MANUAL

OF

STEAM ENGINEERING

BY

W. H. WAKEMAN

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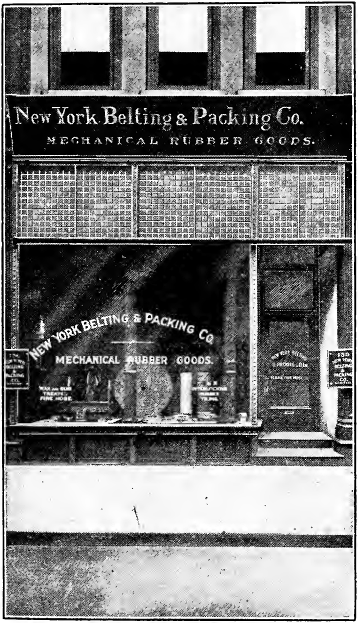

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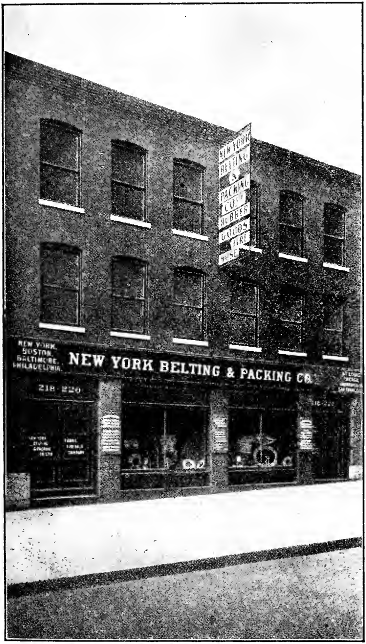

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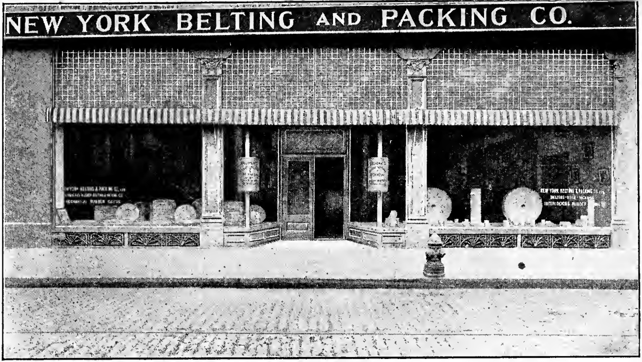

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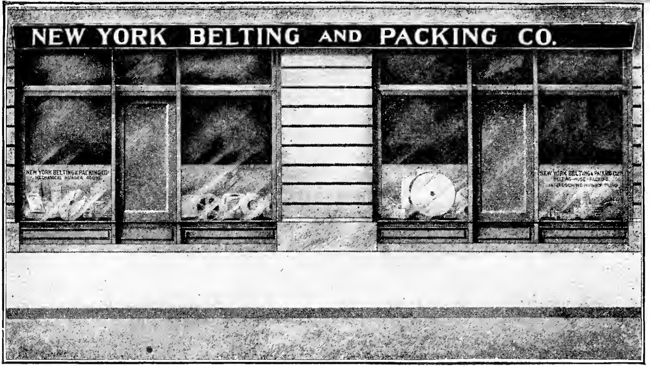

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NEW

A MANUAL

OF

STEAM ENGINEERING

COMPRISING

INSTRUCTIONS, SUGGESTIONS AND ILLUSTRATIONS FOR PROGRESSIVE STEAM ENGINEERS, CONCERNING THE APPLICATION TO MODERN DAILY PRACTICE OF THE APPROVED THEORY OF STEAM ENGINEERING

BY

W. H. WAKEMAN

AUTHOR OF

"ENGINEERING PRACTICE AND THEORY"
"PRACTICAL GUIDE FOR FIREMEN"
AND NUMEROUS ARTICLES FOR THE MECHANICAL PRESS

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PREFACE

THE object sought in presenting this volume to steam engineers and others interested in steam engineering is to provide a convenient and reliable reference book, containing data wanted in every-day practice, arranged in convenient form, with sufficient explanation to render the matter both interesting and instructive.

It contains many tables, which are arranged for use in connection with the reading matter on the same subject, thus condensing information into convenient space, and rendering it available for the busy workers. That the work will prove satisfactory to readers for whom it is intended is the sincere wish of

THE AUTHOR

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DEDICATION

TO the intelligent men who design, construct and operate the steam machinery of this country, keeping it in repair under a great variety of conditions, and securing the best results in practice from every part of both complicated and simple plants, this book is respectfully dedicated by

THE AUTHOR.

SECTION 1

BOILERS

STRENGTH OF STEAM BOILERS

AN erroneous idea concerning the strength of steam boilers prevails among some of the men who are employed in steam plants, which is well illustrated by their answers to the following question: "Suppose that two boilers are made of the same material, with riveted joints of equal strength, and are alike in every respect, except that one is 48 inches and the other 72 inches in diameter. On which of them will it be safe to carry the higher pressure?"

Many of the men to whom this question is referred reply that the larger one will safely carry the higher pressure, and the reason for this is found in the fact that to those who have not intelligently studied the subject the larger a boiler is, the greater its strength appears to be, which is a mistake as the following lines fully explain.

While pressure acts on every inch of the circumference of a boiler shell, it operates exactly as if it pressed upon a space equal to the diameter, therefore if the shell is 72 inches in diameter and the pressure is 100 pounds per square inch, the force which acts to rupture the shell is equal to the diameter multiplied by the pressure and $72 \times 100 = 7,200$ pounds. If we draw a circle representing an end view of the shell, and then draw a straight line dividing it into two equal parts it will be plain that the strain is supported by the two sides of the shell, therefore each must sustain a load equal to $7,200 \div 2 = 3,600$ pounds.

If the shell is but 48 inches in diameter the total load is $48 \times 100 = 4,800$ pounds, or 2,400 pounds on each side. The pressure required to produce a load equal to that carried on each side of the 72 inch shell is found by dividing 3,600 by one-half of the diameter of the boiler in question. In this case it is $3,600 \div 24 = 150$ pounds, thus plainly demonstrating that with 100 pounds on the 72 inch shell and 150 on the 48 inch, the loads are equal.

The iron and steel plates of which boiler shells are made, vary in strength, hence it becomes necessary to know how much stress can be safely put on the plates to be used. This can only be determined by pulling a sample of it apart in a testing machine and noting the strain in pounds that was required to break it. If the plate is .5 inch thick and we take a strip 1 inch wide, it might require 30,000 pounds or more to part it, and although it is quite possible to do this it is better to adopt a plan whereby much less strain will answer the same purpose.

If a piece of this plate 1 inch wide is forged down until it is .3 inch thick, its area will be $1 \times .3 = .3$ square inch, and if pulled apart the strain required to do it may be 16,611 pounds. Dividing the strain required by the area of the test piece where it fractured, gives the tensile strength of the plate in pounds per square inch of section. In this case it is $16{,}611 \div .3 = 55{,}370$ pounds.

When calculating the strength of a boiler, it is necessary to take into account the thickness of the shell, as its strength will vary directly with its thickness. For this calculation it is customary to take a strip of the shell 1 inch wide following the entire circumference, determine its strength and assume that all other parts are equally strong.

In order to be on the safe side, it is necessary to take this strip from the most unfavorable location, as the strength of any structure is determined by the strain that can be safely carried by the weakest part.

In some parts of the shell holes are cut for steam nozzles and other purposes, therefore if the strip found in or near the middle of such a place should be taken without qualification it would possess no strength at all, as it is cut in two, but in all well made boilers these places are strengthened by reinforcing rings, or by special flanging.

The above explanation makes it plain that if a strip 1 inch wide is cut from a sheet .5 inch thick, with a tensile strength of 55,370 pounds, its ultimate strength is $55{,}370 \times .5 = 27{,}685$ pounds.

The riveted joint in the shell of a steam boiler can never be as strong as the solid plate, and as calculations for determining the comparative strength of joints require full explanation, it is here assumed that the design of joint gives .75 of the strength of the solid plate.

Taking this into account we find that the strip 1 inch wide has strength equal to $55{,}370 \times .5 \times .75 = 20{,}763$ pounds.

Again referring to the circle with a straight line drawn through the center of it, showing that the above calculation gives the strength of one side only, it is plain that we must calculate on one-half of the diameter.

For a 72 inch boiler this is 36 inches, and $20{,}763 \div 36 = 576$ pounds, which is the pressure per square inch required to rupture this shell. As a large margin must be assumed for safety, a factor of 5 will be allowed, or in other words one-fifth of the bursting pressure will be taken as the safe working pressure, and $576 \div 5 = 115$, showing that it is safe to carry 115 pounds on such a boiler. This explanation will make clear the following rule for determining the safe working pressure of steam boilers.

Multiply the tensile strength of the plate in pounds per square inch, by its thickness in decimals of an inch, and by the comparative strength of the riveted joint. Divide by one-half of the diamater in inches, and the quotient is the bursting pressure. Divide this by the factor of safety (for which 5 is recommended), and the final quotient is the safe working pressure.

This rule is fully explained in order that the reader may clearly understand the principles involved, and if results secured by it differ from other cases, the cause of difference may be plainly seen.

The following table gives the safe working pressures of boilers from 42 to 84 inches in diameter, with plates of suitable thickness, the tensile strength of which is stated. The table is based on the foregoing rule.

DIAMETER OF SHELL 42 INCHES

FACTOR OF SAFETY 5

Tensile strength 50,000 pounds			Tensile strength 55,000 pounds			Tensile strength 60,000 pounds		
Thickness of plate	Strength of joint	Safe pressure	Thickness of plate	Strength of joint	Safe pressure	Thickness of plate	Strength of joint	Safe pressure
25	.60	71	.25	60	78	.25	.60	85
3125	.60	89	.3125	.60	98	.3125	.60	107
375	.70	125	.375	70	137	.375	.70	150
4375	.75	156	.4375	.75	171	.4375	.75	187
5	.75	178	.5	.75	196	.5	.75	214

DIAMETER OF SHELL 48 INCHES

FACTOR OF SAFETY 5

.25	.60	61	.25	.60	69	.25	60	75
.3125	.60	78	.3125	.60	86	.3125	.60	94
.375	.70	109	.375	.70	120	.375	70	131
.4375	.75	136	.4375	.75	150	.4375	.75	164
.5	.75	156	.5	.75	172	.5	.75	188

DIAMETER OF SHELL 54 INCHES

FACTOR OF SAFETY 5

.25	.60	56	.25	.60	61	.25	60	67
.3125	.60	69	.3125	.60	76	.3125	.60	83
.375	.70	97	.375	.70	107	.375	.70	117
.4375	.75	121	.4375	.75	134	.4375	.75	146
5	.75	139	.5	.75	153	.5	.75	167

DIAMETER OF SHELL 60 INCHES

FACTOR OF SAFETY 5

.3125	.65	68	.3125	.65	74	.3125	.65	81
.375	.70	87	.375	.70	96	.375	.70	105
.4375	.75	109	.4375	.75	120	.4375	.75	130
.5	.75	125	.5	.75	137	.5	.75	150
.5625	.80	150	.5625	.80	165	.5625	.80	180

DIAMETER OF SHELL 66 INCHES

FACTOR OF SAFETY 5

Tensile strength 50,000 pounds			Tensile strength 55,000 pounds			Tensile strength 60,000 pounds		
Thickness of plate	Strength of joint	Safe pressure	Thickness of plate	Strength of joint	Safe pressure	Thickness of plate	Strength of joint	Safe pressure
.3125	.65	61	.3125	.65	68	.3125	.65	74
.375	.70	79	.375	.70	87	.375	.70	95
.4375	.75	100	.4375	.75	109	.4375	.75	119
.5	.75	113	.5	.75	125	.5	.75	136
.5625	.80	136	.5625	.80	150	.5625	.80	164

DIAMETER OF SHELL 72 INCHES

FACTOR OF SAFETY 5

.375	.70	73	.375	.70	80	.375	.70	87
.4375	.70	85	.4375	.70	93	.4375	.70	102
.5	.75	104	.5	.75	115	.5	.75	125
.5625	.75	117	.5625	.75	129	.5625	.75	141
.625	80	140	.625	.80	153	.625	.80	166

DIAMETER OF SHELL 78 INCHES.

FACTOR OF SAFETY 5

.375	.70	67	.375	70	74	.375	.70	81
.4375	.70	78	.4375	.70	86	.4375	.70	94
.5	.75	96	.5	.75	105	.5	.75	115
.5625	.75	108	.5625	.75	119	.5625	.75	130
.625	.80	128	.625	.80	141	.625	.80	154

DIAMETER OF SHELL 84 INCHES

FACTOR OF SAFETY 5

.375	.70	62	.375	.70	68	.375	.70	75
.4375	.75	78	.4375	75	86	.4375	.75	94
.5	.75	89	.5	.75	98	.5	.75	107
.5625	.80	107	.5625	.80	118	.5625	.80	128
.625	.85	126	.625	.85	139	.625	.85	152

As the thickness of boiler plates is frequently stated in sixteenths of an inch, the following table will enable the reader to convert it into decimal fractions without calculation.

$\frac{1}{16}$=.0625	$\frac{5}{16}$=.3125	$\frac{9}{16}$=.5625	$\frac{13}{16}$=.8125
$\frac{1}{8}$=.125	$\frac{3}{8}$=.375	$\frac{5}{8}$=.625	$\frac{7}{8}$=.875
$\frac{3}{16}$=.1875	$\frac{7}{16}$=.4375	$\frac{11}{16}$=.6875	$\frac{15}{16}$=.9375
$\frac{1}{4}$=.25	$\frac{1}{2}$=.5	$\frac{3}{4}$=.75	1= 1000

RULES FOR USING DECIMAL FRACTIONS

ADDITION

Write down the numbers to be added so that each decimal point (after the first), will be directly under the preceding, and proceed to add as with whole numbers.

Example.	Proof.
17.87	.096
.096	41.69
3.7	17.87
41.69	3.7
63.356	63.356

If the same sum is obtained when the numbers to be added are located in different positions, it is evidence that the result is correct.

SUBTRACTION

Set down the minuend, or larger number, and locate the subtrahend, or smaller number so that one decimal point will be directly under the other, and proceed as with whole numbers.

Example.	Proof.
107.8672	93.7526
93.7526	14.1146
14.1146	107.8672

MULTIPLICATION

Proceed as with whole numbers and point off as many decimals as there are in the multiplicand and multiplier, or the first and second numbers added together.

```
Example.          Proof.
  .375       .375).23250(.62
  .62             2250
 -----            ----
  750              750
 2250              750
 -----            ----
.23250               0
```

DIVISION

Divide the dividend by the divisor as with whole numbers, then point off as many decimals in the quotient or answer as those in the dividend exceed those in the divisor.

```
Example.             Proof.
.235).43750(1.86      .235
      235             1.86
      ----            ----
      2025            1410
      1880           1880
      ----
       1450          235
       1410             40
       ----          -----
         40          .43750
```

In all cases where there are not enough figures in the answer to provide the required number of decimal places, ciphers must be added at the left hand until enough are secured.

Every cipher added at the left hand reduces the value of the fraction, but when added at the right hand for convenience, they do not affect the result.

THE UNITED STATES GOVERNMENT RULE FOR SAFE PRESSURES

Many engineers prefer to use this rule especially where they desire to carry as much pressure on their boilers as they can find authority for, and this idea is correct provided the rule is used in an intelligent manner. This rule is as follows:

Multiply one-sixth of the tensile strength by the thickness of plate and divide by one-half of the diameter, for single riveted seams. For double riveted seams add 20 per cent.

A very common error when considering this rule is to conclude that a factor of safety of 6 is used, but this is not the case, because one-sixth of the strength of the whole plate before it is punched, or drilled, is taken, and as the riveted seams are not as strong as the full plate the difference between the two operate to reduce the factor of safety. For illustration of this point, see the following example:

Selecting from the foregoing table a boiler 72 inches in diameter made of plates .5 inch thick whose tensile strength is 60,000 pounds, with a double riveted joint that has 75 per cent. of the strength of the solid plate, the safe pressure is 125 pounds. As the factor of safety is 5, the bursting pressure is $125 \times 5 = 625$ pounds.

Applying the U. S. Government rule to the above case gives the following result:

$$(60{,}000 \div 6) \times .5 \div (72 \div 2) = 139,$$

and adding 20 per cent. brings the result up to 166 pounds for a safe pressure. Dividing the bursting pressure by the latter, shows that the real factor of safety is $625 \div 166 = 3.76$ although the apparent factor is 6.

From the foregoing it will be plain that the U. S. Government rule does not take into consideration the varying strength of riveted joints of different designs, except that in all cases a double riveted joint is 20 per cent. stronger than the single riveted design. It is quite possible to design a single riveted joint that is stronger than another having two rows of rivets, but the U. S. Government rule taken as a whole is not so short and simple as it appears, for it includes directions for making riveted joints that are strong and durable, therefore the only mistake made along this line is when careless engineers or ignorant steam users apply the rule to cases where inferior joints have been made by incompetent boiler makers, but it was never intended for such applications.

The following table has been prepared for the benefit of those who prefer this rule, but it should be remembered that every single riveted joint to which the table refers must contain 60 per cent. of the strength of the solid plate, and every double riveted joint must have 75 per cent. of it. Furthermore the factor of safety averages about 3.5. For these conditions the following safe pressures are allowed.

DIAMETER OF SHELL 42 INCHES

	Tensile strength 50,000 lbs.		Tensile strength 55,000 lbs.		Tensile strength 60,000 lbs.	
Thickness of plate	Single riveted	Double riveted	Single riveted	Double riveted	Single riveted	Double riveted
.25	99	119	109	131	119	143
.3125	124	148	136	163	148	178
.375	148	178	163	196	178	214
.4375	173	207	191	229	208	249
.5	198	237	218	261	238	285

DIAMETER OF SHELL 48 INCHES

Thickness of plate	Tensile strength 50,000 lbs.		Tensile strength 55,000 lbs.		Tensile strength 60,000 lbs.	
	Single riveted	Double riveted	Single riveted	Double riveted	Single riveted	Double riveted
.25	87	104	95	114	104	125
.3125	108	130	119	143	130	156
.375	130	156	143	171	156	187
.4375	152	182	167	200	182	218
.5	173	207	191	229	208	249
DIAMETER OF SHELL 54 INCHES						
.25	77	92	85	102	92	111
.3125	96	115	106	127	115	138
.375	116	139	127	152	139	167
.4375	135	162	148	177	162	194
.5	154	184	169	202	185	222
DIAMETER OF SHELL 60 INCHES						
.3125	87	104	95	114	104	125
.375	104	125	114	137	125	150
.4375	120	145	133	159	146	175
.5	139	166	153	183	166	199
.5625	156	187	171	205	187	224
DIAMETER OF SHELL 66 INCHES						
.3125	79	95	87	104	95	114
.375	95	114	104	125	113	136
.4375	110	132	121	145	132	158
.5	126	151	139	166	151	181
.5625	142	170	156	187	170	204
DIAMETER OF SHELL 72 INCHES						
.375	87	104	95	114	104	125
.4375	101	121	111	133	121	145
.5	115	138	127	152	139	166
.5625	130	156	143	171	156	187
.625	144	173	159	190	173	207
DIAMETER OF SHELL 78 INCHES						
.375	80	96	88	106	96	115
.4375	93	111	103	123	112	134
.5	107	128	117	140	128	153
.5625	120	144	132	158	144	172
.625	133	159	147	176	160	192

DIAMETER OF SHELL 84 INCHES

	Tensile strength 50,000 lbs.		Tensile strength 55,000 lbs.		Tensile strength 60,000 lbs.	
Thickness of plate	Single riveted	Double riveted	Single riveted	Double riveted	Single riveted	Double riveted
.375	74	89	82	98	89	107
.4375	86	103	95	114	104	125
.5	99	119	109	130	119	142
.5625	111	133	122	146	134	160
.625	124	148	136	163	149	178

THE STRENGTH OF RIVETED JOINTS

One of the most important problems presented in connection with steam boilers is to calculate the strength of riveted joints of various designs. As a general rule (or what may more properly be called an estimate), the strength of single riveted joints have 56 per cent. and double riveted joints 70 per cent. of the strength of the solid plate.

However, it is not a good plan to assume this proportion of strength, for while some joints may greatly exceed it, others will prove to be much weaker, hence the only safe plan is to calculate the strength of each joint separately when the safe working pressure of a boiler is to be determined.

While studying this subject the principles involved should first receive attention, for if they are well understood it will assist the engineer not only in his efforts to learn the rules, but also in remembering them afterward.

The comparative strength of the plate before and after it is punched or drilled for the rivets, is usually determined first. If we take a strip of boiler plate 3 inches wide as shown at 2 in Fig. 1,

FIG.1

it represents **100** per cent. of strength, because none of it has been cut away for rivet holes. Proceeding to drill a ¾-inch hole in it, as shown at 3, it is plain that one-quarter of the metal has been cut away and it does not need any calculation to demonstrate it, but in order to fully illustrate the point the following may be considered.

Having removed .75 inch of the metal there is $3-.75=2.25$ inches left, and dividing this by the original width shows the percentage remaining, or $3-.75\div 3=.75$ of the strength of the solid plate hence the rule.

From the pitch of the rivets subtract the diameter of one rivet, and divide the remainder by the pitch. The quotient is the percentage of the strength of the plate.

As a formula it appears as follows:

$$\frac{P-D}{P}=S$$

P = Pitch of rivets.

D = Diameter of rivets.

S = Percentage of strength.

As the pitch of the rivets is used the strip appears with one-half of a rivet hole in each edge, but the result is not changed by it. See Fig. 2. The thickness of the plate is not considered in this connection as it is not necessary, and the use of it would make the calculation more complicated without corresponding benefit.

Fig. 2

The next point to be considered is the strength of the rivets, and for convenience of illustration it is assumed that the shearing strength of the rivets and the tensile strength of the plate are equal. While this is true in some cases it is not in others, therefore it cannot be laid down as a general rule.

The area of a circle $\frac{3}{4}$ inch in diameter is .44 square inch, and assuming that there are two rows of rivets, the area of two as illustrated in Fig. 3 is

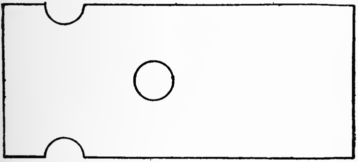


FIG. 3

$.44 \times 2 = .88$ square inch. As this must be compared with the full area of the plate, its thickness must now be taken into consideration and for this purpose it is assumed to be .375 inch. Multiplying this by the pitch of rivets shows that the full area is $.375 \times 3 = 1.12$ square inches.

By dividing the area of rivets by the area of plate the strength of the rivets is found to be $.88 \div 1.12 = .78$ of the solid plate, hence the rule.

Divide the area of one rivet multiplied by the number of rows, by the thickness of the plate multiplied by the pitch of the rivets. The quotient is the percentage of strength of the solid plate. As a formula it appears as follows:

$$\frac{A \times N}{T \times P} = S$$

A = Area of one rivet.
N = Number of rows.
T = Thickness of plate.
P = Pitch of rivets.
S = Percentage of strength.

In the above example the strength would be taken as .75, because the lower result must always be adopted in order to be on the safe side.

This example seems to show that it is safe to call the strength of all double riveted joints .70 or more, but the defect in it is due to the fact that it only illustrates plates of a low tensile strength, and these are not fit for the construction of modern boilers.

The Hartford Steam Boiler Inspection and Insurance Co. have adopted 38,000 pounds per square inch of sectional area as the shearing strength of rivets in single shear, and while it does not seem right to use an arbitrary rating for all joints, as rivets vary in strength, still if they are put into a joint in a workmanlike manner they will always be as strong as these figures indicate, and where they prove to be stronger, it renders the boiler safer.

Adopting this standard for the strength of rivets, and using plates with a tensile strength of 55,000 pounds, the result is much less, as follows:

Pitch of rivets 3 inches.
Diameter of rivets .75 inch.
Number of rows 2.
Thickness of plate .375 inch.

$$\frac{3-.75}{3}=.75 \text{ for the plates.}$$

$$\frac{.44\times2\times38{,}000}{.375\times3\times55{,}000}=.54 \text{ for the rivets.}$$

If the strength of this joint is taken at .70 it becomes dangerous because its real strength is only .54. It can be improved by reducing the pitch to 2.25 inches.

$$\frac{2.25-.75}{2.25}=.66 \text{ for the plates.}$$

$$\frac{.44\times2\times38{,}000}{.375\times2.25\times55{,}000}=.72 \text{ for the rivets.}$$

This increases the strength of joint to .66 when calculated on a very conservative basis.

For convenience in calculating the strength of rivets the following table gives the areas of circles from ½ to 1 inch inclusive.

DIAMETER AND AREA OF RIVETS.

Diameter of rivet	Decimal equivalent	Area
½	.5	.196
9/16	.5625	.248
⅝	.625	.306
11/16	.6875	.371
¾	.75	.441
13/16	.8125	.518
⅞	.875	.601
15/16	.9375	.690
1	1.	.785

In some cases it is advisable to increase the pitch and add another row of rivets, making what is known as the triple riveted joint. The foregoing rules for strength of plate and rivets applies to such a joint and shows it to be efficient when well designed, but like others its actual efficiency should be calculated and not assumed. The following proportions will give good results. See Fig. 4.

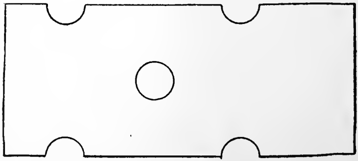


FIG 4

Pitch of rivets 3.5 inches.
Diameter of rivets 13/16 inch.
Thickness of plate .375 inch.

$$\frac{3.5-.8125}{3.5}=.76 \text{ for the plates.}$$

$$\frac{.518\times3\times38{,}000}{.375\times3.5\times55{,}000}=.81 \text{ for the rivets.}$$

This demonstrates that the strength of such a joint may be taken at .76 of the strength of the solid plate. The same proportions may be used for a joint in plates of 60,000 pounds tensile strength, but owing to the increase in strength of the plates, the comparative strength of the rivets is less.

$$\frac{.518\times3\times38{,}000}{.375\times3.5\times60{,}000}=.75 \text{ for the rivets.}$$

As the comparative strength of the plate, when punched or drilled ready for use, remains the same, or .76, the strength of the joint must be taken at .75 which is practically the same as before.

Using the same proportions of joint for plates of 50,000 pounds tensile strength raises the comparative strength of rivets.

$$\frac{518\times3\times38{,}000}{.375\times3.5\times50{,}000}=.90$$

Therefore this joint may be taken at .75 for all plates mentioned in the foregoing tables, so far as their tensile strength is concerned.

Modern practice in steam engineering, calling for high boiler pressures, has proved that every form of lap joint is unreliable, owing to the tendency to bend the plate as shown in Fig. 5.

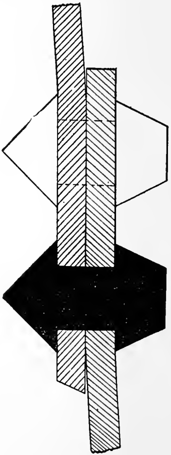


FIG. 5

and as this is repeated every time that pressure is raised and removed, it has caused disastrous failure in many cases.

To overcome this objection the double strap butt joint is used, a very simple form of which is shown in Fig. 6. Such

FIG 6

a joint is very efficient even when only a single row of rivets is used, because the rivets are in double shear, or in other words they must be sheared in two places before they fail. If the strength in single shear is 38,000 pounds it is raised to 70,300 pounds for double shear if we consider double shear equal 1.85 single shear. For illustration of the strength of this form of joint, the following proportions are taken.

Pitch of rivets	3 inches.
Diameter of rivets	.⅞ "
Number of rows	1
Thickness of plate	.375
Tensile strength	50,000 pounds.

$$\frac{3-.875}{.3}=.70 \text{ for the plates.}$$

$$\frac{.601\times1\times70{,}300}{.375\times3\times50{,}000}=.75 \text{ for the rivets.}$$

The strength of this single riveted joint is thus shown to be .70 of the strength of the solid plates.

Increasing the pitch and adding another row of rivets makes a better joint, for which the following proportions are assumed:

Pitch of rivets	3.5 inches.
Diameter of rivets	¾ "
Number of rows	2 "
Thickness of plate	.375 "
Tensile strength	60,000 pounds.

$$\frac{3.5-.75}{3.5}=.78 \text{ for the plates.}$$

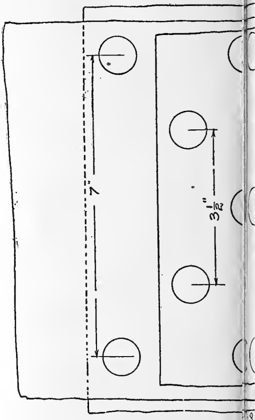
$$\frac{.44\times2\times70{,}300}{.375\times3.5\times60{,}000}=.78 \text{ for the rivets.}$$

In this case the strength of both plates and rivets equals .78 of the solid plates. Here the tensile strength is 60,000 pounds while in the preceding it is 50,000, but it may be taken at any less value in either case without bad effect on the joint, as such changes only raise the percentage of strength of the rivets.

This joint may be still further im-

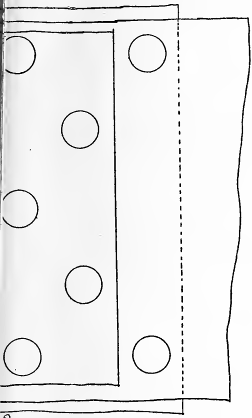
FIG. 7

proved by extending the inner strap as illustrated in Fig. 7, and adding an inner row of rivets as shown in Fig. 8 with double pitch. For illustration of the principles involved, the following proportions are assumed for this joint:



Pitch of outer rivets	3.5 inches.
" " inner "	7 "
Diameter of rivets	¾ "
Thickness of plates	.375 "
Tensile strength	60,000 pounds.

To determine the strength of this joint use the following rules: From the pitch of the inner row of rivets (or the double pitch), subtract the area of one rivet and divide the remainder by the double pitch. The quotient is the strength of the plates.



8.

Multiply the combined area of all rivets in double shear by 70,300. Multiply the combined area of all rivets in single shear by 38,000 and add the two products together. Divide the sum by the strength of the section of plate as found by multiplying the area of plate between centers of the rivets, by the tensile strength. The quotient is the percentage of strength of the rivets.

Stated as a formula these rules appear as follows:

$$\frac{P - D}{P} = S$$

P = Double pitch, or pitch of the inner row.
D = Diameter of rivet.
S = Percentage of strength of plate.

$$\frac{(A \times R \times 70{,}300) + (E \times V \times 38{,}000)}{T \times P \times H} = S$$

A = Area of one rivet in double shear.
R = Number of rivets in double shear.
E = Area of rivets in single shear.
V = Number of rivets in single shear.
T = Thickness of plate.
P = Pitch of rivets.
H = Tensile strength.
S = Strength of rivets.

Application of these formulas to the above example results as follows:

$$\frac{7 - .75}{7} = .89 \text{ for the plates.}$$

$$\frac{(.44 \times 4 \times 70{,}300) + (.44 \times 1 \times 38{,}000)}{.375 \times 7 \times 60{,}000} =$$

$$\frac{140{,}448}{157{,}500} = .89 \text{ for the rivets.}$$

While it is not necessary to make the straps as thick as the plates, they must not be less than $\frac{5}{8}$ of their thickness, therefore for $\frac{3}{8}$ inch plates they should be $.375 \times .625 = .234$ inch or say, $\frac{1}{4}$ inch thick.

THE U. S. GOVERNMENT RULE FOR THE STRENGTH OF RIVETED JOINTS, IRON PLATES AND IRON RIVETS

As these rules are similar to the foregoing they will be stated as formulas only.

$$\frac{P - D}{P} = S$$

P = Pitch of rivets.
D = Diameter of rivets.
S = Strength of plates.

$$\frac{A \times R}{T \times P} = S$$

A = Area of rivet.
R = Number of rows.
T = Thickness of plates.
P = Pitch of rivets.
S = Strength of rivets.

As neither the shearing strength of rivets, nor the tensile strength of plates are mentioned, it shows that they are equal for iron plates and iron rivets.

The above formulas apply to single, double and triple riveted lap joints.

STEEL PLATES AND STEEL RIVETS

To determine the strength of plates at the joint proceed as for iron plates.

The strength of rivets is determined by the following formula:

$$\frac{A \times R \times .8}{T \times P} = S$$

This is the same as for iron rivets except that the constant .8 is added, because the shearing strength of steel rivets is taken as .8 of the tensile strength of the plate.

In preceding rules and formulas the shearing strength of rivets is taken as a certain fixed value and while this may not seem to be correct for all cases, still it is the lowest value determined by several tests, hence a riveted joint that is made in a workmanlike manner should always show the value given and more in many cases.

There is a difference between the diameter of a rivet and the diameter of the hole before the rivet is driven into place, but when the joint is finished the hole is or ought to be, entirely filled by the rivet, hence the diameter of the hole is used in these calculations.

DOUBLE BUTT-STRAP JOINTS WITH STEEL PLATES AND STEEL RIVETS

DOUBLE RIVETED

The percentage of strength of plate is found as in preceding examples. For strength of rivets the following formula applies:

$$\frac{A \times R \times 1.75 \times .8}{T \times P} = S$$

A = Area of rivets which are in double shear.

R = Number of rows, which is 2 in this case.

1.75 = A constant for rivets in double shear. This represents their strength compared with those in single shear.

.8 = A constant as above mentioned.

T = Thickness of plate.

P = Pitch of rivets.

S = Strength of rivets.

TRIPLE RIVETED

Strength of plates is determined by the above rule for other joints. The strength of rivets is determined by the preceding rule for double riveted joints, except that there are 2.5 rows instead of 2. This assumes that the inner row of rivets has a double pitch or else that the inside strap is long enough for three rows, and the inner strap for two rows of rivets.

When designing riveted joints for steam boilers it is necessary to avoid very narrow pitches as they bring the rivet holes too near together for good results. Too wide pitches are also detrimental, as the plate will spring between the rivets, and the joint will not remain tight when calked.

It is sometimes claimed that all calculations given in books are applicable to horizontal fire tube boilers only, but this is a mistaken idea, as they must be used in calculating the strength of all vertical fire tube boilers and many of the water tube type contain steam and water drums that are shells carrying internal pressure, and are fitted with riveted joints the same as fire tube boilers.

BRACING FLAT SURFACES

All flat surfaces in steam boilers, except those of small area, require braces to support them under ordinary working pressures, therefore it becomes necessary to know how much pressure a given surface can safely carry. These surfaces are strong enough to carry a light pressure without bracing, but this fact is not generally taken into consideration when calculating the strength of these parts, because if there is an error in such a plan it is on the safe side, consequently it can do no harm.

It is assumed that all of the pressure is carried by the braces, or stay bolts which are only short braces, therefore, the safe load for them must be known and not exceeded in practice.

The following rule determines the safe working pressure for stay bolts and braces where the plates are not more than 7/16 inch thick:

Multiply the area of the brace by 6,000 and divide the product by the horizontal multiplied by the vertical pitch in inches. The quotient is the safe working pressure.

Analysis of this rule shows that braces are limited to a strain of 6,000 pounds per square inch of sectional area, which affords a liberal factor of safety as the tensile strength of such iron ought to be at least 45,000 pounds.

Where this rule is applied to threaded stay bolts, the area must be calculated from the diameter at the base of the thread, as that represents the real strength of it. For illustration of this rule the following example is given:

Diameter of stay bolt	.75 inch
Area " " "	.44 square inch
Horizontal pitch	5 inches
Vertical pitch	6 "

Then $\frac{.44 \times 6{,}000}{5 \times 6} = 88$ pounds safe working pressure.

If any form of braces are located too far apart, the plate between them will bulge and finally cause leaks or perhaps a disastrous rupture, consequently the safe load for these plates must be determined and the actual working load kept below the safe limit. This may be determined by the following rule, for plates not more than 7/16 inch thick.

Multiply 28,672 by the square of the thickness of the plate and divide the product by the horizontal multiplied by the vertical pitch. The quotient is the safe working pressure.

For example, take the water leg of a locomotive boiler made in the following proportions:

Thickness of plate	.375 inch
Horizontal pitch	6 inches
Vertical "	6 "

Then

$$\frac{28{,}672 \times (.375 \times .375)}{6 \times 6} = 112$$ pounds safe pressure.

Analysis of this rule shows that as the pitch increases, the safe pressure decreases, which is logical. If the horizontal and vertical pitches were always equal the rule would read "square the pitch," but they sometimes differ in practice, hence the rule covers any difference that is desired or required.

Where the plates are more than $\frac{7}{16}$ inch thick, the same rule applies if the safe pressure is based on the diameter and pitch of stay bolts, but larger bolts and greater pitches are adopted in order to be consistent with the increased thickness of plate, as the following example illustrates:

Diameter of stay bolt	$1\frac{1}{2}$ inches.
Area " " "	1.767 square in.
Horizontal pitch	9.5 inches.
Vertical "	9.5 "

Then

$$\frac{1.767 \times 6{,}000}{9.5 \times 9.5} = 117 \text{ pounds safe pressure.}$$

The following table of areas of circles from 1 to 2 inches in diameter is given for convenience in making calculations that are based on the foregoing rule:

AREA OF CIRCLES FROM 1 TO 2 INCHES

Diameter	Decimal	Area
1	1.0000	.785
$1\frac{1}{16}$	1.0625	.886
$1\frac{1}{8}$	1.125	.994
$1\frac{3}{16}$	1.1875	1.107
$1\frac{1}{4}$	1.25	1.227
$1\frac{5}{16}$	1.3125	1.353
$1\frac{3}{8}$	1.375	1.484
$1\frac{7}{16}$	1.4375	1.622
$1\frac{1}{2}$	1.5	1.767
$1\frac{9}{16}$	1.5625	1.917
$1\frac{5}{8}$	1.625	2.073
$1\frac{11}{16}$	1.6875	2.236
$1\frac{3}{4}$	1.75	2.405
$1\frac{13}{16}$	1.8125	2.580
$1\frac{7}{8}$	1.875	2.761
$1\frac{15}{16}$	1.9375	2.948
2	2.0000	3.141

Where the safe pressure is based on the thickness of the plate and the pitch of stay bolts, the following rule applies:

Multiply 30,720 by the square of the thickness of the plate and divide the product by the horizontal multiplied by the vertical pitch.

This rule contains a larger constant number than the former given for this purpose, which is the only difference between them. For illustration assume the following proportions:

Thickness of plate	.625 inch
Horizontal pitch	10 inches
Vertical "	10 "

Then

$$\frac{30{,}720 \times (.625 \times .625)}{10 \times 10} = 120 \text{ pounds safe}$$

pressure.

The stay bolts for this case should not be less than 1⅝ inches in diameter, because the strain to be supported is $10 \times 10 \times 120 = 12{,}000$ pounds and as the strain on stay bolts is limited to 6,000 pounds per square inch of sectional area, these must have an area of at least 2 square inches, and the nearest to this according to the table is 1⅝ inches in diameter.

The following table contains the safe loads for stay bolts and braces from ½ to 2 inches in diameter. The first column gives the diameter, the second is the safe load at 6,000 pounds per square inch sectional area, which is the U. S. Government rule, but inasmuch as this affords a large factor of safety, the Hartford Steam Boiler Inspection and Insurance Co. allow 7,500 pounds and this is found in the third column.

SAFE LOADS FOR STAY BOLTS AND BRACES

Diameter in inches	Safe load 6,000 pounds	Safe load 7,500 pounds
½	1,176	1,470
⅝	1,836	2,295
¾	2,646	3,307
⅞	3,606	4,507
1	4,710	5,887
1⅛	5,964	7,355
1¼	7,362	9,202
1⅜	8,904	11,130
1½	10,602	13,252
1⅝	12,438	15,547
1¾	14,430	18,037
1⅞	16,566	20,707
2	18,840	23,557

SURFACE SUPPORTED BY STAY BOLTS AND BRACES

Every stay bolt and brace occupies more or less space on the plate that is to be supported, the amount varying with the diameter, and this space is covered, therefore it is not subjected to pressure. It is customary to ignore this fact when making these calculations, because if there is an error it is on the safe side, hence no harm can result.

When braces contain toggle joints they should be well made, so that there will be no lost motion to be taken up when pressure is put on, and let out when it is removed, thus causing the plate to bend many times, the ultimate result of which is that the metal is weakened until it may fail under an ordinary working pressure, causing loss of life and property.

The following table contains the number of square inches on a flat surface that stay bolts and braces will safely support, the diameter ranging from ½ inch to 2 inches and the pressure from 80 to 120 pounds. The strain is limited to 6,000 pounds per square inch of sectional area.

Following this is another table which is the same except that the strain is limited to 7,500 pounds per square inch of sectional area.

To use these tables proceed as follows: Suppose that a space on the water leg of a locomotive boiler that is 24×56 inches, is to be made strong enough to carry 90 pounds pressure and stay bolts $\frac{7}{8}$ inch in diameter are to be used. How many will be required? The space contains $24 \times 56 = 1{,}344$ square inches.

The first table, limiting the stress to 6,000 pounds, shows that each $\frac{7}{8}$ inch stay bolt will support 40 square inches. under 90 pounds pressure. Then $1{,}344 \div 40 = 33$ with a remainder of 24 inches to be provided for, therefore it will require 34 stay bolts.

If the strain is limited to 7,500 pounds an examination of the next table shows that a $\frac{7}{8}$ inch stay bolt will support 50 square inches under 90 pounds pressure. Then, $1{,}344 \div 50 = 26$ with a remainder of 44 square inches, therefore 27 stay bolts will be required.

SQUARE INCHES OF SURFACE SUPPORTED BY STAY BOLTS AND BRACES

Diameter of brace in inches	Strain limited to 6,000 pounds Boiler Pressure				
	80	90	100	110	120
½	14.7	13.0	11.7	10.7	9.8
⅝	23.0	20.4	18.3	16.7	15.3
¾	33.0	29.4	26.4	24.0	22.0
⅞	45.0	40.0	36.0	32.7	30.0
1	58.8	52.3	47.1	42.8	39.2
1⅛	74.5	66.2	59.6	54.2	49.7
1¼	92.0	81.8	73.6	66.9	61.3
1⅜	111.3	98.9	89.0	80.9	74.2
1½	132.5	117.8	106.0	96.3	88.3
1⅝	155.4	138.2	124.3	113.0	103.6
1¾	180.3	160.3	144.3	131.1	120.2
1⅞	207.0	184.0	165.6	150.6	138.0
2	235.5	209 3	188.4	171.2	157.0

SQUARE INCHES OF SURFACE SUPPORTED BY STAY BOLTS AND BRACES

Diameter of brace in inches	Strain Limited to 7,500 pounds Boiler Pressure				
	80	90	100	110	120
½	18.3	16.3	14.7	13.3	12.2
⅝	28.7	25.5	22.9	20.8	19.1
¾	41.3	36.7	33.0	30.0	27.5
⅞	56.3	50.0	45.0	40.9	37.5
1	73.5	65.4	58.8	53.5	49.0
1⅛	91.9	81.7	73.5	66.8	61.2
1¼	115.0	102.2	92.0	83.6	76.6
1⅜	139.1	123.6	111.3	101.1	92.7
1½	165.6	147.2	132.5	120.4	110.4
1⅝	194.3	172.7	155.4	141.3	129.5
1¾	225.4	200.4	180.3	163.9	150.2
1⅞	258.8	230.0	207.0	188.2	172.5
2	294.4	261.7	235.5	214.1	196.3

THE ANGULARITY OF BRACES

As a general rule, to which there may be a few exceptions, stay bolts are set at right angles to the plates, and all of the foregoing rules and directions are based on this condition, but braces must frequently be located at angles that are less than 90 degrees, and the effect of this is to put a greater load on the brace, although the same surface is supported and the same pressure carried.

Fig. 9 illustrates a stay bolt in place

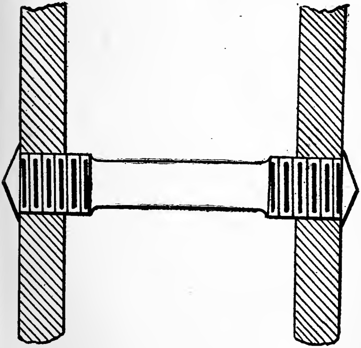


FIG. 9

at an angle of 90 degrees to the plate, therefore the pressure puts a strain on the brace that is represented by the surface of plate multiplied by the pressure per square inch.

Fig. 10 shows a brace that is located at an angle of 45 degrees to the boiler head that it supports. The load on this

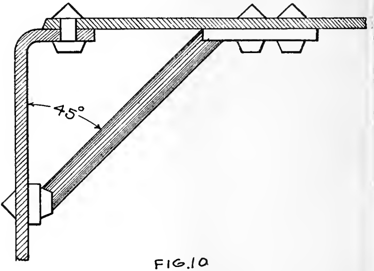


FIG. 10

brace is determined by the surface supported in square inches, multiplied by the pressure per square inch, divided by the natural sine of the angle.

Suppose that this brace is 1¼ inches in diameter and the strain is limited to 6,000 pounds per square inch of sectional area. According to the table of safe loads this brace will carry 7,362 pounds which will be secured in accordance with the next table if it supports 73.6 square inches at 100 pounds pressure.

The natural sine of an angle of 45

degrees is .7071, therefore the true load on this brace under these conditions is $7{,}362 \div .7071. = 10{,}411$ pounds, which is an overload even if the limit is placed at 7,500 pounds per square inch of sectional area.

To overcome this objection it becomes necessary to ascertain the number of square inches that can be allowed without exceeding the safe limit. This is accomplished by multiplying the number of square inches that could be supported safely if the brace stood at an angle of 90 degrees as stated in the table, by the natural size of the angle. If this number is 73.6 and the brace stands at an angle of 45 degrees, the natural sine of which is .7071 then the surface that can safely be allowed to this brace is $73.6 \times .7071 = 52$ square inches. While braces are not usually located at such an acute angle, this example illustrates the effect of angularity and shows that it should not be neglected where accuracy is desired. If the angle is increased to 75 degrees, the space becomes 71 square inches.

The following table contains the natural sines of angles from 10 to 90 degrees which fully covers all that can be required for use in connection with boiler bracing, and the use of the data given in it has already been fully explained.

This is an important branch of the subject of boiler bracing, and is earnestly recommended to those who have not heretofore given it due attention.

NATURAL SINES OF ANGLES

Angle	Sine	Angle	Sine
10	.1736	51	.7771
11	.1908	52	.7880
12	2079	53	.7986
13	.2249	54	.8090
14	.2419	55	.8191
15	.2588	56	.8290
16	.2756	57	.8386
17	.2923	58	.8480
18	.3090	59	.8571
19	3255	60	.8660
20	.3420	61	.8746
21	.3583	62	.8829
22	.3746	63	.8910
23	.3907	64	.8987
24	.4067	65	.9063
25	.4226	66	.9135
26	.4383	67	.9205
27	.4539	68	.9271
28	.4694	69	.9335
29	.4848	70	.9396
30	.5000	71	.9455
31	.5150	72	.9510
32	.5299	73	.9563
33	.5446	74	.9612
34	.5591	75	.9659
35	.5735	76	.9703
36	.5877	77	.9743
37	.6018	78	.9781
38	.6156	79	.9816
39	.6293	80	.9848
40	.6427	81	.9876
41	.6560	82	.9902
42	.6691	83	.9925
43	.6820	84	.9945
44	.6946	85	.9961
45	.7071	86	.9975
46	.7193	87	.9986
47	.7313	88	.9993
48	.7431	89	.9998
49	.7547	90	1.0000
50	.7660		

BRACING THE HEADS OF TUBULAR BOILERS

Every tube that is put into the head of a tubular boiler acts as a brace of more or less efficiency, and as a large portion of the head is removed in making places for these tubes, they are sufficient to hold what is left except a small space below them which usually requires two braces, and a larger space above them which assumes the form of a segment of a circle.

It is more difficult to compute the area of this space than to determine the number of square inches on the water leg of a locomotive boiler, or on any other part that can be put into the form of a square, as there are several points to be taken into consideration.

These heads are usually about one-half inch thick, and taking this as a basis the head is self-supporting for 2 inches above the upper row of tubes, and for 3 inches from the shell, consequently if we take the head of a 72-inch boiler in which the upper tubes are 7 inches above the center line, the height of the segment to be braced is 24 inches, and it is part of a circle $72-(3\times2)=66$ inches in diameter, as 6 inches must be subtracted for the self-supporting portion above mentioned. The chord or base is 63.375 inches long as shown by measurement on a full sized drawing. This is illustrated in Fig. 11.

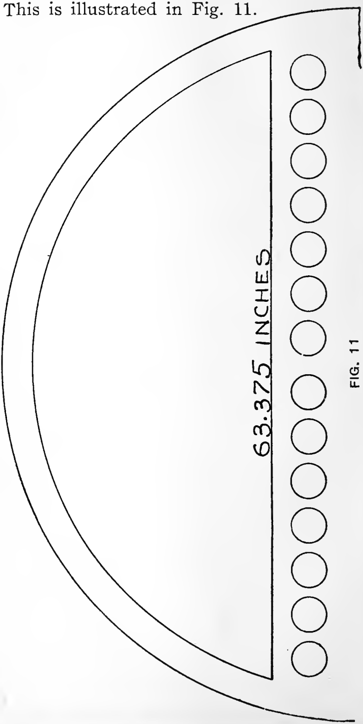


FIG. 11

RULE FOR CALCULATING THE AREA OF A SEGMENT OF A CIRCLE

Ascertain the area of a sector of the circle having the same arc as the segment. Determine the area of the triangle formed by the chord of the segment and the radii of the sector. Subtract the latter from the former and the remainder is the area of the segment.

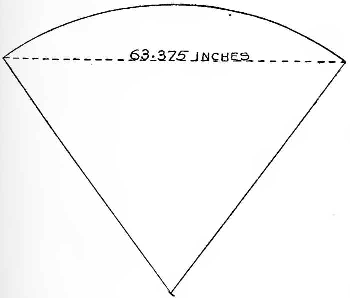


FIG. 12

Fig. 12 represents a sector the upper part of which constitutes the segment. Applying this rule shows that the segment in this case contains 1,125 square inches. It is assumed that the head is full 72 inches in diameter in order to agree with calculations made to determine the bursting and safe working pressure of boiler shells as the diameters given are determined by internal measurements.

Fig. 13 illustrates another rule for determining the area of the segment to be braced, as it was laid out full size and divided into equal spaces 1 inch wide by ordinates, commencing at the center and working both ways. The illustration is reduced for convenience, but the correct proportions are retained.

There are 63 ordinates and their total length is 1,127.7 inches, therefore the average height of this segment is $1{,}127.7 \div 63 = 17.9$ inches. As it is 63.375 inches long it contains $17.9 \times 63.375 =$ 1,134 square inches. The rule on which this is based may be stated as follows:

Draw a portion of a circle that is 6 inches smaller than the diameter of the boiler, to form the arc. Draw a horizontal line 2 inches above the upper row of tubes to form the chord. Begin at the center and lay off ordinates 1 inch apart, extending from the chord to the arc. Determine the length of each, add them together, divide by the number of ordinates and the quotient is the average height. Multiply this by the length and the product is the number of square inches contained in the segment.

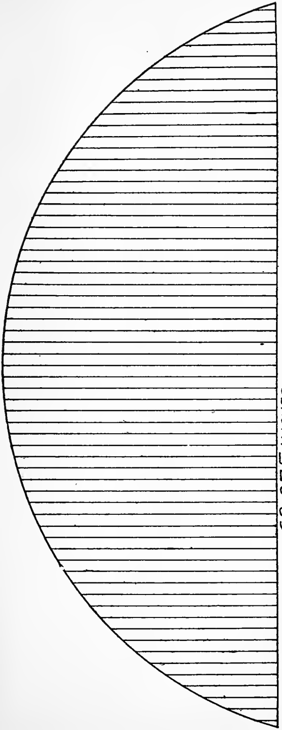


FIG 13

It is difficult or impossible to secure exactly the same result by two different rules, but the difference is not sufficient to affect the number of braces required.

EXPLANATION OF THE FOLLOWING TABLE

This table contains the areas of segments to be braced on tubular boiler heads from 42 to 84 inches in diameter. The distance from the center of the boiler to the top of the tubes is given, and it will be noted that this arrangement of tubes gives a very liberal steam space. If more tubes are put in the heating surface will be increased, but taken as a whole the boiler will not be benefited thereby. The columns in this table contain the following data:

1. Diameter of boiler head.
2. Diameter of circle on which the calculation is based.
3. Distance from center of head to top of tubes.
4. Distance from center of head to chord or base of segment.
5. Length of chord.
6. Greatest height of segment.
7. Average height of segment.
8. Area of segment.

All measurements are stated in inches and areas in square inches.

TABLE OF AREAS OF SEGMENTS DETERMINED BY ORDINATES

1	2	3	4	5	6	7	8
42	36	4	6	34 20	12	8.61	294
48	42	4	6	40.20	15	10.86	436
54	48	5	7	46.00	17	12.36	568
60	54	5	7	52.25	20	14.93	780
66	60	6	8	57.75	22	16.04	926
72	66	7	9	63.37	24	17.90	1,134
78	72	8	10	68.88	26	19.20	1,322
84	78	9	11	74.75	28	20.70	1,547

ANOTHER RULE

A rule that is frequently used to determine the area of a segment of a circle, is expressed briefly in the following formula:

$$\frac{H}{D} \times C \times D^2 = A$$

H = Height of segment.
D = Diameter of circle.
C = A constant taken from a table.
A = Area of segment.

As the height of segment and the diameter of circle are always known or easily determined, this constitutes a very simple formula, but a table of constants is required.

The following table gives the area of segments previously mentioned in order that the reader may compare them with results obtained by the plan of laying them out by ordinates, which is here considered the standard.

AREA OF SEGMENTS
DETERMINED BY THE FOREGOING FORMULA

Diam. of head	$\frac{\text{Height}}{\text{Diam.}}$	Constant	Diam.	Area
42	$^{12}/_{36}$=.333	.2288	1,296	296
48	$^{15}/_{42}$=.357	.2516	1,764	443
54	$^{17}/_{48}$=.354	.2488	2,304	573
60	$^{20}/_{54}$=.370	.2641	2,916	770
66	$^{22}/_{60}$=.367	.2574	3,600	926
72	$^{24}/_{66}$=.363	.2622	4,356	1,142
78	$^{26}/_{72}$=.361	.2555	5,184	1,324
84	$^{28}/_{78}$=.359	.2535	6,084	1,542

The following table contains all of the constants that are required for ordinary cases along this line. The first column contains the quotients found by dividing the greatest height of segment by the diameter of the circle. The second column contains the corresponding constant. When the latter is multiplied by the square of the diameter of the circle, the product is the area of the segment.

TABLE OF CONSTANTS

Height / Diameter	Constant	Height / Diameter	Constant
.150	.0738	.198	.1102
.151	.0745	199	.1110
.152	.0753	.200	.1118
.153	.0760	.201	.1126
154	.0767	.202	.1134
155	.0774	.203	.1142
.156	.0781	.204	.1150
157	.0789	205	.1158
158	.0796	.206	.1166
159	.0803	.207	.1174
160	.0811	.208	.1182
.161	.0818	.209	.1190
.162	.0825	.210	.1199
163	.0833	.211	.1207
.164	.0840	.212	.1215
.165	.0848	.213	.1223
.166	.0855	.214	.1231
.167	.0862	.215	.1239
.168	.0870	.216	.1248
.169	.0877	.217	.1256
.170	.0885	.218	.1264
.171	.0892	.219	.1272
.172	.0900	.220	.1281
.173	.0908	.221	.1289
174	.0915	.222	.1297
.175	.0923	.223	.1306
.176	.0930	.224	.1314
.177	.0938	.225	.1322
.178	.0946	.226	.1331
.179	.0953	.227	.1339
.180	.0961	.228	.1347
.181	.0969	.229	.1356
.182	.0976	.230	.1364
.183	.0984	.231	.1373
.184	0992	.232	.1384
185	.1000	.233	.1390
.186	.1007	.234	.1398
.187	.1015	.235	.1406
.188	.1023	.236	.1415
.189	.1031	.237	.1423
.190	.1039	.238	.1432
.191	.1046	.239	.1440
.192	.1054	.240	.1449
.193	.1062	.241	.1458
.194	.1070	.242	.1466
.195	.1078	.243	.1475
196	.1086	.244	.1483
.197	.1092	.245	.1492

Table of Constants—*Continued*

Height / Diameter	Constant	Height / Diameter	Constant
.246	.1500	.294	.1926
.247	.1509	.295	.1936
.248	.1518	.296	.1945
.249	.1526	.297	.1954
.250	.1535	.298	.1963
.251	.1544	.299	.1972
.252	.1552	.300	.1981
.253	.1561	.301	.1990
.254	.1570	.302	.2000
.255	.1578	.303	.2009
.256	.1587	.304	.2018
.257	.1596	.305	.2027
.258	.1605	.306	.2036
.259	.1613	.307	.2046
.260	.1622	.308	.2053
.261	.1631	.309	.2064
.262	.1640	.310	.2073
.263	.1649	.311	.2083
.264	.1657	.312	.2092
.265	.1666	.313	.2101
.266	.1675	.314	.2110
.267	.1684	.315	.2120
.268	.1693	.316	.2129
.269	.1702	.317	.2138
.270	.1710	.318	.2148
.271	.1719	.319	.2157
.272	.1728	.320	.2166
.273	.1737	.321	.2176
.274	.1746	.322	.2185
.275	.1755	.323	.2194
.276	.1764	.324	.2204
.277	.1773	.325	.2213
.278	.1782	.326	.2222
.279	.1791	.327	.2232
.280	.1800	.328	.2241
.281	.1809	.329	.2250
.282	.1818	.330	.2260
.283	.1827	.331	.2269
.284	.1836	.332	.2279
.285	.1845	.333	.2288
.286	.1854	.334	.2298
.287	.1863	.335	.2307
.288	.1872	.336	.2316
.289	.1881	.337	.2326
.290	.1890	.338	.2335
.291	.1899	.339	.2345
.292	.1906	.340	.2354
.293	.1917	.341	.2364

Table of Constants—*Continued*

Height/Diameter	Constant	Height/Diameter	Constant
.342	.2373	.390	.2835
.343	.2383	.391	.2845
.344	.2392	.392	.2855
.345	.2402	.393	.2865
.346	.2411	.394	.2875
.347	.2421	.395	.2884
.348	.2430	.396	.2894
.349	.2440	.397	.2904
.350	.2449	.398	.2914
.351	.2459	.399	.2923
.352	.2468	.400	.2933
.353	.2478	.401	.2943
.354	.2488	.402	.2953
.355	2497	.403	.2963
.356	.2507	.404	.2972
.357	.2516	.405	.2982
.358	.2526	.406	.2992
.359	.2535	.407	.3002
.360	.2545	.408	.3012
.361	.2555	.409	.3022
.362	.2564	.410	.3031
.363	.2574	.411	.3041
.364	.2583	.412	.3051
.365	.2593	.413	.3061
.366	.2603	.414	.3071
.367	.2612	415	.3081
.368	.2622	.416	.3091
.369	.2632	.417	.3100
.370	.2641	.418	.3110
.371	.2651	.419	.3120
.372	.2661	.420	.3130
.373	.2670	.421	.3140
.374	.2680	.422	.3150
.375	.2690	.423	.3160
.376	.2699	.424	.3169
.377	.2709	.425	.3179
378	.2719	.426	.3189
.379	.2728	.427	.3199
380	.2738	.428	.3209
.381	.2748	.429	.3219
.382	.2758	.430	.3229
.383	.2767	.431	.3239
.384	.2775	.432	.3249
.385	.2787	.433	.3259
.386	.2796	.434	.3268
.387	.2806	.435	.3278
.388	.2816	.436	.3288
.389	.2826	.437	.3298

Table of Constants—*Continued*

Height / Diameter	Constant	Height / Diameter	Constant
.438	.3308	.445	.3378
.439	.3318	.446	.3388
.440	.3328	.447	.3398
.441	.3338	.448	.3407
.442	.3348	.449	.3417
.443	.3358	.450	.3427
.444	.3368		

To illustrate the operation of finding the area of a segment by the use of this table, suppose that on the head of an 84 inch boiler, there is a segment 28 inches high to be braced. As the circle is $84 - 6 = 78$ inches in diameter, the height divided by the diameter is $28 \div 78 = .359$ and when this number is found in the table, the constant opposite is .2535. Squaring the diameter of the circle and multiplying by this constant shows that the area of this segment is $78 \times 78 \times .2535 = 1{,}542$ square inches.

DIRECTIONS FOR USING THE FOREGOING TABLES

For illustration, suppose that on the head of a 72-inch boiler there is a segment 24 inches high to be braced, to make it safe at 110 pounds pressure, and it is proposed to use round braces 1¼ inches in diameter, or their equivalent in some other form. They are to be located at an angle of 65 degrees from the head. How many braces will be required?

The table on page 51 shows that if the head of a 72-inch boiler has a segment 24 inches high its area is 1,134 square inches. The table of "Square Inches of Surface Supported by Stay Bolts and Braces, Strain Limited to 6,000 Pounds," shows that a brace $1\frac{1}{4}$ inches in diameter will support 66.9 square inches under 110 pounds pressure, if the brace is located at an angle of 90 degrees from the head. In this case it is at 65 degrees, therefore as the table of "Natural Sines of Angles" shows that the sine of 65 is .9063 this brace will safely support $66.9 \times .9063 = 60.6$ square inches.

As the total area to be supported is 1,134 square inches, it will require $1{,}134 \div 60.6 = 18.7$ braces, or in practice it would be called 19. If a strain of 7,500 pounds per square inch of sectional area is allowed, each brace will support 83.6 square inches at 90 degrees under 110 pounds pressure as per table. Taking it at an angle of 65 degrees it will support $83.6 \times .9063 = 75.7$ square inches therefore it will require $1{,}134 \div 75.7 = 15$ braces under these conditions. These tables provide for the solution of any problem along this line that is found under ordinary working conditions.

It is not a difficult matter to adapt them to other conditions not mentioned directly, by the following plan. Under head of "Square Inches Supported by Stay Bolts and Braces. Strain Limited to 7,500 pounds," boiler pressures from 80 to 120 pounds are included, but if a brace will support 18.3 square inches under 80 pounds pressure, it will support $18.3 \times 2 = 36.3$ square inches under 40 pounds. If it will support 200.4 square inches under 90 pounds pressure, it will support $200.4 \div 2 = 100.2$ square inches under 180 pounds pressure, and many other combinations can be secured from this table.

The "Table of Areas of Segments" can be adapted to other sizes, as follows: Column 1 includes a boiler head 66 inches in diameter in which the distance from center of head to top of tubes is 6 inches. (See column 3.) The distance to the chord of this segment is 8 inches, the length of the chord is 57.75 inches, and the area is 926 square inches.

Suppose that in another case the greatest height of segment is 21 inches or 1 inch less than given in column 6. As the chord in the table is 57.75 inches long, the area will be nearly 57.75 square inches less. If the segment is 1 inch higher or 23 inches, its area is about 57.75 square inches more. These results are practically correct for the calculations to which they belong, as the slight difference due to approximate measurements will not change the number of braces required in a given case.

BOILER FLUES

SAFE PRESSURE FOR FLUES

Flues in a shell boiler are subject to collapsing pressure, hence they must be treated in a different way from parts that resist bursting pressure. When calculating the safe internal pressure of the shell of a boiler it is not necessary to take the length of it into consideration, but experiments on flues show that when collapsing pressure is under consideration, the length becomes a factor. The following rule determines the safe pressure of flues:

Multiply the square of the thickness of the iron by 806,300. Divide the product by the diameter multiplied by the length and by 3. The quotient is the safe pressure.

For illustration take a flue made of iron $\frac{1}{4}$ inch thick with riveted seams. It is 12 inches in diameter and 20 ft. long,

$\frac{.25\times.25\times806{,}300}{12\times20\times3}=70$ pounds safe pressure.

THICKNESS OF FLUES

The required thickness of flues for given conditions, is determined by the following rule:

Multiply the diameter in inches, by the length in feet, by the steam pressure and by 3. Divide the final product by 806,300 and extract the square root of the quotient.

The above mentioned example gives this result:

$$\sqrt{\frac{20\times12\times70\times3}{806{,}300}}=.25 \text{ inches thick.}$$

DIAMETER OF FLUES

To determine the proper diameter of a flue for given conditions the following rule applies:

Multiply the square of the thickness by 806,300. Divide the product by the length in feet, multipled by the steam pressure and by 3. The quotient is the diameter in inches.

Using the foregoing, example results as follows:

$$\frac{.25\times.25\times806{,}300}{20\times70\times3}=12 \text{ inches diameter.}$$

LENGTH OF FLUES

As the length of a flue is a factor in determining the safe pressure, it becomes necessary in certain cases to determine how long a flue can be made without exceeding the safe limit. This point can be decided by the next rule.

Multiply the square of the thickness by 806,300 and divide by the diameter multiplied by the safe pressure and by **3.** The quotient is the length in feet.

Applying this to the same example gives the following result:

$$\frac{.25 \times .25 \times 806{,}300}{12 \times 70 \times 3} = 20 \text{ feet long.}$$

In all the foregoing calculations concerning boiler flues, a factor of safety of 3 is used, which seems to have proved sufficient for all practical use in the judgment of Inspectors in the U. S. Government service, but if a greater factor is desired it may be substituted, giving results in accordance with the change.

The length of a flue, so far as its safe pressure is concerned is the distance between supports. If it is a plain flue this means the extreme length, but if it is made in sections with flanged ends, each of these sections is called the length, and if re-inforcing rings are used, the distance between them is taken as the length in the above rules for safe pressure, etc.

Furthermore, throughout this work a boiler flue is taken as being 4½ inches or more in diameter while tubes are 4 inches or less. In accordance with this assumption, the foregoing rules are not recommended for anything less than 5 inches, and they probably give very conservative results in practice.

There does not seem to be any really satisfactory data of a simple nature concerning the strength of boiler tubes, but fortunately they are strong enough to safely withstand much more pressure than is given for boiler shells of various diameters, consequently they are safer than other parts.

The same reasoning applies to tubes for water tube boilers which are subjected to bursting pressure, hence they are properly termed "safety boilers" provided they do not include a large steam drum.

As a general rule a water tube does comparatively little damage when it bursts, but exceptions to this rule are not unknown.

The next table contains the standard dimensions and weight per linear foot of boiler flues from 4½ to 21 inches in diameter.

When referring to the size of standard boiler flues and tubes it is well to remember that when the diameter is mentioned it is the outside diameter, consequently when ordering flue cleaners this fact must be taken into consideration. If the internal diameter is meant in any special case it should be plainly stated in order to avoid misunderstandings.

Following this is a table of dimensions of tubes from 1 to 4 inches in diameter, and this refers to the external diameter as above mentioned.

SIZES OF STANDARD BOILER FLUES FROM 4½ TO 21 INCHES

Diameter		Thickness	Circumference		Weight
Ext.	Int.		Ext.	Int.	
4½	4.232	.134	14.137	13.295	6.17
5	4.704	.148	15.708	14.778	7.58
6	5.670	.165	18.850	17.813	10.16
7	6.670	.165	21.991	20.954	11.90
8	7.670	.165	25.133	24.006	13.65
9	8.640	.180	28.274	27.143	16.76
10	9.594	.203	31.416	30.141	21.00
11	10.560	.220	34.558	33.175	25.00
12	11.542	.229	37.699	36.260	28.50
13	12.524	.238	40.841	39.345	32.06
14	13.504	.248	43.982	42.424	36.00
15	14.482	.259	47.124	45.497	40.60
16	15.458	.271	50.266	48.563	45.20
17	16.432	.284	53.407	51.623	49.90
18	17.416	.292	56.549	54.714	54.82
19	18.400	.300	59.690	57.805	59.48
20	19.360	.320	62.832	60.821	66.77
21	20.320	.340	65.974	63.837	73.40

SIZES OF STANDARD BOILER TUBES FROM 1 TO 4 INCHES

Diameter		Thick-ness	Circumference		Weight
Ext.	Int.		Ext.	Int.	
1	.810	.095	3.142	2.545	.90
1¼	1.060	.095	3.927	3.330	1.15
1½	1.310	.095	4.712	4.115	1.40
1¾	1.360	.095	5.498	4.901	1.65
2	1.810	.095	6.283	5.686	1.91
2¼	2.060	.095	7.069	6.472	2.16
2½	2.282	.109	7.854	7.169	2.75
2¾	2.532	.109	8.639	7.995	3.04
3	2.782	.109	9.425	8.740	3.33
3¼	3.010	.120	10.210	9.456	3.96
3½	3.260	.120	10.996	10.242	4.28
3¾	3.510	.120	11.781	11.027	4.60
4	3.732	.134	12.566	11.724	5.47

EXTRA STRONG BOILER TUBES

There are many cases where extra strong tubes are required for special purposes and they are made to meet this demand. As a thick tube will hold more than a thin one in the capacity of a brace, they are sometimes used near the center of large boiler heads on this account. They should be used in some of the horizontal water tube boilers now in the market, in order to make them more reliable, as more material is needed in a water tube than in a fire tube for the same pressure, as the former must support the weight of water from which the latter is free. However, the water tube is supported between the ends, while a fire tube cannot be so strengthened. The next table gives dimensions of extra strong boiler tubes.

SIZE OF EXTRA STRONG BOILER TUBES

Nominal diameter	Outside diameter	Inside diameter	Thickness	Thickness double extra strong	Inside diam. double extra strong
½	.84	.542	.149	.298	.244
¾	1.05	.736	.157	.314	.422
1	1.315	.951	.182	.364	.587
1¼	1.66	1.272	.194	.388	.884
1½	1.9	1.494	.203	.406	1.088
2	2.375	1.933	.221	.442	1.491
2½	2.875	2.315	.280	.560	1.755
3	3.5	2.892	.304	.608	2.284
3½	4.0	3.358	.321	.642	2.716
4	4.5	3.818	341	.682	3.136

HEATING SURFACE OF STEAM BOILERS

The heating surface of a steam boiler consists of the parts that are in contact with the fire or the hot gases produced by it, on one side, and are covered by water on the other. It may be possible to show an exception to this rule in the case of a vertical fire tube boiler in which the tubes extend through the steam space to the upper head. For a comparatively short distance they are not covered by water but they tend to superheat the steam, or at least to evaporate particles of water that are taken up by the steam as it ascends from the water surface, hence the surface above the water level in such cases is of more or less value.

The capacity of a boiler so far as the evaporation of water is concerned, depends on the amount of heating surface it contains and the rapidity with which the water circulates through it. The number of square feet of heating surface is easily determined, but the efficiency of the circulation can only be determined by actual trial of various kinds of boilers.

After it is once ascertained for a certain design, all others of the same kind may be depended upon to show the same results, provided the conditions are alike.

The amount of heating surface offers a suggestion concerning the power that a boiler can develop, but it should not be taken as definite information concerning what it can do or of what it actually is doing in practice.

A boiler may have heating surface enough to develop 100 horse power according to standard rating for given conditions, but the actual conditions may be more favorable for evaporating water, hence it can be made to develop 150 horse power, although it would probably result in low efficiency so far as the amount of water evaporated per pound of coal is concerned, and repair bills may be large.

On the other hand a boiler may have enough heating surface to warrant its rating at 100 horse power, yet owing to light demand for steam it may not develop more than 50 horse power, hence this must be taken into consideration when noting the amount of coal burned in different places.

Some heating surface is much more efficient than others, but this is not considered when the amount is stated, because it is difficult or impossible to always tell just which is the most efficient, and to draw the line of separation between the two kinds. Where heat is travelling in one direction and water in the opposite, the best results are secured.

When calculating the heating surface in the shell of a tubular boiler, or in the steam drum of one of the water tube type, multiply the circumference in feet by the length and divide the product by 2. In some cases the actual surface with which heat comes in contact may exceed the result secured by the above rule, because the brick work may be drawn in to touch the shell above the center line, thus increasing the surface by a few square feet. Where accuracy is required it is necessary to measure the exact distance from the wall on one side to a similar place on the other, following the curve of the shell. This will give the width, and multiplying it by the length gives the square feet of heating surface in the shell.

The tubes seem to present a more complicated problem, but this can be greatly modified by modern methods. If we find the circumference of a tube in inches, then multiply it by the length in inches and divide the product by 144, the quotient will be the number of square feet in the tube, but it requires the use of many figures, and is therefore too long for convenience.

The use of a table shortens the process, and thus proves satisfactory to the busy engineer. The following contains the required information, and further explanation follows it.

Information in this table is classified as follows in the several columns:

1. External diameter in inches.
2. External area in square inches.
3. Internal " " "
4. Length of tube required for one square foot of heating surface, inside measurement.
5. Length of tube required for one square foot of heating surface, outside measurement.

6. Length of tube required for one square foot of heating surface, based on the mean diameter, or the external diameter less the thickness.

BOILER TUBES

1	2	3	4	5	6
1	.785	.515	4.479	3.820	4.149
1¼	1.227	.882	3.604	3.056	3.330
1½	1.767	1.348	2.916	2.547	2.730
1¾	2.405	1.911	2.448	2.183	2.316
2	3.142	2.573	2.110	1.910	2.010
2¼	3.976	3.333	1.854	1.698	1.770
2½	4.909	4.000	1.674	1.528	1.601
2¾	5.940	5.035	1.508	1.389	1.449
3	7.069	6.079	1.373	1.273	1.329
3¼	8.296	7.116	1.269	1.175	1.222
3½	9.621	8.347	1.172	1.091	1.132
3¾	11.045	9.676	1.088	1.019	1.054
4	12.566	10.939	1.024	.955	.990

Suppose that a fire tube or tubular boiler is fitted with 96 3-inch tubes, 16 feet long. How many square feet of heating surface are in the tubes? The total length is $16 \times 96 = 1{,}536$ feet. As the fire can only come in contact with the internal surface of these tubes, the total length is to be divided by 1.373 because column 4 in the foregoing table shows that it requires 1.373 feet in length to make one square foot on the tube. The heating surface in this case is $1{,}536 \div 1.373 = 1{,}118$ square feet.

A water tube boiler contains 126 tubes 4 inches external diameter 20 feet long.

How many square feet of heating surface do they contain? The total length is $20 \times 126 = 2{,}520$ feet. The fire comes in contact with the external surface of these tubes, hence it requires .955 feet in length to make one square foot. (See column 5 in the foregoing table.) Then $2{,}520 \div .955 = 2{,}638$ square feet in these tubes.

If two-thirds of the heads in the above mentioned tubular boiler, minus the area of the tubes, is considered effective heating surface, how many square feet do both heads contain?

The area of a 66-inch circle is $66 \times 66 \times .7854 = 3{,}421.19$ square inches, two-thirds of which is 2,280.78. The external area of a 3-inch tube is 7.069 square inches. (See column 2 in the preceding table.) The combined area of 96 is then $7.069 \times 96 = 678.62$. Then $2{,}280.78 - 678.62 = 1{,}602.16$ square inches which is 15.83 square feet for each, or 31.66 for both heads.

If it is claimed that only the internal area of the tubes should be subtracted, because the thickness of the metal accounts for the remainder, then the area of each tube is 6.079 square inches. (See column 3 in the table.) The combined area is $6.079 \times 96 = 583.58$ square inches and subtracting this from two-thirds of the area of the head leaves $2,280.78 - 583.58 = 1,697.20$ square inches or 11.78 square feet, or 23.56 in both heads. These directions apply to all boiler heads and any number of tubes.

The next table gives a list of boiler heads from 42 to 84 inches in diameter the area of each in square feet, the circumference and one-half of the circumference of the shells of equal diameter in feet.

It enables the reader to readily calculate the heating surface in any boiler head after accounting for the tubes.

For illustration suppose that a boiler head 78 inches in diameter contains 120 three inch tubes, and the heating surface in it is to be determined, taking the whole head. The combined area of the tubes is $7.069 \times 120 \div 144 = 5.89$ square feet.

The table shows that this head contains 33.18 square feet and $33.18 - 5.89 = 27.29$ square feet.

This table may also be used for determining the heating surface in boiler shells. The circumference of each is given in feet, and it follows that this is the number of square feet of heating surface for each foot in length of the boiler.

For illustration suppose that a boiler 84 inches in diameter is 20 feet long. How many square feet of heating surface are in the shell? The table shows that there are 21.99 for each foot or $21.99 \times 20 = 439.8$ in the whole shell.

If only one-half of the shell is taken the table shows that there are 10.99 square feet for each foot in length, or 219.8 square feet for one-half of the shell. In the case of a vertical fire tube boiler, the shell cannot be counted as heating surface because the fire does not touch it, but the furnace contains some that is very effective. Suppose that in a given case the circular furnace is 60 inches in diameter. The foregoing table shows that the circumference of this furnace is 15.70 feet, or in other words there are 15.70 square feet for each foot in height, therefore, if it is 4 feet high above the grates there are $15.70 \times 4 = 60.28$ square feet in this part. The surface in the lower head may be determined by the rule already explained.

TABLE OF AREAS OF BOILER HEADS

In Square Feet, the Circumference, and One-half the Circumference in Feet

Diameter in inches	Area in square ft.	Circumference in feet	One-half Circumference in feet
42	9.62	10.99	5.49
44	10.55	11.51	5.75
46	11.54	12.04	6.02
48	12.56	12.56	6.28
50	13.63	13.09	6.54
52	14.74	13.61	6.80
54	15.90	14.13	7.06
56	17.10	14.66	7.33
58	18.34	15.18	7.59
60	19.63	15.70	7.85
62	20.96	16.23	8.11
64	22.34	16.75	8.37
66	23.75	17.27	8.63
68	25.22	17.80	8.90
70	26.72	18.32	9.16
72	28.27	18.84	9.42
74	29.86	19.37	9.68
76	31.50	19.89	9.94
78	33.18	20.42	10.21
80	34.90	20.94	10.47
82	36.67	21.46	10.73
84	38.48	21.99	10.99

The heating surface in a vertical fire tube boiler consists of the furnace and the tubes. In a horizontal tubular boiler it includes the shell, tubes and heads. The locomotive type utilizes the sides of the furnace, also the crown sheet, the heads and the tubes. The water tube kind takes all of the tubes, the headers and the lower half of the steam drum. Flue boilers include about two-thirds of the shell, all of the flues and what is left of the heads. The cylinder only has about two-thirds of the shell and the same proportion of the heads.

When the total heating surface in either or all of the above types has been ascertained in accordance with these rules, tables and suggestions, it forms a basis on which to make an estimate of what a certain kind and size of a boiler will do under fair conditions.

The following figures show what can be secured along this line. They indicate the number of square feet that will develop one horse power.

HEATING SURFACE PER HORSE POWER

Vertical	18 square feet
Horizontal	15 " "
Locomotive	15 " "
Water tube	12 " "
Flue	10 " "
Cylinder	8 " "

Vertical boilers are usually designed to include a large amount of heating surface in a comparatively small shell, but the vertical portion of the tubes render them slightly less effective than when in a horizontal position, as they are found in the common tubular and locomotive boilers.

The water tube type is more efficient in evaporating water per square foot of heating surface, hence the number required per horse power is less. The flue boiler is efficient in this respect, but owing to its design the heating surface that can be secured with a shell of given diameter is less than with others above mentioned. The cylinder boiler is credited with developing a horse power with less than any of the preceding kinds, but the square feet of surface secured in a given boiler is very small, hence a much larger shell is required for a given power.

The author is aware that a horse power has been developed in numerous cases with much less heating surface than these figures represent, but the natural tendency of steam users to load their machinery to the extreme limit, and the ambition of faithful engineers to meet every responsbility put upon them needs no encouragement here. On the contrary there is more need of conservative advice now than ever before, and to this end it is earnestly recommended that boilers never be loaded beyond what these figures indicate.

The worst case that has come to our notice in detail is where a horse power was actually developed for every three square feet of heating surface that the boiler contained, but the conditions showed an utter disregard of the danger to human life, and indifference to the cost of maintenance and repairs.

ACTUAL HORSE POWER OF BOILERS

When men who are interested in steam engineering consider the subject of the power of a boiler, they seem to naturally divide themselves into three classes.

First, those who claim that there is no such thing as the horse power of a boiler.

Second, those who admit that a boiler does develop power, but think that the present way of determining it is illogical and wrong, hence wish to have a different standard adopted.

Third, those who admit that the present practice along this line is apparently not up-to-date, but cannot understand how any standard can be devised which will suit even a majority of the cases in common practice, and so long as this is true now and liable to be for a long time to come, it is useless to introduce a decidedly disturbing element into the theory of steam engineering, which cannot be of sufficient advantage to recompense for the trouble and expense of changing set rules in thousands of volumes on this important subject.

In reply to the first it is proper to call attention to the fact that all standards of this or any other kind are simply the result of the deliberations of one man that subsequently were adopted by men in council, or they were accepted by bodies of men congregated for the purpose, hence have become laws on the various subjects to which they apply.

It is just as logical and proper to decide on what shall be known as one horse power in a boiler, as it is to decide that 5,280 feet shall constitute a mile.

Concerning the objections of the second class, it is only necessary to say that the present standard was adopted many years ago when the average boiler pressure was much lower than it is at the present time. However, even at that time there were many boilers that were operated under much lower pressure than the standard calls for, while others carried higher pressures.

Now the only difference between conditions at that time and at the present is that some boilers are carrying much higher pressure than ever before. On the other hand thousands of boilers are operated every day under less pressure than the standard calls for, but this standard can be applied to every boiler in use, without regard to the conditions under which it is operated. Could any other standard do more?

The third class are a conservative body of men who are willing to accept anything that is decidedly better than what has been, or is now in use, but they insist on being convinced that old standards are defective, and that new ones are free from defects before they discard the former and adopt the latter. Surely this is safe ground on which to stand, and without this element the whole superstructure of steam engineering would become unstable and unreliable so far as theoretical standards are concerned, and this means almost every branch of the subject.

The foregoing remarks refer to what is known as the Centennial rating of the power of steam boilers, as follows:

The evaporation of 30 pounds of water per hour, from feed water at 100 degrees Fah. into steam at 70 pounds gauge pressure, or its equivalent under other conditions, constitutes one horse power.

If boilers were always run under these conditions, it would only be necessary to divide the weight of water evaporated per hour by 30 and the quotient would be the power developed. In practice it becomes necessary to reduce the actual results secured, to terms of the above rating in order to make it agree, and this proves a stumbling block to the working engineer and the owners of boilers who have not given the subject due attention. The matter will be clear to all who study the following statements:

THE OBJECT OF A BOILER TEST

The object sought in conducting a boiler test is to accurately determine the amount of water evaporated into dry steam, in order to calculate the power developed, and to find the weight of coal actually burned in order to ascertain the efficiency of the boiler.

The foregoing seem to be self-evident facts, but past experience shows that they are not, for numerous so-called tests have been reported which neither tell how much water was evaporated nor the weight of coal burned to evaporate it. Where these points are not settled, the whole proves unreliable and unsatisfactory.

Reports of conditions that can have no effect on the results will not be introduced for the purpose of making the matter appear more complicated, as the object is to simplify the process and retain accuracy.

STARTING A TEST

Before a test is started the water level should be brought to the point where it is considered advisable to carry it which will be at about two gauges usually. Tie a string around the gauge glass at this point, maintain the same water level as closely as possible throughout the test, and bring it to exactly the same place at the conclusion of it. As a further precaution the steam pressure should be the same at the beginning and the end of the test.

If these directions are followed there will be no error on account of more or less water in the boiler than there should be.

Care should be taken to know that none of the water pumped in is lost through a leaky blow off valve, or at any other point below the water line.

It is very convenient to use a water meter to determine the amount delivered to the boiler, and there is no good reason why this method should not be used. If a water meter does not correctly indicate the amount of water passing through it, and its indications are accepted without correction, the final result will not be correct, but it is not so difficult to calibrate, or prove a meter as it appears.

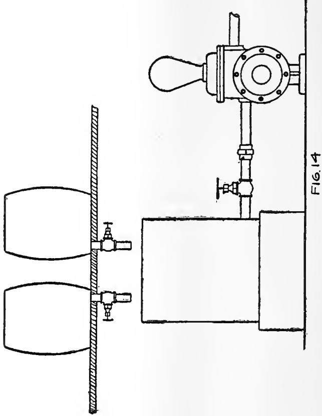
Set a barrel on a pair of platform scales and note its exact weight. Let water pass through the meter into the barrel until the meter indicates that say, 4 cubic feet have passed. Put a thermometer into this water and note its temperature, which is assumed to be 68 degrees.

By referring to a table of the properties of water the weight per cubic foot at 68 degrees is found to be 62.33 pounds, therefore 4 cubic feet weighs $62.33 \times 4 = 249.32$ pounds. Suppose that the water in the barrel actually weighs 244.25 pounds. In that case the water actually delivered to the boiler is $244.25 \div 249.32 = .98$ of what the meter indicates, consequently when the meter is read at the end of the test, the result must be multiplied by .98 to ascertain the true quantity in cubic feet.

Suppose that the water actually weighed 253 pounds, then the indications of the meter would have to be multiplied by $253 \div 249.32 = 1.015$ in order to determine the actual quantity used.

WEIGHING THE WATER

Water for testing an ordinary boiler may be weighed by means of three barrels as shown in Fig. 14. Two of them



are mounted on a platform and are fitted with outlet pipes 2 inches in diameter, both of which discharge into another barrel under them. Valves are provided to shut off either one at pleasure.

Before the lower barrel is placed in position and connected to the pump, both of the upper ones should be filled with water and each of them drained separately into a barrel on a pair of scales, thus determining the exact weight of water that each holds.

The third barrel is then placed in position and connected as shown, so that during the test water may be drawn continuously from the lower barrel as it is fed from each of the others alternately. By keeping a record of the number of times these barrels are filled and emptied, the exact weight of water used may be known without further calculation.

QUALITY OF STEAM PRODUCED

A boiler test made without determining the quality of the steam produced, is of no value, because much of the water pumped into the boiler may not be evaporated, but pass away with the steam in the form of hot water.

This water may be raised to a temperature equal to the steam with which it mingles, and the boiler should be credited with this heat, but the latent heat of evaporation has not passed into it, and as this is much greater than the sensible heat, it cannot be ignored.

A sample of the steam to be tested should be taken from a vertical pipe if possible, by means of a small perforated pipe screwed into it as shown in Fig. 15. It should always be arranged so as to prevent taking steam from the inner

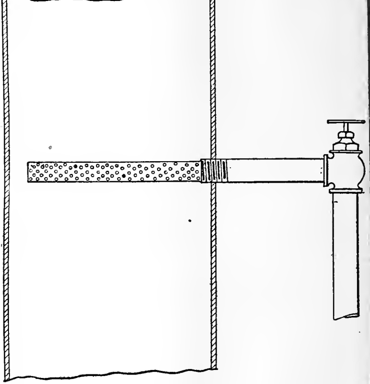


FIG. 15

surface of the large pipe, as water may trickle down on this surface, thus not giving a true sample of the steam.

If it is necessary to take the sample from a horizontal pipe, special care should be taken to avoid the water which always runs along the bottom of such a pipe.

Having properly connected this pipe the steam may be tested by blowing a portion of it into a certain weight of cold water, as illustrated in Fig. 16. A tee should be put on the end of this pipe to

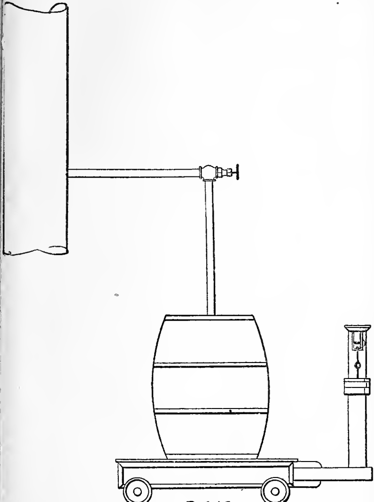


FIG 16

prevent the steam from interfering with correct weighing of the whole. The temperature of this water should be raised to not less than 110, nor more than 150 degrees Fah. The former ought to be secured in order to give sufficient rise in temperature to lessen the possibility of error, and it is a good idea not to exceed the latter, as the danger of loss of heat increases with the temperature. It should be thoroughly stirred to make the temperature even throughout the whole of it.

It is not necessary to use any given weight of cold water (provided the correct weight is known), but from 300 to 400 pounds is suggested, because more reliable results are usually secured with a large quantity. An ordinary oil barrel will hold 320 pounds without being too full for convenience, and as one of them is usually available around a steam plant it can be cleaned and used for this purpose, therefore a uniform weight of 320 pounds is recommended for these tests.

The percentage of moisture in the steam tested may be determined by the following formula:

$$\left[\frac{W}{S}\times(H-C)-(T-H)\right]\frac{I}{L}=Q$$

W = Weight of cold water used.
S = " " steam condensed.
H = Total heat of one pound of the heated water.
C = Total heat of one pound of the cold water.

T = Total heat of one pound of the water at a temperature corresponding to the pressure of steam used.

L = Latent heat of the steam.

Q = Quality of the steam tested, taking dry steam at unity or I.

The weight of steam condensed is found by subtracting the weight of the cold water from the weight of the heated water.

In all cases where there is moisture in the steam tested, the value of Q is less than one, as it represents the comparative value of the steam, as for illustration, if the formula is applied to a given case and the value of Q is .95 the steam is .95 dry or $I - Q = I - .95 = .05$ moist, or in other words it is 95 per cent. dry and there is 5 per cent. of moisture in it.

For illustration, suppose that 320 pounds of water are put into the barrel at a temperature of 65 degrees Fah., the total heat of which is 33.01. Steam is blown into this water until it weighs 340 pounds or $340 - 320 = 20$ pounds more and the temperature is raised to 125 degrees Fah., the total heat of which is 93.17. Pressure is maintained at 95 pounds absolute, or 80 by the gauge. The total heat of the water under this pressure is 295.1 and the latent heat of the steam is 885.6.

An application of the formula results as follows:

$$\left[\frac{320}{20}\times(93.17-33.1)-(295.1-93.17)\right]$$

$$\frac{\mathrm{I}}{885.6}=(16\times 60.07-201.93)\times .0011. = .83$$

Therefore .83 of the mixture coming from the boiler is dry steam and 1.00 − .83 = .17 of it is water. If it only required 19 pounds of condensed steam to secure the same temperature in the barrel, then the quality of the steam would be .89 and if 18 pounds were sufficient it would be raised to .94. As 17.5 pounds will raise it to .98 it illustrates the necessity of taking great care to secure correct weights for every experiment.

In case that only 17 pounds of condensed steam gives the required rise in temperature under the same conditions the final result is 1.02 and as this is more than I, further explanation is necessary. It denotes that the steam is superheated slightly.

It now becomes necessary to proceed as follows, in order to determine the degrees of superheat. From the quality of the steam subtract 1 and multiply the remainder by 2.0833. In this case it is $1.02-1\times2.0833=.04$ degree, showing that the steam is practically dry.

Suppose that it required only 5 pounds of water to secure the same rise in temperature, then there would be 6 degrees of superheat present. The temperature of this steam as it comes from the boiler would be 6 degrees higher than the temperature of saturated or dry steam at the same pressure.

THERMOMETERS

In order to make the necessary calculations after testing the steam, a table of the properties of water is required, and it follows this explanation. The temperatures are given from the freezing to the boiling point on the Fahrenheit, Centigrade and Reaumur thermometers, to enable the reader to easily use the kind that is preferred. It should be remembered, however, that zero on the Fahrenheit scale is 32 degrees below the freezing point, while the boiling point under atmospheric pressure at sea level is 212. Zero and the freezing point are the same on the Centigrade scale and the boiling point is found at 100, while the Reaumur takes the freezing point at zero and places the boiling point at 80 degrees above it.

For temperatures higher than the table contains the following rules may be used to convert the value of one scale into a corresponding value on another.

TO CHANGE FAHRENHEIT TO CENTIGRADE

Subtract 32, multiply the remainder by 5 and divide the quotient by 9. Example: A barrel contains 320 pounds of water at 110 degrees Fahrenheit. What is its temperature by the Centigrade scale?

$110-32\times5\div9=43.3$ degrees.

TO CHANGE FAHRENHEIT TO REAUMUR

Subtract 32, multiply the remainder by 4 and divide the quotient by 9.

In the foregoing example what is the temperature of the water by the Reaumur scale?

$110-32\times4\div9=34.7$ degrees.

TO CHANGE CENTIGRADE TO FAHRENHEIT

Multiply by 9, divide the product by 5 and add 32 to the quotient. Example: A barrel contains 320 pounds of water at 70 degrees Centigrade. What is its temperature by the Fahrenheit scale?

$70\times9\div5+32=158$ degrees.

TO CHANGE CENTIGRADE TO REAUMUR

Multiply by 4 and divide the product by 5.

In the preceding example what is the temperature of the water by the Reaumur scale?

$70\times4\div5=56$ degrees.

TO CHANGE REAUMUR TO CENTIGRADE

Multiply by 5 and divide the product by 4. Example: A barrel contains 320 pounds of water at 60 degrees Reaumur. What is its temperature on the Centigrade scale?

$60\times5\div4=75$ degrees.

TO CHANGE REAUMUR TO FAHRENHEIT

Multiply by 9, divide the product by 4 and add 32 to the quotient. In the preceding example what is the temperature of the water on the Fahrenheit scale?

$60\times9\div4+32=167$ degrees.

These rules may be applied to any case with correct results.

PROPERTIES OF WATER

Temperature			Heat units	Weight per cubic foot
F	C	R		
32	0	0	0	62.42
33	0.6	0.4	1	62.42
34	1.1	0.9	2	62.42
35	1.7	1.3	3	62.42
36	2.2	1.8	4	62.42
37	2.8	2.2	5	62.42
38	3.3	2.7	6	62.42
39	3.9	3.1	7	62.42
40	4.4	3.6	8	62.42
41	5.0	4.0	9	62.42
42	5.6	4.4	10	62.42
43	6.1	4.9	11	62.42
44	6.7	5.3	12	62.42
45	7.2	5.8	13	62.42
46	7.8	6.2	14	62.42
47	8.3	6.7	15	62.42
48	8.9	7.1	16	62.41
49	9.4	7.6	17	62.41
50	10.0	8.0	18	62.41
51	10.6	8.4	19	62.41
52	11.1	8.9	20	62.40
53	11.7	9.3	20.01	62.40
54	12.2	9.8	22.01	62.40
55	12.8	10.2	23.01	62.39
56	13.3	10.7	24.01	62.39
57	13.9	11.1	25.01	62.39
58	14.4	11.6	26.01	62.38
59	15.0	12.0	27.01	62.38
60	15.6	12.4	28.01	62.37
61	16.1	12.9	29.01	62.37
62	16.7	13.3	30.01	62.36
63	17.2	13.8	31.01	62.36
64	17.8	14.2	32.01	62.35
65	18.3	14.7	33.01	62.34
66	18.9	15.1	34.02	62.34
67	19.4	15.6	35.02	62.33
68	20.0	16.0	36.02	62.33
69	20.6	16.4	37.02	62.32
70	21.1	16.9	38.02	62.31
71	21.7	17.3	39.02	62.31
72	22.2	17.8	40.02	62.30
73	22.8	18.2	41.02	62.29
74	23.3	18.7	42.03	62.28
75	23.9	19.1	43.03	62.28
76	24.4	19.6	44.03	62.27

Properties of Water—*Continued*

Temperature			Heat units	Weight per cubic foot
F	C	R		
77	25.0	20.0	45.03	62.26
78	25.6	20.4	46.03	62.25
79	26.1	20.9	47.03	62.24
80	26.7	21.3	48.04	62.23
81	27.2	21.8	49.04	62.22
82	27.8	22.2	50.04	62.21
83	28.3	22.7	51.04	62.20
84	28.9	23.1	52.04	62.19
85	29.4	23.6	53.05	62.18
86	30.0	24.0	54.05	62.17
87	30.6	24.4	55.05	62.16
88	31.1	24.9	56.05	62.15
89	31.7	25.3	57.05	62.14
90	32.2	25.8	58.06	62.13
91	32.8	26.2	59.06	62.12
92	33.3	26.7	60.06	62.11
93	33.9	27.1	61.06	62.10
94	34.4	27.6	62.06	62.09
95	35.0	28.0	63.07	62.08
96	35.6	28.4	64.07	62.07
97	36.1	28.9	65.07	62.06
98	36.7	29.3	66.07	62.05
99	37.2	29.8	67.08	62.03
100	37.8	30.2	68.08	62.02
101	38.3	30.7	69.08	62.01
102	38.9	31.1	70.09	62.00
103	39.4	31.6	71.09	61.99
104	40.0	32.0	72.09	61.97
105	40.6	32.4	73.10	61.96
106	41.1	32.9	74.10	61.95
107	41.7	33.3	75.10	61.93
108	42.2	33.8	76.10	61.92
109	42.8	34.2	77.11	61.91
110	43.3	34.7	78.11	61.89
111	43.9	35.1	79.11	61.88
112	44.4	35.6	80.12	61.86
113	45.0	36.0	81.12	61.85
114	45.6	36.4	82.13	61.83
115	46.1	36.9	83.13	61.82
116	46.7	37.3	84.13	61.80
117	47.2	37.8	85.14	61.78
118	47.8	38.2	86.14	61.77
119	48.3	38.7	87.15	61.75
120	48.9	39.1	88.15	61.74
121	49.4	39.6	89.15	61.72
122	50.0	40.0	90.16	61.70

Properties of Water—*Continued*

Temperature F	C	R	Heat units	Weight per cubic foot
123	50.6	40.4	91.16	61.68
124	51.1	40.9	92.17	61.67
125	51.7	41.3	93.17	61.65
126	52.2	41.8	94.17	61.63
127	52.8	42.2	95.18	61.61
128	53.3	42.7	96.18	61.60
129	53.9	43.1	97.19	61.58
130	54.4	43.6	98.19	61.56
131	55.0	44.0	99.20	61.54
132	55.6	44.4	100.20	61.52
133	56.1	44.9	101.21	61.51
134	56.7	45.3	102.21	61.49
135	57.2	45.8	103.22	61.47
136	57.8	46.2	104.22	61.45
137	58.3	46.7	105.23	61.43
138	58.9	47.1	106.23	61.41
139	59.4	47.6	107.24	61.39
140	60.0	48.0	108.25	61.37
141	60.6	48.4	109.25	61.36
142	61.1	48.9	110.26	61.34
143	61.7	49.3	111.26	61.32
144	62.2	49.8	112.27	61.30
145	62.8	50.2	113.28	61.28
146	63.3	50.7	114.28	61.26
147	63.9	51.1	115.29	61.24
148	64.4	51.6	116.29	61.22
149	65.0	52.0	117.30	61.20
150	65.6	52.4	118.31	61.18
151	66.1	52.9	119.31	61.16
152	66.7	53.3	120.32	61.14
153	67.2	53.8	121.33	61.12
154	67.8	54.2	122.33	61.10
155	68.3	54.7	123.34	61.08
156	68.9	55.1	124.35	61.06
157	69.4	55.6	125.35	61.04
158	70.0	56.0	126.36	61.02
159	70.6	56.4	127.37	61.00
160	71.1	56.9	128.37	60.98
161	71.7	57.3	129.38	60.96
162	72.2	57.8	130.39	60.94
163	72.8	58.2	131.40	60 92
164	73.3	58.7	132.41	60.90
165	73.9	59.1	133.41	60.87
166	74.4	59.6	134.42	60.85
167	75.0	60.0	135.43	60.83

Properties of Water—*Continued*

Temperature			Heat units	Weight per cubic foot
F	C	R		
168	75 6	60.4	136 44	60.81
169	76 1	60 9	137.45	60.79
170	76 7	61 3	138.45	60.77
171	77 2	61 8	139 46	60.75
172	77 8	62.2	140 47	60.73
173	78.3	62.7	141 48	60.70
174	78.9	63.1	142 49	60.68
175	79 4	63.6	143.50	60.66
176	80.0	64.0	144 51	60.64
177	80.6	64 4	145.52	60.62
178	81 1	64 9	146.52	60.59
179	81 7	65.3	147.53	60.57
180	82 2	65.8	148.54	60.55
181	82.8	66.2	149.55	60.53
182	83.3	66.7	150 56	60.52
183	83.9	67 1	151.57	60.48
184	84.4	67 6	152 58	60.46
185	85 0	68.0	153 50	60.44
186	85.6	68 4	154.60	60.41
187	86 1	68 9	155 61	60.39
188	86.7	69 3	156.62	60.37
189	87 2	69.8	157 63	60.34
190	87 8	70 2	158.64	60.32
191	88 3	70 7	159 65	60.29
192	88 9	71 1	160.67	60.27
193	89 4	71.6	161.68	60.25
194	90 0	72 0	162 69	60.22
195	90 6	72.4	163 70	60.20
196	91 1	72.9	164.71	60.17
197	91 7	73.3	165 72	60.15
198	92 2	73.8	166 73	60.12
199	92 8	74 2	167.74	60.10
200	93.3	74 7	168.75	60.07
201	93 9	75 1	169.77	60.05
202	94 4	75.6	170.78	60 02
203	95.0	76.0	171 79	60.00
204	95.6	76.4	172.80	59.97
205	96.1	76 9	173.81	59.95
206	96 7	77.3	174.83	59.92
207	97 2	77.8	175.85	59.89
208	97 8	78.2	176.85	59.87
209	98 3	78 7	177.86	59.84
210	98.9	79 1	178 87	59.82
211	99.4	79 6	179.89	59.79
212	100.0	80 0	180.90	59.76

The foregoing tables show that as the temperature of water is increased it expands, hence if a cubic foot at 40 degrees Fah. or at any higher temperature is still further heated, it will occupy more than a cubic foot of space, and it follows that if the volume is kept constant the weight must decrease.

The difference is so small that it does not appear in the table because the figures are given to two decimal places only, until the temperature is raised to 48 degrees or more. If it is raised from 32 to 39 degrees it contracts slightly, thus occupying less space, but the difference is very small. Maximum density is attained at 39.1 degrees Fahrenheit, 3.9 Centigrade or 3.1 Reaumur.

It is useless for engineers, under even the best working conditions, to attempt to prove these figures, on account of difficulty of measuring out exactly a cubic foot, and of obtaining a sample of perfectly pure water.

SATURATED STEAM

This name does not always seem to be appropriate, as it suggests to the average working engineer, steam that is saturated with water, whereas it is intended for dry steam, as otherwise it could not constitute a standard for comparison, for as soon as steam becomes mixed with water, or when there is water suspended in the steam, its quality becomes variable with the amount of water present, hence it could not be used with profit for comparison. Saturated steam is therefore the dividing quality between wet and superheated steam.

In order to solve problems that have already been presented, as well as some of those that follow, it is necessary to know something of the properties of saturated steam. These are given in the next table which will be explained here in order that the working engineer who has not enjoyed the advantages of a technical education may understand them, also for firemen who wish to be advanced to engineers, and all others interested in this important subject.

Absolute pressures are used because all pressure must be reckoned from a perfect vacuum in this work, as otherwise there would be no standard for a base of operations. For all practical purposes it may be taken at 15 pounds above the gauge pressure, for steam gauges indicate the unbalanced pressure, and safety valves are designed to operate on the same principle. To ascertain the gauge pressures, subtract 15 from those given in the table.

Temperatures are stated in the Fahrenheit scale in which zero is 32 degrees below the freezing point. It is not convenient or practicable to state the heat units in water above zero, because the amount of heat required to raise the temperatures through a given range below 32 on this scale is less than to raise it through the same range above 32 degrees. It is therefore better to base all such calculations in which heat units are factors, on the freezing point, as it saves confusion and trouble.

It is claimed that the temperature of water and of steam in a boiler is the same, and this is true after the water has been in the boiler under working conditions long enough to attain its maximum temperature, but it does not immediately flash into steam, because it lacks the latent heat of evaporation. This is a wise provision as otherwise we could not operate steam boilers as we do at present. When the heat units in a pound of water and the latent heat of evaporation are added, the sum is the total heat of steam at given pressure.

We are sometimes told that the total heat of steam is the same for all pressures, but this is not true, hence should not be accepted.

The latent heat decreases as the temperature or sensible heat increases, but not in the same proportion as the total for 15 pounds, absolute pressure, is 1,146.9 while for 200 pounds it is, 1,198.3 a difference of 51.4 heat units, or more than 4 per cent.

It is necessary to know the weight of steam under various pressures when calculating the weight of a given volume and this is stated in the table.

PROPERTIES OF SATURATED STEAM

P = Absolute pressure in pounds per square inch, or 15 pounds above gauge pressure.

T = Temperature of steam under the given pressure.

W = Heat units in a pound of water under pressure that corresponds to the temperature.

L = Latent heat, or the number of heat units required to convert one pound of water at a given temperature and pressure into steam.

S = Total heat of steam per pound above 32 degrees Fah.

C = Weight of one cubic foot of steam at stated pressure.

PROPERTIES OF SATURATED STEAM

P	T	W	L	S	C
14.7	212.0	180.9	965.7	1,146.6	.03794
15	213.0	181.9	965.0	1,146.9	.03868
16	216.3	185.3	962.7	1,147.9	.04110
17	219.4	188.4	960.5	1,148.9	.04352
18	222.4	191.4	958.3	1,149.8	.04592
19	225.2	194.3	956.3	1,150.6	.04831
20	227.9	197.0	954.4	1,151.5	.05070
21	230.5	199.7	952.6	1,152.2	.05308
22	233.0	202.2	950.8	1,153.0	.05545
23	235.4	204.7	949.1	1,153.7	.05782
24	237.8	207.0	947.4	1,154.5	.06018
25	240.0	209.3	945.8	1,155.1	.06253
26	242.2	211.5	944.3	1,155.8	.06487
27	244.3	213.7	942.8	1,156.4	.06721
28	246.3	215.7	941.3	1,157.1	.06955
29	248.3	217.8	939.9	1,157.7	.07188
30	250.2	219.7	938.9	1,158.3	.07420
31	252.1	221.6	937.2	1,158.8	.07652
32	254.0	223.5	935.9	1,159.4	.07884
33	255.7	225.3	934.6	1,159.9	.08115
34	257.5	227.1	933.4	1,160.5	.08346
35	259.2	228.8	932.2	1,161.0	.08576
36	260.8	230.5	931.0	1,161.5	.08806
37	262.5	232.1	929.8	1,162.0	.09035
38	264.0	233.8	928.7	1,162.5	.09264
39	265.6	235.4	927.6	1,162.9	.09493
40	267.1	236.9	926.5	1,163.4	.09721
41	268.6	238.5	925.4	1,163.9	.09949
42	270.1	240.0	924.4	1,164.3	.1018
43	271.5	241.4	923.3	1,164.7	.1040
44	272.9	242.9	922.3	1,165.2	.1063
45	274.3	244.3	921.3	1,165.6	.1086
46	275.7	245.7	920.4	1,166.0	.1108
47	277.0	247.0	919.4	1,166.4	.1131
48	278.3	248.4	918.5	1,166.8	.1153
49	279.6	249.7	917.5	1,167.2	.1176
50	280.9	251.0	916.6	1,167.6	.1198
51	282.1	252.2	915.7	1,168.0	.1221
52	283.3	253.5	914.9	1,168.4	.1243
53	284.5	254.7	914.0	1,168.7	.1266
54	285.7	256.0	913.1	1,169.1	.1288
55	286.9	257.2	912.3	1,169.4	.1311
56	288.1	258.3	911.5	1,169.8	.1333
57	289.1	259.5	910.6	1,170.1	.1355
58	290.3	260.7	909.8	1,170.5	.1377
59	291.4	261.8	909.0	1,170.8	.1400
60	292.5	262.9	908.2	1,171.2	.1422

Properties of Saturated Steam—*Continued*

P	T	W	L	S	C
61	293.6	264.0	907.5	1,171.5	.1444
62	294.7	265.1	906.7	1,171.8	.1466
63	295.7	266.2	905.9	1,172.1	.1488
64	296.8	267.2	905.2	1,172.4	.1511
65	297.8	268.3	904.5	1,172.8	.1533
66	298.8	269.3	903.7	1,173.1	.1555
67	299.8	270.4	903.0	1,173.4	.1577
68	300.8	271.4	902.3	1,173.7	.1599
69	301.8	272.4	901.6	1,174.0	.1621
70	302.7	273.4	900.9	1,174.3	.1643
71	303.7	274.4	900.2	1,174.6	.1665
72	304.6	275.3	899.5	1,174.8	.1687
73	305.6	276.3	898.9	1,175.1	.1709
74	306.5	277.2	898.2	1,175.4	.1731
75	307.4	278.2	897.5	1,175.7	.1753
76	308.3	279.1	896.9	1,176.0	.1775
77	309.2	280.0	896.2	1,176.2	.1797
78	310.1	280.9	895.6	1,176.5	.1819
79	310.9	281.8	895.0	1,176.8	.1840
80	311.8	282.7	894.3	1,177.0	.1862
81	312.7	283.6	893.7	1,177.3	.1884
82	313.5	284.5	893.1	1,177.6	.1906
83	314.4	285.3	892.5	1,177.8	.1928
84	315.2	286.2	891.9	1,178.1	.1950
85	316.0	287.0	891.3	1,178.3	.1971
86	316.8	287.9	890.7	1,178.6	.1993
87	317.7	288.7	890.1	1,178.8	.2015
88	318.5	289.5	889.5	1,179.1	.2036
89	319.3	290.4	888.9	1,179.3	.2058
90	320.0	291.2	888.4	1,179.6	.2080
91	320.8	292.0	887.8	1,179.8	.2102
92	321.6	292.8	887.2	1,180.0	.2123
93	322.4	293.6	886.7	1,180.3	.2145
94	323.1	294.4	886.1	1,180.5	.2166
95	323.9	295.1	885.6	1,180.7	.2188
96	324.6	295.9	885.0	1,181.0	.2210
97	325.4	296.7	884.5	1,181.2	.2231
98	326.1	297.4	884.0	1,181.4	.2253
99	326.8	298.2	883.4	1,181.6	.2274
100	327.6	298.9	882.9	1,181.8	.2296
101	328.3	299.7	882.4	1,182.1	.2317
102	329.0	300.4	881.9	1,182.3	.2339
103	329.7	301.1	881.4	1,182.5	.2360
104	330.4	301.9	880.8	1,182.7	.2382
105	331.1	302.6	880.3	1,182.9	.2403
106	331.8	303.3	879.8	1,183.1	.2425

Properties of Saturated Steam—*Continued*

P	T	W	L	S	C
107	332.5	304.0	879.3	1,183.4	.2446
108	333.2	304.7	878.8	1,183.6	.2467
109	333.9	305.4	878.3	1,183.8	.2489
110	334.5	306.1	877.9	1,184.0	.2510
111	335.2	306.8	877.4	1,184.2	.2531
112	335.9	307.5	876.9	1,184.4	.2553
113	336.5	308.2	876.4	1,184.6	.2574
114	337.2	308.8	875.9	1,184.8	.2596
115	337.8	309.5	875.5	1,185.0	.2617
116	338.5	310.2	875.0	1,185.2	.2638
117	339.1	310.8	874.5	1,185.4	.2660
118	339.7	311.5	874.1	1,185.6	.2681
119	340.4	312.1	873.6	1,185.8	.2703
120	341.0	312.8	873.2	1,185.9	.2724
121	341.6	313.4	872.7	1,186.1	.2745
122	342.2	314.1	872.3	1,186.3	.2766
123	342.9	314.7	871.8	1,186.5	.2788
124	343.5	315.3	871.4	1,186.7	.2809
125	344.1	316.0	870.9	1,186.9	.2830
126	344.7	316.6	870.5	1,187.1	.2851
127	345.3	317.2	870.0	1,187.3	.2872
128	345.9	317.8	869.6	1,187.4	.2894
129	346.5	318.4	869.2	1,187.6	.2915
130	347.1	319.1	868.7	1,187.8	.2936
131	347.6	319.7	868.3	1,188.0	.2957
132	348.2	320.3	867.9	1,188.2	.2978
133	348.8	320.8	867.5	1,188.3	.3000
134	349.4	321.5	867.0	1,188.5	.3021
135	350.0	322.1	866.6	1,188.7	.3042
136	350.5	322.6	866.2	1,188.9	.3063
137	351.1	323.2	865.8	1,189.0	.3084
138	351.8	323.8	865.4	1,189.2	.3105
139	352.2	324.4	865.0	1,189.4	.3126
140	352.8	325.0	864.6	1,189.5	.3147
141	353.3	325.5	864.2	1,189.7	.3169
142	353.9	326.1	863.8	1,189.9	.3190
143	354.4	326.7	863.4	1,190.0	.3211
144	355.0	327.2	863.0	1,190.2	.3232
145	355.5	327.8	862.6	1,190.4	.3253
146	356.0	328.4	862.2	1,190.5	.3274
147	356.6	328.9	861.8	1,190.7	.3295
148	357.1	329.5	861.4	1,190.9	.3316
149	357.6	330.0	861.0	1,191.0	.3337
150	358.2	330.6	860.6	1,191.2	.3358
151	358.7	331.1	860.2	1,191.3	.3379
152	359.2	331.6	859.9	1,191.5	.3400
153	359.7	332.2	859.5	1,191.7	.3421

Properties of Saturated Steam—*Continued*

P	T	W	L	S	C
154	360.2	332.7	859.1	1,191.8	.3442
155	360.7	333.2	858.7	1,192.0	.3463
156	361.3	333.8	858.4	1,192.1	.3483
157	361.8	334.3	858.0	1,192.3	.3504
158	362.3	334.8	857.6	1,192.4	.3525
159	362.8	335.3	857.2	1,192.6	.3546
160	363.3	335.9	856.9	1,192.7	.3567
161	363.8	336.4	856.5	1,192.9	.3588
162	364.3	336.9	856.1	1,193.0	.3609
163	364.8	337.4	855.8	1,193.2	.3630
164	365.3	337.9	855.4	1,193.3	.3650
165	365.7	338.4	855.1	1,193.5	.3671
166	366.2	338.9	854.7	1,193.6	.3692
167	366.7	339.4	854.4	1,193.8	.3713
168	367.2	339.9	854.0	1,193.9	.3731
169	367.7	340.4	853.6	1,194.1	.3754
170	368.2	340.9	853.3	1,194.2	.3775
171	368.6	341.4	852.9	1,194.4	.3796
172	369.1	341.9	852.6	1,194.5	.3817
173	369.6	342.4	852.3	1,194.7	.3838
174	370.0	342.9	851.9	1,194.8	.3858
175	370.5	343.4	851.6	1,194.9	.3879
176	371.0	343.9	851.2	1,195.1	.3900
177	371.4	344.3	850.9	1,195.2	.3921
178	371.9	344.8	850.5	1,195.4	.3942
179	372.4	345.3	850.2	1,195.5	.3962
180	372.8	345.8	849.9	1,195.7	.3983
181	373.3	346.3	849.5	1,195.8	.4004
182	373.7	346.7	849.2	1,195.9	.4025
183	374.2	347.2	848.9	1,196.1	.4046
184	374.6	347.7	848.5	1,196.2	.4066
185	375.1	348.1	848.2	1,196.3	.4087
186	375.5	348.6	847.9	1,196.5	.4108
187	375.9	349.1	847.6	1,196.6	.4129
188	376.4	349.5	847.2	1,196.7	.4150
189	376.9	350.0	846.9	1,196.9	.4170
190	377.3	350.4	846.6	1,197.0	.4190
191	377.7	350.9	846.3	1,197.1	.4212
192	378.2	351.3	845.9	1,197.3	.4233
193	378.6	351.8	845.6	1,197.4	.4254
194	379.0	352.2	845.3	1,197.5	.4275
195	379.5	352.7	845.0	1,197.7	.4296
196	380.0	353.1	844.7	1,197.8	.4317
197	380.3	353.6	844.4	1,197.9	.4337
198	380.7	354.0	844.1	1,198.1	.4358
199	381.2	354.4	843.7	1,198.2	.4379
200	381.6	354.9	843.4	1,198.3	.4400

WATER EVAPORATED UNDER WORKING CONDITIONS

When water that is used in conducting a boiler test is measured in barrels, it is only necessary to multiply the number of barrel fulls by the weight of each to secure the total weight, but if a meter is used, the quantity that has passed through it is indicated in cubic feet. Care must be taken to note the temperature as that determines the weight per cubic foot.

Suppose that during a test lasting 10 hours, the meter indicates that 658 cubic feet have passed, or 65.8 per hour, and calibration of the meter according to directions already given shows that .98 of its indications are the true quantity. Then $65.8 \times .98 = 64.484$ cubic feet. Taking the temperature at 65 degrees, the weight per cubic foot is 62.34 pounds, or $64.484 \times 62.34 = 4{,}019.93$ pounds per hour.

For a simple illustration it is assumed that after the feed water passes through the meter at 65 degrees it goes to a heater where its temperature is raised to 100 degrees Fah., and it is then forced into the boiler which carries 70 pounds pressure by the gauge or 85 pounds absolute. The atmospheric pressure is 14.7 pounds or less, but inasmuch as our ordinary steam gauges do not designate fractions of a pound it is not necessary to take them into account in this calculation.

Under these conditions every 30 pounds of water pumped into the boiler represents one horse power, therefore this boiler developed $4{,}019.93 \div 30 = 134$ horse power.

Particular attention is called to the fact that no mention is made of the purpose for which this steam is used, because it makes no difference in the calculations. Some of it may be used to run an engine, another portion to operate pumps, and the remainder to heat dry kilns, or for any other purpose for which steam is required, but this has no effect on the power developed by the boiler.

This does not necessarily mean that it will supply enough steam to run an engine and develop 134 horse power, because it might require 50 pounds of steam per hour for each horse power developed. This steam would then furnish $4{,}019.93 \div 50 = 80.4$ horse power.

On the other hand, it might be used to run a high grade engine requiring only 15 pounds of steam per hour for each horse power developed. Under this condition the boiler would supply $4{,}019.93 \div 15 = 268$ horse power.

These two examples clearly illustrate the injustice of rating a boiler by the power developed in an engine. In the former case the engine might be rated at 134 horse power, still the boiler could not supply the steam required to run it, although it might be forced much beyond its rated capacity in an effort to keep the engine in operation. On this basis the boiler would be condemned, although developing more power than it was designed for.

In the latter case the engine might be rated at 134 horse power and the boiler might be highly commended because it supplies the required steam with a slow fire in the furnace and a small amount of coal per hour, but this would not be a fair decision, because the boiler would be supplying only $134 \times 15 \div 30 = 67$ horse power which fully explains its easy performance.

ACCOUNTING FOR MOISTURE IN STEAM

Frequently there is moisture in steam supplied by a boiler, therefore, it becomes necessary to take this into account in order that the boiler may be given credit for its exact performance. In this case 4,019.93 pounds of water were pumped into the boiler. If the calorimeter test shows that 3 per cent. of this water is not evaporated, but passes out as hot water mixed with the steam, it must be subtracted from the total weight used. Then $4{,}019.93 - (4{,}019.93 \times .03) = 4{,}019.93 - 120.59 = 3{,}899.34$ pounds.

This water went into the boiler at a temperature of 100 degrees and passed out at 85 pounds absolute pressure, the corresponding temperature of which is 316 degrees Fah. and this heat must be accounted for. Water at 100 degrees contains 68.08 heat units per pound, which is increased to 287 at 85 pounds absolute pressure, therefore $287 - 68.08 = 218.92$ heat units were put into each pound of it, or $120.59 \times 218.92 = 26{,}399.56$ heat units for the whole.

In order to reduce this to proper terms it is necessary to ascertain how many pounds of water this amount of heat will evaporate under given conditions. The total heat of steam at 85 pounds absolute pressure is 1,178.3. (See table.) It already contains 68.08 heat units per pound, therefore, it requires $1{,}178.3 - 68.08 = 1{,}110.22$ heat units. Then 26,399.56 will evaporate $26{,}399.56 \div 1{,}110.22 = 23.78$ pounds of water, if it was utilized for this purpose.

Adding this to the amount actually evaporated shows that if all of the heat accounted for had been used to convert water into steam, the amount would have been $3{,}899.34 + 23.78 = 3{,}923.12$ pounds. Dividing this by 30 shows under these conditions the boiler would develop 130.77 horse power.

EQUIVALENT HORSE POWER

The above title is used because it expresses concisely the meaning of this paragraph when fully explained. Boilers are seldom operated under the exact conditions laid down for standard tests when the horse power developed is to be determined, but fortunately it is not necessary to comply with these conditions so far as steam pressure carried and temperature of feed water are concerned, as it is not difficult to reduce the results secured under any given conditions to terms that will admit of comparison on a common basis with the standard, which is the evaporation of 30 pounds of water per hour, when carrying 70 pounds gauge, or 85 pounds absolute pressure, with feed water at 100 degrees Fah.

Every pound (in weight) of steam at this pressure contains 1,178.3 heat units, and every pound of feed water at 100 degrees contains 68.08 heat units, therefore heat from the furnace must supply $1{,}178.3 - 68.08 = 1{,}110.22$ heat units. This demonstrates that the generation and application to water in a boiler of enough heat to evaporate 30 pounds in one hour where each pound requires 1,110.22 heat units, constitutes a boiler horse power.

It naturally follows that if steam is generated under conditions that require less heat per pound, a greater weight of water will be evaporated by the same amount of heat. Consequently if conditions are such that more heat is required per pound of steam, less water will be evaporated by the same quantity of heat.

The quantity of water required per hour to constitute one horse power under different conditions can be determined by the following formula:

$$\frac{1{,}110.22}{T-F}\times 30 = W.$$

T = Total heat of steam at given pressure.

F = Heat units in the feed water above 32 degrees, at given temperature.

W = Weight of water in pounds.

For an illustration of the application of this formula, suppose that a boiler evaporates into dry steam 4,019.93 pounds of water per hour, under 165 pounds absolute pressure, with feed water at 210 degrees Fah. How much water constitutes one horse power under these conditions and how much power is developed?

The total heat of steam at 165 absolute, or 150 pounds gauge pressure is 1,193.5 and water at 210 degrees contains 178.87 heat units

$\frac{1{,}110.22}{1{,}193.5-178.87}\times 30 = 32.826$ pounds of water required to constitute one horse power. $4{,}019.92 \div 32{,}826 = 122.46$ horse power developed.

As the evaporation of the same weight of water under less favorable conditions developed 134 horse power, the improved conditions, consisting of heating the feed water to a higher temperature, reduced the load by 11.54 horse power.

This reduction of load results in a corresponding saving in fuel, calling attention to great benefits derived from heating the feed water by exhaust steam which is frequently a waste product.

The next table gives the weight, in pounds, of water that must be evaporated per hour to constitute one horse power under different conditions. It is based on the foregoing formula, and will be very useful to engineers and others who wish to know at a glance how much water is required for one horse power under conditions found in their respective plants.

For illustration, suppose that a certain boiler is operated under 125 pounds gauge pressure and the feed water enters at 210 degrees after passing through any kind of a heater that utilizes exhaust steam. Following the column under 140 pounds absolute pressure (which is equal to 125 by the gauge) until it intersects with the line beginning with 210 it shows that 32.95 pounds are required under these conditions.

POUNDS OF WATER PER HORSE POWER

Absolute Boiler Pressure

Feed water	40	50	60	70	80	90	100
40	28.83	28.74	28.63	28.56	28.49	28.43	28.37
50	29.08	28.97	28.88	28.80	28.73	28.67	28.61
60	29.33	29.23	29.13	29.05	28.98	28.92	28.86
70	29.57	29.48	29.39	29.31	29.24	29.14	29.12
80	29.86	29.75	29.65	29.57	29.50	29.43	29.37
90	30.13	30.02	29.92	29.84	29.76	29.69	29.63
100	30.41	30.29	30.19	30.11	30.03	29.96	29.90
110	30.86	30.57	30.47	30.39	30.31	30.23	30.17
120	30.97	30.85	30.75	30.66	30.59	30.51	30.45
130	31.27	31.14	31.04	30.95	30.87	30.79	30.73
140	31.56	31.44	31.33	31.24	31.16	31.08	31.02
150	31.87	31.74	31.63	31.54	31.46	31.38	31.32
160	32.18	32.05	31.93	31.84	31.76	31.68	31.61
170	32.49	32.36	32.24	32.15	32.07	31.99	31.92
180	32.82	32.68	32.57	32.46	32.38	32.30	32.23
190	33.15	33.01	32.90	32.79	32.70	32.62	32.55
200	33.48	33.34	33.22	33.12	33.03	32.95	32.87
210	33.80	33.69	33.56	33.46	33.36	33.28	33.21
212	33.90	33.75	33.63	33.53	33.43	33.35	33.27

POUNDS OF WATER PER HORSE POWER

Absolute Boiler Pressure

Feed water	110	120	130	140	150	160	170
40	28.32	28.26	28.23	28.19	28.15	28.11	28.08
50	28.56	28.51	28.47	28.43	28.39	28.35	28.32
60	28.81	28.76	28.72	28.68	28.63	28.59	28.56
70	29.06	29.01	28.97	28.92	28.88	28.84	28.80
80	29.32	29.27	29.22	29.18	29.14	29.09	29.06
90	29.58	29.53	29.48	29.44	29.39	29.35	29.31
100	29.84	29.79	29.73	29.70	29.65	29.61	29.57
110	30.12	30.06	30.01	29.97	29.92	29.88	29.84
120	30.39	30.34	30.29	30.24	30.19	30.15	30.11
130	30.67	30.62	30.57	50.32	30.48	30.43	30.39
140	30.96	30.90	30.85	30.80	30.76	30.71	30.67
150	31.25	31.19	31.14	31.09	31.04	31.00	30.96
160	31.55	31.49	31.43	31.39	31.34	31.29	31.25
170	31.85	31.79	31.74	31.69	31.63	31.59	31.55
180	32.16	32.10	32.05	31.99	31.94	31.89	31.85
190	32.48	32.42	32.36	32.31	32.25	32.21	32.16
200	32.80	32.74	32.68	32.63	32.57	32.53	32.48
210	33.13	33.07	33.01	32.95	32.91	32.85	32.80
212	33.20	33.14	33.08	33.02	32.96	32.91	32.87

BENEFITS OF FEED WATER HEATERS

Although the saving in fuel is more than enough benefit to pay for the installation of a good feed water heater, it is not the only advantage gained by its use, and under some conditions the saving of unnecessary strains on the boiler is a greater compensation than the saving in fuel.

This is especially true of locomotive boilers or any other type that is fitted with a water leg into which the cool feed water is discharged. The Hartford Steam Boiler Inspection & Insurance Co. made experiments some time ago which demonstrated that cool water coming in contact with a heated boiler plate causes it to contract, and as such a plate is rigidly connected to others, a very great strain on these parts is the sure result.

While it is difficult or impossible to calculate with even a reasonable degree of accuracy what these strains amount to, it is a well known fact that they cause cracks in the plates which greatly weaken them, and if such cracks are not given intelligent attention, the result may be disastrous explosions.

As plates contract directly in proportion to the difference between the temperature of the feed water striking directly against them, and that of the steam or hot water in contact with adjacent parts, anything that reduces this difference cannot fail to be beneficial. The girth seams of horizontal boilers frequently leak from the same cause.

The percentage of fuel saved by the installation of a good feed water heater that utilizes exhaust steam, or by an economizer that takes heat from waste gases on their way to the chimney, can be determined by the following rule.

From the total heat, or heat units in the heated water, subtract the heat units in the cold water. Divide the remainder by the total heat of steam at the pressure carried, minus the heat units in the cold water. Multiply the quotient by 100.

When written as a formula it appears as follows:

$$\frac{H-C}{S-C} \times 100 = P.$$

H = Total heat in the heated water.
C = Total heat in the cold water.
S = Total heat in the steam at the pressure carried.
P = Percentage of gain, or the portion of fuel saved by heating the feed water.

For illustration, suppose that a boiler carries 140 pounds absolute pressure and uses water from the street main at a temperature of 50 degrees Fah. What percentage of the fuel now burned will be saved by the installation of an exhaust steam feed water heater that will raise the temperature of it to 211 degrees Fah., assuming that the exhaust steam is not utilized for any other purpose?

Applying the formula to this case results as follows:

$$\frac{179.89-18}{1{,}189.5-18}\times 100=13.8 \text{ per cent.}$$

It is not a difficult matter to determine the saving that will result under any conditions that can be found in practice by the foregoing rule and formula, using tables of properties of water and steam that are found on preceding pages, but for the convenience of readers the next table contains the results of calculation to determine these values. The initial temperatures refer to the temperature of water as it enters the heater. This is given from 40 to 100 degrees Fah. as this range will be sufficient to cover all ordinary cases. For lower or higher temperatures use the formula.

The table includes cases where the water is heated to 180, 190, 200, and 210 degrees Fah., as a heater that will not deliver water at the former temperature is of little value, and the latter is seldom exceeded without back pressure on the engine. The pressures are absolute, therefore 15 must be subtracted from them to secure gauge pressures. The results given are the percentage of saving on all fuel burned where water is forced into boilers at the initial temperatures.

Saving by the use of Feed Water Heaters, when water is heated to 180 degrees Fah.

ABSOLUTE BOILER PRESSURE

Initial temperature	40	50	60	70	80	90	100	110	120	130	140	150	160	170
40	12.16	12.12	12 08	12.05	12.01	11.99	11.97	11.95	11.93	11.91	11.89	11.87	11.86	11.85
50	11.39	11.35	11.32	11.29	11.26	11.24	11.21	11.19	11.18	11.16	11.14	11.12	11.11	11.10
60	10.61	10.58	10.54	10.51	10.49	10.46	10.44	10.42	10.41	10.39	10.38	10.35	10.34	10.33
70	9.82	9.78	9.75	9.73	9.70	9.67	9.66	9.64	9.63	9.61	9.59	9.58	9.57	9.56
80	9.01	8.97	8.95	8.92	8.90	8.88	8.86	8.84	8.83	8.82	8.80	8.79	8.78	8.77
90	8.18	8.15	8.13	8.10	8.08	8.06	8.05	8.03	8.02	8.00	7.99	7.98	7.97	7.96
100	7.35	7.31	7.29	7.27	7.25	7.24	7.22	7.20	7.19	7.18	7.17	7.16	7.15	7.14

Saving by the use of Feed Water Heaters, when the water is heated to 190 degrees Fah,

ABSOLUTE BOILER PRESSURE

Initial temperature	40	50	60	70	80	90	100	110	120	130	140	150	160	170
40	13.04	12.99	12.95	12.91	12.88	12.86	12.83	12.80	12.78	12.77	12.75	12.73	12.71	12.69
50	12.28	12.23	12.19	12.16	12.13	12.10	12.08	12.06	12.04	12.02	12.00	11.99	11.97	11.96
60	11.50	11.46	11.43	11.40	11.37	11.34	11.32	11.30	11.28	11.26	11.25	11.23	11.21	11.20
70	10.72	10.67	10.63	10.61	10.59	10.56	10.54	10.52	10.51	10.49	10.47	10.46	10.44	10.43
80	9.91	9.88	9.85	9.82	9.80	9.77	9.75	9.73	9.71	9.70	9.69	9.67	9.66	9.65
90	9.09	9.06	9.03	9.01	8.99	8.97	8.95	8.93	8.92	8.90	8.89	8.87	8.86	8.85
100	8.27	8.23	8.20	8.18	8.16	8.14	8.12	8.11	8.09	8.08	8.07	8.06	8.05	8.05

Saving by the use of Feed Water Heaters, when water is heated to 200 degrees Fah.

ABSOLUTE BOILER PRESSURE

Initial temperature	40	50	60	70	80	90	100	110	120	130	140	150	160	170
40	13.91	13.86	13.82	13.78	13.75	13.72	13.69	13.67	13.65	13.62	13.60	13.58	13.56	13.55
50	13.16	13.11	13.07	13.04	13.00	12.98	12.95	12.93	12.90	12.88	12.86	12 85	12.83	12.81
60	12.39	12.35	12.31	12.28	12.25	12.22	12.19	12.17	12.15	12.13	12.11	12.09	12.08	12.06
70	11.61	11.57	11.53	11.50	11.47	11.45	11.43	11.40	11.39	11.37	11.35	11.33	11.32	11.31
80	10.82	10.78	10.75	10.72	10.69	10.66	10.64	10.62	10.61	10.59	10.57	10.56	10.54	10.53
90	10.01	9.98	9.94	9.91	9.89	9.87	9.85	9.83	9.81	9.79	9.78	9.77	9.76	9.74
100	9.19	9.14	9.12	9.10	9.08	9.05	9.03	9.02	9.00	8.99	8.97	8.96	8.95	8.94

Saving by the use of Feed Water Heaters, when water is heated to 210 degrees Fah.

ABSOLUTE BOILER PRESSURE

40	14.78	14.73	14.70	14.67	14.62	14.58	14.56	14.52	14.50	14.48	14.46	14.42	14.41	14.39
50	14.04	13.99	13.95	13.91	13.88	13.85	13.82	13.79	13.78	13.75	13.73	13.71	13.69	13.67
60	13.28	13.24	13.20	13.16	13.13	13.10	13.08	13.05	13.03	13.00	12.98	12.96	12.95	12.93
70	12.51	12.47	12.43	12.39	12.36	12.34	12.31	12.29	12.27	12.25	12.23	12.20	12.19	12.18
80	11.72	11.68	11.64	11.61	11.58	11.56	11.54	11.52	11.50	11.48	11.46	11.44	11.42	11.41
90	10.92	10.88	10.85	10.81	10.79	10.77	10.75	10.73	10.71	10.69	10.68	10.66	10.64	10.63
100	10.11	10.08	10.04	10.01	9.99	9.97	9.96	9.93	9.91	9.89	9.88	9.86	9.85	9.84

EQUIVALENT EVAPORATION

When a certain weight of water is evaporated in a steam boiler at a known pressure, it forms a basis for computing the power developed, as before explained, also for comparing results secured with other boilers, but for convenience and accuracy it is customary to reduce all of them to a common level.

Boilers are fed with water whose temperature varies through a wide range, and steam is generated under many different pressures, but when it is assumed that the feed water is heated to 212 degrees Fah. and that steam is generated under atmospheric pressure at a temperature of 212 degrees Fah. and all results are reduced to these standards, comparisons can be intelligently made at short notice.

As the feed water and the steam generated have the same temperature, it demonstrates that only the latent heat of steam at atmospheric pressure is put into the water after it enters the boiler. This amounts to 965.7 heat units per pound above 32 degrees Fah., and this result is obtained by subtracting the heat units in a pound of the feed water at this temperature, from the total heat of steam at this pressure, and 1,146.6—180.9 = 965.7. If water at 100 degrees is pumped into a boiler carrying 85 pounds absolute pressure, each pound of it requires 1,178.3 − 68.08 = 1,110.22 heat units to evaporate it into dry steam.

Thus it becomes plain that the numbers 965.7 and 1,110.22 are used for comparisons along this line.

The evaporation of 30 pounds of water in one hour, where each pound of it requires 1,110.22 heat units, takes 33,306.6 heat units. Under other conditions the evaporation of one pound of water requires 965.7 heat units, therefore the total just mentioned will evaporate $33{,}306.6 \div 965.7 = 34.489$ pounds of water, which is called 34.5 for convenience in calculating results. Thus the evaporation of 34.5 pounds of water in one hour from and at 212 degrees constitutes one boiler horse power, as it is equivalent to the standard explained on foregoing pages.

Another way in which this can be used is to show how much water would have been evaporated in any given case found in practice, provided the feed water was at a temperature of 212 degrees and there was no pressure by the gauge. Such a problem is solved by the following rule:

Multiply the weight of water actually evaporated into dry steam, by the number of heat units required to evaporate one pound and divide the product by 965.7. The quotient is the weight that would be evaporated under assumed conditions.

For example, suppose that a boiler evaporates into dry steam 4,019.93 pounds of water in one hour, from feed water at 100 degrees, into steam at 70 pounds gauge pressure. How much would have been evaporated from and at 212 degrees? How much power would have been developed?

$4{,}019.93 \times (1{,}178.3 - 68.08) \div 965.7 =$ 4,621.5 pounds.

When this is divided by 34.5 it shows that 134 horse power was developed.

COAL REQUIRED TO EVAPORATE WATER

Owners of steam plants and superintendents of mills, shops and factories frequently say that if two kinds of coal are to be tested it is only necessary to compare the weight used to run the machinery one week, with what is required to operate it another week, and that comparison decides the comparative value of two kinds of coal.

To a certain limited extent this conclusion is correct, because the weight of coal burned, in connection with the price per ton determines the cost of fuel for a given time, but it is unjust and unfair to base an opinion concerning the real merit of any kind of fuel on the result of such a so-called test, which is so crude and unsatisfactory that it is not worthy to be called a test at all.

When discussing the merits of such a transaction the steam user always claims that the same amount of machinery was operated during the time mentioned, but while such a claim may seem reasonable from his point of view, it really is not, because an engine will seldom or never develop just the same power for many days in succession, even if the machines operated are engaged on a class of work that apparently does not vary, and for some other kinds the variation is great from day to day. Where live steam is used in varying quantities there is much more chance for a difference in actual results.

The quality of the steam produced may not change, but this is not known definitely unless careful tests are made of it in accordance with instructions found on foregoing pages of this work.

The actual weight of coal consumed may not have been correctly reported, because some kinds contain much more moisture than others, and water in the furnace should not be counted as coal. When a boiler test is to be made, a fair sample of the coal should be selected, carefully dried and the percentage of moisture it contains accurately determined. It is usually convenient to take a small wooden box, fill it with a sample of the coal, then place it on the boiler setting to dry. Unless the box has just been kiln dried it is sure to contain moisture, and as this evaporates when the coal is dried, it makes enough difference to spoil the effort to fairly rate the coal used in making the boiler test.

In a certain case the small box used for this purpose weighed one-half pound less after it had been thoroughly dried, and the moisture actually evaporated in drying the sample of coal was .7 pound. If the moisture coming out of the box had been credited to the coal, and the whole pile had been judged by the sample, the entire test would have been worthless.

If the sample, as taken from the pile, weighed 18.25 pounds and after being thoroughly dried it weighed 17.5 pounds it shows that $18.25-17.5 \div 18.25 \times 100 = 4.1$ per cent. of the weight of material brought from the coal yard was water and 95.9 per cent. was coal. By courtesy the whole weight is called coal, and what is left after the weight of moisture is subtracted is called dry coal.

Assuming that 4,019.93 pounds of water were pumped into the boiler in one hour, and that there was 3 per cent. of moisture in the steam produced, the total weight evaporated is 3,899.34 pounds, and the heat used in raising the temperature of this moisture is equivalent to the evaporation of 23.78 pounds, then the final weight accounted for is 3,923.12 pounds.

If 448 pounds of coal were required and it contained 4.1 per cent. of moisture which is 18.36 pounds, the actual weight of dry coal is $448 - 18.36 = 429.64$ pounds. The actual evaporation is therefore $3{,}923.12 \div 429.64 = 9.13$ pounds of water per pound of dry coal.

A rule has already been given and explained for determining the weight of water that would have been evaporated with feed water at 212 degrees and steam at zero by the gauge. In that case the total weight of water evaporated per hour was taken, but in this case it is the weight evaporated per pound of coal that is to be compared with the given standard.

Then $9.13 \times (1{,}178.3 - 68.08) \div 965.7 = 10.5$ pounds from and at 212 degrees.

WATER EVAPORATED PER POUND OF COMBUSTIBLE UNDER GIVEN CONDITIONS

Another point to be taken into consideration in this connection is that after a boiler test is finished, so far as the burning of coal is concerned, there is more or less ash left which cannot be burned. When we consider that this varies greatly with different kinds of coal, the injustice of charging it to the boiler as actually burned, is at once plain, therefore it must be taken from the weight of dry coal burned.

Examination of several reports of boiler tests shows that the ashes remaining ranged from 6.4 to 26.2 per cent. of the entire weight of dry coal fed into the furnace. This shows that it is necessary in every case to determine the weight of dry ashes remaining at the conclusion of a test.

Suppose that in this case there were 62.75 pounds of ashes left, for each hour covered by the test. This would be $429.64 - 62.75 \div 429.64 \times 100 = 85.3$ per cent. of combustible, and as the entire weight is represented by 100 the ash amounts to $100 - 85.3 = 14.7$ per cent.

Subtracting 62.75 pounds from 429.64 shows that the weight of combustible is 366.89 pounds and as this evaporated 3,923.12 pounds of water, it demonstrates that $3{,}923.12 \div 366.89 = 10.7$ pounds were evaporated for each pound of combustible burned under given conditions.

WATER EVAPORATED PER POUND OF COMBUSTIBLE FROM AND AT 212 DEGREES FAH.

While the foregoing example seems to be complete, there is another point to be disposed of in order to cover the whole process. This consists of demonstrating the weight of water that would be evaporated per pound of combustible from and at 212 degrees Fah.

As 10.7 pounds were evaporated under given conditions which required, 1,110.22 heat units for each pound of water, then each pound of coal yielded $1{,}110.22 \times 10.7 = 11{,}879.35$ heat units. As it requires 965.7 to evaporate one pound "from and at 212 degrees," this is equivalent to the evaporation of 12.3 pounds under the assumed conditions, according to a rule given and explained on previous pages.

This forms a correct and equitable standard for comparison, and it is the only one worthy of serious consideration where accuracy is desired, as anything else that is worked out with less care is of less value, and in many cases the results secured by incorrect methods are worse than useless.

Where the actual results secured from several tests made on different boilers, to determine their efficiency in service, and these results have all been reduced to this common standard, it is not necessary to study all of the reports as the weight of water evaporated into dry steam per pound of combustible, "from and at 212 degrees," enables the engineer to compare them intelligently at short notice.

THE EFFICIENCY OF STEAM BOILERS

Steam users are frequently asked to buy appliances that will save coal, and in hundreds of plants there are opportunities to do all that salesmen guarantee for their goods, but in some cases the conditions have not been thoroughly investigated, causing enthusiastic advocates of certain inventions to make extravagant claims for them which cannot be realized in practice.

These remarks do not apply to cases where it is proposed to save steam and thus save coal indirectly, as this is not the proper place to treat them, but only to plants where it is expected to save coal directly by some appliance for the furnace.

Efforts are made in some of these cases to convince steam users that only 10 or 15 per cent. of the coal burned in a boiler furnace is taken up by the water when it is converted into steam. If this was true there would be a good opportunity to save a portion of such a great loss, but unfortunately for these enthusiasts, and fortunately for the steam user, these claims are not true, as the following example demonstrates.

The report of a test made on a vertical fire tube boiler of the most simple kind, shows that for each pound of coal burned in the furnace, 11.34 pounds of water were evaporated into dry steam "from and at 212 degrees." As it requires 965.7 heat units to evaporate one pound under these conditions, to evaporate 11.34 pounds it is necessary to use 10,951 heat units.

The number of heat units in the brand of coal used is not stated, but much of our ordinary steam coal does not contain more than 13,000 heat units per pound, in which case the efficiency of this boiler under ordinary working conditions is $10{,}951 \div 13{,}000 \times 100 = 84$ per cent. Assuming that the coal contains 13,500 heat units, the efficiency is 81 per cent. and for 14,000 heat units it is 78 per cent. This should show every reader that it is impossible to save from 25 to 50 per cent. which has been claimed in some cases.

The vertical fire tube boiler apparently affords an excellent chance for heat to ascend in the tubes and escape to the stack, but this is more apparent than real provided the boiler is well proportioned. During the above mentioned test, the temperature of the escaping gases was 427 degrees. The steam pressure was 60 pounds absolute, the temperature of which is 292.5 degrees, and it was superheated 18 degrees by the tubes passing through the steam space, therefore the final temperature was 310.5 degrees.

As the escaping gases were only 116.5 degrees above this, there cannot be much improvement made at this point, because there must be more heat in the gases than there is in the steam, or else heat will escape from the latter to the former, thus lowering the efficiency of the boiler.

This is the principal source of loss from a well set boiler and an ordinary furnace, but measures should be taken to prevent radiation of heat from all other parts, and all holes in brick setting around cast iron fronts, in connection with stacks, and in all other places where cold air can be drawn in to cool the heated surfaces, should be carefully closed, as each one represents loss of drawing power on the fire, or of heat from the coal consumed. This is an important point, for when cold air is drawn inward the defect is not noticed as quickly as when heat comes outward, yet all such imperfections are a direct source of loss.

THE LOAD AND THE STEAM PRESSURE

The load on a steam boiler is frequently estimated by the pressure carried on it in everyday service, as men who are not well informed on the subject of steam engineering conclude that when a boiler carries a high pressure there must necessarily be a heavy load on it, but while this may be true, and it frequently is, still it cannot be laid down as a principle because there are many boilers in use carrying a low pressure for heating purposes, yet the load on them is heavy.

The load on a boiler is determined by the weight of water evaporated into dry steam in a given time, and if 20 pounds gauge pressure is enough for a certain place, that does not prevent conditions from demanding a rapid rate of evaporation.

On the other hand the nature of the business for which another boiler supplies steam may be such that while a gauge pressure of 200 pounds or more may be required, only a comparatively small quantity of steam is used, hence the load is light.

Nobody would allow a boiler used under such conditions to be run without a fireman or an engineer in constant attendance, yet we are sometimes asked if steam cannot be left on a building all night, without anybody in attendance on the boiler, because there is not a very high pressure on, consequently there can be no danger.

Suppose a boiler should be left alone all night, and soon after the fireman has departed the gauge glass should break. What would be the consequence by the time that he returned in the morning? What would become of a fire if left without attention for 10 or 12 hours?

Boilers are frequently run all day, the fires banked at night, while the pressure is but little lower than is required during the day, and when the fireman returns the next morning he finds a working pressure of steam on.

This does not prove that steam could be used during the night without loss, as some managers seem to think, for if heat is taken from the boilers after the fires are banked there is just so much less left for use in the morning. Whatever remains at night after a day's run, is not lost as it is only stored for future use. Coal that is used for banking a fire is either there for use when a bright fire is wanted or else a portion of it is burned and the resulting heat has kept the boiler just so much nearer ready for use when wanted at short notice.

SECTION 2

BOILER FEEDERS

REMARKS ON BOILER FEEDERS

When a boiler feeder is to be selected for a steam plant, there are several points to be considered, which are apparently not always remembered, judging by the illogical selections made in some cases. This does not include the efficiency of the boiler feeder as a machine, for that will be considered separately, because its importance warrants it, but to other conditions which make a certain kind suitable for some places and unsuitable for others.

For illustration, take a plant where the nearest revolving shaft from which power can be obtained is perhaps 100 feet distant from the boiler room. A power pump is not suitable for such a plant because the fireman will have to leave his boilers too many times in order to attend to the pump and regulate the supply of feed water delivered, thus not only diverting his attention from duties in front of his boilers which require constant care and labor, but also causing much useless walking, and furthermore such practice results in his being absent from the boiler room without good reasons whenever he feels so inclined, to the detriment of good service.

If more or less of the feed water is warm before it passes through a heater it is not a good idea to use an injector, as it is not reliable under such conditions. Many of them will take warm or even hot water when new and in perfect order, but after they have been used long enough to cause slight wear in the tubes they "kick" especially if the water is warmer than usual, for in all cases the incoming water must be cool enough to condense the steam used to operate the injector.

It will pay to install a good feed water heater in practically every case, but where this valuable feature must be omitted for any cause, an injector should always be used as it delivers hot water to the boilers, thus preventing the great strains on plates, due to feeding cold water.

In electric power and lighting plants where current is always available a power pump driven by a motor is considered an up-to-date machine. As it is practicable to vary the speed of such a motor to meet the requirements of one or more boilers, and as it may be located in the most convenient place without regard to the nearest shaft or the most available steam pipe, it forms a very desirable combination for feeding boilers.

THE EFFICIENCY OF BOILER FEEDERS

The mechanical efficiency of a boiler feeder is found by dividing the power actually used in forcing water through pipes and valves into boilers against the pressure carried, by the power required to operate the machine. In the form of a formula it appears as follows:

$$\frac{A}{P} = M.E.$$

A = Actual power required to force water into boilers.

P = Power required to operate the machine.

M E = Mechanical efficiency.

For example, suppose that 4.8 horse power are actually used in forcing water into a battery of boilers, and it requires 6 horse power to operate the machine. The mechanical efficiency is therefore

$$\frac{4.8}{6} = .80$$

From this example it will be plain that the mechanical efficiency of a boiler feeder (or any other machine), is always a fraction, because it requires a perfect machine in which there is absolutely no power lost, to equal unity, or 1, and it is impossible to devise and construct such a machine.

If a pump is used for this purpose and the pistons are packed very tightly, the value of P will be large, consequently the fraction resulting from an application of the formula is small, or in other words the mechanical efficiency is low.

Where it is desired to express the efficiency in the form of a percentage of the whole power used the formula becomes

$$\frac{A}{P} \times 100 = \text{M. E.}$$

The thermal efficiency of a boiler feeder is the amount of heat used in forcing water into the boilers, divided by the heat put into the machine to operate it. It is found in accordance with the above explanation relating to mechanical efficiency.

POWER PUMPS

When selecting a boiler feeder the mechanical efficiency of all kinds taken into consideration should be noted, but a certain kind may show high mechanical efficiency when taken as a machine designed for this purpose, and when used in connection with some other appliance the combination may give excellent results, but when the boiler feeder is considered alone it may be wasteful and unsatisfactory.

The improved up-to-date power pump is a good illustration of this principle because when in good repair its mechanical efficiency is high. Furthermore, the quantity of steam required to operate it is small, as the power needed is developed in the main engine which is generally economical in the use of steam. Still it would be very poor practice to use a power pump to force water into one or more boilers without using a heater to raise the temperature as high as possible. This combination gives excellent satisfaction in practice, provided the pump can be conveniently located near a shaft used for other purposes, so that but little power will be used in operating the transmission devices, whatever they may be, between the main engine and the pump.

Fig. 17 illustrates a single acting power pump fitted with one cylinder. Water is drawn in on the upward stroke and forced out on the downward stroke of the plunger. It is a very simple pump, but its capacity is small and its mechanical efficiency is comparatively low, as there is but one effective stroke for each revolution of the shaft.

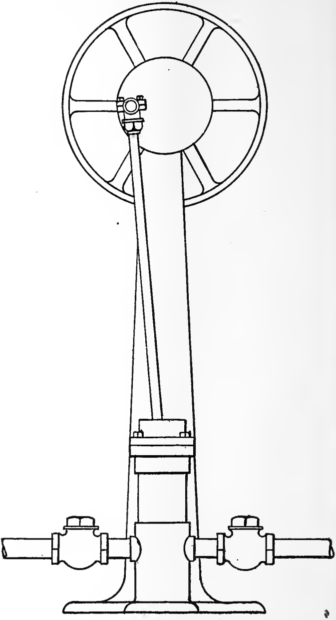


FIG. 17

When a pump of this kind is fitted with two cylinders, its capacity is doubled and its mechanical efficiency is higher, because the friction is not increased in proportion to the increase in capacity. As there are two effective strokes for each revolution, the flow of water is nearly continuous, although its speed is not constant. When fitted with three cylinders this is known as the

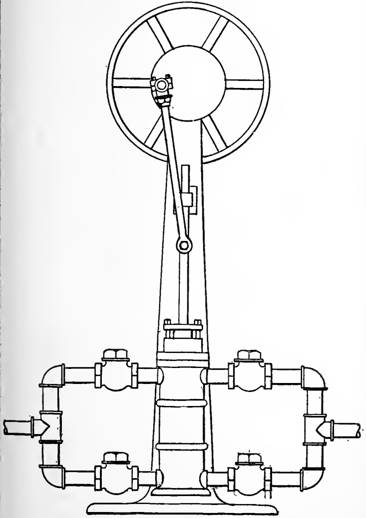


FIG. 18

single acting triplex pump. It delivers a nearly constant stream of water.

Fig. 18 illustrates a double acting power pump fitted with one cylinder. As it draws in water and expels it at each stroke, it gives fair results in practice.

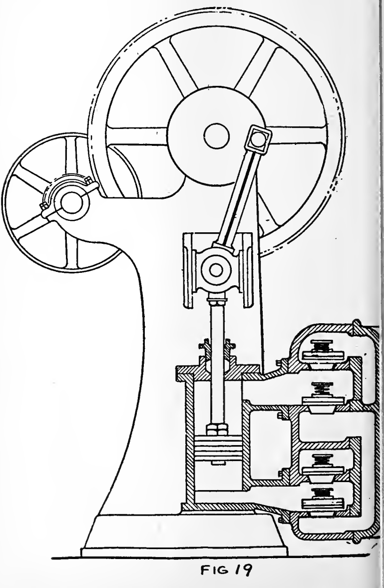


FIG 19

When this kind of a pump is fitted with two cylinders, and the driving cranks are set at an angle of 90 degrees, the flow of water is practically continuous, and the mechanical efficiency is satisfactory.

These pumps are sometimes fitted with three cylinders, and as the pistons are double acting, an even and continuous flow of water is secured, when the driving cranks are set at an angle of 120 degrees.

Fig. 19 illustrates the arrangement of valves in this machine, which is known as the double acting triplex pump.

HORSE POWER REQUIRED BY PUMPS

The first part of the next table gives the power required to operate Gourd's single acting triplex pumps, when forcing water against columns of water of stated height, or in other words the given head in feet, which gives corresponding pressure. The head includes the height from the point where water is lifted to the highest part of the delivery pipe.

The second part applies to double acting triplex pumps at given capacities.

When comparing the power required by these two kinds of pumps, the capacity must be considered in connection with the size of pump.

For illustration, a 7×8 inch single acting pump when working against 108 pounds pressure requires 15 horse power for the given capacity. A 7×8 inch double acting pump would require 30 horse power at the same rate, but the capacity of the latter is not twice as much as the former, therefore when the table states that 28 horse power will be required, it is a logical conclusion.

The first column contains the diameter and stroke of the pump under the heading "D. & S." The second gives the usual capacity, or the number of gallons that each will deliver at normal speed under the heading of "Capacity." Succeeding columns give first the head or height of the column of water, and under it the pressure due to this height of water. This represents boiler pressure where the pump is used to feed boilers instead of raising the water to the given height. Figures under these headings are the horse power required for stated conditions.

POWER REQUIRED TO OPERATE TRIPLEX PUMPS

Single Acting

D. & S.	Capacity	Height in feet of a column of water. 50	100	150	200	250	300	350
		Corresponding pressure in lbs. per sq. in. 21	43	65	87	108	130	150
2 x3	6	50	.50	.70	.90	1.05	1.20	1.33
2½x4	12	.60	1.00	1.36	1.50	1.85	2.00	2.33
3 x4	18	.75	1.10	1.60	2.00	2.50	2.90	3.15
3½x4	25	.83	1.30	1.80	2.25	2.70	3.10	3.25
4 x4	32	1.20	1.50	2.00	2.40	3.00	3.70	4.00
4 x6	50	1.90	2.50	3.10	3.75	4.80	5.60	6.25
5 x6	75	2.00	3.50	4.00	5.00	6.25	7.50	8.75
5 x8	90	2.50	4.00	5.00	6.00	7.50	9.00	10.50
5½x8	110	3.50	4.50	6.00	7.50	9.70	11.50	12.50
6 x8	132	3.60	4.50	7.00	8.70	11.00	13.00	15.50
6 x10	154	4.05	6.00	8.00	10.00	12.75	15.00	17.80
6½x8	153	4.25	6.00	8.00	10.00	12.75	15.00	17.80
7 x8	180	5.00	7.00	9.50	12.00	15.00	18.00	21.00
7 x10	209	5.25	7.80	10.75	13.50	17.25	21.00	23.33
8 x8	234	5.85	9.00	12.00	15.00	19.50	23.50	25.50

Power Required to Operate Triplex Pumps—*Continued*

Single Acting

D. & S.	Capacity	Height in feet of a column of water: 50	100	150	200	250	300	350
		Corresponding pressure in lbs. per sq. in.: 21	43	65	87	108	130	150
8 x10	273	7.00	10.50	15.00	18.25	22.75	26.00	30.00
8 x12	312	8.25	12.00	17.00	22.00	26.00	30.00	34.50
9 x10	344	8.50	13.00	18.00	23.00	28.00	33.00	37.50
10 x10	428	10.60	16.20	22.50	28.70	35.00	41.10	46.80
11 x10	516	12.70	19.50	27.00	34.50	42.00	49.50	56.20
12 x10	617	15.30	23.40	32.40	41.40	50.40	59.40	67.50
8½x12	352	8.50	14.00	19.00	23.50	28.00	33.00	38.00
9 x12	396	9.50	15.60	21.30	26.30	31.30	36.90	42.50
10 x12	488	11.10	19.00	26.00	33.00	40.50	46.00	53.00
11 x12	592	13.50	23.00	32.00	40.50	48.25	56.00	64.75
10 x16	652	14.80	25.30	35.20	44.50	53.00	61.60	71.20
12 x12	704	16.00	28.00	38.00	47.50	57.75	68.00	76.50
12 x14	779	17.60	30.80	41.80	52.25	63.32	74.80	84.10
13 x12	826	18.70	32.70	44.50	55.60	67.50	79.50	89.50
14 x12	958	21.70	38.00	51.60	64.60	78.50	92.50	104.00
15 x12	1,100	24.90	43.60	59.20	74.10	90.00	106.00	119.00

Power Required to Operate Triplex Pumps—*Continued*

Single Acting

12 x16	846	19.00	33.00	45.00	55.00	69.00	81.50	93.00
13 x16	990	23.00	40.00	54.00	68.00	82.50	97.50	108.00
14 x16	1,152	26.50	44.70	62.00	77.60	93.00	109.00	127.00
15 x16	1,321	30.50	51.50	71.30	89.00	107.00	125.00	146.00
12 x18	897	19.50	35.00	48.00	67.00	74.00	84.00	98.00
13 x18	1,050	23.00	41.00	56.00	70.50	85.00	101.00	113.00
14 x18	1,222	27.50	47.00	64.00	81.50	98.00	114.50	134.00
15 x18	1,403	32.00	56.00	76.00	95.00	115.00	136.00	153.00

POWER REQUIRED TO OPERATE TRIPLEX PUMPS

Double Acting

4 x6	94	2.40	3.90	5.20	6.30	7.50	8.80	10.10
4½x8	140	3.50	5.90	7.80	9.40	11.20	13.20	15.20
5 x8	175	4.40	7.30	9.75	11.70	14.00	16.40	19.00
5½x8	211	5.30	8.80	11.70	14.00	17.00	19.70	22.80
6 x8	252	5.75	10.20	13.50	16.80	20.00	23.50	27.00
6½x8	297	6.75	12.00	16.00	19.70	23.50	27.60	31.70

Power Required to Operate Triplex Pumps—*Continued*

Double Acting

D. & S.	Capacity	Height in feet of a column of water 50	100	150	200	250	300	350
		Corresponding pressure in lbs. per sq. in. 21	43	65	87	108	130	150
7 x8	346	8.50	13.00	18.00	23.00	28.00	32.00	37.00
7 x10	411	9.35	15.70	21.50	27.00	32.50	38.00	44.00
8 x10	533	12.00	20.00	28.50	34.00	42.00	50.00	58.00
8 x12	612	14.00	24.00	33.00	41.60	50.00	58.50	68.00
8½x12	684	15.70	26.50	36.70	45.60	55.00	64.30	75.00
9 x12	776	18.00	29.80	40.00	51.70	62.00	72.70	84.00
10 x12	952	21.60	36.60	50.00	63.40	76.00	89.20	103.00
11 x12	1,164	26.50	44.70	62.00	77.60	93.00	109.10	127.00
12 x12	1,368	31.50	53.00	73.50	91.60	110.00	128.70	150.00
11 x14	1,276	29.00	49.00	67.00	85.00	103.00	120.00	140.00
12 x14	1,531	35.00	58.00	80.00	101.00	122.00	143.00	166.00
10 x16	1,170	26.70	44.30	61.10	77.10	93.20	109.00	126.00
12 x16	1,700	38.50	64.00	88.00	111.00	134.00	157.00	183.00
14 x16	2,117	43.75	80.00	110.00	140.00	170.00	200.00	233.00

HEAD NECESSARY TO GIVE REQUIRED PRESSURE

When investigating this matter the engineer in charge of a plant may wish to know the head in feet, necessary to secure a given pressure, and while it is an easy matter to calculate it by multiplying the required pressure by 2.3, as this gives results that are practically correct, it saves time and labor to obtain these from a table, therefore the next table contains the "Feet Head" or the height of a column of water necessary to produce the pressures given in the first column.

PRESSURE AND HEAD OF WATER

Pounds per square inch	Feet head	Pounds per square inch	Feet head
1	2.31	110	253.98
2	4.62	120	277.07
3	6.93	125	288.62
4	9.24	130	300.16
5	11.54	140	323.25
6	13.85	150	346.34
7	16.16	160	369.43
8	18.47	170	392.52
9	20.78	180	415.61
10	23.09	190	438.90
15	34.63	200	461.78
20	46.18	225	519.51
25	57.72	250	577.24
30	69.27	275	643.03
40	92.36	300	692.69
50	115.45	325	750.41
60	138.54	350	808.13
70	161.63	375	865.89
80	184.72	400	922.58
90	207.81	500	1,154.48
100	230.90	1,000	2,308.00

PRESSURE SECURED BY A GIVEN HEAD

For other purposes it may be convenient to know how much pressure will be secured with a given head in feet, and while it can be calculated by multiplying the head by .434 as this will give practically correct results, as before mentioned, it is frequently much more convenient to consult a table and secure the required information at a glance.

Working engineers and others sometimes find it difficult to learn the use of tables, but a few hours spent in studying the subject will usually be sufficient to learn the value of these tables, especially if the more simple ones are studied first.

HEAD AND PRESSURE OF WATER

Head in feet	Pounds per square inch	Head in feet	Pounds per square inch
1	.43	110	47.64
2	.87	120	51.97
3	1.30	130	56.30
4	1.73	140	60.63
5	2.17	150	64.96
6	2.60	160	69.29
7	3.03	170	73.63
8	3.40	180	77.96
9	3.90	190	82.29
10	4.34	200	86.62
20	8.67	225	97.45
30	12.99	250	108.27
40	17.32	275	119.10
50	21.65	300	129.93
60	25.99	325	140.75
70	30.32	350	151.58
80	34.65	400	173.24
90	38.98	500	216.55
100	43.33	1,000	433.30

HORSE POWER REQUIRED TO RAISE WATER UNDER PERFECT CONDITIONS

Actual conditions under which water is raised to different heights are always imperfect, as water passing through rough pipes, crooked passages in valves, around corners and past other obstructions causes friction, which in turn wastes power and increases the cost of operation.

The power thus developed, and used for no profitable purpose is a variable quantity, therefore when a plant is designed, this lost power can only be estimated after taking various conditions into consideration. After the plant is put into operation the amount of power wasted can be accurately determined by ascertaining the power actually used in raising the water, then subtracting the power that would be required if the conditions were perfect.

When assuming perfect conditions all friction is neglected, and the power required is then determined by multiplying the weight of water raised per minute by the height to which it is elevated, and dividing the product by 33,000.

While the horse power lost by friction is determined by subtracting the amount required under perfect conditions from the actual power used in practice, this process does not determine the friction of the pump as it includes all lost power previously mentioned, therefore it ought not to be charged to the pump only.

When calculating power for perfect conditions, it is customary to use the term "theoretical horse power." While this is brief and convenient it is not correct, therefore its use should be discouraged on all occasions. Correct theory always agrees with practice along these lines, for when all conditions are definitely known, theory takes them all into account. When they are not all known it is customary to ignore the uncertain points and call what is left the theoretical result. While custom has established this precedent, it gives a wrong idea concerning the value of theoretical calculations.

The next table gives the power that would be required to raise water to various heights, provided there was no friction to cause waste of power, or under perfect conditions. It is based on the following rule.

Multiple the number of gallons pumped per minute, by the total height to which it is raised and divide by 4,000.

The results secured by this rule and the one preceding it are practically identical, hence either may be used at pleasure.

POWER REQUIRED TO RAISE WATER

Friction not included

Feet / Gallons	5	10	15	20	25	30	35
5	.006	.012	.019	.025	.031	.037	.044
10	.012	.025	.037	.050	.062	.075	.087
15	.019	.037	.056	.075	.094	.112	.131
20	.025	.050	.075	.100	.125	.150	.175
25	.031	.062	.093	.125	.156	.187	.219
30	.037	.075	.112	.150	.187	.225	.262
35	.043	.087	.131	.175	.219	.262	.306
40	.050	.100	.150	.200	.250	.300	.350
45	.056	.112	.168	.225	.281	.337	.394
50	.062	.125	.187	.250	.312	.375	.437
60	.075	.150	.225	.300	.375	.450	.525
75	.093	.187	.281	.375	.469	.562	.656
90	.112	.225	.337	.450	.562	.675	.787
100	125	.250	.375	.500	.625	.750	.875
125	.156	.312	.469	.625	.781	.937	1.094
150	.187	.375	.562	.750	.937	1.125	1.312
175	.219	.437	.656	.875	1.093	1.312	1.531
200	.250	.500	.750	1.000	1.250	1.500	1.750
250	.312	.625	.937	1.250	1.562	1.875	2.187
300	.375	750	1.125	1.500	1.875	2.250	2.625
350	.437	.875	1.312	1.750	2.187	2.625	3.062
400	.500	1.000	1.500	2.000	2.500	3.000	3.500
500	.625	1.250	1.875	2.500	3.125	3.750	4.375

Feet / Gallons	40	45	50	60	75	90	100	125
5	.05	.06	.06	.07	.09	.11	.12	.16
10	.10	.11	.12	.15	.19	.22	.25	.31
15	.15	.17	.19	.22	.28	.34	.37	.47
20	.20	.22	.25	.30	.37	.45	.50	.62
25	.25	.28	.31	.37	.47	.56	.62	.78
30	.30	.34	.37	.45	.56	.67	.75	.94
35	.35	.39	.44	.52	.66	.79	.87	1.08
40	.40	.45	.50	.60	.75	.90	1.00	1.25
45	.45	.51	.56	.67	.84	1.01	1.12	1.41
50	.50	.56	.62	.75	.94	1.12	1.25	1.56
60	.60	.67	.75	.90	1.12	1.35	1.50	1.87
75	.75	.84	.94	1.12	1.40	1.69	1.87	2.34

Power required to raise water—*Continued*

Friction not included

Ft. Gallons	40	45	50	60	75	90	100	125
90	.90	1.01	1.12	1.35	1.68	2.02	2.25	2.81
100	1.00	1.12	1.25	1.50	1.87	2.25	2.50	3.12
125	1.25	1.41	1.56	1.87	2.34	2.81	3.12	3.91
150	1.50	1.69	1.87	2.25	2.81	3.37	3.75	4.69
175	1.75	1.97	2.19	2.62	3.28	3.94	4.37	5.47
200	2.00	2.25	2.50	3.00	3.75	4.50	5.00	6.25
250	2.50	2.81	3.12	3.75	4.69	5.62	6.25	7.81
300	3.00	3.37	3.75	4.50	5.62	6.75	7.50	9.37
350	3.50	3.94	4.37	5.25	6.56	7.87	8.75	10.94
400	4.00	4.50	5.00	6.00	7.50	9.00	10.00	12.50
500	5.00	5.62	6.25	7.50	9.37	11.25	12.50	15.62

Feet Gallons	150	175	200	250	300	350	400
5	.19	.22	.25	.31	.37	.44	.50
10	.37	.44	.50	.62	.75	.87	1.00
15	.56	.66	.75	.94	1.12	1.31	1.50
20	.75	.87	1.00	1.25	1.50	1.75	2.00
25	.94	1.09	1.25	1.56	1.87	2.19	2.50
30	1.12	1.31	1.50	1.87	2.25	2.62	3.00
35	1.31	1.53	1.75	2.19	2.62	3.06	3.50
40	1.50	1.75	2.00	2.50	3.00	3.50	4.00
45	1.69	1.97	2.25	2.81	3.37	3.94	4.50
50	1.87	2.19	2.50	3.12	3.75	4.37	5.00
60	2.25	2.62	3.00	3.75	4.50	5.25	6.00
75	2.81	3.28	3.75	4.69	5.62	6.56	7.50
90	3.37	3.94	4.50	5.62	6.75	7.87	9.00
100	3.75	4.37	5.00	6.25	7.50	8.75	10.00
125	4.60	5.47	6.25	7.81	9.37	10.94	12.50
150	5.62	6.56	7.50	9.37	11.25	13.12	15.00
175	6.56	7.66	8.75	10.94	13.12	15.31	17.50
200	7.50	8.75	10.00	12.50	15.00	17.50	20.00
250	9.37	10.94	12.50	15.72	18.75	21.87	25.00
300	11.25	13.12	15.00	18.75	22.50	26.25	30.00
350	13.12	15.31	17.50	21.87	26.25	30.62	35.00
400	15.00	17.50	20.00	25.00	30.00	35.00	40.00
500	18.75	21.87	25.00	31.25	37.50	43.75	50.00

CONTENTS OF WATER PIPES

When making calculations relating to boiler feeders, and for other purposes, it is frequently convenient to know the contents of water pipes used. In some cases it is necessary to suspend these pipes from joists or to place them on trusses, hence the weight to be carried should be known and provisions made for safely supporting it. In the case of very long pipes the water required to fill them, and the amount available when they are emptied, make more or less difference in the results secured hence there ought to be a more convenient way at hand to determine this, than to calculate the area of pipe and its resulting capacity.

If a vertical pipe is to be carried to an elevated tank, its weight when filled with water becomes a factor to be reckoned with that should not be overlooked. The pressure due to the head of water in all such cases can be ascertained by consulting the table already given for this purpose.

The following table contains data relating to pipes, that are valuable in this connection. The several columns will be readily understood from the explanation given herewith.

N = The nominal inside diameter of pipe, or in other words the size as known to the trade.

A = The actual inside diameter in inches.

G = The number of U. S. gallons for each foot in length.

W = The weight in pounds of water at standard temperature, or 62 degrees Fah. for each foot in length.

P = The weight of pipe per foot.

C = The combined weight of water and pipe per foot.

DIAMETER AND WEIGHT OF WATER PIPES

N	A	G	W	P	C
⅛	.270	.0029	.024	.24	.264
¼	.364	.0054	.045	.42	.465
⅜	.493	.0099	.083	.56	.643
½	.622	.0158	.132	.84	.972
¾	.824	.0277	.231	1.12	1.351
1	1.048	.0447	.373	1.67	2.043
1¼	1.380	.0777	.648	2.24	2.888
1½	1.610	.1058	.882	2.68	3.562
2	2.067	.1743	1.453	3.61	5.063
2½	2.468	.2483	2.070	5.74	7.810
3	3.067	.3835	3.197	7.54	10.737
3½	3.548	.5136	4.291	9.00	13.291
4	4.026	.6613	5.512	10.66	16.172
4½	4.508	.8290	6.910	12.34	19.250
5	5.045	1.0380	8.652	14.50	23.152
6	6.065	1.5000	12.503	18.76	31.263
7	7.023	2.0120	16.771	23.27	40.041
8	7.981	2.5990	21.664	28.18	49.844
9	8.937	3.2590	27.166	33.70	60.866
10	10.018	4.0950	34.134	40.06	74.194
11	11.000	4.9370	41.153	45.02	86.173
12	12.000	5.8750	48.972	49.00	97.972
13	13.250	7.1630	59.708	54.00	113.708
14	14.250	8.2850	69.060	58.00	127.060
15	15.250	9.4890	79.097	62.00	141.097

DIRECT ACTING STEAM PUMPS

The mechanical efficiency of a direct acting steam pump is low because the friction of it is excessive under many conditions found in practice, and although this may be reduced by expert management, it is not practicable to secure best results in a majority of cases. If more attention should be paid to the proper lubrication of these pumps it would prove to be a paying investment. The common brass lubricator (so called by courtesy) is still found in numerous cases, and it is filled at irregular intervals by the engineer or the fireman. One reason for this is that there is no way whereby they can tell at a glance whether it is full or empty, and it is neglected as a natural consequence.

Another bad feature of the appliance is that after it is filled with oil, there is no principle involved in its operation whereby it feeds the oil regularly, consequently it frequently becomes empty in a few minutes, after which the valves and the steam pistons must run without oil until more happens to be supplied.

At other times the oil remains in the lubricator for a long time, leaving the internal moving parts to run dry when they need lubrication. Every direct acting steam pump should be fitted with a sight feed lubricator that feeds oil as regularly as it is fed to the main engine.

The thermal efficiency of the direct acting steam pump is low, because but little of the steam supplied to it is actually used to produce power, and some of it is condensed in the cylinders owing to the slow piston speed which conditions at the water end impose, and nearly all of the remainder passes out through the exhaust pipe.

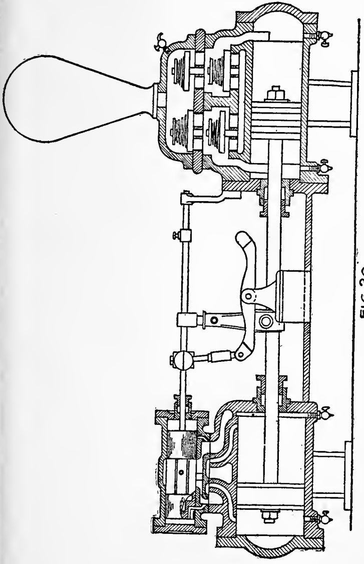
Notwithstanding these unfavorable conditions this boiler feeder is used in thousands of plants, and when all points are considered it is one of the best machines in use at the present time for this purpose.

The latter statement apparently contradicts the two which precede it, but the reasons for it are that this type of pump can easily be regulated to supply any quantity of water wanted from nothing to the full capacity of the machine, and the exhaust steam from it is frequently used to good advantage elsewhere.

This secures economy at the boiler by maintaining a constant water level, and where live steam must be used for heating provided there is not enough exhaust steam to answer the purpose, low thermal efficiency is not detrimental.

SINGLE PUMPS

Fig. 20 illustrates a single direct acting steam pump. This is an ideal kind,



provided means are supplied for reversing the piston when it has reached the end of a stroke. As there is no fly wheel used, nor any substitute for it, there is no momentum developed that can be used to operate a steam valve and cause it to admit steam to the other end of the cylinder, therefore, a device must be employed that will operate a secondary valve and admit steam to a supplemental piston, and when this moves it takes the main steam valve with it, thus securing the desired effect. A great variety of devices have been patented and used for this purpose with more or less success until perfect machines are now manufactured, leaving nothing to be desired along this line. Many working engineers remember the time when a perfect working single pump was the exception rather than the rule, but competition has changed these conditions until satisfactory single pumps are the rule to which there are very few exceptions.

It is claimed that some of this type of pumps are so well designed and constructed that they have but ⅛ inch clearance at each end of the stroke, still the steam piston does not strike the cylinder heads, neither does it make short strokes under ordinary working conditions.

A single pump must of necessity cease drawing and forcing water at the end of each stroke, unless there is a high lift by suction, or the speed is excessive, causing a body of water in the pipes to attain more or less momentum, so that when the water piston stops at the end of each stroke, the body of water continues its motion at reduced speed, until another stroke is begun which again forces the water forward. This action does not always prevent a check valve in the discharge pipe from seating frequently, and causing more or less shock and jar during the operation, but this is not objectionable at slow speed. Pump makers do not recommend such high speeds now as formerly, which is a move in the right direction. When the capacity of a pump is given, the number of strokes necessary to secure the given amount of water should be noted, especially if two or more pumps are to be compared, as otherwise the comparison may not be fair to both parties.

Where competition is great, there is a tendency to rate a pump at an excessively high speed, thus giving the manufacturer who adopts these tactics a temporary and unfair advantage over a more conservative competitor, but careful comparison of all sizes will do much towards correcting this evil.

The next table gives sizes of pumps that are recommended for boiler feeding and pumping water for other purposes under heavy pressure.

The comparative sizes of steam and water pistons, also the length of strokes will be satisfactory in practice. When selecting a pump for a certain boiler it should be remembered that a pump must supply water for the actual service required of this boiler, and not for an arbitrary rating which may or may not be correct.

It is useless to compare sizes of pipes for different pumps, as a pump may not take an excessive amount of steam to run it, because provision is made for fitting it with a large steam pipe, neither will a pump draw a large quantity of water solely on account of having a large suction pipe.

Letters at the head of these columns refer to sizes and capacities as follows:

S = Diameter of steam piston
W = " " water "
L = Length of stroke.
G = Gallons per stroke.
N = Number of strokes per minute at which the pump can be operated.
C = Capacity at given speed.

SIZE AND CAPACITY OF SINGLE PUMPS

S	W	L	G	N	C
3	1¾	3	.031	80 to 175	2.48 to 5.42
3	2	3	.04	80 " 175	32 " 7
4	2½	5	.106	75 " 150	8 " 16
5	3	6	.183	75 " 150	14 " 27
5½	3½	7	.291	75 " 150	22 " 44
6⅛	4	8	.44	75 " 150	36 " 66
7⅝	4¼	9	.55	75 " 125	41 " 69
7⅝	4½	9	.62	75 " 125	46 " 78
8½	5	10	.85	75 " 125	64 " 106
8½	5½	10	1.02	75 " 125	75 " 128
10	6	12	1.46	75 " 125	110 " 183
10	6½	12	1.72	75 " 125	129 " 215
12	7	12	2.00	75 " 125	150 " 250
12	7½	12	2.29	75 " 125	172 " 286
12	7	16	2.66	50 " 100	133 " 266
12	7½	16	3.06	50 " 100	153 " 306
14	7½	12	2.29	75 " 125	172 " 286
14	8	12	2.61	75 " 125	196 " 326
14	8½	12	2.94	75 " 125	221 " 367
14	8	16	3.48	50 " 100	174 " 348
14	8½	16	3.93	50 " 100	197 " 394
14	9	16	4.40	50 " 100	220 " 440
16	9	16	4.40	50 " 100	220 " 440
16	10	16	5.44	50 " 100	272 " 544
18	12	16	7.83	50 " 100	392 " 784

REMARKS ON SINGLE PUMPS

The manufacturers of these pumps state that they can be run at a much higher speed than their catalogues state, in case of emergency, but are careful to add that for continuous work against high pressure, it is better to run them much slower, all of which is herewith recommended for intelligent consideration of all who are interested in giving pumps a reasonable chance for long life and much usefulness.

A pump that is designed to run without outside valve gear can usually be run at a high speed without excessive vibration, but that may not be a good reason for careless use of it in running it at excessive speed for a time and then letting it stand for perhaps an equal time when it could be run at a slow speed continuously, thus greatly prolonging its life.

On the other hand a single pump that is fitted with a valve gear in which a blow is struck every time that the piston is reversed, should never be run at a high speed.

The pump illustrated in Fig. 20 is a submerged piston pump, because both suction and discharge valves are above the water piston, hence while in operation it must always be covered by water.

The natural result of this is that it has superior lifting power, as the water seal so provided assists in keeping it air tight. On this account a pump of this style is recommended where it is necessary to lift water 20 feet or more.

LIFTING POWER OF PUMPS

A pump cannot lift water higher than a column of water will stand to balance the weight of the atmosphere at that particular locality. This is about 14.7 pounds at sea level, which is equal to a column of water 33.95 feet high. The practical working lift is much less, because it is impossible to create and maintain a perfect vacuum in the suction pipe, also owing to the fact that the pump must be set lower in order to allow the water to run into it when in operation, after it has been lifted.

Furthermore, the limit of suction power of a pump is not the same in all localities, because the weight of the atmosphere becomes less as high ground is reached. The next table contains information on this point.

The Limit of Suction Power for Cold Water

ATMOSPHERIC PRESSURE AT VARIOUS ALTITUDES

Height	Absolute pressure	Equivalent head of water	Practical limit of suction
Sea level	14.70 lbs.	33.95 ft.	25 ft.
1/4 mile or 1,320 ft.	14.02 lbs.	32.38 ft.	24 ft.
1/2 mile or 2,640 ft.	13.33 lbs.	30.79 ft.	23 ft.
3/4 mile or 3,960 ft.	12.66 lbs.	29.24 ft.	21 ft.
1 mile or 5,280 ft.	12.02 lbs.	27.76 ft.	20 ft.
1 1/4 mile or 6,600 ft.	11.42 lbs.	26.38 ft.	19 ft.
1 1/2 mile or 7,920 ft.	10.88 lbs.	25.13 ft.	18 ft.
2 miles or 10,560 ft.	9.88 lbs.	22.82 ft.	17 ft.

For the foregoing table the water is assumed to be at standard temperature or 62 degrees Fah., and although it may vary several degrees in either direction without affecting the final result, the temperature can easily be raised high enough to change the limit of suction power because reducing atmospheric pressure on the surface of a body of water lowers its boiling point, hence it will evaporate, or in other words allow the pipe to fill with vapor instead of raising a solid body of the hot water.

The next table gives the maximum depth of suction power, and as the practical limit for general service is much less than the maximum, it follows that very hot water must run to a pump by gravity or else it will not be delivered to the boiler.

THE LIMIT OF SUCTION POWER
For Hot Water

Temperature Fah.	Absolute pressure	Vacuum in inches	Limit of suction power
101.4	1	27.88	31.6
126.2	2	25.85	29.3
144.7	3	23.81	27.0
153.3	4	21.77	24.7
162.5	5	19.74	22.4
170.3	6	17.70	20.1
177.0	7	15.66	17.8
183.0	8	13.63	15.5
188.4	9	11.59	13.2
193.2	10	9.55	10.9
197.6	11	7.51	8.5
201.9	12	5.48	6.2
205.8	13	3.44	3.9
209.6	14	1.40	1.6

DUPLEX PUMPS

The duplex direct acting steam pump owes its existence and great popularity largely to the fact that before it was brought into extensive use, the single pumps then in service were considered unreliable, as they frequently failed to operate when most needed.

This does not refer to any particular kind, neither does it necessarily mean that there were no good ones in evidence at that time. Although this style of pump has several defects that have not been eliminated, it will always go when steam is turned on, provided it has not been purposely disabled, or is badly out of repair.

Its mechanical efficiency is low and its thermal efficiency is not high, yet it is popular and probably always will be. If it does discharge an enormous quantity of steam for the power it develops, in many cases this steam can still be used for heating buildings in winter, and dry kilns and other departments in summer, thus reducing the real waste to a low percentage.

Fig. 21 is a side view in section of one of these pumps, fitted with submerged pistons. The whole machine is very simple, hence not liable to be deranged in rough service.

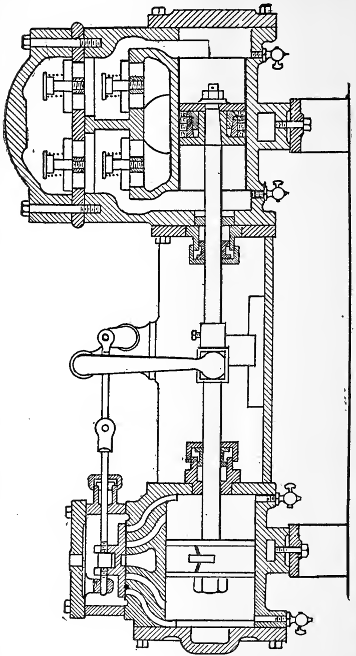


FIG. 21

As the crosshead of one pump operates the valve of the other, it is imposible to run one of them separately in case the other is disabled from any cause, for a duplex pump is only two single pumps located on one base, but they are not fitted with independent valve gears.

SETTING THE VALVES OF DUPLEX PUMPS

This process seems to be a great mystery to those who have not given the matter due attention, but when the principles involved are understood, it becomes an easy matter to apply them in practice.

By observing Fig. 21 carefully it will be noted that the steam piston is in the center of its travel, hence the first move to make in setting the valves is to move the steam piston as far in one direction as it will go, then mark the rod with a lead pencil at the edge of the stuffing box, or at some other suitable place.

Next move it as far as it will go in the opposite direction, and mark it in the same way. Place another mark central between these two and when the latter is flush with the face of the stuffing box gland, the piston is in the center of its travel.

Remove the steam chest cover and place the flat slide valve in the center of its travel, and adjust the lost motion at the valve rod so that it will be equally divided on each side of the valve.

The same device is not used for this purpose on all pumps, hence only general directions can be given at this time. If the device is adjustable, allow enough lost motion on each side of the valve to equal one-half of the width of the steam port. Repeat the operation for the other pump and the valves are set. If the stroke proves to be too long, allowing the piston to strike the cylinder head, reduce the lost motion on the valve rod. If the stroke is too short, give more lost motion.

BRASS AND FIBROUS PACKED PISTONS

Fig. 21 illustrates a piston that is made in several adjustable pieces and packed with a suitable number of rings of square fibrous packing. One advantage of this design is that if pieces of iron scale or other foreign matter from the inside of heating pipes and radiators finds its way to the cylinder of such a pump when it is taking the returns from heating systems, it is not liable to score the cylinder, although there is some danger of it.

Another advantage is found in the fact that when this packing is worn out, it can easily be replaced by the engineer, without making it necessary to employ a machinist to make repairs, or send some of the parts back to the manufacturer.

One disadvantage of this design is that if an inferior grade of packing is used by mistake or otherwise, it will not last long, and it may fail when needed badly, although it usually gives warning to the observing engineer, by partially failing before it wholly disappears.

Another disadvantage is that it is impossible to always pack the two pistons of a duplex pump exactly alike, therefore if both of them are too tight, they will both make short strokes, and greatly reduce the capacity of the pump for a given number of strokes per minute. If one is packed tighter than the other, then one will give a shorter stroke than it was intended for, reducing the capacity and making an unfavorable appearance.

Fig. 22 illustrates a pump in which no fibrous packing is used for the water end. It is fitted with a brass plunger made in one piece, working in a brass or composition ring.

The advantages of this design are that it proves durable in practice, as only comparatively hard metal is used in its construction, and as it is not adjustable there is no danger of packing one, or both of them on a duplex pump, too

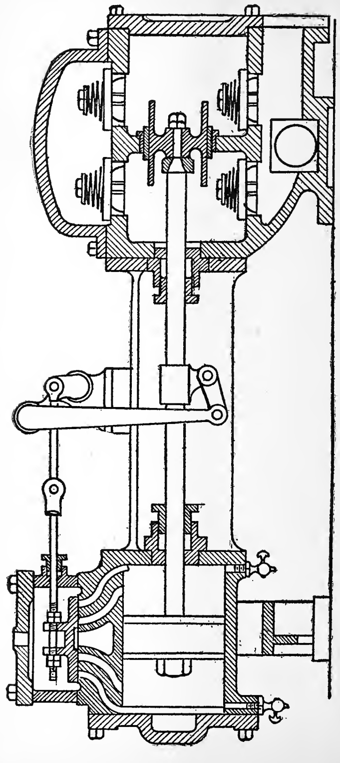


FIG. 22

tight, thus increasing friction and proving unsatisfactory.

Its disadvantages are that when it does need repairs they cannot be made by the average working engineer with the tools commonly found at hand, therefore it will usually be used in poor condition for some time before it is considered actually in need of repairs, as new parts must be sent for and put in place. If such a pump fails at a critical time it cannot be repaired quickly, unless the extra parts are kept in stock for such an emergency.

The piston in Fig. 22 is not submerged, hence this kind of a pump is not suitable for a high suction lift. Water passes through it with the least possible friction, as it enters the suction chamber, passes directly up through the lower valves and is discharged through the upper valves into the discharge chamber without changing the direction in which it travels.

All of the pumps illustrated are suitable for either hot or cold water, but valves that are appropriate for cold water will soon fail if hot water is admitted to them. Hot water valves will answer for cold water, provided they fit their seats perfectly. As hot water valves are usually made of comparatively hard material, a slightly uneven surface on either the valve or its seat will cause a leak and prevent the pump from working properly.

The next table contains the size and capacity of duplex pumps. The several columns are designated by the same letters used for the table of size and capacity of single pumps, and they refer to the same parts and conditions except the last. This is added in order to show the diameter of water cylinder that must be used to secure equal capacity in a single pump, assuming that both pistons of the duplex pump make full strokes.

S = Diameter of steam piston.

W = " " water " or plunger.

G = Gallons of water per single stroke.

N = Number of strokes per minute, at which each piston can be operated.

C = Capacity of both cylinders at given speed.

H = Diameter of the water piston of a single pump that will equal both water pistons of a duplex pump of given size.

SIZE AND CAPACITY OF DUPLEX PUMPS

S	W	L	G	N	C	H
2	1⅛	2¼	.013	100 to 300	2 to 7	1⅝
3	2	3	.04	100 " 250	8 " 20	2⅞
4½	2¼	4	.10	100 " 200	20 " 40	4
5¼	3½	5	.20	100 " 200	40 " 80	5
6	4	6	.33	100 " 150	70 " 100	5⅝
7½	4½	6	.42	100 " 150	85 " 125	6⅜
7½	5	6	.51	100 " 150	100 " 150	7
7½	4½	10	.69	75 " 125	100 " 170	6⅜
9	5¼	10	.93	75 " 125	135 " 230	7½
10	6	10	1.22	75 " 125	180 " 300	8½
10	7	10	1.66	75 " 125	245 " 410	9⅞
12	7	10	1.66	75 " 125	245 " 410	9⅞
14	7	10	1.66	75 " 125	245 " 410	9⅞
12	8½	10	2.45	75 " 125	365 " 610	12
14	8½	10.	2.45	75 " 125	365 " 610	12
16	8½	10	2.45	75 " 125	365 " 610	12
18½	8½	10	2.45	75 " 125	365 " 610	12
20	8½	10	2.45	75 " 125	365 " 610	12
12	10¼	10	3.57	75 " 125	530 " 890	14¼
14	10¼	10	3.57	75 " 125	530 " 890	14¼
16	10¼	10	3.57	75 " 125	530 " 890	14¼
18½	10¼	10	3.57	75 " 125	530 " 890	14¼
20	10¼	10	3.57	75 " 125	530 " 890	14¼
14	12	10	4.89	75 " 125	730 "1220	17
16	12	10	4.89	75 " 125	730 "1220	17

COMPOUND STEAM PUMPS

Where fuel is expensive and the exhaust steam from a pump cannot be used for other purposes, the compound pump, either single or duplex is recommended, as it will save about 30 per cent. of fuel over what a single pump requires to do the same work.

The principle involved in the use of steam in these two kinds of pumps is that in the single pump (which may be either single or duplex), steam is admitted for the whole length of every stroke, and then exhausted at practically the same pressure that it had at the beginning of the stroke, hence there is great waste of heat.

In the compound pump (which may be either single or duplex), steam is admitted to the high pressure cylinder and after its work is completed here, it is exhausted into the low pressure cylinder (which is much larger than the high pressure), where it is expanded to a comparatively low terminal pressure. As work is done during the expanding process the results so far as the consumption of fuel is concerned, are satisfactory.

Fig. 23 illustrates this type of pump, and examination of it shows that it is not a complicated machine, as it can be cared for by anybody who is competent to take care of a single pump to do the same work.

The use of steam in the second cylinder causes back pressure in the first, and this lessens the economy that would be secured if this back pressure could be eliminated, but owing to the fact that it opposes the progress of a small piston and assists a much larger one, there is a material gain by the process.

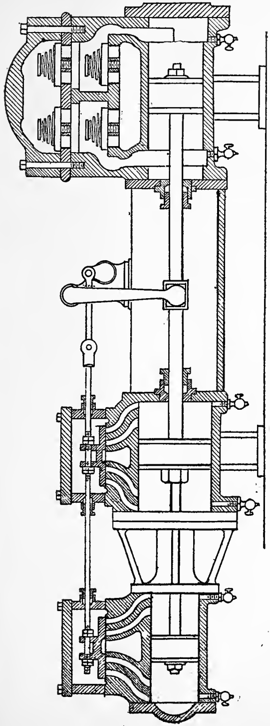


FIG. 23

The next table contains the dimensions and capacity of compound steam pumps to which the following explanation applies:

H P = Diameter of high pressure cylinder.

L P = Diameter of low pressure cylinder.

W C = Diameter of water cylinder.

S = Stroke in inches.

G = Gallons per stroke.

N = Number of strokes per minute for a single pump that can be secured in every day practice without excessive wear. The number will be doubled in case of a duplex pump.

C = Capacity in pounds per minute for a single pump, at given speed.

D = Capacity in pounds per minute for a duplex pump giving full strokes, at given speed.

Attention is called to the fact that the capacity of each is high because the piston speed is high, although the number of strokes per minute is comparatively low. These desirable features are secured by adopting a long stroke.

SIZE AND CAPACITY OF COMPOUND PUMPS

HP	LP	WC	S	G	N	C	D
4	7	4	10	.55	100	456	912
5½	8	4	10	.55	100	456	912
6	10	4	10	.55	100	456	912
7	12	4	10	.55	100	456	912
5½	8	5	10	.85	100	705	1,410
6	10	5	10	.85	100	705	1,410
7	12	5	10	.85	100	705	1,410
8	12	5	10	.85	100	705	1,410
6	10	6	10	1.22	100	1,012	2,024
7	12	6	10	1.22	100	1,012	2,024
8	12	6	10	1.22	100	1,012	2,024
9	14	6	12	1.47	100	1,220	2,440
10	16	6	12	1.47	100	1,220	2,440
12	18	6	12	1.47	100	1,220	2,440
7	12	7	10	1.66	100	1,377	2,754
8	12	7	10	1.66	100	1,377	2,754
9	14	7	10	1.66	100	1,377	2,754
10	16	7	12	2.00	100	1,660	3,320
12	18	7	12	2.00	100	1,660	3,320
14	20	7	12	2.00	100	1,660	3,320
8	12	8	12	2.61	100	2,166	4,332
9	14	8	12	2.61	100	2,166	4,332
10	16	8	12	2.61	100	2,166	4,332
12	18	8	12	2.61	100	2,166	4,332
14	20	8	12	2.61	100	2,166	4,332
9	14	9	12	3.30	100	2,739	5,478
10	16	9	12	3.30	100	2,739	5,478
12	18	9	12	3.30	100	2,739	5,478
14	20	9	12	3.30	100	2,739	5,478
16	24	9	12	3.30	100	2,739	5,478
10	16	10	12	4.08	100	3,386	6,772
12	18	10	12	4.08	100	3,386	6,772
14	20	10	12	4.08	100	3,386	6,772
16	24	10	12	4.08	100	3,386	6,772
10	16	10	14	4.75	80	3,154	6,308
12	18	10	14	4.75	80	3,154	6,308
14	20	10	14	4.75	80	3,154	6,308
16	24	10	14	4.75	80	3,154	6,308
10	16	10	20	6.80	80	4,515	9,030
12	18	10	20	6.80	80	4,515	9,030
14	20	10	20	6.80	80	4,515	9,030
16	24	10	20	6.80	80	4,515	9,030
12	18	12	14	6.85	80	4,548	9,096
14	20	12	14	6.85	80	4,548	9,096
16	24	12	14	6.85	80	4,548	9,096
12	18	12	24	11.75	50	4,872	9,744
14	20	12	24	11.75	50	4,872	9,744
16	24	12	24	11.75	50	4,872	9,744
18	30	12	24	11.75	50	4,872	9,744

THE SPEED OF STEAM PUMPS

The most difficult point to decide when ordering a steam pump, is the speed at which it will give the best results when everything is considered.

The purchaser does not want to buy a pump that is too large for the service required, as that means extra expense, hence he is inclined to take the maker's statement of the speed at which it can be run in case of emergency, for what will give good results in practice; but this is a mistaken idea because excessive speed wears a pump more than it would be worn if it were run at a slow speed for a much longer time.

Furthermore, it greatly increases the danger of failure on account of severe shocks and jars.

It is claimed that a piston speed of 100 feet per minute is the highest limit that can be allowed for a pump, and it is quite safe to say that it should never be exceeded for boiler feeding purposes, and that in many cases the limit ought to be placed at 50 feet.

It is a self-evident truth, although it may not always be remembered, that a high piston speed never damages a pump while the piston is swiftly moving in one direction, but when its motion is quickly reversed the resulting concussion causes rapid wear and loosening of the joints.

The lesson to be learned from these observations, and practical experience along the same line is that a pump should always be designed with a long stroke. It is not practicable to lay down a cast iron rule for the length of stroke, but it should be from 2 to 3 times the diameter of the water cylinder.

Some pumps are designed with a stroke longer than this indicates, as it is 4 or even 5 times the diameter of the water cylinder, but they are designated as a special kind. There is no good reason why the stroke of boiler feeders should not be made much longer than it now is in many cases.

The following dimensions are taken from the catalogue of a prominent firm who have long been engaged in the manufacture of pumps of many kinds. They would make first class boiler feeders.

LONG STROKE PUMPS

Diameter of steam cylinder	Diameter of water cylinder	Length of stroke
10	5	25
10	6	25
12	7	25
14	9	33
16	10½	33
18	12	38

The next table contains the number of strokes that pumps must make to attain speeds from 50 to 125 feet per minute, with strokes ranging from 3 to 18 inches in length.

It not only gives valuable information along this line, but it enables the reader to make comparisons readily which show that a statement of the piston speed of a pump does not always give an intelligent idea of its operation.

For illustration, a pump with an 18 inch stroke will attain a piston speed of 100 feet per minute by making 67 strokes, while another with a stroke of 5 inches must make 240 strokes to attain the same speed. A pump with a 4-inch stroke attains a speed of only 50 feet per minute when making 150 strokes, but when a pump with a 10-inch stroke makes 150 strokes it has a piston speed of 125 feet.

NUMBER OF STROKES REQUIRED TO ATTAIN A GIVEN PISTON SPEED

Speed per minute	Stroke in inches									
	3	4	5	6	7	8	10	12	15	18
	Number per minute									
50	200	150	120	100	86	75	60	50	40	33
55	220	165	132	110	94	82.5	66	55	44	37
60	240	180	144	120	103	90	72	60	48	40
65	260	195	156	130	111	97.5	78	65	52	43
70	280	210	168	140	120	105	84	70	56	47
75	300	225	180	150	128	112.5	90	75	60	50
80	320	240	192	160	137	120	96	80	64	53
85	340	255	204	170	146	127.5	102	85	68	57
90	360	270	216	180	154	135	108	90	72	60
95	380	285	228	190	163	142.5	114	95	76	63
100	400	300	240	200	171	150	120	100	80	67
105	420	315	252	210	180	157.5	126	105	84	70
110	440	330	264	220	188	165	132	110	88	73
115	460	345	276	230	197	172.5	138	115	92	77
120	480	360	288	240	206	180	144	120	96	80
125	500	375	300	250	214	187.5	150	125	100	83

CAPACITY OF PUMPS IN GALLONS

Diam.	Piston speed per minute							
in inches	40	50	60	70	80	90	100	125
1½	3.67	4.58	5.51	6.42	7.34	8.25	9.17	11.45
1¾	5.00	6.25	7.49	8.75	10.00	11.25	12.50	15.62
2	6.53	8.15	9.79	11.41	13.06	14.67	16.32	20.37
2¼	8.26	10.32	12.39	14.45	16.52	18.58	20.65	25.80
2½	10.20	12.75	15.30	17.85	20.40	22.95	25.50	31.87
2¾	12.34	15.42	18.51	21.59	24.68	27.67	30.85	38.55
3	14.69	18.36	22.03	25.70	29.38	33.04	36.72	45.90
3¼	17.24	21.54	25.86	30.16	34.48	38.78	43.09	53.85
3½	19.99	24.99	29.99	34.98	39.98	44.98	49.98	62.47
3¾	22.95	28.68	34.42	40.15	45.90	51.63	57.37	71.72
4	26.11	32.64	39.17	45.19	52.22	58.75	62.28	81.60
4¼	29.48	36.84	44.22	51.58	58.96	66.32	73.69	92.12
4½	33.05	41:31	49.57	57.83	66.10	75.35	82.62	103.27
4¾	36.82	46.02	55.23	64.43	73.64	82.84	92.05	115.07
5	40.80	51.00	61.20	71.00	81.60	91.80	102.00	127.50
5¼	44.78	56.22	67.47	78.71	89.56	101.20	112.45	140.51
5½	49.37	61.70	74.05	86.39	98.74	111.07	123.42	154.27
5¾	53.96	67.44	80.94	94.42	107.92	121.40	134.89	168.62

Capacity of Pumps in Gallons—*Continued*

Diam.	Piston speed per minute							
in inches	40	50	60	70	80	90	100	125
6	58.75	73.44	88.13	102.71	117.50	132.19	146.88	183.60
6½	68.95	86.19	103.43	120.66	137.90	155.14	172.38	215.47
7	79.97	99.96	119.95	139.94	159.94	179.92	199.92	249.90
7½	91.80	114.75	137.70	160.65	183.60	206.55	229.50	286.88
8	104.45	130.56	156.67	182.78	208.90	235.00	261.12	326.40
8½	117.91	147.39	176.87	206.34	235.82	265.30	294.78	368.47
9	132.10	165.24	198.29	231.33	264.38	297.43	330.48	413.10
9½	147.29	184.11	220.93	257.75	294.58	331.39	368.22	460.27
10	163.20	204.00	244.80	285.60	326.40	367.20	408.00	510.00
10½	179.93	224.91	269.89	314.87	329.86	404.83	449.82	562.27
11	197.47	246.84	296.21	345.57	394.94	444.31	493.68	617.10
12	235.00	293.75	352.50	411.25	470.00	528.75	587.50	734.40
13	275.80	344.75	413.70	482.65	551.60	622.55	689.50	861.90
14	319.90	399.85	479.80	559.79	639.70	719.73	799.70	999.60
15	367.20	459.00	550.80	642.60	734.40	826.20	918.00	1,147.50
16	417.80	522.25	626.70	731.15	835.60	940.05	1,044.50	1,305.60
18	528.80	660.95	793.20	925.33	1,057.50	1,189.71	1,321.90	1,652.40

EXPLANATION OF THE PRECEDING TABLES

An explanation of the practical operation of the two foregoing tables will assist the reader in making ready application of them and point out interesting features that otherwise might be overlooked.

Suppose that in a certain case 160 gallons of water are wanted per minute, and the proper size of pump is required. The piston speed is limited to 50 feet per minute for ordinary use, which is to be doubled in case of emergency. Taking the table of "Capacities of Pumps in Gallons," finding the column under 50 feet per minute and following it down we find that the next quantity above 160 is 165.24 which will answer the purpose, as the quantity delivered should never be less than the requirements. In the column headed "Diameter in inches," opposite the quantity delivered as above stated is a figure 9 indicating that the water piston must (or at least ought to) be 9 inches in diameter.

Referring to the table entitled "Number of Strokes Required to Attain a Given Piston Speed," we find that if a stroke of 15 inches is adopted it will only be necessary to make 40 strokes per minute to attain a piston speed of 50 feet, thus securing conditions that result in ease of operation, and durability of the machinery, as we will have a 9×15 inch pump running 40 strokes per minute.

Again suppose that 280 gallons of water are wanted per minute and it is to be secured with a pump that does not make more than 50 strokes per minute, and the piston speed is limited to 70 feet. What is the proper size?

Following down the column under 70 until 285.60 gallons are noted we find that a piston 10 inches in diameter will give the desired quantity. In the other table we find 70 in the column entitled "Speed per minute," and opposite this the only number below 50 is 47 strokes per minute, and by following this column up to the top we find that the proper stroke is 18 inches, therefore the pump should be 10×18 inches.

It may be necessary to determine the capacity of a pump that is available for a certain place, provided it is large enough. Suppose that the water cylinder of this pump is 4 inches in diameter with a stroke of 8 inches and it is considered safe to operate it at 120 strokes per minute for maximum speed. What is the capacity of this pump?

Under 8 in the proper column we find 120 and by following the line to the left the piston speed is found to be 80 feet per minute, which is not excessive for a maximum speed. In the other table we find 80 and by following it down until it intersects the line where 4 is located at the left hand, the capacity is found to be 52.22 gallons per minute.

A pump with a water cylinder 4 inches in diameter and a stroke of 4 inches, delivers 60 gallons of water per minute when running 300 strokes, but this speed is excessive, causing unecessary wear and much trouble. How fast will a 6 × 8 inch pump run if used to supply the same quantity of water?

In the column entitled "Diameter in inches," we find 6 and following out that line until we find 73.44 gallons, which is the next number above 60. Following this column upward shows that in order to deliver this quantity of water the piston speed will be 50 feet per minute. Referring to the other table we find that if a pump with a stroke of 8 inches is run 50 feet per minute it will make 75 strokes giving satisfactory service, therefore if the 4×4 inch pump is replaced by a 6×8 inch, the number of strokes will be reduced from 300 to less than 75, for the same amount of water, or if it makes 75 strokes it will deliver more water than the other did at 300.

STEAM AND WATER CYLINDERS

Attempts have been made to use pumps for boiler feeding that were not properly proportioned for this service, hence all efforts to make them work failed because the available force acting on the steam piston was not sufficient to overcome the resistance opposing the water piston. As this was due to ignorance on the subject, or carelessness in failing to ascertain the relative diameters of the steam and water pistons, attention is called to it here in order that readers may not make the same mistake.

Fig. 24 is a pump piston rod with a steam piston 10 inches, and a water

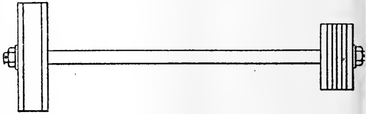


FIG. 24

piston 6 inches in diameter on it. Assuming that a steam pressure of 100 pounds to the square inch is realized, the total pressure on the steam piston is 7,854 pounds, but with the same pressure per square inch opposing the water piston, the total is 2,827 pounds, or a difference of 5,027 pounds.

This is graphically illustrated by Fig. 25 which shows the two pistons together.

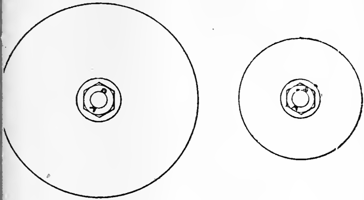


FIG. 25

If both of these pistons were the same diameter the pump would not work because the total resistance would be greater than the total force acting on the steam piston, for if there is 100 pounds pressure on a boiler it will usually be necessary to pump against at least 110 pounds.

However, it is neither advisable nor practicable to use full boiler pressure in the steam cylinder of a pump, hence the piston must be larger accordingly. With 50 pounds pressure on the steam piston, the total is 3,927 pounds, and if the water piston works against 110 pounds the total is 3,109 pounds, leaving a difference of 818 pounds, which is enough to cause the pump to run as desired. This comparison demonstrates that much less than boiler pressure will drive such a pump, and it also shows that the steam piston must always be larger than the water piston.

The next table gives the height to which pumps with given steam and water pistons or cylinders will elevate water. As it is impossible to determine the friction that will result from forcing water through a system of pipes until all conditions are known, or a trial made in practice, no attempt is made to include it in this table, therefore, the figures where the horizontal and the vertical lines intersect represent the height in feet of a column of water acting on the water piston that will balance a pressure of 50 pounds to the square inch on the steam piston. This height may be reduced to pounds by multiplying it by .43.

For illustration, take a steam piston 12 inches and a water piston 7 inches in diameter. By following the horizontal line on a level with 12 until it intersects the vertical line under 7, we find that the column of water must be 220 feet high, or $220 \times .43 = 94.6$ pounds will balance 50 pounds on the steam piston. It will be noted that wherever the diameter of the steam piston is twice the diameter of the water piston, the column of water is 300 feet high.

HEIGHT OF WATER ACTING ON A WATER PISTON TO BALANCE 50 POUNDS PRESSURE ON A STEAM PISTON

Diam. of steam pistons	Diameter of Water Pistons											
	2	2½	3	3½	4	5	6	7	8	9	10	12
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	33
9		972	675	496	380	243	169	124	95	75	61	42
10			833	612	469	300	208	153	117	94	75	50
12				881	675	432	300	220	169	133	108	75
14					920	588	408	300	228	182	147	102
16						762	564	392	300	236	192	141
18						972	650	490	379	300	243	162
20							833	600	469	370	300	208

ORDERING STEAM PUMPS

Having decided on the proper size of pump for the maximum quantity of water wanted, the engineer should take into consideration the following conditions, and inform the parties of whom the pump is to be purchased, concerning them, that there may be no misunderstanding.

Whether the water to be pumped is hot or cold, or each kind alternately. If only cold water is to be used, how high must it be lifted by suction, and how long is the suction pipe? Is the water pure or does it contain impurities that must be prevented from passing the pump?

The available steam pressure must be considered, also the pressure against which water is to be forced. This includes consideration of the length of the discharge pipe, also the number of short turns and valves in it.

It should be known whether the pump is to exhaust into a condenser, the atmosphere, or a heating system of any kind, as these conditions effect the back pressure on the steam piston, or in other words they make a difference in the total load on the pump.

When ordering parts for repairs give the shop number of the pump, also the size and serial number if there is one provided. The parts wanted are usually illustrated in the manufacturer's catalogue, therefore one of them should be kept on hand for use in an emergency. Reading and studying these catalogues enables an engineer to thoroughly understand the machines that he uses.

DIRECTIONS FOR SETTING AND OPERATING STEAM PUMPS

Having selected a place for the pump to be set, have a brick foundation built for it in order to hold it firmly in position. Do not bolt it to a light wooden floor that will not be stiff enough to prevent vibration when new, and that will soon rot and become useless.

Where a pump is to take water under pressure from a street main, the suction pipe may be made smaller than the suction opening calls for, but where water is to be lifted from a well, pond or brook, it should never be reduced even for a low lift, and for a high lift or a long suction pipe, it should be made one size larger.

Always make the suction as short and straight as conditions will admit, as short ells and globe valves add much to the necessary friction of water flowing through pipes. The highest part of a long suction pipe should always be at the pump, and the grade should be continually downward to the water. If it is raised and lowered to suit the ground on which it is located, air will collect in the high places or air pockets, and cause trouble.

Where it is to be laid under ground, cast iron flanged pipe is recommended as durable and satisfactory. Special care should be taken to keep sand, stones, and other foreign matter out of the pipe while it is being laid, as but a small quantity will cause much trouble and annoyance.

Where it is necessary to locate a valve in the straight part of a suction pipe, it should be a gate valve in order to create as little friction as possible. If there is a convenient turn in the pipe, an angle valve may be used instead of an ell. If the suction pipe is long, a foot valve should be located on the end of it, then by providing a small priming pipe, water can be admitted to the suction pipe, from an overhead tank, or a street main, thus filling the suction pipe and insuring prompt operation of the pump when steam is turned on. If this suction pipe is exposed, provision must be made for draining the water out of it when the pump is stopped in cold weather. Frost plugs in the water cylinder should be removed to allow all water to drain out.

A large air chamber should be located on the suction pipe in addition to one provided on the pump in some cases as the latter provides a cushion for the discharge pipe only. Fig. 26 illustrates

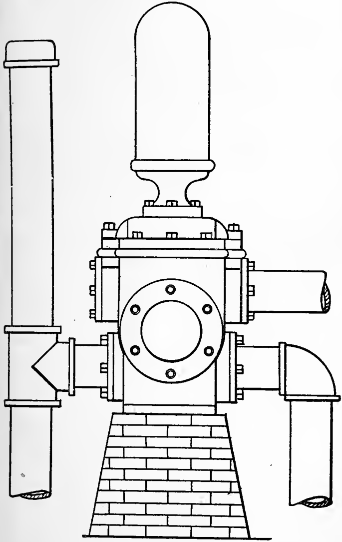


FIG. 26

the proper location of the air chamber, for either one or two suction pipes. If two suction pipes are provided, the water coming in through them will move slowly and quietly. The discharge pipe should be of ample size to allow water to move at a comparatively slow speed when the pump is operated at its full capacity.

When locating the steam pipe, make due allowance for expansion when it is heated by steam. Put a globe valve near the pump and a good double connection sight feed lubricator above it. Before admitting steam to the pump, blow out the pipes thoroughly under full pressure in order to remove red lead, iron chips, and dirt, thus preventing such foreign matter from injuring the steam cylinder.

Keep the stuffing boxes well filled with first-class packing and do not let it remain in use long enough to become hard, as it will score the rods, making it impossible to keep them from leaking steam and water.

Use as good oil on all pump bearings as you would on the best kind of an engine, and only first-class cylinder oil in the steam cylinder. If the pump is to remain idle for several days, use an extra quantity of cylinder oil during the last five minutes that it is run, in order to keep it from rusting when standing still.

The advantage gained by the use of pipes of ample size for both the suction and discharge of pumps will be made plain by studying the next table, which treats pipes from 1 to 6 inches in diameter.

G = Gallons discharged per minute.
V = Velocity of water in feet per second.
F = Friction of water in pounds per square inch for each 100 feet of clean iron pipe, or the pressure required to overcome friction.

It will be noted that as the velocity is increased, the friction increases very fast, therefore, a constant loss results from the use of pipes that are too small for the required service, hence while the first cost of them is less, they are ultimately very expensive.

For illustration, suppose that 20 gallons of water per minute are forced through a 1½ in. pipe, causing a friction loss of 1.66 pounds. If the amount of water is doubled the loss is increased to 6.52 pounds. If 35 gallons are forced through 1 inch pipe the loss is 37 pounds, but 1½ inch pipe will convey the same amount with 5.05 pounds loss.

FRICTION IN POUNDS PRESSURE PER SQUARE INCH

For each 100 feet of clean iron pipe. G. A. Ellis, C. E.

G	1 inch		1¼ inch		1½ inch		2 inch		2½ inch		3 inch		4 inch		6 inch	
	V	F	V	F	V	F	V	F	V	F	V	F	V	F	V	F
5	2.04	0.84	1.31	0.31	0.81	0.12										
10	4.08	3.16	2.61	1.05	1.82	0.47	1.02	0.12								
15	6.13	6.98	3.92	2.38	2.73	0.97										
20	8.17	12.30	5.22	4.07	3.63	1.66	2.04	0.42								
25	10.20	19.00	6.53	6.40	4.54	2.62			1.63	0.21	1.13	0.10				
30	12.30	27.50	7.84	9.15	5.45	3.75	3.06	0.91								
35	14.30	37.00	9.14	12.04	6.36	5.05										
40	16.30	48.00	10.40	16.10	7.26	6.52	4.09	1.60								
45			11.70	20.20	8.17	8.15										
50			13.10	24.90	9.08	10.00	5.11	2.44	3.26	0.81	2.27	0.35	1.28	0.09		
75			19.60	56.10	13.60	22.40	7.66	5.32	4.90	1.80	3.40	0.74				
100					18.20	39.00	10.20	9.46	6.53	3.20	4.54	1.31	2.55	0.33	1.13	0.05
125							12.80	14.90	8.16	4.89	5.67	1.99				
150							15.30	21.20	9.80	7.00	6.81	2.85	3.83	0.69	1.70	0.10
175							17.10	28.10	11.40	9.46	7.94	3.85				
200							20.40	37.50	13.10	12.48	9.08	5.02	5.11	1.22	2.27	0.71
250									16.30	19.66	11.30	7.76	6.39	1.89	2.84	0.26
300									19.60	28.06	13.60	11.20	7.66	2.66	3.40	0.37
350											15.90	15.20	8.94	3.65	3.97	0.50
400											18.20	19.50	10.20	4.73	4.54	0.65
450											20.40	25.00	11.50	6.01	5.11	0.81
500											22.70	30.80	12.80	7.43	5.67	0.96

RECEIVER PUMPS AS BOILER FEEDERS

When a receiver is located in the lower part of a building that is heated by steam, or in a dry kiln, or in any other similar place, all water resulting from the condensation of steam flows into this receiver, from whence it is taken by a pump and returned to the boilers. This pump is operated automatically by a float in the receiver which rises when the water line is raised, and opens a balanced valve in the steam pipe of the pump, thus starting it and taking out the water. When the water level falls, the float follows and shutting the balanced valve stops the pump. If water flows continuously to the receiver, the float will operate the pump at a proper speed to remove it as fast as it comes.

This is not only an excellent way to remove the water of condensation from heating systems, but it constitutes a very good boiler feeder.

As there is usually more or less water lost through drips, etc., a comparatively small quantity of fresh water must be put into the boilers every day. It is not necessary to run an extra pump for this purpose, for if a cold water pipe is tapped into the receiver, or into a convenient return pipe from the heating system, it will only be necessary to

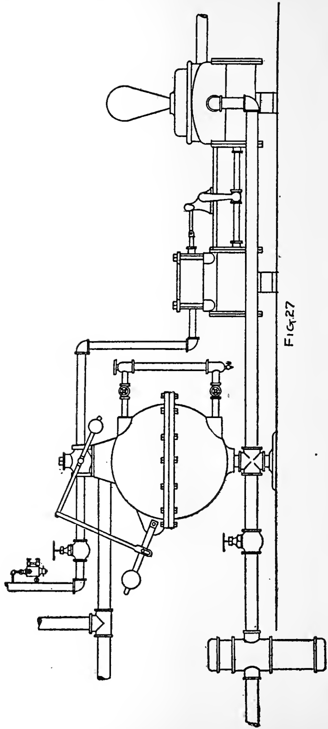


FIG. 27

admit cold water as needed and the float will do the rest. Fig. 27 illustrates a unique device for this purpose. In the cast iron receiver there is a float which operates the lever shown, and this opens and closes the balanced valve on the receiver, admitting steam to run the pump and return the water to the boiler. A double connection lubricator is on the steam pipe where it feeds oil through

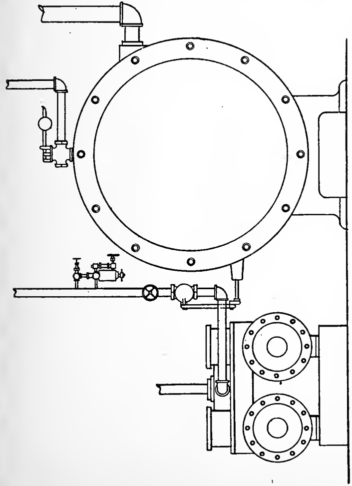


FIG 28

the balanced valve, but none of it can leak into the receiver and be carried into the boilers. A sediment trap is provided on the main return pipe which prevents foreign matter from going into the pump. If this is not provided in some form, the water cylinder of the pump will probably be badly scored by red lead, iron scale, mud, etc., coming down from the heating system.

Fig. 28 is an end view of a pump and receiver, showing the steam end of it. This has done good service in the author's plant for 15 years, and is still in good order. A double connection lubricator is used as shown, except that it is not so near the pump, but is located in the boiler room, where it can be seen by the assistant engineer. All cylinder oil passes through the balanced valve thus keeping it in good working order, but none of the oil can get into the receiver.

Fig. 29 is another end view of the same machine, showing the water end of it. The outlet is located in the head, thus allowing all sediment to stay in the receiver where it can do no harm.

No vent pipe is provided for this receiver as it forms part of a closed system

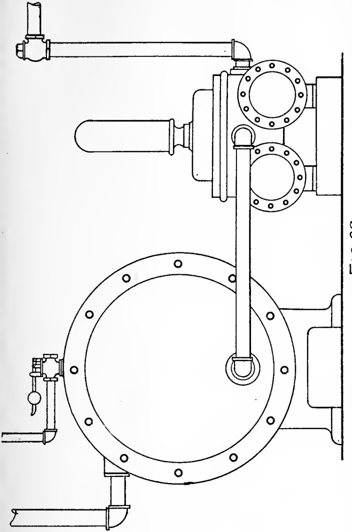


FIG. 29

in which the hot water is not released from pressure, but is returned without loss of heat. A safety valve is provided to protect the receiver from overpressure.

The following table contains the size and capacity of these pumps and receivers. The third column indicates the number of square feet of radiating surface that the pump will drain.

SIZE AND CAPACITY OF PUMPS AND RECEIVERS

Size of pump	Gallons per minute	Square feet	Space occupied		
			Length	Width	Height
3x2x3	12	5,000	34	25	31
4½x2¾x4	20	10,000	37	30	35
5¼x3½x5	35	20,000	48	37	44
6x4x6	60	40,000	54	37	45
7½x5x6	90	60,000	54	39	53
7½x4½x10	90	60,000	65	42	53

PUMPS DRIVEN BY ELECTRIC TRANSMISSION OF POWER

Fig. 30 illustrates a horizontal power pump driven by an electric motor located at the center of the frame. As a high rate of speed at the motor is desirable while the pump operates at a very moderate speed, the reduction is made by gears of suitable size.

This makes a very desirable form of boiler feeder wherever electricity is available, as the speed can be adjusted to meet the requirements of the boilers, and the machine can be located in any convenient place without regard to convenience in transmitting power as it must be considered in other cases, because a pair of wires can be carried any-

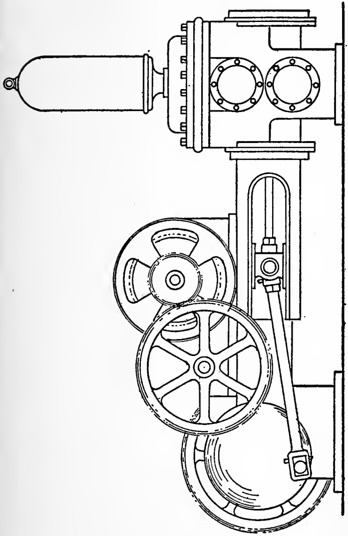


FIG. 30

where, and a reasonable length of pipe does not add much to the cost of operation, provided it is of ample size.

The following table gives size and capacity of these pumps, although larger ones are built to order.

SIZE AND CAPACITY OF ELECTRIC PUMPS

Size in inches	Speed per minute	Gallons per revolution	Gallons per minute
3 x 6	60	.366	21.96
3½ x 6	60	.498	29.88
4 x 8	50	.880	44.00
4½ x 8	50	1.100	55.00
5 x 10	40	1.700	68.00
6 x 12	30	2.920	87.60
7 x 12	30	4.000	120.00
8 x 16	25	6.960	174.80
9 x 16	25	8.800	220.00
10 x 16	25	10.880	272.00
12 x 16	25	15.660	391.50
14 x 16	25	21.320	533.00

Fig. 31 illustrates a single acting vertical triplex pump driven by an electric motor geared 5 to 1. The design and construction of electric motors have been brought so near perfection that any kind of current can be used to drive them, but when laying out a machine of this kind, it is a good idea to take the kind available into consideration, especially when corresponding with parties who make motors, as the proper kind must be selected to use the available current.

The next table gives the size and capacity of these pumps as ordinarily used. It is not good policy to select a small electrically driven pump, and

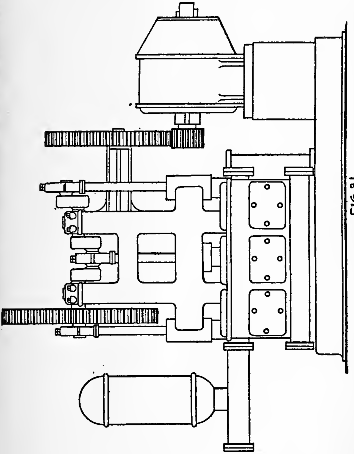


FIG. 31

operate it at a higher speed that the list calls for, either for the horizontal or the vertical types, as vibration and wear will be greatly increased by such action.

SIZE AND CAPACITY OF DIRECT CONNECTED ELECTRIC PUMPS

Size in inches	Speed per minute	Gallons per revolution	Gallons per minute
2 x 2	70	.081	5.67
2½ x 2	70	.127	8.89
2½ x 3	60	.19	11.40
3 x 3	60	.27	16.20
3½ x 3	60	.37	22
3½ x 4	60	.50	30
4 x 4	60	.65	39
4 x 6	60	.98	59
5 x 6	60	1.53	91
5½ x 8	60	2.46	147
6 x 8	55	2.94	161
7 x 8	55	4.00	220
8 x 8	55	5.22	287
8½ x 8	55	5.90	324
9 x 10	50	8.26	413
10 x 10	45	10.20	459
11 x 12	42	14.81	622
12 x 12	42	17.62	740

INJECTORS

The fact that a jet of steam when issuing from a properly proportioned and correctly located nozzle, can be used to move a body of water from one place to another, was known more than 100 years ago. It was utilized to the extent of raising water and allowing it to flow into a tank, or other receptacle below the apparatus, and later it was found practicable to raise water with it into a tank located above the apparatus, thus forming what we now call an ejector.

In 1858 the eminent French engineer, Giffard, discovered that if a jet of steam was used to raise water and it was first allowed to flow into the air on about a level with the apparatus, after working in this way until a rapid circulation was established, the overflow could be slowly shut off, when the moving jet of water and steam would force itself into the boiler against the pressure that was used to operate the apparatus, thus giving us the injector as we have it in use today. To Giffard, then, belongs the honor, not of actually inventing the injector, but of perfecting it and making a practical application of the principles involved as we understand them now.

THEORY OF THE INJECTOR

Many so-called theories which claim to explain the operation of this machine have been advanced, some of them by men who simply guess at it without having knowledge of the nature of fluids and gases, but the following is both reasonable and logical.

The jet of steam used in this machine puts into motion a body of water and as the steam strikes the water and mingles with it, a portion of the great velocity of steam flowing into the atmosphere is imparted to the mixture which is thus thrown forward. If the velocity of this mass is greater than the velocity of water flowing out of the boiler under given pressure, then the injector will feed the boiler, but otherwise it will not. This statement applies to ordinary injectors designed for boiler feeding.

In order to intelligently consider this matter it is necessary to take into account what may properly be called the three active parts of an injector.

First—A nozzle of peculiar shape and correct proportions through which steam is admitted from the boiler to the injector.

Second—A combining tube or nozzle in which steam from the nozzle above mentioned and water from the suction pipe meet, condensing the steam and imparting some of its velocity and the resulting force to the water.

Third—A delivery nozzle designed and proportioned to give the greatest possible velocity to the mixture.

THE VELOCITY OF STEAM

When considering the first part it becomes necessary to compute the velocity of steam when it is allowed to escape into the atmosphere, assuming that it has the same weight per cubic foot as when it was confined in the boiler and not taking into account the shape of the nozzle. The next formula can be used for this purpose.

$$\sqrt{64.32 \times \frac{B \times 144}{W}} = V.$$

B = Boiler pressure as indicated by the steam gauge. As this is on the square inch it is multiplied by 144 to give the total pressure per square foot.

W = Weight of one cubic foot of steam taken at absolute pressure or B + 14.7 pounds.

V = Velocity in feet per second.

For illustration take steam at 100 pounds gauge pressure, the weight of which is .2296 pound per cubic foot.

Substituting these values for the letters, and making the necessary calculation gives the following result:

$$\sqrt{64.32 \times \frac{100 \times 144}{.2296}} = 2{,}008.5$$

feet per second.

VELOCITY OF THE MIXTURE

When solving this problem the first value to determine is the weight of water taken up by the injector compared with the weight of steam used. It may be calculated by the next formula.

$$\frac{T-H}{H-W}=P.$$

T = Total heat of steam at given pressure.

H = Heat units in the water at given temperature as it leaves the injector.

W = Heat units in the water at given temperature as it enters the injector.

P = Pounds of water taken up by the injector for each pound of steam used.

For illustration take steam at 100 pounds gauge pressure as above mentioned, the total heat of which is 1,178 above 32 degrees. If water enters the injector at 65 degrees each pound of it contains 33.01 heat units and if it is discharged from the injector at 155 degrees it contains 123.34 heat units. Application of the formula gives the following result:

$$\frac{1{,}178-123.34}{123.34-33.01}=11.67 \text{ pounds of water}$$

taken by the injector for each pound of steam used, therefore the ratio is 11.67 to 1.

The velocity of the mixture can now be determined by the next formula.

$$\frac{I}{I+P}\times V=M.$$

I = Quantity of steam used which is taken at unity or 1.

P = Pounds of water taken by the injector for each pound of steam used.

V = Velocity of steam at given pressure when flowing into the atmosphere.

M = Velocity of the mixture when flowing into the atmosphere.

This is not the actual velocity of the mixture of steam and water entering the boiler, because it must overcome boiler pressure, which reduces the speed that would be attained if there was no resistance except what is due to atmospheric pressure.

Application of this formula to the example stated gives the following result:

$$\frac{1}{1+11.67}\times 2{,}008.5=158.67$$

feet per second.

This mixture of steam and water will flow into the boiler providing the velocity of it as above calculated is greater than the velocity of a jet of water would be if it was allowed to flow freely from the boiler at stated pressure. This velocity may be determined by the following formula:

$$\sqrt{64.32 \times H} = V.$$

H = Height of a column of water whose weight or pressure per square inch at the base is equal to the boiler pressure.

V = Velocity in feet per second.

The height of this column is found by dividing the number of cubic inches in a cubic foot by 12 and dividing the quotient by the weight of water per cubic foot at stated temperature. This quotient is the height of water in feet required for one pound pressure per square inch, and when this is multiplied by the pressure, the product is the total height of a column of water the pressure of which is equal to the boiler pressure.

In this case it is 100 pounds by the gauge, the temperature of which is 337.8 degrees. Water at this temperature weighs 56.62 pounds per cubic foot. Stated as a formula it appears as follows:

$$\frac{1{,}728}{12} \div W \times P = H.$$

W = Weight of a cubic foot of water at given pressure and temperature.

P = Boiler pressure by the gauge.

H = Height in feet.

Then,

$$\frac{1,728}{12} \div 56.62 \times 100 = 254 \text{ feet.}$$

The velocity with which water would flow from this boiler under 100 pounds pressure is

$\sqrt{64.32 \times 254} = 127.82$ feet per second.

The difference between the velocity of the mixture coming from the delivery tube, and water issuing from the boiler is $158.67 - 127.82 = 30.85$ feet per second, which is sufficient to account for the action of the injector.

Gauge pressure is used in these calculations because both steam and water are supposed to be flowing into the atmosphere. If they were flowing into a vacuum it would be necessary to use absolute pressure.

When calculating these results it is necessary to know the weight of water per cubic foot under various pressures with corresponding temperatures. This will be found in the next table, from zero by the gauge to 135 pounds pressure.

WEIGHT OF WATER PER CUBIC FOOT ABOVE 212° FAH.

Pressure	Temperature	Weight	Pressure	Temperature	Weight
0	212	59.83	43	290.37	57.87
0.3125	213.04	59.81	44	291.48	57.84
1	216.33	59.73	45	292.58	57.81
2	219.45	59.65	46	293.66	57.86
3	222.40	59.59	47	294.73	57.76
4	225.25	59.52	48	295.78	57.74
5	227.95	59.45	49	296.82	57.71
6	230.60	59.38	50	297.84	57.68
7	233.10	59.32	51	298.85	57.66
8	235.49	59.27	52	299.85	57.63
9	237.81	59.21	53	300.84	57.60
10	240.07	59.15	54	301.81	57.58
11	242.24	59.09	55	302.77	57.55
12	244.32	59.05	56	303.72	57.52
13	246.35	59.00	57	304.69	57.50
14	248.33	58.95	58	305.60	57.48
15	250.26	58.90	59	306.52	57.45
16	252.13	58.85	60	307.42	57.43
17	253.98	58.80	61	308.38	57.41
18	255.77	58.75	62	309.22	57.38
19	257.52	58.71	63	310.11	57.36
20	259.22	58.67	64	310.99	57.34
21	260.88	58.63	65	311.86	57.32
22	262.50	58.59	66	312.72	57.30
23	264.09	58.56	67	313.57	57.27
24	265.65	58.51	68	314.42	57.25
25	267.17	58.48	69	315.25	57.23
26	268.66	58.43	70	316.08	57.21
27	270.12	58.40	71	316.90	57.18
28	271.55	58.36	72	317.71	57.16
29	272.96	58.33	73	318.51	57.14
30	274.33	58.29	74	319.31	57.12
31	275.68	58.26	75	320.10	57.10
32	277.01	58.23	76	320.88	57.08
33	278.32	58.19	77	321.66	57.05
34	279.62	58.16	78	322.42	57.03
35	280.89	58.13	79	323.18	57.01
36	282.14	58.09	80	323.94	56.99
37	283.39	58.06	81	324.67	56.97
38	284.58	58.03	82	325.43	56.95
39	285.76	58.00	83	326.17	56.93
40	286.96	57.97	84	326.90	56.91
41	288.06	57.94	85	327.63	56.89
42	289.24	57.91	86	328.35	56.87

Weight of Water per Cubic Foot Above 212° Fah.

Pressure	Temperature	Weight	Pressure	Temperature	Weight
87	329.07	56.86	98	336.58	56.66
88	329.78	56.84	99	337.23	56.64
89	330.48	56.82	100	337.89	56.62
90	331.18	56.80	105	341.00	56.54
91	331.87	56.79	110	344.1	56.46
92	332.56	56.72	115	347.1	56.38
93	333.24	56.75	120	350.0	56.30
94	333.92	56.73	125	352.8	56.22
95	334.59	56.71	130	355.6	56.14
96	335.26	56.69	135	358.4	56.07

EFFICIENCY OF THE INJECTOR

Engineers who take pride in claiming to be practical men only, seem to be still further satisfied with their position when told that theoretically an injector is the most efficient boiler feeder known to modern science. This conclusion on their part is due to the fact that in practice more fuel is required to maintain a given steam pressure when feeding with one of these very simple machines, than when using a direct-acting steam pump, or one of the many forms of power pumps, hence it does not seem to be an efficient machine.

While this appears to be a conclusive argument, it is not really so. Experiments made by Prof. Carpenter show that when considered only as a pump, an injector has a very low efficiency when compared with other kinds, as it developed from 161,000 to 2,752,000 foot pounds of work for each 1,000 pounds of steam used, or its equivalent in burning 100 pounds of coal.

As small direct acting steam pumps, which are known to be very wasteful, develop from 4,000,000 to 8,000,000 foot pounds on the same basis, the difference is apparent, and when compared with high grade pumping engines which develop from 100,000,000 to 140,000,000 foot pounds, the theoretical rating of injectors when considered as pumps only, agrees very closely with the practical engineer's idea of their performance in every day use.

The fact remains, however, that the injector is a very efficient boiler feeder, because all heat in the steam used (except a small part lost by radiation) is returned to the boiler in the feed water. Its seeming inefficiency is due to the fact that live steam is used for the entire process, while with pumps the heating is usually done with exhaust steam that is a waste product. In other words an injector alone is compared with a pump and a heater operating in combination, which is evidently unfair.

CAPACITY OF INJECTORS

The maximum capacity of injectors is given in the manufacturers' catalogue, but this does not necessarily mean that they will deliver the given amount of water under all conditions, therefore, when selecting an injector to feed one or more boilers, it is a good idea to decide on the kind wanted, then select one that will deliver fully as much as can be used under stated conditions.

Put a globe valve in the suction pipe and regulate the supply by it to meet the varying demands.

If a valve in the suction pipe can be used to regulate the supply, it is plain that if the pipe is lengthened it will affect the amount of water brought to the injector, hence this must be taken into consideration. A careful test was made on a certain injector that delivered 108 cubic feet of water per hour when lifting it one foot. The lift was increased one foot for each trial until it measured 8 feet, then 2 feet for each until it was 20 feet, but the steam pressure was carefully kept uniform during the whole test.

The increased lift reduced the capacity to about 52 cubic feet per hour. As these trials were made in rapid succession and the capacity was less every time that the lift was increased, the reduction could not be charged to any other change. As this shows a reduction of more than 50 per cent. it may not be practicable to depend on so great a change in every day practice, but a reduction of 40 per cent. can be made without affecting the reliability of the machine as a boiler feeder, and that should be considered sufficient.

It may not be possible to do this with every injector in the market, but where it is impossible to thus reduce the capacity, provided the water is not warmer than 70 degrees Fah., it is evidence of an inferior machine. If a greater variation than this is required in any special case, two injectors should be installed, thus giving a range of from 60 per cent. of the capacity of one to the full capacity of both, or in other words this plan allows a reduction of 70 per cent. from the full capacity.

When the manufacturer of an injector publishes a list of sizes and the capacity of each size, the given quantities are usually based on the use of a stated steam pressure. If a higher pressure is carried in service, the capacity will be increased, and where it is lower the amount delivered will be less accordingly.

TYPES OF INJECTORS

Fig. 32 represents a type of injector that was considered very good until

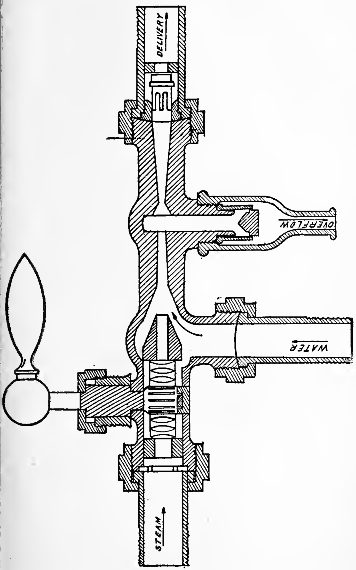


FIG. 32

better ones were invented. It would lift water and deliver it to a boiler, but it required adjusting for every change in steam pressure, which is not considered good practice now. If too much steam was admitted some of it would escape at the overflow, and if the supply of water was too great, a portion of it came out of the same place, thus showing the need of attention in either case.

Fig. 33 is an up-to-date automatic injector. It requires no adjustment

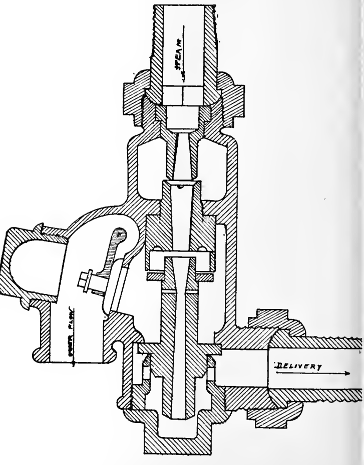


FIG. 33

for varying steam pressure, and when lifting water it is not necessary to give

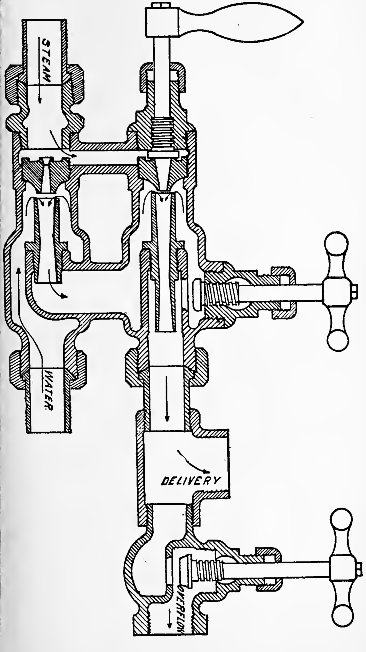


FIG. 34

this part any attention unless too much is taken up, when the supply may be reduced.

Fig. 34 represents one kind of a Hancock inspirator, which is another name for a double tube injector. One set of tubes lifts the water and delivers it to the other which forces it into the boiler. Although it has three valves to operate it is not a complicated machine, and as each valve must be handled separately, it is not difficult to locate a defect which

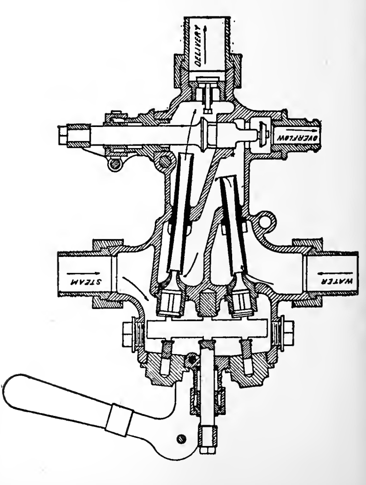


FIG. 35

prevents smooth operation on account of this feature.

Fig. 35 represents a double tube injector in which all valves are connected and are operated by one lever, therefore the starting and stopping process consists of one operation only, so far as the injector is concerned.

Fig. 36 is an exhaust steam injector that will feed a boiler against 75 pounds pressure, without using live steam. If more than 75 pounds are carried, a small quantity of live steam must be used. In the former case there can be no doubt of the economy of its performance, as only a waste product is used to heat the feed water and force it into the boiler. This statement applies to cases where exhaust steam is not used for heating or other useful purpose, and even then it is a superior machine, because it uses only what is actually necessary, allowing the remainder to go where it is wanted. It is not necessary to create back pressure in the exhaust pipe, to operate this injector, but the piping must be arranged so as to afford an ample supply directly to the injector. Do not arrange the piping so that exhaust steam will rush through a straight pipe in which there is a tee with a side outlet through which to supply the injector, as it will not always work well

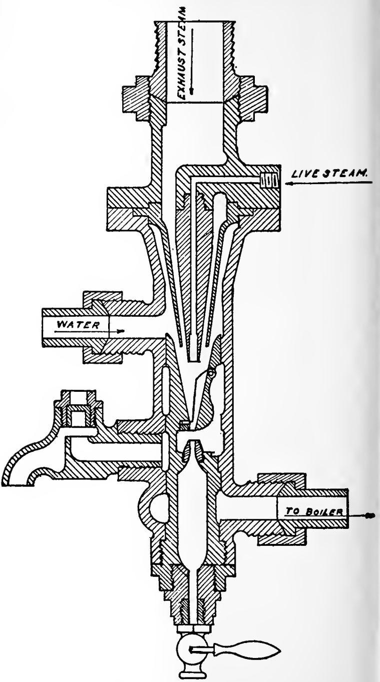


FIG.36

unless there is more or less back pressure. Let the side outlet take away what steam the injector does not use but the direct supply should come through the straight line.

PIPING INJECTORS

Fig. 37 illustrates a good plan for piping an injector, where water is taken from a street main or an elevated tank, especially if the same supply pipe is used for other purposes. If such a pipe delivers water directly to an injector under pressure it will work well if nothing is done to change the pressure, but if water is drawn off at other places along the line sudden opening and closing of the cocks of valves will quickly change the pressure, and as a general rule will cause the injector to "kick" or stop working.

If there is air confined in such a pipe it will cause trouble when it comes to the injector, because water cannot condense it and thus reduce its volume, consequently the injector stops, and if the engineer or fireman does not discover the failure until the water line in the boiler falls too low, the trouble may be serious. Steam will be blown into the water pipes under such conditions, and carry heat where it is not wanted.

The illustration shows a water main 2 located under the floor with an outlet 3 that delivers water through an angle valve 4 and a float valve 5 to a tank. This arrangement of piping and valves allows water to flow freely at the proper rate to secure an even supply. Where attempts are made to regulate the supply by hand, using a valve like 4 or any similar arrangement without the float or tank valve, the water level is sure to

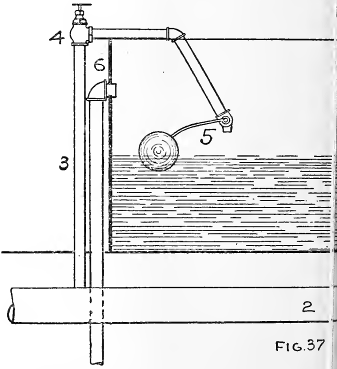
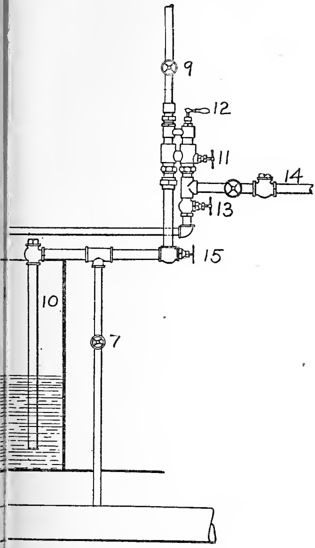


FIG. 37

be too high or too low sooner or later, and cause trouble.

When water is no longer wanted it should be shut off at 4, as the automatic valve 5 is intended to regulate the supply while the injector is in use. If



water leaks into this tank when not in use, it passes out through the overflow pipe 6.

A Hancock Inspirator is shown in this

particular case, but any other kind of an injector should be piped in the same way. If for any reason the injector is heated by steam leaking through it when not in use, cold water from the main 2 can be admitted through 7 and cool it, also the suction pipe without spilling water on the floor, as it passes through 8 into the tank. This may also be used to supply the injector when the steam pressure is too low to take water from the tank.

Under ordinary conditions, to start the machine, admit steam at 9 drawing water through the suction pipe 10 which is fitted with an angle check valve. (This does not assist the injector, but it prevents water from flowing into the tank through 10 when admitted at 7.)

After water has circulated rapidly through the pipes for a few minutes, and there is no more hot water in the steam pipe, close 11, open 12 one-quarter of a turn, and slowly close 13 with a steady movement, as none of the valves used to control an injector of any kind should be operated with an erratic motion.

This will cause hot water to flow through 14 to the boilers. If the full capacity is not required, the supply may be regulated by the angle valve 15. When not in use, valves 11 and 13 should be left open and 12 closed.

SECTION 3

STEAM ENGINES

DEFINITIONS

Various terms which are employed in the descriptions of steam engines are used in such different ways by engineers and others, that it is often difficult to decide definitely what is meant unless many words are added in explanation. As this leads to misunderstandings, the practice should be reformed and put on a uniform basis.

For illustration of this point it is only necessary to call attention to the fact that formerly when an engineer mentioned a "slide valve engine" we knew that he meant an engine that was fitted with a D slide valve which always cut off steam at one point in the stroke, and the speed was regulated by a throttling governor.

At the present time of writing he may mean that same kind of an engine, or he may mean one that is fitted with a balanced valve working between the cylinder and a pressure plate. It cuts off steam automatically according to the load on the engine, and the speed is regulated by a shaft governor.

The following definitions are suggested as being concise and expressive when properly used.

SLIDE VALVE ENGINE

An engine in which steam is admitted to the cylinder, cut-off and released by an unbalanced D slide valve, operated by direct connection to a fixed eccentric, therefore the point of cut-off is constant under all conditions. The speed is regulated by a throttling governor which wiredraws the steam and thus varies the mean effective pressure to suit changes in the load.

AUTOMATIC ENGINES

An engine in which steam is admitted to the cylinder at little less than boiler pressure under good conditions, cut-off at varying points according to the load carried, thus regulating the speed, after which it is released and passes to the exhaust pipe. Such an engine may have four valves, or it may have but one, and it may be run at a low or a high speed, but neither the number of valves nor the speed at which it operates, has any effect on the principles underlying its operation, hence do not form a basis for the name given.

LOW SPEED AUTOMATIC FOUR VALVE ENGINE

An engine designed for a low rotative speed, but not necessarily a low piston speed. Steam is admitted through two ports to the cylinder at the proper time, by two steam valves which also cut off the supply according to the load carried. This steam is exhausted through two separate ports by two exhaust valves. The Corliss is the best known member of this class, but it also includes many others. The speed is usually regulated by a fly ball governor, but there are exceptions to this general rule.

HIGH SPEED AUTOMATIC FOUR VALVE ENGINE

An engine that is operated at a high rotative speed which generally insures a comparatively high piston speed. Steam is admitted to the cylinder by two separate valves which also cut off the supply when the proper amount has been admitted. Steam is exhausted after its work is done, by two other separate valves. The speed is controlled by a shaft governor.

HIGH SPEED AUTOMATIC SINGLE VALVE ENGINE

This engine is run at a high speed, and the steam is admitted to the cylinder, cut off and released by a single valve, which may be either of the flat or of the piston type. The speed is regulated by a shaft governor.

SINGLE ACTING ENGINE

In this type steam is admitted to one end only of the cylinder, and duly exhausted from it. It may be horizontal or vertical or a combination of the two kinds. If there are two cylinders, each taking steam directly from the boiler, and operating independently, it is a simple engine, but if steam is admitted to one cylinder directly from the boiler, and from thence is exhausted into another, it is a compound engine. The number of cylinders does not affect the name, as it is determined by the way in which steam is used.

DOUBLE ACTING ENGINE

Steam is admitted to both ends of the cylinder alternately, and after doing its work it is exhausted independently. The number of cylinders or of valves used does not determine its name neither does the speed at which it is operated, nor the means adopted for controlling the speed. In the absence of a statement to the contrary, every engine is assumed to be double acting.

SINGLE ENGINE

Any kind of a steam engine that is fitted with one cylinder and one crank. Neither the way in which steam is controlled, nor the speed at which it is operated are responsible for the name given.

DOUBLE ENGINE

Any kind of a steam engine that is fitted with two cylinders and two cranks, each operating entirely independently of the other.

SIMPLE ENGINE

An engine in which steam is admitted directly from the boiler to one cylinder, after which it is exhausted to the atmosphere, or to some form of condenser.

COMPOUND ENGINE

An engine in which steam is admitted directly from the boiler to one cylinder, and after its work here is done it is exhausted to another much larger in diameter where it is still further expanded and caused to do useful work.

TRIPLE EXPANSION ENGINE

In this kind of an engine steam is admitted to one cylinder, then exhausted into another of greater diameter, after which it passes to another still larger, where it is further expanded and then exhausted to the atmosphere or to some form of condenser. In some cases this would make the third cylinder and piston very large and heavy, hence to overcome this objection two cylinders are provided, the combined area of which is equal to the area of one of suitable size, and one-half of the steam that comes from the second or intermediate cylinder, passes to each of these, and is duly expanded to a low pressure. Although such an engine has four cylinders, it is still a triple expansion engine.

NON-CONDENSING ENGINE

Any kind of an engine, or a combination of two or more of the above mentioned kinds, in which steam is exhausted into the atmosphere after performing the work for which it was generated. It is sometimes called a "high pressure engine," but this is not correct because the term has no definite meaning at the present time.

CONDENSING ENGINE

An engine in which steam is used in one or more cylinders, then exhausted into a partial vacuum, and thence into a condenser. It was formerly called a "low pressure engine," but the term is obsolete and should not be used, because many condensing engines are run under very high pressures.

SELECTING AN ENGINE

When a steam engine is to be installed in a given place, there are many points to be taken into consideration before an intelligent conclusion can be reached as to the kind and size that will give best results in that particular place.

If for any reason skilled labor to operate it is not available, either on account of unfavorable conditions in regard to location, or because the owners have decided in advance that they will not pay an engineer what a good one is worth, the simplest and cheapest engine should be purchased, for an economical machine is of necessity more or less complicated, or other conditions are its equivalent, and the best one that can be made will prove unsatisfactory if not taken care of by a competent engineer.

Where fuel is expensive, the cost of it becomes an important item in the expense account, and failure to properly manage this part of a business has caused more than one failure where competition is sharp and margins are small, and the same blunder will cause many more in the future. The problem will not be solved in any case by purchasing a complicated engine and then hiring a laborer without qualifications for this important work to take charge of it, expecting to get a competent engineer to examine the plant once or twice each year, and to correct any mismanagement that may be found. These matters need constant intelligent care and nothing else will answer the purpose.

For some places the short stroke, high speed engine is the most appropriate, but for others it is not suitable, and it is not possible to mention all points bearing on the subject in a treatise that can only grasp the general outlines of it. As the low speed engine of two or three decades ago is out of the question, a medium between the two extremes will be considered, or in other words an engine in which the diameter of the cylinder is about equal to one-half of the stroke, and the piston speed is approximately 600 feet per minute.

For the purpose of illustrating several points it is assumed that 300 indicated horse power is wanted, and that a simple non-condensing automatic engine will be used. The boiler pressure is limited to 110 pounds by the gauge, with a reduction of 5 pounds for the initial pressure in the cylinder. Under ordinary conditions the mean effective pressure of a non-condensing engine may be taken at about one-half of the boiler pressure by the gauge, or a few pounds less than this indicates, for economical results in practice.

From this explanation it will be plain that when *estimating* the size of an engine that will be required for a given case the horse power of it is a fixed quantity the piston speed is approximately determined by economical considerations, also the mean effective pressure in connection with the available boiler pressure. It should be remembered that this is only an estimate and not a close calculation. This leaves only the diameter of the cylinder to be determined, and the rule for this purpose may be stated as follows:

Multiply the horse power required by the number of foot pounds constituting one horse power. Divide the product by the piston speed in feet per minute, multiplied by the mean effective pressure. Divide the quotient by .7854 and extract the square root, which is the required diameter. It may be stated as a formula as follows:

$$\sqrt{\frac{H\text{-}P \times F\text{-}P}{S \times P}} \div .7854 = D.$$

H-P = Horse power required.
F-P = Ft. pounds for one horse power.
S = Piston speed in feet per minute.
P = mean effective pressure.
D = Diameter of the cylinder, or piston.

In this case the value of H-P is 300 while S equals 600 and P is taken at 50 for reasons previously explained, and the value of F-P is always 33,000. Substituting these values for the letters gives the following result:

$$\sqrt{\frac{300 \times 33{,}000}{600 \times 50}} \div .7854 = 20.5 \text{ inches,}$$

which in practice would be called 20 in. As the stroke is to be about twice the diameter of the cylinder, it will be fixed at 42 inches, or 7 feet for each revolution therefore, in order to secure a piston speed of 600 feet the fly wheel must revolve nearly 90 times per minute.

Placing these results in concise form, the following data appears:

Boiler pressure,	110 pounds
Initial pressure,	105 "
Mean effective pressure,	50 "
Diameter of cylinder,	20 inches
Length of stroke,	42 "
Revolutions per minute,	90

The diameter of this cylinder may be determined by a shorter method which does not include the extraction of square root. Omitting this part it appears as follows:

$$\frac{300 \times 33{,}000}{600 \times 50} = 330$$

Taking a table of the areas of circles and following down the column of areas until 330 or the nearest value to it is found, the corresponding diameter is 20½ inches, which is called 20 as this is only an estimate of the requirements.

In order to prove the value of this estimate, it becomes necessary to determine the power that this engine will develop under these conditions, and the following rule can be used for this purpose.

Multiply the area of piston by the mean effective pressure, and by the piston speed in feet per minute. Divide the product by 33,000.

As some engineers prefer such a rule stated as a formula, it is herewith given:

$$\frac{A \times P \times S}{33{,}000} = \text{I.H.P.}$$

A = Area of piston in square inches.

P = Mean effective pressure.

S = Piston speed in feet per minute, or the stroke in feed $\times 2 \times$ the number of revolutions per minute.

I. H. P. = Indicated horse power.

Applying this formula to the example given to illustrate a rule for estimating the size of an engine required for a given case, gives the following result:

$$\frac{314.16 \times 50 \times 630}{33{,}000} = 299.88$$

indicated horse power, which shows a very close approximation or estimate.

As a table of the areas of circles is very convenient for use in this connection, it is given on the following pages for this purpose.

CIRCUMFERENCES AND AREAS OF CIRCLES

Advancing by Eighths

Diam.	Circum.	Area	Diam.	Circum.	Area
1	3.1416	.7854	3 3/4	11.781	11.045
1 1/16	3.3379	.8866	3 13/16	11.977	11.416
1 1/8	3.5343	.9940	3 7/8	12.174	11.793
1 3/16	3.7306	1.1075	3 15/16	12.370	12.177
1 1/4	3.9270	1.2272	4	12.566	12.566
1 5/16	4.1233	1.3530	4 1/16	12.763	12.962
1 3/8	4.3197	1.4849	4 1/8	12.959	13.364
1 7/16	4.5160	1.6230	4 3/16	13.155	13.772
1 1/2	4.7124	1.7671	4 1/4	13.352	14.186
1 9/16	4.9087	1.9175	4 5/16	13.548	14.607
1 5/8	5.1051	2.0739	4 3/8	13.744	15.033
1 11/16	5.3014	2.2365	4 7/16	13.941	15.466
1 3/4	5.4978	2.4053	4 1/2	14.137	15.904
1 13/16	5.6941	2.5802	4 9/16	14.334	16.349
1 7/8	5.8905	2.7612	4 5/8	14.530	16.800
1 15/16	6.0868	2.9483	4 11/16	14.726	17.257
2	6.2832	3.1416	4 3/4	14.923	17.721
2 1/16	6.4795	3.3410	4 13/16	15.119	18.190
2 1/8	6.6759	3.5466	4 7/8	15.315	18.665
2 3/16	6.8722	3.7583	4 15/16	15.512	19.147
2 1/4	7.0686	3.9761	5	15.708	19.635
2 5/16	7.2649	4.2000	5 1/16	15.904	20.129
2 3/8	7.4613	4.4301	5 1/8	16.101	20.629
2 7/16	7.6576	4.6664	5 3/16	16.297	21.135
2 1/2	7.8540	4.9087	5 1/4	16.493	21.648
2 9/16	8.0503	5.1572	5 5/16	16.690	22.166
2 5/8	8.2467	5.4119	5 3/8	16.886	22.691
2 11/16	8.4430	5.6727	5 7/16	17.082	23.221
2 3/4	8.6394	5.9396	5 1/2	17.279	23.758
2 13/16	8.8357	6.2126	5 9/16	17.475	24.301
2 7/8	9.0321	6.4918	5 5/8	17.671	24.850
2 15/16	9.2284	6.7771	5 11/16	17.868	25.406
3	9.4248	7.0686	5 3/4	18.064	25.967
3 1/16	9.6211	7.3662	5 13/16	18.261	26.535
3 1/8	9.8175	7.6699	5 7/8	18.457	27.109
3 3/16	10.014	7.9798	5 15/16	18.653	27.688
3 1/4	10.210	8.2958	6	18.850	28.274
3 5/16	10.407	8.6179	6 1/8	19.242	29.465
3 3/8	10.603	8.9462	6 1/4	19.635	30.680
3 7/16	10.799	9.2806	6 3/8	20.028	31.919
3 1/2	10.996	9.6211	6 1/2	20.420	33.183
3 9/16	11.192	9.9678	6 5/8	20.813	34.472
3 5/8	11.388	10.321	6 3/4	21.206	35.785
3 11/16	11.585	10.680	6 7/8	21.598	37.122

Circumferences and Areas of Circles—*Continued*

Diam.	Circum.	Area	Diam.	Circum.	Area
7	21.991	38.485	12⅞	40.448	130.19
7⅛	22.384	39.871	13	40.841	132.73
7¼	22.776	41.282	13⅛	41.233	135.30
7⅜	23.169	42.718	13¼	41.626	137.89
7½	23.562	44.179	13⅜	42.019	140.50
7⅝	23.955	45.664	13½	42.412	143.14
7¾	24.347	47.173	13⅝	42.804	145.80
7⅞	24.740	48.707	13¾	43.197	148.49
8	25.133	50.265	13⅞	43.590	151.20
8⅛	25.525	51.849	14	43.982	153.94
8¼	25.918	53.456	14⅛	44.375	156.70
8⅜	26.311	55.088	14¼	44.768	159.48
8½	26.704	56.745	14⅜	45.160	162.30
8⅝	27.096	58.426	14½	45.553	165.13
8¾	27.489	60.132	14⅝	45.946	167.99
8⅞	27.882	61.862	14¾	46.338	170.87
9	28.274	63.617	14⅞	46.731	173.78
9⅛	28.667	65.397	15	47.124	176.71
9¼	29.060	67.201	15⅛	47.517	179.67
9⅜	29.452	69.029	15¼	47.909	182.65
9½	29.845	70.882	15⅜	48.302	185.66
9⅝	30.238	72.760	15½	48.695	188.69
9¾	30.631	74.662	15⅝	49.087	191.75
9⅞	31.023	76.589	15¾	49.480	194.83
10	31.416	78.540	15⅞	49.873	197.93
10⅛	31.809	80.516	16	50.265	201.06
10¼	32.201	82.516	16⅛	50.658	204.22
10⅜	32.594	84.541	16¼	51.051	207.39
10½	32.987	86.590	16⅜	51.444	210.60
10⅝	33.379	88.664	16½	51.836	213.82
10¾	33.772	90.763	16⅝	52.229	217.08
10⅞	34.165	92.886	16¾	52.622	220.35
11	34.558	95.033	16⅞	53.014	223.65
11⅛	34.950	97.205	17	53.407	226.98
11¼	35.343	99.402	17⅛	53.800	230.33
11⅜	35.736	101.62	17¼	54.192	233.71
11½	36.128	103.87	17⅜	54.585	237.10
11⅝	36.521	106.14	17½	54.978	240.53
11¾	36.914	108.43	17⅝	55.371	243.98
11⅞	37.306	110.75	17¾	55.763	247.45
12	37.699	113.10	17⅞	56.156	250.95
12⅛	38.092	115.47	18	56.549	254.47
12¼	38.485	117.86	18⅛	56.941	258.02
12⅜	38.877	120.28	18¼	57.334	261.59
12½	39.270	122.72	18⅜	57.727	265.18
12⅝	39.663	125.19	18½	58.119	268.80
12¾	40.055	127.68	18⅝	58.512	272.45

Circumferences and Areas of Circles—*Continued*

Diam.	Circum.	Area	Diam.	Circum.	Area
18¾	58.905	276.12	24⅝	77.362	476.26
18⅞	59.298	279.81	24¾	77.754	481.11
19	59.690	283.53	24⅞	78.147	485.98
19⅛	60.083	287.27	25	78.540	490.87
19¼	60.476	291.04	25⅛	78.933	495.79
19⅜	60.868	294.83	25¼	79.325	500.74
19½	61.261	298.65	25⅜	79.718	505.71
19⅝	61.654	302.49	25½	80.111	510.71
19¾	62.046	306.35	25⅝	80.503	515.72
19⅞	62.439	310.24	25¾	80.896	520.77
20	62.832	314.16	25⅞	81.289	525.84
20⅛	63.225	318.10	26	81.681	530.93
20¼	63.617	322.06	26⅛	82.074	536.05
20⅜	64.010	326.05	26¼	82.467	541.19
20½	64.403	330.06	26⅜	82.860	546.35
20⅝	64.795	334.10	26½	83.252	551.55
20¾	65.188	338.16	26⅝	83.645	556.76
20⅞	65.581	342.25	26¾	84.038	562.00
21	65.973	346.36	26⅞	84.430	567.27
21⅛	66.366	350.50	27	84.823	572.56
21¼	66.759	354.66	27⅛	85.216	577.87
21⅜	67.152	358.84	27¼	85.608	583.21
21½	67.544	363.05	27⅜	86.001	588.57
21⅝	67.937	367.28	27½	86.394	593.96
21¾	68.330	371.54	27⅝	86.786	599.37
21⅞	68.722	375.83	27¾	87.179	604.81
22	69.115	380.13	27⅞	87.572	610.27
22⅛	69.508	384.46	28	87.965	615.75
22¼	69.900	388.82	28⅛	88.357	621.26
22⅜	70.293	393.20	28¼	88.750	626.80
22½	70.686	397.61	28⅜	89.143	632.36
22⅝	71.079	402.04	28½	89.535	637.94
22¾	71.471	406.49	28⅝	89.928	643.55
22⅞	71.864	410.97	28¾	90.321	649.18
23	72.257	415.48	28⅞	90.713	654.84
23⅛	72.649	420.00	29	91.106	660.52
23¼	73.042	424.56	29⅛	91.499	666.23
23⅜	73.435	429.13	29¼	91.892	671.96
23½	73.827	433.74	29⅜	92.284	677.71
23⅝	74.220	438.36	29½	92.677	683.49
23¾	74.613	443.01	29⅝	93.070	689.30
23⅞	75.006	447.69	29¾	93.462	695.13
24	75.398	452.39	29⅞	93.855	700.98
24⅛	75.791	457.11	30	94.248	706.86
24¼	76.184	461.86	30⅛	94.640	712.76
24⅜	76.576	466.64	30¼	95.033	718.69
24½	76.969	471.44	30⅜	95.426	724.64

Circumferences and Areas of Circles—*Continued*

Diam.	Circum.	Area	Diam.	Circum.	Area
30½	95.819	730.62	36⅜	114.275	1039.2
30⅝	96.211	736.62	36½	114.668	1046.3
30¾	96.604	742.64	36⅝	115.061	1053.5
30⅞	96.997	748.69	36¾	115.454	1060.7
31	97.389	754.77	36⅞	115.846	1068.0
31⅛	97.782	760.87	37	116.239	1075.2
31¼	98.175	766.99	37⅛	116.632	1082.5
31⅜	98.567	773.14	37¼	117.024	1089.8
31½	98.960	779.31	37⅜	117.417	1097.1
31⅝	99.353	785.51	37½	117.810	1104.5
31¾	99.746	791.73	37⅝	118.202	1111.8
31⅞	100.138	797.98	37¾	118.596	1119.2
32	100.531	804.25	37⅞	118.988	1126.7
32⅛	100.924	810.54	38	119.381	1134.1
32¼	101.316	816.86	38⅛	119.773	1141.0
32⅜	101.709	823.21	38¼	120.166	1149.1
32½	102.102	829.58	38⅜	120.559	1156.6
32⅝	102.494	835.97	38½	120.951	1164.2
32¾	102.887	842.39	38⅝	121.344	1171.7
32⅞	103.280	848.83	38¾	121.737	1179.3
33	103.673	855.30	38⅞	122.129	1186.9
33⅛	104.065	861.79	39	122.522	1194.6
33¼	104.458	868.31	39⅛	122.915	1202.3
33⅜	104.851	874.85	39¼	123.308	1210.0
33½	105.243	881.41	39⅜	123.700	1217.7
33⅝	105.636	888.00	39½	124.093	1225.4
33¾	106.029	894.62	39⅝	124.486	1233.2
33⅞	106.421	901.26	39¾	124.878	1241.0
34	106.814	907.92	39⅞	125.271	1248.8
34⅛	107.207	914.61	40	125.664	1256.6
34¼	107.600	921.32	40⅛	126.056	1264.5
34⅜	107.992	928.06	40¼	126.449	1272.4
34½	108.385	934.82	40⅜	126.842	1280.3
34⅝	108.778	941.61	40½	127.235	1288.2
34¾	109.170	948.42	40⅝	127.627	1296.2
34⅞	109.563	955.25	40¾	128.020	1304.2
35	106.956	962.11	40⅞	128.413	1312.2
35⅛	110.348	969.00	41	128.805	1320.3
35¼	110.741	975.91	41⅛	129.198	1328.3
35⅜	111.134	982.84	41¼	129.591	1336.4
35½	111.527	989.80	41⅜	129.983	1344.5
35⅝	111.919	996.78	41½	130.376	1352.7
35¾	112.312	1003.8	41⅝	130.769	1360.8
35⅞	112.705	1010.8	41¾	131.161	1369.0
36	113.097	1017.9	41⅞	131.554	1377.2
36⅛	113.490	1025.0	42	131.947	1385.4
36¼	113.883	1032.1	42⅛	132.340	1393.7

Circumferences and Areas of Circles—*Continued*

Diam.	Circum.	Area	Diam.	Circum.	Area
42 1/4	132.732	1402.0	48 1/8	151.189	1819.0
42 3/8	133.125	1410.3	48 1/4	151.582	1828.5
42 1/2	133.518	1418.6	48 3/8	151.975	1837.9
42 5/8	133.910	1427.0	48 1/2	152.367	1847.5
42 3/4	134.303	1435.4	48 5/8	152.760	1857.0
42 7/8	134.696	1443.8	48 3/4	153.153	1866.5
43	135.088	1452.2	48 7/8	153.545	1876.1
43 1/8	135.481	1460.7	49	153.938	1885.7
43 1/4	135.874	1469.1	49 1/8	154.331	1895.4
43 3/8	136.267	1477.6	49 1/4	154.723	1905.0
43 1/2	136.659	1486.2	49 3/8	155.116	1914.7
43 5/8	137.052	1494.7	49 1/2	155.509	1924.4
43 3/4	137.445	1503.3	49 5/8	155.902	1934.2
43 7/8	137.837	1511.9	49 3/4	156.294	1943.9
44	138.230	1520.5	49 7/8	156.687	1953.7
44 1/8	138.623	1529.2	50	157.080	1963.5
44 1/4	139.015	1537.9	50 1/8	157.472	1973.3
44 3/8	139.408	1546.6	50 1/4	157.865	1983.2
44 1/2	139.801	1555.3	50 3/8	158.258	1993.1
44 5/8	140.194	1564.0	50 1/2	158.650	2003.0
44 3/4	140.586	1572.8	50 5/8	159.043	2012.9
44 7/8	140.979	1581.6	50 3/4	159.436	2022.8
45	141.372	1590.4	50 7/8	159.829	2032.8
45 1/8	141.764	1599.3	51	160.221	2042.8
45 1/4	142.157	1608.2	51 1/8	160.614	2052.8
45 3/8	142.550	1617.0	51 1/4	161.007	2062.9
45 1/2	142.942	1626.0	51 3/8	161.399	2073.0
45 5/8	143.335	1634.9	51 1/2	161.792	2083.1
45 3/4	143.728	1643.9	51 5/8	162.185	2093.2
45 7/8	144.121	1652.9	51 3/4	162.577	2103.3
46	144.513	1661.9	51 7/8	162.970	2113.5
46 1/8	144.906	1670.9	52	163.363	2123.7
46 1/4	145.299	1680.0	52 1/8	163.756	2133.9
46 3/8	145.691	1689.1	52 1/4	164.148	2144.2
46 1/2	146.084	1698.2	52 3/8	164.541	2154.5
46 5/8	146.477	1707.4	52 1/2	164.934	2164.8
46 3/4	146.869	1716.5	52 5/8	165.326	2175.1
46 7/8	147.262	1725.7	52 3/4	165.719	2185.4
47	147.655	1734.9	52 7/8	166.112	2195.8
47 1/8	148.048	1744.2	53	166.504	2206.2
47 1/4	148.440	1753.5	53 1/8	166.897	2216.6
47 3/8	148.833	1762.7	53 1/4	167.290	2227.0
47 1/2	149.226	1772.1	53 3/8	167.683	2237.5
47 5/8	149.618	1781.4	53 1/2	168.075	2248.0
47 3/4	150.011	1790.8	53 5/8	168.468	2258.5
47 7/8	150.404	1800.1	53 3/4	168.861	2269.1
48	150.796	1809.6	53 7/8	169.253	2279.6

Circumferences and Areas of Circles—*Continued*

Diam.	Circum.	Area	Diam.	Circum.	Area
54	169.646	2290.2	59¾	187.710	2803.9
54⅛	170.039	2300.8	59⅞	188.103	2815.7
54¼	170.431	2311.5	60	188.496	2827.4
54⅜	170.824	2322.1	60⅛	188.888	2839.2
54½	171.217	2332.8	60¼	189.281	2851.0
54⅝	171.609	2343.5	60⅜	189.674	2862.9
54¾	172.002	2354.3	60½	190.066	2874.8
54⅞	172.395	2365.0	60⅝	190.459	2886.6
55	172.788	2375.8	60¾	190.852	2898.6
55⅛	173.180	2386.6	60⅞	191.244	2910.5
55¼	173.573	2397.5	61	191.637	2922.5
55⅜	173.966	2408.3	61⅛	192.030	2934.5
55½	174.358	2419.2	61¼	192.423	2946.5
55⅝	174.751	2430.1	61⅜	192.815	2958.5
55¾	175.144	2441.1	61½	193.208	2970.6
55⅞	175.536	2452.0	61⅝	193.601	2982.7
56	175.929	2463.0	61¾	193.993	2994.8
56⅛	176.322	2474.0	61⅞	194.386	3006.9
56¼	176.715	2485.0	62	194.779	3019.1
56⅜	177.107	2496.1	62⅛	195.171	3031.3
56½	177.500	2507.2	62¼	195.564	3043.5
56⅝	177.893	2518.3	62⅜	195.957	3055.7
56¾	178.285	2529.4	62½	196.350	3068.0
56⅞	178.678	2540.6	62⅝	196.742	3080.3
57	179.071	2551.8	62¾	197.135	3092.6
57⅛	179.463	2563.0	62⅞	197.528	3104.9
57¼	179.856	2574.2	63	197.920	3117.2
57⅜	180.249	2585.4	63⅛	198.313	3129.6
57½	180.642	2596.7	63¼	198.706	3142.0
57⅝	181.034	2608.0	63⅜	199.098	3154.5
57¾	181.427	2619.4	63½	199.491	3166.9
57⅞	181.820	2630.7	63⅝	199.884	3179.4
58	182.212	2642.1	63¾	200.277	3191.9
58⅛	182.605	2653.5	63⅞	200.669	3204.4
58¼	182.998	2664.9	64	201.062	3217.0
58⅜	183.390	2676.4	64⅛	201.455	3229.6
58½	183.783	2687.8	64¼	201.847	3242.2
58⅝	184.176	2699.3	64⅜	202.240	3254.8
58¾	184.569	2710.9	64½	202.633	3267.5
58⅞	184.961	2722.4	64⅝	203.025	3280.1
59	185.354	2734.0	64¾	203.418	3292.8
59⅛	185.747	2745.6	64⅞	203.811	3305.6
59¼	186.139	2757.2	65	204.204	3318.3
59⅜	186.532	2768.8	65⅛	204.596	3331.1
59½	186.925	2780.5	65¼	204.989	3343.9
59⅝	187.317	2792.2	65⅜	205.382	3356.7

Circumferences and Areas of Circles—*Continued*

Diam.	Circum.	Area	Diam.	Circum.	Area
65 1/2	205.774	3369.6	71 3/8	224.231	4001.1
65 5/8	206.167	3382.4	71 1/2	224.624	4015.2
65 3/4	206.560	3395.3	71 5/8	225.017	4029.2
65 7/8	206.952	3408.2	71 3/4	225.409	4043.3
66	207.345	3421.2	71 7/8	225.802	4057.4
66 1/8	207.738	3434.2	72	226.195	4071.5
66 1/4	208.131	3447.2	72 1/8	226.587	4085.7
66 3/8	208.523	3460.2	72 1/4	226.980	4099.8
66 1/2	208.916	3473.2	72 3/8	227.373	4114.0
66 5/8	209.309	3486.3	72 1/2	227.765	4128.2
66 3/4	209.701	3499.4	72 5/8	228.158	4142.5
66 7/8	210.094	3512.5	72 3/4	228.551	4156.8
67	210.487	3525.7	72 7/8	228.944	4171.1
67 1/8	210.879	3538.8	73	229.336	4185.4
67 1/4	211.272	3552.0	73 1/8	229.729	4199.7
67 3/8	211.665	3565.2	73 1/4	230.122	4214.1
67 1/2	212.058	3578.5	73 3/8	230.514	4228.5
67 5/8	212.450	3591.7	73 1/2	230.907	4242.9
67 3/4	212.843	3605.0	73 5/8	231.300	4257.4
67 7/8	213.236	3618.3	73 3/4	231.692	4271.8
68	213.628	3631.7	73 7/8	232.085	4286.3
68 1/8	214.021	3645.0	74	232.478	4300.8
68 1/4	214.414	3658.4	74 1/8	232.871	4315.4
68 3/8	214.806	3671.8	74 1/4	233.263	4329.9
68 1/2	215.199	3685.3	74 3/8	233.656	4344.5
68 5/8	215.592	3698.7	74 1/2	234.049	4359.2
68 3/4	215.984	3712.2	74 5/8	234.441	4373.8
68 7/8	216.377	3725.7	74 3/4	234.834	4388.5
69	216.770	3739.3	74 7/8	235.227	4403.1
69 1/8	217.163	3752.8	75	235.619	4417.9
69 1/4	217.555	3766.4	75 1/8	236.012	4432.6
69 3/8	217.948	3780.0	75 1/4	236.405	4447.4
69 1/2	218.341	3793.7	75 3/8	236.798	4462.2
69 5/8	218.733	3807.3	75 1/2	237.190	4477.0
69 3/4	219.126	3821.0	75 5/8	237.583	4491.8
69 7/8	219.519	3834.7	75 3/4	237.976	4506.7
70	219.911	3848.5	75 7/8	238.368	4521.5
70 1/8	220.304	3862.2	76	238.761	4536.5
70 1/4	220.697	3876.0	76 1/8	239.154	4551.4
70 3/8	221.090	3889.8	76 1/4	239.546	4566.4
70 1/2	221.482	3903.6	76 3/8	239.939	4581.3
70 5/8	221.875	3917.5	76 1/2	240.332	4596.3
70 3/4	222.268	3931.4	76 5/8	240.725	4611.4
70 7/8	222.660	3945.3	76 3/4	241.117	4626.4
71	223.053	3959.2	76 7/8	241.510	4641.5
71 1/8	223.446	3973.1	77	241.903	4656.6
71 1/4	223.838	3987.1	77 1/8	242.295	4671.8

Circumferences and Areas of Circles—*Continued*

Diam.	Circum.	Area	Diam.	Circum.	Area
77¼	242.688	4686.9	83⅛	261.145	5426.9
77⅜	243.081	4702.1	83¼	261.538	5443.3
77½	243.473	4717.3	83⅜	261.930	5459.6
77⅝	243.866	4732.5	83½	262.323	5476.0
77¾	244.259	4747.8	83⅝	262.716	5492.4
77⅞	244.652	4763.1	83¾	263.108	5508.8
78	245.044	4778.4	83⅞	263.501	5525.3
78⅛	245.437	4793.7	84	263.894	5541.8
78¼	245.830	4809.0	84⅛	264.286	5558.3
78⅜	246.222	4824.4	84¼	264.679	5574.8
78½	246.615	4839.8	84⅜	265.072	5591.4
78⅝	247.008	4855.2	84½	265.465	5607.9
78¾	247.400	4870.7	84⅝	265.857	5624.5
78⅞	247.793	4886.2	84¾	266.250	5641.2
79	248.186	4901.7	84⅞	266.643	5657.8
79⅛	248.579	4917.2	85	267.035	5674.5
79¼	248.971	4932.7	85⅛	267.428	5691.2
79⅜	249.364	4948.3	85¼	267.821	5707.9
79½	249.757	4963.9	85⅜	268.213	5724.7
79⅝	250.149	4979.5	85½	268.606	5741.5
79¾	250.542	4995.2	85⅝	268.999	5758.3
79⅞	250.935	5010.9	85¾	269.392	5775.1
80	251.327	5026.5	85⅞	269.784	5791.9
80⅛	251.720	5042.3	86	270.177	5808.8
80¼	252.113	5058.0	86⅛	270.570	5825.7
80⅜	252.506	5073.8	86¼	270.962	5842.6
80½	252.898	5089.6	86⅜	271.355	5859.6
80⅝	253.291	5105.4	86½	271.748	5876.5
80¾	253.684	5121.2	86⅝	272.140	5893.5
80⅞	254.076	5137.1	86¾	272.533	5910.6
81	254.469	5153.0	86⅞	272.926	5927.6
81⅛	254.862	5168.9	87	273.319	5944.7
81¼	255.254	5184.9	87⅛	273.711	5961.8
81⅜	255.647	5200.8	87¼	274.104	5978.9
81½	256.040	5216.8	87⅜	274.497	5996.0
81⅝	256.433	5232.8	87½	274.889	6013.2
81¾	256.825	5248.9	87⅝	275.282	6030.4
81⅞	257.218	5264.9	87¾	275.675	6047.6
82	257.611	5281.0	87⅞	276.067	6064.9
82⅛	258.003	5297.1	88	276.460	6082.1
82¼	258.396	5313.3	88⅛	276.853	6099.4
82⅜	258.789	5329.4	88¼	277.246	6116.7
82½	259.181	5345.6	88⅜	277.638	6134.1
82⅝	259.574	5361.8	88½	278.031	6151.4
82¾	259.967	5378.1	88⅝	278.424	6168.8
82⅞	260.359	5394.3	88¾	278.618	6186.2
83	260.752	5410.6	88⅞	279.209	6203.7

Circumferences and Areas of Circles—*Continued*

Diam.	Circum.	Area	Diam.	Circum.	Area
89	279.602	6221.1	94 7/8	298.059	7069.6
89 1/8	279.994	6238.6	95	298.451	7088.2
89 1/4	280.387	6256.1	95 1/8	298.844	7106.9
89 3/8	280.780	6273.7	95 1/4	299.237	7125.6
89 1/2	281.173	6291.2	95 3/8	299.629	7144.3
89 5/8	281.565	6308.8	95 1/2	300.022	7163.0
89 3/4	281.958	6326.4	95 5/8	300.415	7181.8
89 7/8	282.351	6344.1	95 3/4	300.807	7200.6
90	282.743	6361.7	95 7/8	301.200	7219.4
90 1/8	283.136	6379.4	96	301.593	7238.2
90 1/4	283.529	6397.1	96 1/8	301.986	7257.1
90 3/8	283.921	6414.9	96 1/4	302.378	7276.0
90 1/2	284.314	6432.6	96 3/8	302.771	7294.9
90 5/8	284.707	6450.4	96 1/2	303.164	7313.8
90 3/4	285.100	6468.2	96 5/8	303.556	7332.8
90 7/8	285.492	6486.0	96 3/4	303.949	7351.8
91	285.885	6503.9	96 7/8	304.342	7370.8
91 1/8	286.278	6521.8	97	304.734	7389.8
91 1/4	286.670	6539.7	97 1/8	305.127	7408.9
91 3/8	287.063	6557.6	97 1/4	305.520	7428.0
91 1/2	287.456	6575.5	97 3/8	305.913	7447.1
91 5/8	287.848	6593.5	97 1/2	306.305	7466.2
91 3/4	288.241	6611.5	97 5/8	306.698	7485.3
91 7/8	288.634	6629.6	97 3/4	307.091	7504.5
92	289.027	6647.6	97 7/8	307.483	7523.7
92 1/8	289.419	6665.7	98	307.876	7543.0
92 1/4	289.812	6683.8	98 1/8	308.269	7562.2
92 3/8	290.205	6701.9	98 1/4	308.661	7581.5
92 1/2	290.597	6720.1	98 3/8	309.054	7600.8
92 5/8	290.990	6738.2	98 1/2	309.447	7620.1
92 3/4	291.383	6756.4	98 5/8	309.840	7639.5
92 7/8	291.775	6774.7	98 3/4	310.232	7658.9
93	292.168	6792.9	98 7/8	310.625	7678.3
93 1/8	292.561	6811.2	99	311.018	7697.7
93 1/4	292.954	6829.5	99 1/8	311.410	7717.1
93 3/8	293.346	6847.8	99 1/4	311.803	7736.6
93 1/2	293.739	6866.1	99 3/8	312.196	7756.1
93 5/8	294.132	6884.5	99 1/2	312.588	7775.6
93 3/4	294.524	6902.9	99 5/8	312.981	7795.2
93 7/8	294.917	6921.3	99 3/4	313.374	7814.8
94	295.310	6939.8	99 7/8	313.767	7834.4
94 1/8	295.702	6958.2	100	314.159	7854.0
94 1/4	296.095	6976.7			
94 3/8	296.488	6995.3			
94 1/2	296.881	7013.8			
94 5/8	297.273	7032.4			
94 3/4	297.666	7051.0			

DETERMINING THE MEAN EFFECTIVE PRESSURE

There are many cases in which it becomes necessary to determine the mean effective pressure that will result from given conditions in advance of the installation of the engine, or in other words before an indicator diagram can be secured. As this is an important matter it will be explained in detail.

The terms "average pressure" and "mean effective pressure" are used indiscriminately, not only by engineers in daily practice, but in mechanical books and papers by men who in some cases do not understand the difference, and in others are too careless to properly separate them. The effect in either case on the student is precisely the same, as it gives him a wrong idea of the whole subject.

The average pressure acting on the piston of a steam engine is found by taking into consideration the initial pressure, or pressure at the beginning of the stroke in connection with the pressure realized at short succeeding intervals until the end is reached. When the average of them is found it constitutes the average pressure for those conditions, and this must necessarily be the pressure above a perfect vacuum in all cases.

Special attention is called to the fact that the back pressure is not considered, as it can have no effect on the average

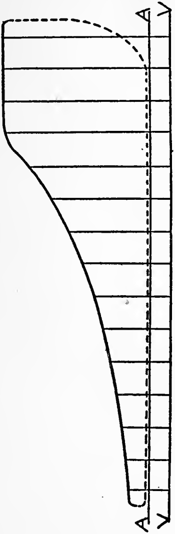


FIG. 38

pressure, because they act on opposite sides of the piston.

Fig. 38 is a theoretical indicator diagram in which the full lines only are used to determine the average pressure. The dotted lines are inserted to make the diagram complete, but they have no other use here. This diagram was originally laid out accurately by a No. 40 scale, showing an initial pressure of 120 pounds absolute with the point of cut-off at one-quarter stroke. While it is necessary to reduce the size of this diagram for use here, the correct proportions are preserved, but it is not practicable to measure it with a No. 40 scale now. When the ordinates are laid out as shown their total length in inches ascertained and divided by the number used, the quotient is the average height of the diagram. Multiply this by the scale adopted (which is No. 40 in this case), and the product is the average pressure acting on the engine piston. These ordinates are the exact length to be measured in order to make the matter as plain as possible. The atmospheric line is shown at AA and the perfect vacuum line at VV.

While a diagram clearly illustrates the principle involved, the same result can be secured with less trouble by calculation, using the following rule:

Ascertain the ratio of expansion and find the corresponding hyperbolic logarithm in a table prepared for this purpose. Add 1, multiply the sum by the initial pressure absolute, and divide by the ratio of expansion. The quotient is the average pressure for stated conditions.

When given as a formula it appears as follows:

$$\frac{(\text{Hyp Log}+1)\times \text{P}}{\text{R}}=\text{A}.$$

Hyp Log = Hyperbolic logarithm of the ratio of expansion.

P = Initial pressure absolute.

R = Ratio of expansion.

A = Average pressure.

For illustration and explanation take a plant carrying 110 pounds pressure by the gauge, or 125 pounds absolute, with an initial pressure of 120 pounds in the cylinder, cutting off steam at one-quarter or .25 of the stroke.

Under these conditions the ratio of expansion is 4 the hyperbolic logarithm of which is 1.3863 and when due application is made of the formula it gives the following result:

$$\frac{(1.3863+1)\times 120}{4}=71.58$$

pounds average pressure.

Fig. 39 illustrates the proper method for finding the mean effective pressure from an indicator diagram. It is a reproduction of Fig. 38, except that there are no dotted lines, and the ordinates are shortened to show the exact measurements required When the total length of these ordinates in inches is found and divided by the number

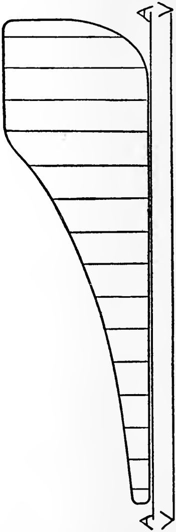


FIG. 39

used, the quotient is the mean effective height of the diagram, and when this is multiplied by the scale used, the product is the mean effective pressure.

As these ordinates are shorter than

before, the effect is to cut off or subtract the back pressure. No stated number of ordinates is required, as the desired result is found by dividing the total length by the number adopted.

Examination of the two preceding diagrams illustrates the fact that when the rule for finding the average pressure on the piston of an engine is stated, it is only necessary to add another clause and it becomes a rule for calculating the mean effective pressure as follows:

Ascertain the ratio of expansion and find the corresponding hyperbolic logarithm in a table prepared for this purpose. Add 1, multiply the sum by the initial pressure absolute, divide by the ratio of expansion, and subtract the back pressure. The remainder is the mean effective pressure for stated conditions.

When given as a formula it appears as follows:

$$\frac{(\text{Hyp Log}+1)\times P}{R}-B=\text{M E P.}$$

Hyp Log = Hyperbolic logarithm of the ratio of expansion.
P = Initial pressure absolute.
R = Ratio of expansion.
B = Back pressure above a vacuum.
M E P = Mean effective pressure for given conditions,

Applying this formula to a case where the initial pressure is 120 pounds absolute, the ratio of expansion 4 and the back pressure 16 pounds gives the following result:

$$\frac{(1.3863+1)\times 120}{4}-16=55.58$$

pounds mean effective pressure.

When calculating the size of an engine that would be required to develop 300 horse power, the mean effective pressure was assumed to be 50 pounds. As the above result is more than the required pressure, the point of cut off would be a trifle shorter than at one-quarter stroke, which is satisfactory, as it will give a lower terminal pressure.

THE RATIO OF EXPANSION AND THE BACK PRESSURE

When the cut-off valve on an automatic, or any other kind of an engine, closes, the piston has travelled a certain part of the stroke, and the relation that this part bears to the whole stroke is the ratio of expansion. For illustration, suppose that the cut-off valve closes when the piston has travelled one-quarter or .25 of the stroke. Taking the whole stroke as unity or 1 and dividing it by .25 gives the ratio of expansion. Then $1 \div .25 = 4$.

For estimating the mean effective pressure the clearance is not taken into account, as it makes the calculation more complicated for what is not intended to be an accurate computation. If greate accuracy is required, the clearance should be stated as the percentage of the whole stroke. It is then added to unity or 1 which represents the whole stroke, also to the fraction which represents the point of cut off and the former is divided by the latter as before.

Suppose that the clearance is 5 per cent. or .05 of the whole stroke, and the cut off takes place at one-quarter stroke. The ratio of expansion is then:

$$\frac{1+.05}{.25+.05}=3.5$$

The next table gives the actual ratios of expansion taking clearance from 1 to 10 per cent. into consideration. The first column gives the point of cut off as a fraction of the whole stroke. The second gives the ratio of expansion without clearance, and succeeding columns give the ratio with the amount of clearance given for each as a per cent. of the whole stroke.

ACTUAL RATIOS OF EXPANSION.

Per cent. of Clearance.

P of C	0	1	2	3	4
.01	100.00	50.5	34.0	25.75	20.8
.02	50.00	33.67	25.50	20.60	17.33
.03	33.33	25.25	20.40	17.16	14.86
.04	25.00	20.20	17.00	14.71	13.00
.05	20.00	16.83	14.57	12.87	11.55
.06	16.67	14.43	12.75	11.44	10.40
.07	14.28	12.62	11.33	10.30	9.46
.08	12.50	11.22	10.2	9.36	8.67
.09	11.11	10.10	9.27	8.58	8.00
.10	10.00	9.18	8.50	7.92	7.43
.11	9.09	8.42	7.84	7.36	6.93
.12	8.33	7.78	7.25	6.86	6.50
.14	7.14	6.73	6.37	6.06	5.78
.16	6.25	5.94	5.67	5.42	5.20
.20	5.00	4.81	4.64	4.48	4.33
.25	4.00	3.88	3.77	3.68	3.58
.30	3.33	3.26	3.19	3.12	3.06
.40	2.50	2.46	2.43	2.40	2.36
.50	2.00	1.98	1.96	1.94	1.92
.60	1.67	1.66	1.65	1.64	1.63
.70	1.43	1.42	1.42	1.41	1.41
.80	1.25	1.25	1.244	1.241	1.238
.90	1.111	1.11	1.109	1.108	1.106
1.00	1.00	1.00	1.000	1.000	1.000

ACTUAL RATIOS OF EXPANSION.

Per cent. of Clearance.

P of C	5	6	7	8	9	10
.01	17.5	15.14	13.38	12.00	10.9	10.
.02	15.00	13.25	11.89	10.80	9.91	8.17
.03	13.12	11.78	10.70	9.82	9.08	8.46
.04	11.66	10.60	9.73	9.00	8.39	7.86
.05	10.50	9.64	8.92	8.31	7.79	7.33
.06	9.55	8.33	8.23	7.71	7.27	6.88
.07	8.75	8.15	7.64	7.20	6.81	6.47
.08	8.08	7.57	7.13	6.75	6.41	6.11
.09	7.50	7.07	6.69	6.35	6.06	5.79
.10	7.00	6.62	6.30	6.00	5.74	5.50
.11	6.56	6.24	5.94	5.68	5.45	5.24

Actual Ratios of Expansion—*Continued*

Per cent. of Clearance

P of C	5	6	7	8	9	10
.12	6.18	5.89	5.63	5.40	5.19	5.00
.14	5.53	5.30	5.10	4.91	4.74	4.58
.16	5.00	4.82	4.65	4.50	4.36	4.23
.20	4.20	4.08	3.96	3.86	3.76	3.61
.25	3.50	3.42	3.34	3.27	3.21	3.14
.30	3.00	2.94	2.90	2.84	2.80	2.75
.40	2.33	2.30	2.28	2.25	2.22	2.20
.50	1.90	1.89	1.88	1.86	1.85	1.83
.60	1.615	1.606	1.597	1.588	1.580	1.571
.70	1.400	1.395	1.390	1.385	1.380	1.375
.80	1.235	1.233	1.230	1.227	1.224	1.222
.90	1.105	1.104	1.103	1.102	1.101	1.100
1.00	1.000	1.000	1.000	1.000	1.000	1.000

Having determined the assumed or the actual ratio of expansion, it becomes necessary to find the corresponding hyperbolic logarithm in a table, and add one to it in each case. The reason for this is because the work done by the steam from the beginning of the stroke to the point of cut off is represented by unity or 1 and the work performed after the cut off has taken place, or during expansion of the steam, is shown comparatively by the hyperbolic logarithm of the ratio of expansion. As these must be added in order to get the total work done during the whole stroke it appears in the rules and formulas given, to make the process complete.

This is illustrated in Fig. 40 which shows a cut off at one-quarter stroke which makes the expansion rate 4 as previously explained. The vertical line extends from the point of cut off to the

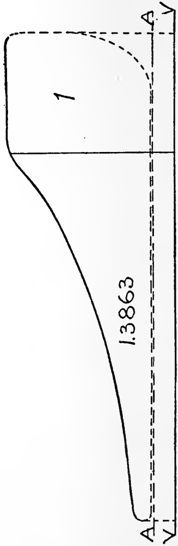


FIG 40

vacuum line, and the space to the right of it represents work done by steam direct from the boiler, and this is unity or 1. The space to the left shows work done by expansion of the steam, and it is represented by the hyperbolic logarithm of 4, which is 1.3863. When these two factors are added the sum is the total work done during the complete stroke. These measurements are taken to the vacuum line, because it is necessary in order to include the whole. The counter pressure line and the atmospheric line are shown in dotted form to make the diagram complete, but their only practical use in this connection is to show how the total work is divided, as they cannot make it either more or less.

Where an engine is located at or near the sea level, and is run non-condensing, the back pressure should be estimated at not less than 16 pounds, as the atmosphere causes nearly 15 and the remainder is allowed for friction in exhaust passages and pipes, as the steam travels to the outer air. If a condenser is to be used under good conditions, the back pressure may be taken at 3 pounds, as that represents good practice.

The following table contains hyperbolic logarithms of numbers from 1.01 to 20 to be used in connection with rules and formulas for determining the average and the mean effective pressures. When the ratio of expansion is determined, find the number in this table and the corresponding hyperbolic logarithm will be found in the next column, and this is to be used as directed.

HYPERBOLIC LOGARITHMS OF NUMBERS

from 1.01 to 20.00.

No.	Log	No.	Log.	No.	Log.
1.01	.0099	1.43	.3577	1.85	.6152
1.02	.0198	1.44	.3646	1.86	.6206
1.03	.0296	1.45	.3716	1.87	.6259
1.04	.0392	1.46	.3784	1.88	.6313
1.05	.0488	1.47	.3853	1.89	.6336
1.06	.0583	1.48	.3920	1.90	.6419
1.07	.0677	1.49	.3988	1.91	.6471
1.08	.0770	1.50	.4055	1.92	.6523
1.09	.0862	1.51	.4121	1.93	.6575
1.10	.0953	1.52	.4187	1.94	.6627
1.11	.1044	1.53	.4253	1.95	.6678
1.12	.1133	1.54	.4318	1.96	.6729
1.13	.1222	1.55	.4383	1.97	.6780
1.14	.1310	1.56	.4447	1.98	.6831
1.15	.1398	1.57	.4511	1.99	.6881
1.16	.1484	1.58	.4574	2.00	.6931
1.17	.1570	1.59	.4637	2.01	.6981
1.18	.1655	1.60	.4700	2.02	.7031
1.19	.1740	1.61	.4762	2.03	.7080
1.20	.1823	1.62	.4824	2.04	.7129
1.21	.1906	1.63	.4886	2.05	.7178
1.22	.1988	1.64	.4947	2.06	.7227
1.23	.2070	1.65	.5008	2.07	.7275
1.24	.2151	1.66	.5068	2.08	.7324
1.25	.2231	1.67	.5128	2.09	.7372
1.26	.2311	1.68	.5188	2.10	.7419
1.27	.2390	1.69	.5247	2.11	.7467
1.28	.2469	1.70	.5306	2.12	.7514
1.29	.2546	1.71	.5365	2.13	.7561
1.30	.2624	1.72	.5423	2.14	.7608
1.31	.2700	1.73	.5481	2.15	.7655
1.32	.2776	1.74	.5539	2.16	.7701
1.33	.2852	1.75	.5596	2.17	.7747
1.34	.2927	1.76	.5653	2.18	.7793
1.35	.3001	1.77	5710	2.19	.7839
1.36	.3075	1.78	·5766	2.20	.7885
1.37	.3148	1.79	·5822	2.21	.7930
1.38	.3221	1.80	·5878	2.22	.7975
1.39	.3293	1.81	·5933	2.23	.8020
1.40	.3365	1.82	·5988	2.24	.8065
1.41	.3436	1.83	·6043	2.25	.8109
1.42	.3507	1.84	·6098	2.26	.8154

Hyperbolic Logarithms of Numbers—*Continued*

No.	Log.	No.	Log.	No.	Log.
2.27	.8198	2.74	1.0080	3.21	1.1663
2.28	.8242	2.75	1.0116	3.22	1.1694
2.29	.8286	2.76	1.0152	3.23	1.1725
2.30	.8329	2.77	1.0188	3.24	1.1756
2.31	.8372	2.78	1.0225	3.25	1.1787
2.32	.8416	2.79	1.0260	3.26	1.1817
2.33	.8458	2.80	1.0296	3.27	1.1848
2.34	.8502	2.81	1.0332	3.28	1.1878
2.35	.8544	2.82	1.0367	3.29	1.1909
2.36	.8587	2.83	1.0403	3.30	1.1939
2.37	.8629	2.84	1.0438	3.31	1.1969
2.38	.8671	2.85	1.0473	3.32	1.1999
2.39	.8713	2.86	1.0508	3.33	1.2030
2.40	.8755	2.87	1.0543	3.34	1.2060
2.41	.8796	2.88	1.0578	3.35	1.2090
2.42	.8838	2.89	1.0613	3.36	1.2119
2.43	.8879	2.90	1.0647	3.37	1.2149
2.44	.8920	2.91	1.0682	3.38	1.2179
2.45	.8961	2.92	1.0716	3.39	1.2208
2.46	.9002	2.93	1.0750	3.40	1.2238
2.47	.9042	2.94	1.0784	3.41	1.2267
2.48	.9083	2.95	1.0813	3.42	1.2296
2.49	.9123	2.96	1.0852	3.48	1.2326
2.50	.9163	2.97	1.0886	3.44	1.2355
2.51	.9203	2.98	1.0919	3.45	1.2384
2.52	.9243	2.99	1.0953	3.46	1.2413
2.53	.9282	3.00	1.0986	3.47	1.2442
2.54	.9322	3.01	1.0009	3.48	1.2470
2.55	.9361	3.02	1.1053	3.49	1.2499
2.56	.9400	3.03	1.1086	3.50	1.2528
2.57	.9439	3.04	1.1119	3.51	1.2556
2.58	.9478	3.05	1.1151	3.52	1.2585
2.59	.9517	3.06	1.1184	3.53	1.2613
2.60	.9555	3.07	1.1217	3.54	1.2641
2.61	.9594	3.08	1.1249	3.55	1.2669
2.62	.9632	3.09	1.1282	3.56	1.2698
2.63	.9670	3.10	1.1314	3.57	1.2726
2.64	.9708	3.11	1.1346	3.58	1.2754
2.65	.9746	3.12	1.1378	3.59	1.2782
2.66	.9783	3.13	1.1410	3.60	1.2809
2.67	.9821	3.14	1.1442	3.61	1.2837
2.68	.9858	3.15	1.1474	3.62	1.2865
2.69	.9895	3.16	1.1506	3.63	1.2892
2.70	.9933	3.17	1.1537	3.64	1.2920
2.71	.9969	3.18	1.1569	3.65	1.2947
2.72	1.0006	3.19	1.1600	3.66	1.2975
2.73	1.0043	3.20	1.1632	3.67	1.3002

Hyperbolic Logarithms of Numbers—*Continued*

No.	Log.	No.	Log.	No.	Log.
3.68	1.3029	4.13	1.4183	4.58	1.5217
3.69	1.3056	4.14	1.4207	4.59	1.5239
3.70	1.3083	4.15	1.4231	4.60	1.5261
3.71	1.3110	4.16	1.4255	4.61	1.5282
3.72	1.3137	4.17	1.4279	4.62	1.5304
3.73	1.3164	4.18	1.4303	4.63	1.5326
3.74	1.3191	4.19	1.4327	4.64	1.5347
3.75	1.3218	4.20	1.4351	4.65	1.5369
3.76	1.3244	4.21	1.4375	4.66	1.5390
3.77	1.3271	4.22	1.4398	4.67	1.5412
3.78	1.3297	4.23	1.4422	4.68	1.5433
3.79	1.3324	4.24	1.4446	4.69	1.5454
3.80	1.3350	4.25	1.4469	4.70	1.5476
3.81	1.3376	4.26	1.4493	4.71	1.5497
3.82	1.3403	4.27	1.4516	4.72	1.5518
3.83	1.3429	4.28	1.4540	4.73	1.5539
3.84	1.3455	4.29	1.4563	4.74	1.5560
3.85	1.3481	4.30	1.4586	4.75	1.5581
3.86	1.3507	4.31	1.4609	4.76	1.5602
3.87	1.3533	4.32	1.4633	4.77	1.5623
3.88	1.3558	4.33	1.4656	4.78	1.5644
3.89	1.3584	4.34	1.4679	4.79	1.5665
3.90	1.3610	4.35	1.4702	4.80	1.5686
3.91	1.3635	4.36	1.4725	4.81	1.5707
3.92	1.3661	4.37	1.4748	4.82	1.5728
3.93	1.3686	4.38	1.4770	4.83	1.5748
3.94	1.3712	4.39	1.4793	4.84	1.5769
3.95	1.3737	4.40	1.4816	4.85	1.5790
3.96	1.3762	4.41	1.4839	4.86	1.5810
3.97	1.3788	4.42	1.4861	4.87	1.5831
3.98	1.3813	4.43	1.4884	4.88	1.5851
3.99	1.3838	4.44	1.4907	4.89	1.5872
4.00	1.3863	4.45	1.4929	4.90	1.5892
4.01	1.3888	4.46	1.4951	4.91	1.5913
4.02	1.3913	4.47	1.4974	4.92	1.5933
4.03	1.3938	4.48	1.4996	4.93	1.5953
4.04	1.3962	4.49	1.5019	4.94	1.5974
4.05	1.3987	4.50	1.5041	4.95	1.5994
4.06	1.4012	4.51	1.5063	4.96	1.6014
4.07	1.4036	4.52	1.5085	4.97	1.6034
4.08	1.4061	4.53	1.5107	4.98	1.6054
4.09	1.4085	4.54	1.5129	4.99	1.6074
4.10	1.4110	4.55	1.5151	5.00	1.6094
4.11	1.4134	4.56	1.5173	5.01	1.6114
4.12	1.4159	4.57	1.5195	5.02	1.6134

Hyperbolic Logarithms of Numbers—*Continued*

No.	Log.	No.	Log.	No.	Log.
5.03	1.6154	5.49	1.7029	5.95	1.7834
5.04	1.6174	5.50	1.7047	5.96	1.7851
5.05	1.6194	5.51	1.7066	5.97	1.7867
5.06	1.6214	5.52	1.7084	5.98	1.7884
5.07	1.6233	5.53	1.7102	5.99	1.7901
5.08	1.6253	5.54	1.7120	6.00	1.7918
5.09	1.6273	5.55	1.7138	6.01	1.7934
5.10	1.6292	5.56	1.7156	6.02	1.7951
5.11	1.6312	5.57	1.7174	6.03	1.7967
5.12	1.6332	5.58	1.7192	6.04	1.7984
5.13	1.6351	5.59	1.7210	6.05	1.8001
5.14	1.6371	5.60	1.7228	6.06	1.8017
5.15	1.6390	5.61	1.7246	6.07	1.8034
5.16	1.6409	5.62	1.7263	6.08	1.8050
5.17	1.6429	5.63	1.7281	6.09	1.8066
5.18	1.6448	5.64	1.7299	6.10	1.83
5.19	1.6467	5.65	1.7317	6.11	1.8099
5.20	1.6487	5.66	1.7334	6.12	1.8116
5.21	1.6506	5.67	1.7352	6.13	1.8132
5.22	1.6525	5.68	1.7370	6.14	1.8148
5.23	1.6544	5.69	1.7387	6.15	1.8165
5.24	1.6563	5.70	1.7405	6.16	1.8181
5.25	1.6582	5.71	1.7422	6.17	1.8197
5.26	1.6601	5.72	1.7440	6.18	1.8213
5.27	1.6620	5.73	1.7457	6.19	1.8229
5.28	1.6639	5.74	1.7475	6.20	1.8245
5.29	1.6658	5.75	1.7492	6.21	1.8262
5.30	1.6677	5.76	1.7509	6.22	1.8278
5.31	1.6696	5.77	1.7527	6.23	1.8294
5.32	1.6715	5.78	1.7544	6.24	1.8310
5.33	1.6734	5.79	1.7561	6.25	1.8326
5.34	1.6752	5.80	1.7579	6.26	1.8342
5.35	1.6771	5.81	1.7596	6.27	1.8358
5.36	1.6790	5.82	1.7613	6.28	1.8374
5.37	1.6808	5.83	1.7630	6.29	1.8390
5.38	1.6827	5.84	1.7647	6.30	1.8405
5.39	1.6845	5.85	1.7664	6.31	1.8421
5.40	1.6864	5.86	1.7681	6.32	1.8437
5.41	1.6882	5.87	1.7699	6.33	1.8453
5.42	1.6901	5.88	1.7716	6.34	1.8469
5.43	1.6919	5.89	1.7733	6.35	1.8485
5.44	1.6938	5.90	1.7750	6.36	1.8500
5.45	1.6956	5.91	1.7760	6.37	1.8516
5.46	1.6974	5.92	1.7783	6.38	1.8532
5.47	1.6993	5.93	1.7800	6.39	1.8547
5.48	1.7011	5.94	1.7817	6.40	1.8563

Hyperbolic Logarithms of Numbers—*Continued.*

No.	Log.	No.	Log.	No.	Log
6.41	1.8579	6.87	1.9272	7.33	1.9920
6.42	1.8594	6.88	1.9286	7.34	1.9933
6.43	1.8610	6.89	1.9301	7.35	1.9947
6.44	1.8625	6.90	1.9315	7.36	1.9961
6.45	1.8641	6.91	1.9330	7.37	1.9974
6.46	1.8656	6.92	1.9344	7.38	1.9988
6.47	1.8672	6.93	1.9359	7.39	2.0001
6.48	1.8687	6.94	1.9373	7.40	2.0015
6.49	1.8703	6.95	1.9387	7.41	2.0028
6.50	1.8718	6.96	1.9402	7.42	2.0042
6.51	1.8733	6.97	1.9416	7.43	2.0055
6.52	1.8749	6.98	1.9430	7.44	2.0069
6.53	1.8764	6.99	1.9445	7.45	2.0082
6.54	1.8779	7.00	1.9459	7.46	2.0096
6.55	1.8795	7.01	1.9473	7.47	2.0109
6.56	1.8810	7.02	1.9488	7.48	2.0122
6.57	1.8825	7.03	1.9502	7.49	2.0136
6.58	1.8840	7.04	1.9516	7.50	2.0149
6.59	1.8856	7.05	1.9530	7.51	2.0162
6.60	1.8871	7.06	1.9544	7.52	2.0176
6.61	1.8886	7.07	1.9559	7.53	2.0189
6.62	1.8901	7.08	1.9573	7.54	2.0202
6.63	1.8916	7.09	1.9587	7.55	2.0215
6.64	1.8931	7.10	1.9601	7.56	2.0229
6.65	1.8946	7.11	1.9615	7.57	2.0242
6.66	1.8961	7.12	1.9629	7.58	2.0255
6.67	1.8976	7.13	1.9643	7.59	2.0268
6.68	1.8991	7.14	1.9657	7.60	2.0281
6.69	1.9006	7.15	1.9671	7.61	2.0295
6.70	1.9021	7.16	1.9685	7.62	2.0308
6.71	1.9036	7.17	1.9699	7.63	2.0321
6.72	1.9051	7.18	1.9713	7.64	2.0334
6.73	1.9066	7.19	1.9727	7.65	2.0347
6.74	1.9081	7.20	1.9741	7.66	2.0360
6.75	1.9095	7.21	1.9755	7.67	2.0373
6.76	1.9110	7.22	1.9769	7.68	2.0386
6.77	1.9125	7.23	1.9782	7.69	2.0399
6.78	1.9140	7.24	1.9796	7.70	2.0412
6.79	1.9155	7.25	1.9810	7.71	2.0425
6.80	1.9169	7.26	1.9824	7.72	2.0438
6.81	1.9184	7.27	1.9838	7.73	2.0451
6.82	1.9199	7.28	1.9851	7.74	2.0464
6.83	1.9213	7.29	1.9865	7.75	2.0477
6.84	1.9228	7.30	1.9879	7.76	2.0490
6.85	1.9242	7.31	1.9892	7.77	2.0503
6.86	1.9257	7.32	1.9906	7.78	2.0516

Hyperbolic Logarithms of Numbers—*Continued*

No.	Log.	No.	Log.	No.	Log.
7.79	2.0528	8.25	2.1102	8.71	2.1645
7.80	2.0541	8.26	2.1114	8.72	2.1656
7.81	2.0554	8.27	2.1126	8.73	2.1668
7.82	2.0567	8.28	2.1138	8.74	2.1679
7.83	2.0580	8.29	2.1150	8.75	2.1691
7.84	2.0592	8.30	2.1163	8.76	2.1702
7.85	2.0605	8.31	2.1175	8.77	2.1713
7.86	2.0618	8.32	2.1187	8.78	2.1725
7.87	2.0631	8.33	2.1199	8.79	2.1736
7.88	2.0643	8.34	2.1211	8.80	2.1748
7.89	2.0656	8.35	2.1223	8.81	2.1759
7.90	2.0669	8.36	2.1235	8.82	2.1770
7.91	2.0681	8.37	2.1247	8.83	2.1782
7.92	2.0694	8.38	2.1258	8.84	2.1793
7.93	2.0707	8.39	2.1270	8.85	2.1804
7.94	2.0719	8.40	2.1282	8.86	2.1815
7.95	2.0732	8.41	2.1294	8.87	2.1827
7.96	2.0744	8.42	2.1306	8.88	2.1838
7.97	2.0757	8.43	2.1318	8.89	2.1849
7.98	2.0769	8.44	2.1330	8.90	2.1861
7.99	2.0782	8.45	2.1342	8.91	2.1872
8.00	2.0794	8.46	2.1353	8.92	2.1883
8.01	2.0807	8.47	2.1365	8.93	2.1894
8.02	2.0819	8.48	2.1377	8.94	2.1905
8.03	2.0832	8.49	2.1389	8.95	2.1917
8.04	2.0844	8.50	2.1401	8.96	2.1928
8.05	2.0857	8.51	2.1412	8.97	2.1939
8.06	2.0869	8.52	2.1424	8.98	2.1950
8.07	2.0882	8.53	2.1436	8.99	2.1961
8.08	2.0894	8.54	2.1448	9.00	2.1972
8.09	2.0906	8.55	2.1459	9.01	2.1983
8.10	2.0919	8.56	2.1471	9.02	2.1994
8.11	2.0931	8.57	2.1483	9.03	2.2006
8.12	2.0943	8.58	2.1494	9.04	2.2017
8.13	2.0956	8.59	2.1506	9.05	2.2028
8.14	2.0968	8.60	2.1518	9.06	2.2039
8.15	2.0980	8.61	2.1529	9.07	2.2050
8.16	2.0992	8.62	2.1541	9.08	2.2061
8.17	2.1005	8.63	2.1552	9.09	2.2072
8.18	2.1017	8.64	2.1564	9.10	2.2083
8.19	2.1029	8.65	2.1576	9.11	2.2094
8.20	2.1041	8.66	2.1587	9.12	2.2105
8.21	2.1054	8.67	2.1599	9.13	2.2116
8.22	2.1066	8.68	2.1610	9.14	2.2127
8.23	2.1078	8.69	2.1622	9.15	2.2138
8.24	2.1090	8.70	2.1633	9.16	2.2148

Hyperbolic Logarithms of Numbers—*Continued*

No.	Log.	No.	Log.	No.	Log.
9.17	2.2159	9.55	2.2565	9.93	2.2956
9.18	2.2170	9.56	2.2576	9.94	2.2966
9.19	2.2181	9.57	2.2586	9.95	2.2976
9.20	2.2192	9.58	2.2597	9.96	2.2983
9.21	2.2203	9.59	2.2607	9.97	2.2996
9.22	2.2214	9.60	2.2618	9.98	2.3006
9.23	2.2225	9.61	2.2628	9.99	2.3016
9.24	2.2235	9.62	2.2638	10.00	2.3026
9.25	2.2246	9.63	2.2649	10.25	2.3279
9.26	2.2257	9.64	2.2659	10.50	2.3513
9.27	2.2268	9.65	2.2670	10.75	2.3749
9.28	2.2279	9.66	2.2680	11.00	2.3979
9.29	2.2289	9.67	2.2690	11.25	2.4201
9.30	2.2300	9.68	2.2701	11.50	2.4430
9.31	2.2311	9.69	2.2711	11.75	2.4636
9.32	2.2322	9.70	2.2721	12.00	2.4849
9.33	2.2332	9.71	2.2732	12.25	2.5052
9.34	2.2343	9.72	2.2742	12.50	2.5262
9.35	2.2354	9.73	2.2752	12.75	2.5455
9.36	2.2364	9.74	2.2762	13.00	2.5649
9.37	2.2375	9.75	2.2773	13.25	2.5840
9.38	2.2386	9.76	2.2783	13.50	2.6027
9.39	2.2396	9.77	2.2793	13.75	2.6211
9.40	2.2407	9.78	2.2803	14.00	2.6391
9.41	2.2418	9.79	2.2814	14.25	2.6567
9.42	2.2428	9.80	2.2824	14.50	2.6740
9.43	2.2439	9.81	2.2834	14.75	2.6913
9.44	2.2450	9.82	2.2844	15.00	2.7081
9.45	2.2460	9.83	2.2854	15.50	2.7408
9.46	2.2471	9.84	2.2865	16.00	2.7726
9.47	2.2481	9.85	2.2875	16.50	2.8034
9.48	2.2492	9.86	2.2885	17.00	2.8332
9.49	2.2502	9.87	2.2895	17.50	2.8621
9.50	2.2513	9.88	2.2905	18.00	2.8904
9.51	2.2523	9.89	2.2915	18.50	2.9173
9.52	2.2534	9.90	2.2925	19.00	2.9444
9.53	2.2544	9.91	2.2935	19.50	2.9703
9.54	2.2555	9.92	2.2946	20.00	2.9957

In order to save the trouble of applying a rule or a formula for the purpose of determining the mean effective pressure for stated conditions, the next table is given, but the following explanation must be carefully noted, as otherwise correct results will not be secured from it.

The first horizontal line contains the various points at which steam is cut off from the cylinder, ranging from 1/10 to 9/10 of the stroke. The second line is the corresponding ratio of expansion, provided that clearance is not taken into account. The first column contains initial pressures from 10 to 200 pounds, all absolute or above a perfect vacuum, therefore, when taking gauge pressure for an example in practice at or near the sea level, 15 pounds should be added to it, and the resulting number found in the first column. Gauge pressure might have been used in place of absolute pressures, but in that case it would only apply to places where the atmosphere weighs nearly 15 pounds to the square inch, whereas the plan adopted makes it available for all conditions. For the same reason the back pressure was not subtracted, as there is no universal rule to follow.

To fully illustrate the use of this table of "Average Pressure of Steam" suppose that the initial pressure as shown by measuring upward from the atmospheric line of an indicator diagram, is 105 pounds, the atmospheric pressure is practically 15 pounds, and the point of cut off is at ¼ stroke. The engine is run non-condensing. What is the mean effective pressure?

The absolute pressure is $105+15=120$ pounds. Finding 120 in the first column and following the line to the right hand to the sixth column under ¼ we find that the average pressure is 71.58 pounds. Subtracting 15 for the atmosphere and one for friction of steam, or 16 pounds back pressure, shows that under these conditions the mean effective pressure is 55.58 pounds. This agrees with the result secured by calculation as demonstrated on preceding pages.

Assuming that a condensing engine is run under conditions that agree with the preceding example, except that while the load is increased enough to maintain the cut off at ¼ stroke, the back pressure is reduced to 3 pounds. What is the mean effective pressure?

We have demonstrated that the average pressure is 71.58 pounds, therefore the mean effective pressure is $71.58-3 = 68.58$ pounds.

Assuming that the non-condensing engine has 7 per cent. of clearance, what is the mean effective pressure, provided the clearance is taken into consideration?

Referring to the table entitled "Actual Ratios of Expansion," and following down the first column under P of C until .25 is found (meaning that the point of cut off is at $\frac{1}{4}$ or .25 of the stroke), then taking the horizontal line until the column under 7 per cent. is reached, we find that the actual ratio of expansion is 3.34. In the table of "Average Pressure of Steam" the nearest ratio of expansion is 3.33 which practically agrees with the conditions above mentioned. In this column on the horizontal line beginning with 120 pounds pressure we learn that the average pressure is 79.31 pounds from which must be subtracted the back pressure of 16 pounds, therefore the mean effective pressure is 63.31 pounds.

Suppose that a condenser is added to this engine, and more machinery is installed in the mill or shop, causing the cut off to take place at 1/5 or .20 of the stroke, the clearance remaining at 7 per cent., what is the mean effective pressure?

Following the directions above given the actual ratio of expansion is 3.96, which is 4 for all practical purposes. With 120 pounds initial pressure, and 4 expansions, or a ratio of expansion of 4, the average pressure is 71.58 pounds and the mean effective pressure is $71.58-3=68.58$ pounds.

The following explanation of the next table is given in condensed form for the convenience of application:

Cut off = The proportion of the stroke completed by the piston when the cut off valve closes.

R of E = Ratio of expansion, or 1 divided by the "cut off" as above described.

I P = Initial pressure absolute, or the pressure at the beginning of the stroke above a vacuum, which is maintained until the cut off valve closes.

AVERAGE PRESSURE OF STEAM

Cut-off	1/10	1/8	1/6	1/5	1/4
R.of E.	10	8	6	5	4
I. P.	Average Pressure				
10	3.30	3.84	4.64	5.21	5.96
15	4.94	5.77	6.98	7.82	8.94
20	6.60	7.69	9.29	10.43	11.93
25	8.25	9.62	11.63	13.04	14.91
30	9.90	11.54	13.96	15.65	17.90
35	11.56	13.47	16.28	18.26	20.87
40	13.21	15.39	18.59	20.87	23.86
45	14.86	17.32	20.94	23.48	26.84
50	16.51	19.24	23.26	26.09	29.82
55	18.16	21.16	25.59	28.69	32.80
60	19.81	23.09	27.92	31.31	35.79
65	21.46	25.01	30.24	33.91	38.77
70	23.10	26.94	32.55	36.53	41.75
75	24.77	28.86	34.90	39.13	44.73
80	26.42	30.79	37.21	41.74	47.72
85	28.07	32.71	39.55	44.35	50.57
90	29.73	34.64	41.88	46.96	53.68
95	31.37	35.56	44.20	49.57	56.66
100	33.03	38.48	46.51	52.18	59.65
105	34.68	40.41	48.56	54.78	62.63
110	36.38	42.34	51.17	57.40	65.61
115	37.98	44.26	53.51	60.00	68.59
120	39.64	46.18	55.84	62.62	71.58
125	41.28	48.10	58.16	65.22	74.56
130	42.92	50.03	60.47	67.83	77.54
135	44.59	51.95	62.82	70.40	80.52
140	46.23	53.88	65.13	73.05	83.51
145	47.89	55.80	67.47	75.66	86.49
150	49.54	57.73	69.80	78.27	89.47
155	51.19	59.65	72.13	80.87	92.45
160	52.84	61.58	74.45	83.48	95.44
165	54.49	63.50	76.78	86.09	98.42
170	56.15	65.42	79.10	88.70	101.4
175	57.80	67.35	81.43	91.31	103.3
180	59.45	69.27	83.76	93.77	107.3
185	61.10	71.20	86.08	96.53	110.3
190	62.75	73.12	88.41	99.14	113.3
195	64.40	75.05	90.74	101.7	116.3
200	66.06	76.97	93.05	104.4	119.3

AVERAGE PRESSURE OF STEAM

Cut-off	3/10	1/3	4/10	1/2	6/10	2/3
R.ofE.	3.33	3	2.5	2	1.66	1.5
I. P.	Average Pressure.					
10	6.60	6.98	7.66	8.46	9.03	9.36
15	9.91	10.49	11.49	12.69	13.55	14.05
20	13.21	13.98	15.32	16.93	18.07	18.73
25	16.52	17.48	19.16	21.16	22.59	23.41
30	19.82	20.99	22.99	25.39	27.11	28.10
35	23.13	24.48	26.82	29.64	31.62	32.78
40	26.43	27.98	30.65	33.86	36.15	37.43
45	29.74	31.48	34.38	38.09	40.66	42.14
50	33.04	34.96	38.32	42.32	45.18	46.82
55	36.38	38.48	42.15	46.55	49.93	51.51
60	39.65	41.98	45.98	50.79	54.22	56.20
65	44.00	45.47	49.81	55.02	59.00	60.88
70	46.26	48.96	53.65	59.25	63.26	65.56
75	49.61	52.47	57.48	63.48	68.08	70.25
80	52.87	55.96	61.31	67.72	72.30	74.86
85	56.23	59.47	65.14	71.95	77.16	79.61
90	59.48	62.97	68.97	76.18	81.34	84.28
95	62.84	66.46	72.80	80.41	86.24	88.98
100	66.08	69.92	76.64	84.65	90.36	93.64
105	69.46	73.46	80.47	88.88	95.32	98.35
110	72.70	76.94	84.30	93.11	99.40	103.0
115	76.07	80.46	88.13	97.34	104.4	107.7
120	79.31	83.96	91.96	101.6	108.4	112.4
125	82.69	87.45	95.80	105.8	113.4	117.0
130	85.91	90.92	99.63	110.0	117.5	121.7
135	89.31	94.45	103.4	114.2	122.5	126.0
140	92.52	97.94	107.3	118.5	126.5	131.1
145	95.92	101.4	111.1	122.7	131.6	135.8
150	99.13	104.9	114.9	126.9	135.5	140.5
155	102.5	108.4	118.7	131.2	140.7	145.5
160	105.8	111.9	122.6	135.4	145.2	149.9
165	109.1	115.4	126.4	139.6	149.7	154.5
170	112.4	118.9	130.2	143.9	154.3	159.2
175	115.7	122.6	134.1	148.1	158.9	163.9
180	119.0	125.9	137.9	152.3	163.4	168.6
185	122.3	129.4	141.7	156.6	167.9	173.2
190	125.6	132.9	145.6	160.8	172.4	177.3
195	129.0	136.4	149.4	165.0	177.0	182.6
200	132.3	139.9	153.3	169.3	180.7	187.3

AVERAGE PRESSURE OF STEAM

Cut-off	7/10	3/4	8/10	7/8	9/10
R.ofE.	1.43	1.33	1.25	1.14	1.11
I. P.	Average Pressure				
10	9.48	9.65	9.78	9.92	9.94
15	12.23	14.48	14.67	14.88	14.92
20	18.98	19.30	19.56	19.84	19.89
25	23.72	24.13	24.46	24.80	24.86
30	28.46	28.96	29.35	29.76	29.84
35	33.21	33.79	34.24	34.72	34.81
40	37.95	38.61	39.13	39.68	39.78
45	42.70	43.44	44.02	44.64	44.75
50	47.44	48.27	48.92	49.60	49.73
55	52.19	53.09	53.81	54.56	54.70
60	56.93	57.92	58.70	59.52	59.67
65	61.68	62.75	63.59	64.48	64.64
70	66.42	67.57	68.48	69.44	69.62
75	71.17	72.40	73.38	74.40	74.59
80	75.91	77.23	78.27	79.37	79.56
85	80.66	82.06	83.16	84.32	84.54
90	85.40	86.88	88.05	89.28	89.50
95	90.15	91.71	92.94	94.25	94.48
100	94.89	96.54	97.84	99.21	99.46
105	99.63	101.3	102.7	104.1	104.4
110	104.4	106.2	107.6	109.1	109.4
115	109.1	111.0	112.5	114.0	114.3
120	113.8	115.8	117.4	119.0	119.3
125	118.6	120.6	122.3	124.0	124.3
130	123.3	125.5	127.2	128.9	129.3
135	128.1	130.3	132.0	133.9	134.2
140	132.8	135.1	136.9	138.8	139.2
145	137.5	139.9	141.8	143.8	144.2
150	142.3	144.8	146.7	148.8	149.2
155	147.0	149.6	151.6	153.7	154.1
160	151.8	154.4	156.5	158.7	159.1
165	156.5	159.2	161.4	163.6	164.1
170	161.3	164.1	166.3	168.6	169.0
175	166.0	168.9	171.2	173.6	174.0
180	170.8	173.7	176.1	178.5	179.0
185	175.5	178.6	181.0	183,5	184.0
190	180.3	183.4	185.9	188.5	188.9
195	185.0	188.2	192.3	193.4	193.9
200	189.8	193.0	195.6	198.4	198.9

CAUTION

Do not attempt to use the foregoing table until you have carefully read the explanation of it which precedes the table, as otherwise correct results may not be obtained owing to misapplication of the figures given.

As a further illustration of its value take the case of an engine located on very high ground where the atmosphere pressure is only about two-thirds of what we find it at sea level, or say 10 pounds, for convenience. If the steam gauge indicates 110 pounds and the initial pressure is 5 pounds less, the initial pressure absolute is $110-5+10=115$ pounds. Assuming that the steam is cut off at $\frac{1}{4}$ stroke and ignoring the clearance, the table indicates that the average pressure is 68.59 pounds. If this engine is operated without a condenser the mean effective pressure is $68.59-16=52.59$ pounds. If the condenser is used it is $68.59-3=65.59$ pounds.

REMARKS ON CALCULATING THE HORSE POWER OF STEAM ENGINES

Preceding calculations on this important subject relate to simple double acting engines, which constitute a majority of the engines now in use, but there are others in service, and as the proper way to determine the power they develop may not be plain to the young engineer, explanations are herewith given that will prove useful in such cases.

Where a single acting engine is in use the effective piston speed is only one-half of what it is for a double acting engine, therefore the indicated horse power will be but one-half of that obtained by preceding rules and formulas.

Where there are two single acting cylinders the engine is to be treated as if it were a double acting engine with one cylinder. When calculating the power developed by these engines, the piston speed is taken while steam is acting on the piston. If this principle is always applied, correct results will be obtained. When treating a double acting engine, the total piston speed is to be used, but the mean effective pressure for two diagrams should not be added together and the result used in making the calculation, as that gives twice as much power as really is developed. It is proper to add them but the sum must be divided by 2.

A double engine is to be treated as if it consisted of two separate engines located in different parts of the works where machinery is used. The power of each is to be computed, in accordance with rules given in this book, and the results added together.

Where great accuracy is required it becomes necessary to make allowance for the space occupied by the piston rod on the area of the piston, because steam pressure cannot act on this space. In the case of a single acting engine with two cylinders, no allowance is to be made for the rod as steam only acts on the head end, where there is no rod to reduce the area.

With a double acting engine (not fitted with a tail rod), it is proper to subtract one-half of the area of the piston rod, and call the remainder the effective area of the piston. The piston rod occupies space on one side of the piston, while the other side is clear, therefore if one-half of it is assumed to be on each side of it, the result is correct.

For example, take a 20=inch piston with a piston rod $3\frac{7}{8}$ inches in diameter. The area of a 20=inch circle is 314.16 square inches, and the area of a $3\frac{7}{8}$ circle is 11.793 square inches. Then $(314.16 - 11.793) + 314.16 \div 2 = 308.26$ square inches. The area of the piston rod is 11.793 square inches, one-half of which is 5.896; then $314.16 - 5.896 =$ 308.26 square inches as before.

Large horizontal engines are fitted with heavy pistons that rest on the bottom of the cylinders and cause much friction. Unless superior lubrication is provided, the wear is excessive, and in order to improve these conditions, a tail rod is added which is illustrated in Fig. 41. The piston rod is continued through the cylinder head on the head end, which is fitted with a stuffing box similar to the crank end cylinder head. The outer end of this piston rod is carried by a second crosshead that travels on a lower guide, and as this is always in sight, it can be properly lubricated, thus preventing undue wear on the piston and the cylinder.

When calculating the power of such engines the whole area of the rod should be subtracted from the area of the piston. For a 20-inch piston with a rod $3\frac{7}{8}$ inches in diameter the effective area is $314.16 - 11.793 = 302.36$ square inches.

Where several pounds back pressure are added to an engine in order to use the exhaust steam for heating a mill or shop, or for any other purpose, it increases the load and makes it necessary to burn more coal. As this change makes no difference in the indicated horse power, it does not seem to be consistent to some engineers, but it admits of a perfectly logical explanation. The indicated horse power of an engine is represented by the area of the indicator

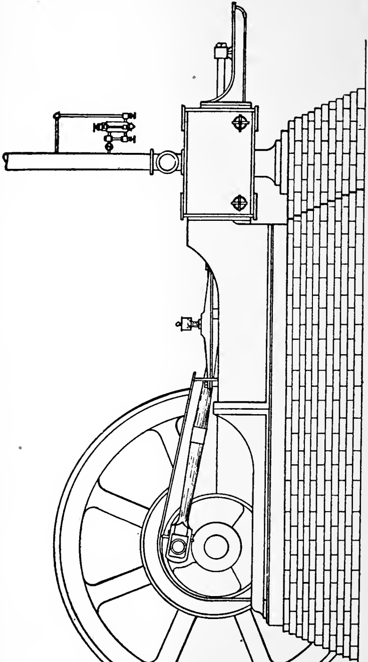


Fig. 41.

diagram taken from it, and while this varies with changes made in the machinery driven by this engine, it is not affected by changes in the back pressure. How then is the apparent change in the load accounted for?

Careful study of the following statements will make this clear. Taking for standard conditions an engine in which there is no back pressure above the atmosphere, the steam and expansion lines average a certain number of inches above the vacuum line, and this height represents the total load on the engine. If back pressure is added to the piston by forcing exhaust steam around the shop until the back pressure line is raised a certain part of an inch, the average height of the steam and expansion lines will be raised the same fraction of an inch, showing that the total load has been increased.

On the other hand, if a condenser is added and the back pressure line is lowered a certain fraction of an inch by it, the average height of the steam and the expansion lines will be lowered the same fraction of an inch, showing that the total load has been decreased, and the coal required to generate steam to operate the same machinery is less. Under all of the foregoing changes it is assumed that the same amount of power is required to drive the works, as any change in this respect will prevent proper comparison of the effects of the above mentioned difference in back pressure.

HORSE POWER CONSTANTS

It is frequently necessary to determine the power that an engine is developing under varying conditions, which cause changes in the mean effective pressure. As the speed is assumed to be constant under these changes (and it is practically so with an up-to-date engine), the mean effective pressure is the only factor that changes, hence it follows that if a constant is determined which represents the power developed for one pound mean effective pressure, it is only necessary to multiply this constant by the mean effective pressure to determine the power developed. This constant is found by use of the following rule.

Multiply the area of the piston by the piston speed in feet per minute, and divide by 33,000. The quotient is the horse power constant for that engine under given conditions.

For illustration suppose that an engine is fitted with a piston 20 inches in diameter, the area of which is 314.16 square inches. The stroke is 42 inches and the speed 90 revolutions per minute, giving a piston speed of 630 feet. Then $314.16 \times 630 \div 33{,}000 = 5.9976$ which is the horse power constant. If the mean effective pressure is 50 pounds to the square inch, it develops $5.9976 \times 50 = 299.88$ horse power.

The next table gives the horse power constants for cylinders from 4 to 60 inches in diameter, with piston speed from 10 to 900 feet per minute. To illustrate its practical application, suppose that a 24×48 inch engine, at 100 revolutions per minute shows a mean effective pressure of 56 pounds; what power is developed?

The piston speed is 8 feet per revolution, or 800 feet per minute. Finding the column under 800 and following it downward until the horizontal line opposite 24 is found, the horse power constant is 10.967. Then $10.967 \times 56 = 614.15$ horse power developed under these conditions.

Suppose that an 18×36 inch engine revolves 80 times per minute, and indicator diagrams from it show 47 pounds mean effective pressure; what power is developed?

The piston speed is 6 feet per revolution, or 480 feet per minute. In the column under 400 and on the horizontal line opposite 18 the horse power constant is 3.0845. On the same line under 80 the constant is .6169. These two constants are to be added together to account for 480 feet per minute, and $3.0845 + .6169 = 3.7014$. Multiplying this by 47 shows that 173.96 horse power was developed when the diagrams were taken.

HORSE POWER CONSTANTS

	Piston speed per minute.					
Diam.	10	20	30	40	50	60
4	.0038	.0076	.0114	.0152	.0190	.0228
5	.0059	.0119	.0178	.0238	.0297	.0357
6	.0085	.0171	.0257	.0342	.0428	.0514
7	.0116	.0233	.0349	.0466	.0583	.0699
8	.0152	.0304	.0457	.0609	.0761	.0913
9	.0192	.0385	.0578	.0771	.0963	.1156
10	.0238	.0476	.0714	.0952	.1190	.1428
11	.0288	.0576	.0863	.1151	.1439	.1729
12	.0342	.0685	.1028	.1370	.1713	.2056
13	.0402	.0804	.1206	.1608	.2011	.2413
14	.0466	.0933	.1399	.1865	.2332	.2798
15	.0535	.1071	.1606	.2142	.2677	.3213
16	.0609	.1218	.1827	.2437	.3046	.3655
17	.0687	.1275	.1963	.2651	.3339	.4026
18	.0771	.1542	.2313	.3084	.3855	.4626
19	.0859	.1718	.2577	.3436	.4295	.5155
20	.0952	.1904	.2856	.3808	.4760	.5712
21	.1049	.2099	.3148	.4198	.5247	.6297
22	.1151	.2303	.3455	.4607	.5759	.6911
23	.1259	.2518	.3777	.5036	.6295	.7554
24	.1370	.2741	.4112	.5483	.6854	.8225
25	.1487	.2975	.4462	.5950	.7437	.8925
26	.1608	.3217	.4826	.6435	.8044	.9653
27	.1735	.3470	.5205	.6940	.8675	1.0410
28	.1865	.3731	.5597	.7463	.9329	1.1196
29	.2001	.4003	.6004	.8006	1.0008	1.2009
30	.2142	.4284	.6426	.8568	1.0710	1.2852
31	.2287	.4574	.6861	.9148	1.1436	1.3723
32	.2437	.4874	.7311	.9748	1.2186	1.4623
33	.2591	.5183	.7775	1.0367	1.2959	1.5551
34	.2751	.5502	.8253	1.1005	1.3756	1.6508

Horse Power Constants—*Continued.*

Diam.	Piston speed per minute. 10	20	30	40	50	60
35	.2915	.5831	.8746	1.1662	1.4578	1.7493
36	.3084	.6169	.9253	1.2338	1.5422	1.8507
37	.3258	.6516	.9774	1.3033	1.6291	1.9549
38	.3436	.6873	1.0310	1.3747	1.7184	2.0620
39	.3620	.7240	1.0860	1.4480	1.8100	2.1720
40	.3808	.7616	1.1424	1.5232	1.9040	2.2848
41	.4000	.8001	1.2002	1.6003	2.0004	2.4005
42	.4198	.8386	1.2585	1.6783	2.0982	2.5180
43	.4400	.8801	1.3202	1.7602	2.2003	2.6404
44	.4607	.9215	1.3823	1.8431	2.3038	2.7646
45	.4819	.9639	1.4459	1.9278	2.4098	2.8917
46	.5036	1.0072	1.5108	2.0144	2.5180	3.0216
47	.5257	1.0515	1.5772	2.1030	2.6287	3.1545
48	.5483	1.0967	1.6451	2.1934	2.7418	3.2901
49	.5714	1.1429	1.7143	2.2858	2.8572	3.4286
50	.5950	1.1900	1.7850	2.3800	2.9750	3.5700
51	.6190	1.2381	1.8571	2.4762	3.0952	3.7142
52	.6435	1.2871	1.9307	2.5742	3.2178	3.8613
53	.6685	1.3371	2.0036	2.6742	3.3427	4.0113
54	.6940	1.3880	2.0820	2.7760	3.4700	4.1640
55	.7199	1.4399	2.1599	2.8798	3.5998	4.3197
56	.7463	1.4927	2.2391	2.9855	3.7318	4.4782
57	.7732	1.5465	2.3198	3.0930	3.8663	4.6396
58	.8006	1.6013	2.4019	3.2025	4.0032	4.8038
59	.8284	1.6570	2.4854	3.3139	4.1424	4.9709
60	.8568	1.7136	2.5704	3.4272	4.2840	5.1408

Diam.	Piston speed per minute. 70	80	90	100	200	300
4	.0266	.0304	.0342	.0381	.0762	.1142
5	.0416	.0476	.0535	.0595	.1190	.1785
6	.0599	.0685	.0771	.0857	.1714	.2570
7	.0816	.0933	.1049	.1166	.2332	.3499
8	.1066	.1218	.1370	.1523	.3046	.4570
9	.1349	.1542	.1735	.1928	.3856	.5783
10	.1666	.1904	.2142	.2380	.4760	.7140
11	.2015	.2303	.2581	.2880	.5760	.8639
12	.2399	.2741	.3084	.3427	.6854	1.0282
13	.2815	.3217	.3620	.4022	.8044	1.2067
14	.3265	.3731	.4198	.4665	.9330	1.3994
15	.3748	.4284	.4819	.5355	1.0710	1.6065

Horse Power Constants—*Continued.*

Diam.	Piston speed per minute.					
	70	80	90	100	200	300
16	.4265	.4874	.5483	.6093	1.2186	1.8278
17	.4614	.5402	.6190	.6878	1.2756	1.9635
18	.5397	.6169	.6940	.7711	1.5422	2.3134
19	.6014	.6873	.7732	.8592	1.7184	2.5775
20	.6664	.7616	.8568	.9520	1.9040	2.8560
21	.7347	.8396	.9446	1.0496	2.0992	3.1488
22	.8063	.9215	1.0367	1.1519	2.3038	3.4558
23	.8813	1.0072	1.1331	1.2590	2.5180	3.7771
24	.9516	1.0967	1.2338	1.3709	2.7418	4.1126
25	1.0413	1.1900	1.3388	1.4875	2.9750	4.4625
26	1.1262	1.2871	1.4480	1.6089	3.2178	4.8266
27	1.2145	1.3880	1.5615	1.7350	3.4700	5.2051
28	1.3061	1.4927	1.6793	1.8659	3.7318	5.5978
29	1.4011	1.6013	1.8014	2.0016	4.0032	6.0047
30	1.4994	1.7136	1.9278	2.1420	4.2840	6.4260
31	1.6010	1.8297	2.0585	2.2872	4.5744	6.8615
32	1.7060	1.4497	2.1934	2.4371	4.8742	7.3114
33	1.8143	2.0735	2.3326	2.5918	5.1836	7.7755
34	1.9259	2.2010	2.4762	2.7513	5.5026	8.2538
35	2.0409	2.3324	2.6240	2.9155	5.8310	8.7465
36	2.1591	2.4676	2.7760	3.0845	6.1690	9.2534
37	2.2808	2.6066	2.9324	3.2582	6.5164	9.7747
38	2.4057	2.7494	3.0930	3.4367	6.8734	10.310
39	2.5340	2.8960	3.2580	3.6200	7.2400	10.860
40	2.6656	3.0464	3.4272	3.8080	7.6160	11.424
41	2.8005	3.2006	3.6007	4.0008	8.0016	12.002
42	2.9378	3.3577	3.7775	4.1983	8.3866	12.585
43	3.0804	3.5205	3.9606	4.4006	8.8012	13.202
44	3.2254	3.6861	4.1469	4.6077	9.2154	13.823
45	3.3737	3.8556	4.3376	4.8195	9.6390	14.459
46	3.5253	4.0289	4.5325	5.0361	10.072	15.108
47	3.6802	4.2059	4.7317	5.2574	10.515	15.772
48	3.8385	4.3868	4.9352	5.4835	10.967	16.451
49	4.0001	4.5715	5.1429	5.7144	11.429	17.143
50	4.1650	4.7600	5.3550	5.9500	11.900	17.850
51	4.3333	4.9523	5.5713	6.1904	12.381	18.571
52	4.5049	5.1484	5.7920	6.4355	12.871	19.307
53	4.6790	5.3483	6.0169	6.6854	13.371	20.056
54	4.8581	5.5521	6.2461	6.9401	13.880	20.820
55	5.0397	5.7596	6.4796	7.1995	14.399	21.599
56	5.2246	5.9709	6.7173	7.4637	14.297	22.391
57	5.4128	6.1861	6.9584	7.7326	15.465	23.198
58	5.6044	6.4051	7.2057	8.0063	16.013	24.019
59	5.7993	6.6278	7.4563	8.2849	16.570	24.854
60	5.9976	6.8544	7.7112	8.5680	17.136	25.704

Horse Power Constants—*Continued*

Piston speed per minute.

Diam.	400	500	600	700	800	900
4	.1523	.1904	.2285	.2666	.3046	.3427
5	.2380	.2975	.3570	.4165	.4760	.5355
6	.3427	.4284	.5141	.5998	.6854	.7711
7	.4665	.5831	.6997	.8163	.9330	1.0496
8	.6093	.7616	.9139	1.0662	1.2186	1.3709
9	.7711	.9639	1.1567	1.3495	1.5422	1.7350
10	.9520	1.1900	1.4280	1.6660	1.9040	2.1420
11	1.1519	1.4399	1.7279	2.0159	2.3038	2.5818
12	1.3709	1.7136	2.0563	2.3990	2.7418	3.0845
13	1.6089	2.0111	2.4133	2.8155	3.2178	3.6200
14	1.8659	2.3324	2.7989	3.2654	3.7318	4.1983
15	2.1420	2.6775	3.2130	3.7485	4.2840	4.8195
16	2.4371	3.0464	3.6557	4.2650	4.8742	5.4835
17	2.6513	3.3391	4.0269	4.6147	5.4026	6.1904
18	3.0845	3.8556	4.6267	5.3978	6.1690	6.9401
19	3.4367	4.2959	5.1551	6.0143	6.8734	7.7326
20	3.8080	4.7600	5.7120	6.6640	7.6160	8.5680
21	4.1983	5.2479	6.2975	7.3471	8.3966	9.4462
22	4.6077	5.7596	6.9115	8.0634	9.2154	10.367
23	5.0361	6.2951	7.5541	8.8131	10.072	11.331
24	5.4835	6.8544	8.2253	9.5962	10.967	12.338
25	5.9500	7.4375	8.9250	10.413	11.900	13.388
26	6.4355	8.0444	9.6534	11.262	12.871	14.480
27	6.9401	8.6751	10.410	12.145	13.880	15.615
28	7.4637	9.3296	11.196	13.061	14.927	16.793
29	8.0063	10.008	12.009	14.011	16.013	18.014
30	8.5680	10.710	12.852	14.994	17.136	19.278
31	9.1487	11.436	13.723	16.010	18.297	20.585
32	9.7485	12.186	14.623	17.060	19.497	21.934
33	10.367	12.959	15.551	18.143	20.735	23.326
34	11.005	13.756	16.508	19.259	22.010	24.762
35	11.662	14.578	17.493	20.409	23.324	26.240
36	12.338	15.422	18.507	21.591	24.676	27.760
37	13.033	16.291	19.549	22.808	26.066	29.324
38	13.747	17.184	20.620	24.057	27.494	30.930
39	14.480	18.100	21.720	25.340	28.960	32.580
40	15.232	19.040	22.848	26.656	30.464	34.272
41	16.003	20.004	24.005	28.005	32.006	36.007
42	16.783	20.982	25.180	29.378	33.577	37.775
43	17.602	22.003	26.404	30.804	35.205	39.606
44	18.431	23.038	27.646	32.254	36.861	41.469
45	19.278	24,098	28.917	33.737	38.556	43.376
46	20.144	25.180	30.216	35.253	40.289	45.325

Horse Power Constants—*Continued*

	Piston speed per minute.					
Diam.	400	500	600	700	800	900
47	21.030	26.287	31.545	36.802	42.059	47.317
48	21.934	27.418	32.901	38.385	43.868	49.352
49	22.858	28.572	34.286	40.001	45.715	51.429
50	23.800	29.750	35.700	41.650	47.600	53.550
51	24.762	30.952	37.142	43.333	49.523	55.713
52	25.742	32.178	38.613	45.049	51.484	57.920
53	26.742	33.427	40.113	46.798	53.483	60.169
54	27.760	34.700	41.640	48.581	55.521	62.461
55	28.798	35.998	43.197	50.397	57.596	64.796
56	29.855	37.318	44.782	52.246	59.709	67.173
57	30.930	38.663	46.396	54.128	61.861	69.594
58	32.025	40.032	48.038	56.044	64.051	72.057
59	33.139	41.424	49.709	57.993	66.278	74.563
60	34.272	42.840	51.408	59.976	68.544	77.112

FLY WHEELS

When the speed of an engine is to be determined there are several points that ought to be considered, one of which is the safe speed for the fly wheel. This cannot be stated in revolutions per minute alone, but must be taken in connection with the diameter of the wheel. For illustration of this point we cannot say that a fly wheel should never be run more than 300 revolutions per minute, because the safe limit for a large wheel is much less and for a small one it may be a great deal more.

The safe limit of any wheel is the number of feet that the rim or the face of the wheel can travel in a minute without danger of disruption by centrifugal force.

The rim speed of a fly wheel may be determined by the following rule.

Multiply the diameter in feet by 3.1416 and by the number of revolutions per minute. The product is the speed in feet per minute.

Now the safe speed of cast iron fly wheels was formerly taken at 5,000 feet per minute (which is about one mile), but as higher speeds are imperatively demanded in modern practice, the limit has been raised to 100 feet per second or 6,000 feet per minute. Even this allows a large factor of safety, provided the iron is of fair quality, and the casting is free from defects, but these desirable qualities cannot be guaranteed in every case. All the parts of a fly wheel should be designed with a large factor of safety to withstand even greater strain than it is subjected to under ordinary working conditions, especially in view of the fact that if a governor does not control the speed perfectly, allowing it to increase above normal conditions, the strain may be greatly increased before the speed can be reduced.

When the rim of a fly wheel is made thicker than usual it is not a guarantee of greater safety, because centrifugal force increases with the weight, hence the strain is much greater on a thick rim than on a thin one, when both are run at the same speed, and the danger of hidden flaws in cast iron increases with the size of the casting, therefore after a certain thickness is secured, based on the shape of rim, number of arms supporting the same, etc., it is useless to make it thicker.

The number of revolutions per minute that can safely be allowed for well designed cast iron wheels, free from defective castings, is determined by the following rule:

Divide 6,000 by the diameter multiplied by 3.1416. The quotient is the safe number of revolutions per minute. For illustration take a wheel 10 feet in diameter. Then $6{,}000 \div (10 \times 3.1416) =$ 191 revolutions.

The next table gives the safe speed of wheels from 4 to 30 feet in diameter based on a rim speed of 6,000 feet per minute.

For well designed and constructed wooden fly wheels, it is probably safe to add 25 per cent. to the figures given in the table.

SAFE SPEED OF FLY WHEELS

Diameter in feet	Revolutions per minute	Diameter in feet	Revolutions per minute
3	636	17	112
4	477	18	106
5	381	19	100
6	318	20	95
7	272	21	91
8	238	22	86
9	212	23	83
10	191	24	79
11	173	25	76
12	159	26	73
13	146	27	70
14	136	28	68
15	127	29	66
16	119	30	63

MORE ABOUT HORSE POWER

Men who are not engineers and who have not given the matter much attention, seem to think that if an engineer is told the horse power of an engine that he has never seen, he should know at once the size of it, or in other words the diameter of cylinder and length of stroke. It is impossible to do this because a given horse power may be secured from a great variety of sizes, and on the other hand a given size may represent a wide range of power developed under different conditions.

In order to fully illustrate this matter several tables will be given which represent good practice at the present time as they are published by up-to-date engine builders as representing their product to prospective customers. For the benefit of the reader who is seeking practical information along this line, they will prove valuable as showing combinations of sizes that will give satisfaction in service.

The first of these tables shows the wide range of power that can be obtained from each engine of given size, beginning with a small one with a 10×24 inch cylinder rated at 38 horse power, and ending with a 34×60 inch cylinder rated at 647 horse power. There are four ratings given for each engine although only one speed is included in each case, and the boiler pressure is 80 pounds by the gauge, in all cases mentioned in the table, the columns of which contain the following information:

D = Diameter of cylinder in inches.
S = Stroke in inches.
R = Revolutions per minute.
P = Piston speed in feet per minute.
V = Variation in power without seriously affecting economy.
M = Maximum power that can be secured under given conditions.
B = Power developed with best economy.

HORSE POWER WITH 80 LBS. PRESSURE

D	S	R	P	V	M	B
10	24	110	440	33 to 43	56	38
12	28	95	443	48 " 59	76	53
14	32	85	453	68 " 83	109	75
16	30	90	450	90 " 109	142	100
16	38	85	538	106 " 128	170	117
16	42	85	595	119 " 144	188	131
18	36	90	540	135 " 151	213	143
18	42	85	595	149 " 181	234	165
20	42	80	560	174 " 211	275	192
20	48	75	600	186 " 227	300	205
22	42	75	525	197 " 240	313	218
22	48	75	600	225 " 274	360	250
22	52	75	649	244 " 296	390	270
24	42	82	560	257 " 312	398	284
24	48	75	600	270 " 326	429	295
24	56	70	653	294 " 350	465	322
26	48	75	600	315 " 383	503	350
26	54	70	630	331 " 400	528	367
26	60	65	660	344 " 416	545	380
28	54	67	603	367 " 444	584	405
28	60	65	650	400 " 483	632	440
30	60	65	650	459 " 556	726	507
32	48	80	640	510 " 619	809	564
32	60	65	650	520 " 633	826	576
34	60	65	650	585 " 710	932	647

The indicated horse power of an engine varies directly with the speed, provided this is not excessive, for it was probably designed to run at a given rate and if the number of revolutions is increased, the mean effective pressure may not be as high as it was with a lower speed, although the boiler pressure remains unchanged. While the preceding table gives various ratings at 80 pounds boiler pressure, the next states the indicated horse power that can be realized with higher boiler pressures. Both relate to simple, non-condensing engines

The several columns contain the following information.

D = Diameter of cylinder in inches
S = Stroke in inches.
R = Revolutions per minute.

There are two columns under 90 pounds, the first giving the builder's rating when steam is cut off at 1/5 stroke, and the second when the point of cut off is lengthened to 1/4 stroke, which is not unreasonable. The next two columns give the ratings with the same points of cut off, provided the boiler pressure is increased to 100 pounds, while the next two give ratings based on the same points of cut off, with a boiler pressure of 110 pounds by the gauge.

INDICATED HORSE POWER—LOW SPEED

D	S	R	90		100		110	
			1/5	1/4	1/5	1/4	1/5	1/4
11	24	110	50	79	55	65	61	72
12	30	90	62	54	69	83	77	92
12	36	85	70	84	78	94	87	105
14	36	85	95	114	107	128	121	142
16	36	82	120	144	135	162	151	180
16	42	78	133	159	150	179	168	200
18	36	80	148	177	166	199	186	222
18	42	78	168	202	189	227	212	253
18	48	75	195	222	208	249	234	271
20	42	75	200	240	225	270	252	300
20	48	72	219	263	246	296	275	330
22	42	75	242	290	271	326	303	364
22	48	72	265	318	298	358	333	400
24	48	70	307	368	345	414	386	460
26	48	70	360	432	405	486	454	541
28	48	68	406	487	457	548	514	595
30	48	68	444	526	507	594	590	683

LOW AND HIGH SPEED

When the speed of an engine is mentioned, it usually refers to revolutions of the crank shaft, but it may mean the piston speed in feet per minute. However, the latter may be the same for two engines where the former differs widely. For illustration, take a 12×36 inch engine at 90 revolutions, giving a piston speed of 540 per minute. If a 12×12 inch engine revolves three times as fast, making 270 revolutions, the piston speed will be 540 feet as in the preceding case, but the former is generally called a low speed engine, while the latter is known as a high speed machine. With the same initial pressure, and cutting off at equal points in the stroke, the indicated horse power is alike, therefore the difference between them is not so great as it appears at first.

If power is to be transmitted by belt from the engine to a shaft in a mill, shop or factory, the low speed engine with a long stroke is appropriate, because it will prove much more durable owing to the fact that the valve gear reverses its motion only 180 times per minute, and it is the reversing process that causes wear and tear in an engine. If a comparative high speed must be obtained at once, making it necessary to connect the engine directly to the machine to be driven as in the case of a dynamo or generator, the high speed engine with a short stroke must be used, but the valve gear is of a very different type, as a general rule, for its direction of travel must be reversed 540 times per minute, therefore, any form of cut off in which a moving part engages a part that is at rest (as with the Corliss valve gear) is impracticable and cannot be used successfully.

The following table gives the indicated horse power of high speed engines under various conditions. The sizes given refer to the diameter of cylinder and length of stroke. The speed mentioned is the number of revolutions per minute. Three rates are given for each engine, showing the range of speed for which it is adapted. Boiler pressures from 80 to 120 pounds are included with cut off at ¼, ⅜ and ½ stroke. The former should not be exceeded in general service, but if a heavy load is thrown on occasionally, the longer points of cut off may be utilized for the emergency. When selecting an engine of this class it should always be large enough to carry the estimated load, when running at the lowest speed given, and cutting off steam at the shortest point mentioned.

INDICATED HORSE POWER. HIGH SPEED

Size	Speed	Cut off at ¼ stroke					Cut off at ⅜ stroke					Cut off at ½ stroke				
		80	90	100	110	120	80	90	100	110	120	80	90	100	110	120
5x7	325	8	10	11	13	14	11	12	14	15	17	12	14	16	17	19
"	350	9	11	12	14	15	12	13	15	16	18	13	15	17	19	21
"	375	10	12	13	15	16	13	14	16	18	19	14	16	18	20	23
6x7	325	13	15	17	19	20	16	18	20	22	25	18	20	23	25	28
"	350	14	16	18	20	22	17	19	22	24	27	19	22	25	27	30
"	375	15	17	19	21	24	18	21	23	26	28	20	23	26	29	32
7x7	325	18	20	23	25	28	21	24	27	30	33	24	28	31	34	38
"	350	19	22	25	27	30	23	26	30	33	36	26	30	33	37	40
"	375	21	23	26	29	32	25	28	32	35	39	28	32	36	40	43
8x7	325	23	26	30	33	36	28	32	36	40	44	32	36	40	45	49
"	350	25	28	32	36	39	30	34	39	43	47	34	39	44	48	53
"	375	28	30	34	38	42	32	37	41	46	50	37	42	48	52	57
8x10	300	31	35	39	44	48	37	42	47	53	58	42	48	53	59	65
"	325	33	38	43	47	52	40	46	51	57	63	45	52	58	64	70
"	350	36	41	46	51	56	43	49	55	61	67	49	56	62	69	76

INDICATED HORSE POWER. HIGH SPEED

Size	Speed	Cut off at ¼ stroke					Cut off at ⅜ stroke					Cut off at ½ stroke				
		80	90	100	110	120	80	90	100	110	120	80	90	100	110	120
9x10	300	39	44	50	55	61	47	54	60	66	73	53	60	68	75	82
"	325	42	48	54	60	66	51	58	65	72	79	57	65	73	81	89
"	350	45	52	58	64	71	55	62	70	78	85	62	70	79	87	96
10x10	300	48	55	62	68	75	58	66	74	82	90	65	75	84	93	101
"	325	52	59	67	74	81	63	72	80	89	98	71	81	90	100	110
"	350	56	64	72	80	87	68	77	86	96	105	76	87	98	108	118
11x10	300	58	66	74	82	90	70	80	89	99	109	79	90	101	112	123
"	325	63	71	80	89	98	76	86	97	108	118	85	97	110	121	133
"	350	68	77	87	96	106	82	93	105	116	127	92	105	118	130	143
10x12	250	48	55	62	68	75	58	66	74	82	90	66	75	84	93	102
"	275	53	60	68	75	82	64	73	82	91	100	72	82	92	102	112
"	300	58	66	74	82	90	70	79	89	99	108	79	89	100	111	122
11x12	250	58	66	75	83	91	70	80	90	100	109	79	90	101	112	123
"	275	64	73	82	91	100	78	88	99	110	120	87	100	112	124	135
"	300	70	80	90	99	109	85	96	108	119	131	95	108	122	135	148

INDICATED HORSE POWER. HIGH SPEED

Size	Speed	Cut off at ¼ stroke					Cut off at ⅜ stroke					Cut off at ½ stroke				
		80	90	100	110	120	80	90	100	110	120	80	90	100	110	120
12x12	250	69	79	89	98	108	84	95	107	118	130	94	107	121	134	147
"	275	76	87	98	108	119	92	105	118	131	143	104	118	133	147	161
"	300	83	95	107	118	130	101	115	128	142	156	113	129	145	161	176
13x12	250	81	92	104	115	127	98	112	125	139	153	111	126	142	157	172
"	275	89	102	115	127	140	108	123	138	153	168	122	139	156	172	189
"	300	97	111	125	139	153	118	135	151	167	183	133	151	170	188	206
12x14	225	72	82	93	103	113	88	100	112	124	136	98	112	126	139	153
"	250	80	91	103	114	126	97	111	124	138	151	110	125	140	155	170
"	275	89	101	114	126	138	106	121	136	151	166	120	137	154	170	187
13x14	225	85	97	109	121	133	103	117	131	145	159	115	131	148	163	179
"	250	95	108	121	134	148	114	130	145	161	177	128	146	164	182	200
"	275	104	118	133	147	162	125	143	160	177	195	141	161	180	200	220
14x14	225	98	112	126	140	154	119	136	152	169	185	134	153	172	190	208
"	250	109	124	140	155	171	132	151	169	187	206	149	170	191	211	231
"	275	120	137	155	171	188	146	166	186	206	226	164	187	210	232	255

INDICATED HORSE POWER. HIGH SPEED

Size	Speed	Cut off at ¼ stroke					Cut off at ⅜ stroke					Cut off at ½ stroke				
		80	90	100	110	120	80	90	100	110	120	80	90	100	110	120
15x14	225	113	129	145	161	177	137	156	175	194	213	154	175	197	218	239
"	250	126	143	161	179	197	152	173	194	215	236	171	195	219	243	266
"	275	138	157	177	197	217	167	190	213	236	260	188	215	241	267	292
14x16	200	101	115	129	143	157	122	139	156	173	190	137	156	175	194	213
"	225	113	129	145	161	177	137	156	175	194	213	154	176	197	218	239
"	250	124	143	162	179	197	152	173	194	215	236	171	195	219	243	266
15x16	200	114	130	147	163	179	138	158	177	196	215	156	178	199	221	242
"	225	129	147	166	184	202	156	178	199	221	243	176	200	225	249	273
"	250	144	163	184	204	224	174	198	222	246	270	196	223	250	277	304
16x16	200	131	149	168	187	205	159	181	203	225	247	179	203	228	253	278
"	225	148	168	189	210	231	179	203	228	253	278	201	229	257	284	312
"	250	164	187	210	233	257	199	226	253	281	308	223	254	286	316	347
17x16	200	148	169	190	210	231	179	203	228	253	278	201	229	257	285	313
"	225	167	190	214	237	260	202	229	257	285	313	227	258	290	321	352
"	250	185	211	238	263	289	224	255	286	317	348	252	287	322	357	391

INDICATED HORSE POWER. HIGH SPEED

Size	Speed	Cut off at ¼ stroke					Cut off at ⅜ stroke					Cut off at ½ stroke				
		80	90	100	110	120	80	90	100	110	120	80	90	100	110	120
16x18	175	129	148	166	183	202	156	178	200	222	244	176	201	226	250	274
"	200	148	168	190	210	231	179	204	228	253	278	201	229	257	285	313
"	225	166	189	214	237	260	201	229	257	285	312	227	258	290	320	352
17x18	175	146	166	187	207	228	178	201	225	250	275	199	226	254	282	309
"	200	167	190	214	237	261	203	230	257	285	313	227	258	290	322	353
"	225	188	214	241	267	294	227	259	290	321	352	256	291	327	362	397
18x18	175	164	187	210	233	256	198	225	252	280	308	223	254	285	316	346
"	200	187	203	240	266	292	226	257	288	320	351	254	289	325	360	395
"	225	210	240	270	300	329	254	290	325	360	395	286	326	366	406	445
19x18	175	182	208	234	260	285	221	251	281	312	343	248	283	318	352	486
"	200	208	238	267	296	326	252	287	322	357	392	284	323	362	402	441
"	225	234	267	300	333	366	283	323	362	401	440	319	363	407	452	496
18x20	160	165	189	213	236	259	200	228	256	284	311	226	257	288	319	350
"	180	186	212	239	265	291	225	256	287	318	350	254	289	324	359	394
"	200	207	236	265	294	323	250	285	319	354	389	282	321	360	399	437

INDICATED HORSE POWER. HIGH SPEED.

Size	Speed	Cut off at ¼ stroke					Cut off at ⅜ stroke					Cut off at ½ stroke				
		80	90	100	110	120	80	90	100	110	120	80	90	100	110	120
19x20	160	184	211	237	262	288	223	254	285	316	346	252	286	321	356	390
"	180	208	237	266	295	324	251	286	320	355	390	283	322	361	400	439
"	200	231	263	296	328	360	279	318	356	394	433	314	358	401	445	488
20x20	160	205	234	263	292	321	248	282	316	350	385	279	318	357	396	434
"	180	230	263	296	328	360	278	317	355	394	433	314	357	401	444	487
"	200	256	292	328	364	400	309	352	394	437	480	348	396	445	493	541
21x20	160	226	257	289	320	353	273	311	348	386	424	307	349	392	435	477
"	180	254	289	326	361	397	307	350	392	434	476	346	394	442	490	537
"	200	282	322	362	402	442	341	389	435	483	530	384	438	491	544	597
20x22	150	211	241	271	301	331	256	292	327	362	398	288	328	368	408	447
"	165	232	265	298	331	364	281	320	359	398	437	317	360	405	448	492
"	180	254	289	325	361	397	307	349	392	434	476	346	393	441	488	536
21x22	150	232	265	298	331	363	281	320	359	397	436	316	360	404	448	492
"	165	255	292	328	364	398	309	352	394	437	480	348	396	445	493	540
"	180	278	318	358	396	435	337	384	430	477	524	380	432	485	537	590

INDICATED HORSE POWER—HIGH SPEED

Size	Speed	Cut off at ¼ stroke				
		80	90	100	110	120
22x22	150	255	291	328	363	399
"	165	281	320	361	400	439
"	180	306	349	393	436	479
23x22	150	280	319	359	398	437
"	165	308	351	395	438	481
"	180	336	383	431	478	525

Size	Speed	Cut off at ⅜ stroke				
		80	90	100	110	120
22x22	150	309	352	394	437	480
"	165	340	387	434	481	527
"	180	371	422	473	524	575
23x22	150	338	386	433	479	526
"	165	372	424	475	527	578
"	180	406	462	518	574	630

Size	Speed	Cut off at ½ stroke				
		80	90	100	110	120
22x22	150	348	396	444	492	540
"	165	383	435	488	542	594
"	180	418	475	533	591	648
23x22	150	381	434	487	540	592
"	165	420	478	536	594	652
"	180	457	521	585	648	711

COMPOUND ENGINES

Many steam engineers seem to believe that the principal object in adopting a compound engine is to secure more power than could be developed in any other kind occupying the same space. This idea is probably based on the well known fact that two cylinders are used and as they are supposed to develop more power than could be secured from one, it constitutes conclusive proof in their estimation. It follows as a natural consequence that if pressure in the receiver between the two cylinders is light, the second cylinder is considered of little or no value, because, it is claimed that the small piston does all of the work, and in addition to this, it must drive the large piston.

It is possible to find a few places where this state of affairs exists, but they are not so common as engineers who are not well educated along this line seem to believe. One reason for this belief is found in the following statement of facts: A small piston driven by high steam pressure may operate against a comparatively high back pressure, therefore, the net power available for driving machinery is not as great as the initial pressure indicates when considered alone. On the other hand a large piston may be driven by a low steam pressure, but it operates against a very low back pressure, consequently the net power available for useful work is nearly or quite equal to that secured in the high pressure cylinder.

If a simple engine is overloaded until it cannot maintain the required speed, when a superior cylinder oil is used to lubricate its internal parts, and the valves are properly set, the best way to secure more power is to remove the old machine and put in another simple engine with a cylinder large enough to do the work easily, provided economy in the use of steam is not of great importance. If a simple engine is run under conditions that prove wasteful of steam, the remedy is to install a compound engine that is well adapted to the necessary load. Where the exhaust steam can be used profitably, it should be run non-condensing, but otherwise a condenser should be added to remove back pressure from the large piston.

From this it will be plain that the real object in installing a compound engine is to develop power with the least possible amount of steam. This is consistent because this type makes it practicable to actually use steam at high pressure by expanding it to the greatest profitable extent, and at the same time cylinder condensation is reduced to a lower point than can be secured when expanding steam to a low pressure in a single cylinder.

If a very high pressure is used in a simple engine carrying a light load, the terminal pressure will be low, and the temperature of the cylinder at the end of each stroke will be low, consequently when another charge of steam is admitted, some of it will be condensed in raising the temperature of the cylinder, thus causing a loss of heat. With a compound engine this difference is divided between two cylinders, hence it is less for each, resulting in economical use of the steam.

DESIGNING COMPOUND ENGINES

A very good method of designing a simple engine, assuming that 300 horse power would be required, has been illustrated on previous pages, and in order to show the economy of the compound engine it becomes necessary to design one for a load of 300 horse power, and compare the results. It is assumed that this engine is run non-condensing.

Modern practice with these engines has demonstrated that within certain reasonable limits the pressure, total ratio of expansion, and comparative size of cylinders must follow general rules in order to secure good results, and these have been observed in the example given. Data to be used as a basis for the calculations may be stated as follows:

Gauge pressure at the boiler,	130	pounds
Atmospheric pressure,	15	"
Initial pressure absolute,	140	"
Ratio of expansion, high pressure cylinder,	3	
Ratio of expansion, low pressure cylinder,	2.5	
Total ratio of expansion,	7.5	

TO DEVELOP 300 HORSE POWER

In order to make an intelligent comparison that can be easily understood and appreciated, the stroke of this engine is assumed to be 42 inches and the speed 90 revolutions, making the piston speed 630 feet per minute, to correspond with the simple engine above mentioned.

The first point in the problem is to determine the mean effective pressure that will result from these conditions, using the following formula.

$$\frac{(\text{Hyp Log}+1)\times P}{R} - B = \text{M E P.}$$

Hyp Log = Hyperbolic logarithm of the ratio of expansion.
P = Initial pressure absolute.
R = Ratio of expansion.
B = Back pressure absolute.
M E P = Mean effective pressure.

Applying it to these conditions gives the following results:

$$\frac{(2.0149+1)\times 140}{7.5}-15=41 \text{ pounds}$$

mean effective pressure.

When designing a compound engine it is assumed that all of the power is developed in the low pressure cylinder, consequently it becomes necessary to determine the diameter of a cylinder that will prove large enough for this purpose, after which it will be used for a low pressure cylinder. A high pressure cylinder will then be designed and the total power developed is to be divided between them as nearly even as possible. The formula for a simple engine must be used here.

$$\frac{\text{H-P}\times\text{F-P}}{\text{M E P}\times\text{S}}=\text{A}$$

H-P = Horse power required.
F-P = Foot pounds for 1 horse power.
M E P = Mean effective pressure.
S = Piston speed in feet per minute.
A = Area of piston.

Applying it to stated conditions gives the following result:

$$\frac{300 \times 33{,}000}{41 \times 630} = 383 \text{ square inches.}$$

Referring to the table of the area of circles we find that the nearest area is 380 square inches, which corresponds to a circle 22 inches in diameter, therefore the cylinder is 22×42 inches. Proof is shown by calculating the power that this engine will develop as follows:

$$\frac{380 \times 41 \times 630}{33{,}000} = 297.4 \text{ horse power}$$

which is near enough for all practical purposes, as this is an estimate only.

Now the total ratio of expansion in a compound engine is found by multiplying the ratio in the high pressure cylinder by the ratio of the two cylinders, or by the comparative areas of them. To illustrate this point we may assume that the ratio of expansion in the high pressure cylinder is 3. The area of the low pressure cylinder is 2.5 times as great as the high pressure, therefore the total ratio of expansion is $3 \times 2.5 = 7.5$. The stroke of the two cylinders is supposed to be equal in these explanations.

Taking these facts into consideration, and having the area of the low pressure cylinder given, by dividing it by the ratio between the two, the quotient determines the area of the high pressure cylinder. In this case it is $380 \div 2.5 = 152$ square inches, which nearly corresponds to the area of a circle 14 inches in diameter, therefore the high pressure cylinder is 14 inches, with an area of 153.93 square inches. The mean effective pressure for the low pressure cylinder alone is 41 pounds, but when considering the complete engine, one-half of this must be applied to the low pressure cylinder, or $41 \div 2 = 20.5$ pounds. When the other half is multiplied by the ratio between the two cylinders, the product is the necessary mean effective pressure for the high pressure cylinder, and $20.5 \times 2.5 = 51.25$ pounds. This process is known as raising the mean effective pressure of the low pressure cylinder to terms of the high pressure. Under these conditions this cylinder will develop—

$$\frac{153.93 \times 51.25 \times 630}{33{,}000} = 150.6 \text{ horse}$$

power.

The low pressure cylinder will develop—

$$\frac{380 \times 20.5 \times 630}{33{,}000} = 148.7 \text{ horse power.}$$

The whole engine will develop **150.6** $+$**148.7** $=$**299.3** horse power.

THE MEAN EFFECTIVE PRESSURES IN COMPOUND ENGINES

It now becomes necessary to determine the mean effective pressures that can be secured under given conditions, and compare them with those required to secure the desired power, as before demonstrated. Before proceeding to do this a few general rules must be considered in order to make the matter plain.

1. When the initial pressure in the high pressure cylinder of a compound engine is divided by the ratio of expansion for that cylinder, the quotient is the terminal pressure.

2. The terminal pressure in the high pressure cylinder is equal to the back pressure in the same cylinder.

3. The back pressure in the high pressure cylinder is equal to the initial pressure in the low pressure cylinder.

4. When the initial pressure in the low pressure cylinder is divided by the ratio between the two cylinders, the quotient is the terminal pressure for that cylinder and for the whole engine.

5. The terminal pressure in the low pressure cylinder bears no definite relation to the back pressure in it, but they should be nearly equal in the case of a non-condensing engine.

6. The ratio of expansion for the low pressure cylinder is fixed by the comparative areas, or the ratio between the two cylinders. Thus, if the ratio of cylinders is 1 to 4 the ratio of expansion for the low pressure is 4 regardless of other conditions.

7. The total ratio of expansion divided by the ratio of cylinders, gives the ration of expansion for the high pressure cylinder.

In this case the initial pressure is 140 pounds, and the ratio of expansion is 3, therefore the terminal and the back pressure is $140 \div 3 = 46.66$ pounds. The mean effective pressure for this cylinder is—

$$\frac{(1.0986+1)\times 140}{3}-46.66=51.27 \text{ pounds.}$$

The mean effective pressure for the low pressure cylinder is—

$$\frac{(.9163+1)\times 46.66}{2.5}-15=20.76 \text{ pounds.}$$

These pressures very nearly agree with those determined by a different calculation, which proves their value.

The total power of this engine using the pressures found by the latter calculation is next given. For the high pressure cylinder it is—

$$\frac{153.93\times 51.27\times 630}{33,000}=150.6 \text{ horse power.}$$

For the low pressure cylinder it is—

$$\frac{380\times 20.76\times 630}{33,000}=150.6 \text{ horse power.}$$

The whole engine develops $150.6+150.6=301.2$ horse power.

The next table contains a list of long stroke compound non-condensing engines that are proportioned to suit the given steam pressure by the gauge, and when developing the power stated they will prove economical in the use of steam.

They will develop 25 per cent. more power without seriously affecting their efficiency, and the speed may be increased 20 per cent. with a corresponding increase in available power. However, the speeds given in the table are about right in order to secure durability of the valve gear and noiseless operation of the same.

LOW SPEED COMPOUND NON-CONDENSING ENGINES.

For 100 pounds gauge pressure.

Size	Speed	Power	Size	Speed	Power
11&16x30	100	100	26&38x60	65	734
11&16x36	90	108	27&40x54	70	782
12&18x36	90	135	27&40x60	65	812
12&18x42	80	140	28&42x54	70	869
14&20x30	100	155	28&42x60	65	897
14&20x36	90	167	30&44x54	70	958
15&22x36	90	205	30&44x60	65	988
15&22x44	80	213	31&46x66	60	1,095
16&24x36	90	243	31&46x72	55	1,095
16&24x42	80	252	32&48x66	55	1,089
18&26x42	80	297	32&48x72	50	1,090
18&26x48	75	318	34&50x66	55	1,286
19&28x42	80	342	34&50x72	50	1,176
19&28x48	75	366	35&52x66	55	1,283
20&30x42	80	392	35&52x72	50	1,272
20&30x48	75	420	36&54x66	55	1,385
22&32x48	75	480	36&54x72	50	1,374
22&32x54	70	504	38&56x66	55	1,488
23&34x48	75	546	38&56x72	50	1,476
23&34x54	70	573	39&58x66	55	1,597
24&36x54	70	636	39&58x72	50	1,584
24&36x60	65	656	42&62x66	55	1,821
26&38x54	70	712	42&62x72	50	1,806

LOW SPEED COMPOUND NON-CONDENSING ENGINES.

For 125 pounds gauge pressure.

Size	Speed	Power	Size	Speed	Power
9&14x30	100	92	22&34x54	70	687
9&14x36	90	99	22&34x60	65	708
10&16x36	90	130	23&36x54	70	769
10&16x42	80	135	23&36x60	65	793
11&18x30	100	150	24&38x54	70	857
11&18x36	90	162	24&38x60	65	884
13&20x36	90	203	25&40x66	60	900
13&20x42	80	210	25&40x72	55	990
14&22x36	90	246	26&42x66	55	1,004
14&22x42	80	255	26&42x72	50	996
16&24x42	80	302	28&44x66	55	1,183
16&24x48	75	324	28&44x72	50	1,092
17&26x48	75	357	29&46x66	55	1,204
17&26x60	65	382	29&46x72	50	1,194
18&28x42	80	413	30&48x66	55	1,313
18&28x48	75	443	30&48x72	50	1,302
19&30x48	75	509	32&50x66	55	1,422
19&30x54	70	534	32&50x72	50	1,410
20&32x54	70	607	34&54x66	55	1,658
20&32x60	65	627	34&54x72	50	1,644

LOW SPEED COMPOUND NON-CONDENSING ENGINES

For 150 pounds gauge pressure

Size	Speed	Power	Size	Speed	Power
9&14x30	100	100	14&22x36	90	267
9&14x36	90	108	14&22x42	80	277
10&16x36	90	141	16&24x42	80	329
10&16x42	80	146	16&24x48	75	353
11&18x30	100	165	17&26x42	80	386
11&18x36	90	179	17&26x48	75	414
13&20x36	90	220	18&28x42	80	448
13&20x42	80	228	18&28x48	75	480

Low Speed Compound Non-Condensing Engines
Continued

For 150 pounds gauge pressure

Size	Speed	Power	Size	Speed	Power
19&30x48	75	551	26&42x66	55	1,089
19&30x54	70	579	26&42x72	50	1,080
20&32x54	70	659	28&44x66	55	1,195
20&32x60	65	680	28&44x72	50	1,186
22&34x54	70	743	29&46x66	55	1,307
22&34x60	65	767	29&46x72	50	1,296
23&36x54	70	832	30&48x66	55	1,423
23&36x60	65	868	30&48x72	50	1,400
24&38x54	70	929	32&50x66	55	1,544
24&38x60	65	958	32&50x72	50	1,532
25&40x66	60	1,076	34&54x66	55	1,801
25&40x72	55	1,076	34&54x72	50	1,786

HIGH SPEED COMPOUND ENGINES

The long stroke engine with a comparatively low rotative speed, which gives a high piston speed, is generally considered the best for shop, mill and factory service, where high speed of the jack shaft can easily be secured by well proportioned pulleys and good belts, but there are cases where a high rotative speed is wanted without using belts, therefore, a high speed engine must be used.

The next table gives sizes of non-condensing engines that can be used with stated pressures to develop the power given in each case. As the cylinder ratio is 1 to 3.5 the higher pressures can be used to good advantage. The first column contains the diameter of cylinders and length of stroke. The second is the speed in revolutions per minute. Succeeding columns give the power developed.

HIGH SPEED COMPOUND NON-CONDENSING ENGINES

Cut off at ¼ Stroke.

Size	Speed	Initial pressure					
		100	110	120	130	140	150
7&11x10	300	30	34	37	41	44	48
"	325	32	36	40	44	48	52
"	350	34	39	43	47	52	56
8&13x10	300	40	45	50	55	60	65
"	325	43	49	54	59	65	70
"	350	46	52	58	64	69	75
8&13x12	250	40	45	50	55	60	65
"	275	44	50	55	60	66	71
"	300	48	54	60	66	72	78
9&16x12	250	55	62	68	75	82	89
"	275	60	68	75	83	90	98
"	300	66	74	82	90	99	107
9&16x14	225	57	65	72	79	86	94
"	250	64	72	80	88	96	104
"	275	70	79	88	96	105	114
11&18x14	225	80	90	100	109	119	129
"	250	88	100	111	122	132	144
"	275	97	110	122	133	145	158
11&18x16	200	81	91	101	111	121	131
"	225	91	102	114	124	136	148
"	250	101	114	127	139	152	164

HIGH SPEED COMPOUND NON-CONDENSING ENGINES

Cut off at 1/4 Stroke—*Continued*

Size	Speed	Initial Pressure					
		100	110	120	130	140	150
12&20x16	200	98	110	122	134	146	159
"	225	110	124	138	151	164	179
"	250	122	138	153	168	183	199
12&20x18	175	96	109	120	132	144	157
"	200	110	124	138	151	165	179
"	225	124	140	155	170	185	201
14&23x18	175	129	146	162	178	194	210
"	200	148	168	185	203	222	240
"	225	166	188	208	228	249	270
15&25x20	160	154	174	193	211	230	250
"	180	173	195	216	238	259	282
"	200	192	217	242	264	288	313
17&28x20	160	194	219	243	267	291	316
"	180	219	247	273	300	328	355
"	200	243	274	304	333	363	394
17&28x22	150	200	226	250	275	300	326
"	165	221	249	275	305	331	358
"	180	240	271	301	330	359	390
19&32x22	150	255	288	319	350	382	415
"	165	280	316	351	386	420	456
"	180	306	345	383	421	458	500

Cut off at 3/8 Stroke.

Size	Speed	Initial pressure.					
		100	110	120	130	140	150
7&11x10	300	41	45	50	55	60	65
"	325	44	49	55	59	65	70
"	350	47	53	59	64	70	76
8&13x10	300	55	61	68	74	81	88
"	325	59	66	73	81	88	95
"	350	64	72	79	87	94	102
8&13x12	250	55	61	68	74	81	88
"	275	60	67	75	82	89	97
"	300	66	74	81	89	97	105

HIGH SPEED COMPOUND NON-CONDENSING ENGINES

Cut off at ⅝ Stroke—*Continued*

Size	Speed	Initial pressure					
		100	110	120	130	140	150
9&16x12	250	76	85	94	103	112	122
"	275	83	93	103	114	124	134
"	300	91	102	113	124	136	146
9&16x14	225	79	89	99	108	118	128
"	250	88	99	110	121	131	142
"	275	97	109	121	132	144	156
9&18x14	225	109	123	136	149	162	176
"	250	121	136	151	165	180	195
"	275	134	149	166	182	198	214
11&18x16	200	111	125	138	151	164	178
"	225	125	410	155	170	185	201
"	250	139	156	712	189	206	223
12&20x16	200	134	151	167	183	199	216
"	225	151	170	188	206	224	243
"	250	168	189	209	229	249	270
12&20x18	175	132	148	164	180	196	213
"	200	151	170	188	206	224	243
"	225	170	191	211	232	252	274
14&23x18	175	178	199	220	242	263	285
"	200	202	228	252	277	301	326
"	225	224	256	284	311	338	367
15&25x20	160	212	237	263	288	314	340
"	180	238	267	296	324	353	383
"	200	264	297	328	361	392	426
17&28x20	160	266	299	331	364	395	429
"	180	300	336	372	409	445	482
"	200	333	374	414	454	494	536
17&28x22	150	274	308	342	375	407	442
"	165	303	339	376	413	449	486
"	180	330	370	410	449	489	531
19&32x22	150	350	394	436	479	520	565
"	165	386	433	479	526	573	621
"	180	420	472	522	574	624	678

HIGH SPEED COMPOUND NON-CONDENSING ENGINES

Cut off at ½ Stroke

Size	Speed	Initial pressure					
		100	110	120	130	140	150
7&11x10	300	47	52	58	63	69	75
"	325	51	57	63	69	75	81
"	350	55	61	68	74	81	87
8&13x10	300	63	71	79	86	94	101
"	325	69	77	85	93	101	110
"	350	74	83	92	100	109	118
8&13x12	250	63	71	79	86	94	101
"	275	70	78	86	95	103	111
"	300	76	85	94	103	112	122
9&16x12	250	88	99	110	120	131	142
"	275	97	109	121	132	144	156
"	300	106	119	132	144	157	170
9&16x14	225	93	104	116	126	138	149
"	250	103	116	128	139	153	166
	275	113	127	141	154	168	182
9&18x14	225	127	142	157	172	197	203
"	250	141	158	175	191	208	225
"	275	155	174	192	210	229	238
11&18x16	200	129	145	160	175	190	206
"	225	145	163	180	197	214	232
"	250	161	181	200	219	238	258
12&20x16	200	157	175	194	212	231	250
"	225	176	197	218	238	260	281
"	250	195	219	242	265	289	312
12&20x18	175	154	173	191	209	228	246
"	200	176	198	218	238	260	282
"	225	198	222	246	268	293	318
14&23x18	175	206	231	256	280	305	330
"	200	236	265	293	320	349	377
"	225	266	298	329	360	392	424
15&25x20	160	246	276	306	334	364	394
"	180	277	311	344	376	410	444
"	200	308	346	383	418	455	493
17&28x20	160	310	348	385	420	458	495
"	180	349	391	433	473	515	558
"	200	387	434	481	525	572	619
17&28x22	150	320	359	397	433	472	510
"	165	352	394	437	477	520	563
"	180	383	429	476	520	567	613
19&32x22	150	408	459	508	554	604	654
"	165	449	504	558	612	665	720
"	180	490	550	609	675	725	784

HIGH SPEED COMPOUND NON-CONDENSING ENGINES

Cut off at ⅔ Stroke

Size	Speed	Initial pressure 100	110	120	130	140	150
7&11x10	300	53	59	65	72	78	84
"	325	57	64	71	79	84	91
"	350	62	69	76	83	91	98
8&13x10	300	72	80	89	97	106	114
"	325	78	87	96	105	114	124
"	350	83	94	104	113	123	133
8&13x12	250	72	80	89	97	106	114
"	275	79	88	98	107	116	126
"	300	86	96	107	116	127	137
9&16x12	250	100	113	125	137	149	161
"	275	111	124	138	150	164	177
"	300	121	136	150	164	178	183
9&16x14	225	105	118	131	143	156	169
"	250	117	132	146	159	173	187
"	275	129	145	160	175	190	206
9&18x14	225	144	161	178	194	210	229
"	250	160	179	198	216	235	254
"	275	175	197	218	237	258	279
11&18x16	200	146	164	181	198	215	232
"	225	164	184	204	222	241	261
"	250	182	204	227	247	269	291
12&20x16	200	177	198	220	240	260	282
"	225	199	223	247	270	293	318
"	250	221	248	275	300	326	353
12&20x18	175	174	196	216	236	257	278
"	200	199	224	248	270	294	318
"	225	224	251	278	304	330	357
14&23x18	175	233	262	290	316	343	372
"	200	267	300	332	362	393	426
"	225	300	336	373	407	442	479
15&25x20	160	279	313	346	378	411	445
"	180	314	352	390	426	462	501
"	200	349	392	434	473	514	556
17&28x20	160	351	393	436	476	516	559
"	180	395	442	490	534	582	629
"	200	438	492	544	594	645	699
17&28x22	150	362	405	450	491	532	576
"	165	398	446	495	539	587	634
"	180	433	487	538	587	638	691
19&32x22	150	461	520	576	628	682	738
"	165	509	571	632	690	750	813
"	180	555	623	690	753	818	886

LOW SPEED COMPOUND CONDENSING ENGINES

As a general rule it is not a good idea to expand steam until the terminal pressure is lower than the back pressure, and while this rule limits the total ratio of expansion in a non-condensing compound engine to a point that will make the terminal pressure about 15 pounds absolute under ordinary conditions, and other considerations may raise it several pounds, the addition of a condenser to the plans and specifications for an engine to develop a given power, makes it possible to expand the steam much lower, as the terminal pressure may be as low as 6 pounds where the back pressure is about 3 pounds, both absolute or above a vacuum. This is accomplished by using a larger low pressure cylinder and cutting off steam earlier in the high pressure.

When designing compound engines the same rules can be used for both condensing and non-condensing service, but inasmuch as there is more than one method for this work, a better illustration of the whole process (or a larger part of it), will result from using a somewhat different plan here than was adopted for the non-condensing engine.

In order to make a complete and logical comparison later on, it is assumed that 300 horse power is wanted, and that the stroke of the engine is 42 inches with a piston speed of 630 feet per minute. When the other data is added the whole appears as follows, giving an intelligent basis for designing the cylinders.

Gauge pressure at the boiler...............	=140	pounds.
Atmospheric pressure...	= 15	"
Initial pressure absolute	=150	"
Terminal " "	= 7.5	"
Total ratio of expansion	=20	"
Ratio of cylinders......	= 1 to 4	
Stroke................	= 42 inches.	
Speed...	=90 revolutions.	
Piston speed per minute	=630 feet.	

To develop 300 horse power.

The first point is to determine the ratio of expansion for the high pressure cylinder. According to a rule previously given, this is found by dividing the total ratio of expansion by the ratio of cylinders. In this case it is $20 \div 4 = 5$. The ratio of expansion for the high pressure cylinder is 5, therefore steam is cut off in this cylinder at $\frac{1}{5}$ or .20 of the stroke, as clearance is ignored in these estimates. There is a drop in pressure between the two cylinders, especially in the case of a cross compound engine due to free expansion in the receiver, but inasmuch as this would seldom be the same in two or more engines in practice, no attempt is made to account for it when estimating cylinders for a compound engine.

The terminal pressure in the high pressure cylinder is $150 \div 5 = 30$ pounds. This is also the back pressure in the same cylinder, and the initial for the low pressure, all absolute. The mean effective pressure for the high pressure cylinder is therefore—

$$\frac{(1.6094+1)\times 150}{5} - 30 = 48.3 \text{ pounds.}$$

One-half of the total of 300 horse power, or 150, is to be developed in this cylinder, therefore the required area is—

$$\frac{150 \times 33{,}000}{48.3 \times 630} = 162.6 \text{ square inches,}$$

which is a circle 14.4 inches in diameter, but would be called 14 inches in order to avoid impracticable fractions. The area is 153.93 square inches, as this cylinder is 14×42 inches.

The given ratio of cylinders is 1 to 4, therefore, the area of the low pressure cylinder is $153.93 \times 4 = 615.7$ square inches, corresponding to a circle 28 inches in diameter.

As the initial pressure in this cylinder is 30 pounds and the ratio of expansion is 4 with a back pressure of 3 pounds, which is in accordance with ordinary practice in condensing engines, the mean effective pressure is—

$$\frac{(1.3863+1) \times 30}{4} - 3 = 14.89 \text{ pounds.}$$

Under these conditions the high pressure cylinder will develop—

$$\frac{153.93 \times 48.3 \times 630}{33{,}000} = 141.9 \text{ horse power.}$$

The low pressure will develop—

$$\frac{615.7 \times 14.89 \times 630}{33{,}000} = 175 \text{ horse power.}$$

The complete engine will develop $141.9 + 175 = 316.9$ horse power, which is about 5 per cent. more than is wanted but it is not always practicable (although it may be possible), to design cylinders under given conditions so that exactly the desired amount of power will be developed, but the automatic cut off device on the high pressure cylinder will respond to the demand for power and give desired results.

The power developed in these cylinders is not equal, but these conditions show the necessity of providing an automatic cut off for the low pressure cylinder, which can be adjusted by the engineer, then by lengthening the comparative range of the governor he can lengthen the point of cut off in the low pressure cylinder, thus throwing some of the load from the low to the high pressure cylinder, and make them nearer equal in this respect, for in these estimates it is assumed that the cut off in the low pressure cylinder corresponds to the ratio of areas between the two cylinders. In this case the ratio is 4, therefore the cut off in the low pressure cylinder is assumed to be at $\frac{1}{4}$ or .25 of the stroke.

A COMPARISON OF RESULTS

Here are three engines designed to develop approximately 300 horse power all having the same stroke, and as they revolve 90 times per minute the piston speed is alike, therefore, it is practicable to illustrate their relative economy by determining the weight of steam required per horse power hour by each, and comparing the results. The weight of steam used is found by the following rule.

Multiply the area of piston in square inches, by the distance in inches travelled by the piston at the point of cut off, and by the number of strokes per hour, and divide by 1,728. Multiply the quotient by the weight of steam per cubic foot at given pressure. Divide by the horse power developed, and the quotient is the weight of steam used per horse power hour. If it is desired to take clearance into account, multiply the above result by the given per cent. of clearance and add it to the amount.

The following data applies to the first engine to be considered:

Size of cylinder.... $= 20 \times 42$ inches.
Area " " $= 314.16$ square inches.
Ratio of expansion............ $= 4$
Distance to point of cut off........ $= 42 \div 4 = 10.5$ inches.

Strokes per hour $= 90 \times 2 \times 60 = 10{,}800$
Absolute pressure $= 120$ pounds

Weight per cubic foot............ $= .2724$ pounds.

Horse power developed........ $= 299.88$

Then $314.16 \times 10.5 \times 10{,}800 \div 1{,}728 =$ 20,616 cubic feet per hour, the weight of which is .2724 pound per cubic foot, therefore the total weight used per hour is $20{,}616 \times .2724 = 5{,}615.798$ pounds. Then $5{,}615.798 \div 299.88 = 18.72$ pounds per horse power hour.

Assuming that the clearance is 3 per cent. of the whole volume of the cylinder will raise this to $18.72 + (18.72 \times .03) =$ 19.28 pounds per horse power hour. Adding 30 per cent. to this increases it to 25.06 pounds, and the reason for this addition is found in the following lines.

The weight of steam accounted for by the above calculation which is based on theoretical conditions, does not represent the weight of water pumped into the boilers, and furthermore it is not intended for this purpose, as the only reliable process is to either measure or weigh the water as it goes into the boilers after which some of it will be used to run the engine forming a part of the plant, a portion will be used for operating pumps and other machines, and the remainder will be lost through leaky joints, or disappear as the result of radiation. These factors are seldom or never the same in two different plants, therefore it is not practicable to make a fixed allowance that will apply to all cases, and no attempt is made to do so here. However, if 30 per cent. is added to the results secured by these calculations, the sum will represent what can be realized under first-class conditions, but whether it is secured or not, can only be determined by direct experiment in each test, as it is made from time to time. As 30 per cent. is added in all cases mentioned here for comparison, the results bear the same relation to each other that existed previously, but the final amount in each case is reasonable, hence is an improvement over the rate secured by the necessarily crude and unfinished calculation previously secured.

The following data applies to the second engine to be considered, which is of the compound non-condensing type:

Size of cylinders. = 14 and 22×42 inches.
Area of high pressure cylinder... = 153.93 square inches.
Ratio of expansion for this cylinder = 3
Distance to point of cut off...... = $42 \div 3 = 14$ inches.
Strokes per hour. = $90 \times 2 \times 60 = 10,800$
Absolute pressure = 140 pounds.
Weight per cubic foot........... = .3147 pound.
Horse power developed.......... = 301.2

Then $153.93 \times 14 \times 10,800 \div 1,728 =$ 13,468 cubic feet per hour, the weight of which is .3147 pound per cubic foot, therefore the total weight used per hour is $13,468 \times .3147 = 4,238.379$ pounds.

Then $4,238.379 \div 301.2 = 14.07$ pounds per horse power hour. Adding 3 per cent. for clearance raises this to 14.49 pounds and adding 30 per cent. as before gives a total of 18.83 pounds per horse power hour.

The following data applies to the third engine to be considered which is of the compound condensing type:

Size of cylinders = 14 and 28×42 inches.
Area of high pressure cylinder = 153.93 square inches.
Ratio of expansion for this cylinder = 5
Distance to point of cut off .... $= 42 \div 5 = 8.4$ inches.
Strokes per hour $= 90 \times 2 \times 60 = 10{,}800$
Absolute pressure = 150 pounds.
Weight per cubic foot........... = .3358 pound.
Horse power developed....... = 316.9.

Then $153.93 \times 8.4 \times 10{,}800 \div 1{,}728 =$ 8,081 cubic feet per hour, the weight of which is .3358 pound per cubic foot, therefore the total weight used per hour is $8{,}081 \times .3358 = 2{,}713.6$ pounds.

Then $2713.6 \div 316.9 = 8.55$ pounds per horse power hour. Adding 3 per cent. for clearance raises this to 8.8 pounds, and adding 30 per cent. as before gives a total of 11.44 pounds per horse power hour.

The difference between results that can be secured by use of these engines does not seem large as here stated according to common practice, but when it is multiplied by the power developed in a given case, and this goes on from 10 to 24 hours per day according to the service in which an engine is used, the economy of securing a first-class engine and of placing a competent engineer in charge of it, will be very plain.

The next four tables give sizes of compound condensing engines that will render satisfactory service under given conditions. The first contains engines suitable for 125 pounds boiler pressure, when the load causes about 14 expansions and consideration of this matter calls for explanation of what constitutes unreasonable conditions. The fact that a compound engine must run under conditions which give a low terminal pressure if economical results are desired, has been made plain on previous pages, but the prospective customer, or the superintendent or the owner of a plant where one or more of these engines are to be used, frequently forget that a low terminal pressure means a comparatively light load. Now, if a very heavy load is put on a compound engine, resulting in a high terminal pressure in the second cylinder, this improved kind of an engine will be almost as wasteful as a simple engine, when used under similar conditions.

When the manufacturer of a compound engine is asked to state the amount of power that it will develop in an emergency, he usually answers the question just as it is stated, although he knows that while carrying the maximum load it is wasteful of steam. If a steam user proceeds to put this load on his engine for 10 hours or more per day, and the result is very unsatisfactory, as the wear and tear is excessive, and the coal bill proves to be very large, he ought to blame nobody but himself. He should ask the engine builder to state the power that it will develop on an economical basis, and then keep within the limits so determined.

The second table contains sizes of low speed engines where the ratio of cylinders is 1 to 4. This means that the area of the low pressure is four times as much as the high pressure. The ratio of diameter is 1 to 2 in all such cases, as stated in the list given. These engines are suitable for 150 pounds pressure by the gauge.

The third table contains the proper sizes for given loads where the pressure is 120 pounds, and the speed is high. The ratio of cylinders is about 1 to 3.5 although it varies with the larger sizes given.

The fourth table contains sizes of cylinders with a ratio of 1 to 4, to carry 150 pounds pressure. In this and in the preceding table three different speeds are given with power developed accordingly. If the speed of an engine is increased it will develop more power with out changing it from an economical to a wasteful machine, provided that the boiler pressure and the point of cut off remain constant. Excessive speed will cause rapid wear, and better lubrication will be necessary, but these details can usually be dealt with properly if an intelligent engineer is in charge and his recommendations are adopted. The speeds given in these tables are not excessive, and there is no good reason why a well made engine should give trouble of any kind while developing power as stated in them. Good judgment should be shown in the selection of a superior cylinder oil for all engines working under high pressure. The cost per gallon should be a secondary consideration, provided it does satisfactory work.

COMPOUND CONDENSING ENGINES.

Long stroke. Low speed. About 14 Expansions. Ratio of Cylinders Approximately 1 to 3⅛.

For 125 pounds boiler pressure.

Size	Speed	Power	Size	Speed	Power
9&16x30	100	96	22&40x54	70	750
10&18x30	100	122	23&42x60	65	864
11&20x36	90	163	24&44x60	65	948
12&22x36	90	190	25&46x60	65	1,036
13&24x36	90	234	26&48x66	55	1,050
14&26x42	80	285	28&50x66	55	1,140
16&28x42	80	330	29&52x66	55	1,233
17&30x42	80	380	30&54x72	50	1,319
18&32x48	75	463	31&56x72	50	1,411
19&34x48	75	523	32&58x72	50	1,520
20&36x54	70	615	34&62x72	50	1,739
21&38x54	70	678	36&64x72	50	1,858

COMPOUND CONDENSING ENGINES.

Long stroke. Low speed. 16 Expansions.
Ratio of cylinders 1 to 4.

For 150 pounds boiler pressure.

Size	Speed	Power	Size	Speed	Power
8&16x30	100	99	20&40x54	70	784
9&18x30	100	126	21&42x60	65	891
10&20x36	90	168	22&44x60	65	978
11&22x36	90	203	23&46x60	65	1,069
12&24x36	90	242	24&48x66	55	1,084
13&26x42	80	294	25&50x66	55	1,176
14&28x42	80	341	26&52x66	55	1,272
15&30x42	80	392	27&54x72	50	1,360
16&32x48	75	478	28&56x72	50	1,463
17&34x48	75	539	29&58x72	50	1,569
18&36x54	70	634	31&62x72	50	1,793
19&38x54	70	707	32&64x72	50	1,900

COMPOUND CONDENSING ENGINES.

Short stroke. High speed. About 14 Expansions.
Ratio of Cylinders approximately 1 to 3.5.

For 120 pounds boiler pressure.

Size	Speed	Power	Speed	Power	Speed	Power
7&11x10	300	45	325	49	350	53
8&13x10	300	61	325	66	350	71
8&13x12	250	61	275	67	300	73
9&16x12	250	84	275	93	300	101
9&16x14	225	89	250	98	275	108
11&18x14	225	122	250	136	275	149
11&18x16	200	124	225	139	250	155
12&20x16	200	150	225	169	250	187
12&20x18	175	148	200	169	225	190
14&23x18	175	198	200	227	225	255
15&25x20	160	236	180	266	200	295
17&28x20	160	298	180	335	200	372
17&28x22	150	308	165	337	180	368
19&32x22	150	391	165	432	180	470

COMPOUND CONDENSING ENGINES

Short stroke. High speed. 16 Expansions
Ratio of cylinders 1 to 4.

For 150 pounds boiler pressure

Size	Speed	Power	Speed	Power	Speed	Power
5½&11x10	300	42	325	45	350	49
6½&13x10	300	58	325	63	350	68
6½&13x12	250	58	275	64	300	70
8 &16x12	250	88	275	97	300	106
8 &16x14	225	93	250	103	275	113
9 &18x14	225	117	250	130	275	143
9 &18x16	200	119	225	134	250	149
10 &20x16	200	147	225	165	250	184
10 &20x18	175	145	200	166	225	186
12 &24x18	175	199	200	228	225	256
12½&25x20	160	240	180	270	200	300
14 &28x20	160	288	180	324	200	360
14 &28x22	150	297	165	326	180	356
16 &32x22	150	387	165	427	180	465

PROVIDING FOR INCREASE OF POWER WHEN INSTALLING A COMPOUND ENGINE

When a steam plant is to be designed constructed and installed, the services of a competent designing engineer should be secured to draw the specifications and superintend the work, as it is impossible to lay down rules for such cases that will cover every contingency that may arise without making them too cumbersome for practical use. It is possible and practicable, however, to make a few suggestions and give rules that apply to this work, that will prove valuable to both engineer in charge, and owner of the plant, as they either give clear directions, or point out the necessity for investigation along these lines.

Suppose that a cross compound condensing engine is preferred, and for one or two years the load is to be very light after which it will be greatly increased. This engine is illustrated in Fig. 42, but while the load is light the low pressure connecting rod may be disconnected from the crank pin, and the eccentric straps taken off, thus leaving it a simple non-condensing engine. In order to illustrate the matter, assume that a single diagram taken from this engine under these conditions appears like Fig. 43. It is now working under economical conditions as the terminal pressure is not enough above the atmosphere to denote serious waste, yet it is sufficient to prevent excessive cylinder condensation. This engine was designed for 150 pounds boiler pressure by the gauge, but only 100 is carried now, as no more is needed, and it is useless to carry a high boiler pressure unless it can actually be used to good advantage.

Fig. 44 is another diagram from the same engine while carrying 100 pounds boiler pressure, but the load is much heavier than before. This is a wasteful

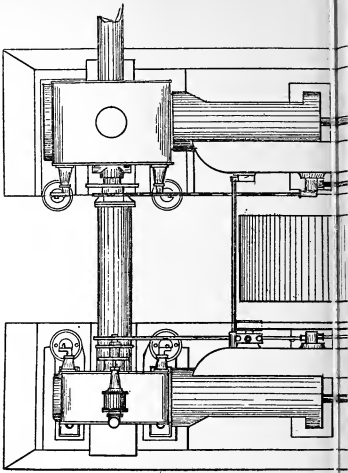


FIG. 42

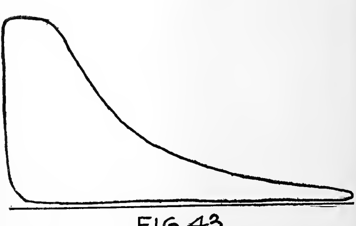
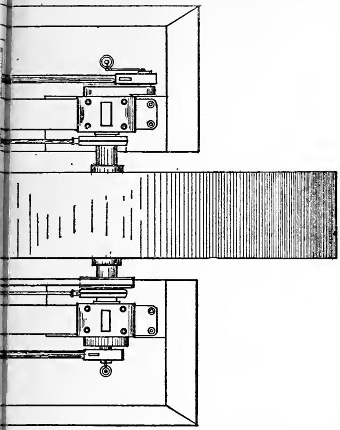


FIG. 43



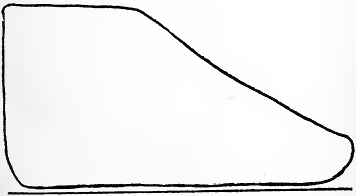


FIG. 44

condition of affairs, therefore, the boiler pressure is raised to 150 pounds and the diagram Fig. 45 is taken. The in-

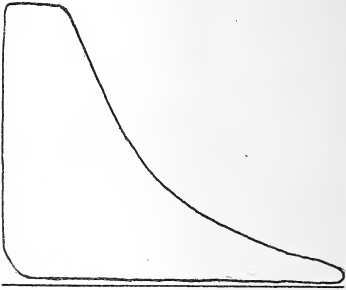


FIG. 45

creased boiler pressure has shortened the point of cut off and lowered the terminal pressure until economical conditions are again secured. However, the load is again increased until it produces a diagram similar to 46. The limit of boiler pressure is shown by the diagram, therefore, the next move is to connect the low pressure connecting rod, adjust the eccentric straps, make proper steam connections, put the condenser into service and run the engine as it was originally intended to be used. A diagram

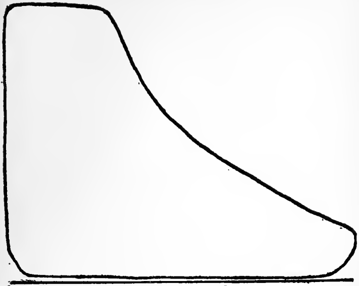


FIG. 46

from the high pressure cylinder as now running, is shown in Fig. 47, and one

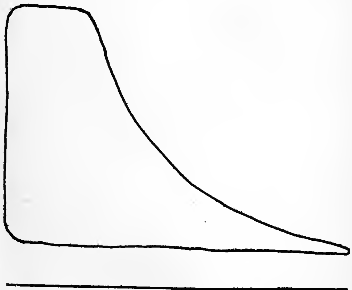


FIG. 47

from the low pressure cylinder appears in Fig. 48.

The flexibility of an engine of this kind is well illustrated by this example, and there are other points that are

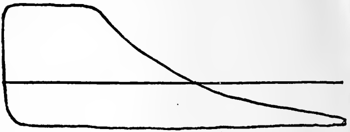


FIG. 48

worthy of special attention. The complete engine should be installed at first, and such parts as are not wanted for immediate use can be held in reserve until needed. While this is the best plan that can be devised, it is objected to by some engineers and owners, as they believe that only the high pressure side should be installed at first, as the low pressure side can be added when wanted, thus saving interest on the investment, etc. This plan is earnestly objected to for the following reason: My observation and experience teaches that although a plant may be started in an unfinished condition, it will be run that way as long as possible.

The engine may be overloaded until the speed is much less than it ought to be, and the terminal pressure is so high that a large quantity of heat is thrown away every day, but so long as the plant continues to run without an absolute stop, there is no time to shut down for improvements, money cannot be spared for that purpose, etc., and matters usually remain in that condition until enough money has been wasted to more than pay the cost of needed improvements, after which they are made, resulting in marked economy of operation causing the owner to wonder why he did not do it before.

Other reasons why the low pressure side of this engine ought to be installed complete when the plant is first erected, are that if it is not done then, the firm who build these engines may go out of business, their works may burn down, or they may be so busy when the extra parts are wanted that it will be necessary to wait a year for the work to be completed.

On the other hand if the low pressure side is completed, it may be put into service at a few hours' notice, thus saving serious loss due to shutting down the plant for several weeks, for such machinery is sometimes wanted when least expected. Suppose that the piston rod on the high pressure side should break, allowing the piston to be forced out through the cylinder head, taking pieces

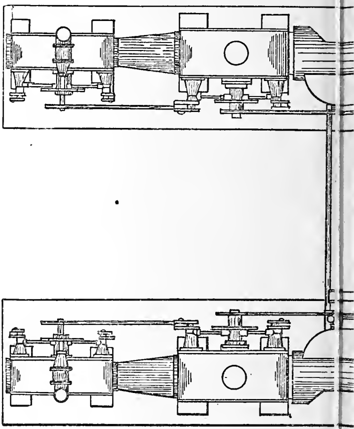
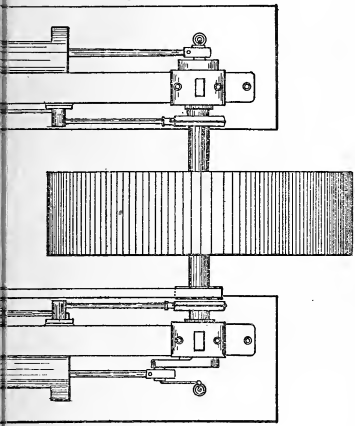


FIG. 9.

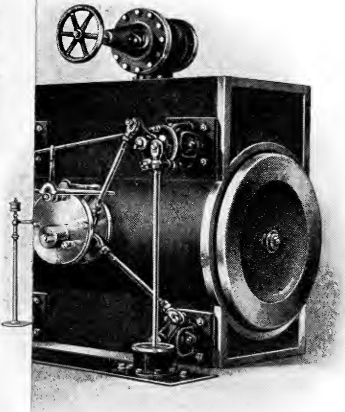
of the cylinder with it, thus making it necessary to build a new cylinder, piston and piston rod. The high pressure connecting rod could then be disconnected, and the eccentric straps taken off. The piping might be quickly changed to deliver steam directly to the low pressure cylinder, the boiler pressure reduced to 30 or 40 pounds and in a few hours



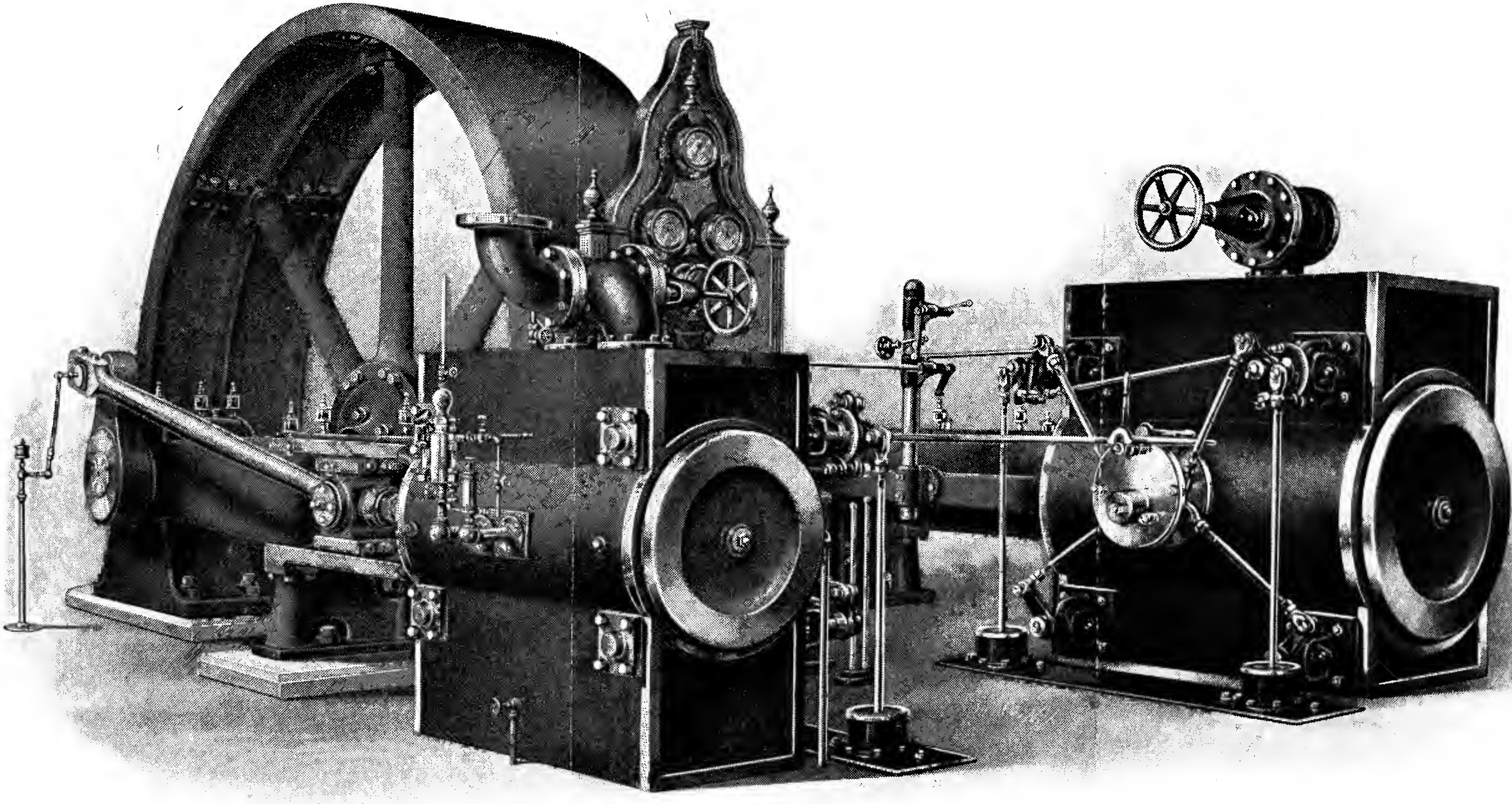
the plant could be in full operation, which might easily be continued indefinitely, or until repairs were completed. This would also make it unnecessary to work overtime on the job, thus saving heavy expense, as overtime must be paid for extra, although the work cannot be carried on to good advantage at night with ordinary lighting facilities, by men who are taxed beyond their strength in working from 24 to 48 hours without rest.

Another advantage of this plan is that it does not include expensive changes in the engine in order to secure the full power, no additions are required, and the speed is not increased, therefore it is not necessary to provide a new and larger main pulley on the jack shaft to keep the machinery speed constant while the engine speed is changed, and the main belt does not have to be lengthened to meet new conditions.

Fig. 49 illustrates a double tandem compound condensing engine that is a very flexible unit. If only a small portion of the full load is to be carried at first, one side can be laid up, and a light boiler pressure will be sufficient for the other. After six months have elapsed the other side can be started and each run alternately, thus keeping one side in reserve, ready for use in case of accident to the other. As the load is increased, the boiler pressure can be



:R FRAME TYPE



CROSS COMPOUND HARRIS-CORLISS ENGINE, STANDARD GIRDER FRAME TYPE

raised, and when it is no longer economical to run with one side they can both be used with a low boiler pressure, then as more load is added it can be raised until the full power of the complete engine is utilized.

While the highest efficiency cannot be expected under all of these varying conditions, it will be realized under some of them, hence the total difference is small when compared with the benefits secured.

SECTION 4

PRACTICAL POINTERS FOR PROGRESSIVE POWER PLANT OPERATORS

PACKING A MAN-HOLE COVER

Putting in a man-hole cover is a rough job, yet it requires a certain amount of care in order to make it successful, for if the gasket is cut in two by coming into contact with a sharp corner it will cause a bad leak, and where a tubular gasket is used, it is necessary to clamp both ends of it in proper place, or else it will be necessary to empty the boiler and make a new joint.

While this is very unpleasant to all concerned, when there is plenty of time to repack the joint, it is much worse to discover such a mistake about an hour before it is time to start the engine, because every self-respecting engineer wishes to start his plant on time, as failure to do so, even when there is apparently a good reason for it, injures his reputation and causes his employer to lose confidence in him. On this account it is always a good idea to raise pressure on a boiler after it has been cleaned and inspected, before it is wanted for actual service, then if it becomes necessary to cool it off and repack one or more joints it will cause but little expense and no delay to the plant.

FAILURE OF MAN-HOLE GASKETS

Sometimes a man-hole or a hand-hole gasket will blow out under pressure, but such accidents are always due to bad design of the parts, carelessness in making the joints, or failure to properly care for them. The surfaces to be packed may be roughened by corrosion, or made winding by abuse of the boiler. It may be necessary in such a case to use two flat gaskets, but this should be avoided as long as possible, because the extra thickness of gasket makes it more liable to blow out. When a boiler is filled with cold water and one of these joints leaks, it is a great temptation to seize a long wrench and screw the nut as far down as possible. Of course this is proper to a limited extent, but if excessive leverage is applied the cast iron dog which spans the hole may be broken. It is well to have an extra one on hand for such emergencies.

PACKING A HAND-HOLE COVER

Horizontal return tubular boilers are frequently made without a hand hole in the rear head. As long as the boiler is in service, there is no need of a hand hole here, but when it is laid off to be cleaned it is very convenient to have a place where at least a little fresh air can be secured, also an opening through which tools can be passed. It is difficult in some cases and impossible in others to properly care for a hand-hole cover owing to its location. When packed with a rubber gasket and the nut is screwed down firmly, the joint may be tight, but when steam is raised the heat softens the gasket, and pressure acting on the back of the cover settles it down into place until the nut is quite loose. If it is made tight again before all pressure is taken off, it will probably make a good joint until it is time to remove the cover again, but if there is no chance to do this for several days or weeks, and the steam pressure goes down to zero, it will probably leak, and water coming through the imperfect joint will cause corrosion that in course of time will waste away the head until a tight joint cannot be secured.

FITTING A HAND-HOLE COVER

Before attempting to pack a hand–hole cover for the first time, on either an old or a new boiler, the engineer should put the cover in place without packing and be sure that the hole is large enough to allow the cover to rest firmly on the head without binding on the edges as otherwise the gasket may not be clamped firmly, and when subjected to pressure it may be blown out. After a tight joint is secured, and all lost motion due to shrinking of the gasket has been taken up, if the bolt projects through the nut, another should be screwed on to protect the end of it, then the dog, nuts and bolt should be covered with asbestos to protect them from the fierce heat.

RUBBER AND METAL GASKETS

As a general rule, to which there may be a few exceptions, a rubber man-hole gasket can be used several times, provided there is a broad surface in contact to afford a good bearing. If it is desired to use a gasket more than once, the side put next to the cover should not contain anything to prevent it from sticking to the metal, but the other side ought to be coated with Dixon's graphite, ground fine and mixed with cylinder oil to the consistency of a thick paste. This will prevent the rubber from sticking to the head, and if care is taken to clean pieces of scale, etc., off from the surfaces to be packed, and the cover is put back in exactly the place from which it was removed, it should make a tight joint. This cannot be done with some of the modern boilers where the head is simply flanged inward and planed off to receive the cover, as the surface is too narrow to admit of it. The metal cuts into the gasket until it is practically metal to metal, with only enough rubber between them to fill the places that would otherwise be vacant, therefore, such a joint will not readily fail under pressure. Lead gaskets are sometimes adopted for this service, and they would be used much more if it was possible to put them back after cleaning and inspecting boilers, and secure tight joints with little trouble, but as a rule this is not practicable owing to the difficulty of getting them back exactly in their former positions.

LOCATING FUSIBLE PLUGS

The fusible plug in a horizontal tubular boiler should be located in the rear head, three inches above the upper tubes. As such a boiler ought to be one inch lower at the rear than at the front, in order to have all water drain to the blow-off pipe, this will leave but two inches of water above the highest part of the tubes when the fusible plug is uncovered, and this is no more than enough to insure safety.

In a locomotive boiler it should be in the highest part of the crown sheet, and in every other kind it ought to be located where it will melt and give emphatic notice of the condition of affairs, before any part of the boiler is injured through lack of water.

LEVELLING BOILERS

The upper row of tubes in a tubular boiler, should be set level crosswise of the boiler, causing the water to cover them evenly, then if the dome or the steam nozzles do not stand plumb, the defect may be counteracted by using pipe flanges that are thicker on one edge than the other.

HEAVILY LOADED BOILERS

Boilers are frequently made to develop more power in practice than their builders intended, and there are engineers who claim that it is a good idea to secure as much power as possible from a boiler, and then put it in the junk pile. One reason why this plan should not be followed is that such boilers will not be put into the junk pile soon enough, but will be used long after they become unsafe, as it is an expensive job to shut down a plant, remove old boilers and install new ones.

SMOKE CONSUMERS AND PREVENTERS

A good smoke consuming device is a valuable apparatus, but a smoke preventer is much better. A very good device for this purpose is a large furnace, and plenty of boiler power, enabling the fireman to run comparatively slow fires, and thus consume the coal properly, leaving no valuable carbon to go up the chimney. It naturally follows that a good fireman will be necessary in order to secure the full advantage of furnaces and boilers that are well proportioned for the service that they were intended to perform.

A boiler should be high enough above the grates to permit proper combustion of the coal, before the flames come into direct contact with the metal, because this contact lowers the temperature, which must be very high in order to secure good results. Under these conditions a fire may be carried 8 or 10 inches thick, and if coal is supplied frequently and in small quantities, with a suitable amount of air admitted above the fire, it is possible to run a plant without having much black smoke come out of the chimney.

A certain plant with which the writer is somewhat familiar, sends large quantities of very black smoke out of its chimneys. One reason for this is that although the boilers are rated fully as high as they ought to be, still they are caused to develop 50 per cent. more power than they were built for. This leaves the fireman no chance to manage his fires to prevent smoke by firing in front and allowing the smoke producing elements to be consumed by passing over the rest of the fire, as he must manage his fires to produce the greatest possible amount of steam without regard to good appearance or economy of fuel.

FORCED DRAFT

When forced draft is mentioned it may mean that boilers under which it is used are worked beyond their rated capacity, but it may be used to force air through a bed of very small coal, that could not be used with ordinary natural draft. In the latter case it is not wise to claim that the forced draft is ruining the boilers of a plant, because no more air passes through such a furnace, and the temperature is no higher in it than would be secured by burning large coal with good natural draft.

TEMPERATURE OF GASES

Whether a boiler is overloaded or not can usually be determined by observing the temperature of the products of combustion as they pass to the stack. If they are less than 100 degrees higher than the temperature of steam at the pressure carried, the boiler cannot be seriously overloaded, but if they are 200 or 300 degrees higher, it indicates that more power is demanded than the boiler can supply at an economical rate. This conclusion is based on the fact that if the grate and the heating surface are properly proportioned, all heat generated on the grate will be absorbed by the heating surface, except enough to cause good draft. This applies to ordinary power plants, and if an exception can be found to this general rule it denotes special conditions.

THROTTLING *VS.* AUTOMATIC ENGINES

Suppose that a certain engine, fitted with a D slide valve and a throttling governor, is running with no cut off, or in other words it takes steam constantly throughout every stroke, and the load calls for a mean effective pressure of 40 pounds. Under these conditions there will be 40 pounds pressure by the gauge at the beginning of each stroke, and this will be continued to the end. Suppose that the size of this engine and the speed at which it runs are such that it now develops 100 horse power. The boiler pressure is 80 pounds by the gauge and the reduction is made by the throttling governor with which it is fitted.

The cylinder and governor are removed, and another cylinder of the same size is substituted, but it is fitted with an automatic cut off valve gear. The speed remains the same, and as the load is not changed it requires 40 pounds mean effective pressure to drive it. With a boiler pressure of 80 pounds and the cut off taking place at ¼ stroke, the desired result is secured provided all conditions are favorable.

In the first case the cylinder was filled with steam flowing directly from the boiler through the full stroke. In the second case the flow of steam is cut off at ¼ stroke and no more enters during that stroke. Do we get the same power by taking from the boiler one-quarter of the amount taken before, thus saving 75 per cent. of the coal bill? No, decidedly not. Provided there were no leaks in the former case, the change will result in saving about 25 per cent. of fuel. Why is not a greater saving realized? The following explanation will make this clear:

When one pound of water at its greatest density is turned into steam, it makes 284.5 cubic feet at 80 pounds gauge pressure. When this steam passes the throttling governor its pressure in this case is reduced to 40 pounds. The same weight of steam now fills 475.9 cubic feet of space, therefore, it fills the cylinder nearly twice as many times as it would at 80 pounds. Furthermore, the throttling process slightly superheats the steam, thus making it more valuable than before. As the pressure remains the same throughout the stroke, there is no change of temperature to cause cylinder condensation, and this condition prevents much loss of heat.

The automatic engine takes enough steam at 80 pounds gauge pressure to fill the cylinder one-quarter full (which would fill it nearly one-half full at the lower pressure) at each stroke, and as the supply is then cut off it begins to expand, and as the pressure is lowered the temperature becomes less until the stroke is completed. As soon as the crank has passed the center at that end, the cylinder receives another charge of steam at nearly boiler pressure, but more or less of it is condensed by raising the temperature of the cylinder and this is a direct loss that is repeated at every stroke. These facts are sufficient to explain why the automatic cut off engine is not more economical than it really is in practice. A saving of 25 per cent. is sufficient to warrant its use, but it appears to the casual observer as if it should be more.

DRAINING THE STEAM PIPE

When it is nearly time to start an engine, do not forget to open the small drip valve with which every engine is, or ought to be fitted, for the purpose of allowing all water to drain out of the steam pipe. Everybody knows that this ought to be done, but when something happens to interfere with the regular routine usually observed in starting the plant, it may be forgotten, and if the throttle valve is opened when the steam pipe above it is full of cold water, something interesting may happen. It may not amount to more than a heavy, sharp pound in the pipe that an engineer does not readily forget after he has heard it once, or a gasket may be blown out of a packed joint. It is quite possible for a valve to be burst in this way, causing the plant to be shut down until a new one is put in. Some of these drip pipes are too large, therefore if the valve in one of them is opened wide it will cause water hammer in the main pipe. A one-half inch drip pipe is large enough for a

12 inch main steam pipe, then there will not be much danger of trouble on this account, even if the valve is opened wide.

WARMING THE CYLINDER.

After all water is removed from the main steam pipe, the cylinder ought to be gradually warmed by admitting steam to it, taking care not to admit enough at first to start the engine before it is time to set the machinery in motion. If the engine is fitted with a detachable valve gear, this feature can be utilized to admit steam to alternate ends of the cylinder, but as soon as it is well heated, the valve gear should be hooked in and steam enough admitted to start the engine slowly. The practice of rocking the valve gear of an engine back and forth when starting it, does not tend to free the cylinder of water, but on the contrary it closes the exhaust valves before they would be otherwise, hence water is prevented from escaping freely. Some engine builders now make their valve gears so that they cannot be unhooked, as they desire to compel the engineer to start his engine without giving it the shocks and jars so frequently resulting from the reversing process. It is claimed that fly wheels are sometimes loosened on the crank shaft by it.

RUNNING AN ENGINE "OVER"

When you write to an engine builder or a firm that was organized for the purpose of building engines of various sizes, and ask them if their horizontal engines will run "under" as well as "over" you must always expect a reply in the affirmative. There are a few engines that were made to run "over" and they cannot be reversed, but with these exceptions all others are advertised to run as well in one direction as the other.

It is a well known fact that when an engine runs "over" (or in other words, when the top of the fly wheel travels from the cylinder), the crosshead is held down on the guides, hence it is only necessary to adjust it to the proper height to bring it into line, as it is not necessary to make fine adjustments to prevent it from being lifted.

On the other hand if an engine runs "under" and is carrying a load that equals one-half of its rated capacity or more, the crosshead will be lifted at every stroke unless it is held firmly in place.

ADJUSTING THE CROSSHEAD

I once had charge of a Corliss engine that run "under" for 132 hours per week. When it was comparatively cold I would carefully adjust the crosshead between the upper and lower guides until it was a perfect fit. After it had run for an hour the vertical space between the guides near the cylinder, would be greater than it was before, hence the crosshead would lift and cause a pound at every revolution which was much worse on Saturday than it was on Monday, but the best engineer in the world could not stop it, because he could not keep the guides parallel vertically, when one end was hot and the other cool. If that fly wheel had revolved in the opposite direction, the crosshead could have been run for months without adjustment, and if there was room enough between the top of it and the upper guide to move a piece of thick writing paper, there would have been no pound, as it would still have run quietly.

SHUTTING DOWN AN ENGINE

When an engine is to be shut down for a month or two, do not simply close the throttle valve and let it stand without further attention, until it is wanted again, as rust and corrosion will damage it much more than it would be worn while in use under fair conditions for a year, and perhaps for double that time. The valve or valves should be taken out after all packing has been removed from the stems, thoroughly inspected, and if repairs are needed, now is a very good time to have them made. If these parts are in good order, cover them with cylinder oil and put them in place ready for use, but do not repack the stems until they are wanted for service.

PREVENTING CORROSION

Remove all packing from the piston rod stuffing box, clean all parts and cover them with cylinder oil, taking care to have a good coat of it between the gland and the rod. Take off the cylinder head, remove the follower bolts, the plate, and every part of the piston. Carefully inspect them for defects, and if springs are worn, studs corroded or jam nuts loose, have such defects repaired or the parts renewed while there is plenty of time for the work. Measure the cylinder and if it is worn much larger in one or more places than in others, it should be re-bored and new packing rings put into the piston. If this is not necessary, put the piston rod exactly in the center of the cylinder, replace every part of the piston ready for use, taking care to cover them all with cylinder oil at the proper time, also every square inch of the internal surface of the cylinder, in order to prevent it from rusting, and replace the cylinder head ready for use.

STARTING AN ENGINE.

When an engine is to be started after remaining idle for several months, the work necessary to be done depends on how the several parts were cared for when the machine was put out of commission. If the preceding directions concerning this important matter were intelligently followed, then it is only necessary to repack the valve stems and the piston rod, turn on the steam and start the engine.

USING A STOP MOTION.

On every Corliss engine, and on several other kinds there is or can be, some kind of a "stop motion," which is a device for preventing the engine from becoming a wreck in case the governor belt breaks or runs off from the pulleys. A certain class of engineers appreciate the value of such a device, keep it in good order and frequently test it to know that it will do the work for which it was installed. Another class seem to resent the idea that any such safeguard is necessary on a machine of which they have charge, consequently it is either removed altogether, or disabled and rendered worthless.

This is a serious mistake, because an engineer never knows when a hidden flaw will cause one of his governor pulleys to collapse, neither can he tell when the governor belt will fail or come off, thus admitting the full force of steam to the cylinder, increasing the speed of the fly wheel in a few seconds until the safe limit is exceeded, and unless steam is quickly shut off a bad wreck will result. The cautious engineer will always take advantage of every possible precaution and keep his plant ready for a test by interested parties at any time.

SPEED OF A GOVERNOR

Do not forget that the governor on your engine was built to revolve at a certain speed, which is the standard for your case. This applies to a fly ball governor without springs, or adjustable weights. If you expect to increase the speed of your engine by changing the diameter of the governor pulley, or the other one, do not state that you intend to run your governor faster in order to secure the desired result, because the speed of it will not be changed. When making calculations to determine the diameter of a pulley required to change the speed of an engine, just assume that the governor is the prime mover and that its speed does not change. This will prevent making statements that are not wise, and facilitate the actual work required by the contemplated change.

TROUBLE WITH A CRANK PIN

If the crank pin of an engine heats when lubricated with good oil, it should be thoroughly examined for defects in the pin and its boxes. If none are found the engine ought to be indicated without delay, as the sole cause of trouble may be located at the valves, for if they are not set to admit, exhaust and compress steam properly, the effect may appear in connection with the crank pin, although it is as far away as possible.

LINING AN ENGINE

Your engine may have been in line when it was erected ten years ago, but that does not prove it to be so now, as it is quite possible for it to be changed a great deal, by taking up lost motion and making other adjustments from time to time, consequently if it pounds now more than it did when new, take out the piston and remove all other parts necessary, then draw a line exactly through the center of the cylinder and see if the crosshead travels parallel to it, also if the crank shaft stands at right angles. You may be surprised at the result of your investigations.

FILING CRANK PIN BRASSES

We occasionally find an engineer who believes that every time his crank pin boxes are keyed up to take out lost motion, they must be filed so that they will come together nicely at both top and bottom, but this is not necessary. On locomotives it is advisable to do this, but it is not required on stationary work, for if they are open 1/16 inch at these points it can do no harm, and it is much more convenient to put a pair of boxes on a planer and take off enough to answer for a year or two, than to file it off every time that lost motion must be taken up.

AVERAGE PRESSURE OF STEAM

The average pressure of steam acting on the piston of an engine is never used when calculating the indicated horse power of it, although it is so stated in some of the rules given for this purpose. The average pressure is always taken above a perfect vacuum and the back pressure can have no effect on it, because the two pressures are acting on opposite sides of the piston. The average pressure shows the total force acting on the face of a piston to propel it forward, but gives no clue whatever to the division of the opposing force. When calculating this pressure it makes no difference whether a condenser is used or not, as only the initial pressure and point of cut off (including clearance), are taken into consideration. Of course, there are times when we want to know how much back pressure opposes the advance of an engine piston, also how much of it is above atmospheric pressure and other particulars, but this only occurs when we wish to determine the mean effective pressure, and when this is known it is used in calculating the indicated horse power developed by an engine.

KEEPING AN ENGINE CLEAN

When a certain engine is correctly adjusted and properly lubricated, it will give good results, whether the cylinder head is kept bright or allowed to become dingy and dirty. There is, however, a connection between the outward appearance and the inward condition of an engine, that is well understood by engineers and steam users, for as a general rule to which there may be a very few exceptions, when an engine is rusty and dirty on the outside, the inside of it will not bear careful inspection. The outward surfaces can be kept clean and bright when the machinery is in operation, consequently if a man will not attend to this during working hours, he will probably not spend any more time taking care of the internal parts, while the machinery is shut down and other employes are enjoying themselves, than will barely keep them in operation.

THE STRIKING POINTS

To ascertain what may be called the "striking points" of an engine, disconnect the connecting rod from the crosshead, move the piston as far as it will go in one direction and put a mark on the guides at one end of the crosshead. Now move the piston as far as it will go in the opposite direction and mark the guide in the same way. This reads very smoothly, and it will work in the same way, provided you take the packing rings out before the piston is moved further than it moves in regular service. If you fail to do this and one of them is caught in a port, it may cause more or less trouble to get it out. When the connecting rod and the packing rings have been put in again, place the crank on each center alternately and mark the extreme travel of the crosshead as before. The difference between these two marks at each end represents what is sometimes called the clearance. This is correct so far as the travel of the piston is concerned, but it does not include the waste space in ports, etc. An engine may be so designed that it will wear unevenly, hence the piston may be drawn gradually towards one of the cylinder heads, as lost motion is taken up, but so long as the crosshead does not reach these marks on the guides, known as the "striking points," there is no danger of the piston striking either cylinder head.

SMALL REPAIR JOBS

Do not attempt to do a job of piping, or a small repair job on the engine during the noon hour, as it will probably take longer than you expect, hence when it is time to start the machinery for the afternoon you will not be ready, the other employes will be unable to work, and you will heartily wish that you had postponed it until evening, when there was plenty of time to do it without interfering with the regular work.

TEMPERATURE OF STEAM AT VARIOUS PRESSURES ABOVE THAT OF THE ATMOSPHERE (14.7 lbs.)

Pounds Pressure	Degrees Fahrenheit	Pounds Pressure	Degrees Fahrenheit	Pounds Pressure	Degrees Fahrenheit
0	212	18	254.5	100	337.5
1	215.5	19	256	105	341
2	219	20	257.5	115	347
3	222	25	265	125	353
4	225	30	272.5	135	358
5	227.5	35	279.5	145	363
6	230	40	285.5	155	368
7	232.5	45	291	165	373
8	235	50	297	175	377
9	237.5	55	302	185	381
10	240	60	307	235	401
11	242	65	311	285	417
12	244	70	315	335	430
13	246	75	320	385	445
14	248	80	323	435	456
15	250	85	327	485	467
16	252	90	331	585	487
17	253.5	95	334	685	504

ALPHABETICAL LIST OF TABLES

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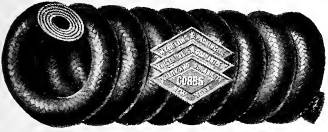
COBBS

High Pressure Spiral Piston and Valve Rod Packing

Suitable for Steam, Gas and Air Pressure

100 POUNDS AND OVER



Spiral-Style 2

IT is made with a rubber core, of an oil and heat resisting compound which is covered with a well frictioned duck and an outer covering of fine asbestos. It is guaranteed not to become hard, or score the rods.

It has proven exceptionally serviceable on the piston rods and valve stems of all kinds of engines, pumps, gas and air compressors, steam hammers, etc.

"Cobbs" High Pressure Spiral Packing is made both square and round, and is carried in stock in all sizes from 1/8 in. to 1 1/4 in.

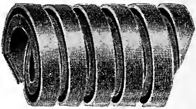
Price per lb., $1.75

Write for Discount

NEW YORK BELTING & PACKING CO.

VULCAN

High Pressure Spiral Piston and Valve Rod Packing



Style 112

IS made of a very fine quality of duck, well frictioned, with the thread end against the rod, making the best kind of wearing surface. It also has a rubber cushion which, when heated, will expand and force the duck against the rod, which has a tendency to steady it if it is a vibrating rod. The cushion is protected by a layer of duck which prevents it from sticking to the stuffing box.

"Vulcan" High Pressure Packing is particularly serviceable on direct connected dynamo engines, high speed engines, rock drills, etc., particularly where there are vibrating rods.

Carried in stock in all sizes from ¼ in. to 1 in.

Price per lb., $1.75

Write for Discount

NEW YORK BELTING & PACKING CO.

MAGIC

Rubber Cushion Diagonal Expansion Spiral Packing

Style 5

IS especially adapted for service where the rod is scored. In cases where it is necessary to screw down tight on the glands, "Magic" Packing gives splendid results, set thumb tight. The wedge side has a soft rubber back which, when heated, expands, in some cases, sufficiently to compensate for worn rods. The flax is braided and interwoven with a soft, frictionless metal wire, binding it together so there is no possible chance for the rod to pick up strands of the flax to get into cylinders and pass from there into the check valves. Magic Packing is filled with the best cold lubrication that can be procured. It gives best service for ammonia, steam, boiling water plungers, pistons, etc.

Carried in stock in all sizes from ¼ in. to 1 in.; larger sizes made to order.

Price per lb., $1.50

Write for Discount

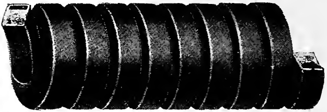
NEW YORK BELTING & PACKING CO.

B. & P.

H. P. Gum Cushion Asbestos Spiral Piston and Valve Rod Packing

Suitable for Steam, Gas and Air Pressure

150 POUNDS AND OVER



Style 1003

OUR High Pressure Cushion Asbestos Spiral and Ring Packings are made of the best quality Asbestos, with a soft red cushion. There is a growing demand for a Packing to withstand extremely high pressures and superheat, which can be supplied with our High Pressure Cushion Asbestos, which is made in Spiral Rod, Coil and Rings and carried in stock in sizes from ¼ in. to 1¼ in.

Price per lb., $1.75

Write for Discount

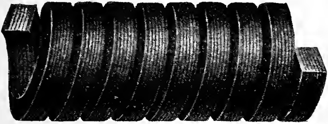
NEW YORK BELTING & PACKING CO.

GRAPHON

Low Pressure Spiral Piston and Valve Rod Packing

Suitable for Steam Pressure

LESS THAN 100 POUNDS



Style 1

WE make for Steam, Ammonia, Hot or Cold Water, in Spiral, Plain and Sectional Cup Rings. We realize the necessity of lubricating this style of Packing for the particular uses to which it will be applied and after years of study and experiment, we have discovered a process of lubrication for steam and ammonia, which gives our Graphon Packing the enviable position of second to none.

Carried in stock in sizes from ⅛ in. to 1¼ in.

Price per lb., $1.50

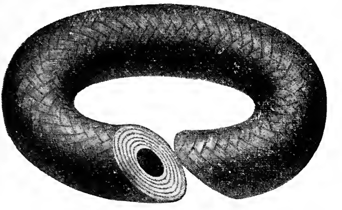
Write for Discount

NEW YORK BELTING & PACKING CO.

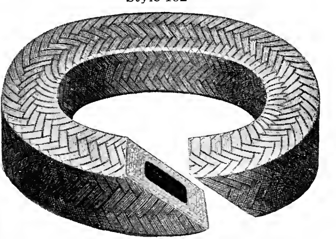
COBBS High Pressure Ring Piston and Valve Rod Packing

Suitable for Steam, Gas and Air Pressure

100 POUNDS AND OVER



Cobbs High Pressure Round Ring Packing
Style 102



Cobbs High Pressure Square Ring Packing
Style 203

"COBBS" Piston and Valve Stem Packing is the outcome of years of experiments by its inventor, Mr. J. H. Cobb, who has been with the New York Belting & Packing Company, for many years.

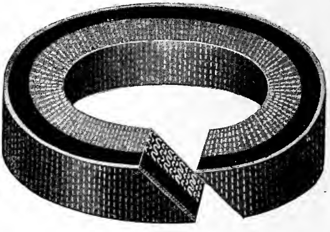
See page 385 for Cobbs Spiral.

Price per lb., $2.00 Write for Discount

NEW YORK BELTING & PACKING CO.

VULCAN

High Pressure Square Ring Packing



Style 212

THIS packing is manufactured of the same material and for the same purposes as "Vulcan" High Pressure Spiral Packing.

When ordering give diameter of rod and inside diameter of stuffing box.

See page 386 for Spiral style.

Price per lb., $2.00

Write for Discount

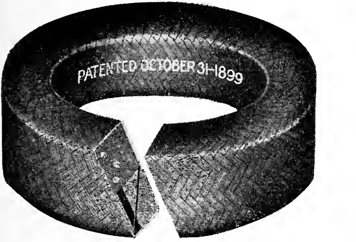
NEW YORK BELTING & PACKING CO.

MAGIC

Diagonal Expansion Ring Packing

Suitable for Hot Water

100 TO 212 DEGREES



Style 105

FOR Piston Rods and Boiler Feed and *outside packed* Plunger Pumps, where water is pumped from wells, traps, heaters.

This packing is manufactured of the same material and for the same purposes as "Magic" Spiral Packing.

When ordering give diameter of rod and inside diameter of stuffing box, also state whether to be used for water, steam or ammonia.

See page 387 for Magic Wedge Spiral.

Price per lb., $2.00

Write for Discount

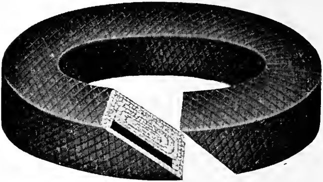
NEW YORK BELTING & PACKING CO.

B. & P.

High Pressure Cushion Asbestos Ring Piston and Valve Rod Packing

Suitable for Steam, Gas and Air Pressure

150 POUNDS AND OVER



Style 1202

TO-DAY, the tendency is toward extremely high pressure and superheat. To overcome these severe conditions, we offer our B. & P. High Pressure Asbestos Ring Packing.

See page 388 for H. P. Asbestos Spiral.

Price per lb., $2.00

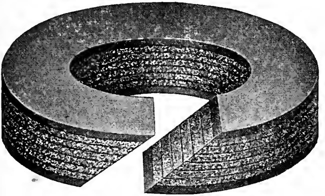
Write for Discount

NEW YORK BELTING & PACKING CO.

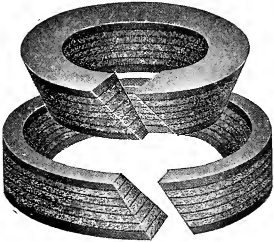
GRAPHON Low Pressure Plain and Sectional Rings

Suitable for Steam Pressure

LESS THAN 100 POUNDS



Graphon Plain Ring Packing Style 101



Graphon Sectional Ring Packing Style 201

In sending sizes of rods and stuffing boxes, please state if possible, the class of work for which they are intended, as we make considerable difference in the lubrication when used for Steam and Ammonia.

See page 389 for Graphon Spiral.

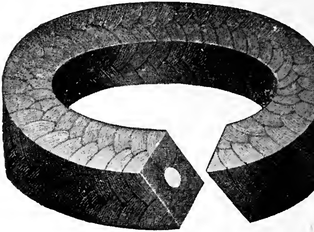
Price per lb., $2.00 Write for Discount

NEW YORK BELTING & PACKING CO.

HIGH PRESSURE
Waterproof Hydraulic Ring Packing

Suitable for Cold Water Pressure

UP TO 3,000 POUNDS



Style 206

HIGH PRESSURE Waterproof Hydraulic Packing is made from long fibre flax and treated with our special waterproof compound, which makes it impervious to water, and work with little or no friction. These Rings are made for Cold Water Pistons, Plungers on Cold Water Pumps, Elevator Cylinders, Hydraulic Presses, Stern Glands and special hydraulic work.

See page 390 for High Pressure Waterproof Hydraulic Spiral.

Price per lb., $2.00

Write for Discount

NEW YORK BELTING & PACKING CO.

COBBS Throttle Packing

Suitable for Steam Pressure

UP TO 250 POUNDS

For use on Locomotive Throttles.

(In sending orders always specify O. D. of the rod, I. D. of the stuffing box, and the depth of the box. There is no exact standard for the sizes of these Throttle Packing Sets as there is so much variation in the different locomotives.)

Style 1820

Does Not Get Hard
Cannot Wear the Rod

IT has a rubber core and a woven asbestos cover. It does not become hard in service as do other throttle packings, but remains soft and pliable, consequently it is very durable.

There is nothing made that gives such good service.

This packing is put up in sets as shown in cut and we strongly recommend that it be purchased in this form. We can, however, supply this packing in spiral coils if desired, put up in boxes of weights and lengths as shown on page 385.

Price per set, $2.00

Write for Discount

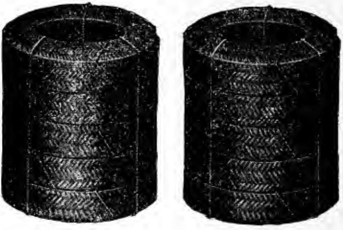
NEW YORK BELTING & PACKING CO.

COBBS Air Pump Packing
Suitable for Steam and Air Pressure
UP TO 250 POUNDS

For use on New York and Westinghouse Air Pumps and Air Compressors.

(Put up in double boxes and exact fitting sets.)

Note that this packing is put up in "double sets," each tied up in rings and packed in boxes. This is sufficient Packing for both ends of "Westinghouse" Air Pump, although the "N. Y. Duplex" Pumps require two (2) double sets.



Style 1830

Does Not Get Hard **Cannot Wear the Rod**

Sizes of Air Pump Packing are as follows:

WESTINGHOUSE AIR BRAKE PUMP

		Ins. Diam.		Out. Diam.
8 in. depth 4 rings	..	$1\frac{1}{4}$ in.	..	2 in.
$9\frac{1}{2}$ in. " 5 "	..	$1\frac{7}{16}$ in.	..	$2\frac{1}{8}$ in.
11 in. " 5 "	..	$1\frac{3}{4}$ in.	..	$2\frac{5}{8}$ in.

NEW YORK AIR BRAKE PUMP

No. 2 Duplex

Depth 4 rings. Ins. Diam. $1\frac{1}{4}$ in. Out. Diam. 2 in.

No. 5 Duplex

Depth 5 rings. Ins. Diam. $1\frac{1}{2}$ in. Out. Diam. $2\frac{1}{4}$ in.

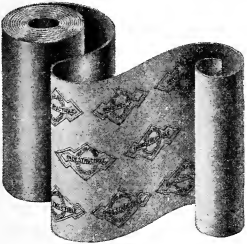
It has a rubber core and a woven asbestos cover. It does not become hard in service as do other air pump packings, but remains soft and pliable, consequently it is very durable.

There is nothing made that gives such good service.

Price per set, $2.00 Write for Discount

NEW YORK BELTING & PACKING CO.

INDESTRUCTIBLE White Sheet Packing



Style 10

THIS is a Sheet Packing that will stand very high pressure and extreme heat. It is not affected by ammonia, liquor, or alkali, making a tight joint when used for steam, air, hot or cold water. It does not dry out or become hard and brittle, but retains its tough elastic qualities either in or out of service. It will not blow out under any pressure, and the joint can be broken numerous times without renewing the packing.
Carried i stock in sizes 1/32 in. to 1/4 in.

Price per lb., $1.00

Write for Discount

Indestructible White Sheet Packing

With Brass Wire Insertion

Style 29

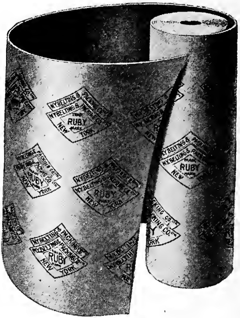
For marine work, vibrating joints, or where there are but a few bolts in flange, or where the ports run close to the edge, and hard places to hold in general.
Made 1/16 and 1/8 in. thick.

Price per lb., $1.25

Write for Discount

NEW YORK BELTING & PACKING CO.

RUBY Red Sheet Packing



Style 11

THIS packing is made of a soft tenacious compound of red rubber that conforms to the inequalities of flanges and will not blow out under the highest pressures. In the hottest joint it remains soft and flexible, and can be removed and used again and again in the same joint, packing it as perfectly as when first inserted.

It also makes a perfect packing for hydraulic and cold joints of all kinds. It perfectly withstands the action of steam, alkalies, ammonia, etc.

The application of graphite to the face of the packing before use prevents adhesion to flanges and makes it easier to break the joint after use. Carried in stock from $\frac{1}{32}$ in. to $\frac{1}{4}$ in. and 36 in. wide.

Price per lb., $1.00 Write for Discount

Ruby Sheet Packing

With Brass Wire Insertion

Carried in stock $\frac{1}{16}$ in. and $\frac{1}{8}$ in.—36 **in. wide.**

Price per lb., $1.25 Write for Discount

NEW YORK BELTING & PACKING CO.

KARBONITE Black Sheet Packing

Style 12

THIS packing is made from the highest grade of rubber combined with the very best ingredients for the purpose, producing a Packing that does not vulcanize. It will not harden when subjected to extreme heat and will not blow out under the highest pressure. The action of oils, alkalies, ammonia, etc. is resisted better by "**Karbonite**" than by any other known rubber compound. It will take up the contraction of metal when line cools off, and prevents joints from leaking. Nothing produced will last as long or give such good service under any and all conditions.

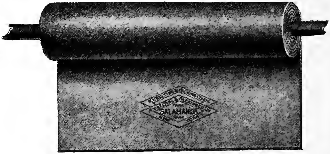
Carried in stock from $\frac{1}{32}$ in. to $\frac{1}{8}$ in.

Price per lb., $1.25

Write for Discount

NEW YORK BELTING & PACKING CO.

SALAMANDA Sheet Packing



Style 13

A HIGH-GRADE self-vulcanizing packing. Because of its semi-cured condition it readily conforms to rough surfaces, and when subjected to heat hardens, making a secure, permanently packed joint.

It can be used between unfinished cast iron flanges, saving the cost of turning and planing.

Carried in stock in thicknesses from $\frac{1}{32}$ in. upward—36 in. wide. Wire insertion made to order.

Price per lb., 65 cents

Write for Discount

NEW YORK BELTING & PACKING CO.

FIRO Superheat Sheet High Pressure Steam Packing

Style 1846

Has every qualification the name implies, and for Gas Engines, Automobiles, Motor Boats, Steel Plants, Turbines; in fact any place where extraordinary high pressures and intense heat exist, Firo has no equal. Firo will never vulcanize, cannot burn or char in a joint, will not flatten out between the flanges when the bolts are drawn up and makes a positively permanent joint, insuring economy to the plant and safety to the workmen. It is absolutely non-absorbent. Firo is made in sheets 40 in. x 40 in., 40 in. x 80 in. and 40 in. x 120 in., in all thicknesses from 1/32 in. up.

A sheet 40 in. x 40 in. will weigh as follows:

1/32 in. thick	3 1/4 pounds,
1/16 in. "	6 1/2 "
1/8 in. "	13 "

Price per lb., $2.00 Write for Discount

ASBESTOS Metallic Sheet Packing

Style 410

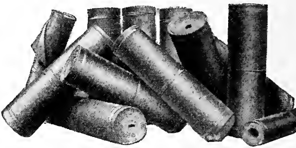
Is made of the pure Asbestos Yarn, each strand having two brass wires interwoven, which give it strength. It is made 40 in. wide in all thicknesses from 1/32 in. up and for all purposes where the usual fibrous and rubber sheet packings would be destroyed by intense heat.

Price per lb., $1.50 Write for Discount

NEW YORK BELTING & PACKING CO.

CLOTH INSERTION

Sheet Packing



FOR steam, hot or cold water, and weather strips.

We make three grades of superior quality of this packing in the following brands:

"Double Diamond"	Style 33
"Carbon"	" 34
"Spider"	" 14

In thicknesses of from $\frac{1}{32}$ in. to $\frac{1}{4}$ in. 1-ply of duck to each $\frac{1}{16}$ in. in thickness

Write for price.

WIRE INSERTION

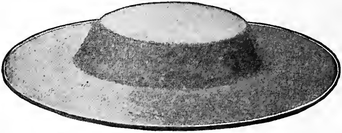
Sheet Packing

Chiefly used in joints where C. I. Packing would be quickly burned out by heat or blown out by excessive pressure.

STEAM REGULATOR DIAPHRAGMS

ON account of the severe work they are subjected to, we make them of a high grade rubber, most suitable for the purpose.

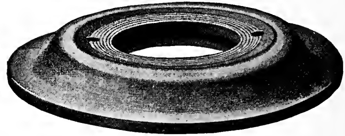
Below, we show cuts and give sizes in which they are made.



High and low pressure. Made to withstand the most severe service.

No. 1,	$6\frac{3}{4}$ in. outside Diam., each ..	$ .55
No. 2,	9 " " " " ..	.85
No. 3	$11\frac{5}{8}$ " " " " ..	1.45
No. 4,	15 " " " " ..	2.65

PUMP DIAPHRAGMS



For "Edson" and "Loud" pumps. These are high grade in every sense of the word.

No. 0 Edson, each	.. ..	$1.50
No. 2 " "	.. ..	2.00
No. 3 " "	.. ..	3.00
No. 4 " "	.. ..	4.00
No. 1 Loud, "	.. ..	2.00
No. 2 " "	.. ..	3.00

Write for Discounts

NEW YORK BELTING & PACKING CO.

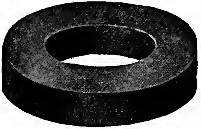
PUMP VALVES

THE proper working of a Pump depends mostly on the Valves. It is necessary that these should be of the very best material and of a density suitable for the work the Pump must perform.

Our experience in manufacturing Valves for Marine, Blower Engines, Acid, Mining, Boiler Feed and other Pumps, enables us to furnish the proper Valves and puts us in a position to make any special Valve you may desire. It is essential that you state explicitly the conditions under which the Valves are to be used, the size of the hole, the thickness and width of the Valves; also working pressure.

NEW YORK BELTING & PACKING CO.

HARD RUBBER VALVE DISCS



New Style

Old Style

For all Makes of Valves

MANUFACTURED of hard rubber composition for the highest steam pressures, also ammonia, oils, acids, etc. Will not soften or wear uneven, thus preventing leaky valves; a rubber composition that has been tested by many years of actual service and warranted to give satisfaction.

Also furnished in soft composition for cold water, low pressures if desired.

In ordering state whether New Style (oblong hole) or Old Style (round hole) is wanted.

Carried in stock in all sizes as listed.

List Prices

Size	Price Each	Size	Price Each
¼ inch	$.03	1¼ inch	$.09
⅜ "	.04	1½ "	.12
½ "	.04	2 "	.18
¾ "	.05	2½ "	.24
1 "	.06	3 "	.33

NEW YORK BELTING & PACKING CO.