THE BOILER

By STEPHEN CHRISTIE

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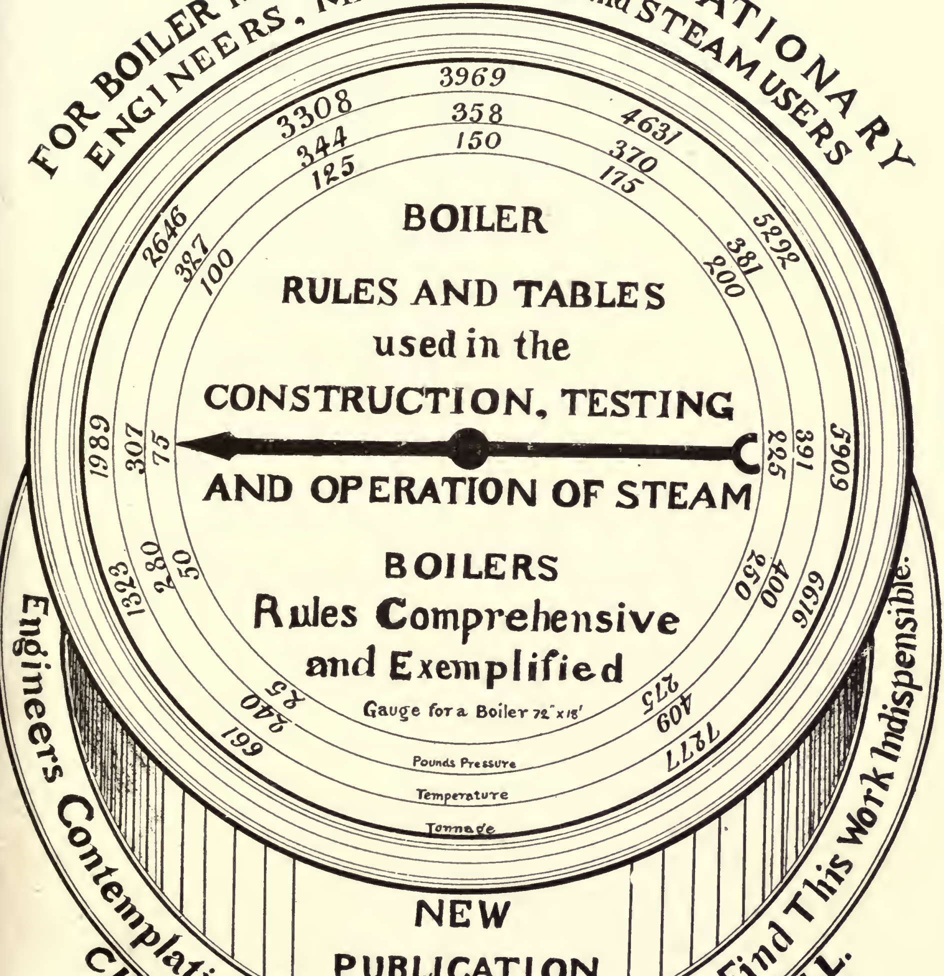
FOR BOILER MAKERS, MARINE and STATIONARY ENGINEERS, MACHINISTS and STEAM USERS



BOILER

RULES AND TABLES used in the CONSTRUCTION, TESTING AND OPERATION OF STEAM BOILERS

Rules Comprehensive and Exemplified

Gauge for a Boiler 72" x 18'

NEW PUBLICATION

1908

Engineers Contemplating Taking Examinations Will Find This Work Indispensible.

Stephen Christie

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PREFACE.

THE writer, after many years of experience in connection with boilers, as a boiler maker, master boiler maker, and boiler inspector, has, in his vocation, found it necessary to use rules, tables and formulas in conjunction with his work and duties and has profited by those of older and wider experience in the craft and, having had ample opportunity, inclination and resource for research for comprehensive, concise and condensed formulas and rules governing his daily duties, has compiled this work.

The author does not claim originality; it is the intention to make the subject as clear as possible, to make it a pleasant study so that the layman can master the many rules that may seem too intricate and attention has been given to the most practical part of estimating values in connection with steam boiler designing.

Many valuable and scientific books have treated the subject of steam boilers and some exhaustively and from them I have learned. I have quoted from those authors' fund of information and from personal experience, and it will be my aim to make this compilation clear and free from any technicalities that would in a measure confuse the student and sincerely hope it may accomplish the mission intended, to interest those whose duties, labors and interests are in connection with the steam boiler.

STEPHEN CHRISTIE.

CHAPTER I.

MATERIALS.

It has been stated by historians that Tubal Cain was an iron worker, no doubt an artificer in plow shares and pruning hooks, but that in remote antiquity, when metals were few in number and knowledge of their uses limited, and it is doubtful if the steam boiler was among the articles made.

Historians record the nature of metals during those early ages as gold, silver, brass, iron, tin and lead, and also state that bronze had been in use before iron, thus we may favor doubt about boilers of some description being in use during those ages of antiquity.

Aristotle seems to be the earliest authority quoted on the subject of iron, saying "that iron was purified from acoria by melting, and after repeated treatments by melting became purified." What state of purification in relation to iron working tools or metals was not stated.

Daimachus, an early writer on the subject mentions different kinds of steel and the purposes to which they were used, and severally suited, viz.:

Chalybdie for carpenter tools.

Lacedoemonian for files and drills and stone cutters' tools.

Lydian for knives and razors.

Thus ancient history records some notice of materials used in boiler construction, but it is doubtful if ancient process of manufacturers or knowledge of material construction brought it up to anything like the state of perfection that could be used in steam boilers of today.

This chapter was not intended to treat on metallurgy only to touch upon materials as now used in these days of high pressure boilers.

Manufacturers assume great responsibilities in selecting material for boilers, hence care in selection.

Boiler making today is a science, demanding scientific education and knowledge gained by research, investigation and reasoning.

The writer can go back mentally to the days when boiler making was apparently in its infancy, this when comparing the boilers of today with the demands for power and when the very low pressures were then well suited to the low grade material manufactured; designs crude, seams out of all proportions, bracing out of reasoning, and the ignorant mechanic, whose only evidence of work was strong in arm, wrought defects without thought of effects.

There is evolution and revolution in boiler making today.

High pressures are necessary, also care in selecting materials and designing boilers. The construction for the demands today are high pressures; due to competition, economy and fuel and space. It is necessary then to have all parts equal in strength, different parts favored with material of specific quality, such as braces, tubes, fire sheets, where circulation is least; corrosion, expansion, contraction or pitting active will necessitate increased thickness of plate; again, to secure complete circulation, combustion of fuel, etc.; to arrange heating surface in proportion to grate area and steam space, to make the form of boiler such that it can be constructed without mechanical difficulty or great expense.

Designs must be made to give strength, durability under the action of hot gases and corrosive elements, to be accessible, for cleaning, repairing and to provide safety appliance of ample proportions and applied properly. Thus the necessity of the greater education in boiler designing and construction and knowledge of material used.

Material for boiler purposes as well as other uses invariably contains in combination some proportion of various elements, and although these may appear small, have very marked influence upon its strength, ductility and working qualities, thus making it necessary to have both chemical as well as physical tests. In the manufacturing of boiler material the process of carburization changes the nature and properties of contained carbon, thus wrought iron contains from 5 per cent to only a trace per cent of carbon, and steel including all kinds of iron contains not more than 1.75 per cent of carbon and varies in fusibility, hardness, susceptibility to tempering and malleability. The first two properties being increased by increase of carbon, while the others are diminished.

All ores go through the process of reduction, and the more impurities they contain the greater amount of work is necessary to treat them; these include carbonic acid, water, combustible and earthy matter.

CAST IRON.

In cast iron these qualities looked for are taken from the fuel and mode of smelting, this materially as much as the character of ore. To convert cast iron into bar, forged or malleable iron, it has to be refined by smelting with coke or charcoal; this process eliminates the oxygen and carbon which may be left, thus bringing it to a state of refined metal, this is forged under hammer, passed through roll and drawn into bars, cut in lengths and formed into bundles or piles, again reheated and once more hammered and rolled into any shape. Cast iron has in its makeup carbon-silicon; this is a slag and its presence makes iron and steel hard and brittle, but up to 6 per cent is harmless providing 3 per cent. of manganese is present with it. Manganese, of which 5 per cent is sufficient to make iron cold short, is valuable in iron to be converted into steel.

Sulphur and phosphorus, when 8 per cent is present, make iron and steel crystallized and unfits it for plate for boiler purposes.

Arsenic increases the hardness in steel at the expense of toughness, as does carbon with it in form of graphite. The gray iron contains most graphite and carbon, making it more fusible and softer than white iron. The latter contains more combined carbon; these constituents vary, thus having various influence on the mechanical properties, and, after repeated fusings, loses its carbon.

THE ELEMENTS IN CAST IRON ARE AS FOLLOWS:

ELEMENTS.	PERCENTAGE.
Combined carbon	.15 to 1.25 per cent
Graphite	1.85 to 3.25 " "
Silicon	.15 to 5. " "
Sulphur	0 to .05 " "
Phosphorus	0 to 1.3 " "
Manganese	0 to 1.5 " "
Iron	90. to 95. " "

Cast iron is not reliable for boiler construction unless for very low pressure, while it resists corrosion it is brittle and to get strength great thickness is necessary.

From cast iron to steel, plate is susceptible to the widest variation in its character; cast iron as extracted from ore, is melted with comparative facility and according to mode of operation in foundry, may be rendered so hard that it requires special tools to work it.

This metal by treatment with heat and air is converted into great tensile strength and ductility, still soft and easily worked into shapes without fracture.

The difference in molecular construction between cast and malleable iron is, the cast iron contains a larger proportion of carbon and some silicon, the malleable iron practically none—thus to produce steel the cast iron is melted first, then wrought iron and steel scraps are added by degrees (these in equal proportion), then an addition of spiegeleisen is added with manganese; as soon as this metal ceases to flow it is removed and poured into moulds, reheated and rolled into plate.

WROUGHT IRON.

Wrought iron is made by the process called puddling to eliminate the graphite and combined carbon from the pig iron, leaving sufficient to give strength in this new combination. In operation the mass is heated and kneaded by the paddles into blooms, and these are compressed under a hammer to remove the slag, again heated, rolled out and further squeezed by passing through rolls, thus forming a puddle bar. These bars are broken up and worked by hammering and rolling more or less according to degree of purity and strength required, thus iron plates retain the fibrous quality imparted to the bar, but owing to the secretion of cinder scale between the layers (thus producing blisters), careful tests are necessary by eye or hammer.

Wrought iron, while possessing great tenacity combined with toughness and ductility is well adapted to resist sudden strains.

While the puddle bars are going through the rolls oxide of iron is formed in scales, caused by the hot iron coming in contact with the air; these scales are collected for the puddling furnace, with use being that of absorbing the carbon from the iron.

The wrought iron is Lamina in its construction, is ductile and has a tensile strength varying up to 55,000 pounds per square inch and a ductility to 40 per cent; its uses in boiler construction are in tubes, rivets, braces and for reinforcement. One objectional feature in iron plates is the smallness of plate that can be manufactured without chance of blistering or lamination; another is the excess of labor due to more seams, thus reducing the strength of boiler.

The great advantages steel has over wrought iron are, plate can be made in sizes of larger dimensions, boilers can be made of lighter material, greater power of conductivity of heat can be secured, but it necessitates greater care in flanging the material and in fitting up.

MATERIAL.

Average crushing and breaking strains of iron and steel:

Breaking strain of wrought iron	23 tons
Crushing " " " "	17 "
Breaking strain of cast iron	7½ "
Crushing " " " "	50 "
Breaking strain of steel bars	55 "
Crushing " " " "	110 "

this per square inch of section.

STEEL PLATE.

Steel is a carburet of iron and the earliest invention of same was prepared by fusion and not by cementation; in this later process the metal is surrounded by charcoal, and thus it draws its supply of carbon, the molecules of iron taking up the latter.

Since that early process there have been several methods employed to produce the steel, viz.:

1st Direct from ores.
2nd By addition of carbon and malleable iron.
3rd By the partial decarburization of pig iron.
4th By diluting the carbon in pig iron and the addition of malleable iron.

Steel plate is termed mild steel, low steel and high steel, which contains a high percentage of carbon. The following table will show the proportion of carbon and corresponding hardness:

NO. OF HARDNESS.	PER CENT OF CARBON.	OBSERVATION.
1	1.58 to 1.38	cannot be welded.
2	1.38 to 1.12	welds easily and used for chisels.
3	1.12 to .88	used for cutting tools.
4	.88 to .62	mild steel for tires, etc.
5	.62 to .38 and	tempers slightly, steel for boiler plates.
6	.38 to .15	
7	.15 to .05	does not temper, used for machinery.

Steel and iron, like all other metals, are composed of atoms grouped in molecules, and any force that alters the relations of the atoms in the molecules modifies the physical properties of the metal, thus in heating, cooling and crushing the physical properties of metals vary with its degree of purity.

Density of a metal is dependent on the intimacy of the contact between the molecules and is influenced by temperature and rate of cooling; its density can be augmented by hammering or any compressing stress; pressure on all sides increases its density.

Malleability is the property of permanently extending in all directions without rupture by pressure produced by slow stress or by impact.

Ductility is the property that enables metal to be worked into flanges or drawn into wire, and this ductility increases with increased temperature.

Tenacity is a property possessed by metals in varying degree, it is the resisting, the separating of the molecules after the limit of elasticity has passed.

Hardness is the resistance offered by the molecules of a substance to their separation by penetrating action of another substance.

Brittleness is the sudden interruption of molecules, cohesion, when substances are subjected to the action of some extraneous force, such as a blow or change of temperature and largely influenced by purity of metal.

Elasticity is the power a body possesses of resuming its original form after removal of an external force which has changed its form, and to measure the strength of metals it is necessary to determine:

First.—The greatest stress the metal can sustain within the limits of elasticity.

Second.—The total exent of strain before rupture takes place.

Third.—The ultimate tensile strength or maximum stress the metals can sustain without rupture.

The difference between steel and iron is seen when subjected to a high temperature and suddenly cooled by plunging in cold water. The iron is affected very little while the steel becomes hardened.

A chemical test to distinguish iron from steel is by placing a drop of diluted nitric acid upon a clean surface of the metal; a greenish-gray stain appears upon iron; on the steel a black spot, this latter is due to the separation of carbon.

The processes of making boiler plate are the Siemens-Martin or open hearth process, and by the Bessemer converter. The latter is costly. The former offers better facilities for testing the quality while still in a molten state and its character modified at will by addition of such material required to produce desired results. While the Bessemer process is not as desirable owing to its not offering facilities for testing or adjustment. The elements that increase tensile strength will reduce ductility, as carbon increases strength up to a certain limit then beyond excess reduces it, as a certain limit separates steel from cast iron.

The hardening elements are carbon, silicon, manganese and phosphorus.

Manganese steel contains a high percentage of the latter, having a little carbon and is avoided in boiler construction.

The qualities in steel for boilers are homogenity, tenacity, elasticity and ductility; distinct from steel used for other purposes boiler plate should be tough and not of such a character that it might harden under the action of sudden great changes of temperature.

Steel is structural and chemical, it is a compound or an alloy of elements, silver, tungsten, chromium, titanium, silicon and cyanogen. It forms an intermediate link between ordinary cast iron and wrought iron, uniting with the properties of both and its distinguishness or characteristic is its capability of being hardened or softened by rapid or slow cooling.

TABLE SHOWING COMPARISONS OF IRON AND STEEL:

	IRON.		STEEL.	
	SWEDISH.	PENN.	MILD.	VERY MILD.
Carbon	.087	.067	.238	.009
Silicon	.56	.020	.105	.163
Sulphur	.005	.001	.012	.009
Phosphorus	.....	.075	.034	.084
Manganese	.....	.009	.184	.620
Iron	99.220	99.828	99.427	99.115

U. S. GOVERNMENT SPECIFICATIONS FOR MATERIAL.

Fire-box steel should show a tensile strength of not less than 52,000 pounds, and not over 62,000 pounds per square inch, an elastic limit not less than one-half (½) the ultimate strength, elongation 25 per cent and tested as follows: Cold and quench bends

180 degrees flat on itself without fracture on outside of bent portion, not over .04 per cent of sulphur or .04 per cent phosphorus.

Flange steel to show a tensile strength of from 55,000 to 65,000 pounds per square inch, elastic limit not less than one-half of its ultimate strength, elongation 25 per cent, cold and quench bends 180 degrees flat on itself, without fracture on one side of bent portion and not over .04 per cent of phosphorus and not over .05 per cent of sulphur.

Extra soft steel to show a tensile strength of 45,000 to 55,C00 pounds per square inch, elastic limit not less than one-half its ultimate strength, elongation 28 per cent, cold and quench bends 180 degrees flat on itself without fracture on outside of bent portion, not over .04 per cent of sulphur or phosphorus.

Plates and steel rivets to be made by the open hearth process and tests to be made to determine tensile strength, ductility, elasticity, elongation; physical and chemical tests to be made at place of manufacture, all plates to be plainly stamped at corner near center. Material for stay bolts and braces to have a tensile strength of not less than 46,000 pounds per square inch when made of iron and not less than 55,000 pounds when made of steel.

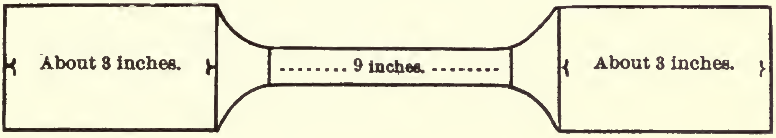
Steel rivet material to have a tensile strength of 50,000 to 60,000 pounds per square inch of sectional area and elastic limit not less than one-half the ultimate strength, a bending test as follows at 180 degrees flat on itself without fracture on outside portion; elongation 26 per cent.

Iron rivet material to have a tensile strength of 40,000 pounds per square inch.

SPECIFICATION AND TESTING OF MATERIALS.

The U. S. Government rules as specified for the construction of boilers coming under federal supervision are as follows:

"That iron or steel plate intended for construction of boiler to be used in steam vessels shall be stamped in at least five different places by the manufacturer at place where made, viz., at corners about eight inches from edges and near center and with number of pounds per square inch of tensile strength; it will be the sectional inch and which must not be less than 45,000 pounds for iron or 50,000 pounds for steel; from plates shall be taken coupons and prepared, by plaining edges, these test pieces shall be at least 16 inches in length and from one and one-half (1½) inches to three and one-half (3½) inches in width at ends, which ends shall join by an easy fillet, a straight in the center of at least 9 inches in length and 1 inch in width, in form to the following diagram marked with light prick punch marks at distances one inch apart, spaced so as to give 8 inches in length."



The strain necessary to break the test pieces as described is taken as the proportion of the T S (tensile strength) per square inch.

EXAMPLES.

Test piece or coupon reduced to smallest part is one-fourth of a square inch and is broken at 15,000 pounds.

15000
4

60000 TS per square inch

To determine the elongation, the part cut out in test piece marked at inch sections and the force necessary to break it asunder is the proportionate part of the T S per square inch, and distance stretched represents percentage of elongation.

EXAMPLE.

To find percentage of elongation in a test piece. Coupon 8″ before testing, elongated to 10½.

10.5 = 10½″ after testing
8 = before testing

10.5) 2.500 (23 per cent of elongation
2 10

400
315

85

Test piece 1 5/8 x 3/8 breaks at 34,000 pounds.

```
         1.625
          .375
         -----
          8125
        11375
        4875
      -------
      .609375)34000,0000(55829 lbs. TS
              3045
              ----
               3550
               3045
               ----
                5050
                4872
                ----
                 1780
                 1218
                 ----
                  5620
                  5481
                  ----
                   139
```

Strain necessary to break a test piece is the proportionate part of the tensile strength per square inch.

A piece of plate sectional area .5 square inch breaks at 30,000 pounds.

```
.5000)30000,0000(60000 lbs. TS
      300000
      ------
           000
```

TABLE.

Showing width of plate expressed in 100th of an inch that will equal one quarter of one square inch of section of the various thickness of plate.

Example.—If plate is ¼ inch in thickness the width should be 100th of an inch wide to equal one quarter of one square inch of section or as follows:

$\frac{3}{16}$ × 133	$\frac{5}{16}$ × 80
.21 × 119	.33 × 76
.23 × 109	.35 × 71
¼ × 100	$\frac{3}{8}$ × 67
.26 × 96	$\frac{7}{16}$ × 57
.29 × 86	½ × 50

Only steel plates manufactured by what is known as the basic or acid open hearth process will be allowed to be used in the construction of boilers for marine purposes and manufacturer shall furnish a certificate with each order of steel tested stating technical process by which said steel was manufactured, this is not intended to apply to plates used in construction of Bessemer steel tubes.

No plate made by acid process shall contain more than 0.06 per cent of phosphorus or 0.04 per cent of sulphur, and no plate made by the basic process shall contain more than .04 per cent of sulphur or phosphorus. This to be determined by analysis by the manufacturer.

Steel plates must have a tensile strength not less than 55,000 pounds and not over 75,000 pounds per square inch of section, but boilers whose construction is commenced after June 30, 1905, where plate will come in contact with fire either in use or in course of construction of the boiler the tensile strength shall not be more than 70,000 pounds per square inch of section.

No plate shall be stamped with a greater tensile strength than 70,000.

Elongation shall show at least 25 per cent in a length of 2 inches for thickness to one-fourth (¼) inclusive in a length of 4 inches for over one-fourth to seven-sixteenths inch, inclusive; in a length of 6 inches for all plates over seven-sixteenths inch. The sample must show a reduction of sectional area as follows:

At least 50 per cent for thickness over one-half to three-fourths inch inclusive, 45 per cent for thickness over one-half to three-fourths inclusive, and 32.5 per cent for thickness over three-fourths of an inch.

Quenching and bending test pieces shall be at least 12 inches in length and from 1 to 3½ inches in width. The sides where sheared or planed must not be rounded, but the edges may have the sharpness taken off with a fine file. The test piece shall be heated to a cherry red (as seen in a dark place) and then plunged into water at a temperature of about 82 degrees F. Thus prepared the sample shall be bent to a curve, the inner radius of which is not greater than one and one-half times the thickness of the sample without cracks or flaws, the ends must be parallel after bending.

Iron plates when tested must show a tensile strength of not less than 45,000 pounds and not over 60,000 pounds per square inch of sectional area and show an elongation of at least 15 per cent in a length of 8 inches and a reduction of area as follows: For plate having 45,000 T S 15 per cent, and for each additional 1,000 pounds up to 55,000 add 1 per cent; for samples over 55,000 pounds up to 60,000 T S 25 per cent shall be required; a bending test as follows: a piece 12 inches in length and from 1 to 3½ inches in width, the edge not to be rounded, then bent cold to an angle of 90 degrees to a curve the inner radius of which no greater than one and one-half times the thickness of the sample without cracks or flaws."

The chemical or analytical test is for the purpose to show right proportions of elements and properties useful in the material's make-up, for specific purposes, and if free from those whose presence are bad, a certain proportion of carbon gives it a given degree of strength, while a small percentage of sulphur will render it useless for boiler purposes. The effect of the latter and phosphorus is crystalization of metal.

Plates are usually ordered by thickness, but there are occasions when weight is defined rather than the thickness and rejected unless up to demands. The effects sometimes are that owing to the plates being made of large dimensions and cut up to demands for smaller sizes some of uneven thickness are left; this is due to the process of rolling, the center of rolls expanding, thus leaving center of plate thicker; while rolls are turned in center to obviate this effect the heating of rolls must offset the turning down.

BOILER DESIGNING.

Boiler designing is a science and much depends on the accuracy of details.

Modern engines, higher pressure, and that potent factor of the times, competition, demand the greatest efficiency from fuel and engine.

But a few years ago comparatively, the rule was "thumb" in the designing of a boiler, of "what had been done" without any reasoning; this apparently when we see some of the boilers now in use; plates, seams, rivets, location of same, brace design, number, and method of attaching them, tubes, size, number and distribution; domes, their ratio to boiler, old-time makers and engineers said, "one-fifth the size of boiler was a fair ratio;" all giving evidence that it was no defined rule from reasoning, but following what had been done. Today the designing of a boiler is a problem to be worked out, solved by factors entering into the matter; location, space economy, fuel economy, engine design and efficiency, arrangement of furnaces that available heat can be most completely absorbed and utilized, effects of contraction and expansion, the various types of boiler must be considered for their niche of maximum usefulness, for often times one will excel in certain duties and fail in another. Requirements must be looked into and the one factor, location, would change a design completely, for instance, where space is limited, cost and life may be sacrificed, another where fuel would be for life, again, locations where fuel must be sacrificed, where water is bad, and a design must be made to suit the accessibilities to clean. Again, an illustration of what must be considered, and the sacrifice for demands and conditions to obtain results, is the fire engine boiler, life, cost, fuel, and access to clean and repair, all for quick steaming qualities. Then grate proportion for heating surface in different types of boiler, and the necessity of steam space and tube arrangement to avoid obstruction of steam passages that retard circulation; points which in early boiler designing were badly neglected.

Increased pressure has been demanded due to space and type of engine would often times vary proportions.

The power of boilers today is estimated from an evaporative measure, not from the old-time commercial rating, i. e., so many square feet of heating surface per H. P., leaving design or type out of the question. Thus we see the importance of boiler designing. The earliest known steam generator was a sphere. In the boiler of Worcester and Papin and Savery the flue encircled the outside of shell. Newcomen substituted that by having a hemispherical top and flat arch or bottom. The wagon boiler designed by Watt resembled a wagon and hence its name. Boilers have been made in many and various forms, classified by designer's name, their uses or form. Today boilers are generally classed as internal, external, water tube, pipe, and sectional (the latter used extensively for heating), each class usually bearing a name incident to their use, such as locomotive or marine, again boilers are further classed as vertical, horizontal, tubular, cylinder and flue.

CHAPTER II.

SELECTION OF BOILER.

In estimating the power of a boiler it was formerly a custom to have a certain number of square feet of heating surface to represent a H. P. (horse power) and the different types were supposed to have better or inferior efficiencies due to design for instance.

The cylinder type of boiler was reckoned from a unit of 10 square feet of heating surface per horse power, the horizontal tubular type, 12 to 15 square feet; the reason for the difference was the former type of boiler's heating surface was considered as all active and exposed to the highest temperature, while the latter had the heating surface of tubes that was exposed to the waste gases after coming in contact with the bottom thus a lower temperature, while as a fact the tubes were thinner and had more conductivity for heat; thus 15 square feet was considered the unit of measurement for that type.

Internal fire boilers were measured from the 10 square feet standard.

But as fuels now are valued by their heating values, the amount of water they will evaporate per pound of class fuel, so with the boiler, it must be measured from its efficiency from an evaporative point, other factors entering into its performances are hardness of water and temperature of feed water.

As the subject of the steam boiler is one that can be treated almost inexhaustibly, it is the writer's intention to devote this work to boiler rules and tables governing their construction.

ENGINE POWER.

Power, or as it is mechanically expressed, heat, is measured, and the unit of this measurement is the amount of heat which will raise the temperature of one pound of water one degree F at its point of greatest density (39 deg. F.). The number of heat units in one pound of water at any given temperature is called the "Heat in liquid," when heat is applied to water in open vessel the temperature will rise until its boiling point is reached, beyond this point no increase of temperature will result; the heat absorbed being employed in transforming the water from liquid to steam; this is called the "heat of vaporization," and diminishes as the temperature and pressure increases. The "heat in liquid," added to the "heat of vaporization," is equal to the total heat. The ratio of the amount of heat required to make one pound of steam under any given conditions to that required to make a pound of steam from and at 212° is called the "factor of evaporation."

This factor is found by subtracting the heat units in one pound of the feed water at the given temperature from the heat units or total heat of one pound of the steam at the given pressure, and dividing the result by 965.7, which is the heat of vaporization, or number of heat units required to evaporate one pound of water at 212° into steam at 212°.

The total number of pounds of water to be evaporated per hour under a given steam pressure multiplied by its particular factor of evaporating gives us the "equivalent evaporation," from and at 212°, or in other words, the amount of water which would have been evaporated, with the same amount of fuel, had the feed water been at 212 degrees and the pressure that of the atmosphere.

Assuming an engine to be one of 200 H. P. and the boiler to be selected according to the commercial rating of boilers. The given data to determine from would be:

200 HP engine,
engine taking 20 lbs. of steam per HP per hour
120 absolute pressure (by gauge 105)
190° temperature of feed water
the evaporation of 34.5 lbs. of water at 212°.

As stated, the number of pounds of water to be evaporated to produce a horse power from an engine will be computed from the type of engine used. See table of engine efficiencies, Standards of Steam Engine.

TABLE OF STANDARD OF STEAM ENGINES.

TYPE OF BOILERS.		TYPE OF ENGINES.				
Pounds of water evaporated in a common Horizontal Tubular boiler per lbs. of coal burned, 7 to 8 lbs.	Pounds of water evaporated in a modern water tube boiler per lbs. of coal burned, 9 to 10 lbs.	Simple non-condensing automatic cut-off engine, steam pressure 80 to 90 lbs.	Simple condensing automatic cut-off engines, steam pressure 80 to 90 lbs.	Compound non-condensing engines, steam pressure 130 to 140 lbs.	Compound condensing engines, steam pressure 120 to 140 lbs.	Triple cylinder expansion engines, steam pressures 140 to 160 lbs.
Water consumption of different types of engines per 1 HP per hour.		32 lbs.	22 lbs.	20 lbs.	16 lbs.	13 lbs.
Coal consumption per 1 HP per hour with a modern water tube boiler.		3½ lbs.	2½ lbs.	2¼ lbs.	1¾ lbs.	1¼ lbs.
Coal consumption per 1 HP per hour with a common HT boiler.		4 lbs.	3 lbs.	2¾ lbs.	2¼ lbs.	1¾ lbs.

RULES FOR CALCULATION.

THE CIRCLE.

Multiply diameter by 3.1416 to find circumference. Multiply circumference by .31831 to find diameter. Multiply square of diameter by .7854 to find area. Multiply the square root of area by 1.12837 to find diameter. Multiply diameter by .8862 to find side of a square equal to area. Multiply diameter by .7071—product is side of an inscribed square.

Rule to find area of a circular ring formed by two concentric circles: Multiply the sum of the two diameters by their difference and the product by .7854—the result is area. Multiply radius by 6.2831 to find circumference.

Rule to find area of a section of a circle: Multiply one-half the length of arc by the radius of circle.

Rule to find area of a sector: Multiply length of arc by the radius and divide the product by 2 for the area.

Example:

50″ = length of arc of sector
30″ = radius

2)1500

750 = area of sector

Rule to find area of a triangle: Multiply base by height and divide the product by 2 for the area.

Example:

38″ = base of triangle
20″ = height of triangle

2)760

380 = area of triangle

Rule to find area of a segment of a circle: Subtract area of triangle from area of sector. The result will be the area of segment.

Example:

750 = area of sector
380 = area of triangle

370 = area of segment

Rule to find one dimension of triangle when area and one dimension is given: Double the area and divide by given dimension.

Rule to find area of triangle when dimensions of three sides are given: From half the sum of the three sides, subtract each side separately; multiply the half sum and the three remainders together; the square root of the product is the area.

Rule to find hypothenuse of a triangle when dimensions of base and perpendicular are given: Extract the square root of the sum of the squares of the base and the perpendicular; the result is the length of hypothenuse.

Rule to find the base or perpendicular when hypothenuse is given: Extract the square root of the difference between the square of the squares of the base and the perpendicular; the result is the required side.

QUADRILATERALS.

Rule to find area of a parallelogram: Multiply base by altitude.

Rule to find area of a trapezoid: Multiply one-half sum of the parallel sides by the altitude.

Rule to find area of a trapezium: Multiply the diagonal by one-half sum of the perpendiculars drawn to it from the vertices of opposite angle.

Rule to find area of a rectangle: Multiply length by width.

Doubling the diameter of a circle increases its area four times.

The side of a square multiplied by 1.128 equals diameter of circle of equal area.

Rule to find volume of a pyramid or cone: Multiply the area of the base by one-third the altitude.

Rule to find the convex surface of a frustrum of a pyramid or of a cone: Multiply the sum of the perimeters or of the circumference by one-half the slant height.

Rule to find the volume of a frustrum of a pyramid or of a cone: To the sum of the areas of both bases add the square root of the product and multiply this sum by one-third of the altitude.

THE SPHERE.

Rule to find the surface of a sphere: Multiply the diameter by the circumference of a great circle of a sphere.

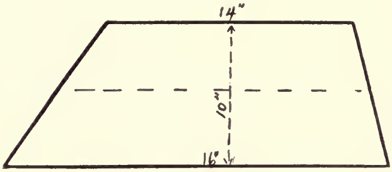
Rule to find the volume of a sphere: Multiply the surface by 1/6 of the diameter or 1/3 of the radius.

Rule to find the three dimensions of a rectangular solid, the volume and ratio of the dimensions being given: First, divide the volume by the product of the terms proportional to the three dimensions, and extract the cube root of the quotient. Second, multiply the root obtained by each proportional term; the products will be the corresponding side.

Rule to find solidity of a sphere: Multiply cube of diameter by .5236.

Rule to find surface of a ball: Multiply square of diameter by 3.1416.

A TRAPEZOID.



A plane four sided figure having two of the opposite sides parallel to each other.

Rule to find area of a trapezoid whose sides are 26″ and 14″ altitude 10″: Multiply one-half the sums of parallel sides by the altitude.

Example:

$$\begin{array}{r@{}l} & 26'' \\ & 14 \\ \hline 2) & 40 \\ \hline & 20 \\ & \;10 \\ \hline & 200 \end{array}\quad \text{area of trapezoid}$$

SOLIDS.

Rule to find volume of a prism or cylinder: Multiply area of the base by the altitude.

Rule to find convex surface of a prism or cylinder: Multiply the perimeter or circumference of the base by the altitude.

Signs Used in Mathematical Calculations.

π	Ratio of circumference of a circle to a diam., as 3.1416
=	Equal, as 12 inches = 1 foot
+	Plus, addition, as $2+4=6$
—	Minus, substraction, as $8-4=4$
×	Multiply, as $4\times4=16$
÷	Divide, as $10\div2=5$
: : :	Proportion, as 2 : 4 : : 8 : 16; or 2 is to 4 as 8 is to 16
√	Square root is required; cube root, $\sqrt[3]{27}=3$
5^2	Number is to be squared, $5^2=25$
5^3	Number is to be cubed, $5^3=125$
.	Decimal point, as $.1=\frac{1}{10}$; $.14=\frac{14}{100}$
()	Parenthesis, all numbers between to be taken as one
——	Vinculum signifies the numbers over which it is placed are to be taken together.
°	Degrees
′	Minutes or feet
″	Seconds or inches

A coefficient is a prescribed amount to make up for any defects reducing the strength of plate due to punching, caulking, etc.

A factor of safety is the difference between the safe working and bursting pressures.

CIRCUMFERENCES AND AREAS OF CIRCLES.

OF ONE INCH.				OF INCHES OR FEET.					
Fract.	Dec.	Circ.	Area	Dia.	Circ.	Area	Dia.	Circ.	Area.
1-64	.015625	.04909	.00019	1	3.1416	.7854	64	201.06	3216.99
1-32	.03125	.09818	.00077	2	6.2832	3.1416	65	204.20	3318.31
3-64	.046875	.14726	.00173	3	9.4248	7.0686	66	207.34	3421.19
1-16	.0625	.19635	.00307	4	12.5664	12.5664	67	210.49	3525.65
5-64	.078125	.24545	.00479	5	15.7080	19.635	68	213.63	3631.68
3-32	.09375	.29452	.00690	6	18.850	28.274	69	216.77	3739.28
7-64	.109375	.34363	.00939	7	21.991	38.485	70	219.91	3848.45
1-8	.125	.39270	.01227	8	25.133	50.266	71	223.05	3959.19
9-64	.140625	.44181	.01553	9	28.274	63.617	72	226.19	4171.50
5-32	.15625	.49087	.01917	10	31.416	78.540	73	229.34	4185.39
11-64	.171875	.53999	.02320	11	34.558	95.033	74	232.48	4300.84
3-16	.1875	.58905	.02761	12	37.699	113.1	75	235.62	4417.86
13-64	.203125	.63817	.03241	13	40.841	132.73	76	238.76	4536.46
7-32	.21875	.68722	.03758	14	43.982	153.94	77	241.90	4656.63
15-64	.234375	.73635	.04314	15	47.124	176.71	78	245.04	4778.36
1-4	.25	.78540	.04909	16	50.265	201.06	79	248.19	4901.67
17-64	.265625	.83453	.05542	17	53.407	226.98	80	251.33	5026.55
9-32	.28125	.88357	.06213	18	56.549	254.47	81	254.47	5153.
19-64	.296875	.93271	.06922	19	59.690	283.53	82	257.61	5281.02
5-16	.3125	.98175	.07670	20	62.832	314.16	83	260.75	5410.61
21-64	.328125	1.0309	.08456	21	65.973	346.36	84	263.89	5541.77
11-32	.34375	1.0799	.09281	22	69.115	380.13	85	267.04	5674.50
23-64	.359375	1.1291	.10144	23	72.257	415.48	86	270.18	5808.80
3-8	.375	1.1781	.11045	24	75.398	452.39	87	273.32	5944.68
25-64	.390625	1.2273	.11984	25	78.540	490.87	88	276.46	6082.12
13-32	.40625	1.2763	.12962	26	81.681	530.93	89	279.60	6221.14
27-64	.421875	1.3254	.13979	27	84.823	572.56	90	282.74	6361.73
7-16	.4375	1.3744	.15033	28	87.965	615.75	91	285.88	6503.88
29-64	.453125	1.4236	.16126	29	91.106	660.52	92	289.03	6647.61
15-32	.46875	1.4726	.17257	30	94.248	706.86	93	292.17	6792.91
31-64	.484375	1.5218	.18427	31	97.389	754.77	94	295.31	6939.78
1-2	.5	1.5708	.19635	32	100.53	804.25	95	298.45	7088.22
33-64	.515625	1.6199	.20880	33	103.67	855.30	96	301.59	7238.23
17-32	.53125	1.6690	.22166	34	106.81	907.92	97	304.73	7339.81
35-64	.546875	1.7181	.23489	35	109.96	962.11	98	307.88	7542.96
9-16	.5625	1.7671	.24850	36	113.10	1017.88	99	311.02	7697.69
37-64	.578125	1.8163	.26248	37	116.24	1075.21	100	314.16	7853.98
19-32	.59375	1.8653	.27688	38	119.38	1134.11	101	317.30	8011.85
39-64	.609375	1.9145	.29164	39	122.52	1194.59	102	320.44	8171.28
5-8	.625	1.9635	.30680	40	125.66	1256.64	103	323.58	8332.29
41-64	.640625	2.0127	.32232	41	128.81	1320.25	104	326.73	8494.87
21-32	.65625	2.0617	.33824	42	131.95	1385.44	105	329.87	8659.01
43-64	.671875	2.1108	.35453	43	135.09	1452.20	106	333.01	8824.73
11-16	.6875	2.1598	.37122	44	138.23	1520.53	107	336.15	8992.02
45-64	.703125	2.2090	.38828	45	141.37	1590.43	108	339.29	9160.88
23-32	.71875	2.2580	.40574	46	144.51	1661.90	109	342.43	9331.32
47-64	.734375	2.3072	.42356	47	147.65	1734.94	110	345.58	9503.32
3-4	.75	2.3562	.44179	48	150.80	1809.56	111	348.72	9676.89
49-64	.765625	2.4054	.45253	49	153.94	1885.74	112	351.86	9852.03
25-32	.78125	2.4544	.47937	50	157.08	1963.50	113	355.	10028.75
51-64	.796875	2.5036	.49872	51	160.22	2042.82	114	358.14	10207.03
13-16	.8125	2.5525	.51849	52	163.36	2123.72	115	361.28	10386.89
53-64	.828125	2.6017	.53862	53	166.50	2206.18	116	364.42	10568.32
27-32	.84375	2.6507	.55914	54	169.65	2290.22	117	367.57	10751.32
55-64	.859375	2.6999	.58003	55	172.79	2375.83	118	370.71	10935.88
7-8	.875	2.7489	.60132	56	175.93	2463.01	119	373.85	11122.02
57-64	.890625	2.7981	.62298	57	179.07	2551.76	120	376.99	11309.73
29-32	.90625	2.8471	.64504	58	182.21	2642.08	121	380.13	11499.01
59-64	.921875	2.8963	.66746	59	185.35	2733.97	122	383.27	11689.87
15-16	.9375	2.9452	.69029	60	188.50	2827.43	123	386.42	11882.29
61-64	.953125	2.9945	.71349	61	191.64	2922.47	124	389.56	12076.28
31-32	.96875	3.0434	.73708	62	194.78	3019.07	125	392.70	12271.85
63-64	.984375	3.0928	.76097	63	197.92	3117.25	126	395.84	12468.98

Areas of Circles from 1/32 Inch up to 10 Inches in Diameter, Advancing by Thirty-Seconds of an Inch.

INCHES.

	0″	1″	2″	3″	4″	5″	6″	7″	8″	9″	
0	....	.7854	3.1416	7.068	12.56	19.63	28.27	38.48	50.26	63.62	0
1/32	.000767	.8352	3.240	7.216	12.76	19.88	28.57	38.83	50.66	64.06	1/32
1/16	.00306	.8866	3.341	7.366	12.96	20.13	28.87	39.17	51.05	64.50	1/16
3/32	.0069	.9395	3.443	7.516	13.16	20.38	29.16	39.52	51.45	64.95	3/32
1/8	.0123	.9940	3.546	7.669	13.36	20.63	29.46	39.87	51.85	65.40	1/8
5/32	.0192	1.050	3.651	7.824	13.57	20.88	29.77	40.22	52.25	65.84	5/32
3/16	.0276	1.107	3.758	7.970	13.77	21.13	30.07	40.57	52.65	66.30	3/16
7/32	.0376	1.166	3.866	8.137	13.98	21.39	30.37	40.93	53.05	66.75	7/32
1/4	.0491	1.227	3.976	8.295	14.19	21.65	30.68	41.28	53.46	67.20	1/4
9/32	.0621	1.289	4.087	8.456	14.40	21.91	30.99	41.64	53.86	67.65	9/32
5/16	.0767	1.353	4.199	8.618	14.61	22.17	31.30	42.00	54.27	68.11	5/16
11/32	.0928	1.418	4.314	8.781	14.82	22.43	31.61	42.36	54.68	68.57	11/32
3/8	.1104	1.484	4.430	8.946	15.03	22.69	31.92	42.72	55.09	69.03	3/8
13/32	.1296	1.553	4.547	9.112	15.25	22.95	32.23	43.08	55.50	69.49	13/32
7/16	.1503	1.623	4.666	9.280	15.47	23.22	32.55	43.45	55.91	69.95	7/16
15/32	.1725	1.694	4.786	9.450	15.68	23.49	32.86	43.81	56.33	70.42	15/32
1/2	.1963	1.767	4.908	9.621	15.90	23.76	33.18	44.18	56.74	70.88	1/2
17/32	.2216	1.840	5.032	9.794	16.13	24.03	33.50	44.55	57.16	71.35	17/32
9/16	.2485	1.917	5.157	9.968	16.35	24.30	33.82	44.92	57.58	71.82	9/16
19/32	.2770	1.994	5.283	10.14	16.57	24.58	34.15	45.29	58.00	72.29	19/32
5/8	.3067	2.073	5.411	10.32	16.80	24.85	34.47	45.66	58.43	72.76	5/8
21/32	.3382	2.154	5.541	10.50	17.03	25.13	34.80	46.04	58.85	73.23	21/32
11/16	.3712	2.236	5.672	10.68	17.26	25.41	35.12	46.41	59.28	73.71	11/16
23/32	.4057	2.319	5.805	10.86	17.49	25.68	35.45	46.79	59.70	74.18	23/32
3/4	.4417	2.405	5.939	11.04	17.72	25.97	35.78	47.17	60.13	74.66	3/4
25/32	.4793	2.492	6.075	11.23	17.95	26.25	36.11	47.55	60.56	75.14	25/32
13/16	.5184	2.581	6.212	11.41	18.19	26.53	36.45	47.94	60.99	75.62	13/16
27/32	.5591	2.669	6.351	11.60	18.43	26.82	36.79	48.32	61.24	76.10	27/32
7/8	.6013	2.761	6.491	11.79	18.66	27.11	37.12	48.71	61.86	76.59	7/8
29/32	.6450	2.854	6.633	11.98	18.91	27.40	37.46	49.09	62.30	77.07	29/32
15/16	.6903	2.948	6.777	12.18	19.15	27.69	37.80	49.48	62.74	77.56	15/16
31/32	.7370	3.044	6.922	12.37	19.39	27.98	38.14	49.87	63.18	78.05	31/32

DECIMALS OF A FOOT FOR EACH $\frac{1}{32}$ND OF AN INCH.

INCH	0″	1″	2″	3″	4″	5″	6″	7″	8″	9″	10″	11″
0	0	.0833	.1667	.2500	.3333	.4167	.5000	.5833	.6667	.7500	.8333	.9167
1/32	.0026	.0859	.1693	.2526	.3359	.4193	.5026	.5859	.6693	.7526	.8359	.9193
1/16	.0052	.0885	.1719	.2552	.3385	.4219	.5052	.5885	.6719	.7552	.8385	.9219
3/32	.0078	.0911	.1745	.2578	.3411	.4245	.5078	.5911	.6745	.7578	.8411	.9245
⅛	.0104	.0937	.1771	.2604	.3437	.4271	.5104	.5937	.6771	.7604	.8437	.9271
5/32	.0130	.0964	.1797	.2630	.3464	.4297	.5130	.5964	.6797	.7630	.8464	.9297
3/16	.0156	.0990	.1823	.2656	.3490	.4323	.5156	.5990	.6823	.7656	.8490	.9323
7/32	.0182	.1016	.1849	.2682	.3516	.4349	.5182	.6016	.6849	.7682	.8516	.9349
¼	.0208	.1042	.1875	.2708	.3542	.4375	.5208	.6042	.6875	.7708	.8542	.9375
9/32	.0234	.1068	.1901	.2734	.3568	.4401	.5234	.6068	.6901	.7734	.8568	.9401
5/16	.0260	.1094	.1927	.2760	.3594	.4427	.5260	.6094	.6927	.7760	.8594	.9427
11/32	.0286	.1120	.1953	.2786	.3620	.4453	.5286	.6120	.6953	.7786	.8620	.9453
⅜	.0312	.1146	.1979	.2812	.3646	.4479	.5312	.6146	.6979	.7812	.8646	.9479
13/32	.0339	.1172	.2005	.2839	.3672	.4505	.5339	.6172	.7005	.7839	.8672	.9505
7/16	.0365	.1198	.2031	.2865	.3698	.4531	.5365	.6198	.7031	.7865	.8698	.9531
15/32	.0391	.1224	.2057	.2891	.3724	.4557	.5391	.6224	.7057	.7891	.8724	.9557
½	.0417	.1250	.2083	.2917	.3750	.4583	.5417	.6250	.7083	.7917	.8750	.9583
17/32	.0443	.1276	.2109	.2943	.3776	.4609	.5443	.6276	.7109	.7943	.8776	.9609
9/16	.0469	.1302	.2135	.2969	.3802	.4635	.5469	.6302	.7135	.7969	.8802	.9635
19/32	.0495	.1328	.2161	.2995	.3828	.4661	.5495	.6328	.7161	.7995	.8828	.9661
⅝	.0521	.1354	.2188	.3021	.3854	.4688	.5521	.6354	.7188	.8021	.8854	.9688
21/32	.0547	.1380	.2214	.3047	.3880	.4714	.5547	.6380	.7214	.8047	.8880	.9714
11/16	.0573	.1406	.2240	.3073	.3906	.4740	.5573	.6406	.7240	.8073	.8906	.9740
23/32	.0599	.1432	.2266	.3099	.3932	.4766	.5599	.6432	.7266	.8099	.8932	.9766
¾	.0625	.1458	.2292	.3125	.3958	.4792	.5625	.6458	.7292	.8125	.8958	.9792
25/32	.0651	.1484	.2318	.3151	.3984	.4818	.5651	.6484	.7318	.8151	.8984	.9818
13/16	.0677	.1510	.2344	.3177	.4010	.4844	.5677	.6510	.7344	.8177	.9010	.9844
27/32	.0703	.1536	.2370	.3203	.4036	.4870	.5703	.6536	.7370	.8203	.9036	.9870
⅞	.0729	.1562	.2396	.3229	.4062	.4896	.5729	.6562	.7396	.8229	.9062	.9896
29/32	.0755	.1589	.2422	.3255	.4089	.4922	.5755	.6589	.7422	.8255	.9089	.9922
15/16	.0781	.1615	.2448	.3281	.4115	.4948	.5781	.6615	.7448	.8281	.9115	.9948
31/32	.0807	.1641	.2474	.3307	.4141	.4974	.5807	.6641	.7474	.8307	.9141	.9974
1	......	......	......	......	......	......	......	......	......	......	......	1.0000

HORSE POWER MEASUREMENT.

In calculating the H. P. boiler required for a given engine it is customary to calculate what amount of water would be evaporated per hour at the temperature of 212 atmospheric pressure.

The ratio of the amount of heat required to make one pound of steam under any given condition to that required to make a pound of steam from 212° is called the factor of evaporation, and this is found by subtracting the heat units in one pound of the feed water at the given temperature, from the heat units in one pound of steam at the given pressure, and dividing the result by 965.7, which is the heat of evaporation, or number of heat units required to evaporate one pound of water at 212° into steam of 212°.

The number of pounds of water to be evaporated per hour under a given steam pressure, multiplied by its particular factor of evaporation, gives the factor of evaporation from and at 212° (or the amount of water which would have been evaporated with the same amount of fuel, had the feed water been at 212 degrees atmospheric pressure.

Hence it is first necessary to find the amount of water the engine is to use per hour; then the factor of evaporation and the product of these two will be the equivalent from and at 212°; 34½ pounds of water at 212° evaporated into steam at atmospheric pressure equals a horse power; dividing the equivalent evaporation by 34½ gives the horse power required.

Rule to find capacity of boiler for any engine, this according to the commercial rating of boilers: Multiply the horse power of the engine by the number of pounds of steam the engine will consume per indicated horse power per hour and call this product No. 1; from the number of heat units contained in one pound of the steam at absolute pressure subtract the number of heat units in one pound of feed water, and divide by 965.7 to get factor of evaporation, and call this product No. 2; multiply product No. 1 by product No. 2 and divide by 34½ (the number of pounds of water evaporated from and at 212°, to develop one horse power), and this product will be the required commercial rating of boiler.

LEGEND:

E =Power of engine
L =Lbs. of steam per horse power
P =Pressure
T =Temperature of feed water
W =Water to be evaporated per HP per hour
TSH =Total heat units in steam
HU =Heat units in feed water
HE =Heat of evaporation
FE =Factor of evaporation
W of W =Weight of water used per HP per hour

FORMULA:

$$\frac{E \times L \times (TSH—HU \div 965.7)}{34\frac{1}{2}} = \text{commercial rating of boiler.}$$

```
        No. 1   200  HP engine
                 20  No. of lbs. of steam per HP per hour
               ----
               4000 =the weight in lbs. of water used per hour
No. 2
Heat units to
evaporate one               1217.9445 =total heat of given steam
lb. of water at              190.643  =heat units in feed water
212° into steam             ----------
at 212° =             965.7)1027.3015(1.063 factor of evaporation
                             965 7
                           --------
                              61 601
                              57 942
                              ------
                               3 6595
                               2 8971
                               ------
                                 7624

               1.063     factor of evaporation =product No. 2
                4000     weight of water in lb. use per hour =product No. 1
               --------
               4252.000  the equivalent evaporation from and at 212° F.

Weight of water
required per hour
per HP for high
pressure engine =     34.5)4252.000(123.24 commercial HP of boiler required
                           345
                           ---
                            802
                            690
                            ----
                            1120
                            1035
                            ----
                              850
                              690
                              ----
                              1600
                              1380
                              ----
                               220
```

This example was figured on a basis of 34½ lbs. of water per engine HP. The consumption of steam of modern engine, per HP, varies in limits, depending on type of engine.

PROPERTIES OF STEAM.

The temperature at which water is converted into steam varies with the pressure. At atmospheric the steaming point is 212 degrees F., less at low pressure and higher at higher pressure. When water reaches the boiling point, further addition of heat effects no change in temperature, the heat absorbed in producing steam having the same temperature and pressure as that at which it is evaporated. The heat thus absorbed is known as the latent heat, so called because it produces effects other than those of change of temperature. The amount of heat rendered latent by each pound of water in becoming steam varies with the pressure, decreasing as the pressure rises. The latent heat added to the sensible heat (this latter as shown by the thermometer) gives the total heat, this term used to designate the number of heat units contained in one pound of steam above a given temperature. Total heat is calculated from 32 degrees F. as the total heat is greater the higher the pressure; the amount of fuel necessary to evaporate a pound of water increases with the pressure; saturated steam cannot be superheated in contact with water, that is, its temperature cannot be raised above the point normal to the pressure, neither can it be cooled without change of pressure, for any loss of heat is compensated by the latent heat of the steam which is condensed.

Saturated steam is that which has the minimum temperature at which it can exist as a vapor under the given pressure.

Superheated steam has a temperature higher than that of saturation at the same pressure. The same pressure above vacuum is the gauge pressure plus 14.7 pounds.

TABLE

Pressure of Steam at Different Temperatures.

Pounds pressure per square inch above vacuum	Temperature Fahr.	Heat units in water above 32°	Latent heat in Heat of Vaporization	Total heat units above 32°	Volume of one pound in cubic foot
1	101.99	70.0	1043.0	1113.1	334.5
5	162.34	130.7	1000.8	1131.5	73.21
10	193.25	161.9	979.0	1140.9	38.15
15	213.03	181.8	965.1	1146.9	26.14
20	227.95	196.9	954.6	1151.5	19.91
25	240.04	209.1	946.0	1155.1	16.13
30	250.72	219.4	938.9	1158.3	13.59
35	259.19	228.4	932.6	1161.0	11.75
40	267.13	236.4	927.0	1163.4	10.37
45	274.29	243.6	922.0	1165.6	9.285
50	280.85	250.2	917.4	1167.6	8.418
55	286.89	256.3	913.1	1169.4	7.698
60	292.51	261.9	909.3	1171.2	7.097
65	297.77	267.2	905.5	1172.7	6.583
70	302.71	272.2	902.1	1174.3	6.143
75	307.38	276.9	898.8	1175.7	5.760

Pressure of Steam at Different Temperatures.

Pounds pressure per square inch above vacuum	Temperature Fahr.	Heat units in water above 32°	Latent heat in Heat of Vaporization	Total heat units above 32°	Volume of of one pound cubic foot
80	311.80	281.4	895.6	1177.0	5.426
85	316.02	285.8	892.5	1178.3	5.126
90	320.04	290.0	889.6	1179.6	4.859
95	323.89	294.0	886.7	1180.7	4.619
100	327.58	297.9	884.0	1181.9	4.403
105	331.13	301.6	881.3	1182.9	4.205
110	334.56	305.2	878.8	1184.0	4.026
115	337.86	308.7	876.3	1185.0	3.862
120	341.05	312.0	874.0	1186.0	3.711
125	344.13	315.2	871.7	1186.9	3.572
130	347.12	318.4	869.4	1187.8	3.444
135	350.03	321.4	867.3	1188.7	3.323
140	352.85	324.4	865.1	1189.5	3.212
145	355.59	327.2	863.2	1190.4	3.107
150	358.26	330.0	861.0	1191.2	3.011
155	360.86	332.7	859.3	1192.0	2.919
160	363.40	335.4	857.4	1192.8	2.833
165	365.88	338.0	855.6	1193.6	2.751
170	368.29	340.5	853.8	1194.3	2.676
175	370.65	343.0	852.0	1195.0	2.603
180	372.97	345.4	850.3	1195.7	2.535
185	375.23	347.8	848.6	1196.4	2.470
190	377.44	350.1	847.0	1197.1	2.408
195	379.61	352.4	845.3	1197.7	2.349
200	381.73	354.6	843.8	1198.4	2.294
205	383.82	356.8	842.2	1199.0	2.241
210	385.87	358.9	840.7	1199.6	2.190
215	387.88	361.0	839.2	1200.2	2.142
220	389.84	363.0	837.8	1200.8	2.096
225	391.79	365.1	836.3	1201.4	2.051
250	400.99	374.7	829.5	1204.2	1.854
275	409.50	383.6	823.2	1206.8	1.691
300	417.42	391.9	817.4	1209.3	1.553
325	424.82	399.6	811.9	1211.5	1.437
350	431.90	406.9	806.8	1213.7	1.337
375	438.40	414.2	801.5	1215.7	1.250
400	445.15	421.4	796.3	1217.7	1.172
500	466.57	444.3	779.9	1224.2	.939

ENGINE NOTES.

Steam at atmospheric pressure flows into a *Vacuum* at the rate of about 1,550 feet per second, and into the *Atmosphere* at the rate of 650 feet per second.

The specific gravity of steam (at atmospheric pressure) is .411, that of air at 34 deg. Fahrenheit, and .0006 that of water at same temperature.

33000 minute foot pounds equal 1 H. P.

396000 minute inch pounds equal 1 H. P.

A cubic inch of water evaporated under atmospheric pressure is approximately converted into 1 cubic foot of steam.

The horse power of boilers, as per standard adopted by the Am. S. M. E., is 30 pounds water evaporated per hour at a pressure of 70 pounds per square inch and from a temperature of 100 degrees Fahr.

Well designed boilers, under successful operation, will evaporate from 7 to 10 pounds of water per pound of first-class coal.

Each square foot of heating surface is considered sufficient to evaporate 3½ pounds of water; therefore, for an engine using 30 pounds of water per horse power per hour, each horse power of the engine requires **8.75** square feet heating surface in the boiler.

On one square foot of fire grate can be burned on an average from 10 to 12 pounds hard coal, or 18 to 35 pounds soft coal, per hour, with natural draft.

Two and one-quarter pounds of dry wood is equal to 1 pound of average quality soft coal.

Condensing engines require from 20 to 30 times the amount of feed water for condensing purposes; approximately for most engines, 1 to 1½ gallons condensing water per minute per indicated horse power, depending on temperature of injection water.

Surface condensers for compound steam engines require about 2 square feet of cooling surface per horse power; ordinary engines will require more surface according to their economy in the use of steam. It is absolutely necessary that the air pump should be set lower than the condenser for satisfactory results.

The effect of a good air pump and condenser should be to get 25 inches of vacuum and to make available about 10 pounds more mean effective pressure with the same terminal pressure, or to give the same mean effective pressure with a correspondingly less terminal pressure. Approximately, a good condenser will save one-fourth of the fuel consumed, or, in other words, increase the power of the engine one-fourth, the fuel consumption remaining the same.

One pound of water evaporated from, and at 212° F. is equivalent to 965.7 British thermal units.

The evaporation of 30 pounds of water per hour, from a temperature of 100° F., into steam at 70 pounds gauge pressure = one H. P. This is equivalent to 34½ pounds of water from and at 212° F.

A common rule to find horse power on an engine: Multiply area of piston by pressure per square inch and by length of stroke and again by number of revolutions per minute; divide this sum by constant 16500.

Legend:

P =pressure = 100 lbs.
A =area of piston = 78.5400
S =length of stroke in feet = 1 ft.
R =number of revolutions = 70
C =constant = 16500

Formula:

$$\frac{A \times P \times S \times R}{C} = \text{H.P.}$$

Example:

78.5400 = area of piston
100 = lbs. pressure
7854.0000
1 ft. stroke
7854.0000
70 = number of revolutions
constant = 16500) 549780.0000 (33.3 = horse power
49500
54780
49500
52800
49500
3300

THE THERMOMETER.

To convert Fahrenheit degrees to centigrade, subtract 32 degrees from number of degrees Fahrenheit; multiply the sum by 5 and divide product by 9.

Legend:

F =Fahrenheit = 32°
C =Centigrade = 100°
R =Reaumur = 80°

Formula:

$$\frac{5 \times (F-32)}{9} = \text{Centigrade}$$

Example:

$$\begin{array}{r l} 212 & = \text{degrees Fahrenheit} \\ 32 & \\ \hline 180 & \\ 5 & \\ \hline 9)900 & \\ \hline 100 & = \text{Centigrade} \end{array}$$

To convert Centigrade degrees to Fahrenheit: Multiply the number of degrees centigrade by 9, divide result by 5 and add 32 to quotient.

Formula:

$$\frac{9 \times C}{5} + 32 = \text{Fahrenheit}$$

Example:

$$\begin{array}{r l} 100 & = \text{degrees Centigrade} \\ 9 & \\ \hline 5)900 & \\ \hline 180 & \\ 32 & \quad \text{to be added} \\ \hline 212 & = \text{degrees Fahrenheit} \end{array}$$

To convert Fahrenheit degrees to Reaumur subtract from number of degrees Fahrenheit 32; multiply result by 4 and divide product by 9.

Formula:

$$\frac{4 \times (F - 32)}{9} = \text{Reaumur}$$

Example:

$$\begin{array}{r l} 212 & = \text{degrees Fahrenheit} \\ 32 & \\ \hline 180 & \\ 4 & \\ \hline 9)720 & \\ \hline 80 & = \text{degrees Reaumur} \end{array}$$

To convert Reaumur degrees to Fahrenheit: Multiply number of degrees of Reaumur by 9; divide product by 4 and add 32 to quotient.

Formula:

$$\frac{9 \times R}{4} + 32 = \text{Fahrenheit}$$

Example:

$$\begin{array}{r l} 80 & = \text{degrees Reaumur} \\ 9 & \\ \hline 4)720 & \\ \hline 180 & \\ 32 & \quad \text{to be added} \\ \hline 212 & = \text{degrees Fahrenheit} \end{array}$$

Comparisons of Thermometer Scales.

Fahrenheit	Centigrade	Reaumur	Fahrenheit	Centigrade	Reaumur
— 4	—20	—16	113	45	36
+ 5	15	12	112	50	40
14	10	8	131	55	44
23	5	4	140	60	48
32	0	0	149	65	52
41	+ 5	+ 4	158	70	56
50	10	8	167	75	60
59	15	12	176	80	64
68	20	16	185	85	68
77	25	20	194	90	72
86	30	24	203	95	76
95	35	28	212	100	80
104	40	32	...	...	..
BOILING POINT	BOILING POINT	BOILING POINT	FREEZING POINT	FREEZING POINT	FREEZING POINT
212	100	80	32	0	0

CHAPTER III.

BOILER CONSTRUCTION.

Boiler construction can be classed as one of the highest among crafts. In old-time boiler making holes were punched leaving initial fractures around edge of holes and often times, when assembling joints, holes were found out of alignment, and to admit a rivet the plate had to be cut by reaming to make the holes coincide, thus reducing the percentage of strength, at best, very low. Today drilled holes are specified by reliable authorities and followed up by reputable boiler makers. Modern machinery of today has developed a wonderful improvement in the craft; it has taken the place of old-time hand methods; accuracy, efficiency and strength have been gained; improved tools to facilitate work, brain and not all muscle employed by the mechanics; he reasons, conceives, then executes with these modern conveniences; his aim is to produce results, betterment of his work. Flanging machines have added factors to safety; that old methods of flanging were not conducive to good effects or results is now apparent; for when the part of work to be flanged was heated, hammered, reheated and hammered again —hot and cold—often resulting in defects in plates that made them unfit for use, time and material would be wasted. With the modern flanging machine time is saved, expense lessened and work turned out as near perfect as possible, one heat and the cooling having an annealing effect, general and gradual, gang punches adjusted accurately, time and labor saved and the efficiency of joint holes not impaired.

Rivet machinery with its power of compression ensures strength of rivet joints and lessens the effect of injury to plate by caulking as done by the old-time hand riveted joint, especially when left to the novice, defects were developed and material operated on was destroyed.

Electric cranes and air lifts are found necessary for facilitating work by aiding in assembling or fitting up parts of boilers under construction.

Thus we find boiler making today one of the scientific mechanical crafts and with the expectations that work carried out as designed produce the best results.

This book will give general rules and tables used in the construction of the steam boiler and governing their use in safety.

RIVETS AND RIVETING.

In designing a joint like any part of the construction of boilers, care in calculation and proportioning of rivet are very essential. Shearing strength and ductility are important factors; perfect alignment of holes, size of same, and method of making same, must not be overlooked.

On the driving of a rivet will depend much. Without going into the details on the subject of riveting it may be well to say that in the old-time methods of hand riveting the structural makeup of a rivet was changed; when the rivet should have been finished, the many repeated blows soon changed its nature, and, unnecessary to say, "it was near finished." But improved machinery has wrought changes and with it the changing of rivet material—this in turn has provided a larger factor of safety using old rules, and has provided greater efficiency by lighter material.

The heating of rivet to proper degree of heat is another important measure and with modern forges as used this can be accomplished with no difficulty or more than ordinary attention.

Table of Rivets and Bolts Without Nuts in 100 Lbs.

Average number.

Length of Rivets.	Diameter of Rivets.								
	¼	5/16	⅜	7/16	½	⅝	11/16	¾	⅞
½	8000	5100	3200	1900	...	...	...	...	...
⅝	7000	4500	2900	1800	...	...	...	...	...
¾	6300	4100	2373	1476	1103	642	...	...	...
⅞	5700	3700	2190	1371	1030	604	...	...	...
1	5200	3400	2034	1280	968	571	400	345	...
1⅛	4700	3100	1898	1200	910	541	382	322	208
1¼	4400	2900	1780	1129	862	514	365	311	206
1⅜	4100	2700	1675	1066	815	489	350	295	204
1½	4000	2500	1582	1010	776	462	335	284	201
1⅝	3800	2300	1498	960	740	446	324	275	199
1¾	3500	2200	1424	914	707	428	311	266	192
1⅞	3400	2000	1356	872	672	411	302	257	185
2	3000	1900	1295	834	648	395	293	249	178
2⅛	....	....	1238	800	623	381	285	240	172
2¼	2800	1800	1187	768	599	367	277	233	167
2⅜	....	....	1139	738	577	354	269	226	162
2½	2500	1700	1095	711	556	343	261	219	157
2⅝	....	....	1052	687	537	332	253	212	152
2¾	....	....	1017	662	519	321	245	206	148
2⅞	....	....	982	636	503	311	237	201	144
3	....	....	949	611	487	302	230	196	140
3¼	....	....	890	581	459	285	218	186	132
3½	....	....	837	548	433	270	208	177	126
3¾	....	....	791	519	411	257	198	168	120
3⅞	....	....	....	....	395	250	195	165	119
4	....	....	749	400	390	244	189	161	115
4¼	....	....	....	....	372	233	180	155	110
4½	....	....	....	....	355	223	172	149	105
4¾	....	....	....	....	339	214	166	143	101
5	....	....	....	....	325	205	160	136	97
5¼	....	....	....	....	312	197	154	131	94
5½	....	....	....	....	300	190	149	127	91
5¾	....	....	....	....	289	183	144	123	88
6	....	....	....	....	279	177	139	118	85

The measurement of a cone or button head rivet is taken under the head; rivets for counter sunk holes measured over all.

Safe loads for any number of iron rivets from one to ten, ranging in diameter from ½ inch to 1⅜ inches, assuming a shearing strength of 42,000 pounds for iron rivets in single shear, as determined by experiments conducted by the Master Steam Boilermakers' Association and reported and endorsed at the 1906 convention of that Association.

Diam. of Rivet.	Diam. of Hole.	Area of Hole.	SHEARING STRENGTH OF IRON RIVETS AT 42,000 LBS. PER SQUARE INCH.									
			1 Rivet.	2 Rivets.	3 Rivets.	4 Rivets.	5 Rivets.	6 Rivets.	7 Rivets.	8 Rivets.	9 Rivets.	10 Rivets.
½	9/16	.2485	10,437	20,874	31,311	41,748	52,185	62,622	73,059	83,496	93,933	104,370
9/16	⅝	.3068	12,885	25,770	38,655	51,540	64,425	77,310	90,195	103,080	115,965	128,850
⅝	11/16	.3712	15,590	31,180	46,770	62,360	77,950	93,540	109,130	124,720	140,310	155,900
11/16	¾	.4417	18,551	37,102	55,653	74,204	92,755	111,306	129,857	148,408	166,959	185,510
¾	13/16	.5185	21,777	43,554	65,331	87,108	108,885	130,662	152,439	174,216	195,993	217,770
13/16	⅞	.6013	25,254	50,508	75,762	101,016	126,270	151,524	176,778	202,032	227,286	252,540
⅞	15/16	.6902	28,988	57,976	86,964	115,952	144,940	173,928	202,916	231,904	260,892	289,880
15/16	1	.7854	32,986	65,972	98,958	131,944	164,930	197,916	230,902	263,888	296,874	329,860
1	1 1/16	.8866	37,237	74,474	111,711	148,948	186,185	223,422	260,659	297,896	335,133	372,370
1 1/16	1⅛	.9940	41,748	83,496	125,244	166,992	208,740	250,488	292,236	333,984	375,732	417,480
1⅛	1 3/16	1.1079	46,515	93,030	139,545	186,060	232,575	279,090	325,605	372,120	418.635	465,150
1 3/16	1¼	1.2271	51,538	103,076	154,614	206,152	257,690	309,228	360,766	412,304	463,842	515,380
1¼	1 5/16	1.3529	56,822	113,644	170,466	227,288	284,110	340,932	397,754	454,576	511,398	568,220
1 5/16	1⅜	1.4848	62,361	124,722	187,083	249,444	311,805	374,166	436,527	498,888	561,249	623,610
1⅜	1 7/16	1.6229	68,162	136,324	204,486	272,648	340,810	408,972	477,134	545,296	613,458	681,620

Safe loads for any number of steel rivets from one to ten, ranging in diameter from ½ inch to 1⅜ inches, assuming a shearing strength of 45,000 pounds for steel rivets in single shear, as determined by experiments conducted by the Master Steam Boilermakers' Association and reported and endorsed at the 1906 convention of that Association.

Diam. of Rivet,	Diam. of Hole.	Area of Hole.	SHEARING STRENGTH OF STEEL RIVETS AT 45,000 LBS. PER SQUARE INCH.									
			1 Rivet.	2 Rivets.	3 Rivets.	4 Rivets.	5 Rivets.	6 Rivets.	7 Rivets.	8 Rivets.	9 Rivets.	10 Rivets.
½	9/16	.2485	11,182	22,364	33,546	44,728	55,910	67,092	78,274	89,456	100,638	111,820
9/16	⅝	.3068	13,806	27,612	41,418	55,224	69,030	82,836	96,642	110,448	124,254	138,060
⅝	11/16	.3712	16,704	33,408	50,112	66,816	83,520	100,224	116,928	133,632	150,336	167,040
11/16	¾	.4417	19,876	39,752	59,628	79,504	99,380	119,256	139,132	159,008	178,884	198,760
¾	13/16	.5185	23,332	46,664	69,996	93,328	116,660	139,992	163,324	186,656	209,988	233,320
13/16	⅞	.6013	27,058	54,116	81,174	108,232	135,290	162,348	189,406	216,464	243,522	270,580
⅞	15/16	.6902	31,059	62,118	93,177	124,236	155,295	186,354	217,413	248,472	279,531	310,590
15/16	1	.7854	35,343	70,686	106,029	141,372	176,715	212,058	247,401	282,744	318,087	353,430
1	1 1/16	.8866	39,888	79,776	119,664	159,552	199,440	239,328	279,216	319,104	358,992	398,880
1 1/16	1⅛	.9940	44,730	89,460	134,190	178,920	223,650	268,380	313,110	357,840	402,570	447,300
1⅛	1 3/16	1.1075	49,837	99,674	149,511	199,348	249,185	299,022	348,859	398,696	448,533	498,370
1 3/16	1¼	1.2271	55,219	110,438	165,657	220,876	276,095	331,314	386,533	441,752	496,971	552,190
1¼	1 5/16	1.3529	60,880	121,760	182,640	243,520	304,400	365,280	426,160	487,040	547,920	608,800
1 5/16	1⅜	1.4848	66,816	133,632	200,448	267,264	334,080	400,896	467,712	534,528	601,344	668,160
1⅜	1 7/16	1.6229	73,030	146,060	219,090	292,120	365,150	438,180	511,210	584,240	657,270	730,300

NOTE: In calculating the strength of rivets in the above tables, the diameter of the driven rivet, or in other words, the diameter of the hole, has been used in all cases

WEIGHT OF CIRCULAR BOILER HEADS.

Diameter in Inches	THICKNESS IN INCHES														
	1/8	3/16	1/4	5/16	3/8	7/16	1/2	9/16	5/8	11/16	3/4	13/16	7/8	15/16	1
16	7	11	14	18	22	25	29	......	......	......	......	......	......	......	......
17	8	12	16	20	24	28	33	......	......	......	......	......	......	......	......
18	9	14	18	23	27	32	36	......	......	......	......	......	......	......	......
19	10	15	20	25	30	36	42	......	......	......	......	......	......	......	......
20	11	17	23	28	34	39	45	......	......	......	......	......	......	......	......
21	12	19	25	31	37	43	50	......	......	......	......	......	......	......	......
22	14	20	27	34	41	48	55	......	......	......	......	......	......	......	......
23	15	22	30	37	45	52	60	......	......	......	......	......	......	......	......
24	16	24	32	41	49	57	65	......	......	......	......	......	......	......	......
25	18	26	35	44	53	62	70	......	......	......	......	......	......	......	......
26	19	29	38	47	57	67	76	......	......	......	......	......	......	......	......
27	21	31	41	51	62	72	82	......	......	......	......	......	......	......	......
28	22	33	44	55	66	77	88	......	......	......	......	......	......	......	......
29	24	36	47	59	71	83	95	......	......	......	......	......	......	......	......
30	25	38	51	63	76	89	101	......	......	......	......	......	......	......	......
31	27	41	54	68	81	95	108	......	......	......	......	......	......	......	......
32	29	43	58	72	86	101	115	......	......	......	......	......	......	......	......
33	31	46	61	76	92	108	122	......	......	......	......	......	......	......	......
34	33	49	65	81	98	114	130	......	......	......	......	......	......	......	......
35	35	52	69	86	103	121	138	......	......	......	......	......	......	......	......
36	37	55	73	91	109	128	146	162	178	194	210	......	......	......	......
37	39	58	77	96	116	136	154	172	190	207	225	......	......	......	......
38	41	61	81	102	123	142	163	182	201	220	239	......	......	......	......
39	43	64	86	107	128	150	171	193	213	233	254	......	......	......	......
40	45	68	90	113	135	158	180	203	225	247	269	......	......	......	......
41	47	71	95	118	142	166	189	213	237	259	283	......	......	......	......
42	50	75	99	124	149	174	199	223	248	272	297	......	......	......	......
43	52	78	104	130	156	182	208	234	260	286	313	......	......	......	......
44	55	82	109	136	164	191	218	245	273	301	329	......	......	......	......
45	57	86	114	143	171	200	228	257	285	315	344	......	......	......	......
46	60	89	119	149	179	209	239	268	299	328	358	......	......	......	......
47	62	93	124	156	187	218	249	280	311	342	374	......	......	......	......
48	65	97	130	162	195	227	259	292	324	357	390	......	......	......	......
49	68	101	135	169	203	237	270	304	338	372	406	......	......	......	......
50	......	106	141	176	211	246	281	317	352	387	422	......	......	......	......
51	......	110	147	183	220	256	293	330	366	403	439	......	......	......	......
52	......	114	152	190	228	266	304	343	381	419	457	......	......	......	......
53	......	119	158	198	237	277	316	356	395	435	474	......	......	......	......
54	......	123	164	205	246	287	328	369	410	451	492	......	......	......	......

Heads below heavy line will run heavier than the weight given.

WEIGHT OF CIRCULAR BOILER HEADS.

Diameter in Inches	THICKNESS IN INCHES														
	1/8	3/16	1/4	5/16	3/8	7/16	1/2	9/16	5/8	11/16	3/4	13/16	7/8	15/16	1
55	......	128	170	213	255	298	341	383	426	468	511	......	......	......	......
56	......	132	177	221	265	309	353	397	441	485	530	......	......	......	......
57	......	137	183	229	274	320	366	412	457	503	549	......	......	......	......
58	......	142	189	237	284	331	379	426	473	521	568	......	......	......	......
59	......	147	196	245	294	343	392	441	490	539	588	......	......	......	......
60	......	......	204	254	305	356	408	457	510	561	612	......	......	......	......
61	......	......	212	264	317	370	423	476	529	582	635	......	......	......	......
62	......	......	219	273	328	382	437	492	546	601	656	......	......	......	......
63	......	......	226	282	338	395	451	508	564	620	677	......	......	......	......
64	......	......	233	291	349	407	466	524	582	640	699	......	......	......	......
65	......	......	240	300	360	420	480	540	600	661	721	......	......	......	......
66	......	......	248	310	371	433	495	557	619	681	743	......	......	......	......
67	......	......	255	319	383	447	510	574	638	702	766	......	......	......	......
68	......	......	263	329	394	460	526	591	657	723	789	......	......	......	......
69	......	......	271	338	406	474	541	609	677	744	812	......	......	......	......
70	......	......	279	348	418	487	557	627	696	766	836	......	......	......	......
71	......	......	287	358	430	502	573	645	717	788	860	......	......	......	......
72	......	......	295	368	442	516	589	663	737	810	884	......	......	......	......
73	......	......	303	379	454	530	606	682	757	833	909	......	......	......	......
74	......	......	311	389	467	545	623	700	778	856	934	......	......	......	......
75	......	......	320	400	480	560	640	719	799	879	959	......	......	......	......
76	......	......	328	410	492	575	657	739	821	903	985	1067	1149	1231	1313
77	......	......	337	421	506	590	674	758	843	927	1011	1095	1180	1264	1348
78	......	......	346	432	519	605	692	778	865	951	1038	1124	1210	1297	1383
79	......	......	355	443	532	621	710	798	887	976	1064	1153	1242	1330	1419
80	......	......	367	458	550	641	733	830	917	1008	1100	1192	1283	1384	1476
81	......	......	378	473	568	662	757	851	946	1041	1135	1230	1324	1419	1514
82	......	......	388	485	582	679	776	872	969	1066	1163	1260	1357	1454	1551
83	......	......	397	497	596	695	795	894	993	1093	1192	1291	1390	1490	1589
84	......	......	407	509	610	712	814	916	1017	1119	1221	1322	1424	1526	1628
85	......	......	417	521	625	729	833	937	1042	1146	1250	1354	1458	1562	1667
86	......	......	427	533	640	746	853	960	1066	1173	1280	1386	1493	1599	1706
87	......	......	437	546	655	764	873	982	1091	1200	1309	1419	1528	1637	1746
88	......	......	447	558	670	782	893	1005	1117	1228	1340	1452	1563	1675	1787
89	......	......	457	571	685	799	914	1028	1142	1256	1370	1485	1599	1712	1827
90	......	......	467	584	701	817	934	1051	1168	1285	1401	1518	1635	1752	1869

Heads below heavy line will run heavier than the weight given.

WEIGHT OF CIRCULAR BOILER HEADS.

Diameter in Inches	THICKNESS IN INCHES														
	1/8	3/16	1/4	5/16	3/8	7/16	1/2	9/16	5/8	11/16	3/4	13/16	7/8	15/16	1
91				597	716	836	955	1075	1194	1313	1433	1552	1671	1791	1910
92				610	732	854	976	1098	1220	1342	1463	1586	1708	1830	1953
93				623	748	873	998	1122	1247	1372	1496	1621	1746	1870	1995
94				637	764	892	1019	1146	1274	1401	1529	1656	1783	1911	2038
95				651	781	911	1041	1171	1301	1431	1561	1692	1822	1952	2082
96				664	797	930	1063	1196	1329	1462	1594	1727	1860	1993	2126
97				678	814	950	1085	1221	1357	1492	1628	1764	1899	2035	2171
98				692	831	969	1108	1246	1384	1523	1662	1800	1939	2077	2215
99				707	848	989	1130	1272	1413	1554	1696	1827	1978	2120	2261
100				721	865	1000	1153	1297	1442	1585	1730	1874	2018	2163	2307
101				735	882	1020	1175	1322	1470	1616	1764	1910	2058	2205	2352
102				750	899	1048	1197	1347	1498	1647	1798	1947	2098	2248	2398
103				764	916	1068	1219	1372	1525	1678	1832	1984	2138	2291	2444
104				779	933	1087	1241	1395	1551	1706	1862	2023	2178	2333	2488
105				794	950	1107	1263	1420	1577	1735	1894	2056	2215	2374	2533
106				808	967	1127	1286	1445	1605	1765	1926	2090	2254	2416	2578
107				822	984	1147	1309	1470	1632	1796	1960	2126	2293	2459	2624
108				836	1001	1166	1331	1496	1660	1827	1994	2162	2331	2499	2666
109				850	1018	1186	1354	1522	1689	1858	2027	2197	2369	2539	2710
110				865	1035	1205	1376	1547	1717	1888	2060	2232	2405	2580	2753
111				879	1052	1225	1398	1572	1745	1919	2094	2268	2443	2620	2795
112				894	1069	1245	1420	1596	1773	1950	2128	2306	2484	2663	2842
113				909	1086	1264	1442	1620	1800	1980	2162	2341	2522	2704	2886
114				923	1103	1284	1464	1645	1827	2010	2194	2377	2560	2745	2930
115				938	1120	1303	1486	1670	1855	2040	2226	2412	2598	2785	2973

Heads below heavy line will run heavier than the weight given.

ESTIMATING THE WEIGHT OF STEEL PLATES.

The table of the weight of steel plates is based upon the assumption that one cubic inch of rolled steel weighs .2833 pounds and that this is increased, by the springage of the rolls, by a certain percentage depending upon the width and thickness of the plate and which is assumed to be in accordance with a table given herewith:

PERCENTAGE OF INCREASE OF DENSITY OF ROLLED STEEL PLATES.

THICKNESS OF PLATE. Inch.	WIDTH OF PLATE.			
	Up to 75 Inches. Per cent.	75 to 100 Inches. Per cent.	100 to 115 Inches. Per cent.	Over 115 Inches. Per cent.
1/4	10	14	18	. .
5/16	8	12	16	. .
3/8	7	10	13	17
7/16	6	8	10	13
1/2	5	7	9	12
9/16	4½	6½	8½	11
5/8	4	6	8	10
Over 5/8	3½	5	6½	9

To illustrate the method used in calculating the table following this article, we will calculate the estimated weight of a 1/4″ plate 38″ wide and 138″ long. Multiplying these three dimensions together gives us the number of cubic inches of steel in the plate as follows: 1/4 × 38 × 138 = 1311. As the increase in density is 10 per cent for this size plate, according to the table, we add 10 per cent to the weight of one cubic inch of steel (.2833) as follows: .2833 × .10 = .02833 and .2833 + .02833 = .31163 — the weight in pounds of one cubic inch of steel in this particular plate. Multiplying the number of cubic inches in the plate (1311) by this gives us the weight of the plate in pounds as follows: 1311 × .3116 = 408.55 = weight of plate in pounds. Taking the nearest unit makes it 409, which agrees with the table, but no allowance has been made here for springage of the rolls and in using this table the percentage given in the table above must be added. By so doing we get a result which will agree very closely with the table.

WEIGHT PER SQUARE FOOT OF ROLLED STEEL PLATE NOT ALLOWING FOR SPRINGAGE OF ROLLS.

Thickness of Plate, inches.	Pounds per Sq. Foot.	Thickness of Plate, inches	Pounds per Sq. Foot.
		$\frac{5}{8}$	25.497
$\frac{1}{32}$	1.2748	$\frac{11}{16}$	28.047
$\frac{1}{16}$	2.5496	$\frac{3}{4}$	30.596
$\frac{3}{32}$	3.8244	$\frac{13}{16}$	33.146
$\frac{1}{8}$	5.0992	$\frac{7}{8}$	35.696
$\frac{5}{32}$	6.3740	$\frac{15}{16}$	38.245
$\frac{3}{16}$	7.6488	1	40.795
$\frac{7}{32}$	8.9236	$1\frac{1}{16}$	43.344
$\frac{1}{4}$	10.199	$1\frac{1}{8}$	45.894
$\frac{9}{32}$	11.474	$1\frac{3}{16}$	48.444
$\frac{5}{16}$	12.749	$1\frac{1}{4}$	50.993
$\frac{11}{32}$	14.024	$1\frac{5}{16}$	53.543
$\frac{3}{8}$	15.299	$1\frac{3}{8}$	56.092
$\frac{13}{32}$	16.574	$1\frac{7}{16}$	58.642
$\frac{7}{16}$	17.849	$1\frac{1}{2}$	61.192
$\frac{15}{32}$	19.124	$1\frac{3}{4}$	71.390
$\frac{1}{2}$	20.398	$1\frac{7}{8}$	76.489
$\frac{9}{16}$	22.948	2	81.588

The weight per square foot of ¼″ plate as given by this table is 10.199 and in a piece of 38″ × 138″, according to the first table, the increase would be 10 per cent, making the increase 10.199 × .10 = 1.0199. Adding the increase to the weight per square foot given in the table makes it 11.2189 as follows: 10.199 + 1.0199 = 11.2189. The area of the plate in square feet is obtained by multiplying its width by its length in inches and dividing by 144 the number of square inches in a square foot, as follows: 38 × 138 = 5244 = number of square inches in plate. Dividing this by 144 gives us the area of the plate in square feet, as follows: 5244 ÷ 144 = 36.417 = number of square feet in plate. Multiplying this by the weight per square foot as calculated above (11.219) gives us the weight of the plate as follows: 36.417 × 11.219 = 408.56 = weight of plate in pounds. This agrees practically with the table given below and the weight calculated by the other method at the beginning of this article.

WEIGHT OF STEEL BOILER PLATES.

¼″ PLATE.

Size.	Weight, Pounds.	Size.	Weight, Pounds.
26 × 120	243	50 × 138	538
26 × 138	280	54 × 120	505
30 × 120	280	57 × 138	613
30 × 138	323	57 × 143	635
36 × 120	337	57 × 156	693
36 × 138	387	60 × 98	458
38 × 120	355	60 × 120	561
38 × 138	409	60 × 138	645
40 × 120	374	64¾ × 138	696
40 × 138	430	64¾ × 143	721
40 × 143	446	64¾ × 156	787
42 × 120	393	64¾ × 175	883
42 × 138	452	64¾ × 194	979
43 × 138	462	72 × 98	550
43 × 143	479	72 × 120	673
43 × 156	523	72 × 138	774
44 × 120	411	72 × 143	802
44 × 138	473	72 × 156	875
46 × 120	430	72 × 175	982
46 × 138	495	72 × 194	1088
48 × 120	449	84 × 98	665
48 × 138	516	84 × 120	814
49 × 98	374	84 × 138	936
49 × 138	552	84 × 143	970
49 × 143	572	84 × 156	1058
49 × 156	624	84 × 175	1187
50 × 120	467	84 × 194	1316

$\frac{5}{16}$″ PLATE.

26 × 80	199	49 × 143	670
26 × 90	223	49 × 156	731
26 × 99	246	49 × 175	820
26 × 120	298	49 × 194	909
26 × 138	343	50 × 120	574
30 × 80	229	50 × 138	660
30 × 90	258	54 × 120	620
30 × 99	284	57 × 80	436
30 × 120	344	57 × 90	490
30 × 138	396	57 × 99	540
36 × 80	275	57 × 138	752
36 × 90	310	57 × 143	779
36 × 99	341	57 × 156	850
36 × 120	413	57 × 175	954
36 × 138	475	57 × 194	1057
38 × 80	291	60 × 120	688
38 × 90	327	60 × 138	792
38 × 99	360	64¾ × 90	557
38 × 120	435	64¾ × 99	613
38 × 138	501	64¾ × 138	854
40 × 80	306	64¾ × 143	885
40 × 90	344	64¾ × 156	966
40 × 99	379	64¾ × 175	1083

5/16″ PLATE.

Size.	Weight, Pounds.	Size.	Weight, Pounds.
40×120	459	64¾×194	1201
40×138	528	72½× 99	686
42×120	482	72½×120	832
42×138	554	72½×138	957
43× 80	329	72½×143	991
43× 90	370	72½×156	1081
43× 99	407	72½×175	1213
43×138	567	72½×194	1345

⅜″ PLATE.

30 ×120	499	72½×108½	894
36 ×120	491	72½×118	972
36 ×138	565	72½×212½	1751
40 ×120	546	72½×231½	1908
40 ×138	627	84 ×108½	1065
44 ×120	600	84 ×118	1158
44 ×138	690	84 ×194	1904
48 ×120	655	84 ×212½	2086
48 ×138	753	84 ×231½	2282
50 ×120	682	96 ×108½	1217
50 ×138	784	96 ×118	1324
54 ×120	737	96 ×194	2176
54 ×138	847	96 ×212½	2384
60 ×120	818	96 ×231½	2597
60 ×138	941	107½×108½	1400
64¾×118	869	107½×118	1523
64¾×194	1428	107½×194	3504
64¾×212½	1564	107½×212½	2742
64¾×231½	1704	107½×231½	2988
65¾×108¼	799		

7/16″ PLATE.

36×120	568	60×120	946
40×120	631	72×120	1135
48×120	757		

½″ PLATE.

36×120	643	60×120	1071
40×120	714	72×120	1285
48×120	857		

¾″ PLATE.

36×120	950	60×120	1583
40×120	1056	72×120	1900
48×120	1267		

⅞″ PLATE.

40×112	1149	53×133	1809
40×154½	1996	53×154	2094
53×112	1523		

TABLES OF WIDTH, LENGTH AND THICKNESS OF PLATES THAT CAN BE MADE FOR BOILER PURPOSES, ALSO DIAMETER OF HEADS.

Thickness.	Diameter of Heads.	Width and Length of Plate.	
		Width.	Length.
¼	115	114″	200″
$\frac{5}{16}$	120	126″	240″
⅜	126	140″	180″
$\frac{7}{16}$	126	140″	180″
½	126	144″	180″
⅝	126	144″	180″

Longer lengths can be made but would be less in width.

Rules adopted by the Association of American Steel Manufacturers: "When ordering plates 12½ pounds to square foot or heavier, up to 100 inches wide, by weight, they shall not average more than 2½ per cent above or below the theoretical weight, when 100 inches and over the limit is 5 per cent."

TABLE OF ALLOWANCES FOR OVERWEIGHT FOR RECTANGULAR PLATE WHEN ORDERED BY GAUGE.

Thickness of Plate.	WIDTH OF PLATE.				
	Up to 50 inches.	50 inches and above.	Up to 75 inches.	75 inches to 100 in.	over 100 inches.
⅛ up to $\frac{5}{32}$	10 per ct.	15 per ct.	....	....	....
$\frac{5}{32}$ up to $\frac{3}{16}$	8½ " "	12½" "	....	....	....
$\frac{3}{16}$ up to ¼	7 " "	10 " "	....	....	....
¼	....	....	10 per ct.	14 per ct.	18 per ct.
$\frac{5}{16}$	....	....	8 " "	12 " "	16 " "
⅜	....	....	7 " "	10 " "	13 " "
$\frac{7}{16}$	....	....	6 " "	8 " "	10 " "
½	....	....	5 " "	7 " "	9 " "
$\frac{9}{16}$	....	....	4½ " "	6½ " "	8½ " "
⅝	....	....	4 " "	6 " "	8 " "
over ⅝	....	....	3½ " "	5 " "	6½ " "

DOME PLATE ALLOWANCES.

Diameter of Domes.	Diameter of Shells.								
	30	36	42	48	54	60	66	72	84
20	6¼	5½	5¼	...	...	...	...	...	...
22	7¼	6¼	5¾	5¼	...	...	...	...	...
24	8½	7¼	6½	5¾	5½	...	...	...	...
26	...	8¼	7¼	6½	6	...	...	...	...
28	...	9½	8	7¼	6½	6	...	...	...
30	...	10¾	9	8	7¼	6¾	6¼	5¾	5¼
32	...	...	10	8¾	8	7¼	6¾	6¼	5¾
34	...	...	...	9¾	8¾	8	7¼	7	6
36	...	...	...	10¾	9½	8½	8	7¼	6½
38	...	...	...	...	10¼	9½	8¾	8	7
40	...	...	...	...	...	10¼	9½	9¾	7½
42	...	...	...	...	...	11¼	10¼	10	8
44	...	...	...	...	...	...	11	10½	9
46	...	...	...	...	...	...	12¼	10¾	9½
48	...	...	...	...	...	...	13	11½	10

The above table is based on single riveting, and the allowances named are those commonly used in figuring the finished length of domes. For double riveting add 2 inches.

STANDARD BOILER TUBES.

Outside Diameter.	Inside Diameter.	THICKNESS.		CIRCUMFERENCE.		TRANSVERSE AREAS.			EXTERNAL HEATING SURFACE.		Nominal Weight per Foot.
			Nearest	External.	Internal.	External.	Internal.	Metal.	Per Foot of Tube Length.	Tube Length, per Sq. Foot.	
Inches.	Inches.	Inches.	B. W. G.	Inches.	Inches.	Square In.	Square In.	Square In.	Sq. Feet.	Feet.	Pounds.
2	1.810	.095	13	6.283	5.686	3.1416	2.5730	.5686	.5236	1.909	1.91
2¼	2.060	.095	13	7.069	6.472	3.9761	3.3329	.6432	.5891	1.698	2.16
2½	2.282	.109	12	7.854	7.169	4.9087	4.0899	.8188	.6545	1.528	2.75
2¾	2.532	.109	12	8.639	7.954	5.9396	5.0349	.9047	.7200	1.389	3.04
3	2.782	.109	12	9.425	8.740	7.0686	6.0787	.9899	.7854	1.273	3.33
3¼	3.010	.120	11	10.210	9.456	8.2958	7.1157	1.1801	.8508	1.175	3.96
3½	3.260	.120	11	10.996	10.242	9.6211	8.3469	1.274	.9163	1.091	4.28
3¾	3.510	.120	11	11.781	11.027	11.045	9.6762	1.369	.9818	1.018	4.60
4	3.732	.134	10	12.566	11.724	12.566	10.939	1.627	1.0472	.955	5.47
4½	4.232	.134	10	14.137	13.295	15.904	14.066	1.838	1.1781	.849	6.17
5	4.704	.148	9	15.708	14.778	19.635	17.379	2.256	1.3090	.764	7.58
6	5.670	.165	8	18.850	17.813	28.274	25.249	3.025	1.5708	.637	10.16
7	6.670	.165	8	21.991	20.954	38.485	34.941	3.544	1.8326	.546	11.90
8	7.670	.165	8	25.133	24.096	50.265	46.204	4.061	2.0944	.477	13.65

Rule to find number of square feet of heating surface in tubes: Multiply the number of tubes by the diameter of a tube in inches and by its length in feet, and by .2618 constant.

LEGEND:

D = Tubes 4″
L = Length = 16′
N = Number = 44
C = Constant = .2618

FORMULA:

N × D × L × .2618 (constant) = heating surface

EXAMPLE:

$$\begin{array}{rl} 44 & = \text{number of tubes} \\ 4 & = \text{diameter in inches} \\ \hline 176 & \\ 16 & = \text{length in feet} \\ \hline 1056 & \\ 176 & \\ \hline 2816 & \\ .2618 & = \text{constant} \\ \hline 22528 & \\ 2816 & \\ 16896 & \\ 5632 & \\ \hline 737.2288 & = \text{total square feet of heating surface in 44 4″ tubes.} \end{array}$$

HEATING SURFACE OF BOILER TUBES.

Diameter X 3.1416 = circumference X 12 = number of square inches in tube one foot of length ÷ 144 = number of square feet (in decimals) one foot of length.

EXAMPLE:

2 inch tube one foot in length:

$$\frac{2 \times 3.1416 = 6.2832 \times 12 = 75.3984}{144} = .5236 \text{ of a square foot}$$

TABLE.

Diam. in.	Multipl'r	Diam. in.	Multipl'r	Diam. in.	Multipl'r	Diam. in.	Multipl'r
1	.2618	11½	3.0107	32	8.3776	53	13.8754
1¼	.3272	11¾	3.0761	32½	8.5085	53½	14.0063
1½	.3927	12	3.1416	33	8.6394	54	14.1372
1¾	.4581	12½	3.2725	33½	8.7703	54½	14.2681
2	.5236	13	3.4037	34	8.9012	55	14.399
2¼	.589	13½	3.5343	34½	9.0321	55½	14.5299
2½	.6545	14	3.6652	35	9.163	56	14.6608
2¾	.7199	14½	3.7961	35½	9.2939	56½	14.7917
3	.7854	15	3.927	36	9.4248	57	14.9226
3¼	.8508	15½	4.0579	36½	9.5557	57½	15.0536
3½	.9163	16	4.1888	37	9.6866	58	15.1844
3¾	.9817	16½	4.3197	37½	9.8175	58½	15.3153
4	1.0472	17	4.4506	38	9.9844	59	15.4462
4¼	1.1126	17½	4.5815	38½	10.0793	59½	15.5771
4½	1.1781	18	4.7124	39	10.2102	60	15.708
4¾	1.2435	18½	4.8433	39½	10.3411	60½	15.8389
5	1.309	19	4.9742	40	10.472	61	15.9698
5¼	1.3744	19½	5.1051	40½	10.6029	61½	16.1007
5½	1.4399	20	5.236	41	10.7338	62	16.2316
5¾	1.5053	20½	5.3669	41½	10.8647	62½	16.3625
6	1.5708	21	5.4978	42	10.9956	63	16.4934
6¼	1.6362	21½	5.6287	42½	11.1265	63½	16.6243
6½	1.7017	22	5.7596	43	11.2574	64	16.7552
6¾	1.7671	22½	5.8905	43½	11.3883	64½	16.8861
7	1.8326	23	6.0214	44	11.5192	65	17.017
7¼	1.8980	23½	6.1523	44½	11.6501	65½	17.1479
7½	1.9335	24	6.2832	45	11.781	66	17.2788
7¾	2.0289	24½	6.4141	45½	11.9119	66½	17.4097
8	2.0944	25	6.545	46	12.0428	67	17.5406
8¼	2.0598	25½	6.6759	46½	12.1735	67½	17.6715
8½	2.2253	26	6.8034	47	12.3045	68	17.8024
8¾	2.2907	26½	6.9377	47½	12.4355	68½	17.9333
9	2.3562	27	7.0686	48	12.5664	69	18.0642
9¼	2.4216	27½	7.1995	48½	12.6973	69½	18.1951
9½	2.4872	28	7.3384	49	12.8282	70	18.326
9¾	2.5525	28½	7.4614	49½	12.9591	70½	18.4569
10	2.618	29	7.5913	50	13.09	71	18.5868
10¼	2.6834	29½	7.7231	50½	13.2209	71½	18.7187
10½	2.7489	30	7.8554	51	13.3518	72	18.8496
10¾	2.8143	30½	7.9849	51½	13.4827	78	20.3370
11	2.8798	31	8.1158	52	13.6136	84	21.9912
11¼	2.9452	31½	8.2467	52½	13.7445	96	25.1328

APPROXIMATE WEIGHT OF ROUND BRACES WITH FLAT ENDS.

Length of Braces, inches	Diameter of Braces, inches	SIZE OF ENDS.		Weight, lbs.
		Width, inches	Thickness, in.	
14	1	2¼	½	7
16	1	2¼	½	7¼
18	1	2¼	½	7½
20	1	2¼	½	8
22	1	2¼	½	8½
24	1	2¼	½	9
26	1	2¼	½	9½
28	1	2¼	½	10
30	1	2¼	½	10½
32	1	2¼	½	11
34	1	2¼	½	11½
36	1	2¼	½	12
38	1	2¼	½	12½
40	1	2¼	½	13
42	1	2¼	½	13½
44	1	2¼	½	14
46	1	2¼	½	14½
48	1	2¼	½	15
50	1	2¼	½	15½
52	1	2¼	½	16
54	1	2¼	½	16½
56	1	2¼	½	17
58	1	2¼	½	17½
60	1	2¼	½	18
14	1⅛	2¼	⅝	7½
16	1⅛	2¼	⅝	8
18	1⅛	2¼	⅝	8½
20	1⅛	2¼	⅝	9
22	1⅛	2¼	⅝	10
24	1⅛	2¼	⅝	11
26	1⅛	2¼	⅝	12
28	1⅛	2¼	⅝	13
30	1⅛	2¼	⅝	14
32	1⅛	2¼	⅝	15
34	1⅛	2¼	⅝	16
36	1⅛	2¼	⅝	17
38	1⅛	2¼	⅝	17½
40	1⅛	2¼	⅝	18
42	1⅛	2¼	⅝	18½
44	1⅛	2¼	⅝	19
46	1⅛	2¼	⅝	19½
48	1⅛	2¼	⅝	20
50	1⅛	2¼	⅝	21
52	1⅛	2¼	⅝	22
54	1⅛	2¼	⅝	23
56	1⅛	2¼	⅝	24
58	1⅛	2¼	⅝	25
60	1⅛	2¼	⅝	26

NUMBER MODERN FORMED BRACES COMMONLY USED IN STANDARD TUBULAR BOILERS.

Length of Brace.	Diameter of Shell.							
	36	42	44	54	60	66	72	84
30	6	6	8	10	10	10	12	16
42	2	4	. .	. .	. .	. .	. .	. .
48	. .	. .	4	6	6	8	8	10
60	. .	. .	. .	. .	4	. .	. .	. .
72	. .	. .	. .	. .	. .	4	4	6

Under the diameter of each shell will be found the number of each length of brace generally used. The thickness of brace varies with thickness of shell.

METALS.

WEIGHT OF SUPERFICIAL FOOT.

Thickness.	W Iron.	C Iron.	Steel.	Copper.	Brass.	Lead.	Zinc.
Inch.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
1/16	2.52	2.34	2.55	2.89	2.73	3.71	2.34
⅛	5.05	4.69	5.10	5.78	5.47	7.42	4.69
3/16	7.58	7.03	7.66	8.67	8.20	11.13	7.03
¼	10.10	9.38	10.21	11.56	10.94	14.83	9.38
5/16	12.63	11.72	12.76	14.45	13.67	18.54	11.72
⅜	15.16	14.06	15.31	17.34	16.41	22.25	14.06
7/16	17.68	16.41	17.87	20.23	19.14	25.96	16.41
½	20.21	18.75	20.42	23.13	21.88	29.67	18.75
⅝	25.27	23.44	25.52	28.91	27.34	37.08	23.44
¾	30.31	28.13	30.63	34.69	32.81	44.50	28.13
⅞	35.37	32.81	35.73	40.47	38.28	51.92	32.81
1	40.42	37.50	40.83	46.25	43.75	59.33	37.50

BIRMINGHAM GAUGE.			
No. of Gauge.	Thickness, Inches.	Weight.	
		Iron.	Steel.
0000	.454	18,22	18.46
000	.425	17.05	17.28
00	.38	15.25	15.45
0	.34	13.64	13.82
1	.3	12.04	12.20
2	.284	11.40	11.55
3	.259	10.39	10.53
4	.238	9.55	9.68
5	.22	8.83	8.95
6	.203	8.15	8.25
7	.18	7.22	7.32
8	.165	6.62	6.71
9	.148	5.94	6.02
10	.134	5.38	5.45
11	.12	4.82	4.88
12	.109	4.37	4.43
13	.095	3.81	3.86
14	.083	3.33	3.37
15	.072	2.89	2.93
16	.065	2.61	2.64
17	.058	2.33	2.36
18	.049	1.97	1.99
19	.042	1.69	1.71
20	.035	1.40	1.42
21	.032	1.28	1.30
22	.028	1.12	1.14
23	.025	1.00	1.02
24	.022	.883	.895
25	.02	.803	.813
26	.018	.722	.732
27	.016	.642	.651
28	.014	.562	.569

U. S. STANDARD GAUGE.			
No. of Gauge.	THICKNESS, IN.		Weight, Iron.
	Fractions.	Decimals.	
0000000	$\frac{1}{2}$	.5	20.
000000	$\frac{15}{32}$	.468	18.75
00000	$\frac{7}{16}$	.437	17.50
0000	$\frac{13}{32}$	.406	16.25
000	$\frac{3}{8}$	.375	15.
00	$\frac{11}{32}$	.343	13.75
0	$\frac{5}{16}$	.312	12.50
1	$\frac{9}{32}$	.281	11.25
2	$\frac{17}{64}$	.265	10.625
3	$\frac{1}{4}$	.25	10.
4	$\frac{15}{64}$	.234	9.375
5	$\frac{7}{32}$	.218	8.75
6	$\frac{13}{64}$	.203	8.125
7	$\frac{3}{16}$	.187	7.5
8	$\frac{11}{64}$	.171	6.875
9	$\frac{5}{32}$	.156	6.25
10	$\frac{9}{64}$	.140	5.625
11	$\frac{1}{8}$	.125	5.
12	$\frac{7}{64}$	.109	4.375
13	$\frac{3}{32}$	.093	3.75
14	$\frac{5}{64}$	.078	3.125
15	$\frac{9}{128}$	.070	2.8125
16	$\frac{1}{16}$	.062	2.5
17	$\frac{9}{160}$	.056	2.25
18	$\frac{1}{20}$	.05	2.
19	$\frac{7}{160}$	.043	1.75
20	$\frac{3}{80}$	.037	1.50
21	$\frac{11}{320}$	.034	1.375
22	$\frac{1}{32}$	.031	1.25
23	$\frac{9}{320}$	.028	1.125
24	$\frac{1}{40}$	.025	1.
25	$\frac{7}{320}$	.021	.875
26	$\frac{3}{160}$	.018	.75
27	$\frac{11}{640}$	.017	.6875
28	$\frac{1}{64}$	.015	.625

The U. S. Standard is the one in common use.

To Convert Weight of Metals Multiply By Following Constants:

Wrought iron into cast iron	× .928
" " " steel	×1.014
" " " zinc	× .918
" " " brass	×1.082
" " " copper	×1.144
" " " lead	×1.468
Square iron into round	× .7854

Weight of Cast Iron Balls.

DIAMETER.	WEIGHT.	DIAMETER.	WEIGHT.	DIAMETER.	WEIGHT.
1	.136	5	17.04	9	99.40
1½	.460	5½	22.68	9½	116.90
2	1.09	6	29.45	10	136.35
2½	2.13	6½	37.44	10½	157.84
3	3.68	7	46.76	11	181.48
3½	5.84	7½	57.52	11½	207.37
4	8.72	8	69.81	12	235.62
4½	12.42	8½	83.73		

ANGLES.

Weights per Foot, Corresponding to Thickness Varying by 1/16 Inch, One Cubic Foot Weighing 480 lbs.

Sizes, inches.	⅛	3/16	¼	5/16	⅜	7/16	½	9/16	⅝	11/16	¾	13/16	⅞
Equal Legs.													
6 x 6	.....	.....	.....	.....	.....	16.75	19.14	21.53	23.92	26.31	28.70	31.10	33.50
5 x 5	.....	.....	.....	.....	12.00	14.28	16.56	18.84	21.13	23.42	25.71	26.85	28.00
4 x 4	.....	.....	.....	8.00	9.50	11.16	12.82	14.49	16.16	17.83	19.20	.....	.....
3½ x 3½	3.10	4.50	5.60	6.95	8.30	9.75	11.20	12.65	14.10	15.55	17.00	.....	.....
3¼ x 3¼	2.90	4.20	.....	.....	7.70	9.05	10.40	11.75	13.10	14.45	15.80	.....	.....
3 x 3	2.60	3.80	4.83	6.05	7.28	8.51	9.74	10.97	12.20	.....	.....	.....	.....
2¾ x 2¾	2.40	3.50	4.41	5.50	6.60	7.70	8.80	.....	.....	.....	.....	.....	.....
2½ x 2½	2.20	3.00	4.00	5.00	6.00	7.00	8.00	.....	.....	.....	.....	.....	.....
2¼ x 2¼	2.00	2.80	3.57	4.47	5.38	6.29	7.20	.....	.....	.....	.....	.....	.....
2 x 2	1.66	2.42	3.21	4.00	4.80	5.60	.....	.....	.....	.....	.....	.....	.....
1¾ x 1¾	1.45	2.13	2.84	3.56	4.28	5.00	.....	.....	.....	.....	.....	.....	.....
1½ x 1½	1.23	1.80	2.40	3.00	3.60	.....	.....	.....	.....	.....	.....	.....	.....
1¼ x 1¼	1.02	1.50	2.00	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
1 x 1	.79	1.20	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
¾ x ¾	.60	.90	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
Unequal Legs.													
6 x 4	.....	.....	.....	.....	12.00	14.44	16.38	18.32	20.26	22.20	24.15	26.10	28.00
6 x 3½	.....	.....	.....	.....	11.50	13.24	14.98	16.72	18.47	20.22	21.97	23.72	26.60
5 x 4	.....	.....	.....	.....	10.80	12.61	14.42	16.24	18.06	19.88	21.70	23.45	25.20
5 x 3½	.....	.....	.....	.....	10.20	11.26	13.72	15.49	17.26	19.03	20.40	.....	.....
5 x 3	.....	.....	.....	.....	9.50	11.16	12.82	14.49	16.16	17.83	19.50	.....	.....
4 x 3½	.....	.....	.....	.....	8.90	10.46	12.02	13.59	15.16	16.73	18.30	.....	.....
4 x 3	.....	.....	.....	.....	8.30	9.75	11.20	12.65	14.10	15.55	17.00	.....	.....
3½ x 2⅝	.....	.....	.....	.....	7.50	8.75	10.00	11.25	12.50	13.75	15.00	.....	.....
3½ x 3	.....	.....	.....	6.50	7.70	9.05	10.40	11.75	13.00	.....	.....	.....	.....
3½ x 2½	.....	.....	4.80	6.05	7.30	8.55	9.80	.....	.....	.....	.....	.....	.....
3 x 2½	.....	.....	4.40	5.55	6.70	7.85	8.80	.....	.....	.....	.....	.....	.....
3¼ x 2	.....	.....	4.20	5.31	6.37	7.43	8.50	.....	.....	.....	.....	.....	.....
3 x 2	.....	3.03	3.98	4.93	5.88	6.84	7.80	.....	.....	.....	.....	.....	.....
2½ x 2	.....	2.72	3.45	4.18	4.92	5.66	6.40	.....	.....	.....	.....	.....	.....
2 x 1½	.....	2.20	2.90	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....

WEIGHTS AND MEASUREMENTS OF STEEL "I" BEAMS.

Depth, Inches.	Min. Weight, lbs. per foot.	Inner Weights.	Max. Weight, lbs. per foot.	Min. Flange, inches.	Min. Web, inches.	Min. Area, square inches.
4	7.5	Vary by 1 lb.	10.5	2.66	.19	2.2
5	9.75	Vary by 2½ lbs.	14.75	3.00	.21	2.9
6	12.25	Vary by 2½ lbs.	17.25	3.33	.23	3.6
7	15.0	Vary by 2½ lbs.	20.0	3.66	.25	4.4
8	17.75	Vary by 2½ lbs.	25.25	4.00	.27	5.2
9	21.0	25 lbs. then vary by 5 lbs.	35.0	4.33	.29	6.3
10	25.0	Vary by 5 lbs.	40.0	4.66	.31	7.4
12	31.5	35 lbs. then vary by 5 lbs.	45.0	5.00	.35	9.3
12	40.0	Vary by 5 lbs.	55.0	5.25	.41	11.85
15	42.0	45 lbs. then vary by 5 lbs.	60.0	5.50	.46	12.5
15	60.0	Vary by 5 lbs.	80.0	6.00	.59	17.68

WEIGHTS AND MEASUREMENTS OF STEEL CHANNELS.

Depth, Inches.	Min. Weight, lbs. per foot.	Inner Weights.	Max. Weight, lbs. per fool.	Min. Flange, inches.	Min. Web, inches.	Min. Area, square inches.
4	5.25	Vary by 1 lb.	7.25	1.58	.18	1.6
5	6.5	Vary by 2½ lbs.	11.5	1.75	.19	2.0
6	8.0	Vary by 2½ lbs.	15.5	1.92	.20	2.4
7	9.75	Vary by 2½ lbs.	19.75	2.09	.21	2.9
8	11.25	Vary by 2½ lbs.	21.25	2.26	.22	3.4
9	13.25	15 lbs. then vary by 5 lbs.	25.0	2.43	.23	3.9
10	15.0	Vary by 5 lbs.	35.0	2.60	.24	4.5
12	20.5	25 lbs. then vary by 5 lbs.	40.0	2.94	.28	6.0
15	33.0	33 lbs. then vary by 5 lbs.	55.0	3.40	.40	9.9

PIPE AND PIPING.

Rule to find pressure allowed on a main steam pipe or header when thickness of pipe and diameter is known: From thickness of plate subtract the constant .1250, then multiply by one-sixth of tensile strength of plate and divide this product by diameter; the sum will be pressure allowed.

LEGEND:

T = Thickness of plate = .4850
C = Constant = .1250
T. S = Tensile strength = 60000
D = Diameter = 24″

FORMULA:

$$\frac{(T—.1250)\times(1/6\text{th of TS})}{D}=\text{pressure}$$

EXAMPLE:
.4850 = thickness of plate
.1250 = constant
.3600
10000 = 1/6 of tensile strength
diameter 24") 3600.0000 (150 lbs. pressure allowed
24
120
120

Rule to find thickness of material for a main, steel or iron, steam pipe or cylinder lap welded: Multiply pressure by diameter and divide by one-sixth of the tensile strength, and add .125

LEGEND:
P = pressure = 150 lbs.
D = diameter = 24"
T.S. = tensile strength = 60,000

FORMULA:

$$\frac{P \times D}{1/6 \text{ of T. S.}} + .125 = \text{thickness}$$

EXAMPLE:
150 = lbs. pressure
24" = diameter
600
300
1/6 of tensile strength = 10,000) 3600 00 (.36
3000 0 .125 added
600 00 .485 = thickness or 31/64 approximately
600 00

Rule to find thickness of plate for a 5" copper pipe: Multiply pressure by inside diameter of pipe and divide by constant 8000; add to quotient the constant .0625.

LEGEND:
P = pressure = 175
ID = inside diameter of pipe = .5
C = constant = 8000

FORMULA:

$$\frac{P \times ID}{C} + .0625 = \text{thickness of plate}$$

EXAMPLE:
175 = pressure
.5" = inside diameter of pipe
8000) 87.50000 (.109
80 00 .0625 = constant
75 00 .1715 = $\frac{11}{64}$ approximately
72 00
3 00

RADIATION OF DIFFERENT SIZES OF WROUGHT-IRON PIPE.

The following table gives the actual lengths of different sizes of pipe sufficient to make ten square feet of radiation:

1 inch Pipe, 28 lineal feet = 10 square feet radiation.
1¼ " " 24 " " = 10 " " "
1½ " " 20 " " = 10 " " "
2 " " 16 " " = 10 " " "
2½ " " 13 " " = 10 " " "
3 " " 11 " " = 10 " " "

Table of Expansion of Wrought-Iron Pipe.

Temperature of the Air when the Pipe is fitted.	Length of Pipe when fitted.	Length of Pipe When Heated to					
		160 Degrees.		180 Degrees.		200 Degrees.	
Degrees Fahr.	Feet.	Feet.	Inches.	Feet	Inches	Feet	Inches
0	100	100	1.28	100	1.44	100	1.60
32	100	100	1.02	100	1.18	100	1.34
64	100	100	.77	100	.93	100	1.09

Standard Flanges. Sizes: Threaded or Plain.

Size Pipe, Inches.	Diameter Flange.	Thickness of Flanges.	Equivalent to Cast Iron.
1- Inch	6- Inch	⅜-Inch	1½-Inch
1¼ "	6 "	⅜ "	1½ "
1½ "	6 "	⅜ "	1½ "
2 "	8 "	½ "	2 "
2½ "	9 "	½ "	2 "
3 "	9 "	½ "	2 "
3½ "	10 "	½ "	2 "
4 "	10 "	½ "	2 "
4½ "	10½ "	½ "	2 "
5 "	11½ "	½ "	2 "
6 "	12½ "	½ "	2 "
7 "	13½ "	½ "	2 "
8 "	15½ "	⅝ "	2¼ "
9 "	16½ "	⅝ "	2¼ "
10 "	17½ "	⅝ "	2¼ "
12 "	21 "	⅝ "	2¼ "

WROUGHT IRON WELDED STEAM, GAS AND WATER PIPE. TABLE OF STANDARD DIMENSIONS

DIAMETER.			Thickness.	CIRCUMFERENCE.		Internal Area.	Length of Pipe per square foot of External Surface.	Square feet of Surface per foot in Length.	Nominal Weight per foot.	Number of Threads per inch of Screw.	Inside Diameter.
Nominal Internal. Inches.	Actual External. Inches.	Actual Internal. Inches.	Inches.	External. Inches.	Internal. Inches.	Sq. Inches.	Feet.		Lbs.		Inches.
1/8	.405	.27	.068	1.272	.848	.0573	9.44	.106	.241	27	1/8
1/4	.54	.364	.088	1.696	1.144	.1041	7.075	.141	.42	18	1/4
3/8	.675	.494	.091	2.121	1.552	.1917	5.657	.177	.559	18	3/8
1/2	.84	.623	.109	2.639	1.957	.3048	4.547	.220	.837	14	1/2
3/4	.105	.824	.113	3.229	2.589	.5333	3.637	.275	1.115	14	3/4
1	1.315	1.048	.134	4.131	3.292	.8626	2.904	.344	1.668	11½	1
1¼	1.66	1.38	.14	5.215	4.335	1.496	2.301	.434	2.244	11½	1¼
1½	1.9	1.611	.145	5.969	5.061	2.038	2.01	.497	2.678	11½	1½
2	2.375	2.067	.154	7.461	6.494	3.356	1.608	.621	3.609	11½	2
2½	2.875	2.468	.204	9.032	7.753	4.784	1.328	.753	5.739	8	2½
3	3.5	3.067	.217	10.996	9.636	7.388	1.091	.916	7.536	8	3
3½	4.	3.548	.226	12.566	11.146	9.887	.955	1.047	9.001	8	3½
4	4.5	4.026	.237	14.137	12.648	12.73	.849	1.178	10.665	8	4
4½	5.	4.508	.246	15.708	14.162	15.961	.764	1.309	12.34	8	4½
5	5.563	5.045	.259	17.477	15.849	19.99	.687	1.456	14.502	8	5
6	6.625	6.065	.28	20.813	19.054	28.888	.577	1.734	18.762	8	6
7	7.625	7.023	.301	23.955	22.063	38.738	.501	1.996	23.271	8	7
8	8.625	7.982	.322	27.096	25.076	50.04	.433	2.256	28.177	8	8
9	9.625	8.937	.344	30.238	28.076	62.73	.397	2.520	33.701	8	9
10	10.75	10.019	.366	33.772	31.477	78.839	.355	2.814	40.065	8	10
11	11.75	11.	.375	36.914	34.558	95.033	.325	3.076	45.028	8	11
12	12.75	12.	.375	40.055	37.7	113.098	.299	3.338	48.985	8	12

Table Giving Diameter and Area at the Bottom of the Thread of Stay-Bolts and Stays of Useful Sizes for Calculating Their Strength, Etc.

Diam. of Stay Bolt	Thread per inch	Diam. at bottom of thread U. S. Standard	Area in sq. inches at bottom of thread U. S. Standard	Diam. at bottom of thread V thread	Area in sq. inches at bottom of thread V thread
⅝	12	.51675	.2097	.48067	.1815
11/16	12	.57925	.2635	.54317	.2317
¾	12	.64175	.3235	.60567	.2881
13/16	12	.70425	.3895	.66817	.3506
⅞	12	.76675	.4617	.73067	.4193
15/16	12	.82925	.5409	.79317	.4941
1	12	.89175	.6246	.85567	.5750
1 1/16	12	.95425	.7152	.91817	.6621
1⅛	12	1.01675	.8119	.98067	.7553
1 3/16	12	1.07925	.9148	1.04317	.8547
1¼	12	1.14175	1.0238	1.10567	.9601
1 5/16	12	1.20425	1.1390	1.16817	1.0718
1⅜	12	1.26675	1.2603	1.23067	1.1895
1½	12	1.39175	1.5213	1.35567	1.4434
1½	6	1.28350	1.2939	1.21134	1.1525
1⅝	5½	1.38882	1.5149	1.31010	1.3480
1¾	5	1.49020	1.7441	1.40350	1.5471
1⅞	5	1.61520	2.0490	1.52850	1.8349
2	4½	1.71134	2.3001	1.61512	2.0487
2⅛	4½	1.83634	2.6485	1.74012	2.3782
2¼	4½	1.96134	3.0213	1.86512	2.7321
2⅜	4	2.05025	3.3014	1.94200	2.9620
2½	4	2.17525	3.7163	2.06700	3.3556
2⅝	4	2.30025	4.1557	2.19200	3.7738
2¾	4	2.42525	4.6196	2.31100	4.1946
2⅞	3½	2.50386	4.9239	2.38015	4.4494
3	3½	2.62886	5.4278	2.50515	4.9290

TAP DRILLS.

THIS TABLE SHOWS THE DIFFERENT SIZES OF DRILL THAT SHOULD BE USED WHEN FULL THREAD IS TO BE TAPPED.

FOR MACHINE AND HAND TAP.

Diameter of Tap	No. of Threads to Inch	Size Drill for V Thread	Size Drill for U. S. Standard Thread	Size Drill for Whitworth Thread
9/32	16 18 20	5/32 5/32 11/64	3/16	3/16
6/32	16 18 20	3/16 13/64 13/64	....	....
5/16	16 18 ..	7/32 15/64 ..	1/4	15/64
11/32	16 18 ..	1/4 17/64 ..	....	....
3/8	14 16 18	1/4 9/32 9/32	9/32	9/32
13/32	14 16 18	19/64 21/64 21/64	....	....
7/16	14 16 ..	21/64 11/32 ..	11/32	11/32
15/32	14 16 ..	23/64 3/8 ..	....	....
1/2	12 13 14	3/8 25/64 25/64	13/32	3/8
17/32	12 13 14	13/32 27/64 27/64	....	....
9/16	12 14 ..	7/16 29/64 ..	7/16	7/16
19/32	12 14 ..	15/32 31/64 ..	....	....
5/8	10 11 12	15/32 1/2 1/2	1/2	1/2
21/32	10 11 12	1/2 17/32 17/32	....	....
3/4	10 11 12	19/32 5/8 5/8	5/8	5/8
25/32	10 11 12	5/8 21/32 21/32	....	....
7/8	9 10 ..	45/64 23/32 ..	23/32	23/32
29/32	9 10 ..	47/64 3/4 ..	....	....
1	8 ...	13/16 ...	27/32	27/32
1 1/32	8 ...	53/64 ...	....	....
1 1/8	7 8	29/32 15/16	15/16	15/16
1 5/32	7 8	15/16 31/32	....	....
1 1/4	7 ...	1 1/32 ...	1 1/16	1 1/16
1 9/32	7 ...	1 1/16 ...	....	....
1 3/8	6 ...	1 1/8 ...	1 5/16	1 5/32
1 13/32	6 ...	1 5/32 ...	....	....
1 1/2	6 ...	1 15/64 ...	1 9/32	1 9/32
1 17/32	6 ...	1 9/32 1 5/16	....	....
1 5/8	5 5 1/2	1 9/32 1 11/32	1 3/8	1 23/64
1 21/32	5 5 1/2	1 5/16 ...	....	....
1 3/4	5 ...	1 13/32 ...	1 1/2	1 1/2
1 25/32	5 ...	1 7/16 ...	....	....
1 7/8	4 1/2 5	1 17/32 1 17/32	1 5/8	1 37/64
1 29/32	4 1/2 5	1 9/16 1 17/32	....	....
2	4 1/2 ...	1 21/32 ...	1 23/32	1 45/64
....	....	... ...	....	....

PIPE TAPS.

Size Pipe	No. of Threads to the Inch	Diameter of Drill	Size Pipe	No. of Threads to the Inch	Diameter of Drill
⅛	27	21/64	3	8	3 5/16
¼	18	29/64	3½	8	3 13/16
⅜	18	19/32	4	8	4 5/16
½	14	23/32	4½	8	4⅞
¾	14	15/16	5	8	5⅜
1	11½	1 3/16	6	8	6 7/16
1¼	11½	1 15/32	7	8	7 7/16
1½	11½	1 23/32	8	8	8½
2	11½	2 3/16	9	8	9½
2½	8	2 11/16	10	8	10½

WEIGHTS OF ROUND AND SQUARE STEEL. PER LINEAL FOOT.

Size, inches.	Round, Weight, lbs.	Square, Weight, lbs.	Size, inches.	Round, Weight, lbs.	Square, Weight, lbs.
3/16	.094	.120	2⅛	12.06	15.36
¼	.167	.213	2¼	13.52	17.22
5/16	.261	.332	2⅜	15.07	19.19
⅜	.375	.478	2½	16.70	21.26
7/16	.511	.651	2⅝	18.41	23.44
½	.668	.851	2¾	20.21	25.73
9/16	.845	1.076	3	24.05	30.62
⅝	1.044	1.329	3¼	28.23	35.94
¾	1.503	1.914	3½	32.74	41.68
⅞	2.046	2.605	3¾	37.57	47.84
1	2.672	3.402	4	42.77	54.45
1⅛	3.382	4.306	4½	54.83	69.81
1¼	4.175	5.316	5	66.82	85.08
1⅜	5.052	6.432	5½	80.85	102.94
1½	6.012	7.655	6	96.22	122.51
1⅝	7.056	8.984	6½	112.92	143.78
1¾	8.183	10.419	7	131.97	166.75
1⅞	9.394	11.961	7½	150.34	191.42
2	10.69	13.61	8	171.04	217.78

WEIGHTS OF FLAT STEEL. PER LINEAL FOOT.

Width. Inches.	Thickness, inches.											
	1/16	⅛	3/16	¼	5/16	⅜	7/16	½	⅝	¾	⅞	1
1	.21	.43	.638	.850	1.06	1.28	1.49	1.70	2.12	2.55	2.98	.....
1⅛	.24	.48	.720	.955	1.20	1.43	1.68	1.92	2.39	2.87	3.35	3.88
1¼	.27	.53	.797	1.06	1.33	1.59	1.86	2.12	2.65	3.19	3.72	4.21
1⅜	.30	.59	.875	1.17	1.46	1.76	2.05	2.34	2.92	3.51	4.09	4.68
1½	.32	.64	.957	1.28	1.59	1.92	2.23	2.55	3.19	3.83	4.47	5.10
1⅝	.35	.69	1.04	1.38	1.73	2.08	2.42	2.77	3.46	4.15	4.84	5.53
1¾	.38	.75	1.11	1.49	1.86	2.23	2.60	2.98	3.72	4.47	5.20	5.95
2	.43	.85	1.28	1.70	2.12	2.55	2.98	3.40	4.25	5.10	5.95	6.80
2¼	.48	.96	1.44	1.91	2.39	2.87	3.35	3.83	4.78	5.75	6.69	7.65
2½	.53	1.06	1.59	2.12	2.65	3.19	3.72	4.25	5.31	6.38	7.44	8.50
2¾	.59	1.17	1.75	2.34	2.92	3.51	4.09	4.67	5.84	7.02	8.18	9.35
3	.64	1.28	1.91	2.55	3.19	3.83	4.46	5.10	6.38	7.65	8.93	10.20
3¼	.69	1.38	2.07	2.76	3.45	4.15	4.83	5.53	6.91	8.29	9.67	11.05
3½	.75	1.49	2.23	2.98	3.72	4.47	5.20	5.95	7.44	8.93	10.41	11.90
3¾	.80	1.60	2.39	3.19	3.99	4.78	5.58	6.38	7.97	9.57	11.16	12.75
4	.85	1.70	2.55	3.40	4.25	5.10	5.95	6.80	8.50	10.20	11.90	13.60
4½	.96	1.92	2.87	3.83	4.78	5.74	6.70	7.65	9.57	11.48	13.39	15.30
5	1.07	2.13	3.19	4.25	5.31	6.38	7.44	8.50	10.63	12.75	14.87	17.00
5½	1.17	2.34	3.51	4.67	5.84	7.02	8.18	9.35	11.69	14.03	16.36	18.70
6	1.28	2.55	3.83	5.10	6.38	7.65	8.93	10.20	12.75	15.30	17.85	20.40
7	1.49	2.98	4.46	5.95	7.44	8.93	10.41	11.90	14.87	17.85	20.83	23.80
8	1.70	3.40	5.10	6.80	8.50	10.20	11.90	13.60	17.00	20.40	23.80	27.20

RULES FOR OBTAINING APPROXIMATE WEIGHT OF WROUGHT IRON.

FOR ROUND BARS.

RULE: Multiply the square of the diameter in inches by the length in feet, and that product by 2.6. The product will be the weight in pounds, nearly.

FOR SQUARE AND FLAT WROUGHT BARS.

RULE: Multiply the area of the end of the bar in inches by the length in feet, and that 3.32. The product will be the weight in pounds, nearly.

WROUGHT IRON, ASSUMED WEIGHT.

A cubic foot .... = 480 lbs.
A square foot, 1 inch thick...................... = 40 lbs.
A bar 1 inch square, 1 foot long.................. = 3 1-3 lbs
A bar 1 inch square, 1 yard long.................. = 10 lbs.

RULE FOR FINDING THE SECTIONAL AREA OF A BAR OF WROUGHT IRON, WHEN WEIGHT PER FOOT IS GIVEN.

Multiply by 3 and divide by 10.

RULE FOR FINDING THE WEIGHT PER FOOT, WHEN AREA IS GIVEN.

Divide by 3 and multiply by 10.

NOTES ON CONSTRUCTION.

The necessity for vigilance and supervision of boiler designing and construction is made apparent in England by the stringent laws and by enforced rules and practices governing the same in way of additional factors for safety. They result in promoting good work and care in the operating and management of steam boilers.

Additional factors for safety are added to the established one of 5 due to deterioration by usage, age or fuel.

The English Board of Trade has established and tabulated a table of percentage of increase of factor of safety and cites reasons for such additional proportions.

All boilers must be designed and constructed according to their specifications, viz.: Holes to be drilled when shell plates have been rolled; straps or cover plates not less than ⅝ of plates they cover; in butt joints rivet sections must be 75 per cent over rivets in single shear and circumferential seams at least one-half the percentage of longitudnal seam.

The increased factor of safety is insisted on when conditions are as follows:

TABLE.

	Percentage of Increase	
A. =	.1	To be added when all holes are fair and good in the long seam, but drilled out of place after bending.
B. =	.2	When all holes are fair and good in longitudinal seams, but drilled before bending.

		Percentage of Increase	
C.	=	.2	When all holes are fair and good in longitudinal seams, but punched after bending.
D.	=	.3	When all holes are fair and good in longitudinal seam but punched before bending.
E.	=	.7	When all holes are not fair and good in longitudinalseam (and increased according to values).
F.	=	.8	When holes are all fair and good in the circumferential seams, but drilled out of place after bending.
G.	=	.1	When all holes are fair and good in the circumferential seams, but drilled before bending.
H.	=	.1	When holes are fair and good in the circumferential seams, but punched after bending.
I.	=	.15	If the holes are all fair and good in the circumferential seams, but punched before bending.
J.	=	.15	If the holes are not fair and good in the circumferential seams (and increased according to values).
K	=	.2	If the double butt straps are not fitted to the longitudinal seams and said seams are lap and double riveted.
L.	=	.07	If double butt straps are not fitted to the longitudinal seams and said seams are lap and triple riveted.
M.	=	.3	If only single butt straps are fitted to the longitudinal seams and said seams are double riveted.
N.	=	.15	If only single butt straps are fitted to the longitudinal seams and said seams are triple riveted.
O.	=	.1	When any description of joint in the longitudinal seam is single riveted.
P.	=	.2	If all holes are punched small and reamed afterwards or drilled out in place.
Q.	=	.4	If the longitudinal seams are fitted with single butt straps and are single riveted.
R.	=	.4	When material or workmanship is according to inspector doubtful or not the best (then the factor is increased accordingly).
S.	=	.1	If the circumferential seams are lap joints and double riveted.
T.	=	.2	If the circumferential seams are lap joints and single riveted.
U.	=	.25	When the circumferential seams are lap and the plates are not entirely under or over covers, and 1.65 to be added if the boiler is not open to inspection during the whole period of its construction.

The benefits derived from these additional factors of safety will be the means of bringing the science of boiler designing and work of construction up to a high standard.

In designing seams reason must govern when calculations are made, for if too great a pitch is used the plate cannot be drawn together without springing of plate or heads of rivets coming off, and so prevent making a tight caulking edge.

Each joint will be taken up separately as the strength of a joint is less than that of the solid plate due to cutting away for rivet holes and the single riveted lap joint is the weakest designed.

Tests have been made on various designed joints, and as it would be impossible to test all joints constructed, calculations from practice, factors and co-efficients must be relied on and followed up; these have proved satisfactory when construction has been carefully complied with according to designs.

The aim in boiler construction is to have the percentage of strength in rivet and plate as near equal as possible.

The maximum strength of a boiler is calculated from its weakest point, and the subject of seams in various forms and design will be taken up later; also boiler diameter, material thickness of same; rivets, their diameter; shearing strength, if single or double; pitch of rivets, number of rivets in joints; butt straps and factors, such as constants, taken into consideration when calculating the strength of a seam and varying according to conditions; methods of construction and design of joint or difference in material.

The necessity for care in designing and constructing to resist great forces is clearly shown by the following calculation: A common size boiler 60″ × 16′ has approximately 32,145 square inches of bursting area and at a pressure of 100 pounds it has a total of 1,607 tons of energy or bursting pressure; with the higher pressures now used, this hazard increases.

The English Board of Trade, a recognized authority on steam boilers, says that the rivet percentage of seam should be in excess of the plate and when computing the rivet section when steel plates and rivets are used the rivet section must be divided by 28/23. If iron rivets are used with steel plates then the rivet section must be ⅝ times greater than plate section and be divided by 13/8.

When describing strains, the action of shearing rivets means to shear across its diameter. The tearing strain refers to the action of tearing apart of plate. The crushing strain is the action to crush or rupture the plate between rivet holes and edge of plate.

In calculations for rivet strength the diameter of the rivet hole will be taken and not the diameter of the rivet, for the rivet must fill the rivet hole.

The reader will observe in following calculations that decimals will be omitted when of minor value.

LEGEND.

SYMBOLS USED IN FORMULAS

P = pressure
p = pitch of rivets
Pm = maximum pitch
N = number of rivets
Pd = diagonal pitch of rivets
D = diameter of boiler
d = diameter of rivet hole
T = Thickness of plate
% = percentage
V = distance between rows
E = distance center of rivets to edge of plate (lap)
TS = tensile strength of plate
AR = area of rivet hole
F = factor of safety

A coefficient is a prescribed amount to make up for any defects reducing strength of plate due to punching, riveting, caulking, &c.

A factor of safety is the difference between the safe working and bursting pressures.

It is well to explain here that calculations of joints are based on the principle that sections of the same do not vary, except according to the joints designed; the boiler, figuratively speaking, is composed of rings, each one having the same amount of plate width and pitch of rivets and the weakest part of this supposed ring is the base of the maximum strength. In the process of computing calculations this will appear clear to the student.

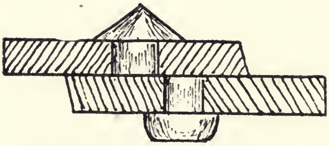
The rules for calculating strength of joints vary in formulas and results, but as stated in previous pages the rules the writer has used in connection with designing, testing and inspecting have been based on experiments and found in practice to have a factor of safety of reasonable margin.

While in computing joints the aim is to get the plate and rivet strength as near equal; favoring the rivet; it must be remembered that a variance in pitch will vary efficiencies as will also the diameter of a rivet, these being of standard sizes and varying in sixteenths; some of the rules will show an excess of rivet strength or even plate, and will appeal to the reader that a smaller diameter of rivet or greater pitch, or a lower or higher tensile strength, would affect the factors in securing the best possible efficiencies.

In the following rules in connection with boiler as outlined there are calculations to make from material and ratios for efficiencies. The strength of rivets has been computed from exhaustive tests and as the subject of rivet shearing will be a factor in calculating seams of efficiency it may be well to make some explanations. The necessary force to shear a rivet in single shear is 38,000 lbs. to square inch of cross section of rivet. The strain necessary to shear a rivet in double shear is 85 per cent more than in single shear.

EXAMPLE:

Rule to find strength of rivet in single shear: Multiply area of rivet hole by shearing resistance of rivet.



SINGLE SHEAR

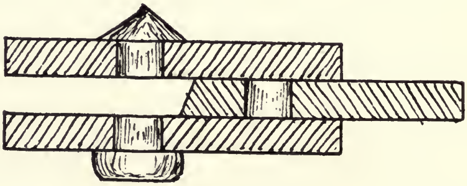
FORMULA:

A × S = strength of rivet in single shear

EXAMPLE:

.5185 = area of rivet hole
38000 = shearing resistance
41480000
15555
19703.0000
19,703 = strength of one rivet in single shear

38000 = lbs. single shear
.85 = % more for double shear
190000
304000
32300.00 lbs. = 85% of 38000 lbs.



DOUBLE SHEAR

adding the value to the above

38000 = single shearing strength
32300 = 85% added
70300 = shearing strength of a rivet in double shear

CHAPTER IV.

BRACES AND REINFORCING.

While there are boilers being made today that have strength in designed circular forms, the many in use and those being constructed have surfaces requiring reinforcements, some having an excess over other types and the high pressures now in demand require the best methods and improved design of brace.

This is a subject of as much importance as the designing of a joint and requires careful selection, proportioning and attaching braces to counteract strains that may be due to resisting bursting pressures, and those of contraction, expansion and collapsing.

Various designed braces and stays have been in use and are as varied in stability, some having minimum amount of strength, due to their structural weakness; again while some have the desired form and strength, location or principle of attaching same has depreciated their value as a reinforcement.

The subject of bracing is broad and could be treated inexhaustively, this owing to the many necessities and forms where each must necessarily be worked out separately. It is the intention to take up the most general methods, such as stay bolts, formed braces, stay tubes, crown bars, and angle irons.

Factors that are taken into consideration are

Structural,
Design,
Tensile strength,
Location, and
Principle of attaching.

In using rivets for braces it is customary to have the combined area equal to 1¼ times the brace area.

STAY BOLTS.

The use of stay bolts or stud stays for bracing is not at best a very satisfactory method of reinforcement, this owing to position and conditions, especially in fire box boilers where strains are caused by a bending force through the expansion of fire sheet, a pulling strain by the collapsing and bursting pressures and by that of vibration.

Care is necessary in selecting the best material; the U. S. Governmen requires the same tests to be made in accordance with those of plate used in connection with boilers coming under the supervision of the Federal Government. In physical and chemical tests results must show according to prescribed rules. Constant vibration is a menace to safety and braces are subject to and effected more by it than the strains from the pressures and more than the shell tubes or rivets are by it.

The best material for this strain is that made from piling material over that which is made from the bloom, this being due to its lamina structure.

Requirements to look for in brace materials are:

Tensile strength,
Elongation,
Reduction of area,
Elasticity.

Vigilance, careful and frequent tests and inspections of the stay bolts are necessary, for the force of expansion, contraction, tension, bending and vibration are severe. In the work of inserting and finishing this part of boiler construction defects often develop, this by stripping of threads when entering inner plate, again by hammering over ends; when this does occur the value of the brace is gone.

The design of the brace (stay bolt) is weak in the first place for the threads act in a measure as an initial fracture, especially so when one portion of thread is cut a little deeper than the balance. The hollow type of stay bolt has commendable features, viz.: The available admission of air to the (rich in heat units) volatile gases from fuel in furnace (these gases having a heat value of 62,000 heat units per pound, while the carbon or coke has only 14,500), the heating of the air before coming in contact and mixing with same, thus producing economical results, from minimum heat absorbed by air from water; another feature that commends itself is instant notice of any failure.

Rule to find safe working pressure on flat surfaces when thickness of plate and pitch of stay bolts are known:

Multiply the constant given for the specified thickness by the thickness of plate squared in sixteenths and divide by the greatest pitch squared.

FORMULA:

$$\frac{C \times T^2}{P^2} = \text{safe working pressure}$$

What is the safe working pressure on a curved surface less than a true circle? Plate 7/16 thick and stay bolts 5″ × 6″ centers.

EXAMPLE:

```
 7 = 7/16 = thickness
 7
 —
49 = thickness squared

                                  112 = constant as provided for
                                   49 = thickness squared
                                 ————
              pitch =     6″     1008
                          6       448
                          —      ————
    pitch squared = 36          )5488 (152 lbs. safe working pressure
                                  36
                                 ————
                                  188
                                  180
                                 ————
                                   88
                                   72
                                   ——
                                   16
```

Note constants for specific conditions as used in following examples:

For a plate three-fourths of an inch thick, stayed 9-inch by 10-inch centers:

$$\text{Working pressure} = \frac{120 \times 144}{100} = 172 \text{ pounds.}$$

For a plate nine-sixteenths of an inch thick, screw stays with nuts, stays pitched 9-inch by 10-inch centers:

$$\text{Working pressure} = \frac{135 \times 81}{100} = 109 \text{ pounds.}$$

For a plate three-fourths of an inch thick, supported by stays with double nuts, without washers or doubling plates, 10-inch by 12-inch centers:

$$\text{Working pressure} = \frac{170 \times 144}{144} = 170 \text{ pounds.}$$

For plate one-half inch thick, with washers three-eighths of an inch thick, stayed 10-inch by 12-inch centers:

$$\text{Working pressure} = \frac{160 \times 101.60}{144} = 112 \text{ pounds.}$$

For plate five-eighths of an inch thick, with doubling plate seven-sixteenths of an inch thick, stayed by 14-inch by 14-inch centers:

$$\text{Working pressure} = \frac{200 \times 149.81}{196} = 152 \text{ pounds.}$$

For plate five-eighths of an inch thick, with tees or angle bars one-half of an inch thick, stayed by 14-inch by 14-inch centers:

$$\text{Working pressure} = \frac{200 \times 167.96}{196} = 171 \text{ pounds.}$$

Plates heated for working must be annealed afterwards.

The diameter of a screw stay shall be taken at the bottom of the thread, provided it is the least diameter of the stay.

Flat heads not exceeding 20 inches in diameter may be used unsupported at pressure allowed by following rule:

Multiplying constant by thickness of head in sixteenths squared, and dividing by half of area to be supported, gives the pressure allowed.

FORMULA:

$$\frac{C \times T^2}{\frac{1}{2} \text{ of } A} = P$$

Where P = steam pressure allowable in pounds.
T = thickness of material = $\frac{3}{4} = \frac{12}{16}$.
A = area of head in inches = 314″.
C = 112 for plates $\frac{7}{16}$ of an inch and under.
C = 120 for plates over $\frac{7}{16}$ of an inch.

Provided, The flanges are made to an inside radius of at least 1½ inches.

EXAMPLE:

Required the working pressure of a flat head 20 inches in diameter and ¾ of an inch thick.

120 = constant as provided for
144 = head in sixteenths squared
480
480
120
one-half area of head = 157) 17280 (110 pounds safe working pressure
157
158
157
10

FLAT SURFACES.

The maximum stress allowable on flat plates supported by stays shall be determined by the following rule:

All stayed surfaces formed to a curve the radius of which is over 21 inches, excepting surfaces otherwise provided for, shall be deemed flat surfaces.

CONSTANTS.

C = 112 for screw stays with riveted heads, plates seven-sixteenths of an inch thick and under.

C = 120 for screw stays with riveted heads, plates above seven-sixteenths of an inch thick.

C = 120 for screw stays with nuts, plates seven-sixteenths of an inch thick and under.

C = 125 for screw stays with nuts, plates above seven-sixteenths of an inch thick and under nine-sixteenths of an inch.

C = 135 for screw stays with nuts, plates nine-sixteenths of an inch thick and above.

C = 170 for stays with double nuts having one nut on the inside and one nut on the outside of plate, without washers or doubling plates.

C = 160 for stays fitted with washers or doubling strips which have a thickness of at least .5 of the thickness of the plate and a diameter of at least .5 of the greatest pitch of the stay, riveted to the outside of the plates, and stays having one nut inside of the plate, and one nut outside of the washer or doubling strip. For T take 72 per cent of the combined thickness of the plate and washer or plate or doubling strip.

C = 200 for stays fitted with doubling strips which have a thickness equal to at least .5 of the thickness of the plate reinforced, and covering the full area braced (up to the curvature of the flange, if any), riveted to either the inside or outside of the plate, and stays having one nut outside and one inside of the plates. Washers or doubling plates to be substantially riveted. For T take 72 per cent of the combined thickness of the two plates.

C = 200 for stays with plates stiffened with tees or angle-bars having a thickness of at least two-thirds the thickness of plate and depth of webs at least one-fourth of the greatest pitch of the stays, and substantially riveted on the inside of the plates, and stays having one nut inside bearing on washers fitted to the edges of the webs, that are at right angles to the plate. For T take 72 per cent of the combined thickness of web and plate.

No flat plates or surfaces shall be unsupported at a greater distance than 18 inches.

Multiply the constant 120 by the thickness squared in sixteenths and divide product by the pitch of stay squared:

FORMULA:

$$\frac{C \times T^2}{P^2} = \text{working pressure}$$

LEGEND:

T = thickness of plate = $\frac{7}{16}$ = 7
P = pitch = 10″
C = constant = 120

EXAMPLE:

120 = constant
49 = plate squared in 16ths
1080
480
pitch squared = 100) 5880 (58.8 lbs. pressure allowed or 59 lbs. nearly
500
880
800
80

Rules adopted by authorities that have proven satisfactory from tests and usage and adopted by the U. S. Government and reputable boiler manufacturers are given in this chapter, and in connection material and workmanship is considered to be the best, fitted accurately and properly secured.

Exhaustive tests have been made by the highest authorities, governments, scientific and mechanical and results have shown that there are some differences; sufficient reasons in the fact show that the majority are near enough to establish formulas that have liberal margins of safety.

Judgment must be governed by conditions and construction when out of the ordinary and special consideration given, always allowing a reasonable factor of safety for an unusual form or position.

For all stays the least sectional area shall be taken in calculating the stress allowable.

All screw stay bolts shall be drilled at the ends with a one-eighth inch hole to at least a depth of one-half inch beyond the inside surface of the sheet. Stays through laps or butt straps may be drilled with larger hole to a depth so that the inner end of said larger hole shall not be nearer than the thickness of the boiler plates from the inner surface of the boiler.

Such screw stay bolts, with or without sockets, may be used in the construction of marine boilers where fresh water is used for generating steam: *Provided, however,* that screw stay bolts of a greater length than 24 inches will not be allowed in any instance, unless the ends of said bolts are fitted with nuts. Water used from a surface condenser shall be deemed fresh water.

Holes for screwed stays must be tapped fair and true and full thread.

The ends of stays which are upset to include the depth of thread shall be thoroughly annealed after being upset.

The sectional area of pins to resist double shear and bending, accurately fitted and secured in crow feet, sling, and similar stays, shall be at least equal to required sectional area of the brace. Breadth across each side and depth to crown of eye shall be not less than .35 to .55 of diameter of pin. In order to compensate for inaccurate distribution the forks should be proportioned to support two-thirds of the load, thickness of forks to be not less than .66 to .75 of the diameter of pins.

The combined sectional area of rivets used in securing tee irons and crow feet to shell, said rivets being in tension, shall be not less than the required sectional area of brace. To insure a well-proportioned rivet point, the total length of shank shall closely approximate the grip plus 1.5 times the diameter of the shank. All rivet holes shall be drilled. Distance from center of rivet hole to edge of tee irons, crow feet, and similar fastenings shall be so proportioned that the net sectional areas through sides at rivet holes shall equal the required rivet section. Rivet holes shall be slightly countersunk in order to form a fillet at point and head.

CONSTANTS PROVIDED FOR THE VARYING REQUIREMENTS.

C = 9,000 for tested steel stays exceeding 2½ inches in diameter.

C = 8,000 for tested steel stays 1¼ inches and not exceeding 2½ inches in diameter, when such stays are not forged or welded. The ends, however, may be upset to a sufficient diameter to allow for the depth of the thread. The diameter shall be taken at the bottom of the thread, provided it is the least diameter of the stay. All such stays after being upset shall be thoroughly annealed.

C = 8,000 for a tested Huston or similar type of brace, the cross-sectional area of which exceeds 5 square inches.

C = 7,000 for such tested braces when the cross-sectional area is not less than 1.227 and not more than 5 square inches, provided such braces are prepared at one heat from a solid piece of plate without welds.

C = 6,000 for all stays not otherwise provided for.

Rule to find sectional area of a brace to support a given area when pressure is known: Multiply area to be supported by pressure per square inch and divide by constant as provided for size and material of brace.

Formula:

$$\frac{A \times P}{C} = \text{sectional area of brace}$$

Legend:

A = area to be supported = 36 square inches
P = pressure = 150 lbs.
C = constant = brace steel having 1¼ diameter = 8000

Example:

36″ = sectional area to be supported
150 = lbs. pressure
1800
36
constant for 1¼ steel brace = 8000)54000000(.6750 = 43/64 or $\frac{15}{16}$ cross-sectional area nearly
48000
60000
56000
40000
40000
0

Rule to find strain on a stay bolt: Multiply the area supported by the stay, by the pressure.

FORMULA:

$A \times P$ =strain on stay

LEGEND:

A =area =6″ × 6″ =36 square inches
P =pressure =150 lbs.

EXAMPLE:

36 square inches =area
150 =lbs. pressure
1800
36
5400 =lbs. strain on bolt

Rule to find greatest area one stay bolt may support: Multiply area of stay bolt by constant and divide by working pressure.

FORMULA:

$$\frac{A \times C}{P}$$ =limit of area to be supported by one bolt

LEGEND:

C =constant =6000 lbs. allowed per cross-sectional area
A =area of stay bolt = $\frac{15}{16}$ = .69029
P =pressure =150 lbs.

EXAMPLE:

.69029 =area of $\frac{15}{16}$ bolt
6000 =constant

pressure =150)41417.~~4000~~(27.6″ =limit of area to be supported by one bolt
300
1141
1050
917
900
17

Rule to find number of stay bolts to support a given area when pressure is given:

Multiply area to be supported by pressure and divide sum by constants as provided for. Constants for the different size bolts to be used are as follows:

for ⅞″ diameter use constant 4000,
" 1⅛″ " " " 6000,
if for over that diameter and up to 2½″ " " " 8000,
being pounds pressure per square inch of cross-sectional area.

FORMULA:

$$\frac{A \times P}{C} = \text{number of stay bolts}$$

The following example is where bolts are ⅞″ in diameter:

LEGEND:

A = area to be supported = 800 square inches
P = pressure = 100 lbs.
C = constant = 4000

EXAMPLE:

800 = area to be supported
100 = lbs. pressure

constant = 4000) 80000 (20 stay bolts required
8000

0

The following example is where bolts are 1⅛″ diameter:

LEGEND:

A = area to be supported = 500
P = pressure = 120 lbs.
C = constant = 6000

EXAMPLE:

500 = area to be supported
120 = lbs. pressure

10000
500

constant = 6000) 60000 (10 stay bolts required
6000

0

Rule to find centers for stay bolts when pressure, area to be supported and constant provided for stay bolt are known: Multiply area of stay bolt by constant and divide by pressure.

FORMULA:

$$\frac{A \times C}{P} = \text{centers of stay bolts}$$

LEGEND:

A = area to be supported = .3750
C = constant = 4000
P = pressure = 150 lbs.

Example:

.3750 = area of stay bolt
4000 = constant

pressure = 150)1500.0000(10″ = centers of stay bolts
150

0

Rule to find area of stay bolt. Multiply centers of stay bolt by pressure and divide by constant 4,000; the quotient is area of stay bolt required.

Formula:

$$\frac{CB \times P}{C} = \text{area of stay bolt}$$

Legend:

P = pressure = 150 lbs.
C = constant = 4000.
CB = center of stay bolt = 10″

Example:

10″ = center of stay bolt
150 = pressure

500
10

constant = 4000)1500.0000(.3750 = area of stay bolt
1200 0

300 00
280 00

20 000
20 000

0

English Board of Trade rule to find safe working pressure when steel stay bolts are used and are screwed into plates and fitted with nuts:

Multiply constant 80 (plus 25% for steel) by thickness of plate in sixteenths plus one sixteenth squared; divide by pitch of rivet squared minus 6; product is safe working pressure.

Formula:

$$\frac{C + \% \times (T + \frac{1}{16})^2}{P^2 - 6} = \text{safe working pressure}$$

LEGEND:

T = thickness of plate = $\frac{7}{16}$
P = pitch = 7
C = constant = 80
% = 25% added for steel

EXAMPLE:

80 = constant
20 = 25% added for steel
———
pitch = 7 100
7 64 = $\frac{7}{16} + \frac{1}{16}$ or $\frac{8}{16}$, squared
— ———
pitch squared = 49 400
minus 6 600
— ———
43)6400(148 = lbs. pressure for steel bolts
43
———
210 7 = $\frac{7}{16}$ = thickness of plate
172 1 = $\frac{1}{16}$ added
——— —
380 8 = $\frac{8}{16}$
344 8
——— —
36 64 = $\frac{8}{16}$ squared

Rule to find pitch of stay bolts:

Multiply constant 112 by the square of the thickness of plate in sixteenths of an inch; divide this product by steam pressure and extract the square root of quotient.

FORMULA:

$$\sqrt{\frac{C \times T^2}{P}} = \text{pitch of stay}$$

LEGEND:

C = constant = 112
T = thickness of plate = $\frac{7}{16}$
P = pressure = 150

EXAMPLE:

112 = constant
49 = the square of $\frac{7}{16}$
———
1008
448
———
150)5488(36
450 square root of 36 is 6″ pitch
———
988
900 6)36(6″ = square root = pitch of bolts
——— 36
88 —

TABLE OF STAY BOLTS, PLATE, PITCH AND PRESSURE.

Pressure in pounds.	Centers of Stay Bolts.		
	⅜″ Plate	7/16″ Plate.	½″ Plate.
20	11¼″ pitch	13″ pitch	15″ pitch
40	8 "	9¼ "	10⅝ "
60	6½ "	7⅝ "	8¾ "
80	5⅝ "	6½ "	7½ "
100	5 "	5¾ "	6¾ "
120	4½ "	5¼ "	6⅛ "
140	4¼ "	4⅞ "	5⅝ "
150	4⅛ "	4¾ "	5½ "
160	4 "	4⅝ "	5¼ "
Diam. of stay bolt	⅞″	1″	1¼″

CROW FOOT OR FORMED BRACES.

As stated in preceding pages the many and varied surfaces to be braced requiring specific methods and application of bracing, the H. T. boiler, having the minimum amount of flat surface and conditions favorable to apply the selection for suitable type of brace, is confined to the one with minimum structural weakness, taking the Huston, McGregor, or of equal stability.

In calculating the necessary reinforcement by bracing—the area of surface to be stayed, and working pressure is considered; while the thickness of head is a factor in its strength, the necessity for braces in lieu of increasing the thickness of head to self supporting, is without comment.

In all types of stays the least sectional area must be taken in calculating the stress allowable and the combined sectional area of rivets used in securing crow feet, angle irons and such form of braces, necessitating rivets, must not be less than the required sectional area of brace; all rivet holes to be drilled, and the distance from center of hole to edge of palm or brace surface shall be so proportionate that the net sectional areas through sides at rivet holes shall equal the rivet section; rivet holes in plate to be slightly countersunk.

Taking a flat surface in head above water line, say 800 square inches, to proportionate a proper thickness of head for that *unstayed* portion it would be necessary to have the thickness of head by rule as follows:

Multiply area by pressure and again by constant; divide product by tensile strength multiplied by 10; the quotient will be the thickness for unstayed portion.

LEGEND:

A = area = 800 square inches
P = pressure = 100
C = constant = 7000 lbs. per square inch
TS = tensile strength = 60000

FORMULA:

$$\frac{A \times P \times C}{TS \times 10} = \text{thickness for unstayed portion}$$

EXAMPLE:

800 = area
100 = pressure
tensile strength = 60000 80000
multiplied by 10 7000 = constant
600000) 560000000 (933 = $\frac{15}{16}$ inch nearly in thickness
5400000
2000000
1800000
2000000
1800000
200000

This would not be desirable for reasons of cost, labor attached to working it and conductivity of heat, therefore heads must be of less thickness and bracing resorted to.

To find the area of an unstayed segment is the first thing necessary and that is a simple rule as used in boiler construction, as calculations for such measures are always favored.

Rule to find minimum area of stay or brace to support a given area: Divide load on stay by allowable strain per square inch of sectional area as provided; the quotient is minimum area of stay.

FORMULA:

$$\frac{L}{S} = \text{area of brace}$$

LEGEND:

L = load on stay = 6750 lbs.
S = strain per square inch of sectional area = 6000 lbs.

Example:

strain allowed per sq. in. = 6000)6750.000(1.125 or $1\frac{1}{8}''$ diameter
6000
750 0
600 0
150 00
120 00
300 00
300 00

Rule to find area of stay beyond maximum of curved surface unsupported when thickness of plate and pressure are known: Multiply constant 112 by thickness of plate in sixteenths of an inch and divide product by the pressure in pounds per square inch; the quotient is area of stay required.

Legend:

C = constant = 112
T = thickness of plate = $\frac{7}{16}$
P = pressure = 150 lbs.

Formula:

$$\frac{C \times T}{P} = \text{area of stay}$$

Example:

112 = constant
.7 = thickness in 16ths
pressure = 150)78.4000(.5226 = area or $\frac{13}{16}$ approximately
75 0
3 40
3 00
400
300
1000
900
100

To determine the areas of diagonal stays: Multiply the area of a direct stay required to support the surface by the slant or diagonal length of the stay; divide this product by the length of a line drawn at right angles to surface supported to center of palm of diagonal stay. The quotient will be the required area of the diagonal stay.

Formula:

$$\frac{A \times L}{1} = \text{sectional area of diagonal stay}$$

LEGEND:

A = sectional area of direct stay = .7854
L = length of diagonal stay = 60″
l = length of line drawn at right angles to boiler head or surface supported to center of palm of diagonal stay = 48″

EXAMPLE:

```
                                 .7854  = area of 1″ direct stay
                                    60 = length of stay
length of line drawn at right   ———————
     angles to boiler = 48″)47.1240(.9817 = sectional area of a diag-
                            43 2                onal brace = 1⅛″ nearly
                            ————
                             3 92
                             3 84
                             ————
                               84
                               48
                              ———
                              360
                              336
                              ———
                               24
```

When diagonal braces are applied the angle should not exceed over 30 degrees.

Rule to find the load on a stay: Multiply area to be supported by pressure and divide by sectional area of stay bolt.

LEGEND:

A = area to be supported = 50″
P = pressure = 160 lbs.
SB = area of stay bolt = .69029

FORMULA:

$$\frac{A \times P}{SB} = \text{strain on sectional area of stay}$$

EXAMPLE:

```
                                50″ = area to be supported
                               160 = pressure
                              ————
                              3000
                              50
                              ————
area of stay bolt = .69029)8000.00000(11589 lbs. = strain on sec-
                           6902 9                  tional area of stay
                           ———————
                           1097 10
                            690 29
                           ————————
                            406 810
                            345 145
                            ————————
                             61 6650
                             55 2232
                             ————————
                              6 44180
                              6 21261
                              ————————
                               22919
```

HEADS.

All heads employed in the construction of cylindrical externally fired boilers, for steamers navigating the Red River of the North and rivers that flow into the Gulf of Mexico, shall have a thickness of material as follows:

For boilers having a diameter—
Over 32 inches and not over 36 inches, not less than ½ inch.
Over 36 inches and not over 40 inches, not less than $\frac{9}{16}$ inch.
Over 40 inches and not over 48 inches, not less than ⅝ inch.
Over 48 inches, not less than ¾ inch.

Where flat heads do not exceed 20 inches in diameter they may be used without being stayed, and the steam pressure allowable shall be determined by the following formula:

$$P = \frac{C \times T^2}{A}$$

Where P = steam pressure allowable in pounds.
T = thickness of material in sixteenths of an inch.
A = one-half the area of head in inches.
C = 112 for plates $\frac{7}{16}$ of an inch and under.
C = 120 for plates over $\frac{7}{16}$ of an inch.

Provided, The flanges are made to an inside radius of at least 1½ inches.

EXAMPLE.

Required the working pressure of a flat head 20 inches in diameter and ¾ of an inch thick. Substituting values, we have

$$P = \frac{120 \times 144}{157} = 110 \text{ pounds}$$

The heads of steam and mud drums of such boilers shall have a thickness of material of not less than half an inch; pressure to be determined by formula for flatheads.

CONVEXED HEAD.

Rule to find pressure allowed on a convexed head: Multiply the thickness of the plate by one-sixth of the tensile strength and divide by one-half of radius to which head is bumped; result gives pressure allowed per square inch.

Add 20 per cent to pressure when the head is double riveted to the shell and the holes are fairly drilled.

LEGEND:

TS = tensile strength = 60000
T = thickness of plate = ⅝ = .625
R = radius of bump = 60″

FORMULA:

$$\frac{T\times(1/6 \text{ of } TS)}{\tfrac{1}{2} \text{ of } R} = \text{lbs. pressure allowed}$$

EXAMPLE:

.625 = thickness of plate
10000 = 1/6 of TS

half of radius = 30)6250.000(208 lbs. = pressure allowed on single riveted circumferential seam
60
250
240
10

208 lbs. = pressure allowed on single riveted
41.6 = 20% added for double riveted

249.6 lbs. pressure allowed double riveted

Rule to find bursting pressure on flat head: Multiply thickness of plate by ten times the tensile strength and divide by area of head in inches; the sum is bursting pressure.

LEGEND:

T = thickness of plate = $\frac{9}{16}$ = .5625
TS = tensile strength = 60000
A = area of head = 934.822 inches
D = diameter of head = 34½″

FORMULA:

$$\frac{T\times(10\times TS)}{A\times C} = \text{bursting pressure}$$

EXAMPLE:

.5625 = thickness of plate
600000 = ten times tensile strength

area of head = 934822)337500.0000(361 lbs. bursting pressure
280446 6
57053 40
56089 32
964 080
934 822
29 258

Divide bursting pressure by 5 and this will give working pressure

CONCAVED HEAD.

Rule to find pressure allowed on a concave head: Multiply the pressure per square inch allowed on a bumped head attached convexly by the constant 6, and the product will give the pressure per square inch allowed on concaved head.

FORMULA:

P × C = pressure on concaved head

LEGEND:

P = pressure allowed on a bumped head = 208 lbs.
C = constant = .6

EXAMPLE:

208 = pressure allowed on a bumped head
.6 = constant

124.8 = lbs. pressure on a concaved head

NOTE ON DISHED HEADS.

Dished or bumped heads have strength due to form and thickness depending on diameter.

Bumped heads may contain a manhole opening flanged inwardly, when such flange is turned to a depth of three times the thickness of the material in the head.

DEPTHS OF DISH AND FLANGE HEADS.

Diam. Heads.	Diam. after Dishing and Flanging.	Depth of Dish.	Depth of Flange.
34	30	3	2
40	36	3	2
46	42	4	2
52½	48	5	2
58½	54	6	2
65	60	6	2
71	66	7	2
77	71½	7	2
78	72	8	2
87	80	8	2½
91	84	9	2½
97	90	10	2½
102	96	12	2½

CAST IRON HEADS.

Rule to find thickness of an unstayed boiler head so it will equal in strength the shell: Multiply square root of radius by the thickness of the shell plate in inches; the product is the required thickness of head.

LEGEND:

T = thickness of plate $\frac{3}{8}''$ = .375
IR = inside radius = 19.9809

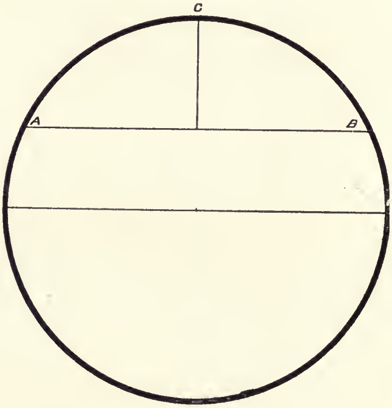
FORMULA:

$(\sqrt{IR}) \times T$ = thickness of head

EXAMPLE:

```
  4.47 =square root of radius
  .375 =thickness of shell
  ----
  2235
 3129
1341
-------
1.67625 = thickness of head required = 1 11/16" approx.
```

A rule to find area of a segment of a circle as outlined by A, B and C.



Divide the diameter of circle by height of the segment, subtract 608 from quotient and extract the square root of the remainder; this result multiplied by four times the square of the height of the segment and divided by three, will give the area.

FORMULA:

$$\left\{\sqrt{\frac{D}{H} - .608}\right\} \times \left\{\frac{4 \times H^2}{3}\right\} = \text{area of segment}$$

Legend:

H = height of segment 22″
D = diam. of boiler 72″
C = constant = .608

Example:

```
                    (diameter)
  height 22″)72.0000(3.2727
              66
              ---
               60
               44
              ----            3.2727
               160               .608 constant
               154            ------
              ----          1)2.66470000(1.6323 sq. root
               154            1
              ----            ----
                 60     26    1 66              1.6323
                 44           1 56                 645
                ---    ---    ------           -------
                160    323      1047            8 1615
                154              969           65 292
                ---    ----    -----          979 38
                  6    3262      7800         ---------
 22″ height of segment           6524       1052.8335
 22                   -----     ------         or 1053″ =area of
 --                   32643     127600                   segment.
 44                              97929
44                              ------
---                              29671
484 height squared
  4 four times
----
3)1936 =4 times square of height
  -----
  645.33
```

Rule to find number of braces to support a segment as just described: Multiply area of segment by pressure in pounds per square inch and divide by number of pounds pressure form or type of brace sectional area is allowed. To illustrate: A modern formed brace by 8,000 when sectional area exceeds 5 square inches; 7,000 when sectional area is less than 5 square inches, and 6,000 for all stays not otherwise provided for.

Formula:

$$\frac{A \times \text{Pressure}}{\text{Brace supporting value}} = \text{number of braces required}$$

Example:

1053 = area of segment required
160 = lbs. pressure

63180
1053

modern brace = 8000)168480(21 + or 22 braces
16000

8480
8000

480

The table given below is an extract from Trautwine's Engineers' Pocket Book, and will be found of great value in arriving at an accurate solution.

The first column marked height, is the height of the segment in parts of the diameter of the boiler. The first number .001 refers to a segment whose height is 1/1000 of the diameter of the boiler, the second number refers to 2/1000 of the diameter of the boiler, and the third 3/1000 of the diameter of the boiler and so on until it reaches a complete semi-circle or half-diameter of the boiler.

CUBICAL CONTENTS.

Suppose now we desire to find the cubical contents by the table of the steam space in a boiler 48 inches in diameter by 14 feet long. The water line say is 4″ above the top row of tubes and the height of the segment is 12 inches.

The area of circles or similar parts of circles of different sizes are directly proportional to the square of their diameter. Hence, it will only be necessary to find what part of the diameter, 12 inches (the height of the steam space), is. This is done by dividing 12 by 48 = .250. Find this quotient in the column of heights in the table, take the corresponding area and multiply it by the square of the diameter. Then 4 × 4 equals 16 and 12 ÷ 48 equals .250. By the table we find that the area of a segment whose height is .250 is seen to be .153546. This multiplied by 16 gives 2.4567 square feet of the cross sectional area of the steam space. This area multiplied by 14, which is the length of the boiler in feet, or 2.4567 × 14 equals 34.39, which is the volume of steam space in cubic feet.

The same result in cubic feet can be obtained by the first method, which I do not think can be simplified any further.

AREAS OF CIRCULAR ARCS.

By This Table May be Obtained the Area of Segments of Circles.

Height	Area	Height	Area	Height	Area	Height	Area
.001	.000 042	.040	.010 538	079	.028 894	.118	.052 090
.002	.000 119	.041	.010 932	.080	.029 435	.119	.052 737
.003	.000 219	.042	.011 331	.081	.029 979	.120	.053 385
.004	.000 337	.043	.011 734	.082	.030 526	.121	.054 037
.005	.000 471	.044	.012 142	.083	.031 077	.122	.054 690
.006	.000 619	.045	.012 555	.084	.031 630	.123	.055 346
.007	.000 779	.046	.012 971	.085	.032 186	.124	.056 004
.008	.000 952	.047	.013 393	.086	.032 746	.125	.056 664
.009	.001 135	.048	.013 818	.087	.033 308	.126	.057 327
.010	.001 329	.049	.014 248	.088	.033 873	.127	.057 991
.011	.001 533	.050	.014 681	.089	.034 441	.128	.058 658
.012	.001 746	.051	.015 119	.090	.035 012	.129	.059 328
.013	.001 969	.052	.015 561	.091	.035 586	.130	.059 999
.014	.002 199	.053	.016 008	.092	.036 162	.131	.060 673
.015	.002 438	.054	.016 458	.093	.036 742	.132	.061 349
.016	.002 685	.055	.016 912	.094	.037 324	.133	.062 027
.017	.002 940	.056	.017 369	.095	.037 909	.134	.062 707
.018	.003 202	.057	.017 831	.096	.038 497	.135	.063 389
.019	.003 472	.058	.018 297	.097	.039 087	.136	.064 074
.020	.003 749	.059	.018 766	.098	.039 681	.137	.064 761
021	.004 032	.060	.019 239	.099	.040 277	.138	.065 449
.022	.004 322	.061	.019 716	.100	.040 875	.139	.066 140
.023	.004 619	.062	.020 197	.101	.041 477	.140	.066 833
.024	.004 922	.063	.020 681	.102	.042 081	.141	.067 528
.025	.005 231	.064	.021 168	.103	.042 687	.142	.068 225
.026	.005 546	.065	.021 660	.104	.043 296	.143	.068 924
.027	.005 867	.066	.022 155	.105	.043 908	.144	.069 626
.028	.006 194	.067	.022 653	.106	.044 523	.145	.070 329
.029	.006 527	.068	.023 155	.107	.045 140	.146	.071 034
.030	.006 866	.069	.023 660	.108	.045 759	.147	.071 741
.031	.007 209	.070	.024 168	.109	.046 381	.148	.072 450
.032	.007 559	.071	.024 680	.110	.047 006	.149	.073 162
.033	.007 913	.072	.025 196	.111	.047 633	.150	.073 875
.034	.008 273	.073	.025 714	.112	.048 262	.151	.074 590
.035	.008 638	.074	.026 236	.113	.048 894	.152	.075 307
.036	.009 008	.075	.026 761	.114	.049 529	.153	.076 026
.037	.009 383	.076	.027 290	.115	.050 165	.154	.076 747
.038	.009 764	.077	.027 821	.116	.050 805	.155	.077 470
.039	.010 148	.078	.028 356	.117	.051 446	.156	.078 194

Height	Area	Height	Area	Height	Area	Height	Area
.157	.078 921	.199	.111 025	.241	.145 800	.281	.180 918
.158	.079 650	.200	.111 824	.242	.146 656	.282	.181 818
.159	.080 380	.201	.112 625	.243	.147 513	.283	.182 718
.160	.081 112	.202	.113 427	.244	.148 371	.284	.183 619
.161	.081 847	.203	.114 231	.245	.149 231	.285	.184 522
.162	.082 582	.204	.115 036	.246	.150 091	.286	.185 425
.163	.083 320	.205	.115 842	.247	.150 953	.287	.186 329
.164	.084 090	.206	.116 651	.248	.151 816	.288	.187 235
.165	.084 801	.207	.117 460	.249	.152 681	.289	.188 141
.166	.085 545	.208	.118 271	.250	.153 546	.290	.189 048
.167	.086 200	.209	.119 084	. . . .	. . . .	.291	.189 956
.168	.087 037	.210	.119 898	. . . .	. . . .	.292	.190 865
.169	.087 785	.211	.120 713	.251	.154 413	.293	.191 774
.170	.088 536	.212	.121 530	.252	.155 281	.294	.192 685
.171	.089 288	.213	.122 348	.253	.156 149	.295	.193 597
.172	.090 042	.214	.123 167	.254	.157 019	.296	.194 509
.173	.090 797	.215	.123 988	.255	.157 891	.297	.195 423
.174	.091 555	.216	.124 811	.256	.158 763	.298	.196 337
.175	.092 314	.217	.125 634	.257	.159 636	.299	.197 252
.176	.093 074	.218	.126 459	.258	.160 511	.300	.198 168
.177	.093 837	.219	.127 286	.259	.161 386	.301	.199 085
.178	.094 601	.220	.128 114	.260	.162 263	.302	.200 003
.179	.095 367	.221	.128 943	.261	.163 141	.303	.200 922
.180	.096 135	.222	.129 773	.262	.164 020	.304	.201 841
.181	.096 904	.223	.130 605	.263	.164 900	.305	.202 762
.182	.097 675	.224	.131 438	.264	.165 781	.306	.203 683
.183	.098 447	.225	.132 273	.265	.166 663	.307	.204 605
.184	.099 221	.226	.133 109	.266	.167 546	.308	.205 528
.185	.099 997	.227	.133 946	.267	.168 431	.309	.206 452
.186	.100 774	.228	.134 784	.268	.169 316	.310	.207 376
.187	.101 553	.229	.135 624	.269	.170 202	.311	.208 302
.188	.102 334	.230	.136 465	.270	.171 090	.312	.209 228
.189	.103 116	.231	.137 307	.271	.171 978	.313	.210 155
.190	.103 900	.232	.138 151	.272	.172 868	.314	.211 083
.191	.104 686	.233	.138 996	.273	.173 758	.315	.212 011
.192	.105 472	.234	.139 842	.274	.174 650	.316	.212 941
.193	.106 261	.235	.140 689	.275	.175 542	.317	.213 871
.194	.107 051	.236	.141 538	.276	.176 436	.318	.214 802
.195	.107 843	.237	.142 388	.277	.177 330	.319	.215 734
.196	.108 636	.238	.143 239	.278	.178 226	.320	.216 666
.197	.109 431	.239	.144 091	.279	.179 122	.321	.217 600
.198	.110 227	.240	.144 945	.280	.180 020	.322	.218 534

Height	Area	Height	Area	Height	Area	Height	Area
.323	.219 469	.368	.262 249	.413	.306 140	.458	.350 749
.324	.220 404	.369	.263 214	.414	.307 125	.459	.351 745
.325	.221 341	.370	.264 179	.415	.308 110	.460	.352 742
.326	.222 278	.371	.265 145	.416	.309 096	.461	.353 739
.327	.223 216	.372	266 111	.417	.310 082	.462	.354 736
.328	.224 154	373	.267 078	.418	.311 068	.463	.355 733
.329	.225 094	.374	.268 046	.419	.312 055	.464	.356 730
.330	.226 034	.375	.269 014	.420	.313 042	.465	.357 728
.331	.226 974	.376	.269 982	.421	.314 029	.466	.358 725
.332	.227 916	.377	.270 951	.422	.315 017	.467	.359 723
.333	.228 858	.378	.271 921	.423	.316 005	.468	.360 721
.334	.229 801	.379	.272 891	.424	.316 993	.469	.361 719
.335	.230 745	.380	.273 861	.425	.317 981	.470	.362 717
.336	.231 689	.381	.274 832	.426	.318 970	.471	.363 715
.337	.232 634	.382	.275 804	.427	.319 959	.472	.364 714
.338	.233 580	.383	.276 776	.428	.320 949	.473	.365 712
.339	.234 526	.384	.277 748	.429	.321 938	.474	.366 711
.340	.235 473	.385	.278 721	.430	.322 928	.475	.367 710
.341	.236 421	.386	.279 695	.431	.323 919	.476	.368 708
.342	.237 369	.387	.280 669	.432	.324 909	.477	.369 707
.343	.238 319	.388	.281 643	.433	.325 900	.478	.370 706
.344	.239 268	.389	.282 618	.434	.326 891	.479	.371 705
.345	.240 219	.390	.283 593	.435	.327 883	.480	.372 704
.346	.241 170	.391	.284 569	.436	.328 874	.481	.373 704
.347	.242 122	.392	.285 545	.437	.329 866	.482	.374 703
.348	.243 074	.393	.286 521	.438	.330 858	.483	.375 702
.349	.244 027	.394	.287 499	.439	.331 851	.484	.376 702
.350	.244 980	.395	.288 476	.440	.332 843	.485	.377 701
.351	.245 935	.396	.289 454	.441	.333 836	.486	.378 701
.352	.246 890	.397	.290 432	.442	.334 829	.487	.379 701
.353	.247 845	.398	.291 411	.443	.335 823	.488	.380 700
.354	.248 801	.399	.292 390	.444	.336 816	.489	.381 700
.355	.249 758	.400	.293 370	.445	.337 810	.490	.382 700
.356	.250 715	.401	.294 350	.446	.338 804	.491	.383 700
.357	.251 673	.402	.295 330	.447	.339 799	.492	.384 699
.358	.252 632	.403	.296 311	.448	.340 793	.493	.385 699
.359	.253 591	.404	.297 292	.449	.341 788	.494	.386 699
.360	.254 551	.405	.298 274	.450	.342 783	.495	.387 699
.361	.255 511	.406	.299 256	.451	.343 778	.496	.388 699
.362	.256 472	.407	.300 238	.452	.344 773	.497	.389 699
.363	.257 433	.408	.301 221	.453	.345 768	.498	.390 699
.364	.258 395	.409	.302 204	.454	.346 764	.499	.391 699
.365	.259 358	.410	.303 187	.455	.347 760	.500	.392 699
.366	.260 321	.411	.304 171	.456	.348 756	....	....
.367	.261 285	.412	.305 156	.457	.349 752	....	....

Rule to find pressure allowed on a brace for given size: Multiply area of brace by pressure allowed per square inch cross sectional area.

LEGEND:

A = area of brace 3″x½″ = 1.5″ area
S = strain allowed = 6000 lbs.
that size brace

FORMULA:

A × S = pressure allowed

EXAMPLE:

3″
.5
———
1.5 = area
6000 lbs. allowed per square inch
———
$9000\cancel{0}$ lbs. allowed on brace of that size

THROUGH BRACE RODS.

Through brace rods are often used when conditions are favorable, space ample for cleaning and inspection.

These rods are usually 1¼ to 2½ inches diameter and washer or plates riveted to heads to increase holding or breaking surface; thickness of heads are governed by pressure, also by the size and number of rods. Same rule is used that governs the palm or formed brace.

Rule to find working pressure allowed on a through brace rod. Multiply area of rod by strain allowed according to corresponding diameter and divide by area supported by rod.

LEGEND:

AR = 2″ rod = 3.1416 = area of rod
A = 16x14 surface = 224″ area
S = strain allowed on that size brace = 8000

FORMULA:

$$\frac{AR \times S}{A} = \text{working pressure}$$

EXAMPLE:

3.1416 = area of 2″ rod
8000 lbs. allowed on sectional area
———
surface area = 224)$25132.\cancel{8000}$(112 lbs. working pressure
224
———
273
224
———
492
448
———
44

CURVED SURFACES.

To find safe working pressure on curved sufrace when stiffened by angle, single or double, or tee bars; for single, the angle iron should have a thickness of at least eight-tenths that of plate and a depth of at least one-half pitch;—where stiffened with double angle or tee irons, to have at least two-thirds that of thickness of plate and a depth of at least one-fourth of pitch; angles or tee bars being substantially riveted to the plate supported.

Where rounded tops of combustion chambers are stiffened with single or double angle-iron stiffeners, or tee bars, such angles or tee bars, shall be of thickness and depth of leaf not less than specified for rounded bottoms of combustion chambers. Said angles or tee bars shall be supported on thimbles and riveted through with rivets not less than one inch in diameter and spaced not to exceed six inches between centers.

Rule to find working pressure allowed on rounded surfaces supported by angle irons or tee bars: Multiply constant by thickness squared in sixteenths and divide by the pitch multiplied by the diameter of curve.

FORMULA:

$$\frac{C \times T^2}{P \times D} = \text{working pressure}$$

LEGEND:

T = thickness of plate in sixteenths of an inch = $\frac{9}{16}$ = 81
P = pitch of angle or tee stiffeners in inches = 7 inches
D = diameter of curve to which plate is bent, in inches = 51 inches
C = constant = 900

EXAMPLE:

900 = constant
81 = thickness squared in 16ths
900
7200
72900

51″ = diameter
7″ = pitch
357

357)72900(204 lbs. = working pressure
714
1500
1428
72

TUBE PLATE

Rule to find the working pressure of a tube sheet supporting a crown sheet braced by crown bars: Subtract inside diameter of tubes in inches from the least horizontal distance between tube centers in inches; multiply the remainder by thickness of tube plate and then by constant 27,000; divide product by extreme width of combustion chamber multiplied by least horizontal distance between tube centers.

FORMULA:

$$\frac{(D-d)T\times C}{W\times D}=\text{working pressure}$$

LEGEND:

D = least horizontal distance between tube centers in inches = 4⅛ inches
d = inside diameter of tubes in inches = 2.782 inches
T = thickness of tube plate in inches = $\frac{11}{16}$ inches = .6875
W = extreme width of combustion chamber in inches = 34¼ inches
C = 27,000.

EXAMPLE:

4.125 = least horizontal distance
2.782 = inside diameter

1.343
.6875 = thickness of tube plate

6715
9401
10744
8058

.9233125
27000 = constant

646 31875000
1846 6250

24929.~~4375000~~

34.25 = extreme width
4.125 = least horizontal distance

17125
6850
3425
13700

141.~~28125~~

141)24929(176 lbs. = working pressure
141

1082
987

959
846

113

Rule to find thickness of plate for a tube sheet: Multiply pressure by width of fire box and by pitch of tubes (distance between centers) and divide this sum by pitch of tubes minus one inside diameter of one tube multiplied by constant.

FORMULA:

$$\frac{P\times W\times p}{(p-d)\times C}=\text{thickness of plate}$$

LEGEND:

p = pitch of tube $=4\frac{1}{8}$
d = inside diam. of tube = 2.782
P = pressure = 176 lbs.
C = constant = 27000
W = width of combustion chamber $=34\frac{1}{4}$ inches

EXAMPLE:

```
                                   176 =pressure lbs. per square inch
                                 34.25 =width of fire box
                                 -----
                                   880
                                  352
                                 704
                                528
     pitch of tubes =4.125      -------
       inside diam. =2.782     6028.00
                    -----        4.125 =pitch of tubes
                    1.343      -------
         constant =    27000   3014000
                     -------  1205600
                     9401000  602800
                    2686     2411200
                    -------  --------
                    36261000)24865.5000(.6857 or 11/16 nearly
                             217566
                             ------
                              310890
                              290088
                              ------
                               208020
                               181305
                               ------
                                267150
                                253827
                                ------
                                 13323
```

U. S. RULES.

The compressive stress on tube plates, as determined by the following formula, must not exceed 13,500 pounds per square inch, when pressure on tops of combustion chamber is supported by vertical plates of such chamber.

$$\frac{P\times D\times W}{2\times(D-d)\times T}=\text{compressive stress}$$

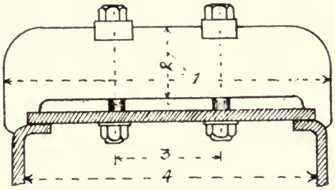
P = working pressure in pounds = 176 lbs.
D = least horizontal distance between tube centers in inches = 4.1250″
d = inside diameter of tube in inches = 2.782.
W = extreme width of combustion chamber in inches = 34¼
T = thickness of tube sheet in inches = $\frac{11}{16}$ = .6875.

EXAMPLE:

```
          176   = pressure
         4.1250 = distance tubes horizontally
         ------
           8800
          352
         176
        704
       --------
       726.0000
          34.25 = width of combustion chamber
       --------
       36300000
      14520000
     29040000
    21780000
    -----------
1.84662500)24865.50000000(13465 = compres-
           18466 2500              sive strain
           ----------
            6399 25000
            5539 87500
            ----------
             859 375000
             738 650000
             ----------
             120 7250000
             110 7975000
             -----------
               9 92750000
               9 23312500
               ----------
                 69437500
```

```
       4.1250 = dis. bet. tubes
       2.782  = inside diam. tube
       ------
       1.3430
            2 = twice
       ------
       2.6860
        .6875 = 11/16 = thickness of
       ------            tube sheet
       134300
      188020
     214880
    161160
   ----------
   1.84662500
```

Sling stays may be used in lieu of girders in all cases, provided, however, that when such sling stays are used, girders or screw stays of the same sectional area must be used for securing the bottom of combustion chamber to the boiler shell.



Rule to find thickness of steel girder: From length of girder subtract pitch of bolts and multiply by centers of girders and by length of same and this sum by pressure; divide this product by depth of girder squared multiplied by constant and then multiplied by the square root of number of supporting bolts.

FORMULA:

$$\frac{(L-P)\times G\times L\times P}{d^2\times C\times\sqrt{N}} = \text{thickness of girder required.}$$

LEGEND:

L = length of girder = 32″
P = pitch of bolts = 9″
G = girder centers = 8½″
d = depth of girder = 5.18″
C = constant = 6000
N = number of bolts = 9

EXAMPLE:

	32.000″ = length of girder
	9.000″ = pitch of bolts
	23.000″
	8.5″ = girder centers
	115000
	184000
depth of girders = 5.18	
5.18	195.5000
	32″ = length of girder
4144	
518	3910000
2590	5865000
26.8324	6256.0000
constant = 6000	160 = pressure
160994.4000	3753600000
sq. rt. of bolts = 3	62560000
$4829832\cancel{000}$)	$1000960\cancel{000}$ (2.0 = thickness of girder
	965966
	349940

In connection with rules covering girder calculations there are constants used and varying according to plate thickness and design of bolt, such as screwed stayed bolts with and without lock nuts, sockets, with riveted heads, number of bolts and water used, as follows:

Use constant 5400 for roof stays, wrought iron.
Use constant 6000 for roof stays, steel.

A constant used by Joshua Rose for computing girder or crown bar supporting bolts 9000 (this for steel).

Rule to find area of supporting bolts (steel) for a girder stay or crown bar. Multiply pressure by area to be supported and divide this product by constant 9000, this will give the pounds strain allowed per square inch of sectional area for a mild steel bolt.

FORMULA:

$$\frac{A \times P}{C} = \text{area supporting bolt required}$$

LEGEND:

A = area to be supported = 8″ × 8″ = 64 square inches
P = pressure = 170 lbs.
C = constant = 9000

EXAMPLE:

64 = square inches to be supported
170 = pressure

4480
64

constant = 9000) 10880.0000 (1.2088 = area = 1½ approximately
9000

1880 0
1800 0

800 00
720 00

80 000
72 000

8 000

Rule to find safe working pressure on a girder supporting a crown sheet of a back smoke box connection, when not subjected to heat in excess of ordinary steam pressures and assuming the combustion chamber ends are fitted to the edge of tube plate and the back of plate of the combustion box, four supporting bolts being used. Multiply constant by depth of girder squared in inches and multiply this sum by thickness of girder in inches; divide product by width of combustion chamber in inches minus pitch of supporting bolts multiplied by distance between girders from center to center in inches and again by length of girder in feet.

FORMULA:

$$\frac{C \times d^2 \times T}{(W-P) \times D \times L} = \text{pressure}$$

LEGEND:

W = width of combustion box in inches = 36″
P = pitch of supporting bolts in inches = 7½ = 7.5
D = distance between girder centers in inches = 7¾ = 7.75
L = length of girder in feet = 3 feet = 3
d = depth of girder in inches = 7½ = 7.5
T = thickness of girder in inches = 2″ = 2
C = constant = 550—when girder is fitted with one supporting bolt
" 825— " " " " " two or three supporting bolts
" 935— " " " " " four supporting bolts

EXAMPLE:

width = 36″
pitch = 7.5
———
28.5 56.25 = depth squared
distance = 7.75 935 = constant
——— ———
1 425 281 25
19 95 1687 5
199 5 50625
——— ———
220.875 52593.75
length = 3 2 = thickness
——— ———
662.~~625~~ 105187.~~50~~

662)105187. (158 or 159 lbs. nearly
662
———
3898
3310
———
5887
5296
———
591

Rule to find depth of steel girder for top of a combustion chamber: Multiply pressure by centers of girder and by length of girder bolts and multiply this sum by length of girder bolts minus pitch of same; divide this product by constant multiplied by thickness of girder and again by square root of number of bolts. The square root of quotient is depth of girder.

FORMULA:

$$\sqrt{\frac{P\times G\times L\times(L—p)}{C\times T\times\sqrt{N}}}=\text{depth of girder}$$

LEGEND:

P = pressure = 160 lbs.
G = girder centers = 8½.
L = length of girder = 32″
C = constant 6000 for steel
C = constant 54000 for iron
T = thickness of girder = 2″
p = pitch of bolts = 9″
N = number of bolts = 9
d = depth of girder = 9″

EXAMPLE:

```
                                     160 =pressure
                                     8.5 =girder centers
                                    ----
                                     800
                                   1280
                                  ------
                                  1360.0
                                      32 =length of girder
                                  ------
                                   27200
                                  40800
                                 -------
        constant =6000           43520.0
thickness of girder =   2″            23 =length of girder minus pitch of bolts
                    -----        -------
square root of      12000        1305600
     no. of bolts =     3        870400
                   ------      ---------
                   36000)1000960.0000(27.8044
                         72000
                         ------
                         280960          5)27.8044(5.272 = 5 9/32 nearly =
                         252000           )25          depth of girder
                         ------            ----
                          289600    102)  280
                          288000       )  204
                          ------         -----
                            160000 1047)   7644
                            144000     )   7329
                            ------        -----
                             160000 10542)  31500
                             144000      )  21084
                             ------         -----
                              16000      )  10416
```

English Board of Trade Rules Governing Girders.

Legend:

P = pressure.
W = width of combustion chamber
p = pitch of bolts
D = distance between girder centers
L = length of girder
d = depth of girder
T = thickness of girder
C = constant for number of bolts

Constants vary according to the iron or steel used, the lower constant for iron.

Constant = 6000 = when only one supporting bolt
“ 9000 to 9900 = when two or three supporting bolts
“ 10200 to 11220 = when four to five supporting bolts
For five bolts use same constant as for four
For six or seven bolts use constant 10500 for iron
“ “ “ “ “ “ “ 11550 “ steel

Formulas:

$$\frac{C \times d^2 \times T}{(W - \text{pitch}) \times D \times L} = \text{working pressure}$$

$$\frac{P \times (W - \text{pitch}) \times D \times L}{C \times d^2} = \text{thickness of girder}$$

$$\frac{P \times (W - \text{pitch}) \times D \times L}{C \times T} = \text{depth of girder}$$

REINFORCEMENT FOR HOLES CUT IN BOILER SHELL.

All holes exceeding 6 inches in diameter cut in either the flat heads or circumferential shell of steel boilers shall be reinforced with wrought or cast steel rings to compensate for the material removed. In lieu of such a reinforce ring, holes in flat heads may, if preferred, be reinforced by flanging the metal about the hole inward to a depth of not less than three-quarters of an inch measured from the inner surface. Reinforce rings on flat heads must be efficiently riveted to the head, and must have a sectional area not less than .8 the section of metal removed, the latter being measured across the shorter axis of the opening.

Reinforce rings on the circumferential shell must be efficiently riveted to the shell, and must have a sectional area not less than .7 the section of metal removed, the latter being measured across the hole in a direction parallel to the length of the boiler.

Reinforce rings should be of thickness not less than that of plate to which attached.

Rule to find width of ring to reinforce an opening in a boiler shell such as a man-hole, when one ring is used: Multiply diameter of opening longitudinally by the thickness of plate and divide the product by twice the thickness of reinforcement ring; add the diameter of rivet hole to quotient. This will be for single riveting and when double riveted add twice the diameter of rivet hole.

FORMULA:

$$\frac{O \times T}{2 \times N} + 1R = \text{width of ring for single riveted}$$

LEGEND:

R = rivet diameter hole = .9375
O = diameter of opening = 11″
T = thickness of shell = $\frac{1}{2}''$ = .5000
N = thickness of ring = $\frac{5}{8}''$ = .6250

EXAMPLE:

thickness of ring = .6250   11″ = diameter of opening
2   .5000 = thickness of shell plate
twice thickness of ring = 1.2500)5.50000(4.4
5 0000   .9375 = diam. of rivet hole
50000   5.3375 = $5\frac{11}{32}''$ nearly
50000

When two rings are used the thickness of each must be at least that of shell and have same tensile strength as that of shell plate; a *single* ring not less than 1¼ the thickness of shell.

Rule to find number of rivets to be used in a reinforcement ring for reinforcing an opening such as a man-hole in boiler shell: Multiply the net section of the ring by four times the tensile strength of the material and divide this product by the product of the shearing strength of rivet multiplied by its area.

FORMULA:

$$\frac{NS \times (4 \times TS)}{SS \times A} = \text{number of rivets required}$$

LEGEND:

NS = net section = 1.5625
SS = shearing strength = 38000
TS = tensile strength = 60000
A = area of rivet = .6013

EXAMPLE:

38000 = shearing strength
.6013 = area of rivet
114000
38000
228000
22849.4000

60000 = tensile strength
4 times
240000

1.5625 = net section
240000 = 4 times tensile strength
625000000
31250
22849)375000.0000(16 rivets ⅞″ diameter required
22849
146510
137094
9416

For a double riveted ring multiply net section of one ring by eight times the tensile strength of material and divide product by the sum obtained by multiplying 1.85 times the shearing strength of rivet's sectional area and the area of rivet.

CHAPTER V.

AMENDMENTS OF STEAMBOAT INSPECTION RULES AND REGULATIONS.

Lap welded boiler flues over 4 inches up to and including 30 inches in diameter shall be made of wrought iron or mild steel made by any process.

A test piece, 2 inches in length, cut from a tube, must stand being flattened by hammering until the sides are brought parallel with the curve on the inside at the ends not greater than three times the thickness of the metal without showing cracks or flaws, with bend at one side in the weld.

Each tube shall be subjected to an internal hydrostatic pressure of 500 pounds per square inch without showing signs of weakness or defects.

All steel tubes shall have ends properly annealed by the manufacturer before shipment. Tubes must stand drilling, riveting, and calking, and work necessary to install them into the tube head without showing any signs of weakness or defects.

No tube increased in thickness by welding one tube inside of another shall be allowed for use.

SEAMLESS STEEL BOILER TUBES.

MATERIAL.

The steel shall be made by the open-hearth process.

SURFACE INSPECTION.

The pipe must be free, inside and outside, from all surface defects that would materially weaken it or form starting points of corrosion. The defects to be especially avoided are snakes, checks, slivers, laps, pits, etc. Pipe must be smooth and straight.

The following tests shall be made before shipment by the manufacturer:

(*a*) A test piece, 2 inches in length, cut from a tube, must stand being flattened by hammering until the sides are brought parallel with the curve on the inside at the ends not greater than three times the thickness of the metal without showing cracks or flaws.

(*b*) Pulling tests must be made from every 50 pieces furnished, or fraction thereof, and must show the following results:

Tensile strength, not less than 48,000 pounds per square inch.

Elongation in 8-inch specimen, not less than 12 per cent.

The results of the pulling tests must be forwarded by the manufacturer to the purchaser of steam pipe, who will forward same to local inspector.

Any pipe used for mud or steam drums must have the ends of same properly annealed before the holes are drilled or the heads are riveted in: *Provided,* That this paragraph shall apply only to drums not exceeding 15 inches in diameter for use on pipe and coil boilers.

When pipe is used for steam lines where flanges are riveted on and calked, the ends of the pipe shall be properly annealed before drilling or riveting the flanges on.

When pipes are expanded into flanges by proper and approved machinery, and flared out at the ends to an angle not exceeding 20° (said angle to be taken in the direction of the length of the pipe) and having a depth of flare equal to *at least* one and one-half times the thickness of the material in said pipe, such pipes may be used for all steam and exhaust pipes when tested to two and one-half times the working pressure and found perfect in every respect.

If the pipe is used for steam lines where the pipe is peened in and flanged over, the ends of the pipe should be properly annealed before the peening or flanging is done.

The use of a square-nosed tool is recommended for cutting tubes and pipe.

Provided, That this entire section shall apply only to tubes and pipes used or to be used in boilers built after June 30, 1905, and to all other pipes referred to in this section subject to pressure installed for use on steam vessels after that date.

TABLES AND EXAMPLES.

Flues and furnaces safe working pressures.

The following table shows diameters, thickness of plate and safe working pressure on flues in sections of 3 feet, maximum length allowed 5 feet ,also sections of 30″ in length, maximum 40″.

Table of Steam Pressure per Square Inch Allowable on Riveted and Lap-Welded Flues Made in Sections and Used in Boilers Whose Construction is Commenced After June 30, 1905.

Thickness of material.	Greatest length of sections allowable, 5 feet.				Greatest length of sections allowable, 3 feet.												
	Least thickness of material allowable.																
	.18 inch.	.20 inch.	.21 inch.	.21 inch.	.22 inch.	.22 inch.	.23 inch.	.24 inch.	.25 inch.	.26 inch.	.27 inch.	.28 inch.	.29 inch.	.30 inch.	.31 inch.	.32 inch.	.33 inch.
	Diameter of flues.																
	Over 6, not over 7 inches	Over 7, not over 8 inches	Over 8, not over 9 inches	Over 9, not over 10 inches	Over 10, not over 11 inches	Over 11, not over 12 inches	Over 12, not over 13 inches	Over 13, not over 14 inches	Over 14, not over 15 inches	Over 15, not over 16 inches	Over 16, not over 17 inches	Over 17, not over 18 inches	Over 18, not over 19 inches	Over 19, not over 20 inches	Over 20, not over 21 inches	Over 21, not over 22 inches	Over 22, not over 23 inches
	Lbs. pressure.	*Lbs. pressure.*	*Lbs. pressure.*	*Lbs. pressure.*	*Lbs. pressure.*	*Lbs. pressure.*	*Lbs. pressuce.*	*Lbs. pressure.*	*Lbs. pressure.*	*Lbs. pressure.*	*Lbs. pressure.*	*Lbs. pressure.*	*Lbs. pressure.*	*Lbs. pressure.*	*Lbs. pressure.*	*Lbs. pressure.*	*Lbs. pressure.*
.18-inch	205																
.19-inch	217																
.20-inch	228	200															
.21-inch	240	210	186	168													
.22-inch		220	195	176	160	146											
.23-inch		230	204	184	167	153	141										
.24-inch		240	213	192	174	160	147	137									
.25-inch			222	200	181	166	153	142	133								
.26-inch			231	208	189	173	160	148	138	130							
.27-inch				216	196	180	166	154	144	135	127						
.28-inch					203	186	172	160	149	140	131	124					
.29-inch						193	178	165	154	145	136	128	122				
.30-inch							184	171	160	150	141	133	126	120			
.31-inch							190	177	165	155	145	137	130	124	118		
.32-inch								182	170	160	150	142	134	128	121	116	
.33-inch								188	176	165	155	146	138	132	125	120	114
.34-inch									181	170	160	151	143	136	129	123	118
.35-inch									186	175	164	155	147	140	133	127	121
.36-inch										180	169	160	151	144	137	130	125
.37-inch											174	164	155	148	140	134	128
.38-inch												168	160	152	144	138	132
.39-inch													164	156	148	141	135
.40-inch														160	152	145	139
.41-inch															156	149	141
.42-inch																152	146
.43-inch																	149

Thickness of material.	Greatest length of sections allowable, 30 inches.																
	Least thickness of material allowable.																
	.34 inch.	.35 inch.	.36 inch.	.37 inch.	.38 inch.	.39 inch.	.40 inch.	.41 inch.	.42 inch.	.43 inch.	.44 inch.	.45 inch.	.46 inch.	.47 inch.	.48 inch.	.49 inch.	.50 inch.
	Diameter of flues.																
	Over 23, not over 24 inches	Over 24, not over 25 inches	Over 25, not over 26 inches	Over 26, not over 27 inches	Over 27, not over 28 inches	Over 28, not over 29 inches	Over 29, not over 30 inches	Over 30, not over 31 inches	Over 31, not over 32 inches	Over 32, not over 33 inches	Over 33, not over 34 inches	Over 34, not over 35 inches	Over 35, not over 36 inches	Over 36, not over 37 inches	Over 37, not over 38 inches	Over 38, not over 39 inches	Over 39, not over 40 inches
	Lbs. pressure.	*Lbs. pressure.*	*Lbs. pressure.*	*Lbs. pressure.*	*Lbs. pressure.*	*Lbs. pressure.*	*Lbs. pressure.*	*Lbs. pressure.*	*Lbs. pressure.*	*Lbs. pressure.*	*Lbs. pressure.*	*Lbs. pressure.*	*Lbs. pressure.*	*Lbs. pressure.*	*Lbs. pressure.*	*Lbs. pressure.*	*Lbs. pressure.*
.34-inch.	113																
.35-inch.	116	112															
.36-inch.	120	115	110														
.37-inch.	123	118	113	109													
.38-inch.	126	121	116	112	108												
.39-inch.	130	124	120	115	111	107											
.40-inch.	133	128	123	118	114	110	106										
.41-inch.	136	131	126	121	117	113	109	105									
.42-inch.	140	134	129	124	120	116	112	108	105								
.43-inch.	143	137	132	127	123	118	114	111	107	103							
.44-inch.	146	140	135	130	125	121	117	113	110	106	103						
.45-inch.		144	138	133	128	124	120	116	112	109	105	102					
.46-inch.			141	136	131	127	122	118	115	111	108	105	102				
.47-inch.				139	134	129	125	121	117	113	110	107	104	101			
.48-inch.					137	132	128	123	120	116	112	109	106	103	101		
.49-inch.						135	130	126	122	118	115	112	108	105	103	100	
.50-inch.							133	128	125	121	117	114	111	108	105	102	100
.51-inch.								131	127	123	120	116	113	110	107	104	102
.52-inch.									130	126	122	118	115	112	109	106	104
.53-inch.										128	124	121	117	114	111	108	106
.54-inch.											127	123	120	116	113	110	108
.55-inch.												125	122	118	115	112	110
.56-inch.													124	121	117	114	112
.57-inch.														123	120	116	114
.58-inch.															122	119	116
.59-inch.																121	118
.60-inch.																	120

Rule to find steam pressure allowed on any flue in table: Multiply crushing strain 8,000 pounds (constant) by thickness noted in column and divide the product by diameter of flue.

FIRST EXAMPLE:

FORMULA:

$$\frac{C \times T}{D} = \text{working pressure allowed}$$

LEGEND:

D = diameter of flue = 15″
T = thickness of plate = ¼ = .25″
C = constant = crushing strain = 8000

EXAMPLE:

8000 = constant
.25 = thickness of plate
40000
16000
flue diameter = 15) 2000.00 (133⅓ lbs. working pressure
15
50
45
50
45
5

SECOND EXAMPLE:

LEGEND:

T = thickness of plate = ⅜ = .375
D = diameter of furnace = 36″
C = constant = 8000 = crushing strain

EXAMPLE:

8000 = constant
.375 = ⅜ = thickness of plate
40000
56000
24000
diameter = 36″) 3000000 (83.3 = lbs. working pressure
288
120
108
120
108
12

FLUES.

The preceding table includes all such riveted and lap-welded flues, exceeding 6 inches in diameter and not exceeding 40 inches in diameter, not otherwise provided for by law.

For any such flue requiring more pressure than is given in table, the same will be determined by proportion of thickness to any given pressure in table to thickness for pressure required, as per example:

A flue not over 19 inches in diameter and 3 feet long requires a thickness of .39 of an inch for 176 pounds pressure; what thickness would be required for 250 pounds pressure?

FORMULA:

$$\frac{\text{Pressure required} \times T}{P} = \text{thickness of plate required}$$

LEGEND:

P = pressure = 176 lbs.
T = thickness of plate = .39 or $\frac{3}{8}$ nearly

EXAMPLE:

250 = increased pressure required
.39 = thickness of plate

2250
750

first pressure = 176) 97.5000 (.5539 = $\frac{9}{16}$ nearly = thickness of plate required
88 0

9 50
8 80

700
528

1720
1580

136

Or, if .39 inch thickness gives a pressure of 176 pounds, what will .554 inch thickness give?

FORMULA:

$$\frac{\text{Thickness of plate required} \times P}{T} = \text{pressure}$$

EXAMPLE:

```
                                      .554 =thickness of plate
                                       176 =first pressure
                                    -------
                                      3324
                                     3878
                                     554
                                    -------
original thickness of plate = .39)97.504(250 =lbs. pressure
                                  78
                                  ----
                                  195
                                  195
                                  ----
                                    0
```

And all such flues shall be made in sections, according to their respective diameters, not to exceed the lengths prescribed in the table, and such sections shall be properly fitted one into the other and substantially riveted, and the thickness of material required for any such flue of a given diameter shall in no case be less than the least thickness prescribed in the table for any such given diameter; and all such flues may be allowed the prescribed working steam pressure if, in the opinion of the inspectors, it is deemed safe to make such allowance. Inspectors are therefore required, from actual measurement of each flue, to make such reduction from the prescribed working steam pressure for any material deviation in the uniformity of the thickness of material, or for any material deviation in the form of the flue from that of a true circle, as in their judgment the safety of navigation may require.

FURNACES.

The tensile strength of steel used in constructing furnaces shall not exceed 67,000, and be not less than 58,000 pounds. The minimum elongation in 8 inches shall be 20 per cent.

All corrugated furnaces having plain parts at the ends not exceeding 9 inches in length (except flues especially provided for), when new, and made to practically true circles, shall be allowed a steam pressure in accordance with the following formula:

$$\frac{C \times T}{D} = \text{pressure}$$

Rule to find collapsing pressure of a spirally corrugated furnace, corrugations 1½″ deep: Multiply square of thickness of flue in thirty-seconds of an inch by the constant 1200 and divide by external diameter of flue in inches multiplied by square root of length in inches.

LEGEND:

L = length = 81″
D = diameter = 40″
T = thickness = ⅝ = 20/32
C = constant = 1200

FORMULA:

$$\frac{T^2 \times 1200}{D \times \sqrt{L}} = \text{collapsing pressure}$$

9)81″(9 = sq. root
81 of length

40″ diameter
9 square root of length
360

EXAMPLE:

20 = thickness in 32nds of an inch
20
400 = thickness squared
1200 = constant
80000
400
360)480000(1333 lbs. collapsing pressure
360
1200
1080
1200
1080
1200
1080
120

MORISON CORRUGATED TYPE.

[In calculating the mean diameter of the Morison furnace, the least inside diameter plus 2 inches may be taken as the mean diameter, thus—

(Mean diameter = least inside diameter + 2 inches.)

Rule to find safe working pressure on a Morison corrugated furnace: Multiply constant 15,600 by thickness of plate and divide by diameter.

T = thickness in inches, not less than five-sixteenths of an inch.
C = 15600, a constant, determined from an actual destructive test under the supervision of the Board of Supervising Inspectors, when corrugations are not more than 8 inches from center to center, and the radius of the outer corrugations is not more than one-half of the suspension curve.

FORMULA:

$$\frac{C \times T}{D} = \text{working pressure}$$

LEGEND:

D = diameter = 42″
T = thickness of plate = ½ = .5
C = constant = 15600

EXAMPLE:

```
                     15600 = constant
                        .5 = thickness of plate
                     -----
diameter = 42″) 7800.0 (185 lbs. working pressure
                42
                ---
                360
                336
                ---
                 240
                 210
                 ---
                  30
```

COLLAPSING.

Rules for determining the collapsing pressures on furnace flues are given by eminent authorities, and these after many tests and experiments. These rules vary in method of computing and in the results; however, there is a reasonable margin for safety in the maximum results.

Hutton's rule for finding collapsing pressure:

Multiply the constant 806,300 by thickness of plate squared in inches and divide product by length of furnace in feet multiplied by diameter in inches.

FORMULA:

$$\frac{C \times T^2}{L \times D} = \text{collapsing pressure}$$

LEGEND:

C = constant = 806300
T = thickness of plate = ⅜ = .3750
D = diameter = 38″
L = length of furnace = 14 feet

EXAMPLE:

806300 = constant
.14062500 = thickness squared

length = 14 403150000
diameter = 38 1612600
4837800
152 3225200
38 806300

532)113385.~~93750000~~(213 lbs = collapsing pressure
1064

698
532

1665
1596

69

Nystrom's rule for finding collapsing pressure:

FORMULA:

C = constant = 200000
Other data same

$$\frac{T^2 \times C}{D \times \sqrt{L}} = \text{collapsing pressure}$$

EXAMPLE:

.14062500 = thickness squared
200000 = constant

142.12)28125.00~~000000~~(197 lbs. = collapsing pressure
14212

diameter = 38 139130
sq. rt. of length = 3.74 127908

1 52
26 6 112220
114 99484

142.12 12736

Rule by Michael Longridge for finding collapsing pressure: Multiply constant 174,000 by thickness of plate squared in inches; divide product by diameter multiplied by the square root of length.

FORMULA:

$$\frac{T^2 \times C}{D \times \sqrt{L}} = \text{collapsing pressure}$$

C = constant = 174000
other data same

EXAMPLE:

length =	38	.14062500 = thickness squared
sq. rt. of diam. =	3.74	174000 = constant
	1 52	56250000000
	26 6	98437500
	114	14062500

142.12)$24468.75\cancel{0}\cancel{0}\cancel{0}\cancel{0}\cancel{0}\cancel{0}$(172 lbs. = collapsing pressure
14212

102567
99484

30835
28424

2411

LEEDS SUSPENSION BULB FURNACE.

Rule to find safe working pressure on a Leeds suspension bulb furnace: Multiply constant 17,300 by thickness of plate and divide by diameter.

T = thickness in inches, not less than five-sixteenths of an inch.
C = a constant, 17300, determined from an actual destructive test under the supervision of the Board, when corrugations are not more than 8 inches from center to center, and not less than 2¼ inches deep.

FORMULA:

$$\frac{C \times T}{D} = \text{working pressure}$$

LEGEND:

C = constant = 17300
T = thickness = ⅜ = .375
D = diameter = 36″

EXAMPLE:

17300 = constant
.375 = thickness of plate

86500
121100
51900

diameter 36″)$6487.\cancel{5}\cancel{0}\cancel{0}$(180 lbs. working pressure
36

288
288

07

FOX TYPE.

Rule to find safe working pressure on the above type of furnace: Multiply constant 14,000 by thickness of plate and divide by diameter.

T = thickness in inches, not less than five-sixteenths.
C = 14000, a constant, when corrugations are not more than 8 inches from center to center and not less than $1\frac{1}{2}$ inches deep.

FORMULA:

$$\frac{C \times T}{D} = \text{safe working pressure}$$

LEGEND:

D = diameter = 40″
T = thickness of plate = $\frac{1}{2}$″ = .5
C = constant = 14000

EXAMPLE:

```
                   14000 = constant
                      .5 = thickness of plate
                   ------
Diameter = 40″)    7000.0 (175 lbs. working pressure
                   40
                   ---
                   300
                   280
                   ---
                    200
                    200
                    ---
```

PURVES TYPE.

FORMULA:

$$\frac{C \times T}{D} = \text{pressure}$$

T = thickness in inches not less than seven-sixteenths.
D = least outside diameter in inches.
C = 14000, a constant, when rib projections are not more than 9 inches from center to center and not less than $1\frac{3}{8}$ inches deep.

BROWN TYPE.

$$\frac{C \times T}{D} = \text{pressure}$$

T = thickness in inches, not less than five-sixteenths.
D = least outside diameter in inches.
C = 14000, a constant (ascertained by an actual destructive test under the supervision of the Board of Supervising Inspectors), when corrugations are not more than 9 inches from center to center and not less than $1\frac{5}{8}$ inches deep.

The thickness of corrugated and ribbed furnaces shall be ascertained by actual measurement. The manufacturer shall have said furnace drilled for a one-fourth inch pipe tap and fitted with a screw plug that can be removed by the inspector when taking this measurement. For the Brown and Purves furnaces the holes shall be in the center of the second flat; for the Morison, Fox, and other similar types in the center of the top corrugation, at least as far in as the fourth corrugation from the end of the furnace.

Type Having Sections 18 Inches Long.

$$\frac{C \times T}{D} = \text{pressure}$$

T = thickness in inches, not less than seven-sixteenths.
D = mean diameter in inches.
C = 100000, a constant, when corrugated by sections not more than 18 inches from center to center and not less than 2½ inches deep, measuring from the least inside to the greatest outside diameter of the corrugations, and having the ends fitted one into the other and substantially riveted together, provided that the plain parts at the ends do not exceed 12 inches in length.

CONES.

Rule to find collapsing pressure on a truncated cone up to 42 inches in length: Multiply twice thickness of plate by the tensile strength and by the hypothenuse length of cone; divide this sum by the square inches in a trapezoid of equal dimensions of truncated cone.

Formula:

$$\frac{2 \times T \times TS \times \text{Hypothenuse}}{\text{Area of trapezoid}} = \text{bursting pressure}$$

Legend:

T = thickness of plate = ⅜ = .375
TS = tensile strength = 60000
Hypothenuse = 40″
Area of trapezoid = 1200

Example:

.7500 = twice thickness of $\frac{3}{8}''$ plate
60000 = tensile strength

45000.0000
40'' = length of hypothenuse of cone

area of a trapezoid = 1200)1800000.0000(1500 lbs. bursting pressure
1200

6000
6000

CONE TOPS.

Flues used in vertical boilers as upper combustion chambers formed in the shape of a cone, when new and made to practically true circles, shall be allowed a steam pressure according to the following formula:

$$\frac{C \times T}{D} = \text{pressure}$$

T = thickness of flue in inches, not less than five-sixteenths.
D = outside diameter in inches, at the center of the length of the flue, not to exceed 42 inches.
C = 10153, a constant, when the length of the flue does not exceed 42 inches, measuring from center of rivet holes in top of head to the center of rivet holes in the tube head.

When the flue exceeds 42 inches in diameter at the center, it shall be deemed flat surface and must be stayed accordingly.

Rule to find safe working pressure on a truncated cone as in a submerged tube upright boiler, length limited to 40'': Multiply constant 8000 by thickness of plate, minimum limit 5/16, and divide by diameter (small and large diameter added together and divided by 2).

Legend:

C = constant = 8000
T = thickness of plate = $\frac{7}{16}$ = .4375
D = diameter, small = 30''
" large = 40''

Formula:

$$\frac{C \times T}{D} = \text{working pressure}$$

Example:

large diam. 40''
small diam. 30''
2)70
35'

8000 = constant
.4375 = $\frac{7}{16}$ plate

35)3500.0000(100 lbs. pressure
35
00

ADAMSON TYPE.

When plain horizontal flues are made in sections not less than 18 inches in length, and not less than five-sixteenths of an inch thick, and flanged to a depth of not less than three times the diameter of rivet hole plus the radius at furnace wall (inside diameter of furnace), the thickness of the flanges shall be as near the thickness of the body of the plate as practicable.

The radii of the flanges on the fire side shall be not less than three times the thickness of plate.

The distance from the edge of the rivet hole to the edge of the flange shall be not less than the diameter of the rivet hole, and the diameter of the rivets before driven shall be at least one-fourth inch larger than the thickness of the plate.

The depth of the ring between the flanges shall be not less than three times the diameter of the rivet holes, and the ring shall be substantially riveted to the flanges. The fire edge of the ring shall terminate at or about the point of tangency to the curve of the flange, and the thickness of the ring shall be not less than one-half inch.

Plain Circular Furnaces or Flues, and Adamson Furnaces Made in Sections Not Less Than 18 Inches in Length.

Rule to find safe working pressure of an Adamson furnace: Multiply length of section by thickness of plate in sixteenths; from this product subtract the length of furnace multiplied by constant 1.03; multiply result by constant 51.5 divided by the diameter.

Formula:

$$[S \times T - (L \times 1.03)] \times \frac{51.5}{D} = \text{pressure}$$

Legend:

S = length of section = 18¾
D = outside diameter of furnace in inches = 44″
L = length of furnace in inches = 48″
T = thickness of plate in sixteenths of an inch = ½ = $\frac{8}{16}$
C = constant = 51.5
C = constant = 1.03

Example:

diameter = 44)51.5(1.17	18.75 = length of section	
41	8 = thickness of plate in 16ths	
7 5	150.00	48 = length of furnace
4 4	49.44	1.03 = constant
3 10	100.56	1 44
3 08	1.17	48
2	7 0392	49.44
	10 056	
	100 56	
	117.6552 = safe working pressure	

VERTICAL TYPE.

Cylindrical flues used as furnaces in vertical boilers, when new, and made to practically true circles, shall be allowed a steam pressure by the following formula:

$$\frac{C \times T}{D} = \text{pressure}$$

T = thickness of flue in inches, not less than one-fourth.
D = outside diameter of flue in inches, not to exceed 42 inches.
C = 10,577, a constant, when the length of the flue does not exceed 42 inches, measuring from the center of the rivet holes in the head to the center of the rivet holes in the leg.

When the flue exceeds 42 inches in diameter, it is deemed to be flat surface and must be stayed accordingly.

STEAM CHIMNEY FLUES.

The Morison, Fox, Purves, or Brown types of corrugated furnaces may be used as flues for steam chimneys or superheaters and shall be allowed a steam pressure by their respective formulas, and other flues, as described below, when new and made to practically true circles and shall be allowed a steam pressure by the following formula:

$$\frac{C \times T}{D} = \text{pressure}$$

T = thickness of material in inches.
D = outside diameter of flue in inches.
C = 12000 for flues under 30 inches in diameter, plates at least five-sixteenths of an inch thick, supported by angle rings at least 2½ by 2½ inches.
C = 12000 for flues 30 inches and under 45 inches in diameter, plates at least three-eighths of an inch thick, supported by angle rings at least 2½ by 2½ inches.
C = 12000 for flues 45 inches and under 55 inches in diameter, plates at least seven-sixteenths of an inch thick, supported by angle rings at least 3 by 3 inches.
C = 12000 for flues 55 inches and under 65 inches in diameter, plates at least one-half inch thick, supported by angle rings at least 3 by 3 inches.
C = 12000 for flues 65 inches and under 75 inches in diameter, plates least nine-sixteenths of an inch thick, supported by angle rings at least 3½ by 3½ inches.
C = 12000 for flues 75 inches and under 85 inches in diameter, plates at least five-eighths of an inch thick, supported by angle rings at least 3½ by 3½ inches.
C = 12000 for flues 85 inches in diameter, plates at least eleven-sixteenths of an inch thick, supported by angle rings at least 4 by 4 inches.

For flues over 85 inches in diameter, add one-sixteenth of an inch to eleven-sixteenths of an inch for every 10 inches increase in the diameter of the flue.

The distance, center to center, between angle rings, or center of angle rings to center of rivets in the heads, shall in no case exceed 2½ feet.

The angle rings shall be accurately fitted and substantially riveted to the flue and connected to the outer shell by braces, which braces shall not exceed 20 inches from center to center on the flue.

ADAMSON RINGS.

Adamson rings may be substituted for the angle rings, but each ring shall not be at a greater distance than 2½ feet from center to center of rings, which rings shall not be required to be braced to the outer shell.

Rule to find the working pressure of an Adamson flue used in a steam chimney: Multiply constant by thickness of plate in inches and divide by diameter.

LEGEND:

T = thickness of plate = ½ = .5
D = diameter = 45″
C = constant = 12000

FORMULA:

$$\frac{C \times T}{D} = \text{working pressure}$$

Example:

```
                     12000 =constant
                        .5 =thickness of plate
                     -----
        diameter=45)60000(133 lbs. pressure
                    45
                    --
                    150
                    135
                    ---
                    150
                    135
                    ---
                     15
```

Rule by Hutton to find collapsing pressure of ribbed furnace with ribs 9 inches centers and not less than 15/16 deep: Multiply thickness of straight or plain part of furnace flue in squared thirty-seconds by constant 1350 and divide by external diameter multiplied by square root of length.

Formula:

$$\frac{T^2 \times 1350}{D \times \sqrt{L}} = \text{collapsing pressure}$$

Legend:

D = diameter = 30″
L = length = 81″
T = thickness of plate = 13/32

Example:

```
                                    169  =13/32 squared
                                    1350 =constant
                                    ----
                                    8450
                 diameter=30"      507
    square root of length=  9     169
                          ---     ------
                          270)228150(845 lbs. collapsing pressure
                              2160
                              ----
                               1215
                               1080
                               ----
                                1350
                                1350
                                ----
```

Rule to get compressive strain on a furnace flue from a collapsing pressure: Multiply diameter of flue by pressure and divide product by twice the thickness of flue plate.

FORMULA:

$$\frac{D \times P}{2 \times T} = \text{compressive strain}$$

LEGEND:

D = diameter = 30″
P = collapsing pressure = 845
T = thickness of flue plate = 13/32 = .40625

EXAMPLE:

```
                               30″ = diameter
                              845 lbs. = collapsing pressure
                              ___
                              150
thickness = .40625           120
                 2          240
          ______        ________
twice thickness = .81250)25350.0000(3120 lbs. compressive strain
                         243750
                         ______
                          97500
                          81250
                         ______
                         162500
                         162500
                         ______
```

PLAIN FLUES.

Rule to find the working pressure of a plain flue used in a steam chimney: Multiply constant by thickness in inches and divide by diameter.

FORMULA:

$$\frac{C \times T}{D} = \text{pressure}$$

LEGEND:

L = length of chimney = 8 ft.
T = thickness of material in inches = $\frac{11}{16}$.
D = outside diameter of flue in inches = 46″.
C = 8000 for flues under 32 inches in diameter, plates at least five-eighths of an inch thick, and not exceeding 8 feet in length.
C = 8000 for flues over 32 inches and under 46 inches in diameter, plates at least eleven-sixteenths of an inch thick, and not exceeding 8 feet in length.

SOCKET BOLTS.

For all boilers carrying a steam pressure of 60 pounds and under per square inch the flue may be braced with socket bolts in lieu of angle rings, such bolts to have heads and the ends to be threaded for nuts, with plate washers not over 12 inches between centers (or equivalent) on the inside of the flue; bolts to be at least 1 inch in diameter at bottom of thread.

For all boilers carrying a steam pressure of over 60 pounds and not over 120 pounds per square inch the flue may be braced with socket bolts in lieu of angle rings, such bolts to have heads and the ends to be threaded for nuts, with plate washers not over 10 inches between centers (or equivalent) on the inside of flue; bolts to be at least 1⅛ inches in diameter at bottom of thread.

Plain flues, Adamson flues, and flues supported by angle bars, when used as furnaces, shall in no case be allowed a greater working pressure than found by the above formulas.

LIMITED FORMULA:

$$\frac{C \times T}{D} = \text{pressure}$$

LEGEND:

C = constant = 9900
T = thickness of plate = ½ = .5
D = diameter = 40″

EXAMPLE:

$$\begin{array}{rl} 9900 & = \text{constant} \\ .5 & = \text{thickness of plate} \\ \hline \end{array}$$

$$\begin{array}{r l} \text{diameter} = 40'')\,49500\,(123\tfrac{3}{4} & \text{lbs. pressure} \\ 40\phantom{(123\tfrac{3}{4}} & \\ \hline 95\phantom{(123\tfrac{3}{4}} & \\ 80\phantom{(123\tfrac{3}{4}} & \\ \hline 150\phantom{(123\tfrac{3}{4}} & \\ 120\phantom{(123\tfrac{3}{4}} & \\ \hline 30\phantom{(123\tfrac{3}{4}} & \end{array}$$

Hutton's rule to find collapsing pressure on a furnace flue lap riveted or flange connected: Multiply thickness of plate squared 32nds by constant 660 and divide by diameter multiplied by square root of length.

LEGEND:

T = thickness = $\frac{3}{8}$ = $.\frac{12}{32}$
C = constant = 660
D = diameter = 36″
L = length = 64″

FORMULA:

$$\frac{T^2 \times C}{D \times \sqrt{L}} = \text{collapsing pressure}$$

EXAMPLE:

.144 = thickness squared in 32nds
660 = constant

diameter = 36 8640
sq. root of length = 8 864

288) 95040 (330 lbs. = collapsing pressure
864

864
864

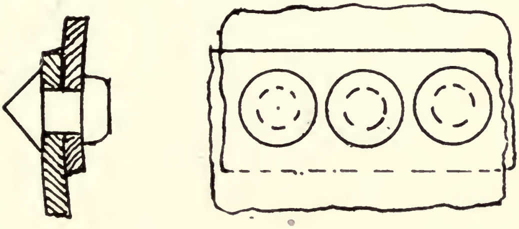
0

TABLE OF COLLAPSING PRESSURES OF FURNACES BY W. S. HUTTON.

Diameter in inches	Length in inches	Thickness in 32nd	Collapsing pressures, lbs. per square inch
33.5	360	11	113
42	420	12	100
42	300	12	119
54	36	8	120
38	86	16	436
36	24	8	218
36	24	12	490
36	48	12	350
43	23	17	842

CHAPTER VI.

SINGLE RIVETED LAP JOINT.



Causes for failure at joint:
First—Shearing of one rivet.
Second—Tearing of plate between rivets.
Third—Crushing of rivet or plate.

In calculating seams it will be necessary to have some data, and to follow this out we will assume it to be as follows:

LEGEND:

TS = tensile strength = 60000
CS = resistance to crushing = 95000
SS = resistance to shearing of rivet = 38000
D = diameter of boiler = 48″
d = diameter of rivet hole = 13/16 = .8125
A = area of rivet hole = .5185
T = thickness of plate = 5/16 = .3125
P = pitch = 1⅞ = 1.8750
F = factor of safety = 5

First:—resistance to shear one rivet

FORMULA:

A × SS = resistance to shearing of one rivet.

EXAMPLE:

.5185 = area of rivet
38000 = shearing strength of rivet, single shear
41480000
15555
19703.0000

19,703 lbs. shearing strength of one rivet.

Second:—tearing the plate between rivets.

Rule to find strength of net section of plate: From pitch of rivet, subtract diameter of rivet hole and multiply this sum by thickness of plate. Multiply this product by the tensile strength of plate.

FORMULA:

(P—d) × T × TS = strength of net section of plate.

EXAMPLE:

1.8750 = pitch
.8125 = diameter of rivet
1.0625
.3125 = thickness of plate
53125
21250
10625
31875
.33203125
60000 = tensile strength
19921.~~87500000~~
19,921 = strength of net section of plate.

Third:—resistance to crushing of rivet or plate.

FORMULA:

P × T × TS = strength of solid plate.

EXAMPLE:

1.8750 = pitch of rivets
.3125 = thickness of plate
93750
37500
18750
56250
.58593750
60000 = tensile strength
35156.~~25000000~~
35,156 lbs. strength solid plate

The net section of rivets is the weakest part of joint. To find the efficiency of joint, multiply the weakest section by constant 100 and divide by the strength of solid plate.

EXAMPLE:

19,703 = shearing resistance of one rivet
35,156 = strength of solid plate

```
35,156)19,703.00(.56 = 56% efficiency of joint
       17 579 0
       --------
        2 125 00
        2 109 36
        --------
```

Rule to find safe working pressure from these calculations: Multiply the tensile strength of plate by the efficiency of joint and this sum by twice the thickness of plate; divide this product by diameter of boiler multiplied by factor of safety.

FORMULA:

$$\frac{TS \times \% \times (2 \times T)}{D \times F} = \text{safe working pressure}$$

EXAMPLE:

```
                             60000 = tensile strength
                               .56 = % efficiency
                           -------
                           3600 00
                          30000 0
                          --------
                          33600.00
                             .6250 = twice thickness of plate
                         ---------
                         168000000
 diam. of boiler = 48"    6720000
 factor of safety =   5  20160000
                   ----  --------
                    240)21000.000000(87.5 lbs. = working pressure
                        1920
                        ----
                         1800
                         1680
                         ----
                          1200
                          1200
                          ----
```

Rule to find thickness of plate: Multiply pressure by factor 6 and multiply again by radius or one-half diameter of boiler and divide product by tensile strength of plate; the quotient will be thickness of plate.

Legend:

F = factor = 6
R = radius = 30″ or one-half diameter.
TS = tensile strength = 60000
P = pressure = 125 lbs.

Formula:

$$\frac{P \times F \times R}{TS} = \text{thickness of plate}$$

Example:

$$\begin{array}{rl} 125 & = \text{lbs. pressure} \\ 6 & = \text{factor} \\ \hline 750 & \\ 30 & = \text{radius} \\ \hline \end{array}$$

$$\text{tensile strength} = 60000)\,22500000\,(3750 = \tfrac{3}{8}\text{ plate}$$

$$\begin{array}{r} 180000 \\ \hline 450000 \\ 420000 \\ \hline 300000 \\ 300000 \\ \hline \end{array}$$

Rule to find pitch of rivets for single, double and triple riveted lap joints when the shearing strength of rivets is near equal to strength of net section of plate: Multiply area of rivet hole by the shearing resistance of rivets and by number of rows of rivets; divide product by thickness of plate multiplied by tensile strength; add to quotient the diameter of rivet hole.

Formula:

$$\frac{A \times SS \times N}{T \times TS} + DH = \text{pitch single riveted joint}$$

Legend:

A = area of rivet = 15/16 = .6903
SS = shearing strength of rivet = 38000
N = number of rows of rivets = 1
T = thickness of plate = .4375
TS = tensile strength = 60000
DH = diameter of rivet hole = .9375

EXAMPLE:

```
                                    .6903    =area of rivet
                                       38000 =shearing strength
                                    --------
 plate thickness = .4375             55224000
tensile strength =      60000      20709
                  ----------       --------
              26250.0000)26231.4000(.9992
                         23625 0      .9375 =diameter of rivet hole
                         -------   -------
                          2606 40  1.9367 =1 15/16 =pitch
                          2362 50
                          -------
                           243 900
                           236 250
                           -------
                             7 6500
                             5 2500
                             ------
                             2 4000
```

Custom through using iron rivets has established a rule to make the rivet hole 1-16 larger than the rivet, but owing to a better rivet material and use of steel rivets, experience has proved that less than 1-16 larger is better.

Rule to find diameter of a rivet for a single riveted lap joint—steel rivets and plate: Add to plate thickness 7-16 of an inch.

FORMULA:

T plus $\frac{7}{16}$ =diameter of rivet for single riveted lap joint

LEGEND:

T =thickness of plate = $\frac{3}{8}$ = .3750

EXAMPLE:

.3750 =thickness of plate
.4375 = $\frac{7}{16}$

.8125 = $\frac{13}{16}$ rivet (this sectional area after rivet has been driven)

The Board of Supervising Inspectors of Steam Vessels, in their rules and regulations governing the construction of steam boilers for marine purposes, prescribe the following rules for single and double riveted lap joints:

$d = T + \frac{3}{8}$ inch for iron plates and iron rivets, single riveted lap joints.
$d = T + \frac{5}{16}$ inch for iron plates and iron rivets, double riveted lap joints.
$d = T + \frac{7}{16}$ inch for steel plates and steel rivets, single riveted lap joints.
$d = T + \frac{3}{8}$ inch for steel plates and steel rivets, double riveted lap joints.

It has been generally considered good practice to have rivet section percentage of strength higher, this for the benefits of caulking and increasing rivet strength and to make up for depreciation due to heating and driving rivet; but one authority on boiler construction says to have plate higher in efficiency to provide for plate deteriorating due to pitting and corrosion; however, in designing seams these conditions can be provided for when computing boiler joints.

Rule to find center of rivet to edge of plate (lap): Multiply diameter of rivet hole by 1.5 (one and a half) constant.

FORMULA:

$d \times C = \text{lap}$

LEGEND:

d = diameter of rivet hole = ¾ = .750
C = constant = 1.5

FORMULA:

.750 = rivet diameter
1.5 = constant
3750
750
1.1250 = 1⅛″ lap

Rule to find percentage of rivet in a single riveted lap joint, steel plate and steel rivets: Multiply area of rivet by number of rows in one pitch and by the constant 100; divide this product by pitch of rivet multiplied by thickness of plate in inches.

LEGEND:

P = pitch of rivets = $1\frac{15}{16}$ = 1.9375
A = area of rivet ¾ hole = .44179
C = constant = 100
T = thickness of plate = ⅜ = .375
N = number of rows of rivets = 1
D = diameter of rivet hole = ¾ = .750

FORMULA:

$$\frac{A \times N \times C}{P \times T} = \text{rivet percentage}$$

EXAMPLE:

pitch of rivet = 1.9375
thickness of plate = .375
96875
135625
58125

.44179 = area of rivet
1 = no. of rows
.44179
100 = constant

.7265625)44.1790000(.60 = 60% rivet percentage
43 593750
5852500

Rule to find percentage of plate in single riveted joint, steel plate and steel rivets, when pitch and diameter of rivet are given: From pitch of rivet subtract sum of diameter of hole, multiply by constant 100 and divide by pitch.

FORMULA:

$$\frac{P—d \times C}{P} = \text{percentage of plate}$$

EXAMPLE:

1.9375 = pitch
.750 = rivet hole diameter
1.1875
100 = constant
pitch = 1.9375) 118.7500 (61% = percentage of plate
116 250
2 5000
1 9375
5625

LLOYDS RULES.

Rule to find working pressure: Multiply constant by thickness of plate and by per cent of joint efficiency; divide this product by diameter of boiler.

CONSTANTS USED.

	Thickness ½	Thickness ½ to ¾	Thickness ¾ & over
For iron plate punched, lap joint	155	165	170
" " " drilled " "	170	180	190
" " " punched double strap	170	180	190
" " " drilled " "	180	190	200

For Steel Plate.	Thickness ⅜ & under	Thickness ⅜ to $\frac{9}{16}$	Thickness $\frac{9}{16}$ to ¾	Thickness ¾ & over
Lap joints punched......... " " drilled	200	215	230	240
" " double strap punched...... " " double strap drilled........	215	230	250	260

LEGEND:
T = thickness of plate
D = diameter
% = joint efficiency

FORMULAS:

$$\frac{C \times T \times \%}{D} = \text{working pressure}$$

$$\frac{P \times D}{C \times \%} = \text{thickness of plate}$$

MANCHESTER STEAM USERS ASSOCIATION FORMULAS:

$$\frac{T \times 2 \times \% \times TS}{D \times 5 \times 100} = \text{working pressure}$$

$$\frac{D \times P \times 5 \times 100}{T \times \% \times 2} = \text{thickness of plate}$$

APPENDIX.

The following formulas are taken from those of the British Board of Trade and are given for the determination of the pitch, distance between rows of rivets, diagonal pitch, maximum pitch and distance from centers of rivets to edge of lap of single and double riveted lap joints, for both iron and steel boilers:

Let p = greatest pitch of rivets in inches.
n = number of rivets in one pitch.
pd = diagonal pitch in inches.
d = diameter of rivets in inches.
T = thickness of plate in inches.
V = distance between rows of rivets in inches.
E = distance from edge of plate to center of rivet in inches.

TO DETERMINE THE PITCH.

Iron plates and rivets:

$$\frac{d^2 \times .7854 \times n}{T} + d = \text{pitch}$$

Example, first, for single-riveted joint: Given, thickness of plate (T) = ½ inch, diameter of rivet (d) = ⅞ inch. In this case n = 1. Required the pitch.

$$\frac{(\frac{7}{8})^2 \times .7854 \times 1}{\frac{1}{2}} + \frac{7}{8} = 2.077 \text{ inches} = \text{pitch}$$

Example for double-riveted joint: Given, $t = \frac{1}{2}$ inch and $d =$ 13/16 inch. In this case $n = 2$.

$$\frac{(\frac{13}{16})^2 \times .7854 \times 2}{\frac{1}{2}} + \frac{13}{16} = 2.886 \text{ inches} = \text{pitch}$$

For *steel* plates and steel rivets:

$$\frac{23 \times d^2 \times .7854 \times n}{28 \times T} + d. = \text{pitch}$$

Example for single-riveted joint: Given, thickness of plate $=$ $\frac{1}{2}$ inch, diameter of rivet $=$ 15/16 inch. In this case $n = 1$.

$$\frac{23 \times (\frac{15}{16})^2 \times .7854 \times 1}{28 \times \frac{1}{2}} + \frac{15}{16} = 2.071 \text{ inches} = \text{pitch}$$

Example for double-riveted joint: Given, thickness of plate $=$ $\frac{1}{2}$ inch, diameter of rivet $= \frac{7}{8}$ inch. $n = 2$.

$$\frac{23 \times (\frac{7}{8})^2 \times .7854 \times 2}{28 \times \frac{1}{2}} + \frac{7}{8} = 2.85 \text{ inches} = \text{pitch}$$

FOR DISTANCE FROM CENTER OF RIVET TO EDGE OF LAP.

$$\frac{3 \times d}{2} = \text{E or lap}$$

Example: Given, diameter of rivet $(d) = \frac{7}{8}$ inch; required the distance from center of rivet to edge of plate.

$$\frac{3 \times \frac{7}{8}}{2} = 1.312 \text{ inches} = \text{E, for single or double riveted lap joint.}$$

FOR DISTANCE BETWEEN ROWS OF RIVETS.

The distance between lines of centers of rows of rivets for double, chain-riveted joints (V) should not be less than twice the diameter of rivet, but it is more desirable that V should not be less than $\frac{4d+1.}{2}$

Example under latter formula: Given, diameter of rivet = ⅞ inch;

$$\frac{(4 \times \frac{7}{8}) + 1}{2} = 2.25 \text{ inches} = V$$

For ordinary, double, zigzag riveted joints:

$$\frac{\sqrt{(11p + 4d)\ (p + 4d)}}{10} = V$$

Example: Given, pitch = 2.85 inches, and diameter of rivet = ⅞ inch:

$$\frac{\sqrt{(11 \times 2.85 + 4 \times \frac{7}{8}) \times (2.85 + 4 \times \frac{7}{8})}}{10} = 1.487 \text{ inches} = V$$

DIAGONAL PITCH.

For double, zigzag riveted lap joint. Iron and steel:

$$\frac{6p + 4d}{10} = pd$$

Example: Given, pitch = 2.85 inches, and d = ⅞ inch;

$$\frac{(6 \times 2.85) + (4 \times \frac{7}{8})}{10} = 2.06 \text{ inches} = pd$$

MAXIMUM PITCHES FOR RIVETED LAP JOINTS.

For single-riveted lap joints:

$$\text{Maximum pitch} = (1.31 \times T) + 1\tfrac{5}{8}$$

For double-riveted lap joints:

$$\text{Maximum pitch} = (2.62 \times T) + 1\tfrac{5}{8}$$

Example: Given, a thickness of plate = ½ inch, required the maximum pitch allowable.

For single-riveted lap joint:

$$\text{Maximum pitch} = (1.31 \times \tfrac{1}{2}) + 1\tfrac{5}{8} = 2.28 \text{ inches}$$

For double-riveted lap joint:

$$\text{Maximum pitch} = (2.62 \times \tfrac{1}{2}) + 1\tfrac{5}{8} = 2.935 \text{ inches}$$

To determine the pitch of rivets from the above formulas, use the diameter and area of the rivet holes. The diameter of the rivets as given in the following tables is the diameter of the driven rivet.

Any riveted joint will be allowed when it is constructed so as to give an equal percentage of strength to that obtained by the use of the formula given.

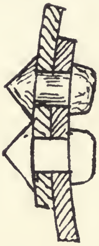
Following are single and double-riveted lap joints tables, taken from the handbook of Thomas W. Traill, entitled Boilers, Marine and Land; Their Construction and Strength, may be taken for use in single and double riveted joints as approximating the formulas of the British Board of Trade for such joints.

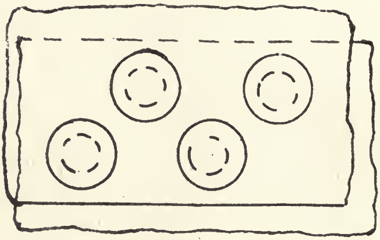
STEEL PLATES AND STEEL RIVETS.

SINGLE-RIVETED LAP JOINTS.

THICKNESS OF PLATE IN INCHES	DIAMETER OF RIVET IN INCHES	PITCH IN INCHES	LAP IN INCHES	EFFICIENCY
1/4	1/2	1 1/4	1	50
1/4	5/8	1 5/8	1 1/8	57
1/4	11/16	1 7/8	1 3/16	60
5/16	5/8	1 3/8	1 1/8	50
5/16	11/16	1 5/8	1 3/16	54
5/16	3/4	1 7/8	1 1/4	56
3/8	3/4	1 11/16	1 1/4	52
3/8	13/16	1 7/8	1 5/16	53
3/8	7/8	2 1/8	1 7/16	55
7/16	3/4	1 9/16	1 1/4	47
7/16	7/8	1 15/16	1 7/16	51
7/16	15/16	2 1/8	1 1/2	53
1/2	7/8	1 13/16	1 7/16	48
1/2	15/16	2	1 1/2	50
1/2	1	2 3/16	1 5/8	51
9/16	15/16	1 7/8	1 1/2	46
9/16	1	2 1/16	1 5/8	48

COMPUTING STRENGTH OF A DOUBLE RIVETED LAP JOINT.





Causes for failure at joint.

1st. Resistance to shearing two rivets.
2nd. Resistance to tearing of plate between two rivets.
3rd. Resistance to crushing in front of two rivets.

Assuming a given boiler of dimensions and data as follows:

LEGEND:

D = diameter of boiler = 54″
T = thickness of plate = $\frac{3}{8}$ = .3750
P = pitch of rivets = 3 $\frac{1}{16}$ = 3.0625
TS = tensile strength = 55000
CS = crushing strength of rivets = 95000
SS = shearing strength of rivets = 38000
A = area of rivet hole = $\frac{15}{16}$ = .69029 or .69
d = diameter of rivet hole = $\frac{15}{16}$ = .9375

First. Resistance to shearing two rivets.

Rule to find shearing strength of rivets in single shear: Multiply area of rivet hole by number of rivets in single shear and this sum by shearing resistance of rivet.

FORMULA:

A × 2 × SS = strength of two rivets in single shear

EXAMPLE:

.69 = area of rivet hole
2 = number of rows
1.38
38000 = shearing strength of rivet in single shear
1104000
414
52440.00
52,440 = strength of two rivets in single shear

Second. Resistance to tearing of plate between two rivets.

Rule to find strength in net section of plate: From pitch of rivet subtract diameter of rivet; multiply this sum by thickness of plate multiplied by tensile strength.

FORMULA:

$(P-d)\times T\times TS$ = strength in net section of plate

EXAMPLE:

3.0625 = pitch
.9375 = diameter of rivet hole
2.1250
20625 = thickness × tensile strength
106250
42500
127500
42500
43828.~~1250~~
43,828 = strength of net section of plate

.3750 = thickness of plate
55000 = tensile strength
18750000
18750
20625.~~0000~~

Third. Resistance to crushing of plate in front of two rivets.

FORMULA:

$d\times 2\times T\times CS$ = resistance to crushing in front of two rivets

EXAMPLE:

.9375 = diameter of rivet hole
2 = two times
1.8750
.375 = thickness of plate
93750
131250
56250
.7031250
95000 = crushing strength
35156250000
63281250
66796.~~8750000~~
66,796 = resistance to crushing in front of two rivets

Rule to find strength of solid plate: Multiply pitch of rivets by the thickness of plate and this sum by tensile strength.

FORMULA:

$P\times T\times TS$ = strength of solid plate

EXAMPLE:

3.0625 = $3\frac{1}{16}$ pitch
.375 = thickness of plate
153125
214375
91875
1.1484375
55000 = tensile strength
5742 1875000
57421 875
63164.~~0625000~~
63,164 = strength of solid plate

Rule to find efficiency from weakest section of joint: Multiply sum of weakest section by 100 and divide by sum of strength of solid plate.

FORMULA:

$$\frac{43{,}828 \times 100}{63{,}163} = \text{efficiency of joint}$$

EXAMPLE:

43,828 = strength of net section of plate
100 = constant
strength of solid plate = 63,164)4382800(69 per cent efficiency of joint
378984
592960
568476
24484

Rule to find safe working pressure from these calculations: Multiply tensile strength of plate by joint's efficiency and multiply this sum by twice the thickness of plate and divide this product by the diameter of boiler in inches multiplied by factor of safety.

FORMULA:

$$\frac{TS \times \% \times (2 \times T)}{D \times F} = \text{working pressure}$$

Example:

```
               55000 =tensile strength
                 .69 =efficiency of joint
              ------
              495000
             330000
             -------
             37950.00
                750 =twice thickness of plate
            --------
diam of boiler=  54"  1897500
factor of safety=  5  265650
              ----  ---------
              270)28462.500(105 lbs. working pressure
                  270
                 ----
                  1462
                  1350
                 -----
                   112
```

When finding diameter of rivet holes for lap and butt joints, the following constants are used:

C=2.25 for lap joints double riveted up to and including ½" plate.

C=1.9 for triple riveted lap joint up to ½" plate.

C=1.8 for butt joints triple and quadruple riveted.

Rule to find diameter of rivet hole: The square root of product of thickness of plate in inches multiplied by constant used in connection with joint form and plate will give diameter of rivet hole.

Legend:

T = thickness of plate = $\frac{7}{16}$ = .4375

C = constant = 2.25

Formula:

$\sqrt{T \times C}$ = diameter of rivet hole

Example:

```
            .4375 =thickness of plate
             2.25 =constant
            -----
            21875
            8750
           8750
           ------
        9).984375(.9921=1" nearly or hole for 15/16
           81                                 rivet
         ) ---
     189)  1743
         ) 1701
         ) ----
    1982)   4275
         )  3964
         ) -----
   19841)   31100
         )  19841
         ) ------
         )  11259
```

Rule to find diameter of shell: Multiply tensile strength by thickness of plate in inches and by per cent of joint; divide this product by pressure multiplied by the factor.

FORMULA:

$$\frac{TS \times T \times \% \text{ of joint}}{P \times F} \times 2 = \text{diameter of shell}$$

LEGEND:

P = pressure = 130
T = thickness of plate = ½ = .5
F = factor = 6
% = percentage of joint = 80
TS = tensile strength = 60000

EXAMPLE:

60000 tensile strength
.5 = thickness of plate
pressure = 130 30000.0
constant = 6 80 = joint efficiency
780)24000.00(30 = radius
2340 2
600 60″ diameter

Rule to find tensile strength of plate for boiler: Multiply given pressure per square inch by tensile strength; multiply this by one-half diameter of boiler; divide by the given thickness of material in inches, and the quotient will give the required tensile strength per square inch in pounds.

FORMULA:

$$\frac{(P \times TS) \times (\tfrac{1}{2} \text{ of } D)}{T} = \text{tensile strength}$$

LEGEND:

TS = tensile strength = 60000
P = pressure = 125 lbs.
D = diameter of boiler = 60″
T = thickness of plate = .3750

EXAMPLE:

125 = pressure
60000 = tensile strength
7500000
30 = ½ the diameter
thickness of plate = .3750)225000000(60000 lbs. = tensile strength
22500
00000

Rule to find thickness of shell plate when percentage of joint is known: Multiply diameter of shell by pressure and again by factor of safety and multiply this sum by 100; divide product by tensile strength multiplied by efficiency of seam multiplied by 2.

LEGEND:

D = diameter = 60″
P = pressure = 150
F = factor of safety = 5
% = percentage of seam strength = 80
TS = tensile strength = 60000
C = constant = 100

FORMULA:

$$\frac{D\times P\times F\times 100}{TS\times \%\times 2} = \text{thickness of shell plate}$$

EXAMPLE:

60″ = diameter of shell
150 = pressure
3000
60
tensile strength = 60000 9000
percentage = 80 5 = factor
4800000 45000
two times = 2 100 = constant
9600000)4500000.0000(.4687 = 15/32″ = thickness required
3840000 0
660000 00
576000 00
84000 000
76800 000
7200 0000
6720 0000
480 0000

Rule to find diameter of steel rivet for steel plate double riveted lap joint: Add ⅜ of an inch to plate thickness.

FORMULA:

⅜ plus T = diameter of rivet

T = thickness of plate = $\frac{7}{16}$ = .4375

EXAMPLE:

.4375 = plate
.375 = ⅜
.8125 = $\frac{13}{16}$ rivet diameter

Rule to find pitch of rivet in a double riveted lap joint — steel plate, steel rivets: Multiply square of diameter of rivet hole by constant 23, this sum by .7854; then multiply this product by the number of rows of rivets; divide by diameter of rivet multiplied by constant 28, and add diameter of rivet hole to quotient. Result gives pitch of rivet.

FORMULA:

$$\frac{d^2\times 23\times .7854\times N}{d\times 28}+d=\text{pitch}$$

d = diameter of rivet hole = $\frac{15}{16}$ = .9375
N = number of rows = 2

EXAMPLE:

.9375 = diameter rivet hole
.9375
46875
65625
28125
84375
.87890625 = diameter of rivet hole [squared
23 = constant
263671875
175781250
20.2148~~4375~~
.7854
808592
1010740
1617184
1415036
15.87670392
2 rows of rivets

diam. of rivet hole = .9375
constant = 28
75000
19750
26.2500

26.2500)31.75340784(1.2096
262500
550340
525000
2534078
2362500
1715784
1575000
140784

1.2096
.9375 = diameter of rivet hole
2.1471 = 2 $\frac{9}{64}$ nearly = pitch

Rule to find distance between rows of chain double riveted joint: To four times the diameter of one rivet hole add one and divide by two.

FORMULA:

$$\frac{4d \text{ plus } 1}{2} = \text{distance between rows chain riveted joint}$$

LEGEND:

d = diameter of rivet hole = $\frac{7}{8}$ = .8750

EXAMPLE:

$$\begin{array}{rl}
.8750 & = \text{diameter of rivet hole} \\
4 & \\
\hline
3.5000 & \\
1.0000 & \text{added} \\
\hline
2)4.5000 & \\
\hline
2.2500 & = 2\tfrac{1}{4}'' \text{ distance between rows}
\end{array}$$

Rule to find diagonal pitch of rivet: To four times the diameter of rivet hole add six times the pitch on straight line and divide by 10.

FORMULA:

$$\frac{4d + 6P}{10} = \text{diagonal pitch}$$

LEGEND:

d = diameter of rivet hole = $\frac{7}{8}$ = .8750
p = pitch = $3\frac{3}{8}$ = 3.3750

EXAMPLE:

$$\begin{array}{rl}
3.3750 & = \text{pitch} \\
6 & \text{times} \\
\hline
20.2500 &
\end{array}$$

$$\begin{array}{rl}
.8750 & = \text{diameter of rivet hole} \\
4 & \text{times} \\
\hline
3.5000 & = 4 \text{ times diameter} \\
20.2500 & = 6 \text{ times pitch} \\
\hline
10)23.7500 & (2.3750 = 2\tfrac{3}{8} \text{ diagonal pitch of rivets} \\
20 & \\
\hline
37 & \\
30 & \\
\hline
75 & \\
70 & \\
\hline
50 & \\
50 & \\
\hline
0 &
\end{array}$$

Rule to find spacings center of rivet to edge of plate. Multiply diameter of rivet by 3 and divide by 2.

FORMULA:

$$\frac{3 \times d}{2} = \text{distance from center of rivet to edge of plate}$$

d = diam. of rivet $\frac{7}{8}$ = .8750

EXAMPLE:

.8750
3

2)2.6250

1.3125 = $1\frac{5}{16}$ inch distance

Rule to find pitch of rivet to give best percentage of strength in a double zig zag riveted joint: Multiply twice the rivet sectional area by the shearing strength of rivet and divide by thickness of plate multiplied by its tensile strength; add to product one diameter of rivet.

FORMULA:

$$\frac{(2 \times A) \times SS}{T \times TS} \text{ plus 1 diam. of rivet} = \text{pitch}$$

LEGEND:

A = rivet area = $\frac{13}{16}$ = .5185
SS = shearing strength one rivet = 38000
T = plate thickness = $\frac{3}{8}$ = .3750
TS = tensile strength of plate = 60000
d = diameter of rivet = $\frac{13}{16}$ = .8125

EXAMPLE:

.5185 = sectional area of rivet
2

1.0370 = twice sectional area of rivet
38000 = shearing strength of one rivet

82960000
31110

$\frac{3}{8}$ plate = .3750
tensile strength = 60000

22500.0000)39406.0000(1.7513
22500

169060
157500

115600
112500

31000
22500

85000
67500

17500

1.7513
.8125 = diam. of one rivet

2.5638 = $2\frac{9}{16}$ inch pitch

Rule to find plate percentage in a double riveted lap joint: From pitch of rivet subtract diameter of rivet and multiply by constant 100; divide this product by pitch of rivet.

LEGEND·

P = pitch = 3⅛ = 3.125
d = diameter of rivet hole = ⅞ = .8750
C = constant = 100

FORMULA:

$$\frac{(P—d)\times 100}{P} = \text{percentage of plate}$$

EXAMPLE:

```
          3.1250 = pitch of rivet
           .8750 = diameter of rivet hole
          ------
          2.2500
               100 = constant
           -------
3.1250)225.0000(72 = percentage of plate
       218750
       ------
         62500
         62500
         -----
```

Rule to find percentage of rivet in a double riveted lap joint: Multiply area of rivet by the number of rows of rivet in one pitch; multiply this product by 100 and by the constant 23; divide this product by pitch multiplied by thickness of plate and constant 28.

LEGEND:

T = thickness of plate = $\frac{7}{16}$ = .4375
P = pitch = 3⅛ = 3.125
A = area of rivet hole = ⅞ = .6013
d = diameter of rivet = ⅞ = .8750
N = number of rows = 2

```
             pitch = 3.125
thickness of plate =  .4375
                     ------
                      15625
                     21875
                     9375
                   1 2500
                   ---------
                   1.3671875
         constant =       28
                   ---------
                  10 9375000
                  27 343750
                  ----------
                  38.2812500
```

FORMULA:

$$\frac{A\times N\times 100\times 23}{P\times T\times 28} = \text{per cent. of rivet section}$$

EXAMPLE:

```
           .6013 = area of rivet hole
             2 rows
          ------
          1.2026
               100 = constant
           --------
          120.2600
                23 = constant
          --------
           3607800
          2405200
          --------
38.281)2765.9800(72.2 = % of rivet strength
       2679 67
       -------
         86 310
         76 562
         ------
          9 7480
          7 6562
          ------
          2 0918
```

Rule to find bursting pressure of boiler: Multiply tensile strength by twice the thickness of plate and divide by the internal diameter of boiler.

FORMULA:

$$\frac{TS \times (2 \times T)}{D} = \text{bursting pressure}$$

LEGEND:

TS = tensile strength = 60000
T = thickness of plate = $\frac{3}{8}$ = .375
D = internal diameter = 60″

EXAMPLE:

thickness of plate = .375
2
twice thickness = .750

60000 = tensile strength
.750 = twice thickness of plate
3000000
420000

internal diameter = 60″) $45000.\cancel{0}\cancel{0}\cancel{0}$ (750 lbs. per square inch bursting pressure
420
300
300
0

The bursting pressure divided by the factor of safety will give the safe working pressure. The factor of safety of 5 has been generally accepted by eminent engineers and boilermakers.

factor = 5) 750 per sq. inch bursting pressure
150 lbs. working pressure

Rule to find working pressure on boilers from a lowest percentage of joint: Multiply tensile strength of material by the lowest percentage of joint, then by twice the thickness of plate and divide by diameter multiplied by factor of safety.

FORMULA:

$$\frac{TS \times \% \times (2 \times T)}{D \times F} = \text{working pressure}$$

LEGEND:

TS = tensile strength = 60000
% = lowest percentage of joint = 80
T = thickness of plate = $\frac{1}{2}$ = .500
D = internal diameter = 71.1250 (outside = 72″)

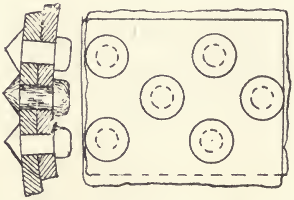
EXAMPLE:

```
                                60000   =tensile strength
                                   80   =percentage of joint
         Internal               ─────
diameter of boiler=71.1250    4800000
   factor of safety =     6           1.0000 =twice thickness of plate
                  ─────────   ─────────
                  426.7500)4800000.0000(112 lbs. working pressure
                           4267500
                           ────────
                            5325000
                            4267500
                            ────────
                             10575000
                              8535000
                              ───────
                              2040000
```

Rule to find safe working pressure according to the U. S. Government rule is as follows: Multiply one sixth of the lowest tensile strength found stamped on any plate by the thickness of same, expressed in inches or decimal parts of same, and divide by the radius or half of diameter expressed in inches. The result will give pressure allowed for a single riveted boiler; when double riveted add 20 per cent. This rule is based on the rivet and plate section being equal and holes drilled.

Thickness of plate	Diameter of rivet	Pitch in inches	Lap in inches	Distance between rows	Efficiency
1/4	1/2	1 13/16	1	1 3/4	69
1/4	5/8	2 5/8	1 1/8	1 7/8	72
1/4	11/16	2 7/8	1 3/16	1 15/16	74
5/16	5/8	2 3/16	1 1/8	1 5/8	68
5/16	11/16	2 1/2	1 3/16	1 3/4	70
5/16	3/4	2 7/8	1 1/4	1 15/16	72
3/8	3/4	2 9/16	1 1/4	1 7/8	68
3/8	13/16	2 7/8	1 5/16	2	69
3/8	7/8	3 1/4	1 7/16	2 3/16	71
7/16	3/4	2 5/16	1 1/4	1 7/8	65
7/16	7/8	2 15/16	1 7/16	2 1/16	67
7/16	15/16	3 5/16	1 1/2	2 3/16	70
1/2	7/8	2 11/16	1 7/16	2	65
1/2	15/16	3	1 1/2	2 1/8	66
1/2	1	3 5/16	1 5/8	2 1/4	68
9/16	15/16	2 3/4	1 1/2	2	63
9/16	1	3 1/16	1 5/8	2 1/8	65

COMPUTING STRENGTH OF TRIPLE RIVETED LAP JOINTS.



Causes for failure at joint.

1st. Resistance to shearing three rivets.
2nd. Resistance to tearing between three rivets.
3rd. Resistance to crushing in front of three rivets.

Assuming a boiler of dimensions and data as follows:

Legend:

T = thickness of plate = $\frac{3}{8}$ = .375
TS = tensile strength = 55000
d = diameter of rivet = 13/16 = .8125
A = area of rivet hole = 13/16 = .5185
P = pitch of rivet = 3¼ = 3.2500
SR = shearing resistance of rivets = 38000
CS = crushing strength of rivet and plate = 95000
D = diameter of boiler = 60″
F = factor of safety = 5

First. Resistance to shearing of three rivets.

Rule to find strength of rivets in single shear: Multiply area of rivet hole by number of rivets, and multiply this sum by the shearing resistance of rivet material.

Formula:

A × No. of rivets × SR = strength of rivets in single shear

Example:

.5185 = area of rivet hole
3 = number of rivets

1.5555
38000 = shearing resistance of rivets

124440000
46665

59109.0000
59,109 lbs. = strength of three rivets in single shear

Second. Resistance to tearing of plate between three rivets.

Rule to find strength of net section of plate: From pitch of rivets subtract diameter of rivet hole and multiply by thickness of plate and multiply this sum by the tensile strength of plate.

FORMULA:

(P—d) × T × TS = strength of net section of plate

EXAMPLE:

3.2500 = pitch of rivet
.8125 = diameter of rivet hole
2.4375
.375 = thickness of plate
121875
170625
73125
.9140625
55000 = tensile strength
45703125000
45703125
50273.4375000
50,273 = strength of net section of plate

Third: Resistance to crushing in front of plate in front of three rivets.

FORMULA:

d × 3 × T × CS = resistance to crushing in front of three rivets

EXAMPLE:

.8125 = diameter of rivet
3 = three rivets
2.4375
.375 = thickness of plate
121875
170625
73125
.9140625
95000 = crushing strength of rivet and plate
4570 3125000
82265 625
86835.9375000
86,835 lbs. = resistance to crushing of material

Rule to find strength of solid plate: Multiply pitch of rivets by thickness of plate and this sum by tensile strength of material.

FORMULA:

P×T×TS = strength of solid plate

EXAMPLE:

```
     3.2500 =pitch
       .375 =thickness of solid plate
     ------
     162500
    227500
    97500
  ---------
  1.2187500
        55000 =tensile strength
  -----------
  6093 7500000
 60937 500000
 ------------
 67031.2500000
67,031 lbs. =strength of solid plate
```

Rule to find efficiency of this joint: Divide net section of plate by strength of solid plate.

EXAMPLE:

```
50,273 =net section of plate
67,031 =strength of solid plate

     67031)50273.000(.749 =efficiency
           46921 7
           -------
            3351 30
            2681 24
            -------
             670 060
             603 279
             -------
              66 781
```

Rule to find safe working pressure from these calculations: Multiply tensile strength of plate by efficiency of joint and multiply this sum by twice thickness of plate; divide this product by diameter of boiler in inches multiplied by factor of safety.

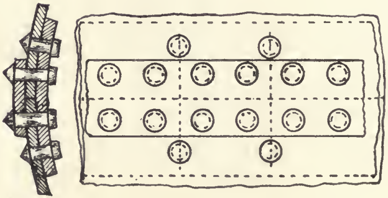
Example:

```
                                  55000 =tensile strength of plate
                                   .749 =percentage of joint
                              -----------
                                495 000
                               2200 00
                              38500 0
                              -----------
                              41195.~~000~~
                                .7500   =twice thickness of plate
                              -----------
diam. of boiler =60"       2059 7500
factor of safety =  5     28836 5
                  ----   ----------
                   300)30896.2~~500~~(102.9 lbs. working pressure
                       300
                       ---
                       896
                       600
                       ----
                       2962
                       2700
                       ----
                        262
```

Thickness of plate	Diameter of rivet	Pitch in in inches	Lap in in inches	Distance between rows	Efficiency
1/4	1/2	2 1/2	1	1 7/8	76
1/4	5/8	3 1/2	1 1/8	2 1/16	80
1/4	11/16	4	1 3/16	2 1/8	81
5/16	5/8	2 15/16	1 1/8	2	76
5/16	11/16	3 1/8	1 3/16	2 1/6	76
5/16	3/4	4	1 1/4	2 3/16	79
3/8	3/4	3 3/8	1 1/4	2 3/16	76
3/8	13/16	3 7/8	1 5/16	2 1/4	77
3/8	7/8	4 7/16	1 7/16	2 1/2	79
7/16	3/4	3	1 1/4	2	73
7/16	7/8	3 15/16	1 7/16	2 1/2	76
7/16	15/16	4 3/8	1 1/2	2 5/8	77
1/2	7/8	3 1/2	1 7/16	2 1/4	73
1/2	15/16	4	1 1/2	2 5/8	74
1/2	1	4 7/16	1 5/8	2 11/16	76
9/16	15/16	3 5/8	1 1/2	2 5/16	72
9/16	1	4 1/16	1 5/8	2 1/2	73

CHAPTER VII.

BUTT JOINT DOUBLE STRAPPED AND DOUBLE RIVETED.



Where butt straps are used in the construction of marine boilers, the straps for single butt strapping shall in no case be less than the thickness of the shell plates; and where double butt straps are used, the thickness of each shall in no case be less than five-eighths (⅝) the thickness of the shell plates.

A rule to find thickness of butt straps is as follows: Multiply the thickness of shell plate by factor 5 and this sum by the wide pitch of rivets in inches minus diameter of one rivet; divide this product by the wide pitch minus two times diameter of rivet multiplied by constant 8.

FORMULA:

$$\frac{T\times F\times (WP-d)}{WP-(2\times d)\times C}=\text{thickness of each butt strap}$$

LEGEND:

T = thickness of plate = $\frac{7}{16}$ = .4375
d = diameter of rivet = ⅞ = .8750
WP = wide pitch = 6¾ = 6.7500
F = factor = 5
C = constant = 8

EXAMPLE:

```
                                  .4375 =thickness of plate
                                      5 =factor
                                 ------
                                 2.1875 =5 times thickness
                                 5.8750       6.7500 =wide pitch
       wide pitch =  6.7500      ------        .8750 =rivet diameter
twice rivet diam. =  1.7500     1093750       ------
                     ------     153125        5.8750
                     5.0000    175000
       constant =         8   109375
                     ------   ---------
                    40.0000)12.85150000(.3212 =thickness of butt strap
                            12 0                   =11/32 approximately
                            ----
                              85
                              80
                              --
                               51
                               40
                               --
                               115
                                80
                               ---
                                35
```

When joints have one strap, butt or lap, the rivets are in single shear only. In triple riveted joints, double strap, the two inner rows are in double shear and the outer in single shear.

Rule to find strength of a solid strip of plate or resistance to a tensile strength: Multiply width of strip by thickness of plate and this product by the tensile strength of material.

FORMULA:

W×T×TS =strength of solid plate

LEGEND:

W =width of strip =6.3750
T =thickness of plate = .4375
TS =tensile strength =60000

EXAMPLE:

```
  6.3750 =width of strip
   .4375 =thickness of plate
  ------
  318750
 446250
191250
2 55000
----------
2.78906250
        60000 =tensile strength
-------------
 167343.7500000
```

167,343 lbs. =strength of solid plate

BUTT JOINT, DOUBLE STRAP AND DOUBLE RIVETED.

Possible causes for failure.

First. Resistance to tearing of plate at outer row of rivets.
Second. Resistance to shearing of two rivets in double shear and one in single shear.
Third. Resistance to tearing of plate at inner row of rivets and shearing one of the outer row single shear.
Fourth. Resistance to crushing in front of three rivets.
Fifth. Crushing in front of two rivets and shearing one rivet.

LEGEND:

T = thickness of plate = $\frac{7}{16}$ = .4375
dh = diameter of rivet hole = $\frac{13}{16}$ = .8125
D = diameter of boiler = 60″
p = pitch of rivets = $4\frac{3}{8}$ = 4.3750
TS = tensile strength = 60000
A = area of rivet hole = $\frac{13}{16}$ = .5185
SS = shearing strength of rivet, single shear = 38000
DS = " " " " double " = 70300
N = number of rows of rivets = 2
CS = crushing strength of material = 95000
F = factor of safety = 5

First. Resistance to tearing at outer row of rivets.

FORMULA:

(p—dh) × T × TS = net section of plate

EXAMPLE:

4.3750 = pitch of rivet
.8125 = diameter of rivet hole

3.5625
.4375 = thickness of plate

178125
249375
106875
1 42500

1.55859375
60000 = tensile strength

93515.~~62500000~~
93,515 lbs. = strength of net section of plate.

Second. The resistance to shearing two rivets in double shear and one in single shear.

FORMULA:

A × N × DS + (A × SS) = total shearing strength of rivets

Example:

		.5185 =area of rivet hole
		2 =number of rows of rivets
		1.0370
		70300 =shearing strength double shear
area of rivet =	.5185	3111000
single shearing strength =	38000	72590
	4148 0000	72901.~~1000~~
	15555	19703. =area multiplied by SS
	19703.~~0000~~	92604 lbs. =total shearing strength of rivets

Third. The resistance to tearing at inner row of rivets and shearing of one rivet.

Formula:

$$(p—2dh) \times T \times TS + (A \times SS) = \text{resistance to tearing at inner row}$$

Example:

4.3750 =pitch of rivets
1.6250 =two diameters of rivet hole

2.7500
.4375 =thickness of plate

137500
192500
82500
1 10000

1.20312500
60000 =tensile strength

72187.~~50000000~~
19703 =area multiplied by SS

91890 lbs. =resistance to tearing at inner row of rivets

Fourth. The resistance to crushing in front of three rivets.

Formula:

$$dh \times 3 \times T \times CS = \text{resistance to crushing}$$

EXAMPLE:

.8125 = diameter of rivet
3 = three rivets

2.4375
.4375 = thickness of plate

121875
170625
73125
97500

1.06640625
95000 = crushing strength

5332 03125000
95976 5625

101308.~~59375000~~ lbs. = resistance to crushing strength in front of three rivets

Fifth. The resistance to crush in front of two rivets and shearing of one rivet

FORMULA:

2×T×CS + (A×SS) = resistance to crushing plate and shearing one rivet

EXAMPLE:

.4375 = thickness of plate
2 = two rivets

.8750 = twice thickness of plate
95000 = crushing strength

43750000
78750

83125.~~0000~~
19703 = area multiplied by SS

102828 lbs. = resistance to crushing plate and shearing one rivet

Strength of solid plate.

FORMULA:

p×T×TS = strength of solid plate

EXAMPLE:

4.3750 = pitch
.4375 = thickness of plate

218750
306250
131250
1 75000

1.91406250
60000 = tensile strength

114843.~~75000000~~ lbs. = strength of solid plate

To find efficiency of joint from these computations: Divide weakest section of plate by strength of solid plate.

EXAMPLE:

Weakest section of plate = 91890
Strength of solid plate = 114843

114843)91890.00(.80 = efficiency of joint
91874 4
15 60

Rule to find safe working pressure from joint efficiency: Multiply tensile strength of plate by joint efficiency and multiply that product by twice the thickness of plate; divide by diameter of boiler multiplied by factor of safety.

FORMULA:

$$\frac{TS \times \% \times (2 \times T)}{D \times F} = \text{safe working pressure}$$

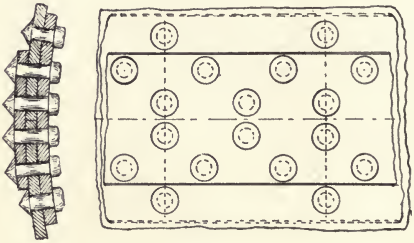
EXAMPLE:

60000 = tensile strength
.80 = efficiency of joint
48000.00
.8750
2400000
diameter of boiler = 60″ 3360000
factor = 5 3840000
300)42000.0000(140 lbs. = working pressure
300
1200
1200
0

DOUBLE RIVETED BUTT JOINTS.

Thickness of Plate.	Diameter of Rivets.	Pitch of Rivets in Inches.	Width of Outside Butt Strap.	Width of Inside Butt Strap.	Thickness of Covering Straps.	Vertical or Transverse Pitch.	Edge of Butt Strap to Center of Rivets.	Pitch of Rivets, Girth Seam.	Edge of Plate to Center of Rivets, Girth Seam.	Strength of Joint. Per Cent.
$\frac{5}{16}$ in	$\frac{11}{16}$ in	$2\frac{1}{4}$x$4\frac{1}{2}$	$4\frac{1}{2}$ in	9 in	$\frac{1}{4}$ in	$2\frac{1}{4}$ in	$1\frac{1}{8}$ in	$2\frac{1}{8}$ in	$1\frac{1}{8}$ in	83
$\frac{3}{8}$ "	$\frac{3}{4}$ "	$2\frac{3}{8}$x$4\frac{3}{4}$	$4\frac{7}{8}$ "	$9\frac{7}{8}$ "	$\frac{5}{16}$ "	$2\frac{7}{16}$ "	$1\frac{1}{4}$ "	$2\frac{1}{8}$ "	$1\frac{7}{32}$ "	82.9
$\frac{7}{16}$ "	$\frac{13}{16}$ "	$2\frac{15}{32}$x$4\frac{15}{16}$	$5\frac{1}{4}$ "	$10\frac{1}{2}$ "	$\frac{3}{8}$ "	$2\frac{5}{8}$ "	$1\frac{15}{16}$ "	$2\frac{1}{4}$ "	$1\frac{7}{32}$ "	82
$\frac{1}{2}$ "	$\frac{7}{8}$ "	$2\frac{9}{16}$x$5\frac{1}{8}$	$5\frac{5}{8}$ "	$11\frac{1}{4}$ "	$\frac{7}{16}$ "	$2\frac{13}{16}$ "	$1\frac{13}{32}$ "	$2\frac{1}{4}$ "	$1\frac{13}{32}$ "	80

BUTT JOINT DOUBLE STRAPPED TRIPLE RIVETED.



Rule to find diagonal pitch of rivets for a butt joint double strap and triple riveted:

To the horizontal pitch multiplied by 6 add diameter of rivet multiplied by 4 and divide result by 10.

FORMULA:

$$\frac{(Hp \times C6) + (d \times C4)}{10} = \text{diagonal pitch}$$

LEGEND:

Hp = horizontal pitch = 3.3750
d = diameter of rivet = .8750

EXAMPLE:

$$\begin{array}{rr} \text{horizontal pitch} = & 3.3750 \\ & 6 \\ \hline & 20.2500 \\ & 3.5000 \\ \hline \end{array} \qquad \begin{array}{rr} \text{diameter of rivet} = & .8750 \\ & \cdot\ 4 \\ \hline & 3.5000 \end{array}$$

$$\begin{array}{l} 10)\,23.7500\,(2.3750 = \text{diagonal pitch} \\ \quad\ \ 20 \\ \quad\ \ \overline{\ \ 37} \\ \quad\ \ \ \ 30 \\ \quad\ \ \ \ \overline{\ \ 75} \\ \quad\ \ \ \ \ \ 70 \\ \quad\ \ \ \ \ \ \overline{\ \ 50} \\ \quad\ \ \ \ \ \ \ \ 50 \\ \quad\ \ \ \ \ \ \ \ \overline{\ \ \ 0} \end{array}$$

Rule to find distance between inner rows of rivets in a butt joint, double or triple riveted chain or zig zag form. Multiply 11 times the pitch plus 8 times the rivet diameter by the pitch, plus 8 times the rivet diameter; extract square root of this product and divide the sum by 10.

FORMULA:

$$\frac{\sqrt{(11\times p+8\times d)\times(p+8\times d)}}{10}=\text{distance between rows of rivets}$$

LEGEND:

p = narrow pitch = $3\frac{3}{8}$ = 3.375
d = diameter of rivet = .875

EXAMPLE:

3.375 = narrow pitch
11 = 11 times
37.125
7.000
44.125
10.375
220625
3 08875
13 2375
441 25
2)458.796875(21.419
)4
41) 58
) 41
424) 1779
) 1696
4281) 8368
) 4281
42829) 408775
) 385461
) 23314

.875 = rivet diam.
8
7.000 = 8 times rivet diam.

3.375 = narrow pitch
7.000 = 8 times diam. rivet
10.375

10)21.419
2.1419 = $2\frac{1}{8}$ approximate distance

Rule to find pitch of rivets in a butt joint double strap and triple riveted inner row: Multiply thickness of plate by 3.5 and add $1\frac{5}{8}$ of an inch to product.

LEGEND:

T = thickness of plate = $\frac{7}{16}$ = .4375
p = pitch 3.5 = 3.5000
$1\frac{5}{8}$ = 1.6250

FORMULA:

$T\times3.5+1\frac{5}{8}=\text{pitch}$

EXAMPLE:

$$\begin{array}{r l}
.4375 & =\text{thickness of plate} \\
3.5 & \\ \hline
21875 & \\
13125 & \\ \hline
1.53125 & \\
1.6250 & =1\frac{5}{8} \\ \hline
3.15625 & =3\frac{5}{32}\text{ pitch}
\end{array}$$

Rule to find plate percentage at wide pitch: From wide pitch subtract diameter of rivet and divide this product by wide pitch of rivet.

FORMULA:

$$\frac{WP - d}{WP} = \text{plate percentage}$$

LEGEND:

WP = wide pitch = 6.7500
d = rivet diameter = $\frac{15}{16}$ = .9375

EXAMPLE:

$$\begin{array}{r r l}
 & 6.7500 & =\text{pitch of rivet} \\
 & .9375 & =\text{diameter of rivet} \\ \cline{2-2}
\text{wide pitch}=6.7500) & 5.812500 & (.86=\text{plate percentage at wide pitch} \\
 & 5\ 40000 & \\ \cline{2-2}
 & 412500 & \\
 & 405000 & \\ \cline{2-2}
 & 7500 &
\end{array}$$

Rule to find percentage of plate at narrow pitch: From narrow pitch subtract rivet diameter and divide this product by narrow pitch.

FORMULA:

$$\frac{NP - d}{NP} = \text{plate percentage}$$

LEGEND:

NP = narrow pitch = 3.5000
d = rivet diameter = $\frac{15}{16}$ = .9375

EXAMPLE:

3.5000 = narrow pitch
.9375 = rivet diameter

narrow pitch = 3.5000)2.562500(.73 = plate percentage at narrow pitch
2 45000

112500
105000

7500

Rule to find safe working pressure on a boiler butt joint double strap, triple riveted: Multiply tensile strength of material by the lowest percentage of joint and this sum by twice the thickness of plate; divide by diameter of boiler multiplied by factor of saftey.

FORMULA:

$$\frac{TS \times \% \times (T \times 2)}{D \times F} = \text{safe working pressure}$$

LEGEND:

TS = tensile strength = 60000
% = lowest percentage of joint = 73%
T = thickness of plate = $\frac{7}{16}$ = .4375
D = diameter of boiler = 72″
F = factor of safety = 5

EXAMPLE:

60000 = tensile strength
.73 = lowest percentage of joint

180000
420000

43800.00
.8750 = twice thickness of plate

219000000
boiler diam. = 72 30660000
factor = 5 35040000

360)38325.~~000000~~(106 lbs. working pressure
360

2325
2160

165

TRIPLE RIVETED BUTT JOINTS.

Plate thickness.	Diameter of rivet.	Thickness of strap.	Width of outer strap.	Width of inner strap.	Pitch of inner row of rivets.	Pitch of outer row of riyets.	Distance between middle and outer row of rivets.	Distance between inner and middle row.	Distance inner row to edge of butt.	Efficiency.
$\frac{1}{4}$	$\frac{1}{2}$	$\frac{1}{4}$	$6\frac{1}{2}$	$11\frac{3}{8}$	$2\frac{1}{4}$	$4\frac{1}{2}$	$2\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{5}{8}$	87
$\frac{9}{32}$	$\frac{9}{16}$	$\frac{1}{4}$	$6\frac{3}{4}$	$12\frac{3}{8}$	$2\frac{9}{32}$	$4\frac{9}{16}$	$2\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{3}{4}$	86
$\frac{5}{16}$	$\frac{11}{16}$	$\frac{1}{4}$	$9\frac{1}{4}$	14	$3\frac{1}{8}$	$6\frac{1}{4}$	$2\frac{3}{8}$	$2\frac{1}{8}$	$2\frac{1}{2}$	88
$\frac{11}{32}$	$\frac{11}{16}$	$\frac{1}{4}$	$9\frac{1}{4}$	14	$3\frac{1}{8}$	$6\frac{1}{4}$	$2\frac{3}{8}$	$2\frac{1}{8}$	$2\frac{1}{2}$	88
$\frac{3}{8}$	$\frac{3}{4}$	$\frac{15}{16}$	$9\frac{3}{4}$	$14\frac{1}{4}$	$3\frac{1}{4}$	$6\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{3}{16}$	$2\frac{7}{16}$	87
$\frac{13}{32}$	$\frac{3}{4}$	$\frac{15}{16}$	$9\frac{3}{4}$	$14\frac{1}{4}$	$3\frac{1}{4}$	$6\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{3}{16}$	$2\frac{7}{16}$	87
$\frac{7}{16}$	$\frac{7}{8}$	$\frac{3}{8}$	$10\frac{1}{8}$	$15\frac{5}{8}$	$3\frac{3}{8}$	$6\frac{3}{4}$	$2\frac{3}{4}$	$2\frac{1}{4}$	$2\frac{13}{16}$	86
$\frac{15}{32}$	$\frac{7}{8}$	$\frac{13}{32}$	$10\frac{3}{8}$	16	$3\frac{1}{2}$	7	$2\frac{13}{16}$	$2\frac{3}{8}$	$2\frac{13}{16}$	86
$\frac{1}{2}$	$\frac{15}{16}$	$\frac{7}{16}$	11	$16\frac{3}{4}$	$3\frac{3}{4}$	$7\frac{1}{2}$	3	$2\frac{3}{8}$	3	86
$\frac{9}{16}$	1	$\frac{7}{16}$	$11\frac{5}{8}$	18	$3\frac{7}{8}$	$7\frac{3}{4}$	$3\frac{3}{16}$	$2\frac{5}{8}$	$3\frac{3}{16}$	85
$\frac{5}{8}$	1	$\frac{1}{2}$	$11\frac{5}{8}$	18	$3\frac{7}{8}$	$7\frac{3}{4}$	$3\frac{3}{16}$	$2\frac{5}{8}$	$3\frac{3}{16}$	84

COMPUTING STRENGTH OF A BUTT JOINT DOUBLE STRAP AND TRIPLE RIVETED.

There are five causes for failure at a butt joint double strap and triple riveted, as follows:

First. By tearing at outer row of rivets.

Second. By shearing of four rivets in double shear and one in single shear.

Third. By the tearing at middle row of rivets and the shearing of one rivet.

Fourth. By the crushing in front of four rivets and shearing of one rivet.

Fifth. By the crushing in front of five rivets, four through strap, the fifth through inner covering of plate only.

LEGEND:

D = diameter of boiler = 72″
ID = internal diameter of boiler = 71.1250
F = factor = 5
TS = tensile strength = 60000
P = pressure
Pt = pitch inner row = $3\frac{3}{8}$ = 3.375
Po = pitch outer row = $6\frac{3}{4}$ = 6.750
SS = shearing strength of rivets = 38000
CS = crushing resistance = 95000
T = thickness of plate = $\frac{7}{16}$ = .4375
d = diameter of rivet = $\frac{7}{8}$ = .8750
DH = diameter of rivet hole = $\frac{15}{16}$ = .9375
A = area of rivet = .6903
CP = cover plate or thickness of strap = .3750

First. The failure by tearing at the outer row of rivets, the resistance is found by the following rule: From pitch of rivet subtract the diameter of rivet and multiply by thickness of plate and then multiply by tensile strength of material.

FORMULA:

(Po—DH) × T × TS = net section of plate

EXAMPLE:

6.7500 = wide pitch
.9375 = diameter of rivet hole
5.8125
.4375 = thickness of plate
290625
406875
174375
2 32500
2.54296875
60000 = tensile strength of plate
152578.~~12500000~~ lbs. = net section of plate

Second. Shearing of four rivets in double shear and one in single shear.

FORMULA:

A × N × DS + 1d of SS = strength of rivets
N = number of rivets = 4
for double shear

EXAMPLE:

.6903 = area of $\frac{15}{16}$ rivet
4 = number of rivets. double shear
2.7612
70300 = strength of rivets double shear
8283600
193284
194112.~~3600~~
26231. = single shearing strength one rivet
220343. lbs. = strength of rivets

area of rivet = .6903
single shearing resistance = 38000
55224000
20709
26231.~~4000~~

Third. Tearing at middle row of rivets and shearing of one rivet, the resistance is:

FORMULA:

(Po—2DH) × T × TS plus (A × SS) = resistance to tearing of plate at middle row and shearing one rivet

EXAMPLE:

6.7500 = wide pitch
1.8750 = 2 diameters of rivet hole
4.8750
.4375 = thickness of plate
243750
341250
146250
1 95000
2.13281250
60000 = tensile strength
127968.~~75000000~~
26231. = shearing strength one rivet single shear
154199. lbs. = resistance to tearing at middle row and shearing one rivet

Fourth. Crushing in front of four rivets and shearing of one rivet.

FORMULA:

(4DH × T × CS) plus (A × SS) = resistance to crushing in front of four rivets and shearing one rivet

EXAMPLE:

3.7500 = four diameters of rivet hole
.4375 = thickness of plate
187500
262500
112500
1 50000
1.64062500
95000 = crushing strength of rivet material
820312500000
14765625
155859.~~37500000~~
26231 = shearing strength one rivet single shear
182090. lbs. = resistance to crushing in front of four rivets and shearing of one

Fifth. Crushing in front of five rivets, four through straps, the fifth through inner cover plate only, the resistance is:

FORMULA:

(4DH × T × CS) plus (DH × CP × CS) = resistance to crushing of plate in front of five rivets

Example:

diameter of rivet hole = .9375
strap thickness = .3750
468750
65625
28125
crushing strength of rivet = .35156250
95000
175781250000
316406250
33398.~~43750000~~

3.7500 = four diameters of rivet hole
.4375 = thickness of plate
187500
262500
112500
1 50000
1.64062500
95000 = crushing strength of rivet
820312500000
1476562500
155859.~~37500000~~
33398
189257 lbs. = crushing strain of plate in front of five rivets

Rule to find strength of strip of plate at wide pitch.

Formula:

Po × T × TS = strength of plate at wide pitch

Example:

6.7500 = wide pitch
.4375 = thickness of plate
337500
472500
202500
2 70000
2.95312500
60000 = tensile strength
177187.~~50000000~~ lbs. = strength of strip of plate at wide pitch

Rule to find efficiency of joint from these calculations.

Legend:

152578 = strength of net section of plate
177187 = strength of solid plate

Example:

177187)152578.00(.86 = efficiency of joint
141749 6
10828 40
10631 22
197 18

Rule to find safe working pressure from efficiency of joint: Multiply tensile strength of plate by percentage of joint; multiply this sum by twice thickness of plate and divide product by diameter multiplied by factor of safety. The quotient will be the safe working pressure of boiler.

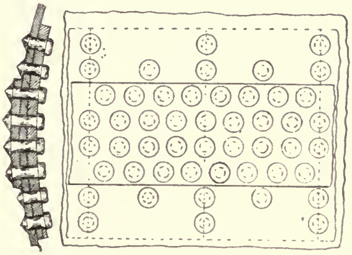
Formula:

$$\frac{TS \times \% \times (2 \times T)}{ID \times F} = \text{safe working pressure}$$

Example:

```
                                       60000 =tensile strength of plate
                                         .86 =percentage of joint
                                     -------
                                     3600 00
                                    48000 0
                                     -------
                                    51600.00
                                        .8750 =twice thickness of plate
                                   ----------
     internal diam. of             258 000000
               boiler = 71.1250   3612 0000
     factor of safety =       5  41280 000
                       -------- -------------
                       355.6250)45150.000000(126.95 =safe working pressure
                                  3556250
                                ---------
                                   9587500
                                   7112500
                                 ---------
                                   24750000
                                   21337500
                                 ----------
                                    34125000
                                    32006250
                                  ----------
                                     21187500
                                     17781250
                                    ---------
                                      3406250
```

QUADRUPLE-RIVETED BUTT JOINT.



Computing strength of a quadruple-riveted butt joint.

Causes for possible failure in a butt joint double strap and quadruple riveted:

First. Tearing of plate through the line of rivets at outer row.
Second. Tearing of plate through line of rivets at second outer row and shearing of outer row of rivets.
Third. Failure of plate through second row of narrow pitch and shearing of the two outer rows of rivets
Fourth. By shearing of all rivets.

Legend:

TS = tensile strength = 60000
SS = shearing strength of rivets material = 38000
CS = crushing strain of material = 95000
T = thickness of plate = $\frac{7}{16}$ = .4375
D = diameter of boiler = 72″
d = diameter of rivets = $\frac{13}{16}$ = .8125
DH = diameter of rivet hole = $\frac{7}{8}$ = .8750
A = area of rivets = $\frac{7}{8}$ = .6013
PN = narrow pitch = $4\frac{1}{16}$ = 4.0625
PW = wide pitch = $8\frac{1}{8}$ = 8.125
Po = outside pitch = $16\frac{1}{4}$″ = 16.2500 or width of strap
N = number of rivets

In connection with this problem it is assumed that the straps or cover plates are three fourths ($\frac{3}{4}$) the thickness of shell plates. Calculations will be made according to points of possible failures.

First. Tearing of plate through the line of rivets at outer row.

Formula:

Po—d = section of plate to resist tearing

EXAMPLE:

16.2500 = outside pitch
.8750 = diameter of ⅞ rivet hole

15.3750 = section of plate to resist tearing less diameter of rivet

To calculate the efficiency of a joint it will be necessary to find out strength of solid plate in strip calculated.

16.2500 = pitch outside row
.4375 = thickness of plate

812500
1137500
487500
650000

7.10937500
60000 = tensile strength

426562.~~50000000~~ lbs. = strength of solid plate at point of calculation.

Second. Tearing of plate at line of rivets next to outer row.

FORMULA:

(Po—2DH) × T × TS + SS of 1d = resistance to tearing of plate at line of 2d outer row

EXAMPLE:

16.2500 = outer pitch or width of strip
1.7500 = two diameters of rivet hole

14.5000
.4375 = thickness of plate

725000
1015000
435000
580000

6.34375000
60000 = tensile strength of plate

380625.~~00000000~~
380625 = lbs. resistance to tearing of plate at second outer row
22849 = strength of the one rivet in outer row

403474 = lbs. resistance at that part of joint

Third. Failure of plate through second row of rivets in narrow pitch and shearing of the two outer rows of rivets.

FORMULA:

(Po—4DH) × T × TS + SS of 3d = lbs. resistance in width of strip

Example:

.6013 = area of one rivet

38000 = shearing strength of rivet

48104000

18039

22849.4000

3 = three rivets

68548.2000

16.2500 = width of strip of plate outer row

3.5000 = diameter of four rivet hole

12.7500

.4375 = thickness of plate

637500

892500

382500

510000

5.57812500

60000 = tensile strength

334687.~~50000000~~

68548 = shearing strength of three rivets in outer rows

403235 = lbs. resistance through net section of plate

Fourth. Point of possible failure by shearing of all rivets. There being three rivets in single shear and eight in double shear.

Formula:

A × SS × N = single shear + N in double shear = shearing strength of rivets in joint

Example:

.6013 = area of 7/8 rivet

38000 = shearing strength in single shear

48104000

18038

22849.4000

3 = number of rivets in single shear

68548.2000 = shearing strength of 3 rivets in single shear

.6013 = area of 7/8 rivet

70300 = shearing strength in double shear

1803900

420910

42271.3900

8 = number of rivets

338171.~~1200~~

Add this latter product to the sum of three rivets in single shear, which gives the total shearing strength of rivets in joint.

68548 = shearing strength of 3 rivets in single shear

338171 = shearing strength of 8 rivets in double shear

406719 lbs. = total shearing strength of rivets in joint

To get the efficiency of joint at this point: Divide resistance of net section of plate by strength of solid plate.

EXAMPLE:

403235 = resistance through net section of plate
426562 = strength of solid plate

426562)403235.000(.945 = per cent. of efficiency
3839058
1932920
1706248
2266720
2132810
133910

Rule to find safe working pressure for boiler from these calculations: Multiply tensile strength by lowest percentage and by twice thickness of plate; divide this product by diameter multiplied by factor of safety.

FORMULA:

$$\frac{TS\times\%\times 2T}{D\times F}=\text{safe working pressure}$$

EXAMPLE:

60000 = tensile strength
.945 = lowest percentage of joint
300000
240000
540000
56700.000
.8750 = twice thickness of plate
2835000000
diam. of boiler = 72″ 396900000
factor of safety = 5 453600000
360)49612.5000000(137.8 = lbs. safe working pressure
360
1361
1080
2812
2520
2925
2880
45

Butt straps or cover plates of a quadruple riveted joint.

Possible causes for failure of butt straps.

First. Both straps breaking across the inner row of rivets.

Second. The plate and inner strap breaking through line of next to inner row of rivets.

Third. The inner strap breaking through the inner row of rivets and shearing rivets.

Fourth. The outer strap breaking through the inside row of rivets and shearing of rivets.

LEGEND:

DH = diameter of rivet hole = ⅞ = .8750
TS = tensile strength = 60000
Po = outer pitch = 16.2500
T = thickness of strap = .3750

First possible cause. Both straps breaking across the inner row of rivets.

FORMULA.

(Po—4DH) × T × N × TS = tensile strength of two straps

EXAMPLE:

16.2500 = outer pitch
3.5000 = four rivet hole diameters

12.7500
.3750 = thickness of strap

6375000
892500
3 82500

4.78125000 = square inches of material at
2 straps (point of possible fracture

9.56250000 = total number of square inches
60000 = tensile strength

573750.00000000 lbs. = tensile strength of the two straps

Showing strength of straps section stronger than plate section.

Second. Point of possible failure—the resistance to fracture at this point is greater than first possible cause.

Third. Possible cause for failure by breaking of strap through line of rivet holes at inner row.

FORMULA:

(Po—4DH) × T × TS + (N × SS) = total resistance to tear plate and shear rivets.

EXAMPLE:

16.2500 = outer pitch
3.5000 = four rivet hole diameters

12.7500
.3750 = thickness of plate

6375000
892500
3 82500

4.78125000
60000 = tensile strength

286875.00000000
182792 = rivet strength

469667 lbs. = resistance to tear plate and shear rivets

22849 = shearing resistance single shear of 7/8 rivets
8 = number of rivets

182792

Fourth. Point of possible failure—same as third point.

These calculations show the straps resistance to strain exceeds the shell plate.

CHAPTER VIII.

SAFE WORKING STEAM PRESSURE OF BOILERS.

AS PRESCRIBED BY THE BOARD OF SUPERVISING INSPECTORS OF STEAM VESSELS OF THE UNITED STATES.

The working steam pressure of a boiler shell is determined by the following rule:

Multiply one-sixth (1-6) of the lowest tensile strength, found stamped on any plate in the cylindrical shell, by the thickness expressed in inches or parts of an inch, of the thinnest plate in the same cylindrical shell, and divide by the radius or half diameter — also expressed in inches — and the sum will be the pressure allowable per square inch of surface for single riveting, to which add 20 per cent. for double riveting when all the holes have been fairly drilled and no part of the hole has been punched.

EXAMPLE.

A boiler 36 inches in diameter, ¼ inch in thickness, tensile strength 60,000 pounds, resolves itself into the following:

$$1/6 \text{ of } 60000 = \frac{10000 \times .25 = 2500}{18} = 138.88 \text{ working steam pressure allowable}$$

for single riveting; for double riveting and drilled holes, 20 per cent. added $=166.65$, this being the pressure allowable by the United States Marine Inspectors.

On the following pages find tables of pressure allowed on various sizes of boiler shells for 50,000, 55,000 and 60,000 pounds tensile strength plates; also a table which simplifies the calculation. Steel plate having a tensile strength of 60,000 pounds is almost universally used by builders of both stationary and marine boilers.

TABLE OF PRESSURE ALLOWABLE ON BOILERS MADE SINCE FEBRUARY 28, 1872.

Diameter of boiler.	Thickness of plates.	45,000 tensile strength, 1–6, 7,500.		50,000 tensile strength, 1–6, 8,333.3.		55,000 tensile strength, 1–6, 9,166.6.		60,000 tensile strength, 1–6, 10,000.		65,000 tensile strength, 1–6, 10,833.3.		70,000 tensile strength, 1–6, 11,666.6.		Diameter of boiler.
		Pressure.	20 per cent additional.	Pressure.	20 per cent additional.	Pressure.	20 per cent additional.	Pressure.	20 per cent additional.	Pressure.	20 per cent additional.	Pressure.	20 per cent additional.	
36 inches	.1875	78.12	93.74	86.8	104.16	95.48	114.57	104.16	124.99	112.84	135.4	121.52	145.82	36 inches
	.21	87.5	105	97.21	116.65	106.94	128.33	116.66	139.99	126.38	151.65	136.11	163.33	
	.23	95.83	114.99	106.47	127.76	117.12	140.54	127.77	153.32	138.41	166.09	149.07	178.88	
	.25	104.16	124.99	115.74	138.88	127.31	152.77	138.88	166.65	150.46	180.55	162.03	194.43	
	.26	108.33	129.99	120.37	144.44	132.4	158.88	144.44	173.32	156.48	187.77	168.51	202.21	
	.29	120.83	144.99	134.25	161.11	147.68	177.21	161.11	193.33	174.53	209.43	187.96	225.55	
	.3125	130.2	156.24	144.67	173.6	159.14	190.96	173.6	208.32	188.07	225.68	202.5	243.04	
	.33	137.5	165	152.77	183.32	168.05	201.66	183.33	219.99	198.61	238.33	213.88	256.65	
	.35	145.83	174.99	162.03	194.43	178.23	213.87	194.44	233.32	210.64	252.76	226.84	272.20	
	.375	156.25	187.5	173.61	208.33	190.97	229.16	208.33	249.99	225.69	270.83	243.05	291.66	
38 inches	.1875	74.01	88.81	82.23	98.67	90.46	108.54	98.68	118.41	106.9	128.28	115.13	138.16	38 inches
	.21	82.89	99.46	92.1	110.52	101.31	121.57	110.52	132.62	119.73	143.67	128.93	154.71	
	.23	90.78	108.93	100.87	121.04	110.96	133.15	121.05	145.26	131.13	157.35	141.22	169.46	
	.25	98.68	118.41	109.64	131.56	120.61	144.73	131.57	157.88	142.54	171.04	153.5	184.20	
	.26	102.63	123.15	114.03	136.83	125.43	150.51	136.84	164.2	148.24	177.88	159.64	191.56	
	.29	114.47	137.36	127.19	152.62	139.91	167.89	152.63	183.15	165.35	198.42	178.06	213.67	
	.3125	123.35	148.02	137.06	164.46	150.76	180.91	164.47	197.36	178.17	213.8	191.88	230.25	
	.33	130.26	156.31	144.73	173.67	159.2	191.04	173.68	208.41	188.15	225.78	202.62	243.14	
	.35	138.15	165.78	153.5	184.21	168.85	202.62	184.21	221.05	199.56	239.47	214.91	257.89	
	.375	148	177.60	164.73	197.67	180.91	217.09	197.36	236.83	213.81	256.57	230.26	276.31	
40 inches	.1875	70.31	84.37	78.12	93.74	85.93	103.11	93.75	112.5	101.56	121.87	109.37	131.24	40 inches
	.21	78.75	94.50	87.49	104.98	96.24	115.48	105	126	113.74	136.48	122.49	146.98	
	.23	86.25	103.5	95.83	114.99	105.41	126.49	115	138	124.58	149.49	134.16	160.99	
	.25	93.75	112.5	104.16	124.99	114.58	137.49	125	150	135.41	162.49	145.83	174.99	
	.26	97.5	117	108.33	129.99	119.16	142.99	130	156	140.83	168.99	151.66	181.99	
	.29	108.75	130.5	120.83	144.99	132.90	159.49	145	174	157.08	188.49	169.16	202.99	
	.3125	117.18	140.61	130.2	156.24	143.22	171.86	156.25	187.45	169.27	203.12	182.29	218.74	
	.33	123.75	148.5	137.49	164.98	151.24	181.48	165	198	178.74	214.48	192.49	230.98	
	.35	131.25	157.5	145.83	174.99	160.41	192.49	175	210	189.58	227.49	204.16	244.99	
	.375	140.62	168.74	156.24	187.48	171.87	206.24	187.5	225	203.12	243.74	218.74	262.48	
42 inches	.1875	66.96	80.35	74.40	89.28	81.84	98.20	89.28	107.13	96.72	116.06	104.16	124.99	42 inches
	.21	75	90	83.32	99.99	91.66	109.99	100	120	108.33	129.99	116.66	139.99	
	.23	82.14	98.56	91.23	109.51	100.39	120.46	109.52	131.42	118.65	142.38	127.77	153.32	
	.25	89.28	107.13	99.2	119.04	109.12	130.94	119.04	142.84	128.96	154.75	138.88	166.65	
	.26	92.85	111.42	103.17	123.8	113.49	136.18	123.8	148.56	134.12	160.94	144.44	173.32	

TABLE OF PRESSURE ALLOWABLE ON BOILERS MADE SINCE FEBRUARY 28, 1872.

Diameter of boiler.	Thickness of plates.	45,000 tensile strength, 1–6, 7,500.		50,000 tensile strength, 1–6, 8,333.3.		55,000 tensile strength, 1–6, 9,166.6.		60,000 tensile strength, 1–6, 10,000.		65,000 tensile strength, 1–6, 10,833.3.		70,000 tensile strength, 1-6, 11,666.6.		Diameter of boiler.
		Pressure.	20 per cent additional.	Pressure.	20 per cent additional.	Pressure.	20 per cent additional.	Pressure.	20 per cent additional.	Pressure.	20 per cent additional.	Pressure.	20 per cent additional.	
	.29	103.57	124.28	115.07	138.08	126.57	151.85	138.09	165.7	149.6	179.52	161.11	193.33	
	.3125	111.6	133.92	124	148.8	136.4	163.68	148.74	178.56	161.2	193.44	173.61	208.23	
	.33	117.85	141.42	130.94	157.12	144.04	172.84	157.14	188.56	170.23	204.27	183.33	219.99	
	.35	125	150	138.88	166.65	152.77	183.32	166.66	199.99	180.55	216.66	194.44	233.32	
	.375	133.92	160.7	148.8	178.56	163.68	196.40	178.57	214.28	193.45	232.14	208.33	249.99	
44 inches	.1875	63.92	76.7	71.02	85.22	78.12	93.74	85.22	102.26	92.32	110.78	99.42	119.3	44 inches
	.21	71.59	85.9	79.54	95.44	87.49	104.98	95.45	114.54	103.4	124.08	111.36	133.63	
	.23	78.4	94.08	87.12	104.54	95.83	114.99	104.54	125.44	113.25	135.9	121.96	146.35	
	.25	85.22	102.26	94.69	113.62	104.16	124.99	113.63	136.35	123.1	147.72	132.56	159.07	
	.26	88.63	106.35	98.48	118.17	108.33	129.99	118.18	141.81	128.02	153.62	137.87	165.44	
	.29	98.86	118.63	109.84	131.80	120.83	144.99	131.81	158.17	142.79	171.33	153.78	184.53	
	.3125	106.53	127.83	118.36	142.03	130.2	156.24	142.04	170.44	153.88	184.65	165.71	198.85	
	.33	112.5	135	124.99	149.98	137.49	164.98	150	180	162.49	194.98	174.99	209.98	
	.35	119.31	143.17	132.57	159.08	145.83	174.99	159.09	190.9	172.34	206.8	185.6	222.72	
	.375	127.81	153.37	142.04	170.44	156.24	187.48	170.45	204.54	184.65	221.58	198.86	238.63	
46 inches	.1875	61.14	73.36	67.93	81.51	74.72	89.66	81.51	97.81	88.31	105.97	95.1	114.12	46 inches
	.21	68.47	82.16	76.08	91.29	83.69	100.42	91.3	109.56	98.91	118.69	106.52	127.82	
	.23	75	90	83.33	100	91.66	109.99	100	120	108.33	129.99	116.66	139.99	
	.25	81.52	97.82	90.57	108.68	99.63	119.55	108.69	130.42	117.75	141.3	126.8	152.16	
	.26	84.78	101.73	94.2	113.04	103.62	124.34	113.44	135.64	122.46	146.95	131.88	158.25	
	.29	94.56	113.47	105.07	126	115.57	138.68	126.09	151.3	136.59	163.92	147.1	176.52	
	.3125	101.9	122.28	113.21	135.86	124.54	149.44	135.86	163.03	147.19	176.62	158.51	190.21	
	.33	107.6	129.12	119.56	143.47	131.52	157.82	143.97	172.16	155.43	186.51	167.39	200.86	
	.35	144.13	136.95	126.8	152.16	139.49	167.38	152.17	182.6	164.85	197.82	177.53	213.03	
	.375	122.28	146.73	135.86	163.03	149.45	179.34	163.04	195.64	176.62	211.94	190.21	228.25	
48 inches	.1875	58.59	70.30	65.1	78.12	71.61	85.93	78.12	93.74	84.63	101.55	91.13	109.35	48 inches
	.21	65.62	78.74	72.91	87.49	80.2	96.24	87.49	104.98	94.79	113.74	102.08	122.49	
	.23	71.87	86.24	79.85	95.82	87.84	105.4	95.83	114.99	103.81	124.57	111.8	133.16	
	.25	78.12	93.74	86.8	104.16	95.48	114.57	104.16	124.99	112.84	135.4	121.52	145.82	
	.26	81.25	97.50	90.27	108.32	99.3	119.16	108.33	129.99	117.36	140.83	126.38	151.65	
	.29	90.62	108.74	100.69	120.82	110.76	132.91	120.83	144.99	130.9	157.08	140.97	169.16	
	.3125	97.65	117.18	108.5	130.2	119.35	143.22	130.21	156.25	141.05	169.26	151.9	182.28	
	.33	103.12	123.74	114.58	137.49	126.04	151.24	137.5	165	148.95	178.74	160.41	192.49	
	.35	109.37	131.24	121.52	145.83	133.67	160.4	145.83	174.99	157.98	189.57	170.13	204.15	
	.375	117.18	140.61	130.2	156.24	143.22	171.86	156.25	187.50	169.27	203.12	182.29	218.74	

Table of Pressure Allowable on Boilers Made Since February 28, 1872.

Diameter of boiler.	Thickness of plates.	45,000 tensile strength, 1-6, 7,500.		50,000 tensile strength, 1-6, 8,333.3.		55,000 tensile strength, 1-6, 9,166.6.		60,000 tensile strength, 1-6, 10,000.		65,000 tensile strength, 1-6, 10,833.3.		70,000 tensile strength, 1-6, 11,666.6.		Diameter of boiler.
		Pressure.	20 per cent additional.	Pressure.	20 per cent additional.	Pressure.	20 per cent additional.	Pressure.	20 per cent additional.	Pressure.	20 per cent additional.	Pressure.	20 per cent additional.	
54 inches	.1875	52.08	62.49	57.87	69.44	63.65	76.38	69.44	82.44	75.23	90.27	81.01	97.21	54 inches
	.21	58.33	69.99	64.81	77.77	71.29	85.54	77.77	93.32	84.25	101.1	90.74	108.88	
	.23	63.88	76.65	70.98	85.17	78.08	93.69	85.18	102.21	92.28	110.73	99.38	119.25	
	.25	69.44	83.32	77.16	92.59	84.87	101.84	92.59	111.10	100.3	120.36	108.02	129.62	
	.26	72.22	86.66	80.24	96.28	88.27	105.92	96.29	115.54	104.31	125.17	112.44	134.8	
	.29	80.55	96.66	89.5	107.40	98.45	118.14	107.41	128.88	116.35	139.62	125.3	150.36	
	.3125	86.8	104.16	96.44	115.73	106.09	127.30	115.55	138.66	125.38	150.45	135.03	162.03	
	.33	91.66	109.99	101.84	122.22	112.03	134.43	122.22	146.66	132.4	158.88	142.59	171.10	
	.35	97.22	116.66	108.02	129.62	118.82	142.58	129.62	155.54	140.43	168.51	151.23	181.47	
	.375	104.16	124.99	115.74	138.88	127.31	152.77	138.88	166.65	150.46	180.55	162.03	194.43	
60 inches	.1875	46.87	56.24	52.08	62.49	57.29	68.74	62.5	75	67.7	81.24	72.91	87.49	60 inches
	.21	52.5	63	58.33	69.99	64.16	76.99	69.99	84	75.83	90.99	81.66	97.99	
	.23	57.5	69	63.88	76.65	70.27	84.32	76.66	91.99	83.05	99.66	89.44	107.32	
	.25	62.5	75	69.44	83.32	76.38	91.65	83.33	99.99	90.27	108.32	97.22	116.66	
	.26	65	78	72.22	86.66	79.44	95.32	86.66	103.99	93.88	112.65	101.11	121.33	
	.29	72.5	87	80.55	96.66	88.61	106.33	96.66	115.99	104.72	125.66	112.77	135.32	
5/16	.3125	78.12	93.74	86.8	104.16	95.48	114.57	104.18	124.99	112.95	135.54	121.52	145.82	
21/64	.33	82.5	99	91.66	109.99	100.83	120.99	109.99	132	119.16	142.99	128.33	153.99	
11/32	.35	87.5	105	97.22	116.66	106.94	128.32	116.66	139.99	126.38	151.65	136.11	163.33	
3/8	.375	93.75	112.5	104.16	124.99	114.58	137.49	125	150	135.41	162.49	145.83	174.99	
66 inches	.1875	42.61	51.13	47.34	56.8	52.07	62.49	56.81	68.17	61.55	73.86	66.28	79.53	66 inches
	.21	47.72	57.26	53	63.63	58.33	69.99	63.63	76.35	68.93	82.71	74.24.	89.08	
	.23	52.27	62.72	58	69.69	63.88	76.65	69.69	83.62	75.5	90.6	81.31	97.57	
	.25	56.81	68.17	63.13	75.75	69.44	83.32	75.75	90.90	82.07	98.48	88.38	106.06	
	.26	59.09	70.9	65.65	78.78	72.22	86.66	78.78	94.53	85.35	102.42	91.91	110.29	
	.29	65.90	79.08	73.23	87.87	80.55	96.66	87.87	105.44	95.2	114.24	102.52	123.02	
	.3125	71	85.2	78.91	94.69	86.89	104.16	94.69	113.62	102.58	123.09	110.47	132.56	
	.33	75	90	83.33	99.99	91.66	109.99	99.99	120	108.33	127.99	116.66	139.99	
	.35	79.56	95.47	88.38	106.05	97.22	116.66	106	127.27	114.89	137.86	123.73	148.47	
	.375	85.22	102.26	94.69	113.62	104.16	124.99	113.62	136.34	123.1	147.72	132.57	159.08	
72 inches	.1875	39.06	46.87	43.4	52.08	47.74	57.28	52.08	62.49	56.42	67.70	60.76	72.91	72 inches
	.21	43.75	52.5	48.6	58.33	53.47	64.16	58.33	69.99	63.19	75.82	68.05	81.66	
	.23	47.91	57.49	53.24	63.88	58.56	70.27	63.88	76.65	69.21	83.05	74.53	89.43	
	.25	52.08	62.49	57.87	69.44	63.65	76.38	69.44	83.32	75.22	90.26	81.01	97.21	
	.26	54.16	64.99	60.18	72.21	66.2	79.44	72.22	86.66	78.24	93.88	84.25	101.10	

TABLE OF PRESSURE ALLOWABLE ON BOILERS MADE SINCE FEBRUARY 28, 1872.

Diameter of boiler.	Thickness of plates.	45,000 tensile strength, 1–6, 7,500.		50,000 tensile strength, 1–6, 8,333.3.		55,000 tensile strength, 1–6, 9,166.6.		60,000 tensile strength, 1–6, 10,000.		65,000 tensile strength, 1–6, 10,833.3.		70,000 tensile strength, 1-6, 11,666.6.		Diameter of boiler.
		Pressure.	20 per cent additional.	Pressure.	20 per cent additional.	Pressure.	20 per cent additional.	Pressure.	20 per cent additional.	Pressure.	20 per cent additional.	Pressure.	20 per cent additional.	
	.29	60.41	72.49	67.12	80.54	73.84	88.60	80.55	96.66	87.26	104.71	93.98	112.77	
	.3125	65.10	78.12	72.33	86.8	79.57	95.48	86.8	104.16	94.03	112.83	101.27	121.52	
	.33	68.75	82.5	76.38	91.65	84.02	100.82	91.66	109.99	99.3	119.16	106.94	128.32	
	.35	72.91	87.49	81.01	97.21	89.11	106.93	97.22	116.66	105.32	126.38	113.42	136.1	
	.375	78.12	93.74	86.8	104.16	95.48	114.57	104.16	124.99	112.84	135.43	121.52	145.82	
78 inches	.1875	36.05	43.21	40.06	48.07	44.07	52.87	48.07	57.68	52.08	62.49	56.08	67.29	78 inches
	.21	40.38	48.45	44.87	53.84	49.35	59.22	53.84	64.60	58.33	69.99	62.82	75.38	
	.23	44.23	53.07	49.14	58.96	54.05	64.86	58.95	70.76	63.88	76.65	68.80	82.56	
	.25	48.07	57.68	53.41	64.09	58.76	70.5	64.4	76.92	69.44	83.32	74.78	89.73	
	.26	50	60	55.55	66.66	66.11	73.33	66.66	79.99	72.22	86.66	77.77	93.32	
	.29	55.76	66.91	61.96	74.35	68.16	81.79	74.35	89.22	80.55	96.66	86.75	104.1	
	.3125	60.09	72.1	66.77	80.12	73.45	88.14	80.12	96.14	86.8	104.16	93.48	112.17	
	.33	63.46	76.15	70.51	84.61	77.56	93.07	84.61	101.53	91.66	109.99	98.71	118.45	
	.35	67.3	80.76	74.78	89.73	82.26	98.71	89.74	107.68	97.22	116.66	104.70	125.64	
	.375	72.11	86.53	80.12	96.14	88.14	105.76	96.15	115.38	104.16	124.99	112.17	134.6	
84 inches	.1875	33.48	40.17	37.2	44.68	40.92	49.1	44.64	53.56	48.36	58.03	52.08	62.49	84 inches
	.21	37.5	45	41.66	49.99	45.83	54.99	50	60	54.16	64.99	58.33	69.99	
	.23	41.02	49.22	45.63	54.75	50.19	60.22	54.75	65.71	59.32	71.18	63.88	76.66	
	.25	44.64	53.56	49.6	59.52	54.56	65.47	59.52	71.42	64.48	77.37	69.44	83.32	
	.26	46.42	55.7	51.58	61.89	56.74	68.08	61.9	74.28	67.05	80.46	72.22	86.66	
	.29	51.78	62.13	57.53	69.03	63.29	75.94	69.04	82.84	74.8	89.76	80.55	96.66	
	.3125	55.8	66.96	62	74.4	68.2	81.84	74.4	89.28	80.6	96.72	86.8	104.16	
	.33	58.92	70.7	65.47	78.56	72.02	86.42	78.57	94.28	85.11	102.13	91.66	109.99	
	.35	62.5	75	69.44	83.32	76.38	91.65	83.33	99.99	90.27	108.32	97.22	116.66	
	.375	66.96	80.35	74.4	89.28	81.84	98.2	89.28	107.13	96.72	116.06	104.16	124.99	
90 inches	.1875	31.25	37.5	34.72	41.66	38.19	45.82	41.66	49.99	45.13	54.15	48.68	58.33	90 inches
	.21	35	42	38.88	46.65	42.77	51.32	46.66	55.99	50.55	60.66	54.44	65.32	
	.23	38.33	45.99	42.59	51.10	46.85	56.22	51.11	61.33	55.37	66.44	59.62	71.54	
	.25	41.66	49.99	46.29	55.54	50.92	61.1	55.55	66.66	60.18	72.21	64.81	77.77	
	.26	43.33	51.99	48.14	57.76	52.96	63.55	57.77	69.32	62.59	75.1	67.4	80.88	
	.29	48.33	57.99	53.7	64.44	59.07	70.8	64.44	77.32	69.81	83.77	75.18	90.21	
	.3125	52.08	62.49	57.86	69.43	63.65	76.38	69.44	83.32	75.23	90.27	81.01	97.21	
	.33	55	66	61.11	73.33	67.22	80.66	73.33	87.99	79.44	95.32	85.55	102.66	
	.35	58.33	69.99	64.81	77.77	71.29	85.54	77.77	93.32	84.25	101.1	90.72	108.88	
	.375	62.5	75	69.44	83.32	76.38	91.65	83.33	99.99	90.27	108.32	97.22	116.66	

Table of Pressure Allowable on Boilers Made Since February 28, 1872.

Diameter of boiler.	Thickness of plates.	45,000 tensile strength, 1-6, 7,500.		50,000 tensile strength, 1-6, 8,333.3.		55,000 tensile strength, 1-6, 9,166.6.		60,000 tensile strength, 1-6, 10,000.		65,000 tensile strength, 1-6, 10,833.3.		70,000 tensile strength, 1-6. 11,666.6.		Diameter of boiler.
		Pressure.	20 per cent additional.	Pressure.	20 per cent additional.	Pressure.	20 per cent additional.	Pressure.	20 per cent additional.	Pressure.	20 per cent additional.	Pressure.	20 per cent additional.	
96 inches	.1875	29.29	35.14	32.55	39.06	35.8	42.96	39.06	46.87	42.31	50.77	45.57	54.68	96 inches
	.21	32.81	39.37	36.45	43.74	40.1	48.12	43.75	52.5	47.39	56.86	51.04	61.24	
	.23	35.93	43.11	39.93	47.91	43.92	52.7	47.91	57.49	51.9	62.28	55.9	67.08	
	.25	39.06	46.87	43.4	52.08	47.74	57.28	52.08	62.49	56.42	67.67	60.76	72.91	
	.26	40.62	48.74	45.14	54.16	49.65	59.58	54.16	64.99	58.78	70.53	63.19	75.82	
	.29	45.31	54.37	50.34	60.4	55.38	66.45	60.41	72.49	65.45	78.54	70.48	84.57	
	.3125	48.82	58.58	54.25	65.1	59.67	71.6	65.1	78.12	70.52	84.62	75.95	91.14	
	.33	51.56	61.87	57.29	68.74	63.02	75.62	68.75	82.5	74.47	89.36	80.2	96.24	
	.35	54.68	65.61	60.76	72.91	66.83	80.19	72.91	87.49	78.99	94.78	85.06	102.07	
	.375	58.58	70.29	65.1	78.12	71.61	85.93	78.12	93.74	84.63	101.55	91.14	109.6	

Note.—At the heads of the double columns will be found the tensile strength of the plates *per square inch of section*, also one-sixth of that amount. The pressure allowable on single-riveted boilers will be found in the first division of the double columns under the tensile strength and opposite the diameters and thickness; and, in the second divisions, the pressures allowable on boilers where all the rivet holes have been fairly drilled and no part of such holes has been punched, and the longitudinal laps of their cylindrical parts double riveted.

The pressure for any dimension of boiler not found in the above table must be ascertained in the manner prescribed.

The following rules and tables are from a commercial rating and only approximate.

STANDARD STEAM BOILER MEAUSUREMENTS.

HORIZONTAL TUBULAR.

Based on 12 square feet of heating surface to a horse power.

A Commercial Rating.

Size.		Thickness.			Boiler with Hand Holes.				Boiler with Man Holes.			
Dia.	Length	Shell.	Heads.	Size of Dome.	Tubes No.	Dia.	Heat. Surf. sq. ft.	Horse Power	Tubes No.	Dia.	Heating. Surf. sq. ft.	Horse Power.
30	6	¼	⅜	16x20	19	2½	106	9				
30	8	¼	⅜	16x20	19	2½	141	12				
36	8	¼	⅜	18x20	38	2½	256	21				
					28	3	226	19				
					25	3½	234	20				
36	10	¼	⅜	18x20	38	2½	311	26				
					28	3	283	24				
					25	3½	292	24				
42	10	¼	⅜	20x24	38	3	372	31				
					34	3½	385	32				
42	12	¼	⅜	20x24	38	3	446	37				
					34	3½	462	39				
42	14	¼	⅜	20x24	38	3	520	43				
					34	3½	539	45				
42	16	¼	⅜	20x24	38	3	595	43				
					34	3½	616	51				
44	12	¼	⅜	24x24	48	3	544	45				
					38	3½	510	43				
44	14	¼	⅜	24x24	48	3	635	53				
					38	3½	491	41				
48	12	5/16	7/16	24x24	58	3	647	54	50	3	572	48
					50	3½	651	54	34	3½	475	40
48	14	5/16	7/16	24x24	58	3	755	63	50	3	667	55
					50	3½	759	63	34	3½	547	46
48	16	5/16	7/16	24x24	58	3	862	72	50	3	762	64
					50	3½	867	72	34	3½	633	53
48	18	5/16	7/16	24x24	58	3	970	81	50	3	857	71
					50	3½	976	81	34	3½	712	59
54	14	5/16	½	30x30	71	3	912	76	59	3	780	65
					56	3½	851	71	48	3½	748	62
					43	4	763	64	40	4	719	60
54	16	5/16	½	30x30	71	3	1042	87	59	3	891	74
					56	3½	972	81	48	3½	855	71
					43	4	802	67	40	4	821	68

The above table is based on rule for ascertaining Heating Surface.

A commercial rating of boiler horse power is obtained by the following rule:

Add to two-thirds of boiler shell area, tube area and the area of one head (this will compensate for tubes holes in both) and divide product by unit of H. P. according to type of boiler. (See table.)

FORMULA:

$$\frac{\frac{2}{3}\,SA+TA+AH}{HP\ \text{unit}}=HP$$

LEGEND:

SA = shell area
TA = tube area
AH = area of head
60″ = boiler diameter
16′ = length
46 4″ tubes
HP unit = 12 sq. ft.

diameter of head = 60″
60
3600
area of one inch = .7854
14400
18000
28800
25200
area head = 2827.4400

EXAMPLE:

3.1416 = circumference of one inch
60″ = diameter of boiler
188.4960
192″ = length of boiler
3769920
16964640
1884960
3)36191.2320 = area of boiler shell
12063.7440 = ⅓
2
24127.4880 = ⅔ of boiler shell area
2827.4400 = area of one head
26954.9280
110986.4448 = tube area

inches per square ft. = 144)137941.3$\cancel{7}\cancel{2}\cancel{8}$(957.9 = square feet of heating surface
1296
834
720
1141
1008
1333
1296
37

3.1416 = circumference of 1 in.
4″ = tube diameter
12.5664
192″ = length of tube
251328
1130976
125664
2412.7488 = heating surface one tube
46 tubes
144764928
96509952
110986.4448 = tube area

calculating 12 square ft. per HP = 12)957.9(79.8 = HP
84
117
108
99
96
3

Heating surface proper means any portion of the boiler where heat is applied to one side of the plate, and water on the other.

The heating surface of a round furnace and tubes is figured by their internal diameter, water tubes and external fired surfaces are measured by their outside diameter, this latter being the surface heated must necessarily be considered as effective heating surface.

The heating surface of boilers can readily be obtained from the following table: In the case of horizontal tubular bricked in boilers, two-thirds of the boiler shell, the whole of the tube surface, and the front and rear head deducting area of tubes and surface above water-line is figured as effective heating surface.

Diameter of boiler, inches	26	28	30	32	34	36	38	40	42	44	46	48
Two-thirds of the heating surface of shell per foot of length	4.54	4.89	5.24	5.59	5.93	6.29	6.63	6.98	7.33	7.68	8.03	8.38
Diameter of boiler, inches	50	52	54	56	58	60	62	64	66	68	70	72
Two-thirds of the heating surface of shell per foot of length	8.73	9.08	9.42	9.77	10.12	10.47	10.82	11.17	11.52	11.87	12.22	12.57

TYPES OF BOILERS AND ESTIMATED GRATE TO HEATING SURFACE PER HORSE POWER.

Types.	Square feet of Heating Surface per horse power.	Square feet of Heating Surface to one foot of grate.
Cylinder	6 to 10	12 to 15
Flue	8 to 12	20 to 25
Horizontal Tubular	12 to 14	25 to 35
Water Tube	11 to 12	35 to 40
Vertical	10 to 12	25 to 30
Internal Fired	12 to 15	50 to 100

RATIO GRATE SURFACE TO HORSE POWER.

Type of Boiler.	Ratio.
HT	4 to 6
WT	3
Loco	.02 " 6
Marine	.12

HEATING SURFACE RATIO TO GRATE SURFACE.

HT	.40 to 50
WT	.34 " 65
Loco	.30 " 34
Marine	.28 " 32

COAL AND GRATE.

The average consumption of coal for steam boilers is 12 pounds per hour for each square foot of grate surface.

Western coals, having a large amount of sulphur, require more space in furnace and more air.

Rule to find area of grate for a given boiler:

Divide pounds of water to be evaporated per hour by number of pounds of water evaporated multiplied by number of pounds of coal burned per hour per square foot of grate.

FORMULA:

$$\frac{\text{number of lbs. of water evaporated per hour}}{\text{water in lbs. evap.} \times \text{per lbs. of coal per hour}} = \text{area of grate}$$

LEGEND:

2400 = lbs. of water to be evaporated
12 = lbs. of coal per square foot of grate
9 = lbs. of water

EXAMPLE:

108)2400 (22 square feet of grate required
216
———
240
216
———
24

12 lbs. of coal per sq. ft. of grate
9 lbs. of water per lbs. of coal
———
108 lbs. of water evaporated per sq. foot of grate

TABLE FOR PRESSED STEEL BOILER LUGS.

Iron rivets have a shearing strength of 38000 lbs.
Steel “ “ “ “ “ “ 45000 “

See tables for boiler weights and rivet strength.

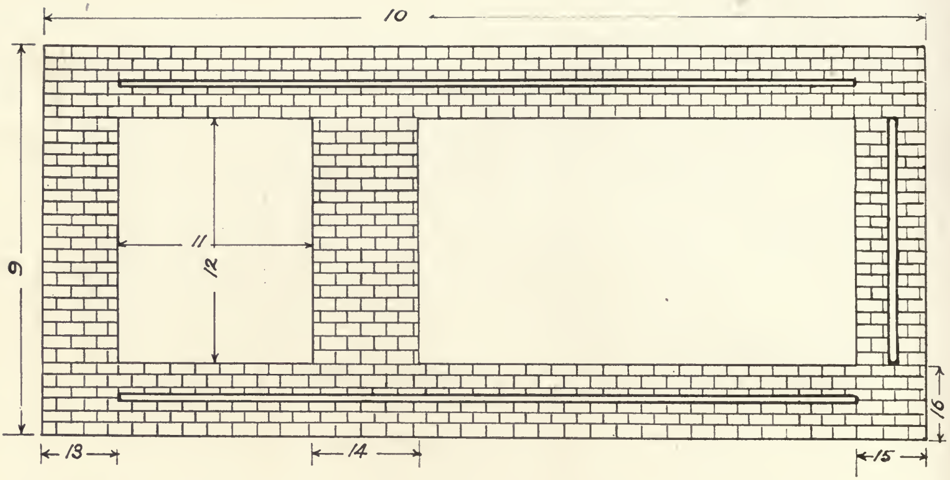
Diameter of boiler, inches.	Height of base of lug above center of boiler.	Width of lug.	Length of lug projection.	Height of lug on boiler.	Thickness.	Weight, lbs.
30	1	7	7	7	3/16	6
36	2	7	7	7	1/4	8
42	2 7/8	8	8	8	1/4	10 1/2
48	3 7/8	8	8	8	5/16	14
54	3 1/2	10	10	10	5/16	20 1/2
60	4 1/2	10	10	10	3/8	23
66	4 1/2	12	12	12	3/8	35
72	5	12	12	12	7/16	40
78	6	12	12	12	1/2	45
84	7	12	12	12	9/16	50

WEIGHT OF HORIZONTAL TUBULAR BOILERS FOR 125 LBS. STEAM PRESSURE COMPLETE WITH FITTINGS FULL OF WATER.

Diameter of boiler, inches	36	36	42	42	44	48	50	54	54	60	60
Length in feet . .	8	10	10	12	12	12	13	13	15	14	15
Weight full of water	6,100	7,600	9,500	10,600	11,600	13,400	14,300	15,400	17,900	20,900	24,900
Diameter of boiler, inches.	60	66	66	72	72	78	78	84	84	90	90
Length in feet	16	16	16	16	18	18	20	18	20	18	20
Weight full of water	27,300	30,400	35,100	40,100	44,100	48,100	56,100	55,100	67,100	65,100	75,100

DIRECTIONS FOR SETTING BOILERS.

Make the excavation to a depth suitable to ground that boiler is to rest upon not less than 24 inches. Build foundation walls at least 12″ wider than walls to floor level, fronts to rest upon two courses of brick above the floor level. Set boiler in place and block it up three or four inches higher than it is to remain, the back side of front to set back four inches from front edge of brick work. Carry up the side and end walls to the proper height for the resting place of brackets (if boiler has brackets place rollers between plates and lugs) leaving space so that walls will not be pushed out of place by expansion of boiler. (Some engineers prefer an air space in setting side and end walls, as a nonconductor of heat.) The walls should be tied together by headers and run every eighteen inches. The headers from outside walls should touch those of inner wall and not be tied together. Fire brick in the furnace should be laid with a course of headers every six courses so that the wall can easily be taken out and repaired at any time when necessary. The rear connection or back arch should be lined with fire brick, the ends of arch resting on side walls and the arch of such radius to permit of easy access to tubes at rear head. A space of one inch should be left between rear end of boiler and inside of arch so that the expansion of boiler will not affect brick work and should be so arranged that it can be removed without injury to walls. It is preferable when covering a boiler to do so with magnesia, as it is light, a non-conductor and will give evidence of any leakage at a local point by discoloration or becoming soft, not like the brick covered boiler that may have leakage many feet from point of steam issuing. If brick is to be used a two inch space should be left between boiler and brick work.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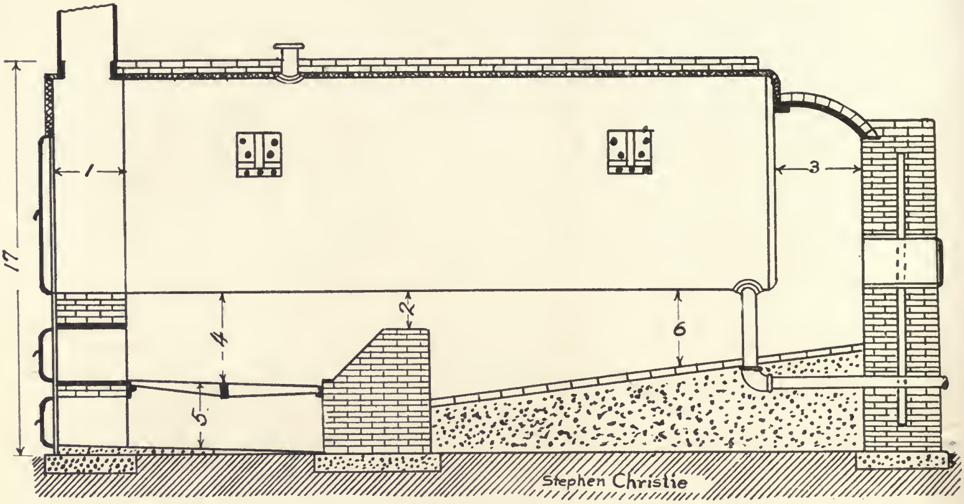
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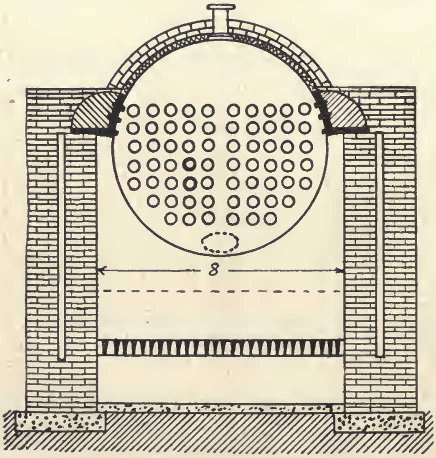
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MEASUREMENTS FOR SETTINGS RETURN TUBULAR BOILERS FULL FLUSH FRONTS.

SEE PLAN AND ELEVATION.

Horse Power.	Diameter of Boiler.	Length of Boiler.	Front-Smoke Box.	Top of Bridge Wall to bottom of Boiler.	Back Head to Back Wall.	Top of Grates to bottom of Boiler.	Top of Grates to Floor Line.	Top of Filling to bottom of Boiler.	Width of Furnace.	Width of Walls over all.	Length of Foundation Walls.	Length of Furnace.	Width of Furnace.	Thickness of Front Brick Work.	Thickness of Bridge Wall.	Thickness of Back Wall.	Thickness of Side Walls.	Height of Walls from Floor Line.
No. H. P.	...... Dia. In.	...... Lgth. Ft.	1 In.	2 In.	3 In.	4 In.	5 In.	6 In.	8 In.	9 Ft. In.	10 Ft. In.	11 In.	12 In.	13 In.	14 In.	15 In.	16 In.	17 Ft. In.
15	36	8	12	10	16	26	20	20	36	7— 4	12—7	36	36	13	18	20	18	7— 0
20	36	10	12	10	16	26	20	20	36	7— 4	14—7	42	36	13	18	20	18	7— 0
25	42	10	12	12	20	26	22	22	42	7—10	14—7	42	42	13	20	20	20	7— 7
30	42	12	12	12	20	26	22	22	42	7—10	16—7	48	42	13	20	20	20	7— 7
35	44	12	12	12	20	26	22	22	44	8— 0	16—7	48	44	13	20	20	20	7—10
40	48.	12	14	12	22	26	24	22	48	8— 4	16—7	48	48	15½	24	24	24	8— 0
45	50	13	14	14	22	28	24	22	50	8— 6	17—7	48	50	15½	24	24	24	8— 1
50	54	13	16	15	24	28	25	24	54	8—10	17—7	54	54	15½	24	24	24	8— 6
60	54	15	16	15	24	30	25	24	54	8—10	19—7	60	54	15½	24	24	24	8— 6
75	60	15	18	15	28	30	26	24	60	10— 0	19—7	60	60	15½	28	26	30	8—11
80	60	16	18	16	28	30	26	24	60.	10— 0	20—7	60	60	15½	28	26	30	8—11
100	66	16	18	16	28	34	26	26	66	10— 6	20—7	66	66	15½	28	26	30	9— 8
125	72	16	20	18	30	34	28	26	72	11— 0	20—7	72	72	16	28	26	30	10— 1
150	72	18	20	18	30	34	28	26	72	11— 0	22—7	78	72	16	30	26	30	10— 1
175	78	18	20	18	30	34	28	26	72	11— 6	22—7	78	78	16	30	26	30	10— 6

MATERIALS FOR BRICKWORK OF REGULAR TUBULAR BOILERS.

SINGLE SETTING.

Boilers.	Common Brick.	Fire Brick.	Sand, bushels	Cement, barrels.	Fire Clay, lbs.	Lime, barrels.
30 inches x 8 feet	5200	320	42	5	192	2
30 " x 10 "	5800	320	46	5½	192	2¼
36 " x 8 "	6200	480	50	6	288	2½
36 " x 9 "	6600	480	53	6½	288	2¾
36 " x 10 "	7000	480	56	7	288	3
36 " x 12 "	7800	480	62	8	288	3¼
42 " x 10 "	10000	720	80	10	432	4
42 " x 12 "	10800	720	86	11	432	4¼
42 " x 14 "	11600	720	92	11¾	432	4½
42 " x 16 "	12400	720	99	12½	432	5
48 " x 10 "	12500	980	100	12½	590	5¼
48 " x 12 "	13200	980	108	13½	590	5½
48 " x 14 "	14200	980	116	14½	590	5¾
48 " x 16 "	15200	980	124	15½	590	6
54 " x 12 "	13800	1150	108	13¾	690	5½
54 " x 14 "	14900	1150	117	15	690	6
54 " x 16 "	16000	1150	126	16	690	6¼
60 " x 10 "	13500	1280	108	13½	768	5½
60 " x 12 "	14800	1280	118	14¾	768	6
60 " x 14 "	16100	1280	128	16	768	6½
60 " x 16 "	17400	1280	140	17½	768	7
60 " x 18 "	18700	1280	148	18¾	768	7½
66 " x 16 "	19700	1400	157	19¾	840	8
72 " x 16 "	20800	1550	166	20¾	930	8½

TWO BOILERS IN A BATTERY.

30 inches x 8 feet	8900	640	70	9	384	3½
30 " x 10 "	9600	640	76	9½	384	4
36 " x 8 "	10500	960	84	10½	576	4¼
36 " x 9 "	11100	960	88	11	576	4½
36 " x 10 "	11800	960	95	12	576	4¾
36 " x 12 "	13000	960	104	13	576	5¼
42 " x 10 "	17500	1440	140	17½	864	7
42 " x 12 "	18600	1440	148	18½	864	7½
42 " x 14 "	19900	1440	159	20	864	8
42 " x 16 "	21200	1440	168	21	864	8½
48 " x 10 "	21400	1960	170	21½	1180	8¾
48 " x 12 "	22300	1960	178	22⅓	1180	9
48 " x 14 "	23900	1960	190	24	1180	9½
48 " x 16 "	25100	1960	200	25	1180	10
54 " x 12 "	23300	2300	186	23⅓	1380	9⅓
54 " x 14 "	24800	2300	198	25	1380	10
54 " x 16 "	26300	2300	210	26⅓	1380	10½
60 " x 10 "	22600	2560	180	22½	1536	9
60 " x 12 "	24800	2560	198	25	1536	10
60 " x 14 "	26800	2560	214	27	1536	10¾
60 " x 16 "	28900	2560	230	29	1536	11½
60 " x 18 "	31000	2560	248	31	1536	12½
66 " x 16 "	33100	2800	264	33	1680	13¼
72 " x 16 "	34000	3100	272	34	1860	13¾

In connection with boiler setting the following information will be useful:

One barrel of lime will lay 800 brick.

Two barrels of lime will lay one perch rubble stone.

To every barrel of lime estimate about ⅝ yards of good sand for brick work.

One and one quarter barrels of cement and three quarters yard of sand will lay 100 feet of rubble stone.

Rule to find number of brick required: Multiply the number of cubic feet by 22.5.

The cubic feet is found by multiplying length by height, then by thickness.

Bricks are usually made 8″ × 4″ × 2″ requiring 27 bricks to make a cubic foot without mortar, the latter is estimated to fill one sixth of space.

CHIMNEYS AND STACKS.

The use for chimneys is necessary in many plants and maintained at great expense of heat units varying as high as 30 per cent of fuel. The necessity arises from following causes, viz.: cost of installing modern methods and the necessity for a chimney to carry off obnoxious gases.

The main object is to obtain air supply for combustion of fuel. Areas for chimneys are calculated from grate area, coal burned in a certain time and usually a ratio of 8 to 1.

The temperatures of gases escaping up a chimney will depend on the material and distance from boilers—the higher the temperature the greater the velocity.

The weight of air necessary for fuels varies, hence the necessity for computing for the maximum amount.

The volume of air is proportional to its temperature; 24 pounds of air at the mean of the atmosphere temperature is 300 cubic feet and at a temperature of 550 degrees F is twice as great.

Rule to find the volume of one pound of air under atmospheric pressure for a given temperature: Divide the absolute temperature of air by the constant 40; the result gives the volume in cubic feet nearly.

LEGEND:

Temp. of atmosphere 80
Constant 40

EXAMPLE:

40)80(2 = volume of one pound in cubic feet
80
—

The intensity of draft is independent of the area of the flue but is proportional to the difference in weight of two columns of air of equal base, one internal and one external. The difference in temperatures between the volume escaping from the inside and the atmosphere increases the draft as the difference between the temperature increases.

The atmospheric pressure or draft is estimated by the height of an equivalent column of water.

CONSIDERATIONS GOVERNING THE HEIGHT OF A CHIMNEY.

It must be high enough to give the required intensity of draft at an economical flue temperature, and to be well above the surrounding objects; increased capacity is much more cheaply gained by increasing the area, it being cheaper to build nearer the ground, and the capacity increases with the square of the diameter and only as the square root of the height. If of brick the height should not exceed ten or eleven times the base, on account of stability.

Rule to find the difference in pressure to be expected between the inside and outside of a chimney for a given height and temperature: Divide 39 by the absolute (actual temperature Fahrenheit plus 461) temperature of the outside air; again, divide 40 by the absolute average temperature of the gases in the stack; subtract the latter from the former quotient, multiply the remainder by the height of the chimney in feet, and divide by 5.2; the final quotient will be the draft in inches in water.

The following table will give the draft power in inches of water for chimneys of specific height basing the temperature as follows:

Escaping gases 552 degrees F.

Atmospheric temperature 62 degrees F.

Height of Chimney in Feet.	Draft Power in Inches of Water.	Theoretical velocity in feet per second.	
		Cold Air Entering.	Hot Gases Escaping.
10	.073	17.8	35.6
20	.146	25.3	50.6
30	.219	31.0	62.0
40	.292	35.7	71.4
50	.365	40.0	80.0
60	.438	43.8	87.6
70	.511	47.3	94.6
80	.585	50.6	101.2
90	.657	53.7	107.4
100	.730	56.5	113.0
120	.876	62.0	124.0
150	1.095	69.3	138.6
175	1.277	74.8	149.6
200	1.460	80.0	160.0

Draft required depends largely on quality and nature of fuel and rate of combustion; it is least for wood and free burning fuels and greatest for fine coal; for slack coal draft equivalent to 1¼ inches of water is necessary.

In designing height of chimney it is the aim to provide for an excess of demands and regulate by dampers to amount required.

Increasing height will increase the flow of escaping gases.

AREA OF CHIMNEY WHEN HORSE POWER IS GIVEN.

Three horse power per square foot of grate surface.

Rule.— Divide the horse power by 3.33 times the square root ot the height. The quotient will be the required effective area in square feet. To the diameter or length of side required to give this area add two inches to compensate for friction.

HORSE POWER OF A GIVEN CHIMNEY.

Rule.— From the area in square feet subtract .6 of the square root of that area and multiply the remainder by the square root of the height and by 3.33.

Or:

Multiply the area in square inches by the square root of the height in feet and divide by 40. The quotient will be the horse power.

SIZE OF CHIMNEYS FOR STEAM BOILERS—KENT.

Formula, $H. P. = 3.33(A - 0.6\sqrt{A})\sqrt{H}$. (Assuming 1 H. P. = 5 lbs. of coal burned per hour.)

Diam. inches	Area A. sq. ft.	Effective Area $E = A - 0.6\sqrt{A}$ sq. ft.	Height of Chimney														Equivalent Square Chimney Side of Square $\sqrt{E} + 4$ inches
			50 ft.	60 ft.	70 ft.	80 ft.	90 ft.	100 ft.	110 ft.	125 ft.	150 ft.	175 ft.	200 ft.	225 ft.	250 ft.	300 ft.	
			Commercial Horse-power of Boiler														
18	1.77	.97	23	25	27	29											16
21	2.41	1.47	35	38	41	44											19
24	3.14	2.08	49	54	58	62	66										22
27	3.98	2.78	65	72	78	83	88										24
30	4.91	3.58	84	92	100	107	113	119									27
33	5.94	4.48		115	125	133	141	149	156								30
36	7.07	5.47		141	152	163	173	182	191	204							32
39	8.30	6.57			183	196	208	219	229	245	268						35
42	9.62	7.76			216	231	245	258	271	289	316	342					38
48	12.57	10.44				311	330	348	365	389	426	460	492				43
54	15.90	13.51					427	449	472	503	551	595	636	675			48
60	19.64	16.98					536	565	593	632	692	748	800	848	894		54
66	23.76	20.83						694	728	776	849	918	981	1040	1097	1201	59
72	28.27	25.08						835	876	934	1023	1105	1181	1253	1320	1447	64
78	33.18	29.73							1038	1107	1212	1310	1400	1485	1565	1715	70
84	38.48	34.76							1214	1294	1418	1531	1637	1736	1830	2005	75
90	44.18	40.19								1496	1639	1770	1893	2008	2116	2318	80
96	50.27	46.01								1712	1876	2027	2167	2298	2423	2654	86
102	56.75	52.23								1944	2130	2300	2459	2609	2750	3012	91
108	63.62	58.83								2090	2399	2592	2771	2939	3098	3393	96
114	70.88	65.83									2685	2900	3100	3288	3466	3797	101
120	78.54	73.22									2986	3226	3448	3657	3855	4223	107
132	95.03	89.18									3637	3929	4200	4455	4696	5144	117
144	113.10	106.72									4352	4701	5026	5331	5618	6155	128

For pounds of coal burned per hour for any given size of chimney, multiply the figures in the table by 5.

Following is a table by Professor Trowbridge:

Height in feet.	Pounds of Coal burned per hour per square foot of section of chimney.	Pounds of Coal burned per hour per square foot, the ratio of grate to chimney being 8 to 1.
20	60	7.5
25	68	8.5
30	76	9.5
35	84	10.5
40	93	11.6
45	99	12.4
50	105	13.1
55	111	13.8
60	116	14.5
65	121	15.1
70	126	15.8
75	131	16.4
80	135	16.9
85	139	17.4
90	144	18.0
95	148	18.5
100	152	19.0
105	156	19.5
110	160	20.0

CHIMNEYS.

Area of chimney for given height and number of square feet of grate surface connected.

Rule.— Multiply the number of square feet of grate surface by 120, and divide by the square root of the height. The quotient will be the required cross section in square inches. See table.

AREA OF CHIMNEYS, IN SQUARE INCHES, WHEN CONNECTED TO GRATE SURFACES FROM TWENTY-FIVE TO ONE THOUSAND SQUARE FEET, AND FOR DIFFERENT HEIGHTS.

Square feet of Grate Surface.	60	65	70	75	80	85	90	95	100	105	110	115	120	125	130	135	140	145	150	155	160	Square feet of Grate Surface
25	387	372	358	346	335	325	316	307	300	292	286	....	....	....	....	....	....	....	....	....	....	25
50	774	744	717	692	670	650	632	615	600	585	572	....	....	....	....	....	....	....	....	....	....	50
75	1161	1116	1055	1039	1006	976	948	923	900	878	858	839	....	....	....	....	....	....	....	....	....	75
100	1549	1488	1434	1385	1341	1301	1264	1231	1200	1171	1144	1119	....	....	....	....	....	....	....	....	....	100
125	1936	1860	1792	1732	1677	1626	1581	1538	1500	1463	1430	1398	1369	....	....	....	....	....	....	....	....	125
150	2322	2232	2151	2078	2012	1952	1897	1846	1800	1756	1716	1678	1643	1609	....	....	....	....	....	....	....	150
175	2711	2604	2509	2424	2347	2277	2213	2154	2100	2049	2002	1958	1917	1878	1841	....	....	....	....	....	....	175
200	3098	2976	2868	2771	2683	2603	2529	2462	2400	2342	2288	2238	2190	2146	2104	2065	....	....	....	....	....	200
225	3485	3348	3227	3117	3018	2928	2846	2770	2700	2634	2574	2517	2464	2314	2368	2323	....	....	....	....	....	225
250	3873	3721	3585	3464	3354	3253	3162	3077	3000	2927	2860	2797	2738	2683	2631	2582	2535	2491	....	....	....	250
275	4260	4093	3944	3810	3689	3579	3478	3385	3300	3220	3146	3077	3012	2951	2894	2840	2789	2740	....	....	....	275
300	4647	4465	4302	4157	4024	3904	3794	3693	3600	3513	3432	3357	3286	3219	3157	3098	3042	2989	2939	....	....	300
325	5034	4837	4661	4503	4360	4230	4110	4001	3900	3801	3718	3636	3560	3488	3420	3356	3296	3238	3184	3132	....	325
350	5422	5209	5019	4849	4695	4555	4427	4309	4200	4098	4004	3916	3834	3756	3683	3614	3459	3487	3429	3373	3320	350
375	....	5581	5378	5196	5031	4880	4743	4616	4500	4391	4290	4196	4107	4024	3946	3873	3803	3737	3674	3614	3557	375
400	....	....	5737	5542	5366	5206	5059	4924	4800	4684	4576	4476	4381	4293	4209	4131	4056	3986	3919	3855	3794	400
425	....	....	....	5889	5702	5531	5375	5232	5100	4977	4862	4755	4655	4561	4473	4389	4310	4235	4164	4096	4031	425
450	....	....	....	....	6037	5857	5692	5540	5400	5269	5148	5635	4929	4829	4736	4647	4563	4484	4409	4337	4269	450
475	....	....	....	....	6372	6182	6208	5848	5700	5562	5434	5315	5203	5098	4999	4905	4817	4733	4654	4578	4506	475
500	....	....	....	....	....	6607	6324	6155	6000	5855	5720	5595	5477	5366	5262	5164	5070	4982	4899	4819	4743	500
525	....	....	....	....	....	6832	6640	6463	6300	6148	6006	5874	5751	5634	5525	5422	5325	5231	5143	5060	4980	525
550	....	....	....	....	....	7158	6957	6771	6600	6440	6292	6154	6024	5903	5788	5680	5578	5480	5388	5301	5217	550
575	....	....	....	....	....	....	7273	7079	6900	6733	6578	6434	6298	6171	6051	5938	5831	5730	5633	5542	5454	575
600	....	....	....	....	....	....	7389	7387	7200	7026	6864	6714	6572	6439	6314	6169	6085	5979	5878	5783	5692	600
625	....	....	....	....	....	....	7905	7694	7500	7319	7151	6993	6846	6708	6577	6455	6338	6228	6123	6024	5929	625
650	....	....	....	....	....	....	8221	8002	7800	7612	7437	7273	7120	6976	6841	6713	6592	6477	6368	6265	6166	650
675	....	....	....	....	....	....	....	8310	8100	7904	7723	7553	7394	7244	7104	6971	6845	6726	6613	6506	6403	675
700	....	....	....	....	....	....	....	8618	8400	8197	8009	7833	7668	7513	7367	7229	7099	6975	6858	6747	6640	700
725	....	....	....	....	....	....	....	8926	8700	8490	8295	8112	7942	7781	7630	7487	7352	7225	7103	6988	6877	725
750	....	....	....	....	....	....	....	9233	9000	8783	8581	8392	8215	8049	7893	7746	7606	7474	7348	7229	7115	750
775	....	....	....	....	....	....	....	9540	9300	9075	8867	8672	8489	8318	8156	8004	7859	7723	7593	7469	7352	775
800	....	....	....	....	....	....	....	....	9600	9368	9153	8952	8763	8586	8419	8262	8113	7972	7838	7710	7589	800
825	....	....	....	....	....	....	....	....	9900	9661	9439	9231	9037	8854	8682	8520	8367	8221	8083	7951	7826	825
850	....	....	....	....	....	....	....	....	10200	9954	9725	9511	9311	9123	8946	8778	8620	8470	8328	8192	8063	850
875	....	....	....	....	....	....	....	....	10500	10246	10011	9791	9585	9391	9209	9037	8874	8719	8573	8433	8300	875
900	....	....	....	....	....	....	....	....	10800	10539	10297	10071	9859	9659	9472	9295	9127	8968	8818	8674	8538	900
925	....	....	....	....	....	....	....	....	11100	10832	10583	10350	10132	9928	9735	9553	9381	9218	9063	8915	8775	925
950	....	....	....	....	....	....	....	....	11400	11125	10869	10630	10406	10196	9998	9811	9634	9467	9308	9156	9012	950
975	....	....	....	....	....	....	....	....	11700	11418	11155	10910	10680	10464	10261	10069	9888	9716	9553	9397	9249	975
1000	....	....	....	....	....	....	....	....	12000	11710	11441	11190	10954	10733	10524	10328	10141	9965	9798	9638	9486	1000

PROPORTIONS OF SELF-SUPPORTING STEEL STACKS.

Horse Power of Stack.	Inside Diameter in Inches.	Height in Feet.	Draft Power in Hydraulic Inches	Tons of Coal Consumed per Hour.	Square Feet of Grate Area.	Diameter at Base.		Diameter at Top of Bell Portion.		Diameter at Top of Stack.		Cubic Yards of Masonry in Foundation.	Fire Brick in Lining.	Common Brick in Lining.	Weight of Stack without Brick Lining, in Tons.
						Feet.	Inches.	Feet.	Inches.	Feet.	Inches.				
100	30	70	0.41	0.25	30	7	6	4	6	3	4	20.6	4500	3500	7.1
115	30	90	0.53	0.28	34	8	6	4	8	3	4	29.5	5000	5000	10.0
130	30	110	0.65	0.32	38	9	0	4	9	3	4	40.0	7000	6500	13.0
125	33	70	0.41	0.31	36	7	4	4	9	3	7	25.0	5000	4500	8.8
140	33	90	0.53	0.35	42	8	3	5	0	3	7	34.5	6000	5500	11.0
160	33	110	0.65	0.40	46	9	3	5	2	3	7	46.0	7500	7000	14.1
150	36	70	0.41	0.37	44	7	7	5	0	3	10	29.6	5000	5000	9.6
175	36	90	0.53	0.44	52	8	6	5	2	3	10	40.0	7000	6000	11.6
200	36	110	0.65	0.50	58	9	6	5	4	3	10	52.5	8000	8000	15.0
250	42	90	0.53	0.62	74	8	9	5	6	4	4	32.1	7500	6000	12.3
275	42	110	0.65	0.68	80	9	3	5	8	4	4	46.0	9000	8000	16.2
300	42	130	0.76	0.75	92	10	9	6	2	4	4	63.3	10500	11000	20.2
350	48	90	0.53	0.87	104	9	2	6	0	4	10	40.0	8000	7000	13.6
375	48	110	0.65	0.93	110	9	9	6	2	4	10	56.0	10000	9000	18.1
400	48	130	0.76	1.00	118	11	4	6	8	4	10	75.7	12000	12000	21.8
430	54	90	0.53	1.07	126	9	8	6	6	5	4	52.6	9000	7500	15.3
470	54	110	0.65	1.17	138	10	8	6	8	5	4	71.9	11000	10000	20.1
510	54	130	0.76	1.27	150	11	9	7	2	5	4	94.7	13000	13000	24.0
580	60	100	0.59	1.45	170	10	6	7	0	5	10	67.3	11000	8000	18.5
675	60	125	0.73	1.62	190	12	0	7	8	5	10	97.0	14000	13000	24.7
700	60	150	0.87	1.75	206	13	2	7	9	5	10	122.0	17000	17000	31.9
700	66	100	0.59	1.75	206	11	0	7	6	6	4	80.0	12000	9000	20.8
800	66	125	0.73	2.00	234	12	6	8	2	6	4	105.0	15000	15000	26.3
900	66	150	0.87	2.25	264	13	8	8	4	6	4	135.0	18000	20000	34.4
950	72	125	0.65	2.37	280	13	0	8	8	6	10	90.0	16000	11000	27.2
1050	72	150	0.87	2.67	310	14	2	8	9	6	10	113.0	20000	19000	38.8
1150	72	175	1.03	2.87	326	15	3	9	0	6	10	155.0	23000	26000	42.3
1150	78	125	0.65	2.87	326	13	6	9	2	7	4	105.0	17000	18000	30.0
1250	78	150	0.87	3.12	368	14	8	9	4	7	4	141.0	21000	24000	38.9
1350	78	175	1.03	3.37	396	15	9	9	6	7	4	185.0	24000	29000	45.0
1400	84	130	0.76	3.50	412	14	0	9	8	7	10	116.0	19000	21000	33.8
1550	84	165	0.97	3.87	456	15	4	9	9	7	10	161.0	25000	28000	44.7
1700	84	200	1.18	4.25	500	16	9	10	0	7	10	217.0	30000	34000	54.3
1800	96	140	0.82	4.50	530	15	4	10	8	8	10	140.0	24000	25000	42.4
2100	96	180	1.06	5.25	620	17	0	10	9	8	10	200.0	30000	31000	52.9
2300	96	220	1.30	5.75	676	18	8	11	0	8	10	273.0	37000	42000	65.3
2400	108	150	0.87	6.00	706	16	9	11	9	9	10	175.0	28000	31000	49.9
2700	108	190	1.12	6.75	794	18	6	12	0	9	10	242.0	35000	37000	62.3
3000	108	240	1.41	7.50	882	20	6	12	2	9	10	325.0	45000	48000	89.0
3000	120	150	0.87	7.50	882	17	9	12	9	10	10	262.0	31000	35000	53.9
3500	120	200	1.18	8.75	1030	19	9	13	0	10	10	304.0	40000	47000	71.0
3900	120	250	1.48	9.75	1148	21	9	13	2	10	10	400.0	51000	60000	94.8
4200	132	200	1.18	10.50	1236	20	9	14	0	11	10	294.0	45000	50000	77.2
4700	132	250	1.48	11.75	1382	23	4	14	4	11	10	400.0	56000	62000	100.5
5200	132	300	1.67	13.00	1528	25	0	14	8	11	10	530.0	67000	75000	140.7

SMOKE STACKS.

APPROXIMATE WEIGHT IN POUNDS OF ONE FOOT OF STACK.

Diameter, inches.	THICKNESS OF MATERIAL.				
	No. 16.	No. 14.	No. 12.	No. 10.	No. 8.
	Weight.	Weight.	Weight.	Weight.	Weight.
10	8	10	13	16	19
12	9	12	14	19	23
14	11	14	16	22	27
16	12	16	20	25	31
18	14	18	23	28	35
20	15	19	25	31	38
22	17	21	28	34	42
24	18	23	30	36	45
26	19	24	32	40	48
28	21	26	35	43	52
30	22	28	37	46	56
32	23	30	39	48	58
34	24	31	41	50	60
36	26	32	43	52	63
38	27	34	44	54	66
40	29	36	47	57	70
42	31	38	49	60	74
44	33	41	54	66	81
48	35	45	59	72	89
54	38	48	64	82	97
60	42	53	71	90	108
66	45	59	77	98	117
72	51	65	86	110	131
78	58	74	98	120	150
84	62	80	105	130	160
96	72	92	130	148	180

SPECIFICATIONS OF VERTICAL TUBULAR BOILERS.

WITH FULL LENGTH TUBES.

Horse power	4	5	6	8	10	12	15	18	21	25	30	34	40	46	50
Diameter of shell, inches	24	24	26	30	30	36	36	36	42	42	42	48	48	48	54
Height of shell, inches	50	60	60	60	72	72	84	96	84	96	108	96	108	120	108
Length of tubes, inches	32	40	38	38	48	48	57	69	54	66	78	63	75	87	75
Number of 2-inch tubes	31	31	37	42	42	57	67	67	98	98	98	138	138	138	174
Thickness of shell, inches	¼	¼	¼	¼	¼	¼	¼	¼	¼	¼	¼	9/32	9/32	9/32	5/16
Thickness of heads, inches	5/16	5/16	5/16	5/16	5/16	⅜	⅜	⅜	⅜	⅜	⅜	7/16	7/16	7/16	7/16
Height of furnace, inches	18	20	22	22	24	24	27	27	30	30	30	33	33	33	33
Diameter of furnace, inches	19	19	21	25	25	30	30	30	36	36	36	42	42	42	48
Heating surface, square feet	50	60	72	96	115	138	185	220	255	306	357	411	483	555	605
Size of lever safety-valve, inches	1	1	1 ¼	1 ¼	1 ½	1 ½	2	2	2	2 ½	2 ½	2 ½	3	3	3 ½
Size of blow-off, inches	¾	¾	1	1	1	1	1 ¼	1 ¼	1 ¼	1 ½	1 ½	1 ½	1 ½	1 ½	1 ½
Weight, complete, lbs.	1100	1200	1350	1630	1850	2000	3000	3350	4000	4350	4700	5700	6100	6500	7500
Diam. of stack required, inches	8	8	10	10	10	13	13	13	16	16	16	18	18	18	20

SPECIFICATIONS OF VERTICAL TUBULAR BOILERS.

WITH SUBMERGED TUBES.

Horse power	5	7	9	12	15	18	21	25	30	34	40	46	50
Diameter of shell, inches	24	30	30	36	36	42	42	42	48	48	48	54	54
Height of shell, inches	72	72	84	84	96	90	96	108	102	108	120	114	120
Length of tubes, inches	30	27	39	39	51	39	45	57	45	51	63	54	60
Number of 2-inch tubes	31	45	45	59	91	91	91	91	134	134	134	174	174
Thickness of shell, inches	¼	¼	¼	¼	¼	¼	¼	¼	9/32	9/32	9/32	5/16	5/16
Thickness of heads, inches	5/16	5/16	5/16	⅜	⅜	⅜	⅜	⅜	7/16	7/16	7/16	7/16	7/16
Height of combustion chamber, inches	18	18	18	18	18	24	24	24	27	27	27	30	30
Height of furnace, inches	24	27	27	27	27	27	27	27	30	30	30	30	30
Diameter of furnace, inches	19	25	25	30	30	36	36	36	42	42	42	48	48
Heating surface, square feet	52	71	94	123	154	184	208	255	300	335	405	455	500
Size of lever safety-valve, inches	1	1 ¼	1 ¼	1 ½	2	2	2	2 ½	2 ½	2 ½	3	3	3 ½
Size of blow-off, inches	¾	1	1	1	1 ¼	1 ¼	1 ¼	1 ½	1 ½	1 ½	1 ½	1 ½	1 ½
Weight complete, lbs	1550	2050	2250	3150	3200	4000	4150	4450	5700	5900	6300	7500	7700
Diameter of stack required, inches	8	10	10	13	13	16	16	16	18	18	18	20	20

Capacities of Boilers for Low Pressure Steam Heating Apparatus.

Boiler Surface, square feet.	Total Direct Radiation, square feet.	Direct Radiation per square foot of Boiler Surface.
40	168	4.20
50	218	4.36
60	272	4.53
80	384	4.80
100	504	5.04
120	626	5.21
140	752	5.37
152	830	5.46
172	962	5.60
194	1114	5.74
211	1232	5.84
252	1522	6.04
292	1816	6.21
295	1840	6.23
347	2240	6.45
399	2642	6.62
421	2820	6.69
482	3321	6.89
541	3818	7.05
580	4247	7.37
720	6210	8.46

The quantities of radiation in the above table are exclusive of all piping.

One square foot of indirect requires the same boiler capacity as 1½ square feet of direct radiation.

TO DETERMINE THE SIZE OF STEAM PIPE MAINS FOR VARYING RADIATION.

For every 100 square feet of radiating surface, allow the area of a one-inch pipe (.7854 square inches).

List of Sizes of Steam Mains.

Radiation, square feet.	One Pipe Work, inches.	Two Pipe Work, inches.
40 to 50	1	¾x ¾
100 “ 125	1¼	1 x ¾
125 “ 250	1½	1¼x1
250 “ 400	2	1½x1¼
400 “ 650	2½	2 x1½
650 “ 900	3	2½x2
900 “ 1250	3½	3 x2½
1250 “ 1600	4	3½x3
1600 “ 2050	4½	4 x3½
2050 “ 2500	5	4½x4
2500 “ 3600	6	5 x4½
3600 “ 5000	7	6 x5
5000 “ 6500	8	7 x6
6500 “ 8100	9	8 x6
8100 “ 10000	10	9 x6

Under ordinary conditions, one square foot of direct radiation surface will heat approximately in:

Bath-room	40	cubic feet.
Living-room	50	" "
Living-room, exposures, ordinary amount of glass	60	" "
Halls	50 to 70	" "
Sleeping rooms	55 " 70	" "
School-rooms	60 " 80	" "
Churches and auditoriums of large cubic contents and with high ceilings	65 " 100	" "
Factories and work-shops	75 " 150	" "

Capacities of Boilers for Hot Water Heating Apparatus.

Boiler Surface, square feet.	Total Direct Radiation, square feet.	Direct Radiation per square foot of boiler surface.
20	110	5.50
30	181	6.03
40	257	6.42
50	338	6.76
60	425	7.08
70	512	7.46
80	603	7.54
90	695	7.72
100	792	7.92
120	991	8.26
140	1198	8.56
159	1400	8.80
199	1842	9.25
225	2142	9.52
279	2788	9.99
323	3332	10.31
372	3976	10.68
453	5065	11.18
517	5938	11.48

The quantities of radiation in the above table are exclusive of all piping.

One square foot of indirect requires the same boiler capacity as 1½ square feet of direct radiation.

CHAPTER IX.

SAFETY VALVES.

A safety valve should have area sufficient for the escape of steam with rapidity to prevent the raising of steam to exceed 10 per cent of pressure allowed and calculations should be from a standard, the maximum water that could be evaporated per pounds of fuel.

Any spring-loaded safety valve constructed so as to give an increased lift by the operation of steam, after being raised from its seat, or any spring-loaded safety valve constructed in any other manner so as to give an effective area equal to that of the aforementioned spring-loaded safety valve, may be used in lieu of the common lever-weighted valve on all boilers on steam vessels, and each spring-loaded valve shall be supplied with a lever that will raise the valve from its seat a distance of not less than that equal to one-eighth of the diameter of the valve opening; but in no case shall any spring-loaded safety valve be used in lieu of the lever-weighted safety valve without first having been approved by the Board of Supervising Inspectors.

The valves shall be so arranged that each boiler shall have at least one separate safety valve, unless the arrangement is such as to preclude the possibility of shutting off the communication of any boiler with the safety valve or valves employed. This arrangement shall also apply to lock-up safety valves when they are employed.

The use of two safety valves may be allowed on any boiler, provided the combined area of such valves is equal to that required by rule for one such valve. Whenever the area of a safety valve, as found by the rule of this section will be greater than that corresponding to 6 inches in diameter, two or more safety valves, the combined area of which shall be equal at least to the area required, must be used.

EXAMPLES:

Boiler pressure = 75 pounds per square inch (gauge).

2 furnaces: Grate surface = 2 × 5 feet 6 inches long × 3 feet wide = 33 square feet.

Water evaporated per pound of coal = 8 pounds.

Coal burned per square foot grate surface per hour = 12½ pounds.

Evaporation per square foot grate surface per hour = 8 × 12½ = 100 lbs.

Hence W = 100 and gauge pressure = 75 pounds.

From table the corresponding value of a is .230 square inches.

Therefore area of safety valve = 33 × .23 = 7.59 square inches.

For which the diameter is 3⅛ inches nearly.

Boiler pressure = 215 pounds.

6 furnaces: Grate surface = 6 × 5 feet 6 inches long × 3 feet 4 inches wide = 110 square feet.

Water evaporated per pound coal = 10 pounds.

Coal burned per square foot grate surface per hour = 30 pounds.

Evaporation per square foot grate surface per hour = 10 × 30 = 300 lbs.

Hence W = 300, gauge pressure = 215, and $a = .270$ (from table).

Therefore area of safety valve = 110 × .270 = 29.7 square inches, which is too large for one valve. Use two.

$$\frac{29.7}{2} = 14.85 \text{ square inches. Diameter} = 4\tfrac{3}{8} \text{ inches.}$$

Rule to determine the area of a safety valve for boiler using oil as fuel or for boilers designed for any evaporation per hour:

Divide the total number of pounds of water evaporated per hour by any number of pounds of water evaporated per square foot of grate surface per hour (W) taken from, and within the limits of, the table. This will give the equivalent number of square feet of grate surface for boiler for estimating the area of valve. Then apply the table as in previous examples.

The areas of all safety valves on boilers contracted for or the construction of which commenced on or after June 1, 1904, shall be determined in accordance with the following formula and table:

EXAMPLE.

Required the area of a safety valve for a boiler using oil as fuel, designed to evaporate 8,000 pounds of water per hour, at 175 pounds gauge pressure.

Make $W = 200$.

$\frac{8,000}{200} = 40$, the equivalent grate surface, in square feet.

For gauge pressure $= 175$ pounds and $W = 200$ from table, $a = .218$ square inch. $.218 \times 40 = 8.72$ square inches, the total area of safety valve required for this boiler, for which the diameter is $3\frac{15}{16}$ square inches nearly.

From which formula the areas required per square foot of grate surface in the following table are found by assuming the different values of W and P.

The figures (*a*) in table multiplied by square feet of grate surface give the area of safety valve or valves required.

When these calculations result in an odd size of safety valve, use next larger standard size.

Table of Area of Safety Valves Required per Square Foot of Grate Surface for Different Pressures and Rates of Evaporation.

P, absolute pressure per square inch.	Gauge pressure per square inch.	These figures represent evaporation in pounds per square foot of grate surface per hour (W) = pounds water evaporated per pound coal × pounds coal burned per square foot grate surface per hour.														
		100	120	140	160	180	200	220	240	260	280	300	320	340	360	380
		The figures below give a, the area in square inches required per square foot of grate surface at the above rate of evaporation.														
65	50	.319	.383	.447	.510	.574	.638	.702	.765	.829	.893	.956	....	....	....	....
70	55	.296	.355	.414	.474	.533	.592	.652	.711	.769	.828	.888	....	....	....	....
75	60	.276	.332	.387	.442	.497	.552	.608	.663	.718	.773	.829	....	....	....	....
80	65	.259	.311	.363	.415	.466	.518	.570	.622	.674	.726	.778	....	....	....	....
85	70	.244	.292	.341	.390	.438	.487	.536	.585	.634	.682	.731	....	....	....	....
90	75	.230	.276	.322	.368	.414	.460	.506	.552	.598	.644	.690	....	....	....	....
95	80	.218	.262	.305	.349	.392	.436	.479	.523	.567	.610	.654	....	....	....	....
100	85	.207	.249	.290	.332	.373	.414	.456	.497	.538	.580	.622	....	....	....	....
105	90	.197	.236	.276	.316	.355	.394	.434	.473	.513	.552	.592	....	....	....	....
110	95	.188	.226	.264	.301	.339	.377	.414	.452	.489	.527	.565	....	....	....	....
115	100	.180	.216	.252	.288	.324	.360	.396	.432	.468	.504	.540	....	....	....	....
120	105	.172	.207	.241	.276	.311	.345	.379	.414	.448	.483	.517	....	....	....	....
125	110	.166	.199	.232	.265	.298	.331	.364	.397	.431	.463	.497	....	....	....	....
130	115	.160	.192	.223	.255	.287	.319	.351	.383	.415	.447	.479	....	....	....	....
135	120	.153	.184	.215	.246	.276	.307	.337	.368	.398	.429	.460	....	....	....	....
140	125	.148	.177	.207	.237	.266	.296	.325	.355	.385	.414	.444	....	....	....	....
145	130	.143	.172	.201	.229	.258	.287	.315	.344	.372	.401	.430	....	....	....	....
150	135	.138	.166	.194	.222	.249	.277	.304	.332	.360	.387	.415	....	....	....	....
155	140	.134	.160	.187	.214	.241	.268	.294	.321	.348	.375	.401	....	....	....	....
160	145	.130	.156	.181	.207	.233	.259	.285	.311	.337	.363	.389	....	....	....	....
165	150	.126	.151	.176	.201	.226	.251	.276	.301	.326	.352	.378	....	....	....	....
170	155	.122	.146	.171	.195	.219	.244	.268	.292	.317	.341	.366	....	....	....	....
175	160	.118	.142	.166	.189	.213	.236	.260	.284	.308	.331	.355	....	....	....	....
180	165	.115	.138	.161	.184	.207	.230	.254	.277	.300	.323	.346	....	....	....	....
185	170	.112	.135	.157	.179	.202	.224	.247	.269	.291	.314	.336	....	....	....	....
190	175	.109	.131	.153	.175	.196	.218	.240	.262	.284	.306	.328	....	....	....	....
195	180	.106	.128	.149	.170	.191	.213	.234	.255	.277	.298	.319	....	....	....	....
200	185	.104	.124	.145	.166	.187	.207	.228	.249	.270	.290	.310	....	....	....	....
205	190	.101	.121	.142	.162	.182	.202	.223	.243	.263	.283	.303	....	....	....	....
210	195	.099	.119	.138	.158	.178	.198	.217	.237	.257	.277	.297	....	....	....	....
215	200	.096	.116	.135	.154	.173	.193	.212	.231	.250	.269	.289	.308	.327	.347	.366
220	205	.094	.113	.132	.151	.170	.189	.208	.226	.245	.264	.283	.302	.321	.340	.358
225	210	.092	.110	.129	.147	.166	.184	.203	.221	.240	.258	.276	.295	.314	.332	.350
230	215	.090	.108	.126	.144	.162	.180	.198	.216	.235	.253	.270	.289	.307	.325	.343
235	220	.088	.106	.124	.141	.159	.176	.194	.212	.229	.247	.264	.282	.300	.318	.336
240	225	.086	.104	.121	.138	.155	.173	.190	.207	.225	.242	.259	.276	.294	.311	.329
245	230	.085	.102	.119	.135	.152	.170	.186	.203	.220	.237	.254	.271	.288	.305	.322
250	235	.083	.100	.117	.133	.149	.167	.183	.199	.216	.233	.249	.266	.282	.299	.315
255	240	.081	.098	.114	.130	.146	.163	.179	.195	.211	.228	.244	.261	.277	.293	.309

Continued on next page.

TABLE OF AREA OF SAFETY VALVES REQUIRED PER SQUARE FOOT OF GRATE SURFACE FOR DIFFERENT PRESSURES AND RATES OF EVAPORATION.

P, absolute pressure per square inch.	Gauge pressure per square inch.	These figures represent evaporation in pounds per square foot of grate surface per hour (W) =pounds water evaporated per pound coal X pounds coal burned per square foot grate surface per hour.														
		100	120	140	160	180	200	220	240	260	280	300	320	340	360	380
		The figures below give a, the area in square inches required per square foot of grate surface at the above rate of evaporation.														
260	245	.080	.096	.112	.128	.144	.160	.176	.192	.208	.224	.240	.255	.271	.287	.303
265	250	.078	.094	.110	.125	.141	.157	.172	.188	.203	.219	.235	.250	.266	.282	.298
270	255	.077	.092	.107	.123	.138	.153	.169	.184	.199	.215	.230	.245	.261	.276	.291
275	260	.075	.090	.105	.121	.136	.151	.166	.181	.196	.211	.226	.241	.256	.271	.286
280	265	.074	.089	.104	.118	.133	.148	.163	.178	.192	.207	.222	.237	.251	.266	.281
285	270	.073	.087	.102	.116	.131	.146	.160	.175	.189	.204	.218	.233	.247	.262	.276
290	275	.072	.086	.100	.114	.129	.143	.157	.172	.186	.200	.214	.228	.242	.257	.271
295	280	.070	.084	.098	.112	.127	.141	.154	.169	.182	.196	.210	.224	.238	.253	.267
300	285	.069	.083	.096	.110	.124	.138	.151	.166	.179	.193	.207	.221	.235	.249	.263
305	290	.068	.082	.095	.109	.122	.136	.149	.163	.177	.190	.204	.217	.231	.245	.258
310	295	.067	.080	.093	.107	.120	.134	.147	.160	.174	.187	.201	.214	.227	.241	.254
315	300	.066	.079	.092	.105	.118	.132	.145	.158	.171	.184	.197	.210	.223	.237	.250

The seats of all safety valves shall have an angle of inclination of 45 degrees to the center line of their axis.

Rule to find area of pop safety valve computed from grate surface, water evaporation and pressure: Multiply constant .2074 by water evaporated per pound of coal per hour and divide by working pressure; this gives area of safety valve per square foot of grate surface. Multiplying this result by total grate surface gives required area of safety valve for furnace grate area.

Formula:

$$\frac{.2074 \times W}{P} = \text{area of safety valve per square foot of grate area}$$

Legend:

C = constant = .2074
W = pounds of water evaporated per square foot of grate surface per hour = 8 pounds of water per pound of coal.
P = absolute pressure plus 15 pounds atmospheric pressure = 90 pounds.
G = grate surface = 30 feet.
Coal burned per square foot of grate per hour = 12.5 pounds.

Example:

lbs. of coal burned per square foot of grate per hour = 12.5
water evaporated = 8
100.0

.2074 = constant
100 = lbs. of water evap. per hour

working pressure = 90)20.7400(.2304 = area of valve per 1 square foot of grate
18 0
2 74
2 70
400
360
40

.2304 = area of valve
30 = total square feet of grate surface
6.9120 = 3″ diameter valve required

REQUIREMENTS IN CONSTRUCTION OF LEVER-SAFETY VALVES.

All the points of bearing on lever must be in the same plane.

The distance of the fulcrum must in no case be less than the diameter of the valve opening.

The length of the lever should not exceed the distance of the fulcrum multiplied by ten.

The width of the bearings of the fulcrum must not be less than three-fourths of 1 inch.

The length of the fulcrum link should not be less than 4 inches.

In all cases the weight must be adjusted on the lever to the pressure of steam allowed in each case by a correct steam gauge attached to the boiler. The weight must then be securely fastened in its position and the lever marked for the purpose of facilitating the replacing of the weight should it be necessary to remove the same, and in no case shall a line or any other device be attached to the lever or weight except in such manner as will enable the engineer to raise the valve from its seat.

When safety valve is blown off always note pressure on gauge; if there is a difference, seek the cause and adjust the gauge or valve until they are as intended.

The lever safety valve, while being very extensively used, is not perfect in action or operation, in not seating itself until pressure has been reduced considerable below point it is set at.

The following rules are used in determining values, viz.: pressure, length of lever and weight of ball.

Rule to find weight of ball when pressure, length of lever and area of valve is known: Multiply pressure in pounds by area of valve in inches and multiply this product by distance of valve center to fulcrum; subtract weight of lever from this product and divide sum by length of lever.

LEGEND:

Va = valve area = 12.5664 = 4″ valve
L = length of lever = 30″
W = weight of lever = 20 lbs.
d = distance valve center to fulcrum = 4″
P = pressure = 100 lbs.

FORMULA:

$$\frac{P \times Va \times d - W}{L} = \text{weight required for ball}$$

Example:

12.5664 = 4″ valve area

100 = pressure

1256.6400

4 = distance valve center to fulcrum

5026.5600

20. = weight of lever

length of lever = 30″) 5006.5600 (166.8853 or 167 lbs. nearly = weight of ball

30

200

180

206

180

265

240

256

240

160

150

100

90

10

Rule to find length of lever when pressure and weight of ball and area of valve is given: Multiply area of valve by pressure in pounds and by distance of center of valve to fulcrum; to this product add weight of lever; divide by weight of ball.

Formula:

$$\frac{Va \times P \times d + W}{Wt} = \text{length of lever}$$

Example

Wt = weight of ball = 166.8853 lbs.

12.5664 = valve area

100 = lbs. pressure

1256.6400

4″ = valve center to fulcrum

5026.5600

20. = weight of lever

weight of ball = 166.8853) 5046.5600 (30 = length of lever

5006 559

40 0010

Rule to find pressure a safety valve will blow off at when weight of ball, length of lever and distance of valve center to fulcrum are known: Multiply weight of ball by length of lever, add weight of lever to this and divide by valve area multiplied by distance of valve center to fulcrum; the quotient will be pressure in pounds.

Formula:

$$\frac{Wt \times L + W}{Va \times d} = \text{pressure}$$

Example:

166.8853 = weight of ball
30″ = length of lever

valve area = 12.5664 5006.5590
distance = 4″ 20. = weight of lever

50.2656) 5026.55900 (99.9 or 100 pounds pressure nearly
4523 904

502 6550
452 3904

50 26460
45 23904

5 02556

Extracts from U. S. Government rules and regulations, prescribed by the Board of Supervising Inspectors, as amended January, 1907:

"No engineer's license shall be issued hereafter or grade increased except upon written examination, which written examination shall be placed on file as records of the office of the inspectors issuing said license. When any person makes application for license it shall be the duty of local inspectors to give the applicant the required examination as soon as practicable."

CLASSIFICATION OF ENGINEERS.

CHIEF.

Chief engineer of ocean steamers.
Chief engineer of condensing lake, bay and sound steamers.
Chief engineer of noncondensing lake, bay and sound steamers.
Chief engineer of condensing river steamers.
Chief engineer of noncondensing river steamers.

Any person holding chief engineer's license shall be permitted to act as first assistant on any steamer of double the tonnage of same class named in said chief's license.

Engineers of all classifications may be allowed to pursue their profession upon all waters of the United States in the class for which they are licensed.

FIRST ASSISTANT.

First assistant engineer of ocean steamers.

First assistant engineer of condensing lake, bay and sound steamers.

First assistant engineer of noncondensing lake, bay and sound steamers.

First assistant engineer of condensing river steamers.

First assistant engineer of noncondensing river steamers.

Engineers of lake, bay and sound steamers, who have actually performed the duties of engineer for a period of three years, shall be entitled to examination for engineer of ocean steamers, applicant to be examined in the use of salt water, method employed in regulating the density of the water in boilers, the application of the hydrometer in determining the density of sea water and the principle of constructing the instrument; and shall be granted such grade as the inspectors having jurisdiction on the Great Lakes and seaboard may find him competent to fill.

Any assistant engineer of steamers of 1,500 gross tons and over, having had actual service in that position for one year, may, if the local inspectors, in their judgment, deem it advisable, have his license indorsed to act as chief engineer on lake, bay, sound, or river steamers of 750 gross tons or under.

Any person having had a first assistant engineer's license for two years and having had two years' experience as second assistant engineer, shall be eligible for examination for chief engineer's license.

SECOND ASSISTANT.

Second assistant engineer of ocean steamers.

Second assistant engineer of condensing lake, bay and sound steamers.

Second assistant engineer of noncondensing lake, bay and sound steamers.

Second assistant engineer of condensing river steamers.

Any person having had a second assistant engineer's license for two years and having had two years' experience as third assistant engineer, shall be eligible for examination for first assistant engineer's license.

THIRD ASSISTANT.

Third assistant engineer of ocean steamers.

Third assistant engineer of condensing lake, bay and sound steamers.

First, second, and third assistant engineers may act as such on any steamer of the grade of which they hold license, or as such assistant engineer on any steamer of a lower grade than those to which they hold a license.

Any person having a third assistant engineer's license for two years and having had two years' experience as oiler or water tender since receiving said license, shall be eligible for examination for second assistant engineer's license.

Inspectors may designate upon the certificate of any chief or assistant engineer the tonnage of the vessel on which he may act.

Any assistant engineer may act as engineer in charge on steamers of 100 tons and under. In all cases where an assistant engineer is permitted to act as engineer in charge, the inspectors shall so state on the face of his certificate of license without further examination.

It shall be the duty of an engineer when he assumes charge of the boilers and machinery of a steamer to forthwith thoroughly examine the same and if he finds any part thereof in bad condition, caused by neglect or inattention on the part of his predecessor, he shall immediately report the facts to the master, owner, or agent and to the local inspectors of the district, who shall thereupon investigate the matter and if the former engineer has been culpably derelict of his duty, they shall suspend or revoke his license.

Before making general repairs to a boiler of a steam vessel the engineer in charge of such steamer shall report, in writing, the nature of such repairs to the local inspector of the district wherein such repairs are to be made.

And it shall be the duty of all engineers when an accident occurs to the boilers or machinery in their charge tending to render the further use of such boilers or machinery unsafe until repairs are made, or when, by reason of ordinary wear, such boilers or machinery have become so unsafe, to report the same to the local inspectors immediately upon the arrival of the vessel at the first port reached subsequent to the accident, or after the discovery of such unsafe condition by said engineer.

Whenever a steamer meets with an accident involving loss of life or damage to property, it shall be the duty of the licensed officers of any such steamer to report the same in writing and in person without delay to the nearest board: *Provided,* That when from distance it may be inconvenient to report in person it may be done in writing only and the report sworn to before any person authorized to administer oaths.

No person shall receive an original license as engineer or assistant engineer (except for special license on small pleasure steamers and ferryboats of 10 tons and under, sawmill boats, pile drivers, boats exclusively engaged as fishing boats and other similar small vessels) who has not served at least three years in the engineer's department of a steam vessel, a portion of which experience must have been obtained within the three years next preceding the application.

Provided, That any person who has served three years as apprentice to the machinist trade in a marine, stationary, or locomotive engine works, and any person who has served for a period of not less than three years as a locomotive or stationary engineer, and any person graduated as a mechanical engineer from a duly recognized school of technology, may be licensed to serve as an engineer of steam vessels after having had not less than one year's experience in the engine department of steam vessels, a portion of which experience must have been obtained within the three years preceding his application; which fact must be verified by the certificate, in writing, of the licensed engineer or master under whom the applicant has served, said certificate to be filed with the application of the candidate; and no person shall receive license as above, except for special license, who is not able to determine the weight necessary to be placed on the lever of a safety valve (the diameter of valve, length of lever, distance from center of valve to fulcrum, weight of lever and weight of valve and stem being known) to withstand any given pressure of steam in a boiler, or who is not able to figure and determine the strain brought on the braces of a boiler with a given pressure of steam, the position and distance apart of braces being known, such knowledge to be determined by an examination in writing, and the report of examination filed with the application in the office of the local inspectors, and no engineer or assistant engineer now holding a license shall have the grade of the same raised without possessing the above qualifications. No original license shall be granted any engineer or assistant engineer who can not read and write and does not understand the plain rules of arithmetic.

Any person may be licensed as engineer (on Form 2130⅞) [New Form 880] on vessels propelled by gas, fluid, naphtha, or electric motors, of 15 gross tons or over, engaged in commerce, if in the judgment of the inspectors, after due examination in writing, he be found duly qualified to take charge of the machinery of vessels so propelled.

Any person holding a license as engineer of steam vessels, desiring to act as engineer of motor vessels, must appear before a board of local inspectors for examination as to his knowledge of the machinery of such motor vessels, and if found qualified shall be licensed as engineer of motor vessels. Form 878, special license to engineers, shall be issued only to engineers in charge of vessels of 10 tons and under. All other licenses to engineers shall be issued on Forms 876 and 877, according to grades specified in this section.

INSPECTING BOILERS.

The necessity of care in inspecting steam boilers is apparent when the amount of power stored up while the boiler is in commission is known—as an illustration: a common sized boiler 60″ × 16′ has 38923square inches, and carrying a pressure of 100 pounds, has 1946 tons of energy. With strains of expansion and contraction not equal all over but varying, and limits to the extreme — (i. e.) the temperature of fire in furnace to that of parts furthest from it, and furthermore when considering that 85% of the boiler is concealed — this by design or principle of installation — the necessity of vigilance can be realized, especially when the causes of failure and defects are numerous, viz.:

- Material,
- Design,
- Construction,
- Appliances,
- Fuel,
- Feed Water,
- Settings, and
- Management and Care.

The hydrostatic test is a method not very satisfactory but often necessary when access to parts is impossible, or where a design of boiler has flat surface and notice of bulging or elongation must be noted before and after pressure; it is necessary when notes of bracing are to be taken and when there are any minor defects such as leaks at rivets or caulking so they can be remedied before more serious results follow. When a hydrostatic test is made of boilers that are accessible, braces and such joints that are weaker than the original plates' tensile strength, must be inspected carefully for any distortions or leaks due to riveting, welds or defective flanges—and hidden defects may give evidence of their presence.

INSPECTIONS.

There are internal and external inspections, both essential in determining the boiler's safety; for to determine the safe working pressures, an internal inspection is absolutely necessary.

The conditions for this latter examination are as follows: The boiler must be cool, water out (this is supposing the boiler has been in commission), ashes and soot removed, the mud only washed out of boiler (it is well to avoid excessive pump pressure when washing out until inspection is made), this so as not to destroy or wash off any evidence of leaks that might be at points inaccessible to view from the outside, but would be in evidence at a point inside, for deposits or precipitation in suspension would collect at point where leakage was, thus giving evidence of leaks that could not be seen from outside; this, of course, applies to boilers of size and design accessible. A thorough examination must be made of all parts of boiler accessible; sounding plates where possible over fire or in furnace; and parts where not possible over fire or in furnace to see or sound, symptoms that would deceive the eye, can, at times, be detected by the sense of touch; flanges and junction of pipes at boilers must be examined, for threads are an initial fracture, and by the pipe or boiler expanding much undue strain results and often causes breaking off of pipe. The tubes at rear and front heads being thin, are often a source of annoyance; examine seams and rivets for leakage and cracks; see that openings to outlets are free from obstructions; sound braces; examine flanges, seams and rivets internally, the condition as to incrustation, corrosion, pitting, and when in doubt, give a hydrostatic test; this would reveal any weakness and leaks impossible to see, or defects developed by closing down the boiler, resulting in contraction. An inspection and sounding of braces should follow the hydrostatic test. Stay bolts must be sounded when type of boiler is braced by them.

The first thing, look at or for the water level, then the steam pressure; view the furnace, tube sheets, crown sheets and sides in internal fired boilers and bottom and furnace walls in external fired boilers, looking at back head from rear doors for leakage; (the doors at rear end were designed for access to back head and to view when the boilers were in commission) the blow-off, and as much of the bottom as possible; brick work; examine the blow off pipe; if it is hot outside of valve it is evidence of leakage at valve (this unless some drips or other steam outlets are connected into same blow-off pipe). A leaky blow-off valve is a source of danger, waste of fuel and energy; the danger lies in the fact that the precipitates will collect at a point where there is leakage and as the blow-off pipe part of it is exposed to heat one can realize there is danger by burning of blow-off pipe.

The outside of brick settings should be examined for fissures or cracks caused by expanding of boiler and excessive heat. These cracks admit cold air, quantity governed by size and draft. These are the cause of much loss of energy, certainly a waste of fuel, and at expense of life or boiler.

Examine the feed appliances; test the steam gauge; following this up by firing up of boiler to point of safe working pressure, then the setting of valve if necessary. When the steam gauge is taken off, blow out the pipe and be sure it is clear, for oftentimes these pipes are neglected, and if there is a syphon or trap for condensation, this latter will generate corrosion and liable to stop up stop-cock, if not, the pipe.

Management and care must be considered, as we have measured the safe working pressure by design, material and construction. The best of man's work would be trivial in the hands of an ignorant boiler attendant, and the only factor for safety in such cases would be to *keep the boilers cold.* Again, the inspector must bear in mind that those in power to hire attendants are oftentimes those whose knowledge of the requirements necessary, for men and duties is very limited.

Fuel should be considered by the inspector, for in these days of coal as fuel it must be remembered that the more sulphur in the fuel, the quicker crystallization will develop in the plates.

Quality of feed water, its temperature and point of admission should be looked after; for these are elements that will, in a measure, give evidence of what one expects.

POINTS TO CONSIDER WHEN INSPECTING BOILERS.

Evidence of excessive firing; piping of boilers for best effect to allow for expansion; avoid rigidity; pipe of sufficient strength for high pressures; deterioration from leakage; corrosion from sulphuric action—soot and moisture develops sulphuric acid. Remember that 75 per cent of the boiler is concealed either by the design or settings and much depends on viewing and examining the minimum portion; that a large amount of energy is stored up in the boiler when in commission; for instance, a boiler 60″ × 16′ at 100 pounds pressure has approximately 1946 tons of energy stored in it. This suggests reasons for thought. There is lamination or blisters and bagging of plates to look for, or to be expected. See that water columns are properly connected and convenient to try at all times; that the safety valve is of sufficient size and operative; that blow-pipes are of proper size and protected; that the feed water appliances are ample and more than one to feed boiler; that the feed water enters at a suitable place; that the check and stop valves are connected and placed a reasonable distance from boiler; that the boiler (if externally fired) is properly set for heat distribution; that the grates are not too close to the boiler (bottom), for space is necessary for combustion and conductivity of heat. Do not for-get that it is a human being who is in charge of the boiler and that it is human to err. This will impress the inspector that if the man in charge knew as much as he does, the inspector's services would not be necessary. It also qualifies the old adage, "No man is the best judge of his own work or actions."

THE SAFE WORKING PRESSURE.

Years ago the Lloyds of Europe adopted a rule to govern the safe working by pressure, viz.: One sixth of the tensile strength of plate, multiplied by thickness of the plate, and divided by the radius; and for years this rule was used universally. It was the supposition that the plate and rivet strength would be near equal and construction the best, 20 per cent was added for double riveted longitudinal seam. At that time low pressures were the rule, consequently security or safety was reasonably expected; but when other factors came to be considered, different types of engines that required higher pressures and fuel became a prime factor, along with space, the demand for higher pressure became apparent and something more than the old time design and construction of boilers had to be considered. The weakest point had to be strengthened, necessitating butt joints, drilled holes, modern flanging, braces and bracing, larger plates and less joints, abandonment of cast iron for man holes and openings. Boiler making tools and machinery had to keep pace, thus the advancement made in the craft necessitates some more definite rules to govern us in the allowing of a safe working pressure. The factor of six, as formerly used, was, no doubt, little enough when iron plates, short and narrow, were used; chipping done by hand, i. e., the grooving by same; punched holes; the drift pin and designing of seams. Thus it was absolutely necessary for a large factor of safety; but as stated, boiler construction to-day is modern and complies to the demand for high pressures. We are too advanced to use such a large safety factor as 6. It is true there are the extremes, but there are things that must be considered in this matter of safety factor, viz., design; tensile strength; thickness of plate; diameter of hole; diameter and pitch of rivet; shearing strength of rivet; diameter of boiler; bracing; lowest percentage of seam. It might be carried further to be more definite, by considering the boiler's use; if boiler would be forced; if loads would vary; type of engine; if the boiler would be used for power or heating only.

It would not be consistent to lay down any specified rule to govern all cases. It may be that the boiler would deteriorate faster in one location than another. This, of course, would be a local consideration, but in these days of modern ideas, designs and construction, a factor of four would be ample to cover all differences in construction and material.

Prepare for inspection by having ashes and deposits removed from under boiler and ash pits, tubes cleaned and soot removed.

Allow boiler and setting time to cool off gradually, open gauge-cocks before letting water run out. Leave dampers open and furnace door closed.

Wash boiler out and have same as dry as possible.

Take steam gauges down for testing.

Steam gauges should be connected with a union between stop cock and gauge, so that the latter can be taken off syphon or pipe without disturbing threads that would alter position when connecting gauge again. It is advisable, when having gauges tested, to raise steam and note point of blowing off, and adjust safety valve if necessary.

If a hydrostatic test is to be made have pump and piping connected and the hydrostatic test applied to a pressure equal to the proportions of 150 pounds to 100 pounds working pressure.

The U. S. Government makes annual inspections and tests and all mandates are carried out to the letter.

Testing of plates, piping and material must fill all requirements, or condemnation or rejection follows. Boilers and appliances must be approved before installing and put into commission.

Some of the requirements are as follows:

CAST STEEL AND CAST IRON.

No cast steel or cast iron subject to pressure shall be allowed to be used in boilers or the pipes connected thereto, except as described as follows:

Cast iron or cast steel may be used in the construction of manhole and hand-hole plates, valves and cocks, water columns, flanges, saddles, ells, tees, crosses or manifolds when such flanges, saddles, ells, tees, crosses, valves and cocks, or manifolds are bolted or riveted directly to the boiler and the valves or cocks; also, casings of slip joints in pipes: *Provided, however,* that the material shall be of the best grade and of suitable thickness and uniform section for the pressure allowed on boilers.

FEED WATER.

The feed water shall not be admitted into any boiler at a temperature less than 100° F., and no marine boiler shall be used without having proper auxiliary appliances for supplying said boilers with water in addition to the usual mode employed.

NAME PLATES.

There shall be fastened to each boiler a plate containing the name of the manufacturer of the material, the place where manufactured, the tensile strength, the name of the builder of the boiler, when and where built.

FUSIBLE PLUGS.

Every boiler, other than boilers of the water-tube type, shall have at least one fusible plug as described below. Plugs shall be made of a bronze casing filled with good Banca tin from end to end. The manufacturers of fusible plugs shall stamp their name or initials thereon for identification and shall file with the local inspectors a certificate, duly sworn to, that such plugs are filled with Banca tin.

Fusible plugs, except as otherwise provided, shall have an external diameter of not less than three-fourths of an inch pipe tap, and the Banca tin shall be at least one-half of an inch in diameter at the smallest end and shall have a larger diameter at the center or at the opposite end of the plug.

Fusible plugs, when used in the tubes of upright boilers, shall have an external diameter of not less than three-eighths of an inch pipe tap, and the Banca tin shall be at least one-fourth of an inch in diameter at the smaller end and shall have a greater diameter at the opposite end of the plug: *Provided, however,* that all plugs used in boilers carrying a steam pressure exceeding 150 pounds to the square inch may be reduced at the smaller end of the Banca tin to five-sixteenths of an inch in diameter.

Externally heated cylindrical boilers, with flues, shall have one plug inserted in one flue and also one plug inserted in shell of each boiler, immediately below the fire line and not less than 4 feet from the front end: *Provided, however,* that when such flues are not more than 6 inches in diameter a fusible plug of not less diameter than three-eighths-inch pipe tap may be used in such flues.

Other shell boilers, except especially provided for, shall have one plug inserted in the crown sheet of the back connection.

Vertical tubular boilers shall have one plug inserted in one of the tubes at least 2 inches below the lowest gauge cock, but in boilers having a cone top the plug shall be inserted in the upper tube sheet.

All plugs shall be inserted so that the small end of the Banca tin shall be exposed to the fire.

It shall be the duty of the inspector at each annual inspection to see that the plugs are in good condition.

GAUGE COCKS AND WATER GLASS.

All boilers shall be supplied with one reliable water gauge and three gauge cocks in each boiler: *Provided,* that when the gauge glass and gauge cocks are connected to the boilers by a water column there must be an additional gauge cock inserted in the head or shell of boiler. The lower gauge cock in boilers more than 48 inches in diameter shall not be less than 4 inches from the top of the flues or tubes. In boilers less than 48 inches in diameter the lower gauge cock shall not be less than 2½ inches above the top of the flues or tubes. A gauge glass shall be considered a reliable water gauge, and a float such as used on western river steamers shall be considered on such boilers as a reliable water gauge.

In vertical boilers or boilers of the water-tube type the location of the lowest gauge cock shall be determined by the local inspectors.

Boilers known as flash boilers constructed of a continuous coil of pipe or series of coils of pipes under three-fourths inch in diameter, whose construction has been approved by the Board of Supervising Inspectors, shall not be required to be supplied with gauge cocks or low-water gauges.

DRILLING TO DETERMINE THICKNESS.

Any boiler ten years old or more shall, at the first annual inspection thereafter, be drilled at points near the water line and at bottom of shell of boiler, or such other points as the local inspectors may direct, to determine the thickness of such material at those points; and the steam pressures allowed shall be governed by such ascertained thickness and the general condition of the boiler.

HYDROSTATIC PRESSURE.

The hydrostatic pressure applied must be in the proportion of 150 pounds to the square inch to 100 pounds to the square inch of the steam pressure allowed and the inspector, after applying the hydrostatic test, must thoroughly examine every part of the boiler.

In applying the hydrostatic test to boilers with a steam chimney, the test gauge should be applied to the water line of such boilers.

All coil and pipe boilers hereafter made, when such boiler is completed and ready for inspection, must be subjected at the first inspection to a hydrostatic pressure double that of the steam pressure allowed in the certificate of inspection.

The use of malleable-iron or cast-steel manifolds, tees, return bends or elbows in the construction of pipe generators shall be allowed and the pressure of steam shall not be restricted to less than one-half the hydrostatic pressure applied to pipe generators unless a weakness should develop under such test as would render it unsafe in the judgment of the inspector making such inspection.

DRUMS AND HEADS.

All drums attached to coil, pipe, sectional or water-tube boilers not already in use or actually contracted for, to be built for use on a steam vessel and its building commenced at or before the date of the approval of this rule, shall be required to have the heads of wrought iron or steel or cast steel flanged and substantially riveted to the drums or secured by bolts and nuts of equal strength with rivets, in all cases where the diameters of such drums exceed 6 inches.

Drums and water cylinders constructed with a bumped head at each or either end, (any opening in the shell or heads to be reinforced as required by the rules of the Board, the circumferential and horizontal seams to be welded and properly annealed after such welding is completed), when tested with a hydrostatic pressure at least double the amount of the steam pressure allowed may be used for marine purposes.

PIPES.

COPPER.

All copper pipe subject to pressure shall be flanged over or outward to a depth of not less than twice the thickness of the material in the pipe and such flanging shall be made to a radius not to exceed the thickness of the pipe. On boilers whose construction was commenced after June 30, 1905, no bend will be allowed in copper pipe of which the radius is less than one and one-half times the diameter of the pipe and such pipe must be so led and flanges so placed that they may be readily taken down if required. Such pipes must be protected by iron casings when run through coal bunkers and must be clear of the coal chutes.

The flanges of all copper steam pipes over 3 inches in diameter shall be made of brass or bronze composition, forged iron or steel, or open-hearth steel castings and shall be securely brazed or riveted to the pipe: *Provided, however,* that when such pipes are properly formed with a taper through the flange, such taper being fully reinforced, the riveting or brazing may be dispensed with: *And provided, also,* that when the pipe has been expanded by proper and capable machinery into grooved flanges and the pipe flared out at the ends to an angle of approximately 20°, said angle to be taken in the direction of the length of the pipe and having a depth of flare equal to at least one and one-half times the thickness of the material in the pipe, said riveting or brazing may be dispensed with. Where copper pipes are expanded into or riveted to flanges it will be necessary for the pipes with their flanges attached to withstand a hydrostatic pressure of two and one-half times the boiler pressure.

Flanges must be of sufficient thickness and must be fitted with such number of good and substantial bolts to make the joints at least equal in strength to all other parts of the pipe.

Any form of joint that will add to the safety or increase the strength of flange and pipe connections over those provided for by this rule, will be allowed on any and all classes of steam pipe.

WATER TUBE AND COIL BOILERS.

Blue prints or drawings of coil boilers and of other boilers, with their specifications, submitted to the Board of Supervising Inspectors for approval under section 4429, Revised Statutes of the United States, must be in duplicate before action thereon will be taken by the Board, with a view of approving the same; one set to be filed with the records of the Board of Supervising Inspectors and the other with the records of the supervising inspector of the district where the manufacturer of the boiler is located.

Rule to find the working pressure allowable on cylindrical shells of water tube or coil boilers, when such shells have a row or rows of pipes or tubes inserted therein: From pitch of holes subtract diameter of pipe, then multiply by thickness of plate and one-sixth of tensile strength. Divide this product by pitch of holes multiplied by radius.

FORMULA:

$$\frac{p - d \times T \times 1/6 \text{ of } TS}{p \times R} = \text{pressure}$$

LEGEND:

p = pitch = 2″
d = diameter of pipe = 1″
T = thickness of plate = ½″ = .5
TS = tensile strength = 60000
R = radius = 10″

EXAMPLE:

```
                     2 =pitch
                     1 =diameter of pipe
                    --
                     1
                    .5 =thickness of plate
                    --
 pitch =  2″        .5
radius = 10″     10000 = one-sixth of tensile strength of
         --      -----                              plate
         20)    5000.0(250 pounds pressure allowed
                40
                ---
                100
                100
                ---
                  0
```

CHAPTER X.

FEED WATER HEATING AND PURIFICATION.

While boiler designing, construction and setting have received the thought and attention of many prominent specialists of this age, this for security against the high pressures necessary to meet the demands of modern engines and that factor, fuel, it is apparent even to the layman that the feed water for steam boilers must be a factor worthy of much consideration, for it means life of boiler and efficiency of same — this under varying conditions even to those who have free fuel and best of water. Various appliances and methods are employed to obtain the best possible results from feed water, for the latter is one of the primaries for disaster and expense in operation. Many well designed and well constructed boilers have been condemned on this account. Reputations that have been built on years of experience and study have been affected by local influences — bad feed water.

Instances can be cited where boilers designed and made by the most progressive boiler makers have been condemned and only material and construction given by the operators as a cause for failures or reduced condition. Feed water is the initial factor in the steam plant. To install the best designed and constructed boiler from the best of material and subject the same to bad feed water, failure of seams or plate are the results expected.

In some localities incrustation and deposits from water are unknown — this where matter which is soluble in land strata are absent — but these locations are very few to the major part of this country. Hence the necessity for an appliance — a vital adjunct to the steam plant — i. e., a feed water purifier.

Many and varied are the appliances now used for this purpose; it would seem that each one has its advocates and no doubt its niche, or suitable place. They all aim to obtain the best possible results, but many fail to accomplish the maximum effect.

A brief description of types mostly in use may be interesting or at least give some food for thought. Possibly future discussions may change views and show that present convictions are wrong. Such subjects are almost inexhaustible and when analyzed they can be made subjects of much merit and of great interest to those whose lives are devoted to steam engineering. For instance, analyzing the boiler, we find:

Material.
Design.
Construction.
Settings.
Appliances.
Management and care, and
Feed water.

It is the latter which I will attempt to digest, not in material value order, or on personal judgment, but as they suggest themselves to the mind when reviewing this subject. A brief description of types in use are:

1. Auxiliary pipes.
2. Water backs.
3. Pipes in uptakes.
4. Closed heaters.
5. Boxes or receptacles in boilers.
6. Live steam heaters.
7. Open heaters.

There is no question but that any or all of these types have some merit in some particular place or under some conditions.

I will take them up in individual order and try to point out their degree of usefulness, or advantages, one over the other.

In order to obtain the best values, we must look for requirements, they must be known; then put them in valued order.

The heater and purifier must have some of these requirements. There is much variance with each type, no two alike, when units of measurement are taken. Quotations of prices are based on individual units of measurement and, like the different types of boilers, are rated on a given quantity of heating surface ranging from 6 to 15 square feet—this irrespective of plate thickness, grate surface, fuel or draft. It is the same with the heaters and so-called purifiers.

1. AUXILIARY PIPES.

These are connected to boiler, water and steam connections. They simply make additional heating surface and have very little merit otherwise. They are not to be recommended for either efficiency, safety or economy. They are short-lived, a menace to security, subject to incrustation and fracture due to expansion and contraction; impossible to clean, making, oftentimes, long and serious delays. It is like courting disaster to apply these to a boiler.

2. WATER BACKS.

These are usually placed back of boiler, top of setting, or in front of or at sides of furnace and shapes are either cylindrical or flat. They are supposed to act in a dual capacity — feed water heater and form an arch or a part of the furnace. It cannot be said that there is any fuel economy. They are a part of the boiler and absorb furnace heat. They have boiler pressure, and are no prevention against solids in suspension going into boiler. They often become incrustated, necessitating repairs, and when one considers the difference in temperature in such a short space, between parts exposed to fire and boiler room, expectations can be realized. The tempering of water by heat before going to boiler, as in case of injectors, is the only point of merit they have. The cylinder type may have some advantages — strength of form and being more accessible to clean. The flat type offers little in that respect. The latter are more costly, owing to the flange and the bracing by stay bolts. Again, either type has the disadvantage of adding weight on settings or walls. The latter are expensive items in keeping up the boiler plant.

3. PIPES IN UPTAKE.

This application for heating feed water has sometimes primary benefits in the way of economy, due to absorbing heat from escaping gases. But this is largely a guess and it is a question if they are often or long economical, for the heat escaping up the stack or uptake is a large factor, in fact very necessary and essential when natural draft is depended on, and supply limited; for to reduce this temperature means less oxygen to fuel.

In some places, and under some conditions, there may be some economy, but in the average plant, none. Incrustated pipes, solids in suspension forced into boilers, fractures, delays in removal or cleaning, can be expected. This type cannot be considered a profitable investment even in plants where induced draft is used, unless water is purified before going through same.

4. CLOSED HEATERS.

Water or steam tubes or pipes, return bends, corrugated or straight, coils, with and without setting chambers.

These appliances are made in varying forms, the aim being to obtain heat from exhaust steam in non-condensing plants, but it is futile to expect anything like purification of feed water from this type. No matter what design they are, their value is limited to that of heating to some extent, the feed water then at a low temperature. They have pressure in excess of the boiler, this owing to the necessity of lifting check valve or overcoming weight of water and pressure in boiler. The exhaust steam temperature must be conducted through plate pipes, coils or tubes, there being no chance for precipitation other than light solids, such as magnesia—this owing to lack of temperature imparted by exhaust and the existing pressure in heater, even with back pressure on engine, for to precipitate other solids the temperature must be increased with pressure obtained in heater. For instance, if pressure was 100 pounds, the temperature necessary would be 338 F., but at atmospheric pressure it would be 212 F. Then what chances could there be even with back pressure when the heat must be conducted through plate? Should light solids be precipitated these would be forced into boiler. Again, this type or class of heater is hard if not almost impossible to clean. Thus, should any solids be in suspension and collect, when the attempt is made to clean exhaust pipes must be disconnected and those of water or steam tube type are difficult for access.

Those with a so-called setting chamber have very little effect from settling, for these have a continuous circulation when feed water passes through. Hence settling is impossible when pumps would be stopped; then the only amount of settling would be equal to that which volume of water at that time would hold.

One argument used in its favor, as heard, is that "only one pump is required." This apparently is enough to convince the layman that to select this type is wise. Some of these closed heaters may have individual merit. For instance, the return bend expands on one end — that is, it is free to do so. Then the corrugated tube has additional heating surface and prevents leaking at ends, expansion and contraction being taken up by the corrugations. But in this form of heater, condensation is usually lost with its purity and heat units. This heater is fast being relegated to one place in the power plant, and that place is the condensing one. Its position being between the engine, cylinder and injection water. Its value, besides giving some heat, is to prevent condensation of steam in cylinder by the injection water.

5. THE BOX OR RECEPTACLE THAT IS PLACED IN THE BOILER.

This idea of a feed water heater and purifier is not new. It is old and has been tried and found wanting. These may be obtained in any shape, or to be put or placed in any part of boiler, on top of tubes or under same. That does not prevent results from being the same. Though feeding impure water into a box having holes or slots, it is a fact the water must find its level, must flow to that point where steam globules are formed and then ascend into space to diffuse. Precipitation does not occur at the instant of contact with heat. Even if it did these receptacles are only settling pans and the perforations are limited — this to confine water inside as long as possible and to aid precipitation. Danger is courted, for should those openings become stopped up danger from low water is the result. If these boxes are open then the solids will find their way to all parts of boiler — this through circulation. These boxes obstruct steam passages, retard circulation and make internal inspections impossible. The price involved in these would be far better invested in something to prevent solids from going into boiler or in aiding to purify feed water before going into boilers, this being done now in modern plants.

6. THE LIVE STEAM HEATER AND PURIFIER.

The live steam purifier, like all other contrivances and appliances for bettering the condition of boilers and increasing efficiency and reducing the hazard and risk in steam boilers, has its advocates.

Much has been claimed for it. Like preceding types it no doubt has some features that might at least appear commendable. But, however, claims are one thing, effects, results and investments are others. The name is somewhat misleading. Its value ceases as an investment when cost and maintenance are experienced. While admitting that it would have one factor, that of precipitation of solids that were held in solution by boiler pressure temperature, this does not alone insure purity of water or establish it as a purifier, for two results are necessary for purification of feed water — viz.: precipitation and filtering.

The pans used are settling surfaces for some of the solids that will settle, but much goes into boiler through gravity circulation. The live steam heaters are selected for only one action — precipitation — and this at the expense of condensation, they being in a position at a considerable distance from water line to grate surface. Some argue that if only some of the solids are prevented from going into boiler, the value of the live steam heater must be considered with fuel saving and efficiency gained, this offsetting the condensation. But there are points of disadvantages. The added hazard, being subjected to the full boiler pressure, has additional energy stored in it. They are placed much higher than boiler water line, access to clean difficult, involve much expense for installation, special frame support and floor. When points of advantages are taken into consideration and weighed with the disadvantages, care should be taken when selection of a feed water purifier is to be made.

7. THE OPEN HEATER AND PURIFIER.

Feed water purification is a possibility and this is when open type of feed water heater and purifier is used, (this is only when care and reason are exercised in selection), and this can be done with minimum loss of furnace heat. It is practically the solution solved when the elements and requirements are adjusted and proportions are proper, viz., time and temperature.

Where a lack of temperature fails time must be increased. Additional body of water will represent time.

This appliance is open to the atmosphere. The feed water supply comes in contact with the exhaust steam or steam used for temperature necessary for precipitation. It will produce a partial vacuum on engine when exhaust steam is used. Precipitation occurs at lowest possible temperature, 15 to 20 per cent of pure water being gained by condensation. There are some open heaters that are so constructed that precipitation is expected at instant of contact of steam and water. Others have so limited a supply of water that no time for action is allowed. In some cases a few strokes of the pump takes all the water out. Others, while they have a copious supply of water, the filtering material is such that it separates, thus leaving water with its solids in suspension free to go to pump, then to the boiler. Others, again, have no facilities for cleaning the filter, unless at expense of closing down or putting cold water into boilers. Most of these are simply receivers, heaters or condensers. They cannot be termed feed water purifiers.

A few suggestions on selection may be in order. Conditions must be observed. First, quality of water to be used; this will determine the filtering surface, but the main requirements are: high temperature, large body of water, large amount of filtering surface, easy to clean.

The two elements, time and temperature, are necessary.

Points to be considered in selecting slow filtering — filter accessible to clean when in use, filtering material and adjustment of same against derangement.

When filtering is operative, deposits will collect on filtering material, thus the necessity of some way to clean off same at any time.

There is the greatest of economy in heating feed water by exhaust steam, even when the latter is used for heating purposes. In this age we are resorting to chemistry as a positive aid in water purification.

FEED WATER HEATERS—KENT

Percentage of saving for each degree of increase in temperature of feed-water heated by waste steam

Initial Temperature of Feed	Pressure of Steam in Boiler, pounds per square inch above Atmosphere										
	0	20	40	60	80	100	120	140	160	180	200
32°	.0872	.0861	.0855	.0851	.0847	.0844	.0841	.0839	.0837	.0835	.0833
40	.0878	.0867	.0861	.0856	.0853	.0850	.0847	.0845	.0843	.0841	.0839
50	.0886	.0875	.0868	.0864	.0860	.0857	.0854	.0852	.0850	.0848	.0846
60	.0894	.0883	.0876	.0872	.0867	.0864	.0862	.0859	.0856	.0855	.0853
70	.0902	.0890	.0884	.0879	.0875	.0872	.0869	.0867	.0864	.0862	.0860
80	.0910	.0898	.0891	.0887	.0883	.0879	.0877	.0874	.0872	.0870	.0868
90	.0919	.0907	.0900	.0895	.0888	.0887	.0884	.0883	.0879	.0877	.0875
100	.0927	.0915	.0908	.0903	.0899	.0895	.0892	.0890	.0887	.0885	.0883
110	.0936	.0923	.0916	.0911	.0907	.0903	.0900	.0898	.0895	.0893	.0891
120	.0945	.0932	.0925	.0919	.0915	.0911	.0908	.0906	.0903	.0901	.0899
130	.0954	.0941	.0934	.0928	.0924	.0920	.0917	.0914	.0912	.0909	.0907
140	.0963	.0950	.0943	.0937	.0932	.0929	.0925	.0923	.0920	.0918	.0916
150	.0973	.0959	.0951	.0946	.0941	.0937	.0934	.0931	.0929	.0926	.0924
160	.0982	.0968	.0961	.0955	.0950	.0946	.0943	.0940	.0937	.0935	.0933
170	.0992	.0978	.0970	.0964	.0959	.0955	.0952	.0949	.0946	.0944	.0941
180	.1002	.0988	.0981	.0973	.0969	.0965	.0961	.0958	.0955	.0953	.0951
190	.1012	.0998	.0989	.0983	.0978	.0974	.0971	.0968	.0964	.0962	.0960
200	.1022	.1008	.0999	.0993	.0988	.0984	.0980	.0977	.0973	.0972	.0969
210	.1033	.1018	.1009	.1003	.0998	.0994	.0990	.0987	.0984	.0981	.0979
220		.1029	.1019	.1013	.1008	.1004	.1000	.0997	.0994	.0991	.0989
230		.1039	.1031	.1024	.1018	.1012	.1010	.1007	.1003	.1001	.0999
240		.1050	.1041	.1034	.1029	.1024	.1020	.1017	.1014	.1011	.1009
250		.1062	.1052	.1045	.1040	.1035	.1031	.1027	.1025	.1022	.1019

Given boiler pressure = 100 lbs. gauge; feed water temperature, original = 60°F. and final = 209°F.; to find the percentage of saving resulting from heating the feed water.

To solve by table look in column of steam pressures headed "100" and opposite to 60° in the first column read .0864, which multiplied by (209—60 = 149) the increase of temperature of feed-water, gives 12.9 per cent.

FORMULA:

$$FT—OT \times C = \text{percentage}$$

FT = final temperature = 209
OT = original temperature = 60
C = constant = .0864

EXAMPLE:

209 = final temperature
60 = original temperature
149 = difference of temperature
.0864 = column constant
596
894
1192
12.8736 = 12 9/10 per cent. nearly

PUMPS AND TANKS.

The efficiency of a pump varies with the type, size, lift, elevation, temperature of water and friction. The steam pump is flexible as regards capacity, a few revolutions faster or slower will greatly increase or diminish the quantity delivered, the maximum efficiency depending on details as to size and connection and locating pump. Hot water cannot be lifted by suction, as its vapor destroys the necessary vacuum, hence the necessity to have the hot water flow to the pump. When long suction pipes are used it will be necessary to have a larger size than with shorter distances, this to allow for friction which might prevent adequate supply to pump. Use as few elbows and sharp bends and valves as possible; avoid traps or air pockets in pipe; suction pipes should be absolutely air tight. A vacuum chamber should be placed on the opposite side of the pump from where suction enters and a foot valve will be found advantageous and desirable, the latter if its location is such that it can be drained when necessary. The valve insures quick starting of pump by keeping suction pipe filled with water. A priming pipe will be convenient when chambers are to be filled to enable pump to start quickly. In starting a pump under pressure it oftentimes happens that the pump will not discharge the water while the pressure is resting on the discharge valve, for the reason that the air in pump cylinders is not discharged, but only compressed by the motion of plungers, then it is necessary to expel air from pump and suction pipe. This can be done by placing a check valve in the discharge pipe near the pump and opening an air vent on the discharge between pump and check, or on a valve chamber on top.

A relief valve is desirable, to prevent damage which might occur by obstruction in discharge line, thus increasing pressure on pump in excess of that which pump was designed for.

Sometimes a pump when first started will deliver a good stream of water, which gradually diminishes in volume until it stops entirely. One reason for this is leak in suction pipes or stuffing box of pump, or, when suction primer is used, in the hand pump stuffing box. Another reason might be that the pump lowers the suction supply, thus increasing the lift until there is not sufficient speed for the elevation. If the pump works indifferently, delivering a stream obviously too small, it is generally because the pump was not properly primed and some air remains in the top part of pump shell. Unless primed by steam ejector the pet cock or plug found on top of pump shell should always be open while priming, and the pump must not be started until water flows out of same.

A pump with horizontal top discharge and short length of discharge pipe is sometimes difficult to start, especially if suction lift is high, owing to the fact that the water is thrown out of the pump shell before the water in suction pipe has got fairly started, thus allowing air to rush back into the pump. If the pump is to work under this condition it is better to use a pump with a vertical discharge and deliver through an elbow, or else lead the discharge pipe upward for a short distance so as to keep a slight pressure or head on the pump, and after priming as high as possible start quickly.

There is generally nothing gained by running above the proper speed required for a given elevation.

To find the theoretical horse power required to elevate water, multiply the gallons pumped per minute by the head in feet and by 8.33 (weight of one gallon of water) and divide product by 33,000. This will be only approximate.

LEGEND:

800 = gallons per minute
20 = feet elevation
8.33 = weight of one gallon of water

EXAMPLE:

800 gallons per minute
20 = feet elevation
16000
8.33 = weight of one gallon of water
48000
48000
128000
33000)133280.000(4.038 H. P. required
132000
1280 00
990 00
290 000
264 000
26 000

Ordinarily pumps will elevate water 50 to 60 feet, and if specially built in regard to strength, could elevate 100 feet, depending on speed.

THEORETICAL STEAM CONSUMPTION.

AT A PISTON TRAVEL OF 100 FEET PER MINUTE.

For use with this table, the effective piston travel is only that portion of the total travel during which the steam valve is open. Thus, if an engine is running 400 feet per minute, and cutting off at ½ stroke, its effective travel will be 200 feet, and its theoretical steam consumption will be 200 divided by 100 multiplied by the amount given in the table for its cylinder diameter and steam pressure. The actual consumption exceeds the theoretical by 25 per cent to 50 per cent.

Diameter of Cylinder	Cubic Feet per Minute	INITIAL STEAM PRESSURE									
		60	70	80	90	100	110	120	130	140	150
		STEAM CONSUMPTION IN POUNDS PER HOUR									
8	34.9	365	410	455	500	540	585	630	670	720	760
9	44.3	465	507	575	630	690	740	800	855	920	964
10	54.5	570	640	710	780	845	915	985	1050	1125	1185
11	66	690	770	860	940	1020	1110	1190	1270	1360	1435
12	78.5	820	920	1020	1120	1220	1320	1420	1520	1620	1710
14	107	1120	1250	1390	1530	1660	1800	1940	2070	2210	2330
16	139.6	1460	1625	1810	2000	2160	2350	2530	2700	2880	3040
18	176.7	1850	2070	2290	2530	2750	2970	3200	3420	3650	3850
20	218.2	2290	2550	2840	3120	3380	3660	3950	4200	4500	4750
22	264	2760	3090	3430	3760	4100	4430	4780	5090	5440	5750
24	314	3290	3660	4070	4490	4860	5270	5680	6060	6480	6820
26	369	3870	4310	4800	5270	5720	6200	6680	7110	7600	8020
28	428	4490	5000	5560	6110	6650	7190	7750	8260	8820	9310
30	491	5160	5750	6390	7010	7610	8250	8880	9490	10120	10680

EXAMPLE: To determine the steam consumption of a 12 and 18 × 12 × 18 *Duplex* Compound Pump: Piston speed 85 feet per minute: Initial Steam pressure 100 pounds.

Since the pump is duplex and since live steam enters the high pressure cylinders only, the theoretical consumption would be double that of a single 12″ cylinder; or at 100 feet piston speed, 1220 × 2 = 2440 pounds per hour.

Theoretical consumption at 85 feet piston speed, 2440 × .85 = 2074 pounds per hour.

The actual steam consumption exceeds the theoretical by 20 per cent to 50 per cent.

The mean pressure of the atmosphere is usually estimated at 14.7 pounds per square inch, so that with a perfect vacuum it will sustain a column of mercury 29.9 inches, or a column of water 33.9 feet high at sea level.

To determine the proportion between the steam and the pump cylinder, multiply the given area of the pump cylinder by the resistance on the pump in pounds per square inch, and divide the product by the available pressure of steam in pounds per square inch. The product equals the area of the steam cylinder. To this must be added an extra area to overcome the friction, which is usually taken at 25 per cent.

The resistance of friction in the flow of water through pipes of uniform diameter is independent of the pressure and increase directly as the length and the square of the velocity of the flow, and inversely as the diameter of the pipe. With wooden pipes the friction is 1.75 times greater than in metallic. Doubling the diameter increases the capacity four times.

To determine the velocity in feet per minute necessary to discharge a given volume of water in a given time, multiply the number of cubic feet of water by 144 and divide the product by the area of the pipe in inches.

To determine the area of a required pipe, the volume and velocity of water being given, mulptily the number of cubic feet of water by 144 and divide the product by the velocity in feet per minute.

To find the diameter of pump plungers to pump a given quantity of water at 100 feet piston speed per minute, divide the number of gallons by 4, then extract the square root, and the result will be the diameter in inches of the plungers.

To find the number of gallons delivered per minute by a single double-acting pump at 100 feet piston speed per minute, square the diameters of the plungers, then multiply by 4.

The area of the steam piston, multiplied by the steam pressure, gives the total amount of pressure that can be exerted. The area of the water piston, multiplied by the pressure of water per square inch, gives the resistance. A margin must be made between the power and resistance.

CAPACITY OF PUMPS AT 100 FEET PISTON SPEED.

A travel of 100 feet piston speed per minute is considered practical and is accepted as standard speed. Slow speed for boiler feeding is recommended. No set rule can be given to cover all conditions. In Fire Pumps, where the largest quantity of water is required, the speed may exceed 200 feet per minute.

THEORETICAL CAPACITY OF PUMPS AT 100 FEET SPEED OF PISTON OR PLUNGER.

Diameter of Pump or Plunger in Inches	U. S. Gallons Per			Diameter of Pump or Plunger in Inches	U. S. Gallons Per		
	Minute	Hour	24 Hours		Minute	Hour	24 Hours
1	4.07	244.7	5875	14¼	828	49704	1192896
1¼	6.37	382.5	9180	14½	858	51468	1235232
1½	9.18	550.8	13219	14¾	887	53256	1278144
1¾	12.49	749	17992	15	918	55070	1321915
2	16.31	979	23500	15¼	949	56928	1366272
2¼	20.6	1239	28180	15½	980	58800	1411200
2½	25.5	1530	36720	15¾	1012	60720	1457280
2¾	30.8	1851	44424	16	1044	62668	1504046
3	36.7	2203	52878	16¼	1077	64638	1551312
3¼	43.1	2586	62064	16½	1110	66642	1599408
3½	49.9	2998	71971	16¾	1144	68676	1648224
3¾	57.3	3442	82619	17	1179	70752	1698048
4	65.2	3916	94002	17¼	1214	72840	1748160
4¼	73.7	4422	106128	17½	1249	74964	1799136
4½	82.6	4957	118971	17¾	1285	77124	1850976
4¾	92	5523	132552	18	1322	79314	1903550
5	102	6120	146880	18¼	1359	81528	1956672
5¼	112	6745	161934	18½	1396	83778	2010672
5½	123	7404	177696	18¾	1434	86060	2065449
5¾	134	8093	194248	19	1473	88368	2120832
6	146	8812	211511	19¼	1511	90660	2175840
6¼	159	9562	229500	19½	1552	93120	2234880
6½	172	10344	248256	19¾	1590	95400	2289600
6¾	185	11152	267660	20	1632	97920	2350080
7	200	11995	287884	20¼	1673	100380	2409120
7¼	214	12867	308808	20½	1714	102840	2468160
7½	229	13769	330478	20¾	1756	105396	2529504
7¾	245	14700	352300	21	1799	107952	2590848
8	261	15667	376011	21¼	1842	110538	2652912
8¼	277	16660	399852	21½	1886	113154	2715696
8½	294	17688	424512	21¾	1930	115800	2779200
8¾	312	18741	449978	22	1974	118482	2843568
9	330	19828	475887	22¼	2020	121194	2908656
9¼	349	20944	502668	22½	2065	123924	2974176
9½	368	22092	530208	22¾	2111	126696	3040704
9¾	388	23280	558720	23	2158	129492	3107808
10	408	24480	587518	23¼	2205	132324	3175776
10¼	428	25716	617184	23½	2253	135186	3244464
10½	449	26989	647789	23¾	2301	138078	3313872
10¾	471	28290	678960	24	2349	140958	3382992
11	493	29616	710784	24¼	2399	143952	3454848
11¼	516	30986	743677	24½	2449	146958	3526992
11½	539	32374	776993	24¾	2499	149952	3598848
11¾	564	33795	811080	25	2550	152994	3671856
12	587	35251	846046	25½	2653	159179	3820300
12¼	612	36735	881640	26	2758	165484	3971630
12½	637	38250	918000	26½	2865	171908	4125800
12¾	663	39816	955584	27	2974	178457	4282967
13	689	41370	992880	27½	3085	185130	4443125
13¼	716	42972	1031328	28	3199	191922	4606125
13½	743	44610	1070640	28½	3314	198838	4772118
13¾	771	46278	1110672	29	3431	205876	4941028
14	799	47980	1151536	30	3672	220320	5287675

For practical purposes, deduct 10 per cent, as no pump will deliver its theoretical capacity.

FRICTION LOSS IN POUNDS PRESSURE.

For each 100 feet of length, in different size, clean iron pipes, discharging given quantities of water per minute.

Galls. per Minute.	SIZES OF PIPES—INSIDE DIAMETER.														
	¾ in.	1 in.	1¼ in.	1½ in.	2 in.	2½ in.	3 in.	4 in.	6 in.	8 in.	10 in.	12 in.	14 in.	16 in.	18 in.
5	3.3	0.84	0.31	0.12	....	....	....	....	....	....	....	....	....	....	....
10	13.0	3.16	1.05	0.47	0.12	....	....	....	....	....	....	....	....	....	....
15	28.7	6.98	2.38	0.97	....	....	....	....	....	....	....	....	....	....	....
20	50.4	12.3	4.07	1.66	0.42	....	....	....	....	....	....	....	....	....	....
25	78.0	19.0	6.40	2.62	....	0.21	1.10	....	....	....	....	....	....	....	....
30	....	27.5	9.15	3.75	0.91	....	....	....	....	....	....	....	....	....	....
35	....	37.0	12.4	5.05	....	....	....	....	....	....	....	....	....	....	....
40	....	48.0	16.1	6.52	1.60	....	....	....	....	....	....	....	....	....	....
45	....	....	20.2	8.15	....	....	....	....	....	....	....	....	....	....	....
50	....	....	24.9	10.0	2.44	0.81	0.35	0.09	....	....	....	....	....	....	....
75	....	....	56.1	22.4	5.32	1.80	0.74	....	....	....	....	....	....	....	....
100	....	....	....	39.0	9.46	7.20	1.31	0.33	0.05	....	....	....	....	....	....
125	....	....	....	....	14.9	4.89	1.99	....	....	....	....	....	....	....	....
150	....	....	....	....	21.2	7.0	2.85	0.69	0.10	....	....	....	....	....	....
175	....	....	....	....	28.1	9.46	3.85	....	....	....	....	....	....	....	....
200	....	....	....	....	37.5	12.47	5.02	1.22	0.17	....	....	....	....	....	....
250	....	....	....	....	....	19.66	7.76	1.89	0.26	0.07	0.03	0.01	....	....	....
300	....	....	....	....	....	28.06	11.2	2.66	0.37	0.09	0.04	....	....	....	....
350	....	....	....	....	....	....	15.2	3.65	0.50	0.12	0.05	0.02	....	....	....
400	....	....	....	....	....	....	19.5	4.73	0.66	0.16	0.06	....	....	....	....
450	....	....	....	....	....	....	25.0	6.01	0.81	0.20	0.07	0.03	....	....	....
500	....	....	....	....	....	....	30.8	7.43	0.96	0.25	0.09	0.04	0.017	0.009	0.005
750	....	....	....	....	....	....	....	....	2.21	0.53	0.18	0.08	....	....	....
1000	....	....	....	....	....	....	....	....	3.88	9.94	0.32	0.13	0.062	0.036	0.020
1250	....	....	....	....	....	....	....	....	....	1.46	0.49	0.20	....	....	....
1500	....	....	....	....	....	....	....	....	....	2.09	0.70	0.29	0.135	0.071	0.040
1750	....	....	....	....	....	....	....	....	....	....	0.95	0.38	....	....	....
2000	....	....	....	....	....	....	....	....	....	....	1.23	0.49	0.234	0.123	0.071
2250	....	....	....	....	....	....	....	....	....	....	....	0.63	....	....	....
2500	....	....	....	....	....	....	....	....	....	....	....	0.77	0.362	0.188	0.107
3000	....	....	....	....	....	....	....	....	....	....	....	1.11	0.515	0.267	0.150
3500	....	....	....	....	....	....	....	....	....	....	....	....	0.697	0.365	0.204
4000	....	....	....	....	....	....	....	....	....	....	....	....	0.910	0.472	0.263
4500	....	....	....	....	....	....	....	....	....	....	....	....	....	0.593	0.333
5000	....	....	....	....	....	....	....	....	....	....	....	....	....	0.730	0.408

HEIGHTS IN FEET TO WHICH PUMPS WILL ELEVATE WATER

Steam pressure, 50 pounds per square inch at the pump. No allowance made for friction in pipes, etc.

Diameter of Steam Cylinders	DIAMETER OF WATER CYLINDERS																
	2 Inch	2½ Inch	3 Inch	3½ Inch	4 Inch	5 Inch	6 Inch	7 Inch	8 Inch	9 Inch	10 Inch	10½ Inch	12 Inch	14 Inch	16 Inch	18 Inch	20 Inch
3½	230	147	102	75	58	37											
4	300	192	134	134	75	48	34										
5	469	300	209	153	117	75	52	38									
6	675	432	300	221	169	108	75	55	42	33							
7	920	588	408	300	230	147	102	75	57	45	37						
8	...	768	533	344	300	192	141	98	75	59	48	44					
9	...	972	675	496	380	243	169	124	95	75	61	55	42				
10	...	...	833	612	469	300	208	153	117	94	75	68	50	38			
12	...	...	...	881	675	432	300	220	169	133	108	97	75	55	42		
14	...	...	...	...	920	588	408	300	228	182	147	133	102	75	57	45	
16	...	...	...	...	...	768	564	392	300	236	192	174	141	98	75	59	48
18	...	...	...	...	...	972	650	490	379	300	243	220	162	122	95	75	61
20	...	...	...	...	...	...	833	600	469	370	300	272	208	150	117	92	75
22	...	...	...	...	...	...	1008	741	567	448	364	329	252	185	142	112	91
24	...	...	...	...	...	...	....	882	675	533	432	392	300	220	169	133	108
26	...	...	...	...	...	...	....	1034	788	626	508	460	356	258	197	156	127
28	...	...	...	...	...	...	....	....	919	726	588	533	407	300	230	181	147
30	...	...	...	...	...	...	....	....	1054	834	676	612	468	345	263	208	169
32	...	...	...	...	...	...	....	....	....	948	798	697	533	391	300	237	192
34	...	...	...	...	...	...	....	....	....	1070	868	786	603	442	339	268	217
36	...	...	...	...	...	...	....	....	....	....	972	881	675	495	380	300	243

The maximum limit of piston speed depends upon the head pumped against.

SIZES FOR BOILER FEED PUMPS.

Diameter of Steam Cylinder	Diam. of Water Cylinder	Stroke	Horse Power Boilers	Steam Pipe	Exhaust Pipe	Suction Pipe	Discharge Pipe
3½″	2¼″	4″	30 to 40	⅜″	½″	1″	¾″
4½	2¾	4	80 to 100	½	¾	2	1¼
5½	3½	5	140 to 160	¾	1¼	2½	1½

When long suction is required use larger suction pipe. Ordinarily allowance for boiler feeding is to deliver 1 cubic foot or 7½ gallons of water per horse power.

THEORETICAL DISCHARGE OF CIRCULAR ORIFICES OR NOZZLES

Diameter in Inches (Ellis.)

NOTE. The actual discharge will be less than the theoretical one given below, ranging with the form of nozzle or tube through which the water flows. For a ring nozzle 64%, and for a good form of tapering smooth nozzle about 82%, can be assumed as the actual discharge.

Head.		Velocity of discharge in feet per second.	Number of United States Gallons of 231 Cubic Inches Discharge per Minute.													
Lbs.	Feet		1/16 Inch	⅛ Inch	3/16 Inch	¼ Inch	⅜ Inch	½ Inch	⅝ Inch	¾ Inch	⅞ Inch	1 Inch	1¼ Inch	1½ Inch	2 Inch	2½ Inch
10	23.1	38.58	0.37	1.48	3.30	5.90	13.2	23.6	36.8	53.2	72.2	94.4	148	212	378	590
15	34.7	47.25	0.45	1.81	4.02	7.23	16.2	28.7	45.0	65.1	88.4	116.	181	260	463	723
20	46.2	54.55	0.52	2.09	4.66	8.35	18.7	33.4	52.0	75.3	102.	134.	209	300	534	835
25	57.8	60.99	0.58	2.33	5.23	9.33	20.9	37.2	58.2	84.1	114.	149.	233	336	597	933
30	69.3	66.82	0.64	2.56	5.71	10.2	22.8	40.9	63.7	92.2	125.	164.	256	368	654	1022
35	80.9	72.16	0.69	2.76	6.16	11.0	24.7	44.2	68.8	99.6	135.	177.	276	397	707	1104
40	92.4	77.14	0.74	2.95	6.60	11.8	26.4	47.2	73.6	106.	144.	189.	295	425	755	1180
45	104.0	81.83	0.78	3.13	6.99	12.5	28.0	50.2	78.1	113.	153.	200.	313	450	801	1252
50	115.5	86.26	0.82	3.30	7.37	13.2	29.5	52.8	82.3	119.	161.	211.	330	475	845	1320
55	127.1	90.46	0.86	3.46	7.73	13.8	30.9	55.4	86.3	125.	169.	221.	346	498	886	1385
60	138.6	96.49	0.90	3.62	8.08	14.5	32.3	57.8	90.1	130.	177.	231.	362	520	925	1446
65	150.2	98.35	0.94	3.77	8.40	15.1	33.6	60.2	93.8	136.	184.	241.	377	542	963	1506
70	161.7	102.06	0.97	3.91	8.73	15.6	34.9	62.5	97.4	141.	191.	250.	391	562	999	1561
75	173.3	105.65	1.01	4.04	9.03	16.2	36.1	64.6	101.	146.	198.	259.	404	582	1034	1616
80	184.8	109.11	1.04	4.18	9.33	16.7	37.8	66.6	104.	150.	204.	267.	418	601	1068	1669
85	196.4	112.46	1.07	4.31	9.62	17.2	38.5	68.8	107.	155.	210.	275.	431	620	1101	1720
90	207.9	115.72	1.10	4.43	9.89	17.7	39.6	70.8	110.	160.	217.	283.	443	637	1133	1770
95	219.5	118.89	1.13	4.55	10.2	18.2	40.7	72.8	113.	164.	223.	291.	455	655	1164	1820
100	231.1	121.98	1.16	4.67	10.4	18.7	41.7	74.6	116.	168.	228.	299.	467	672	1194	1866
105	242.6	125.00	1.19	4.78	10.7	19.1	42.8	76.5	119.	172.	234.	306.	478	688	1224	1912
110	254.2	127.94	1.22	4.90	10.9	19.6	43.8	78.3	122.	177.	239.	313.	490	705	1253	1957
115	265.7	130.82	1.25	5.01	11.2	20.0	44.8	80.1	125.	181.	245.	320.	501	720	1281	2002
120	277.3	133.63	1.27	5.12	11.4	20.4	45.7	81.8	127.	184.	250.	327.	512	736	1308	2044
125	288.8	136.38	1.30	5.22	11.7	20.9	46.7	83.5	130.	188.	255.	334.	522	751	1335	2086
130	300.4	139.08	1.33	5.32	11.9	21.3	47.6	85.1	133.	192.	260.	341.	532	766	1362	2128

PRESSURE OF WATER

The pressure of water in pounds per square inch for every foot in height to 300 feet: and then by intervals to 1,000 feet head. By this table, from the pounds pressure per square inch, the feet head is readily obtained, and vice versa.

Feet Head	Pressure per Square Inch	Feet Head	Pressure per Square Inch	Feet Head	Pressure per Square Inch	Feet Head	Pressure Per Square Inch	Feet Head	Pressure per Square Inch	Feet Head	Pressure per Square Inch
1	0.43	53	22.95	105	45.48	157	68.00	209	90.53	261	113.06
2	0.86	54	23.39	106	45.91	158	68.43	210	90.96	262	113.49
3	1.30	55	23.82	107	46.34	159	68.87	211	91.39	263	113.92
4	1.73	56	24.26	108	46.78	160	69.31	212	91.83	264	114.36
5	2.16	57	24.69	109	47.21	161	69.74	213	92.26	265	114.79
6	2.59	58	25.12	110	47.64	162	70.17	214	92.69	266	115.22
7	3.03	59	25.55	111	48.08	163	70.61	215	93.13	267	115.66
8	3.46	60	25.99	112	48.51	164	71.04	216	93.56	268	116.09
9	3.89	61	26.42	113	48.94	165	71.47	217	93.99	269	116.52
10	4.33	62	26.85	114	49.38	166	71.91	218	94.43	270	116.96
11	4.76	63	27.29	115	49.81	167	72.34	219	94.86	271	117.39
12	5.20	64	27.72	116	50.24	168	72.77	220	95.30	272	117.82
13	5.63	65	28.15	117	50.68	169	73.20	221	95.73	273	118.26
14	6.06	66	28.58	118	51.11	170	73.64	222	96.16	274	118.69
15	6.49	67	29.02	119	51.54	171	74.07	223	96.60	275	119.12
16	6.93	68	29.45	120	51.98	172	74.50	224	97.03	276	119.56
17	7.36	69	29.88	121	52.41	173	74.94	225	97.46	277	119.99
18	7.79	70	30.32	122	52.84	174	75.37	226	97.90	278	120.42
19	8.22	71	30.75	123	53.28	175	75.80	227	98.33	279	120.85
20	8.66	72	31.18	124	53.71	176	76.23	228	98.76	280	121.29
21	9.09	73	31.62	125	54.15	177	76.67	229	99.20	281	121.72
22	9.53	74	32.05	126	54.58	178	77.10	230	99.63	282	122.15
23	9.96	75	32.48	127	55.01	179	77.53	231	100.06	283	122.59
24	10.39	76	32.92	128	55.44	180	77.97	232	100.49	284	123.02
25	10.82	77	33.35	129	55.88	181	78.40	233	100.93	285	123.45
26	11.26	78	33.78	130	56.31	182	78.84	234	101.36	286	123.89

Feet Head	Pressure per Square Inch	Feet Head	Pressure per Square Inch	Feet Head	Pressure per Square Inch	Feet Head	Pressure per Square Inch	Feet Head	Pressure per Square Inch	Feet Head	Pressure per Square Inch
27	11.69	79	34.21	131	56.74	183	79.27	235	101.79	287	124.32
28	12.12	80	34.65	132	57.18	184	79.70	236	102.23	288	124.75
29	12.55	81	35.08	133	57.61	185	80.14	237	102.60	289	125.18
30	12.99	82	35.52	134	58.04	186	80.57	238	103.09	290	125.62
31	13.42	83	35.95	135	58.48	187	81.00	239	103.53	291	126.05
32	13.86	84	36.39	136	58.91	188	81.43	240	103.96	292	126.48
33	14.29	85	36.82	137	59.34	189	81.87	241	104.39	293	126.92
34	14.72	86	37.25	138	59.77	190	82.30	242	104.83	294	127.35
35	15.16	87	37.68	139	60.21	191	82.73	243	105.26	295	127.78
36	15.59	88	38.12	140	60.64	192	83.17	244	105.69	296	128.22
37	16.02	89	38.55	141	61.07	193	83.60	245	106.13	297	128.65
38	16.45	90	38.98	142	61.51	194	84.03	246	106.56	298	129.08
39	16.89	91	39.42	143	61.94	195	84.47	247	106.99	299	129.51
40	17.32	92	39.85	144	62.37	196	84.90	248	107.43	300	129.95
41	17.75	93	40.28	145	62.81	197	85.33	249	107.86	320	138.62
42	18.19	94	40.72	146	63.24	198	85.76	250	108.29	330	142.95
43	18.62	95	41.15	147	63.67	199	86.20	251	108.73	350	151.61
44	19.05	96	41.58	148	64.10	200	86.63	252	109.16	370	160.27
45	19.49	97	42.01	149	64.54	201	87.07	253	109.59	390	168.94
46	19.92	98	42.45	150	64.97	202	87.50	254	110.03	400	173.27
47	20.35	99	42.88	151	65.40	203	87.93	255	110.46	500	216.58
48	20.79	100	43.31	152	65.84	204	88.36	256	110.89	600	259.90
49	21.22	101	43.75	153	66.27	205	88.80	257	111.32	700	303.22
50	21.65	102	44.18	154	66.70	206	89.23	258	111.76	800	346.54
51	22.09	103	44.61	155	67.14	207	89.66	259	112.19	900	389.86
52	22.52	104	45.05	156	67.57	208	90.10	260	112.62	1000	433.18

Rule to find pressure of water head: Multiply constant .434 by number of feet of head.

EXAMPLE:

.434 = constant
45 = feet head

2170
1736

19.530 = pressure or 19½ lbs. approximately

TANKS.

Rule to find capacity of round tank: Square diameter in inches and multiply sum by .7854, then by height in inches; divide this product by 231. This gives capacity in gallons.

FORMULA:

$$\frac{D^2 \times .7854 \times h}{231} = \text{capacity of round tank}$$

LEGEND:

D = diameter of tank = 60″
h = height of tank = 60″
231 cubic inches in one gallon

EXAMPLE:

60″ = diameter of tank
60

3600 = diameter squared
.7854

14400
18000
28800
25200

2827.4400
60 = height

231)169646.4000(734.4 gallons capacity
1617

794
693

1016
924

924
924

U. S. GALLONS IN ROUND TANKS.

For 1 Foot in Depth.

Dia. of Tanks.		No. U. S. Gals.	Cubic Ft. and Area in sq. ft.	Dia. of Tanks.		No. U. S. Gals.	Cubic Ft. and Area in sq. ft.	Dia. of Tanks.		No. U. S. Gals.	Cubic Ft. and Area in sq. ft.
ft.	in			ft.	in.			ft.	in.		
1		5.87	.785	5	8	188.66	25.22	**19**		2120.90	283.53
1	1	6.89	.922	5	9	194.25	25.97	19	3	2177.10	291.04
1	2	8.	1.069	5	10	199.92	26.73	19	6	2234.	298.65
1	3	9.18	1.227	5	11	205.67	27.49	19	9	2291.70	306.25
1	4	10.44	1.396	**6**		211.51	28.27	**20**		2350.10	314.16
1	5	11.79	1.576	6	3	229.50	30.68	20	3	2409.20	322.06
1	6	13.22	1.767	6	6	248.23	33.18	20	6	2469.10	330.06
1	7	14.73	1.989	6	9	267.69	35.78	20	9	2529.60	338.16
1	8	16.32	2.182	**7**		287.88	38.48	**21**		2591.	346.36
1	9	17.99	2.405	7	3	308.81	41.28	21	3	2653.	354.66
1	10	17.95	2.460	7	6	330.48	44.18	21	6	2715.80	363.05
1	11	21.58	2.885	7	9	352.88	47.17	21	9	2779.30	371.54
2		23.50	3.142	**8**		376.01	50.27	**22**		2843.60	380.13
2	1	25.50	3.409	8	3	399.88	53.46	22	3	2908.60	388.82
2	2	27.58	3.687	8	6	424.48	56.75	22	6	2974.30	397.61
2	3	29.74	3.976	8	9	449.82	60.13	22	9	3040.80	406.49
2	4	31.99	4.276	**9**		475.89	63.62	**23**		3108.	415.48
2	5	34.31	4.587	9	3	502.70	67.20	23	3	3179.90	424.56
2	6	36.72	4.909	9	6	530.24	70.88	23	6	3244.60	433.74
2	7	39.21	5.241	9	9	558.51	74.66	23	9	3314.	443.01
2	8	41.78	5.585	**10**		587.52	78.54	**24**		3384.10	452.39
2	9	44.43	5.940	10	3	617.26	82.52	24	3	3455.	461.86
2	10	47.16	6.305	10	6	640.74	86.59	24	6	3526.60	471.44
2	11	49.98	6.681	10	9	678.95	90.76	24	9	3598.90	481.11
3		52.88	7.609	**11**		710.90	95.03	**25**		3672.	490.87
3	1	55.86	7.467	11	3	743.58	99.40	25	3	3745.80	500.74
3	2	58.92	7.876	11	6	766.99	103.87	25	6	3820.30	510.71
3	3	62.06	8.296	11	9	811.14	108.43	25	9	3895.60	520.77
3	4	65.28	8.727	**12**		846.03	113.10	**26**		3971.60	530.93
3	5	68.58	9.168	12	3	881.65	117.86	26	3	4048.40	541.19
3	6	71.97	9.261	12	6	918.	122.72	26	6	4125.90	551.55
3	7	75.44	10.085	12	9	955.09	127.68	26	9	4204.10	562.
3	8	78.99	10.559	**13**		992.91	132.73	**27**		4283.	572.66
3	9	82.62	11.045	13	3	1031.50	137.89	27	3	4362.70	583.21
3	10	86.33	11.541	13	6	1070.80	143.14	27	6	4443.10	593.96
3	11	90.13	12.048	13	9	1110.80	148.49	27	9	4524.30	604.81
4		94.	12.566	**14**		1151.50	153.94	**28**		4606.20	615.75
4	1	97.96	13.095	14	3	1193.0	159.48	28	3	4688.80	626.80
4	2	102.	13.635	14	6	1235.30	165.13	28	6	4772.10	637.94
4	3	106.12	14.186	14	9	1278.20	170.87	28	9	4856.20	649.18
4	4	110.32	14.748	**15**		1321.90	176.71	**29**		4941.	660.52
4	5	114.61	15.321	15	3	1366.40	182.65	29	3	5026.60	671.96
4	6	118.97	15.90	15	6	1411.50	188.69	29	6	5112.90	683.49
4	7	123.42	16.50	15	9	1457.40	194.83	29	9	5199.90	695.13
4	8	127.95	17.10	**16**		1504.10	201.06	**30**		5287.70	706.86
4	9	132.56	17.72	16	3	1551.40	207.39	30	3	5376.20	718.69
4	10	137.25	18.35	16	6	1599.50	213.82	30	6	5465.40	730.62
4	11	142.02	18.99	16	9	1648.40	220.35	30	9	5555.40	742.64
5		146.88	19.63	**17**		1697.90	226.98	**31**		5646.10	754.77
5	1	151.82	20.29	17	3	1748.20	233.71	31	3	5737.50	766.99
5	2	156.83	20.97	17	6	1799.30	240.53	31	6	5829.70	779.31
5	3	161.93	21.65	17	9	1851.10	247.45	31	9	5922.60	791.73
5	4	167.12	22.34	**18**		1903.60	254.47	**32**		6016.20	804.25
5	5	172.38	23.04	18	3	1956.80	261.59	32	3	6110.60	816.86
5	6	177.72	23.76	18	6	2010.80	268.80	32	6	6205.70	829.58
5	7	183.15	24.48	18	9	2065.50	276.12	32	9	6301.50	842.39

31½ Gallons equals 1 Barrel.

To find the capacity of Tanks greater than the largest given in the table, look in the table for a Tank of one-half of the given size and multiply its capacity by 4, or one of one-third its size and multiply its capacity by 9, etc.

STEEL TANK DIMENSIONS.

Diameter, Feet.	Height, Feet.	Thickness, Shell, Inches.	Thickness, Head, Inches.	Size, Angle Iron, Inches.	Weight, Lbs.
3	2½	$\frac{3}{16}$	$\frac{3}{16}$	1½	300
3	3	$\frac{3}{16}$	$\frac{3}{16}$	1½	385
4	3	$\frac{3}{16}$	$\frac{3}{16}$	1½	475
4	4	$\frac{3}{16}$	$\frac{3}{16}$	1½	585
4½	4	$\frac{3}{16}$	$\frac{3}{16}$	1½	670
4½	4½	$\frac{3}{16}$	$\frac{3}{16}$	1½	730
5	4½	$\frac{3}{16}$	¼	2	885
5	5	$\frac{3}{16}$	¼	2	955
5½	5	$\frac{3}{16}$	¼	2	1065
5½	5½	$\frac{3}{16}$	¼	2	1135
6	5½	¼	¼	2	1600
6	6	¼	¼	2	1700
7	6	¼	¼	2	2100
7	7	¼	¼	2	2350
8	7	¼	¼	2½	2800
8	8	¼	¼	2½	3000
9	8	¼	¼	2½	3730
9	9	¼	¼	2½	4060
10	9	$\frac{5}{16}$	$\frac{5}{16}$	2½	4965
10	9	$\frac{5}{16}$	$\frac{5}{16}$	2½	5400
10	10	$\frac{5}{16}$	$\frac{5}{16}$	2½	5850
12	10	$\frac{5}{16}$	$\frac{5}{16}$	2½	7250
12	12	$\frac{5}{16}$	$\frac{5}{16}$	2½	8300

NUMBER OF U. S. GALLONS IN RECTANGULAR TANKS.

For 1 Foot in Depth.

Width of Tank, feet.	LENGTH OF TANK, FEET.																				
	2	2½	3	3½	4	4½	5	5½	6	6½	7	7½	8	8½	9	9½	10	10½	11	11½	12
2	29.92	37.40	44.88	52.36	59.84	67.32	74.81	82.29	89.77	97.25	104.73	112.21	119.69	127.17	134.65	142.13	149.61	157.09	164.57	172.05	179.53
2½	......	46.75	56.10	65.45	74.80	84.16	93.51	102.86	112.21	121.56	130.91	140.26	149.61	158.96	168.31	177.66	187.01	196.36	205.71	215.06	224.41
3	......	......	67.32	78.54	89.77	100.99	112.21	123.43	134.65	145.87	157.09	168.31	179.53	190.75	202.97	213.19	224.41	235.63	246.86	258.07	269.30
3½	......	......	......	91.64	104.73	117.82	130.91	144.00	157.09	170.18	183.27	196.36	209.45	222.54	235.63	248.73	261.82	274.90	288.00	301.09	314.18
4	......	......	......	......	119.69	134.65	149.61	164.57	179.53	194.49	209.45	224.41	239.37	254.34	269.30	284.26	299.22	314.18	329.14	344.10	359.06
4½	......	......	......	......	......	151.48	168.31	185.14	201.97	218.80	235.63	252.47	269.30	286.13	302.96	319.79	336.62	353.45	370.28	387.11	403.94
5	......	......	......	......	......	......	187.01	205.71	224.41	243.11	261.82	280.52	299.22	317.92	336.62	355.32	374.03	392.72	411.43	430.13	448.83
5½	......	......	......	......	......	......	......	226.28	246.86	267.43	288.00	308.57	329.14	349.71	370.28	390.85	411.43	432.00	452.57	473.14	493.71
6	......	......	......	......	......	......	......	......	269.30	291.74	314.18	336.62	359.06	381.50	403.94	426.39	448.83	471.27	493.71	516.15	538.59
6½	......	......	......	......	......	......	......	......	......	316.05	340.36	364.67	388.98	413.30	437.60	461.92	486.23	510.54	534.85	559.16	583.47
7	......	......	......	......	......	......	......	......	......	......	366.54	392.72	418.91	445.09	471.27	497.45	523.64	549.81	575.99	602.18	628.36
7½	......	......	......	......	......	......	......	......	......	......	......	420.78	448.83	476.88	504.93	532.98	561.04	589.08	617.14	645.19	673.24
8	......	......	......	......	......	......	......	......	......	......	......	......	478.75	508.67	538.59	568.51	598.44	628.36	658.28	688.20	718.12
8½	......	......	......	......	......	......	......	......	......	......	......	......	......	540.46	572.25	604.05	635.84	667.63	699.42	731.21	763.00
9	......	......	......	......	......	......	......	......	......	......	......	......	......	......	605.92	639.58	673.25	706.90	740.56	774.23	807.89
9½	......	......	......	......	......	......	......	......	......	......	......	......	......	......	......	675.11	710.65	746.17	781.71	817.24	852.77
10	......	......	......	......	......	......	......	......	......	......	......	......	......	......	......	......	748.05	785.45	822.86	860.26	897.66
10½	......	......	......	......	......	......	......	......	......	......	......	......	......	......	......	......	......	824.73	864.00	903.26	942.56
11	......	......	......	......	......	......	......	......	......	......	......	......	......	......	......	......	......	......	905.14	946.27	987.43
11½	......	......	......	......	......	......	......	......	......	......	......	......	......	......	......	......	......	......	......	989.29	1032.3
12	......	......	......	......	......	......	......	......	......	......	......	......	......	......	......	......	......	......	......	......	1077.2

Rule to find capacity of a square tank: Divide cubic inches of tank by 231. The sum will be the number of gallons.

EXAMPLE:

Tank 60″ × 60″ × 60″

60″ width
60″ long
3600
60 = height
gallons in cubic foot 231)216000(= 935 gallons capacity
2079
810
693
1170
1155
15

Rule to find weight of water in same tank: Multiply the number of gallons by 8.33 (this is weight of one gallon of water). This sum will be weight in pounds.

EXAMPLE:

935 = gallons
8.33 = weight of one gallon of water
28 05
280 5
7480
7788.55 = weight of water in pounds

WATER.

One U. S. gallon equals 231 cubic inches.
One U. S. gallon equals .133 cubic feet.
One U. S. gallon equals 8.33 pounds.
One U. S. gallon equals .83 imperial gallon.

One imperial gallon equals 277.274 cubic inches.
One imperial gallon equals .16 cubic feet.
One imperial gallon equals 10 pounds.
One imperial gallon equals 1.2 U. S. gallon.

One cubic inch of water equals .03607 pound.
One cubic inch of water equals .003607 imperial gallon.
One cubic inch of water equals .004329 U. S. gallon.

One cubic foot of water equals 6.23 imperial gallons.
One cubic foot of water equals 7.48 U. S. gallons.
One cubic foot of water equals 62.321 pounds.
One cubic foot of water equals .028 ton.

One pound of water equals 27.72 cubic inches.
One pound of water equals .10 imperial gallon.
One pound of water equals .12005 U. S. gallon.

One ton of water equals 35.98 cubic feet.
One ton of water equals 224 imperial gallons.
One ton of water equals 268.8 U. S. gallons.

A column of water 1 foot high equals .433 pounds pressure per square inch.

A pressure of 1 pound per square inch equals 2.31 feet of water in height.

A pressure of 1 ounce per square inch equals .144 feet of water in height.

HORSE POWER OF BELTING

Horse-Power which may be transmitted by open single belts to pulleys running 100 revolutions per minute, the diameter of the driving and driven pulley being equal. The horse-power of double belts is 10-7 of that given in the table.

Diameter of Pulley in inches	WIDTH OF BELT IN INCHES.												
	2 H. P.	3 H. P.	4 H. P.	5 H. P.	6 H. P.	8 H. P.	10 H. P.	12 H. P.	14 H. P.	16 H. P.	18 H. P.	20 H. P.	22 H. P.
6	.44	.65	.87	1.09	1.31								
7	.51	.76	1.01	1.27	1.53								
8	.58	.87	1.16	1.45	1.75								
9	.65	.98	1.31	1.64	1.97								
10	.73	1.09	1.45	1.81	2.18								
11	.80	1.20	1.60	2.0	2.40								
12	.87	1.31	1.75	2.18	2.62								
13	.95	1.42	1.89	2.36	2.83								
14	1.02	1.52	2.02	2.53	3.05								
15	1.09	1.64	2.19	2.73	3.29								
16	1.16	1.74	2.32	2.91	3.48								
17	1.24	1.85	2.47	3.09	3.70								
18	1.31	1.96	2.62	3.27	3.92								
19	1.39	2.07	2.76	3.45	4.14								
20	1.45	2.18	2.91	3.64	4.36								
21	1.52	2.29	3.05	3.82	4.58								
22	1.60	2.40	3.20	4.0	4.80								
23	1.67	2.51	3.35	4.18	5.02								
24			3.50	4.4	5.20	7.0	8.7	10.5	12.2	14.0	16.0	17.0	19.0
25			3.6	4.5	5.5	7.3	9.1	10.9	12.7	14.5			
26			3.8	4.7	5.7	7.6	9.5	11.3	13.2	15.1			
27			3.9	4.9	5.9	7.8	9.8	11.8	13.7	15.6			
28			4.1	5.1	6.1	8.1	10.2	12.2	14.3	16.3			
29			4.2	5.3	6.3	8.4	10.5	12.6	14.8	16.9			
30			4.4	5.4	6.6	8.7	10.9	13.1	15.3	17.4	19.0	22.0	24.0
31			4.5	5.6	6.8	9.0	11.3	13.5	15.8	18.0			
32			4.7	5.8	7.0	9.3	11.6	14.0	16.3	18.6			
33			4.8	6.0	7.2	9.6	12.0	14.4	16.8	19.2			
34			4.9	6.2	7.4	9.9	12.4	14.8	17.3	19.8			
35			5.1	6.4	7.6	10.2	12.7	15.3	17.9	20.4			
36			5.2	6.5	7.8	10.5	13.1	15.7	18.3	20.9	24.0	26.0	29.0
37			5.4	6.7	8.1	10.8	13.5	16.2	18.9	21.5			
38			5.5	6.9	8.3	11.0	13.8	16.6	19.3	22.1	25.0	28.0	30.0
39			5.7	7.1	8.5	11.3	14.2	17.0	19.9	22.7			
40			5.8	7.3	8.7	11.6	14.6	17.5	20.4	23.3	26.0	29.0	32.0
42			6.1	7.6	9.2	12.2	15.3	18.2	21.4	24.3	28.0	31.0	34.0
44			6.4	8.0	9.6	12.8	16.0	19.2	22.4	25.6	29.0	32.0	35.0
46			6.7	8.4	10.0	13.4	16.8	20.1	23.4	26.8			
48			7.0	8.8	10.4	14.0	17.4	21.0	24.4	28.0	31.0	35.0	38.0
50			7.2	9.0	10.9	14.6	18.2	21.8	25.4	29.0	33.0	36.0	40.0
54			7.8	9.8	11.8	15.6	19.6	23.6	26.4	31.2	35.0	39.0	43.0
60			8.8	10.8	13.1	17.4	21.8	26.2	30.6	34.8	39.0	44.0	48.0
66			9.6	12.0	14.4	19.2	24.0	28.8	33.6	38.4	43.0	48.0	53.0
72			10.4	13.0	15.6	21.0	26.2	31.4	36.6	41.8	47.0	52.0	58.0
78			11.4	14.2	17.0	22.6	28.4	34.0	39.8	45.4	51.0	57.0	62.0
84			12.2	15.2	19.4	24.4	30.6	36.4	42.8	48.6	55.0	61.0	67.0

Rule to find length of belt: Add together the diameter in inches of the two pulleys; divide this by 2 and multiply the quotient by constant 3¼ (3.25); to this add twice the distance in inches between the centers of shaft; the result will give length of belt approximately.

FORMULA:

$$\left(\frac{2D}{2}\right) \times C + (2 \times d) = \text{length of belt}$$

LEGEND:

D = diameter of pulleys = 30″
20″
C = constant = 3.25
d = distance between shaft centers = 10′ = 120″

EXAMPLE:

30″ and
20 = pulley diameters
2)50
25
3.25 = constant
1 25
5 0
75
81.25
twice distance between centers of shaft = 240.
321.25″ = length of belt

THE USE OF BELTING.

The ultimate strength of a single belt one inch in width and one-quarter inch thick is about 750 pounds, but from the weakening effect of the several methods of joining the ends not more than 200 pounds per inch in width should be depended upon for ultimate strain.

Belts will transmit a force of about 55 pounds for every inch in width, and taking the average thickness of belts at one-sixth of an inch, this means a strain of 330 pounds per square inch of section.

The horse power of a laced belt becomes a maximum at a speed of 87.41 feet per second, or 5,245 feet per minute, or considerably over a mile a minute.

One good method for lacing a belt is to punch the holes in two rows and zigzag, thus a six-inch belt would have seven holes, four nearest the end. The first row should be about three-quarters of an inch from the end of the belt and about the same from the sides. On the larger belts the distance would be somewhat increased. Begin the lacing in the center of the belt and lace both ways; keep the ends of the belt in line and the tension on both ends of the lace the same. The lacing should not be crossed on the side of the belt that runs next the pulley, so that the lacing on that side will be parallel with the edges of the belt, while on the other side it will be at an angle. Loose belts can be run on less power it takes to drive that belt, and in order to run the belt loose it must be in good order; so taking care of belts means less fuel for power and longer life to the belts.

Do not use any belt dressing that will make the belt stick to the pulley. The use of a little good oil occasionally, such as neat's-foot, to keep the leather soft and pliable, will give the very best results.

RULES FOR PULLEY SPEED CALCULATION.

Rule to find size of a pulley for a main line shaft, if the speed of shaft and diameter of pulley on the counter shafts are given: Multiply the diameter in inches of pulley on counter shafts by speed and divide by the revolution of the main shaft; the sum will be the diameter of the pulley.

Example:

Main shaft 150 revolutions per minute; to drive a 15″ pulley 350 revolutions per minute what will be the diameter of pulley on main shaft?

15″ diameter pulley counter shaft
350 revolution of counter shaft

750
45

150)5250(35″ diameter of pulley for main line
450

750
750

To find size of a pulley for counter shaft when revolutions of pulley on main shaft are given: Multiply diameter in inches of driving pulley by the revolutions of the main shaft and divide by the speed required on counter line.

EXAMPLE:

35″ diameter of pulley main shaft
150 revolution main shaft
1750
35
revolution counter shaft 350)5250(15″ pulley for counter line
350
1750
1750

To find speed of counter shaft when revolutions of the main shaft and size of pulleys are known:

Multiply the revolutions of main shaft by the diameter in inches of the pulley and divide by the diameter in inches of the pulley on counter shaft.

EXAMPLE:

35″ pulley main shaft
150 revolutions
1750
35
diameter pulley, counter shaft 15)5250(350 revolution of counter shaft
45
75
75

Slip of belt, also thickness of same, will vary the revolutions some.

Horse Power Shafting Transmission

Diameter of Shaft in Inches	REVOLUTIONS PER MINUTE.									
	100	125	150	175	200	225	250	300	350	400
	HORSE POWER.									
$\frac{15}{16}$	1.2	1.4	1.7	2.1	2.4	2.6	3.1	3.6	4.3	5.0
$1\frac{3}{16}$	2.4	3.1	3.7	4.3	4.9	5.5	6.1	7.3	8.5	9.7
$1\frac{7}{16}$	4.3	5.3	6.4	7.4	8.5	9.5	10.5	12.7	14.8	16.9
$1\frac{11}{16}$	6.7	8.4	10.1	11.7	13.4	15.1	16.7	20.1	23.4	26.8
$1\frac{15}{16}$	10.0	12.5	15.0	17.5	20.0	22.5	25.0	30.0	35.0	40.0
$2\frac{3}{16}$	14.3	17.8	21.4	24.9	28.5	32.1	35.6	42.7	49.8	57.0
$2\frac{7}{16}$	19.5	24.4	29.3	34.1	39.0	44.1	48.7	58.5	68.2	78.0
$2\frac{11}{16}$	26.0	32.5	39.0	43.5	52.0	58.5	65.0	78.0	87.0	104.0
$2\frac{15}{16}$	33.8	42.2	50.6	59.1	67.5	75.9	84.4	101.3	118.2	135.0
$3\frac{3}{16}$	43.0	53.6	64.4	75.1	85.8	96.6	107.3	128.7	150.3	171.6
$3\frac{7}{16}$	53.6	67.0	79.4	93.8	107.2	120.1	134.0	158.8	187.6	214.4
$3\frac{11}{16}$	65.9	82.4	97.9	115.4	121.8	148.3	164.8	195.7	230.7	243.6
$3\frac{15}{16}$	80.0	100.0	120.0	140.0	160.0	180.0	200.0	240.0	280.0	320.0
$4\frac{7}{16}$	113.9	142.4	170.8	199.3	227.8	256.2	284.7	341.7	398.6	455.6
$4\frac{15}{16}$	156.3	195.3	234.4	273.4	312.5	351.5	390.6	468.7	546.8	625.0

The following table gives the maximum permissable distances between bearings of continuous shafts:

Diameterof shaft in inches	Distance between wrought iron	Bearings in feet steel
1	12.27	12.61
2	15.46	15.89
3	17.7	18.19
4	19.48	20.02
5	20.99	21.57
6	22.3	22.92
7	23.48	24.13
8	24.55	25.23
9	25.53	26.24
10	26.4	27.18

The length of a bearing is usually given as three times the diameter of the shaft in inches. The distance between bearings are also given as three times diameter, the product being expressed in feet.

Rule to find diameter of a shaft. Multiply the horse power to be transmitted by the constant 100 for wrought iron; divide the product by the number of revolutions per minute and extract the cube root of quotient; this sum will give safe diameter of shafting. For steel use constant 62.5.

Rule to find diameter of shafts as second movers, transmitting power through long lines. Use preceding rule, using constant 50 for wrought iron and 31.5 for steel.

Rule to find diameter for counter shafting well supported by bearings at short distances. Use preceding rules with constant 33 for wrought iron and 21 for steel.

Rule to find horse power a given shaft will transmit. Multiply the cube of the diameter by the revolutions per minute and divide the product by 100.

For SECOND MOVERS — Multiply the cube of the diameter by twice the revolutions and divide the product by 100.

For THIRD MOVERS — Multiply the cube of the diameter by three times the revolutions and divide by 100.

Approximately a one inch shaft will transmit at 100 revolutions 1 horse power as first mover, 2 horse power as second mover, and 3 horse power as third mover, the power transmitted with safety will vary in proportion as to the speed and as the cube of the diameter.

RULES FOR STEAM BOILERS.

See that water-level has not fallen, and examine joints and seams to detect leakage, and furnaces for evidence of bulging.

Blow through water gages; open blow-off cock to remove sediment; try safety valve to insure free action; raise dampers to clear flues of explosive gases; and stir up fire, heating boiler and setting slowly.

In case of low water, immediately cover the fires with ashes, or, if no ashes are at hand, use fresh coal, and close ash-pit doors. Don't turn on the feed under any circumstances, nor tamper with nor open the safety valve. Let the steam outlets remain as they are.

Close throttle and keep closed long enough to show true level of water. If that level is sufficiently high, feeding and blowing will usually suffice to correct the evil. In case of violent foaming, caused by dirty water, or change from salt to fresh, or vice versa, in addition to the action above stated, check draft and cover fires with fresh coal.

In preparing to get up steam after boilers have been open, or out of service, great care should be exercised in making the man and hand-hole joints. Safety valve should then be opened, and blocked open, and the necessary supply of water run in or pumped into the boilers until it shows at second guage in tubular and locomotive boilers; a higher level is advisable in vertical tubulars as a protection to the top end of the tubes. After this is done fuel may be placed upon the grate, dampers opened, and fires started. If chimney or stack is cold and does not draw properly, burn some oily waste or light kindling at the base. Start fires in ample time so it will not be necessary to force them unduly. When steam issues from the safety valve, lower it carefully to its seat and note pressure and action of steam gauge.

If there are other boilers in operation, and stop valves are to be opened to place boilers in connection with others on a steam pipe line, watch those recently fired up until pressure is up to that of the other boilers to which they are to be connected; and, when that pressure is attained open the stop-valves very slowly and carefully.

Never feed cold water into a boiler as it is injurious to the plates and liable to spring the seams and cause them to leak. A good feed water heater should be used; they not only save early repairs on the boiler but effect a great saving in the consumption of coal.

Boilers should be blown off, a little at least, once or twice a day, and the water should be entirely blown off at least once every two weeks, depending on the nature of the feed water. Never blow out a boiler while it is too hot as the arch plates, flues and braces retain heat enough to bake the deposits of mud into a hard scale that becomes firmly attached to their surface. With the walls and arches too hot while blowing off, the plates are liable to injury. Always allow the setting to cool down before emptying completely as the scale and mud will then be quite soft and can easily be washed out with a hose.

If necessary to blow down, allow the boilers to become cool before filling again. Cold water pumped into hot boilers is very injurious from sudden contraction.

Care should be taken that no water comes in contact with the exterior of the boiler, either from leaky joints or other causes.

In tubular boilers the hand holes should be often opened, and all deposits removed, and fire-plates carefully cleaned.

Keep the boiler clean internally and externally and thoroughly examine plates and seams at frequent intervals, especially those in contact with setting or exposed to direct action of fire.

Always raise steam slowly and never light fire until water shows in gauge glasses. Keep furnace walls in good condition and well pointed up. Allow boiler and brick work to cool before emptying boiler. Prevent oil and greasy matter from entering boiler, as same lead to serious inefficiency and to dangerous heating of plates.

Mud drums should be given careful attention and cleaned and inspected regularly just the same as the boiler.

Try the safety valves cautiously and often, as they are liable to become fast in their seats and useless for the purpose intended. If the valve is of the lever type, do not load it with additional weights. The safety valve is set to blow off at a certain pressure and should blow off when the steam gauge registers this pressure; if it does not, one or the other is wrong and should be corrected.

When a blister appears there must be no delay in having it carefully examined, and trimmed or patched, as the case may require.

Particular care should be taken to keep sheets and parts of boilers exposed to the fire perfectly clean; also all tubes, flues and connections well swept. This is particularly necessary where wood or soft coal is used for fuel.

See that proper water-level is maintained. Keep water gauge classes clean and passages clear, by trying gauges frequently. (Lack of proper attention to water gauges leads to more accidents than any other cause.)

Maintain a fire of even thickness, free from holes and clear of ashes and clinkers. (The proper thickness of fire increases with the hardness and size of coal and with the strength of draft.) Regulate fire and draft and feed to meet demands for steam, keeping water level constant to avoid priming or burning of plates. Ash pits are to be kept clear to avoid burning grate bars and to prevent loss of draft and efficiency.

Never attempt to stop a leak or tighten a joint when boiler is under high pressure. Never cut in a boiler with a battery until its pressure is equal to that of the battery.

Before banking fires run water to proper level, which note, and see that the steam pipe drains are open and in working order.

Water in ash pit has an effect of clinkering, and this varies with the amount of sulphur and iron pyrites and ash in fuel, thus choking up air spaces in grate effecting the life of same. Again the moisture mixing with sulphur has the corrosive effect on boiler and tubes; it also has a cooling effect which detracts from combustion, and volatile gases escape unconsumed.

NOTES.

Slight leakage at joints causes grooving.

Covering of boiler and steam pipes saves fuel and increases efficiency.

A boiler showing pulsations of engine gives evidence of being too small for duty.

Fly wheels should not have a greater speed than one mile per minute to be safe.

Globe valves should always be so placed in steam pipes that their stems are nearly horizontal.

Stack should drain inside — for reasons — appearance —as stacks are in use most of the time, the advantage of having drainage outside is not to be weighed with the advantage of draining inside and appearance.

KNOTS AND MILES.

Knts	Miles	Knts	Miles	Knts	Miles	Knts	Miles	Knts	Miles
1.00	1.1515	6.00	6.9091	11.00	12.6667	16.00	18.4242	21.00	24.1818
1.25	1.4394	6.25	7.1970	11.25	12.9545	16.25	18.7121	21.25	24.4697
1.50	1.7273	6.50	7.4848	11.50	13.2424	16.50	19.0000	21.50	24.7576
1.75	2.0152	6.75	7.7727	11.75	13.5303	16.75	19.2879	21.75	25.0455
2.00	2.3030	7.00	8.0606	12.00	13.8182	17.00	19.5758	22.00	25.3333
2.25	2.5909	7.25	8.3485	12.25	14.1061	17.25	19.8636	22.25	25.6212
2.50	2.8788	7.50	8.6364	12.50	14.3939	17.50	20.1515	22.50	25.9091
2.75	3.1667	7.75	8.9242	12.75	14.6818	17.75	20.4394	22.75	26.1970
3.00	3.4545	8.00	9.2121	13.00	14.9697	18.00	20.7273	23.00	26.4848
3.25	3.7424	8.25	9.5000	13.25	15.2576	18.25	21.0152	23.25	26.7727
3.50	4.0303	8.50	9.7879	13.50	15.5455	18.50	21.3030	23.50	27.0606
3.75	4.3182	8.75	10.0758	13.75	15.8333	18.75	21.5909	23.75	27.3485
4.00	4.6061	9.00	10.3636	14.00	16.1212	19.00	21.8788	24.00	27.6364
4.25	4.8939	9.25	10.6515	14.25	16.4091	19.25	22.1667	24.25	27.9242
4.50	5.1818	9.50	10.9394	14.50	16.6970	19.50	22.4545	24.50	28.2121
4.75	5.4697	9.75	11.2273	14.75	16.9848	19.75	22.7424	24.75	28.5000
5.00	5.7576	10.00	11.5152	15.00	17.2727	20.00	23.0303	25.00	28.7879
5.25	6.0455	10.25	11.8030	15.25	17.5606	20.25	23.3182	25.25	29.0758
5.50	6.3333	10.50	12.0909	15.50	17.8485	20.50	23.6061	25.50	29.3636
5.75	6.6212	10.75	12.3788	15.75	18.1364	20.75	23.8939	25.75	29.6515

TABLE SHOWING KNOTS REDUCED TO MILES.

A nautical mile or knot is 6,080.27 feet.

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