \quad d^3 = \frac{6 M}{b S}$$

The tensile stress for gun metal = 25000 pounds per square inch.

$$\begin{aligned} S &= 5000 \\ M &= 2119 \\ b &= 4 \end{aligned}$$

$$\begin{aligned} \log 6 &= 0.77815 \\ \log 2119 &= 3.32623 \\ \text{colog } 4 &= \frac{1.39794}{4.30103} \\ \text{colog } 5000 &= \frac{1.80335}{1.80335} \end{aligned}$$

$$\text{Antilog } 1.80335 = .6358 = d^2$$

$d = .79 = \text{nearly } \frac{26}{32}$ inches, but making it a convenient size, I took $7/8$ inch as the thickness for the cover to the hand holes.

MAN HOLES.

The man hole is 11 x 15 inches inside, and the area exposed to the steam pressure equals $.785bd$, which is:

$$\begin{aligned} \log .785 &= 1.89487 \\ \log 11 &= 1.04139 \\ \log 15 &= \frac{1.17609}{2.11235} \end{aligned}$$

$$\text{Anti-log } 2.11235 = 129.52 \text{ square inches.}$$

The total pressure that this area is subject to is:

$$\begin{aligned} \log 129.52 &= 2.11235 \\ \log 150 &= \frac{2.17609}{4.28844} \end{aligned}$$

$$\text{Anti-log } 4.28844 = 19429 \text{ pounds.}$$

$$M = S \frac{I}{c} = \frac{W L}{8}$$

The ultimate tensile strength of gun metal equals 25000 pounds per square inch.

Assuming 5 as a factor of safety,

$s = 5000$ pounds per square inch.

The bending moment (M) for a beam loaded uniformly equals $\frac{WT}{8}$, where W is the total weight, and T is the distance in inches;

this gives;

$$\begin{array}{r} \log 19429 = 4.28845 \\ \log 15 = 1.17609 \\ \text{colog } 8 = 1.09691 \\ \hline 4.56145 \end{array}$$

$$\text{Anti-log } 4.56145 = 36429 \text{ pounds}$$

$$d^3 = \frac{6M}{bS}, \text{ and } d' = \sqrt[3]{\frac{6M}{bS}} = 2 \text{ inches}$$

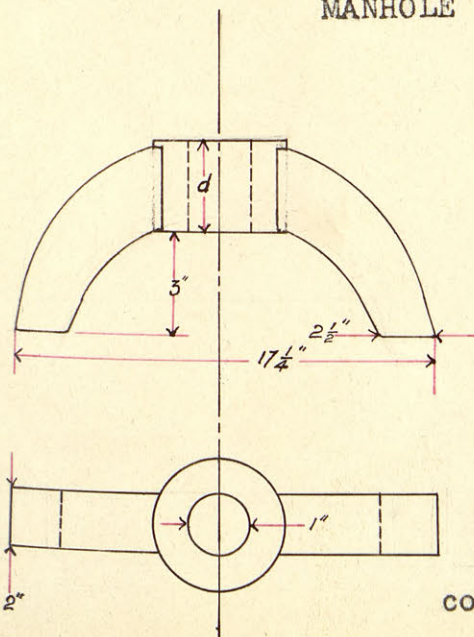
$$\begin{array}{r} \log 6 = 0.77815 \\ \log 36429 = 4.56145 \\ \text{colog } 11 = 2.95861 \\ \text{colog } 5000 = 4.30103 \\ \hline 0.59924 \end{array}$$

$$\text{Anti-log } 0.59924 = 3.974 = 4'' = d^3$$

$$d = 2 \text{ inches}$$

Where d is the thickness of the cover for the manhole.

MANHOLE COVER SUPPORTS.



Load on the bolt = 6000 pounds considering this support as a beam with a concentrated load in the middle; then the bending moment (M) = $\frac{PT}{4}$, where p is the load in pounds and T is the length in inches.

$$M = \frac{PT}{4} \text{ gives}$$

$$\begin{array}{r} \log 6000 = 3.77815 \\ \log 17.25 = 1.23679 \\ \text{colog } 4 = 1.39794 \\ \hline 4.41288 \end{array}$$

$$\text{Anti-log } 4.41288 = 25875 \text{ pounds,}$$

Assuming $b = 2''$

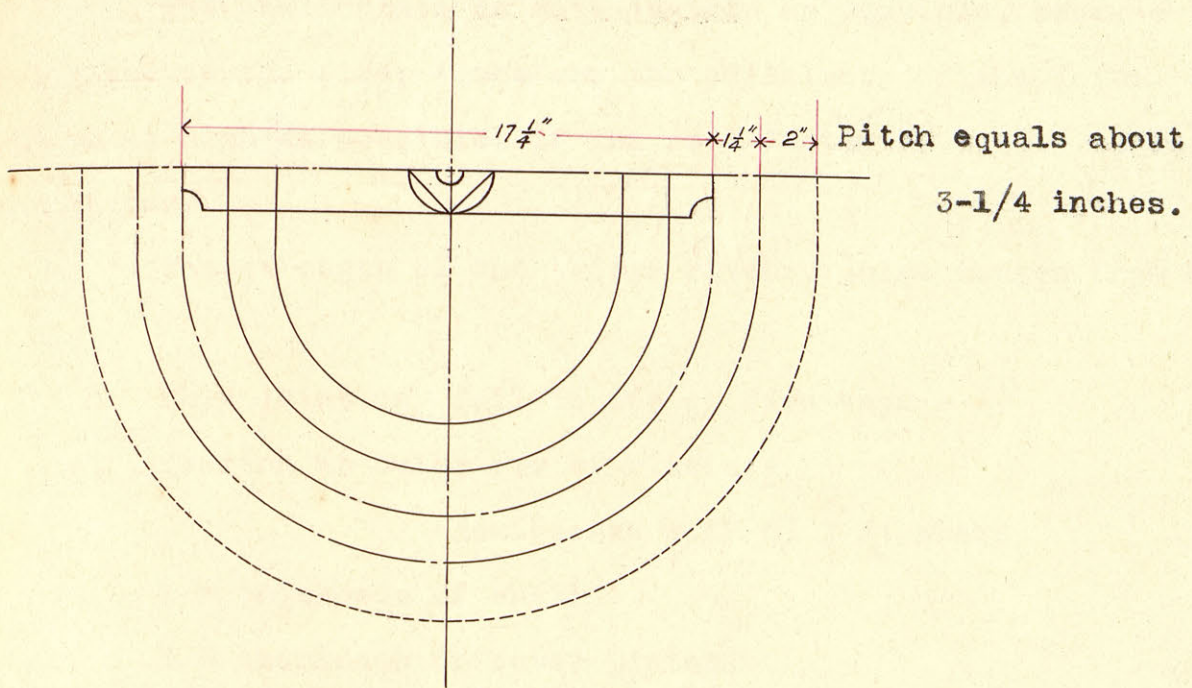
$$d = \frac{2 \quad 6 \text{ M}}{b \text{ S}} \text{ gives}$$

	$\log 6 = 0.77815$
\log	$25875 = 4.41288$
colog	$2 = \underline{1.69897}$
colog	$4000 = 4.39791$
	<u>1.28791</u>

$$\text{Anti-log } 1.28791 = 19.40'' = d^2$$

$d = 4\text{-}1/2''$ nearly.

TOP VIEW OF MAN HOLE AND RINGS.



ARRANGEMENT OF TUBES.

See Figure.

LONGITUDINAL JOINT.

The shell plate is made as thin as possible, because it is exposed to the fire; therefore the efficiency of the joint must be as high as possible, if the real factor of safety is to be satisfactory.

The strength of the triple riveted joint ranges from 85 to 90 %.

This joint may fail in one of five ways:-

A. Tearing at outer row of rivets:-

$$\text{Resistance} = (P-d) T F_t; \text{ where}$$

T = thickness of shell

T_c = thickness of cover plates.

d = diameter of steel rivets.

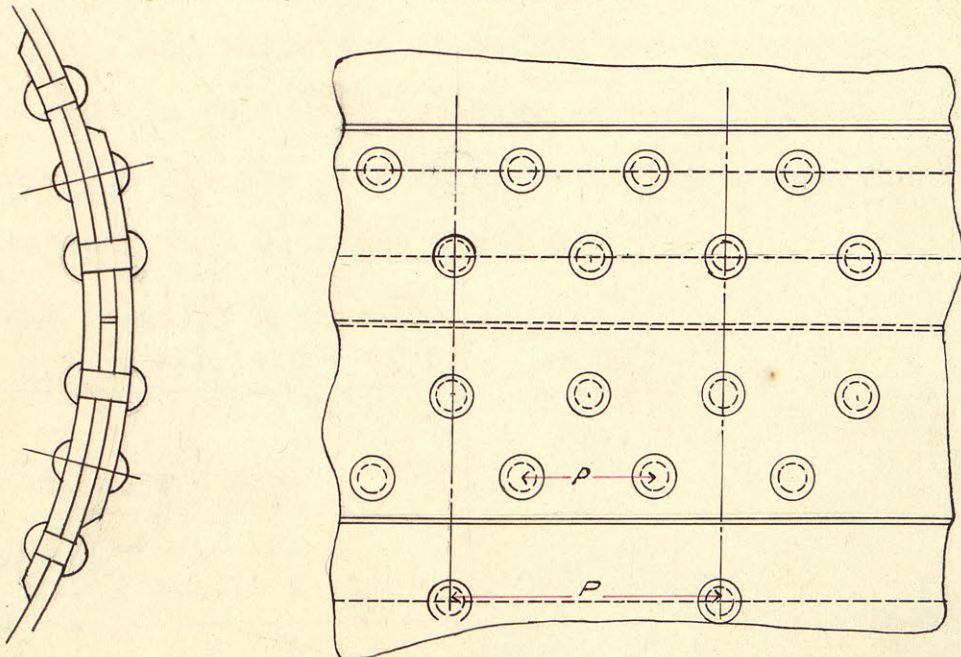
p = pitch of inner row of rivets.

P = pitch of outer row of rivets.

F_t = tearing strength of plate.

F_s = shearing strength of plate.

F_c = crushing strength of mild steel.



B. Shearing four rivets in double shear and one in single shear:

$$\text{Resistance} = \frac{9\pi d^2}{4} F_s$$

C. Tearing at the middle row of rivets and shearing one rivet.

$$\text{Resistance} = (P-2d) T F_t + \frac{\pi d^2}{4} F_s$$

D. Crushing four rivets and shearing one.

$$\text{Resistance} = 4 d T F_c + \frac{\pi d^2}{4} F_s$$

E. Crushing five rivets.

$$\text{Resistance} = 4 d T F_c + d T_c F_c$$

The diameter of the rivet will be found by equating the resistances A and C:-

Therefore,

$$(P-d) T F_t = (P-2d) T F_t + \frac{\pi d^2}{4} F_s$$

$$d = \frac{4 T F_t}{\pi F_s} = \frac{4 \times 7/16 \times 55000}{3.1416 \times 45000} = .68 = 11/16 \text{ inches}$$

which would be large enough for the diameter of the rivet,

but I took 7/8 inch for the diameter of the rivet.

There are several methods in which we may find the way the joint will fail, and then find therefrom the efficiency.

One method is to equate the five several resistances, two and two and calculate the pitch; the least pitch thus found must not be exceeded, thus equating B and C:

$$P = \frac{8\pi d^2}{4} \frac{F_s}{F_t} + 2d = \frac{9\pi d^2}{4} F_s = (P-2d) T F_t + \frac{\pi d^2}{4} F_s$$

$$P = \frac{8\pi d^2}{4} \frac{F_s}{F_t} + 2d = \frac{8 \times 3.1416 \times (7/8)^2}{4 \times 7/16} \times \frac{45000}{55000} + 2 \times 7/8 = 10.74 \text{ inches.}$$

Equating A. and B:

$$(P-d) T F_t = \frac{9\pi d^2}{4} F_s$$

$$P = \frac{9\pi d^2}{4 T} \frac{F_s}{F_t} + d = \frac{9 \times 3.1416 \times (7/8)^2}{4 \times 7/16} \times \frac{45000}{55000} + \frac{7}{8} = 11.3 \text{ inches.}$$

Equating A and E;-

$$P = \frac{4 d F_c}{F_t} + \frac{d T_c F_c}{T F_t} + d = 4 \times \frac{7}{8} \times \frac{95000}{55000} + \frac{7/8 \times 3/8 \times 95000}{7/16} + 7/8$$

$$= 8.2 \text{ inches}$$

Here T the thickness of the cover plate is taken as 3/8 inches thick.

Equating A and D;-

$$(P - d) T F_t = 4 d T F_c + \frac{\pi d^2}{4} F_s$$

$$P = \frac{4 d F_c}{F_t} + \frac{\pi d^2}{4 T} \times \frac{F_s}{F_t} + d$$

$$= 4 \times \frac{7}{8} \times \frac{95000}{55000} + \frac{3.1416 \times (7/8)^2}{4 \times 7/16} \times \frac{45000}{55000} + 7/8 = 8.04 \text{ inches,}$$

which is the greatest allowable pitch at the outer row of rivets.

One method of finding how the joint will fail is to make up equations involving those resistances only. Thus a rivet at the outer row may fail by shearing or by crushing at the cover plate, which is here made thinner than the shell plate.

Equating the resistance of the two methods, we have:

$$\frac{\pi d^2}{4} F_s = T_c d F_c, \text{ or for a cover plate } 3/8 \text{ inches thick,}$$

$$d = \frac{3.1416}{4 \times 3/8} \times \frac{95000}{45000} = 1.01 \text{ inches.}$$

A rivet 1.01 inches in diameter will consequently be just as likely to fail by crushing as by shearing.

But the resistance to shearing increases as the square of the diameter, while the resistance to crushing increases as the diameter.

Therefore a rivet larger than 1.01 inches in diameter will fail by crushing while a smaller rivet will fail by shearing.

A similar calculation at the inner row where the rivet bears against a cover plate both inside and outside, and will consequently crush against the shell plate, gives:

$$\frac{2 \pi d^2}{4} F_s = T d F_c$$

$$d = \frac{2 T}{\pi} \times \frac{F_c}{F_s} = \frac{2 \times 7/16}{3.1416} \times \frac{95000}{45000} = .6 \text{ inches}$$

Here a rivet larger than .6 will crush and one smaller will shear.

It is now evident that a 7/8 inch will shear at the outer row and will crush at the inner row.

For this joint, failure will occur by the method D, but not by the methods B or E. Then equating A and D and solving for P, gives the pitch at the outer row as 8.04 inches.

The corresponding pitch of the outer cover plate is 4.02 inches.

The efficiency of the joint is

$$100 \frac{P - d}{P} = 100 \frac{8.04 - 7/8}{8.04} = 89.1 \%$$

RING SEAM.

The stress on a transverse section of a hollow cylinder from internal fluid pressure is one-half the stress on a longitudinal section.

Single riveted ring seams are the ones that are most commonly used.

In practice it is found that ring seams of horizontal externally fired boilers may have a pitch of about 2-3/16" for all thickness of plate from 1/4 to 1/2 of an inch. The diameters of rivets for such seams may be made about the size given in the following table:-

Thickness of plate 1/4, 5/16, 3/8, 7/16, 1/2

Diameter of rivet 5/8, 11/16, 3/4, 7/8, 7/8

The ring seams' circumference is;

$$3.1416 \times 58 = 182.2 \text{ inches,}$$

which allows 82 rivets with a pitch of 2.22 inches to be used.

This joint will fail by shearing the rivets.

The efficiency of the joint is the ratio of the resistance of a single rivet to shearing, to the resistance of a strip of plate as wide as the pitch.

The efficiency of the joint is:

$$\frac{\pi d^2 F_s}{P T F_t} = \frac{3.1416 \times (7/8)^2 \times 45000}{2.22 \times 7/16 \times 55000} = .505$$

This efficiency is more than half the efficiency of the longitudinal seam, and will therefore be sufficiently strong.

LAP.

It is customary to calculate the width of lap required on the assumption that the metal between the rivet and the edge of the plate may be treated as a beam of uniform depth, fixed at the ends and loaded uniformly by the force which would be required to shear or crush the rivet, taking of course, the larger.

The width of the beam is the thickness of the plate; the depth is the distance from the edge of the hole to the edge of the plate, and the length is the diameter of the rivet.

Rivets in single riveted seams fail by shearing. The load is consequently the shearing resistance, which is

$$\frac{\pi d^2}{4} F_s$$

The maximum bending moment for a beam of uniform cross section, fixed at both ends and loaded uniformly is equal to the load multiplied by 1/8 the span. The moment of resistance is equal to

$$F \frac{I}{C},$$

in which F is the cross-breaking strength (about 55000 pounds), I is the moment of inertia of the section, and C is the distance of the

most strained fibre from the neutral axis. Here

$I = \frac{T H^3}{12}$, $C = \frac{H}{2}$, representing the distance from the edge of the hole to the edge of the plate by H .

Equating the bending moment to the moment of resistance,

$$\frac{1}{8} d \times \frac{\pi d^3}{4} \times F_s = F \frac{T H^3}{6}$$

$$H = \sqrt{\frac{3 \pi d^3}{16 T} \times \frac{F_s}{F}} = \sqrt{\frac{3 \times 3.1416 \times 343}{16 \times 7/16 \times 512} \times \frac{45000}{5000}} = .858 \text{ inches}$$

for the case in hand. The lap is consequently;

$.858 + 1/2 \times 7/8 = 1.295$ inches for the ring seam,; I took it as $1-5/32$ inches.

A similar calculation for the cover plate with the same diameter of rivet, but with a plate $3/8$ inches thick gives:-

$$H = \sqrt{\frac{3 \pi d^3}{16 T} \times \frac{F_s}{F}} = \sqrt{\frac{3 \times 3.1416 \times 343 \times 45000}{16 \times 3/8 \times 512 \times 55000}} = .92 \text{ inches.}$$

The log is; $.92 + 1/2 \times 7/8 = 1.357$ inches;

I took $1-11/32$ inches.

The rivets of the inner rows pass thru both cover plates and are in double shear, and fail by crushing as has been shown.

The load to be used for calculating the lap is therefore the resistance to crushing in front of the rivet which is:-

$$T d F_c.$$

The equation of bending moment and moment of resistance gives:

$$\frac{1}{8} d \times T d F_c = F \frac{T H^3}{6}$$

$$H = d \sqrt{\frac{3 F_c}{4 F}} = 7/8 \sqrt{\frac{3 \times 95000}{4 \times 55000}} = .988$$

The lap is consequently,

$.988 + .437 = 1.425 = 1-7/16$ inches nearly.

AREA OF UPTAKE.

The area of the uptake, like the total area thru the tubes is made from $1/7$ to $1/8$ of the grate area. The area thru the tubes

was found to be 460 square inches.

The uptake may be made 12 inches deep, measured from front to rear. It will then be,

$460 \div 12 = 38.3$ inches wide, measured transversely.

The opening thru the top of the projecting shell at the front end will be made 12 inches deep and must be cut down until it is 38.3 inches wide.

The projecting end of the shell is made long enough so that a space of about 1 inch is left between the uptake and the calking edge of the front tube sheet.

STAYS.

See Figure.

The method of staying selected consists of channel-bars riveted to the head, and supported by thru stays; the upper channel bar is assisted by an angle of iron. The channel-bars selected are six inches wide, and the horizontal rows of rivets in each bar are 3-1/4 inches apart, which brings them as near the flanges of the bar as they can be driven.

The middle of the lower channel-bar is 6-3/4 inches above the top of the tubes, so that the lowest row of rivets is

$6-3/4 - 1/2 \times 3-1/4 = 5-1/8$ inches above the top row of tubes.

But the plate cannot be properly considered to be rigidly supported at a line drawn thru the tops of the tubes; assume the line of support to be 1/4 of the diameter lower down, this makes a space of 3-3/4 inches.

The upper channel-bar is placed 8 inches above the lower one, so that the stay rods are

$29 - (6 + 6 + 8) = 9$ inches below the shell. If the upper rods are much less than 10 inches from the shell, access to the boiler will be difficult.

The space immediately above the upper channel-bar is stayed by aid of an angle-iron which is riveted to the channel-bar.

The distance of the lower row of rivets in the upper channel-bar above the upper row in the lower bar is:

$$8 - 3\frac{1}{4} = 4\frac{3}{4} \text{ inches.}$$

The top row of rivets in the angle iron is

$$4\frac{5}{8} \text{ inches}$$

from the top of the boiler or $3\frac{1}{8}$ inches from the dotted boundary line.

LOWER STAY RODS.

In order to determine the load carried by the lower stay rods, we will assume that half the load on the plate between the lowest row of rivets and the top row of tubes is carried by the rivets, and that the load on the plate between the channel-bars is divided equally between them.

I assumed that the line of support at the tubes is a quarter of their diameter below their tops, and have found this line to be $4\frac{3}{4}$ inches below the lowest row of rivets.

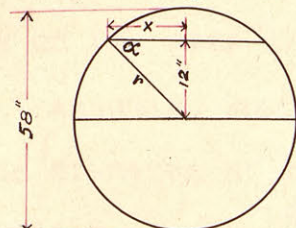
$$\frac{1}{2} \times 4\frac{3}{4} = 2\frac{3}{8} \text{ inches}$$

The distance apart of the two rows of rivets in the channel-bar is $3\frac{1}{4}$ inches.

The total width of plate supported by the channel-bar may therefore be considered to be,

$$\begin{aligned} 3\frac{1}{4} + 5\frac{1}{8} \times \frac{1}{2} + 5\frac{1}{8} \times \frac{1}{2} \\ = 8.37 \text{ inches.} \end{aligned}$$

The length of the lower channel-bar at the middle is



$$\sin \alpha = \frac{12}{29} = .4137 = 24^\circ - 26'$$

$$\cos \alpha = \cos 24^\circ - 26' = \frac{x}{29}$$

$$x = 29 \times .9104 = 26.4 \text{ inches}$$

which equals one-half the length of a line passing thru the centre of the channel-bar and cutting the circumference of the shell.

The total length of the line is 52.8 inches.

The length of this line to the dotted inside circle is

$$52.8 - 2.5 = 50.3 \text{ inches. Therefore,}$$

the length of the channel-bar at the middle is about 50 inches. It is convenient to space the rods 13 inches apart, which leads to an assumed length of 52 inches.

The load on the lower channel-bar is considered to be

$$100 \times 8.37 \times 52 = 43524 \text{ pounds.}$$

Treating the channel-bar as a continuous beam with four equal spaces, and five points of support, of which three are at the stay rods, and two are at the shell of the boiler.

By the theory of continuous beams a uniform load on the channel-bar would be distributed among the five points of support as follows:-

At each point of support at the shell 11/112,

At each outer stay-rod 32/112,

At the middle stay-rod 26/112.

This would bring on each of the outer stay-rods

$$\frac{32}{112} \times 43524 = 12435 \text{ pounds.}$$

Now the load is not uniformly distributed, but is carried in part by the rivets and in part by the nuts and thick washers on the stay-rods; but the actual will bring a less load

on the two outer stays, so that the assumption of the load just found is on the side of safety, and is easily calculated.

Assuming 9000 pounds for the working stress in the stay rods, the diameter of the rod may be calculated by the equation;

$$\frac{\pi d^3}{4} = \frac{12435}{9000}, \quad d^3 = \frac{4 \times 12435}{3.1416 \times 9000} = 1.7591. d = 1.326 \text{ which is}$$

nearly $1-11/32$ inches.

For simplicity the five rods will be the same size, namely, $1-3/8$ inches.

The minimum size of upset end is $1-3/4$ inches, but make it 2 inches.

LOWER CHANNEL-BAR.

The determination of the actual stresses in the channel-bar, allowing for the effect of the nuts and thick washers on the stay rods is very uncertain.

The application of the theory of continuous beams with uniform load may not give us a stress as large as the actual maximum stress, an approximate method will have to be used, which will give a stress at least as great as the greatest stress in the bar.

These lines a b and c d are drawn at $1/4$ of the diameter of the thick washers from the centre of the rod, or at

$$1/4 \times 5-1/2 = 1-5/8 \text{ inches.}$$

Further assume the load on the rivets A & B is due to the pressure of the steam on the area e f g h, bounded by lines drawn half way between them and the nearest point of support.

Thus e g is half way between the rivets and the line a b, g h is half way between the rivets and the line of support at the upper row of tubes, e f is half way between the channel-bars, f h is half way to the next pair of rivets. The rivets are $4-3/4$ inches from the nearest stay rods, and are

$$4 - \frac{3}{4} - 1 - \frac{3}{8} = 3 - \frac{3}{8} \text{ inches from the line}$$

a b; half of this is $1 - \frac{11}{16}$ inches, the two pairs of rivets are

$$13 - \frac{1}{2} - 2 \times 4 - \frac{3}{4} = 4 \text{ inches apart, half}$$

of this is 2 inches. The area e, f, g, h, is

$$(1 - \frac{11}{16} + 2) \times 8 = 29 - \frac{1}{2} \text{ square inches,}$$

and the steam pressure on that area is

$$29 - \frac{1}{2} \times 100 = 2950 \text{ pounds.}$$

This is the load due to each pair of rivets between a pair of stay rods; and since the rivets are symmetrically placed, this is also the supporting force of each end of the beam.

Between the two pairs of rivets the beam is subjected to a uniform bending moment, equal to the load on a pair of rivets multiplied by their distance from the end of the beam; the bending moment is

$$2950 \times 3 - \frac{3}{8} = 9856 \text{ pounds.}$$

The theory of beams given

$$M = F \frac{I}{C}$$

M = bending moment, I = moment of inertia of any section of the beam, C = the distance of most strained fibre from neutral axis, and F is the stress at that fibre, for rolled-steel channel-bars,

F may be taken equal to 16000 pounds, we have

$$\frac{I}{C} = \frac{9857}{16000} = .616 \text{ inches.}$$

Now I and C depend on the form and size of the section of the beam, and conversely, the size and form of beam required may be determined from them. But as the upper channel-bar is exposed to a greater bending moment and consequently must have a larger section than is required for the lower bar, but I will make both of them the same size.

UPPER STAY-RODS.

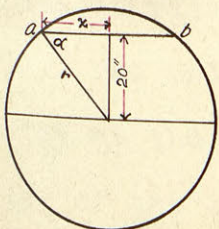
The flat surface of the boiler head above the lower channel-bar is supported by the upper channel-bar aided by the angle iron which is firmly riveted to it, and which will be assumed to act with, and form a part of the channel-bar.

Following the general convention that the pressure on a portion of the head between two lines of support is divided equally between them, assuming that the load on the upper channel-bar is due to the steam-pressure on an area bounded at the bottom by a line half way between the upper and lower channel-bars, and at the top by an arc 3-1/4 inches inside the boiler shell. In the figure, half of this area is represented by $j k l$; the area $j k l$ being about half way between the root of the flange, shown by the outer dotted boundary line and the adjacent rivets. In place of the area $j k l$ a rectangular area $l m n o$, bounded at the end by a line at the middle of the end of the channel-bar, and at the top by a line $m n$, so chosen as to make the rectangular area larger than the area it replaces.

The width of this area, $l m$ is 9-1/4 inches, so that the load per inch of length is

$$9\text{-}1/4 \times 100 = 925 \text{ pounds.}$$

The upper channel-bar may be assimilated to a continuous beam with three unequal spaces, and the middle span between the stay rods is 15-1/2 inches, and the end spans between the stay rods and the roots of the flange of the head are each 11-1/2 inches.



$$\sin \alpha = \frac{20}{29} = .6893 = 43^\circ - 34' - 30''$$

$$\cos \alpha = \cos 43^\circ - 34' - 30'' = .7246$$

$$\cos \alpha = \frac{x}{29}, \quad x = 29 \cos \alpha = 29 \times .7246$$

$$= 21.01 \text{ inches.}$$

The total length of the line a b from one side of the shell to the other is;

$$21.01 \times 2 = 42.02 \text{ inches}$$

$42.02 - 2.5 = 39.52$ inches gives the mean length of the upper channel-bar. This makes the end spans nearly $\frac{3}{4}$ of the middle span.

Now a continuous beam uniformly loaded with w pounds per inch of length, which has a middle span l inches long, and two end spans $\frac{3}{4} l$ inches long, will have for the end supporting forces

$$\frac{233}{864} w l, \text{ and for the middle supporting forces } \frac{847}{864} w l.$$

The end supporting forces are provided by the shell which is well able to carry them. The stay rods which furnish the middle supporting forces, must each carry,

$$\frac{847}{864} \times 15.5 \times 925 = 14055 \text{ pounds.}$$

Assuming 9000 pounds per square inch as a working stress for the stay, the area for the stay is,

$$14055 \div 9000 = 1.56 \text{ square inches.}$$

This is nearly $1\text{-}\frac{3}{16}$ inches. But I make them all the same size, and also $1\text{-}\frac{3}{8}$ inches in diameter. For the upset end the minimum diameter is $1\text{-}\frac{3}{4}$ ", but I will make it two inches.

UPPER CHANNEL- BAR.

The calculation of the stress in the upper channel-bar will be made by an extension of the same approximate method used with the lower channel-bar. Since the middle span is wider than the end span; it will be sufficient to make a calculation for it only.

The calculation is made the same as for a simple beam supported at the ends, the joints of the support being $\frac{1}{4}$ the diameter of the thick washer from the middle stay-rod that is at the distance of $1\text{-}\frac{5}{8}$ inches from the stay rod.

The distance between the upper stay-rods is 15.5 inches, so the span of the beam is,

$$15-1/2 - 2 \times 1-5/8 = 12.75 \text{ inches.}$$

The beam is assumed to be loaded with concentrated loads applied at the rivets C, D, E, F, G, and H. The load on the rivet I is assumed to be carried by the stay-rod directly, and is not included in this calculation. The pair of rivets, D, E and the several rivets C, G, and H are assumed to carry the load due to the pressures on the areas marked off by the dotted lines, each line being drawn half-way between adjacent supporting points, except that the arc at the top is drawn 3-1/4 inches from the shell as mentioned once before. Taking as the bending moment at the middle of the beam, 37390 pounds. Taking as with the lower channel-bar, a working stress of 16000 pounds, we have

$$37390 \times \frac{16000}{C}, \text{ or } \frac{I}{C} = 2.17$$

The moment of resistance of the channel-bar 6 x 2-1/2 x 1/2 inch is 1.08, and the moment of resistance of the 3-1/2 x 3 inch angle iron is 1.55; the sum is equal to 2.63 which is larger than the required moment of resistance given above. These are the forms used.

BRACKETS.

The boiler is supported on four cast iron brackets, each of which is 10 inches wide in the direction of the length of the boiler, 15-1/2 inches long measured circumferentially. Each bracket is riveted to the shell by nine rivets 15/16 of an inch in diameter. The brackets are made wide and long in order that the local strains due to carrying the weight of the boiler may not be excessive. The brackets are set above the middle line of the boiler so that the flanges may be protected by brick work. In the case in

hand they are 3-1/2 inches above the middle line. The brackets are arranged so that the weight of the boiler and accessories is equally divided among them, and so that there is as little bending moment as possible on the shell of the boiler. When four brackets are used they may be somewhat less than a fourth of the length of a tube, from the tube plates.

The load on the brackets may be estimated by calculating the weight of the boiler when entirely full of water, and adding the weight of all parts that are supported by the boiler, such as pipes, valves, and brick work, or covering that may rest on the boiler. One-fourth of this load is assigned to each bracket.

This load on a bracket should be uniformly distributed over the bearing-surface of the flange, which is commonly 8 or 9 inches wide. But to guard against the effect of unequal bearing, it is well to assume the bracket to bear near the outer edge, say two inches from the edge. This bearing force tends to rotate the bracket about its upper edge, and this tendency is resisted by the rivets under the flange, which must be large enough to resist the resulting pull on them. The other rivets are added to give sufficient resistance to shearing all the rivets. There are seldom less than nine rivets in a bracket, all as large as those below the flange, even though fewer would suffice. The bracket is usually made of cast iron, and the dimensions are commonly controlled as much by the conditions required for a sound casting as by calculations for strength. The strength may be calculated by treating it as a cantilever and allowing for the strength of the web.