"VERBAL" NOTES AND SKETCHES FOR MARINE ENGINEERS

J. W. M. SOTHERN
M.I.E.S.

25/- NET.

NINTH EDITION

17^{50}

A. E. Moncaster

SOTHERN'S MARINE ENGINEERING COLLEGE
GLASGOW

"VERBAL" NOTES AND SKETCHES FOR MARINE ENGINEERS

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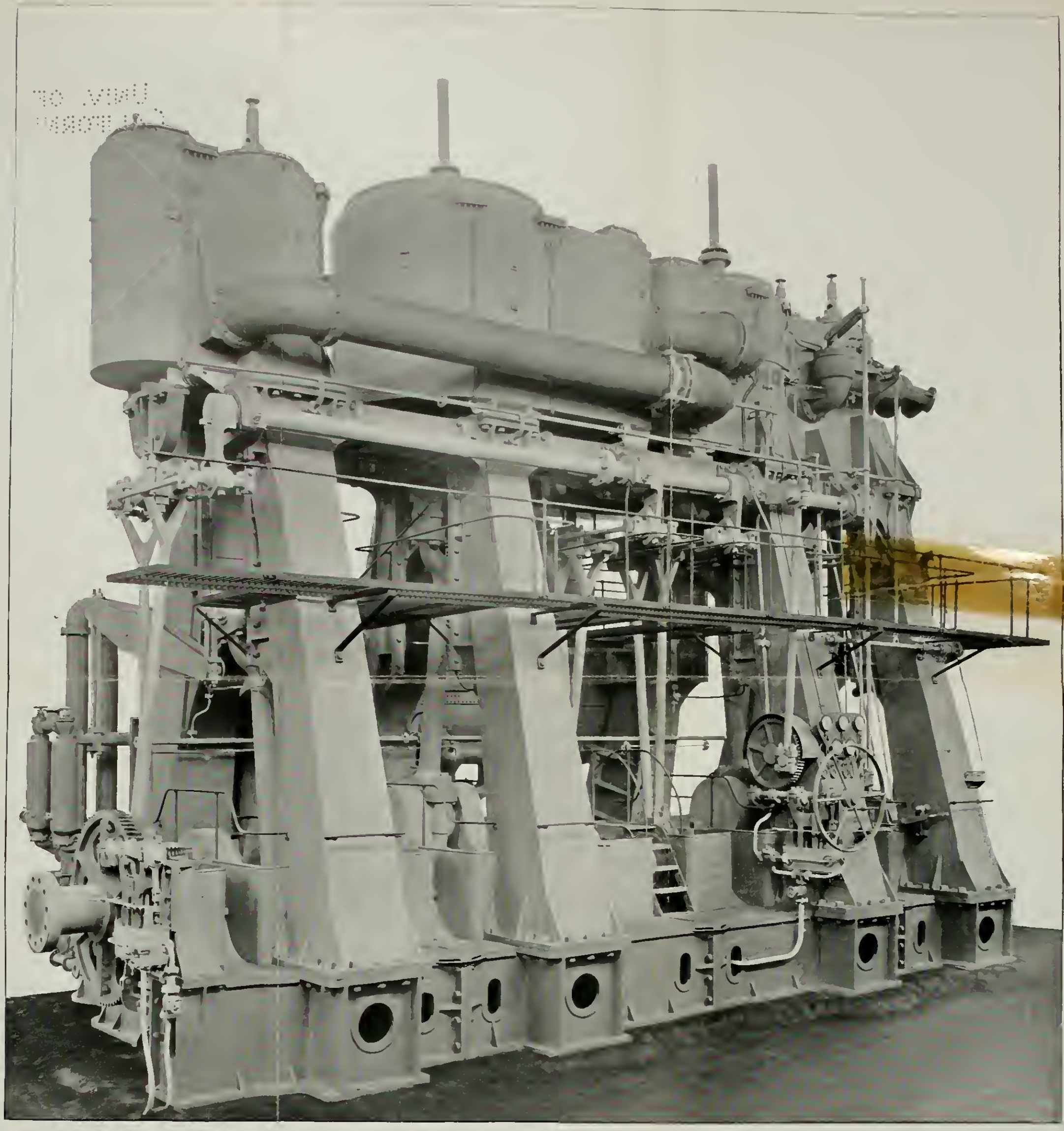


Quadruple Expansion Engines of SS. "Oosterdyk" and "Westerdyk."

Constructed by Messrs Richardson, Westgarth & Co. Ltd., Middlesbrough.

DATA.

Gross tonnage.—8251.

Dimensions of Vessel.—450 feet by 55 feet by 41 feet $7\frac{1}{2}$ inches.

Boilers.—Six single-ended, 15 feet 3 inches by 10 feet 6 inches.
Natural draught.
Total heating surface, 13200 square feet.
Total grate surface, 359 square feet.
Pressure, 215 lbs. per square inch.

Engines.—Cylinders, $27\frac{1}{2}$, $37\frac{3}{4}$, 55, 84 inches.
Stroke, 60 inches.
Cut-off, 70 per cent.

Type of Valve Gear. Stephenson's link motion.

Travel of Valves. 8 inches (L.P. $8\frac{1}{2}$ inches).

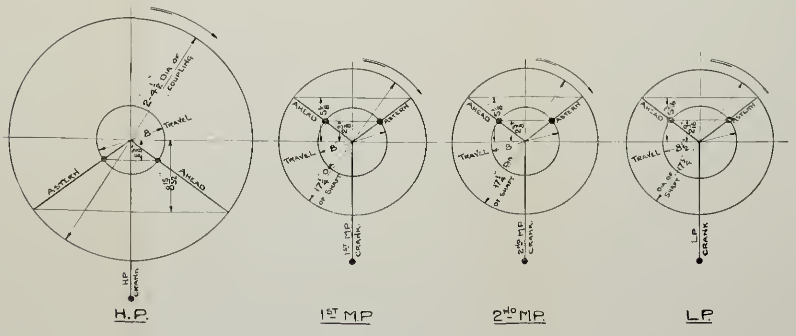
Type of Valves.—H.P. piston valve with inside steam.
First and second M.P. balanced slide valves (Martin & Andrews type).
L.P. double-ported slide valve.

Position of Eccentric Rods with Crank on Bottom Centre.
H.P. rods crossed.
1st I.P. ,, open.
2nd I.P. ,, ,,
L.P. ,, ,,

Total I.H.P.—4800.

Propeller.—Solid manganese bronze.
Diameter, 19 feet 6 inches.
Pitch, 20 feet.
Expanded area, 4 blades, 116 square feet.
Projected area, 4 blades, 99 square feet.

Speed.—$12\frac{1}{2}$ to 13 knots at sea.



Positions of Eccentric Keyseats.

(All cranks on bottom centre.)

NOTE. For H.P., 1st M.P., and 2nd M.P. cylinders, the mean steam lap + lead = $2\frac{3}{8}$, and for the L.P. cylinder the mean steam lap + lead = $2\frac{1}{16}$".

H.P. eccentric keyseat follows crank at an angle of 90 − (steam lap + lead) angle.
1st M.P. ,, ,, leads ,, ,, ,, 90 + (steam lap + lead) angle.
2nd I.P. ,, ,, ,, ,, ,, ,, 90 + (steam lap + lead) angle.
3rd I.P. ,, ,, ,, ,, ,, ,, 90 + (steam lap + lead) angle.

NOTE. The H.P. engine eccentrics are secured to the shaft coupling in place of the shaft.

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A MANUAL OF MARINE ENGINEERING PRACTICE

INTENDED FOR THE USE OF NAVAL AND MERCANTILE MARINE ENGINEERS OF ALL GRADES, AND STUDENTS, FOREMEN ENGINEERS, ETC., AND IS SPECIALLY COMPILED FOR THE USE OF ENGINEERS PREPARING FOR EXAMINATIONS OF COMPETENCY AT HOME OR ABROAD

BY

J. W. M. SOTHERN

Member, Institute of Engineers and Shipbuilders in Scotland; Hon. Member, Glasgow and West of Scotland Association of Foremen Engineers and Draughtsmen.

Author of "Marine Indicator Cards," "The Marine Steam Turbine," "Simple Problems in Marine Engineering Design," "Elementary Mathematics for Marine Engineers," etc.

Principal, Sothern's College of Marine Engineering, Glasgow.

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60 BROWN STREET

1917

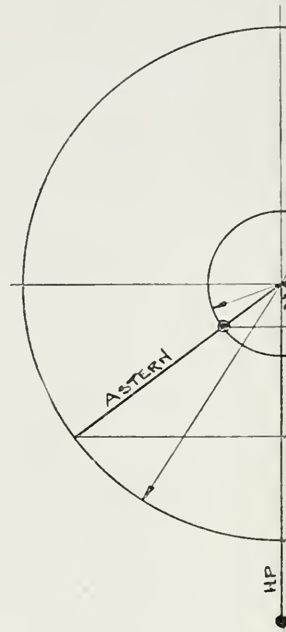
Engines.—Cylinders, 27

Stroke, 60 inc

Cut-off, 70 pe

Type of Valve Gear.—St

Travel of Valves.—8 incl



H.

NOTE.—For H.P., 1st

NOT

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Gift of Arthur E Moncaster

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Fourth Re-issue (with additions) published September 1916

Ninth Edition published October 1917

PREFACE TO NINTH EDITION.

THE present volume, it may be pointed out, contains much new matter and many new illustrations, in addition to which the work, as a whole, has been carefully revised and corrected. Important additions have been made to the sections treating of Diesel Oil Engines, Marine Steam Turbines, and Electric Light, and to the other sections generally and it is hoped that the book will be found to be even more useful than before as a general reference volume for marine engineers of all grades. It may also be pointed out that the work now contains a full set of Board of Trade First Class Examination Drawings with dimensions, and a set of forty Board of Trade Second Class Examination Drawings, also fully dimensioned, for the use of second class students under the new and revised Regulations (1917). The author desires to express his pleasure in the knowledge that the book is steadily rising in popularity yearly, and that the increasing sales now extend to all parts of the world, and it may interest readers to know that the volume enjoys an extensive circulation even in far Japan. "Verbal" Notes is also in general use among the engineering *personnel* of the Royal Navy, and in short, appears to be the most popular technical work on marine engineering published.

The author specially desires to thank A. N. Somerscales, Esq., Hull, for most kindly revising and correcting the work, also for many useful suggestions towards improvement in the text, etc.; also R. M. Sothern, Esq., for preparing many of the drawings and checking the various calculations. The author's thanks are also due to the editor and proprietor of the *Mechanical World* for kind permission to reproduce a number of the illustrations from the pages of that journal; to the proprietors of *Engineering* for illustrations and accompanying descriptions of Diesel Engine sets; to Messrs the British-Thomson-Houston Co. Ltd., Rugby, for kind permission to reproduce illustrations and text relative to Curtis turbines; and specially to Messrs Richardson, Westgarth & Co. Ltd., Hartlepool, for the handsome illustration and the data which form the frontispiece of the volume; also to W. B. Hird, Esq., of Messrs Mavor & Coulson, electrical engineers, Glasgow, for kind permission to reproduce an article on Electric Motor Practice. Finally, and as on former occasions, the author has again to thank numerous friends and students for much valuable data taken from actual practice.

MARINE ENGINEERING COLLEGE,
59 BRIDGE STREET, GLASGOW, *October* 1917.

PREFACE TO EIGHTH EDITION.

In the New Appendix which is included in this Edition the author has introduced a number of new drawings, notes, and calculations, referring chiefly to Diesel Oil Engines, Boilers, and Marine Turbines, the latter including exhaust turbines and geared down turbines, thus bringing the work right up to date.

The author's thanks are due to A. P. Chalkley, Esq., for kind permission to reproduce detail drawings of the Diesel type oil engine from "Diesel Engines," by A. P. Chalkley, B.Sc., A.M.I.C.E.; to the Council of the Institution of Naval Architects for permission to reproduce the illustration of the "Vespasian" gear wheels from a Paper read before that Institution by Sir Charles Parsons, and entitled "The Application of the Marine Steam Turbine and Mechanical Gearing to Merchant Ships"; to the editors of *Engineering* for illustrations of the turbine machinery of Q.SS. "Reina Victoria Eugenia," and SS. "King Orry," and for permission to reprint the descriptions of the machinery which appeared originally in *Engineering*; also for permission to reproduce the illustrations showing the Diesel four-cycle and two-cycle action, from a Paper entitled "The Diesel Oil Engine," by Dr Rudolph Diesel of Munich, and read before the Institution of Mechanical Engineers. The author has also to thank Messrs G. & J. Weir for the illustration of their patent "Dual" type Air Pumps, etc.

MARINE ENGINEERING COLLEGE,
59 BRIDGE STREET, GLASGOW, *September* 1913.

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2. Double-Ended Combustion Chamber.
3. Furnace and Fire Bars.
4. Water Gauge Column.
5. Vertical Donkey Boiler.
6. Fire Bars and Bearers for Vertical Boiler.

Valves—

7. Dead Weight Safety Valve.
8. Spring-Loaded Safety Valve.
9. Boiler Stop Valve.
10. Engine Room Stop Valve.
11. Feed Check Valve.
12. Bilge Suction Valve Chest.

12A. Bilge Injection Valve.

13. Side Discharge Valve.
14. Cylinder Relief Valve.
15. Slide Valve and Spindle.
16. Inside Steam Piston Valve.
17. Double Ported Slide Valve.

Pumps—

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19. Feed Pump Complete.

19A. Feed Relief Air Vessel and Pump Valves.

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20. H.P. Piston and Rod.
21. L.P. Piston and Rod.
22. L.P. Cylinder Cover.
23. Donkey Pump Cylinder and Valve.

Eccentric, etc.—

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Various—

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34. Main Bearing.

34A. Tunnel Bearing Block.

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DESIGN DRAWINGS AND CALCULATIONS.

SECOND CLASS.

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2. Bottom End Bolt, complete.
3. Tapered Coupling Bolt, complete.
4. Distance Piece for Bottom End Brass.
5. Main Bearing "Brass."
6. Bottom End Brass.
7. Steam Stop Valve and Seat.
8. Safety Valve, Seat, and Spindle.
9. Winch Slide Valve.
10. Slide Valve Spindle.
11. Eccentric Pulley.
12. Eccenric Strap.
13. Crank Shaft and Webs.
14. Thrust Shaft.
15. Tail End Shaft.
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25. Spanner.
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28. Fire Bar Bearer.
29. Stay Tube and Common Tube.
30. Tube Stopper.
31. Combustion Chamber Girder.
32. Double Butt Strap Joint.
33. Main Stay.
34. Combustion Chamber stay.
35. Double Riveted Lap Joint.
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38. Hand Wheel for Steam Stop Valve.
39. Air Vessels.
40. Water Gauge Column Bracket.

NOTE.—It is of the utmost importance that each drawing be fully dimensioned before being handed in to the Examiner.

"VERBAL" NOTES AND SKETCHES

SECTION I.

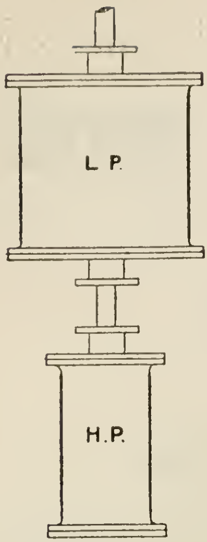
WORKSHOP PRACTICE.

THE reciprocating engine as at present constructed and perfected by the numerous auxiliary specialities now in general use undoubtedly represents vast improvement on the engine of ten or fifteen years ago, and part of this improvement is certainly due to the superior class and make of the machine tools now in general use, and to the ever increasing use of cast steel and mild steel, which materials serve to combine strength with lightness of parts. A brief description of the various types of reciprocating engines found in ordinary marine practice will now be given.

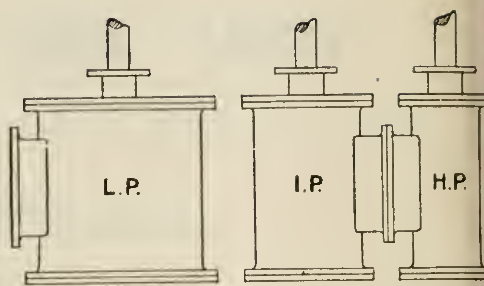
Types of Engines.

Paddle Engines.—For steamers of the paddle type the general practice is to fit an engine of the diagonal pattern (the oscillating type being now nearly obsolete) arranged either single, compound, triple, or quadruple expansion, and generally with two cranks, although three cranks are occasionally arranged for. The Sketches Nos. 1, 2, 3, 4 illustrate the various cylinder and crank arrangements referred to. Piston valves are often fitted to the H.P. and I.P. of triple expansion paddle engines, and flat double-ported slide valves to the L.P. cylinders.

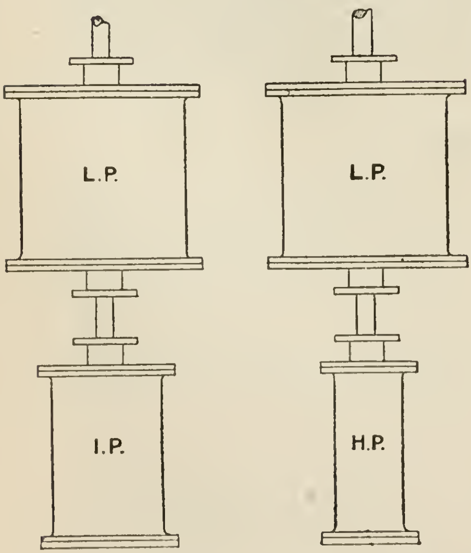
Screw Engines.—In screw steamers the engines are of the inverted type, being either compound, triple, or quadruple expansion, but in Naval practice the turbine has now completely superseded the reciprocating engine for all classes of vessels, and the success of this type of engine, where high power and speeds are required, is beyond dispute. Many cross-channel steamers and deep sea passenger steamers are also fitted with turbine machinery of the Parsons design,



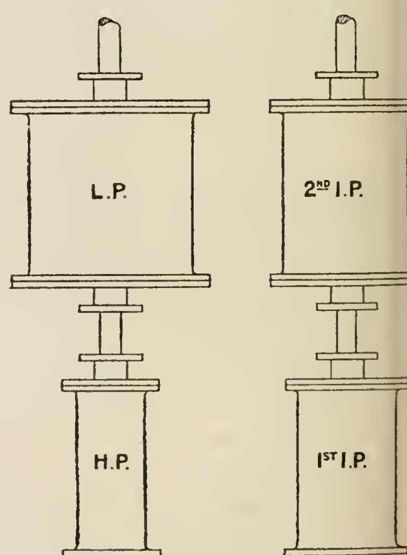
No. 1.—Compound Type Paddle Engines.



No. 2.—Triple Expansion Type Paddle Engines.



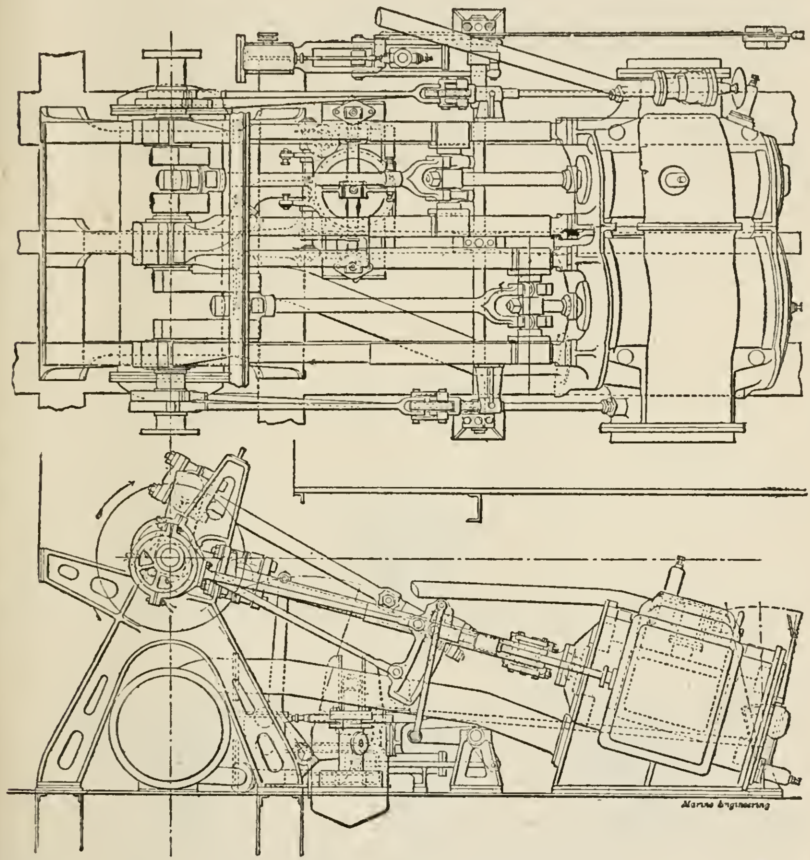
No. 3.—Triple Expansion Type Paddle Engines (Two L.P. Cylinders).



No. 4.—Quadruple Type Paddle Engines.

and the combination arrangement, in which reciprocating engines and turbines are arranged to work conjointly in the same engine-room, is rapidly coming forward into more general practice. (For further information on this subject see author's "Marine Steam Turbine.")

The Sketches numbered 6, 7, 8, 9 illustrate the various cylinder



No. 5.—Diagonal Type Paddle Engine.

and crank arrangements mentioned above, and the flow of the steam through each is as follows :—

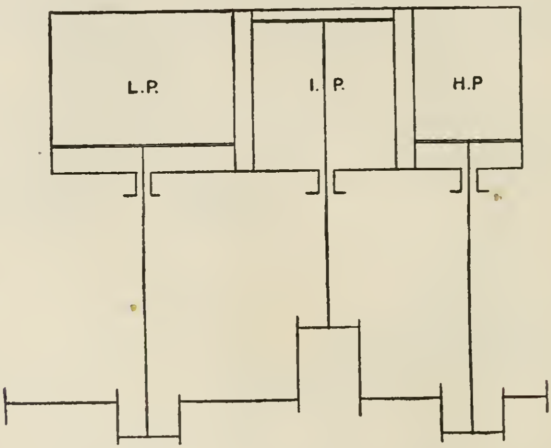
Compound.—Steam flows from boilers through H.P. then L.P. to condenser.

Triple.—Steam flows from boilers through H.P., I.P., and L.P. to condenser.

Triple (with 2 L.P. cylinders and four cranks).—Steam flows from the boilers through H.P., I.P., and then divides into two steam pipes, one led to each L.P. cylinder.

Quadruple.—Steam flows from the boilers through H.P., 1st I.P., 2nd I.P., and L.P. to condenser.

In many designs of large power engines the steam is conveyed from one cylinder to another by means of large pipes, known as "receiver pipes," but in ordinary engines of moderate power the steam flows from one receiver to another through large ports cast in the cylinders themselves. In Sketch No. 9 the cylinders are shown arranged, from forward aft, as H.P., 1st I.P., L.P., and 2nd I.P., which allows of better balancing of the working parts, the crank angles being a few degrees less or more than 90° to each other.



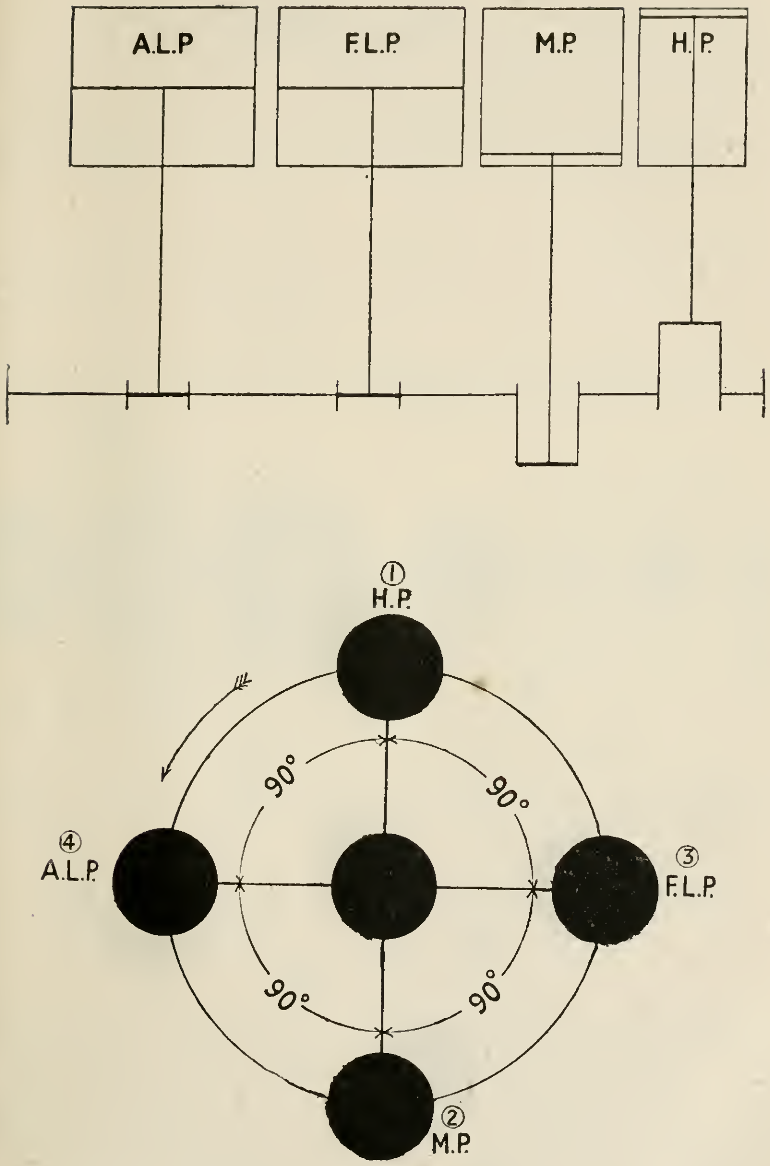
No. 6.—Triple Expansion Engine (Three Cylinders).

Cranks at 120°.

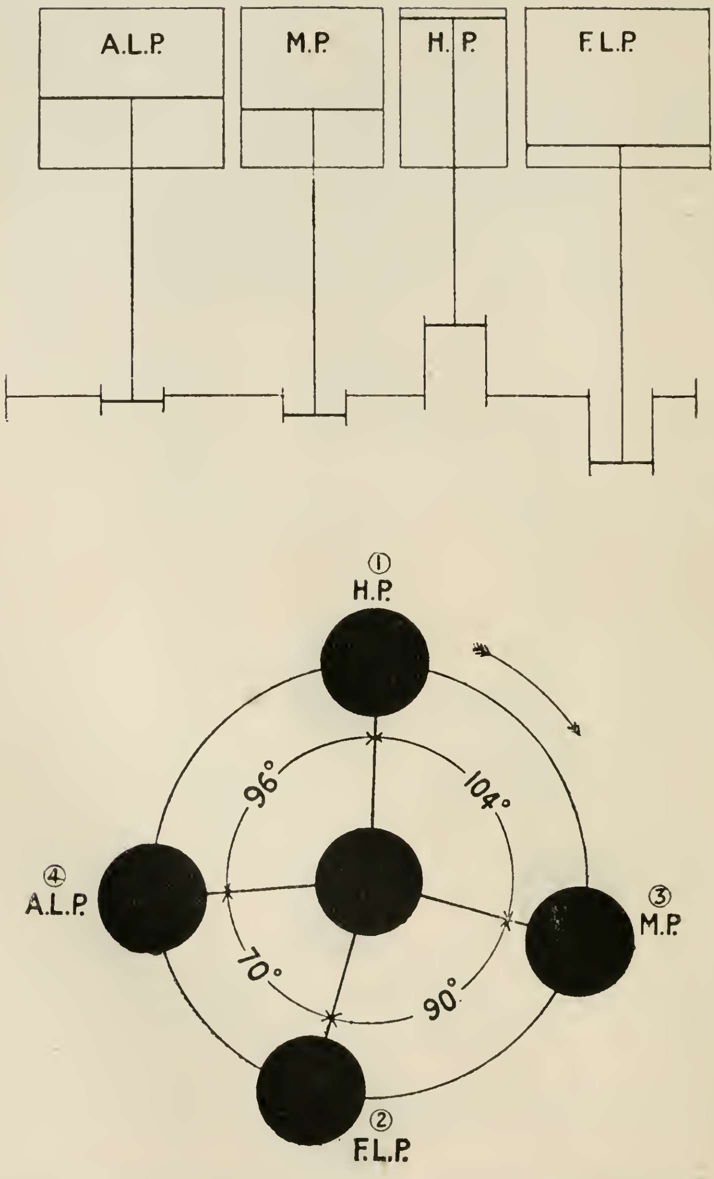
Balanced Engines.—The constantly varying pressures on the crank-pin result in corresponding variations in the twisting stresses exerted by the engine, the range of torsional stresses varying with the type of engine, number and position of cylinders, and the steam distribution in each cylinder.

These unequal stresses continued for long periods often result in the development of flaws on the shaft, and may finally lead to total breakage.

It is therefore desirable to so balance up the moving parts that an even turning movement on the shafting may be obtained, and vibration damped down to a minimum.



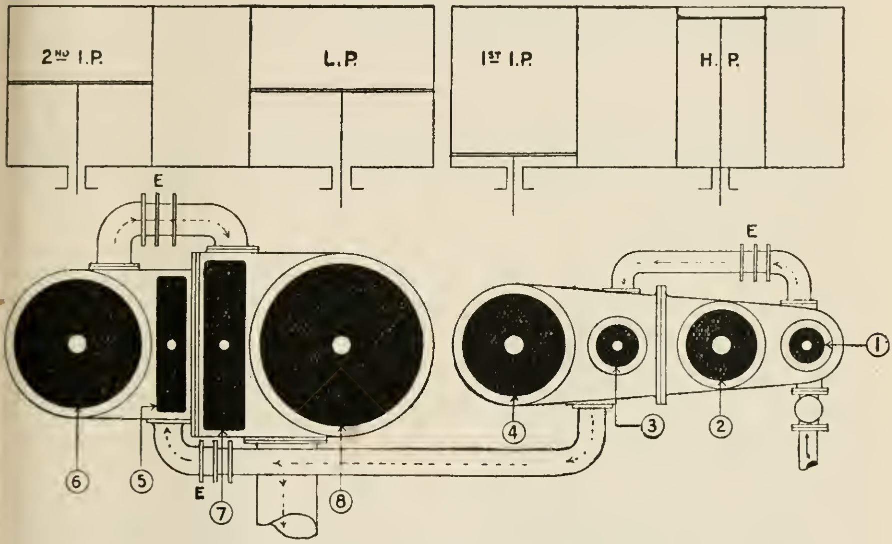
No. 7.—Crank and Cylinder Arrangement, Four-Cylinder Triple Engine.



No. 8.—Crank and Cylinder Arrangement (Yarrow, Schlick, & Tweedy Balanced System).

Regarding this subject Professor W. E. Dalby, M.A., B.Sc., in a paper read at the forty-second meeting of the Institution of Naval Architects, says:—

"The only way of balancing a three-crank marine engine of the usual type is by the addition of balance weights, or bob weights, to the moving parts. In this case, therefore, balancing necessarily means the actual addition of considerable masses of material to the machinery which have no other duty but that of producing forces equal and opposite to the unbalanced forces caused by the motion of the moving parts which are concerned in doing the proper work of the engine. It is well known that Messrs Yarrow,



No. 9.—Quadruple Expansion Type Engine (Large Power).

1, H.P. piston valve.
2, H.P. cylinder.
3, First I.P. piston valve.
4, First I.P. cylinder.
5, Second I.P. slide valve.
6, Second I.P. cylinder.
7, L.P. D.P. slide valve.
8, L.P. cylinder.
E, E, E, Exhaust pipes.

Schlick, & Tweedy made a departure from existing practice when they began to build engines in which the moving parts concerned in doing the proper work of the engine were so arranged that they were in balance amongst themselves. In engines of this kind no part of the machinery merely turns round or reciprocates for the sake of the forces its motion causes on the frame. No such arrangement is possible, however, unless the engine has, at least, four cranks. This condition and the progressive increase in the power of marine engines have together determined the gradual introduction during the last ten years of the four-crank engine into the Navy and the Mercantile Marine. Yet that the possibilities of balancing the four-crank engine have not been generally recognised is shown by the fact that many engines of that type have been and are still being built with their cranks at right angles, even when absence of vibration is imperative. Four cranks at right angles is just the one particular arrangement of a four-crank engine which makes it impossible to effect balance without the addition of balance weights. A change in the crank angles, however, and a small change in the mass of the moving parts is all that is necessary to obtain an engine in which the moving parts are balanced amongst themselves; to change, in fact, a four-crank unbalanced engine into a four-crank balanced engine of the Yarrow, Schlick, & Tweedy type. These changes cannot be made in any arbitrary manner. The masses, crank angles, and centres of cylinders must be mutually adjusted to satisfy certain conditions."

The necessary calculations required in accurately determining the above arrangement of cranks, balance weights, &c., are worked out from the indicator diagrams, crank effort diagrams, and the carefully calculated weights of the various moving parts, and involve a considerable amount of labour.

Balance weights are sometimes fitted to the crank webs of the H.P. and I.P. engines, which are lighter, while the crank-pins of the two L.P. or heavy engines are bored out hollow, so that the weights of the parts may be correctly adjusted.

In the Yarrow-Schlick-Tweedy system of engine balancing, the calculations are usually so carefully determined that the addition of balance weights is not always required, the necessary balance being found by the relative crank angles and crank sequence, or order of rotation.

In an ordinary three-cylinder triple-expansion engine the sequence is either H.P.. I.P., and L.P., or L.P., I.P., and H.P., but when four cylinders are fitted (two L.P.) the sequence is usually as shown in the sketches on page 5.

Observe that the H.P. and I.P. cranks are directly opposite, also that the F.L.P. and A.L.P. are opposite each other, but at right angles to the other two.

It will be thus seen that the crank angles and crank sequence are quite different when the Yarrow-Schlick-Tweedy system is adopted as in the example illustrated on page 6, the H.P. and I.P. cylinders being inside, and the two L.P. placed one forward and one aft. Observe that the heavy engines are placed at the ends to balance up the weight of the moving parts.

The crank sequence is then (1) H.P., (2) F.L.P., (3) I.P., and (4) A.L.P. This arrangement has the effect of reducing the vibration, and also allows of quick and easy handling of the engines.

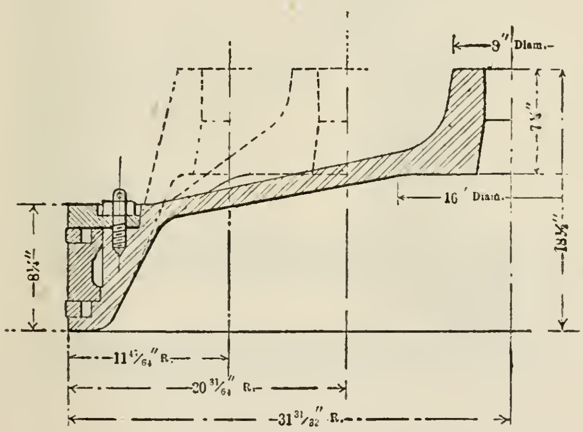
It should be understood that the relative crank angles vary with the size of engine, power, and weight of moving parts.

Valves.—The cylinder valves are either piston valves, single-ported valves, or double-ported valves, a common arrangement being as follows.—

H.P. cylinder - -	Piston valve (inside steam).
I.P. cylinder - -	Piston valve, single-ported or double-ported slide valve.
L.P. cylinder - -	Double-ported slide valve.

Certain builders fit piston valves to all the cylinders of large engines, and in many cases patent valves of the "Trick" double-ported type (see page 200) or of the "Andrews-Martin" type (see page 205) are fitted, the latter giving particularly satisfactory results owing to the good balance obtained.

Pistons.—Pistons are now being constructed in many cases of cast steel, and are fitted with patent rings and springs of approved make,



*** No. 10.—Cast-Steel Piston (with Dimensions).**

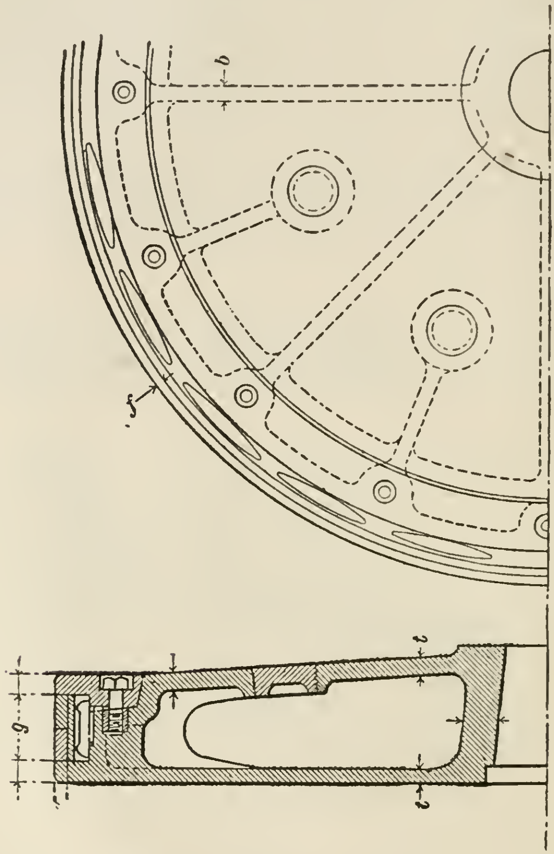
The dotted lines show the I.P. and L.P. pistons equal in depth to the L.P. piston.

NOTE.—The radius of each piston is given.

which, together with the improved design and construction of piston rod metallic packings at present on the market, have assisted to

* Reprinted by permission from "Marine Engine Design." Prof. Edward M. Bragg. D. Van Nostrand Co., New York, 1910.

bring up the efficiency of the marine engine to its present high standard. The introduction of the metallic packings referred to (notably that known as the U.S. packing) (see page 360) has met with

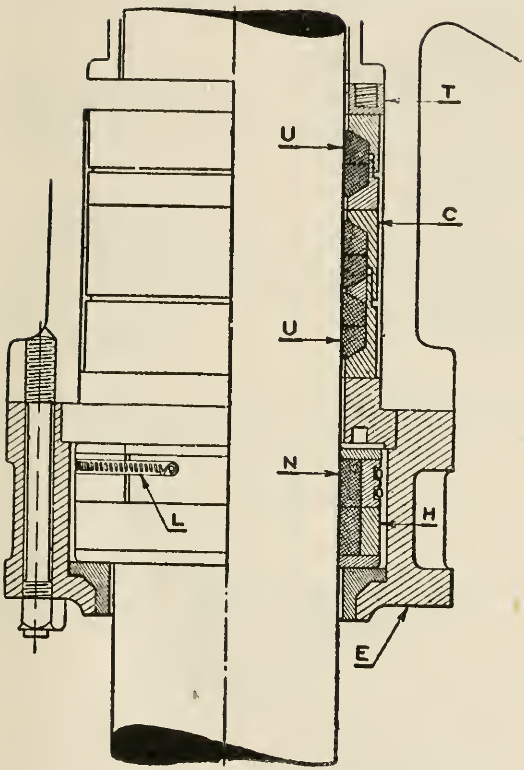


*** No. 11.—Cast-Iron Piston.**

NOTE.—Depth of packing ring $g = 3 \times t$.

the hearty approval of the marine engineering profession as a body, and the form of combination packing supplied by engineering firms in general, if, perhaps, not so effective as the patent types, is yet a great advance on the asbestos and similar packings formerly used for piston rods.

* Reprinted by permission from "Marine Engine Design." Prof. Edward M. Bragg, D. Van Nostrand Co., New York, 1910.



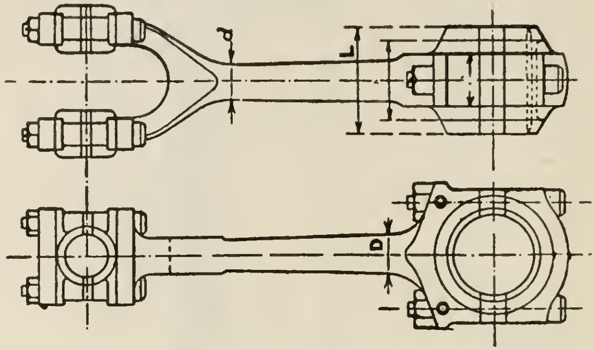
No. 12.—Patent Metallic Packing.

(The Combination Metallic Packing Co., Ltd.)

N, White metal bearing rings.
L, Springs.
E, Extension piece for secondary packing.

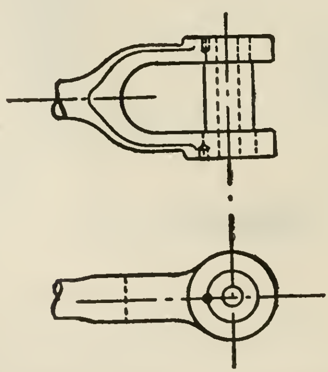
U, White metal bearing blocks
C, Gun-metal case.
T, Springholder.
H, Floating rings.

Connecting Rods.—The two types of connecting rod in common use are known as the single and double top end patterns. The single top end rod is more compact, but the double top end is simpler to manufacture, and is also much easier to overhaul when of large size. In the double type the crosshead is secured to the piston rod by means of a taper and nut, and in the single type the crosshead pin is



*** No. 13.—Double Top End Type Connecting Rod.**

d = Piston rod diameter.
D = $d \times 1{\cdot}2$.
L = Length of crank-pin.



*** No. 14.—Single Top End Type Connecting Rod.**

shrunk into the connecting rod jaws, and sometimes further secured by small locking pins as shown. This pattern of rod costs more to produce than the other type, and is also more difficult to take adrift when overhauling.

The bearing parts of the double top end rod are sometimes made of cast steel with white metal bearing surfaces, but often brass is em-

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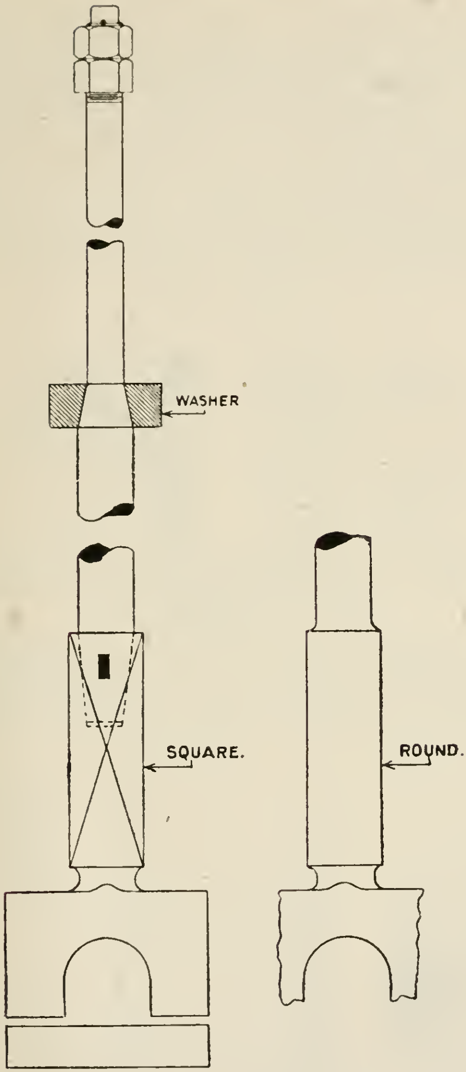
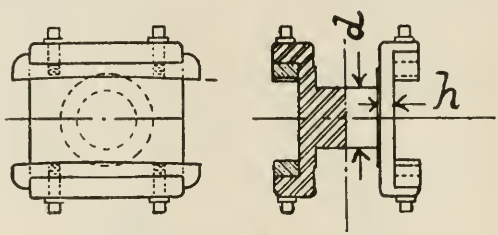


No. 15.—Types of Valve Spindles.

In the square section type shown with the taper and cotter the rod may be drawn out by the top, but with the round section (solid) the rod must be drawn out from below.

ployed for the top ends, and cast steel and white metal for the bottom ends only. In the cheaper class of engines, cast-iron bushes lined with white metal are employed, and this is now the general practice for merchant steamers.

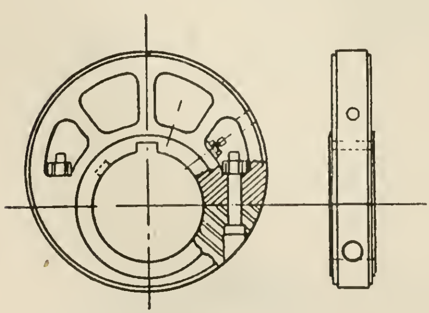
Valve Gear. — Valve gear is generally of the Stephenson link motion type with double bar quadrant; valve spindles are either solid or made in two parts which are connected by a cotter, the upper part fitting by a taper into the lower part. That part of the spindle passing through the guide bracket is sometimes of round section and sometimes of square section. The rod diameter is reduced at the position of the valve, and the washers or cotter under the valve rest on a tapered portion of the spindle. Above the valve a washer is fitted with double nuts (one lock nut), and to further prevent slackening back of these nuts a large split pin or a cotter is run through the rod.



*** No. 16.—Link Block Pin and Liners.**

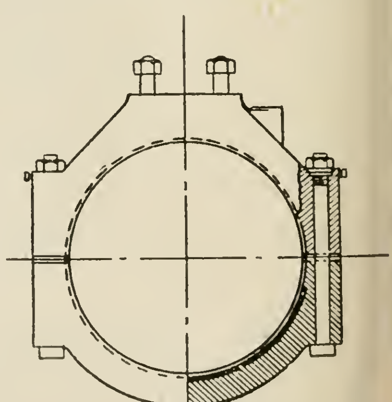
NOTE.—Thickness $h = d \times \cdot 3$.

The bushes in the valve spindle end are usually of brass, and of large bearing surface to reduce wear to a minimum. The saddle or quadrant blocks are of steel fitted with brass liners which bear on the quadrant bars.



*** No. 17.—Eccentric Pulley.**

The pulley is divided into two portions at the shaft centre and these are bolted together as shown, the bolts being arranged with taper heads.



*** No. 18.—Eccentric Strap.**

The strap is recessed out to receiv the pulley and is lined with whit metal, dovetailed in place as shown.

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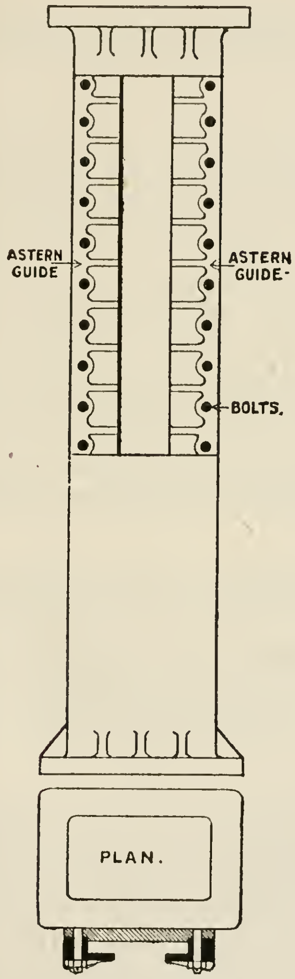
Eccentrics, &c.—Eccentric rods are generally made of steel with brass bushes at the top ends, and the eccentric straps are generally constructed of cast steel lined with white metal as bearing surfaces; the pulleys themselves are of cast iron or cast steel.



No. 19.—"Single" Type Guide showing Ahead and Astern Surfaces.

The plan shows the bolted-on knees, which constitute the astern guides, the bearing surface of which is less than for the ahead guides (about 80 per cent).

Main Bearings.—Main bearing bushes are generally of cast iron lined with white metal, and the bottom half is often made round to facilitate withdrawal. Suitable gutters or oil ways are cut in the white metal surfaces to allow of efficient lubrication, and often the sides are cut away clear altogether, leaving only the top and bottom surfaces effective.

NOTE.—From the foregoing it will be obvious that the term "brasses" is now hardly correct, brass for bearings being generally superseded by cast steel lined with white metal.

Crank-shafts.—The crank-shafts of mild steel are usually of the built pattern with the pins and shaft lengths shrunk into the webs and secured by dowel pins. Sometimes that part of the shaft fitting into the webs is about ½ inch greater in diameter.

Columns.—Columns vary in design, but the usual types fitted are that known as the "box" pattern, and that of the Y type, which is fitted for engines of large power. In some engines of the "open-fronted" type round steel columns are fitted at the front, and the back columns are then arranged with astern guides which overlap the guide shoes, and are held in place by large bolts.

NOTE.—This arrangement of columns was often fitted in the engines of Government torpedo destroyers before the advent of the marine steam turbine.

Description of Construction.

In the construction of the reciprocating engine we will now proceed to deal with the various operations which are performed from the time the castings, forgings, &c., are delivered at the works until the engine is completed in the fitting department and ready for erection in the ship.

Soleplate.—This is generally of box pattern and is made up of several parts, usually three in number, bolted together, one piece forming the forward part, one the centre (Sketch No. 21), and one the after part. In large engines, however, the soleplate sometimes consists of four parts, which are, of course, bolted together. After delivery of the castings the first operation is that of gauging to ascertain if the thickness of metal as required by drawing has been maintained.

This having been found correct, the soleplate is now marked off preparatory to machining the base for columns connecting flanges, and gaps for main bearing bushes. In good practice the bottom of the soleplate is machined, as this ensures good fitting chocks in the ship, and also facilitates the fitting of same. The soleplate is now taken to slotting machine, and has the base for columns and the bolt.

The main bearing bolts being now fitted in all parts of the sole-

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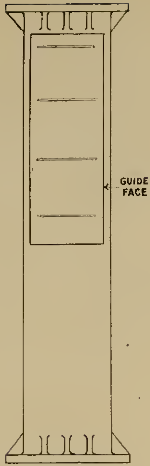
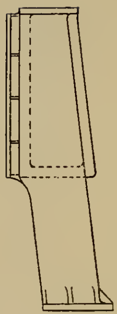
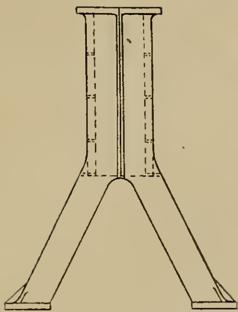


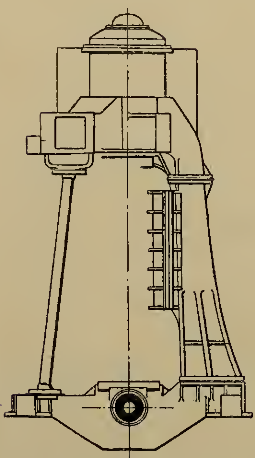
A, Double type column



* B. Double type column.

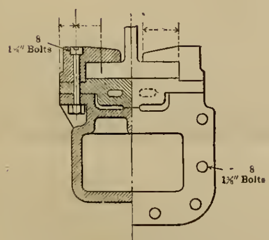


* C, Y type column.



D, Single I type column with open front.

E, Round column for open front.



* F, Plan of single type column showing ahead and astern guides with shoe in position.

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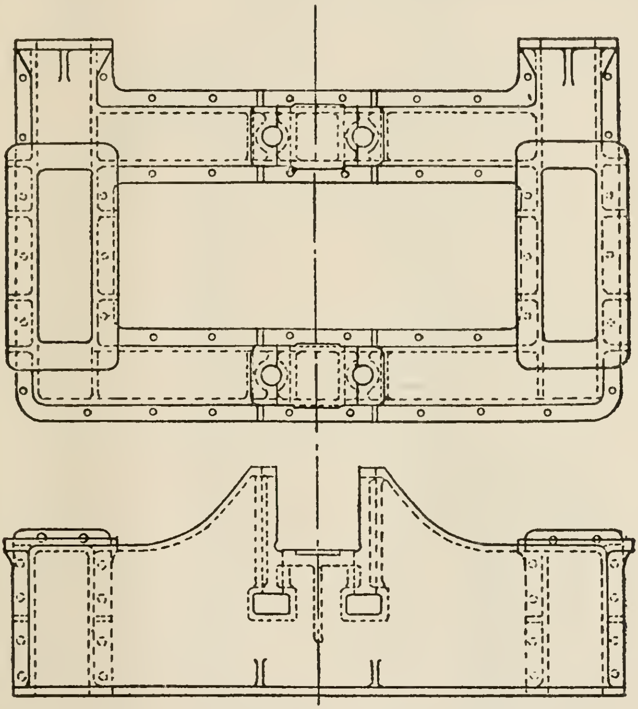
No. 20.—Types of Columns.

[To face page 16.

connecting flanges machined: this operation is carried out on the other parts which form the complete soleplate, and the main bearing gaps are also machined while the soleplate is in this stage. The soleplate being now finished machining, the holes for main bearing bolts, holding-down bolts, and connecting flanges are bored, also the holes for bolting columns to soleplates. The main bearing bolts are now fitted; these bolts are, in ordinary merchant work, a large double-ended stud having a nut at the bottom end which draws the bolt tight up on a collar at the top end (Sketch No. 22). The bolt



*** No. 21.—Part of Soleplate showing Base for Columns.**

s usually reduced to the diameter at the bottom of the thread in he middle, and is a fit in the parallel parts, where it passes through he hole in the soleplate at either end. A feather on stop pin is fitted nder the collar, and this prevents the bolt from turning round when he bolt is being screwed up or slackened. The bottom nut is locked ither by a large split pin through the thimble point at the end of ıe bolt or by a set pin through the nut, and pointed into the screw f the bolt.

The main bearing bolts being now fitted in all parts of the sole-

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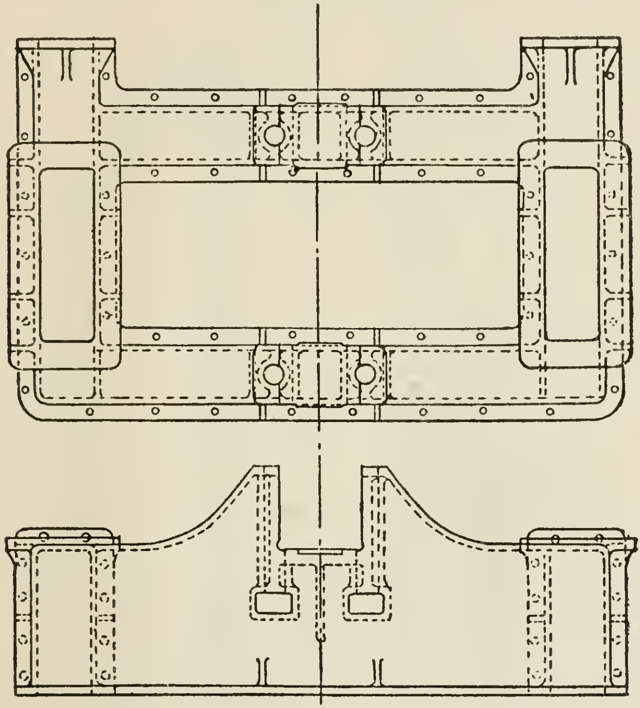
and gaps for main bearing bushes. In good practice the bottom of the soleplate is machined, as this ensures good fitting chocks in the ship, and also facilitates the fitting of same. The soleplate is now taken to slotting machine, and has the base for columns and connecting flanges machined: this operation is carried out on the other parts which form the complete soleplate, and the main bearing gaps are also machined while the soleplate is in this stage. The soleplate being now finished machining, the holes for main bearing bolts, holding-down bolts, and connecting flanges are bored, also the holes for bolting columns to soleplates. The main bearing bolts are now fitted; these bolts are, in ordinary merchant work, a large double-ended stud having a nut at the bottom end which draws the bolt tight up on a collar at the top end (Sketch No. 22). The bolt



*** No. 21.—Part of Soleplate showing Base for Columns.**

is usually reduced to the diameter at the bottom of the thread in the middle, and is a fit in the parallel parts, where it passes through the hole in the soleplate at either end. A feather on stop pin is fitted under the collar, and this prevents the bolt from turning round when the bolt is being screwed up or slackened. The bottom nut is locked either by a large split pin through the thimble point at the end of the bolt or by a set pin through the nut, and pointed into the screw of the bolt.

The main bearing bolts being now fitted in all parts of the sole-

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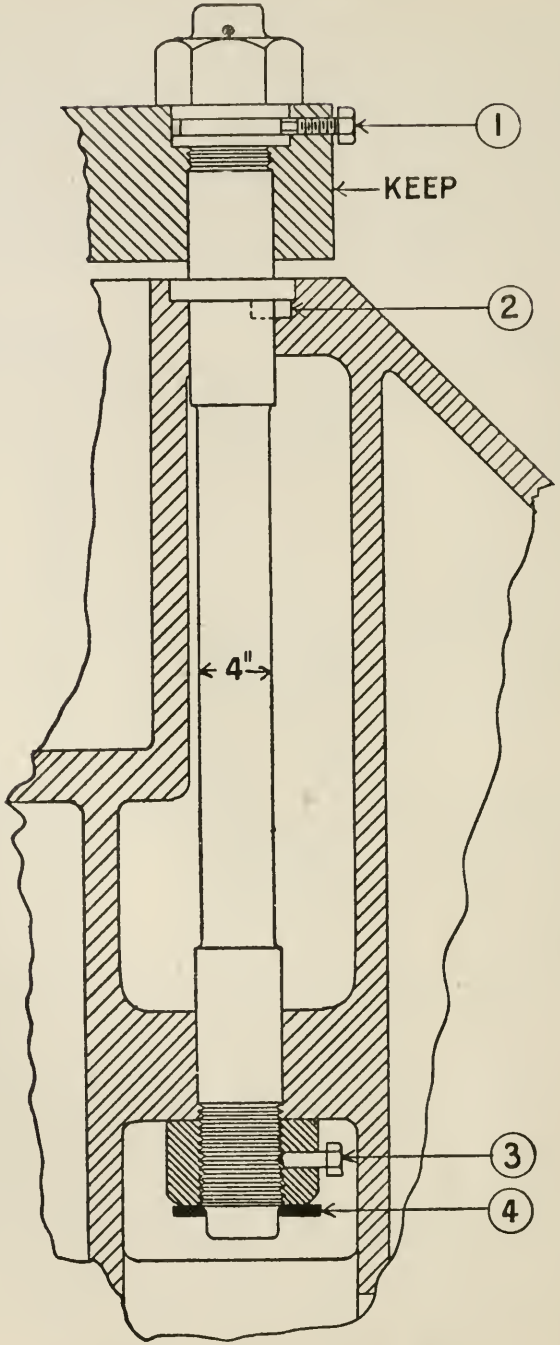
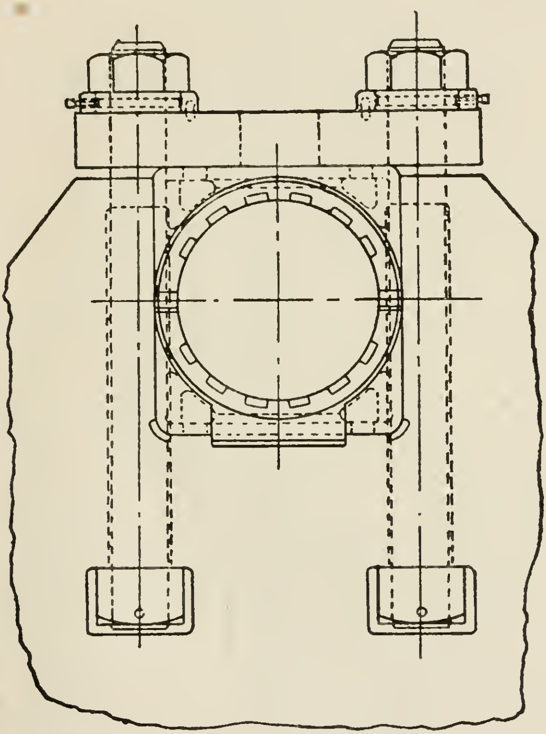


No. 22.—Main Bearing Bolt.

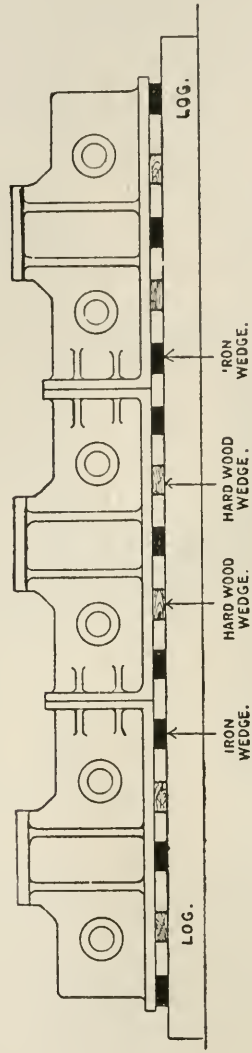
1, Set pin. 2, Feather. 3, Set pin. 4, Split pin.

plate, the next operation is that of setting up or lining off the soleplate. This operation is usually performed on the blocks on which the engine is to be erected. The foundation for the engine usually consists of long logs laid fore and aft, two on each side of soleplate. The soleplate is laid on the logs and levelled up, and the intervening space is filled up with wooden wedges (Sketch No. 24), the three parts of the soleplate being laid on the logs and as near in line as possible by the eye: the different sections are next brought into line. Through the two gaps of the centre portion a straight-edge is laid resting upon



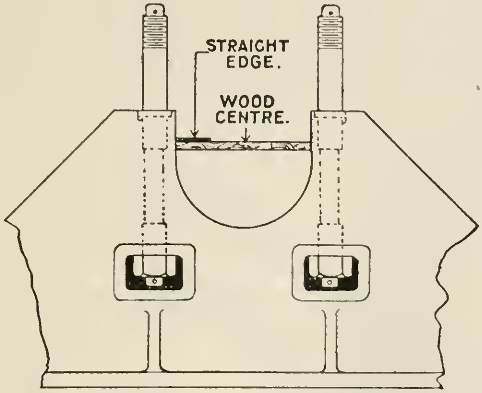
No. 23.—Main Bearing Complete.

ooden centres fitted into the gaps, about 3 inches from the top (Sketch o. 25). The straight-edge is kept bearing hard on the side of the ap; the end of the straight-edge extends into the inside gap of the rward portion of soleplate; this part of soleplate is now moved deways (by means of screw jacks) until side of gap bears on straight-dge. The same operation is performed on aft section, and the edge of e straight-edge is tested by means of feelers until all three parts are ose up to straight-edge. The straight-edge is now put through bottom f gap and all three parts brought up to line in a similar manner, It will be readily understood that after this operation it is necessary to go over the preceding work, so as to ensure that the sides of gaps are still in line. The soleplate is also levelled fore and aft and

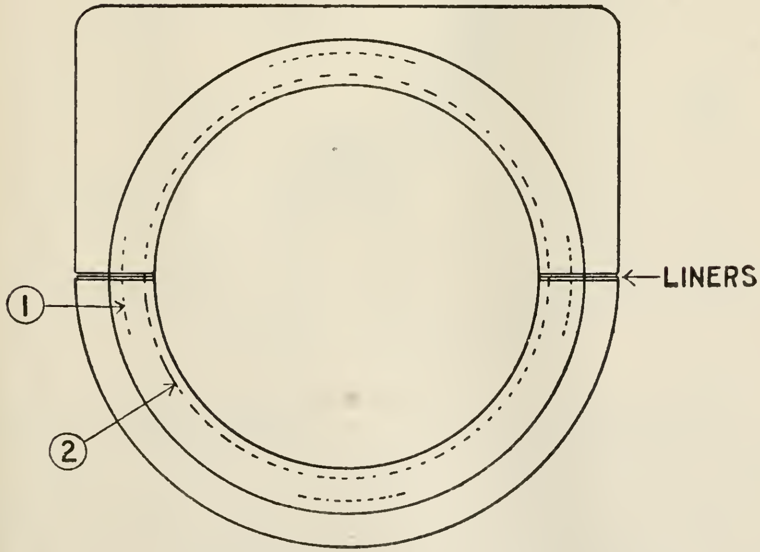


No. 24.—Lining off the Soleplate.

athwart-ships by applying the straight-edge on the base for columns. The soleplate being now set up, the holes in adjoining or connecting flanges are widened and bolts fitted. The space between soleplate and logs is also filled up with wooden wedges, care being exercised while driving up same that the level of soleplate is not altered. The main bearing bushes are now fitted into the gaps, the main bearing covers put on and screwed up. The next operation is that of marking



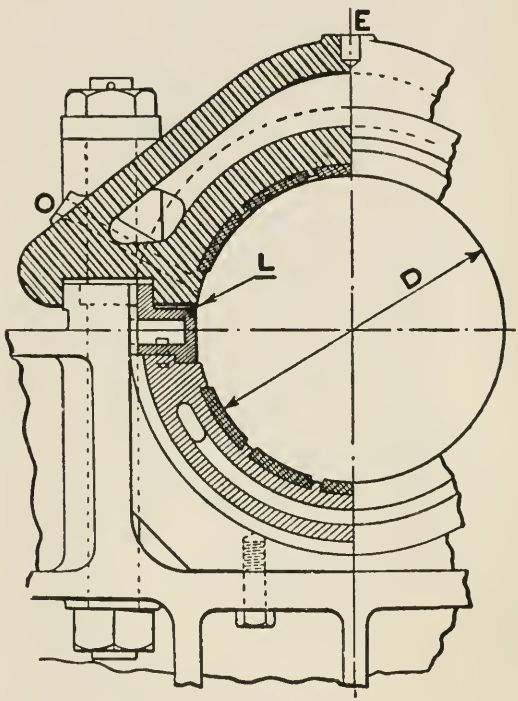
No. 25—Lining off the Soleplate.



No. 26.—Boring out of Main Bearing Bushes.

1, "Proof" lines. 2, Boring out line = diameter of shaft.

off the main bearing bushes: there are several different methods of doing this work. In some works the bushes are not filled with white metal until after being fitted, and the cast iron or steel is bored out to the diameter of the shaft plus the reliefs, and the bush is then filled with white metal and reset up on the machine to the previous machined parts, and bored out to the diameter of the shaft. The general practice, however, is to have the white metal in bush when fitted, and with all bushes in place a fine piano wire is stretched through all the bushes, being carried on supports at each end. This wire is set up athwart ships to the centre of the main bearing gap, which is projected on to the end of the bush at the forward and aft



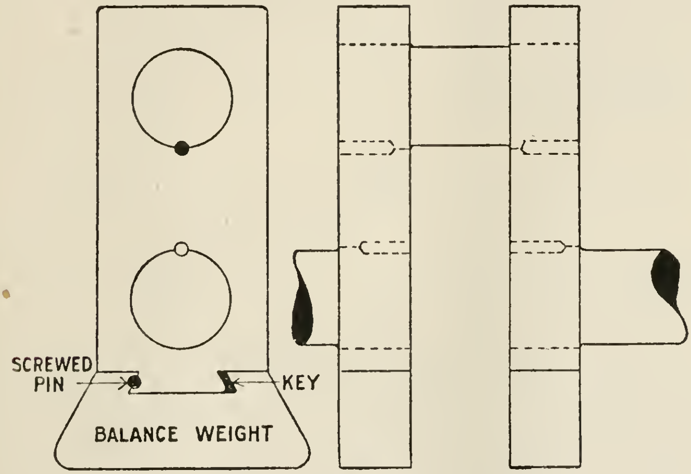
No. 27.—Main Bearing Bush.

L, Liners. D, Shaft diameter.
E, Tapped hole for lifting gear. O, Oil service.

end. The height is taken from drawing and is measured from base for columns. The wire now being set to these points four lines or chords are drawn on each side of each bush with a pair of jennys (Sketch No. 26). The wire is now withdrawn and in each bush a wooden centre is fitted faced with tin, and the centre is picked up from these four lines or chords and the boring out diameter is drawn in on each side of the bush, also a short proof line at four points. This proof line is usually about $\frac{3}{4}$ inch to 1 inch larger in diameter than the boring out size, and is used to test the boring bar when the bush is about bored out to the final size. The bushes are now bored out, and reliefs or gutters cut. The bushes are now put back in place in the soleplate, and are now ready for the bedding down of the crank-shaft.

Crank-shafts and Columns.

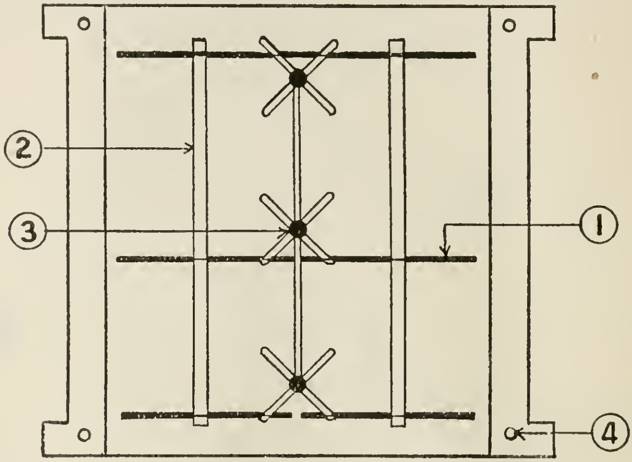
The majority of engineering firms buy in the crank-shafts required for the various engines which they are constructing. This part of the engine is usually delivered in a finished condition, as it is found the steel works which specialise in this work can turn out the finished article much cheaper than it can be constructed in a general engineering concern. The type of crank-shaft now manufactured is that of the built-up type, with the crank pins and shafts



No. 28.—Balanced Crank.

shrunk into the webs, and is usually in three parts having two couplings. This arrangement admits of interchangeability and necessitates one part only for spare. The method of constructing the shaft is as follows :—The webs are machined to shape and the holes bored and turned out for the pins and shafts. The web is laid flat down on a table, over a pit which is of sufficient depth to receive the shaft, when the web is turned upside down to receive the pin. The web is then heated up by means of a bunsen burner sufficiently to allow a gauge, $\frac{1}{32}$ inch larger than the diameter of the part which is to be shrunk, to pass through, and this expansion having taken place the pin or shaft is lowered into the web and the whole part cooled out. The web is next turned upside down and the same operation takes place; that is, the pin being shrunk in first and then the shaft. The other half of the crank is similarly assembled, and then the part consisting of the shaft and pin is suspended above the other part, and the pin is shrunk into the web, thus forming the complete crank. During the operation of joining the two webs by means of the pin care is taken to ensure that the centre lines through the webs are exactly in line. The crank pins and shafts, besides being shrunk into web, are also prevented from turning by round dowel pins being fitted half into the shaft and half into the web (Sketch No. 28).

After the complete shaft is built the coupling holes are bored, and the three parts brought together, the cranks being set to the sequence required. The coupling bolt holes are widened, and bolts



No. 29.—Taking "Leads" off Top Main Bearing Bush.

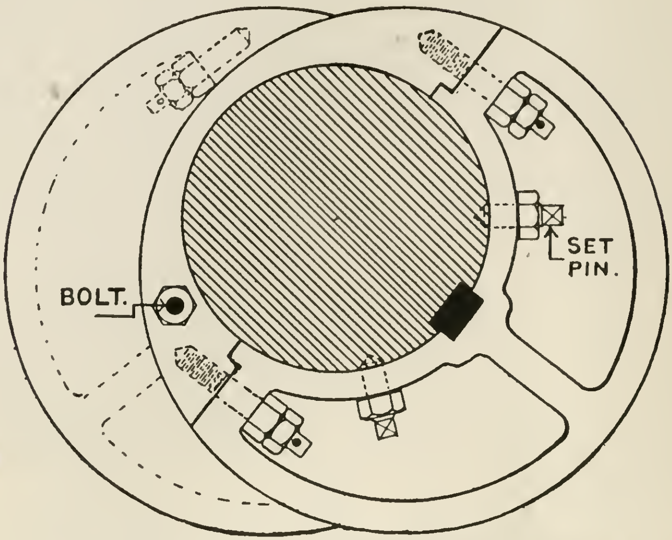
1, Lead wire (18 B.W.G.). 3, Oil hole.
2, Gutters or reliefs. 4, Dowel pins to hold liners in place.

fitted. The shaft is now put into lathe, and the body of the shaft is turned over, also the flanges of the couplings: it is next centred for the turning of the crank-pins, each pin having a separate centre. In large engines and in fast running engines the crank-shafts are usually balanced so as to ensure steady running. This is done theoretically to suit the power of the engines in relation to the power developed in the various cylinders. To attain this balance, balance weights are fitted on the webs of the crank (Sketch No. 28). The crank-shaft is now ready for bedding down in the main bearing bushes. This operation is done by red leading the part which runs in the bearings, the shaft is lowered and turned round, and then lifted, and the parts which show bearings are eased with a scraper, and shaft again tried until it is bearing throughout all the bearings

During the operation of bedding down the shaft, care is taken to ensure that it is thoroughly level when resting in the bearings. After this the top bushes are "leaded," which means that they are fitted down to rest on the liners and give a clearance between the shaft and the bush, usually ·020 inch—this is to ensure passage for oil, and also to allow for expansion of shaft when it acquires what is generally termed a running heat. The clearance of the bush is found by means of lead wire, hence the name "leading." The wire is usually 16 or 18 B.W.G., and is put across the shaft, one part in the middle of the bearings and one at either end (Sketch No. 29). The main bearing cover is put on, and nuts screwed up and hammered. The cover is then slackened back, and bush taken out, and the lead wire is gauged by micrometer. When the leads show an even thickness of the required amount at all parts then the bush is finished "leading." The eccentric pulleys for operating the valve gear are fitted on the crank-shaft, usually after the shaft has been bedded in bearings. The pulleys are bolted on shaft, and set to the required position as shown on drawing, for the marking off of keyseats in the following manner. The H.P. crank-shaft is turned with crank up, and the lines for setting the pulleys are put on the shaft in the following manner.

Cutting of Eccentric Keyseats.

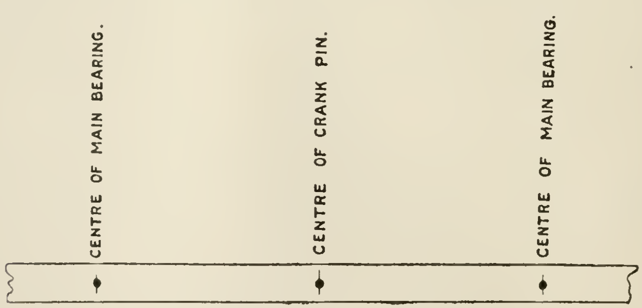
A diagram is prepared in the drawing office, showing the position of pulleys with the crank down. This diagram (see page 233) is mounted on a board, having a half circle, equal to the diameter of the shaft cut out. The vertical line is set as near the crown of the shaft as possible, and a spirit level is applied to the top of the board, and the board brought level. The vertical line is now transferred to the shaft, and by applying a box square, the line is produced along the shaft, a little longer than the distance which will be occupied by the ahead and astern pulleys. From the diagram the distance from the centre to centre of keyseat in ahead pulley is taken by means of dividers, and this distance is marked off on shaft, applying one end of the dividers to the line along the top of the shaft. An arc is drawn at two points, and a straight-edge set to these marks and a line drawn. This line represents the centre of keyseat for the ahead pulley, and the same operation is carried out for the astern. The pulleys are now put on shaft and set with the keyseats exactly central to the mark, and the distance from centre of crank to centre of pulleys is arrived at by means of a length stick marked off from the drawing. In the case of an H.P. engine having an inside lead piston valve, then the ahead pulley will be 90° minus lap and lead behind the crank. An outside lead valve will have the pulleys set as follows (Sketch No. 44):—Ahead pulley 90° plus lap and lead in advance of the crank. The keyseats for the pulleys are generally cut out by hand, and feathers fitted into same. The pulleys are also secured by having bolts passing through both pulleys or having set pins thimble pointed into the shaft (Sketch No. 30).



No. 30.—Method of Locking Eccentric Pulleys.

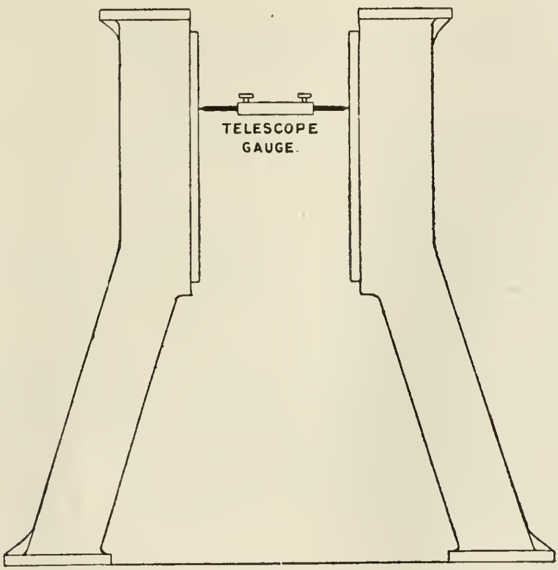
Erecting of Columns.

In the tramp type of engine the back or ahead columns are usually part of the condenser, but in larger engines the condenser is usually separate, being supported on brackets at the back of the engines, and all the columns are fastened direct to the soleplate.



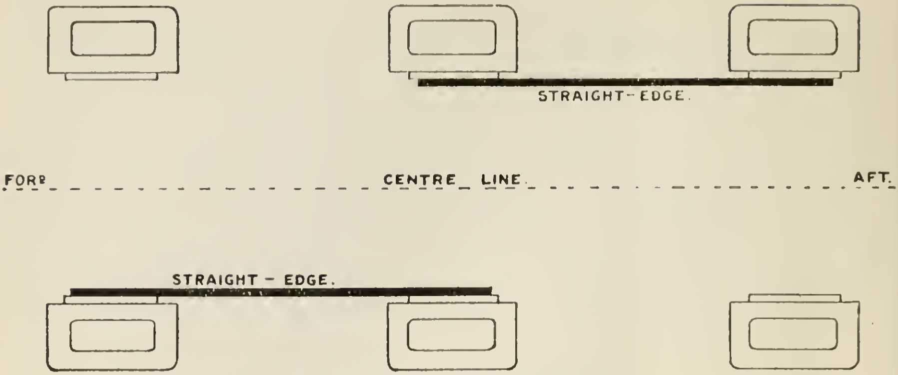
No. 31.—Length Stick for Lining off Columns.

In the first place, we will deal with the erection of a set of engines having the condenser and back columns in one. In this case the bottom of the condenser has flanged faces which adjoin similar faces on the soleplate; these faces are bolted together. In lining off the columns the first operation is to fit centres into the gap for the main bearing bush; along those centres a straight-edge is laid, one edge of which is set to the line representing the centre of crank-shaft. From the top of the columns plumb lines are dropped down. Parallel to the straight-edge and about $\frac{1}{4}$ inch clear of it, a length rod is laid, having marked on it lines corresponding to the centres of the three cranks (Sketch No. 31). The plumb line is now set so that it just grazes the straight-edge, and is directly in line with the line representing the centre of the crank. This being so, the column opposite the one from which the line is suspended is brought into line by



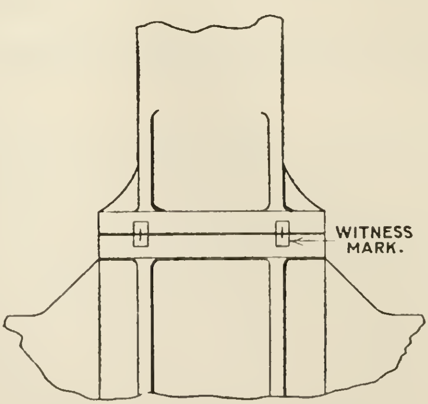
No. 32.—Gauging between Columns.

means of a square applied from a centre line on the face of the column, and by means of a telescope gauge set to half the distance between columns, as shown on drawing. This operation is carried out on all columns, and a plumb rule is applied to the face of each individual column to ensure it being plumb. If it is not plumb, then the column is taken down, and the base on which it rests is either scraped or filed if the amount is not great, but should it be too much for hand labour, then the column itself is machined to suit. The condenser and back columns being one, it is necessary to set this part first, after which the three front columns are brought into position. The distance between the column faces is gauged by means of a telescope gauge (Sketch No. 32). A straight-edge is also passed along



No. 33.—Lining off Columns.

the face of the columns, taking two columns together (Sketch No. 33), and the edge of the straight-edge is tested by means of feelers to ensure all columns being dead in line to each other. It is also advisable to try the straight-edge along the top of the columns to test the height of each, and to ensure the condenser being set up to the height of the front columns. In large engines it is found to be more accurate to work with fine piano wire instead of the plumb line, but in general work the process is similar to that described, but every firm has a system of their own, the result being the same. The columns now being set, the holes in the feet are marked off from the holes on the soleplate flange, witness marks, *i.e.*, at two or four points on each flange a small part is chipped and scraped up and a line put across, either with a drawpoint or with a fine chisel (Sketch No. 34). These points serve to ensure the columns being put back to the position they were in after they are bored, although it

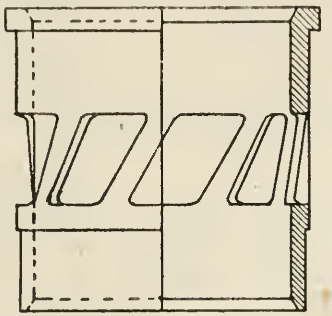


No. 34.—Witness Marks.

is advisable to retest the setting before widening holes. All being right, the holes on the feet of columns and on the flanges connecting the condenser to soleplate are widened and bolts fitted, and the engine is now ready to receive the cylinders.

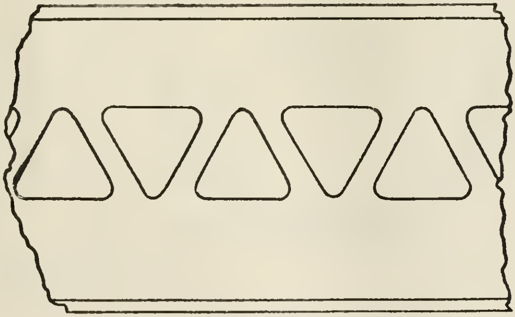
Cylinders and Valve Chests.

The cylinders in a triple expansion engine are termed the High Pressure, Intermediate, and Low Pressure, the usual form



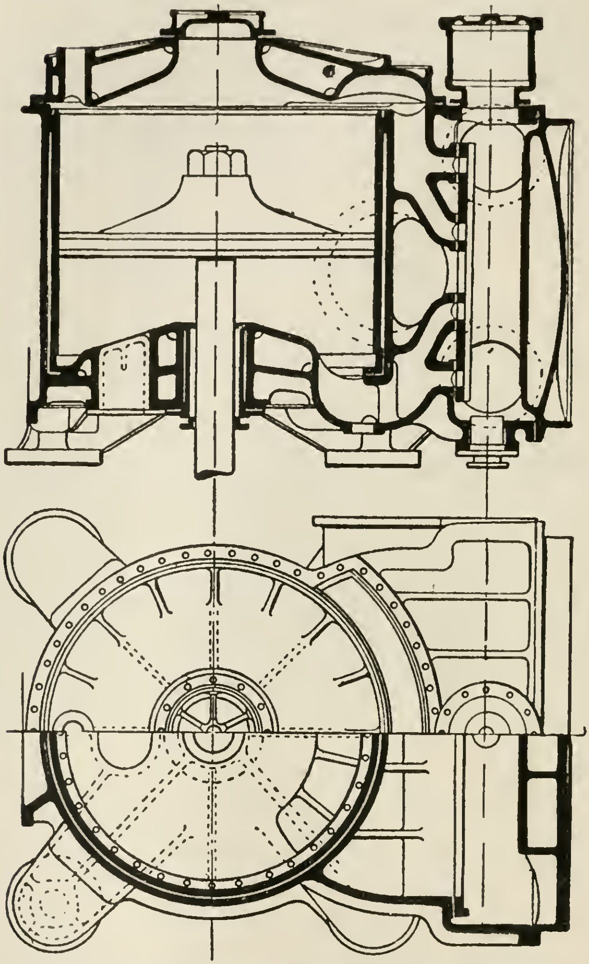
No. 35.—Piston Valve Casing Liner (Brass) (Half in Section).
Showing ports and diagonally cut bars.

of contraction being H.P., I.P. (or M.P.), and L.P. The valve on the H.P. cylinder is usually of the piston type, and in the valve casing liners are fitted (Sketches Nos. 35 and 36). The reason for fitting



No. 36.—Piston Valve Casing Ports and Bars.
The bars are cast as shown to provide a continuous bearing surface for the rings.

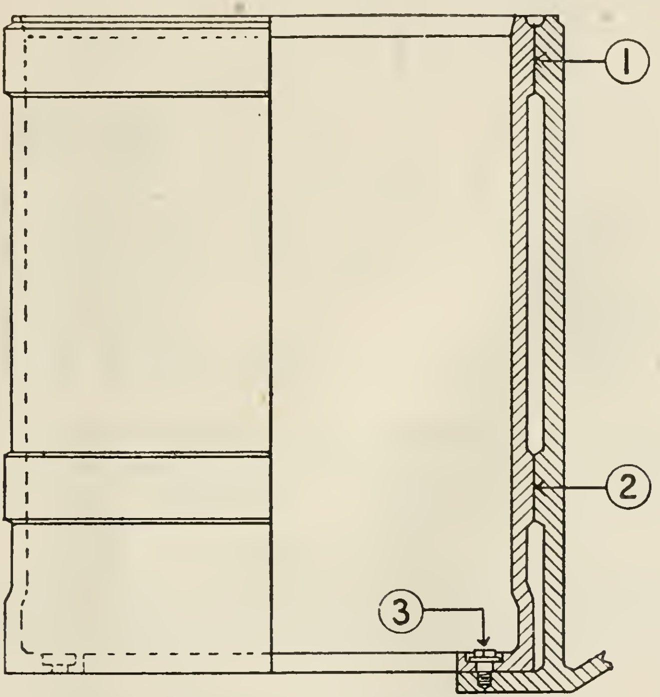
liners instead of boring out the cylinder to the size of the valve is to allow of renewal and also to simplify the cutting of the steam ports. The liners are usually of hard-grained cast iron, and being separate and of small size, it is possible to ensure a close-grained and homogeneous casting. The liners are turned up on the outside and the inside bored out to $\frac{1}{8}$ inch smaller than the finished diameter. The ports are



No. 37.—Cylinder and Valve Chest (Naval Type).

The cylinder is fitted with a liner and the valve casing with a balance cylinder.

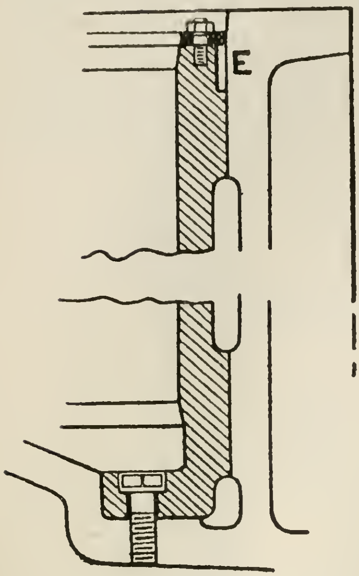
cut out, and after the cylinder is bored out, the liners are fitted into place. The cylinder is put back into machine and the liners are bored out to the finished diameter, thus ensuring a true job in relation to the bore of the cylinder proper. In general work liners are also fitted in the cylinder, this also, as in the case of the valve liners, giving a hard and close-grained casting, and also facilitates renewal. The liners (Sketch No. 38) are secured to the cylinder in various ways, but usually have a flange on the bottom which is bolted to the cylinder and bearing strips at the top and bottom. On the top of the liner a recess is formed into which asbestos packing or similar jointing material is put, and on the top of the packing a piece of steel wire of



No. 38.—Cylinder Liner (Half in Section).

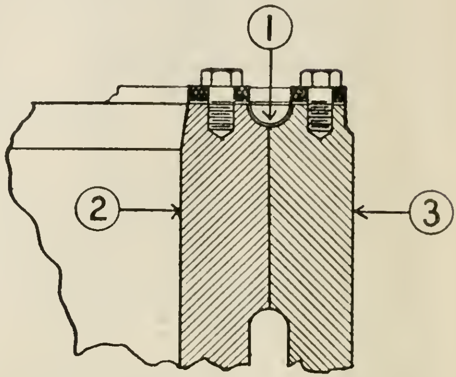
1, 2, Fitting strips.
3, Screwed pins.

the required diameter is put in, having the end scarfed. Above this a flat steel ring is fitted on to studs in the flange of the liner. The studs are square necked, and nuts and split pins are fitted, thus ensuring that no part will slacken back. The space between the liner and the cylinder body is termed a jacket, and to this space a steam connection is sometimes made from the stop valve, and also a drain valve on the bottom. These connections are fitted, to be used when heating up the engines, and to ensure an equal expansion of the cylinder body taking place while the engines are being heated up. The cylinder liner and valve liners being fitted, and other parts of the cylinder machined, the cylinder is now ready for water testing. This operation usually takes place on a set of cylinders which are connected together at the valve casings, when all three cylinders are at the same stage of construction. The I.P. cylinder and L.P. cylinder are



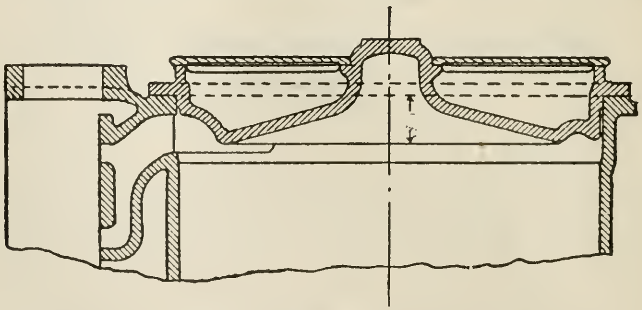
No. 39.—Method of securing Cylinder Liners.

E, Packing space.



No. 40.—Method of keeping Joint of Liner Tight.

1, Copper ring. 2, Cylinder liner. 3, Cylinder wall.



* No. 41.—"Box" Type of Cylinder Cover.

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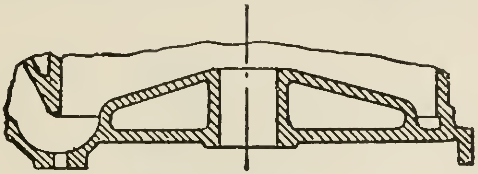
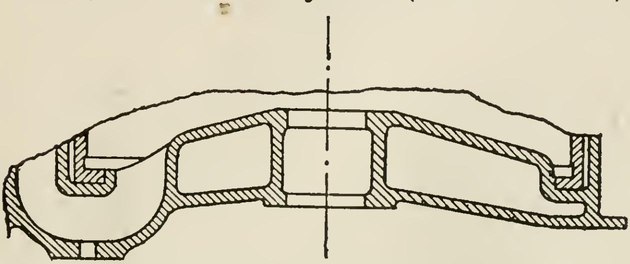
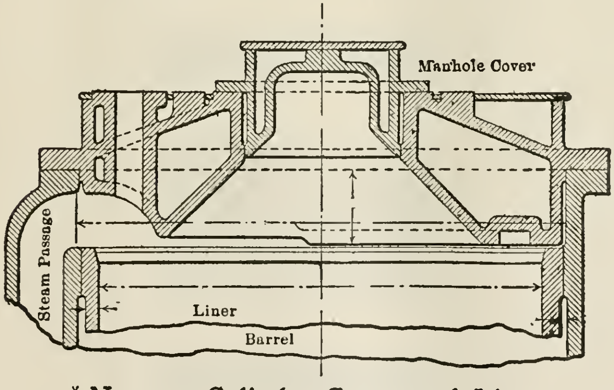
sometimes fitted with a cylinder liner, but this is only in high-class work. The I.P. cylinder having a slide valve is fitted with a valve face. This face is fastened to the face of the cylinder by brass cheese-headed pins (Sketch No. 45), and is of hard close-grained cast iron,



*** No. 42.—Bottom of Cylinder ("Box" Pattern).**



*** No. 43.—Cylinder Bottom with Liner in Position.**



*** No. 44.—Cylinder Cover and Liner.**

Notice how the cover is shaped exactly to fit the piston and so reduce the clearance losses.

and, in some cases, of special metal adapted to resist wear. The intermediate face is usually in one, and the low pressure face in two, one part forming the top half to the centre of the exhaust port, and the other half from same point to the bottom. After the cylinders have been erected on top of the engine columns, and plumbed to that

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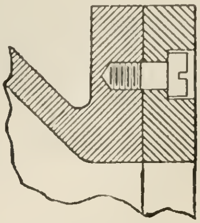
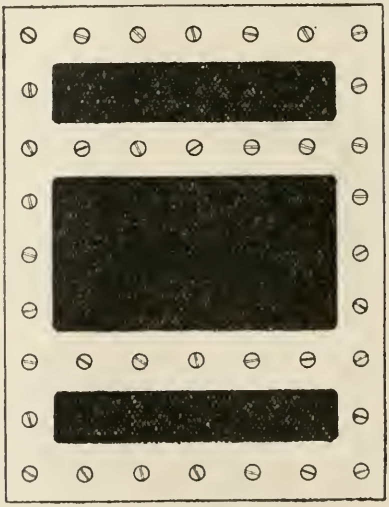
position, the cylinders are taken down and the valve faces machined over, so as to ensure the face of the valve being parallel to the bore of the cylinder. The cylinders being now all the same distance forward in the course of construction, they are brought together and the H.P. cylinder jointed to the I.P. cylinder at the casing or receiver,



No. 45.—Method of securing Cylinder False Face.

and the L.P. cylinder is jointed in a similar manner to the other side of the I.P. cylinder. This joint is usually made of mastic cement or red lead putty, and in some cases a groove is cut down each side and along the bottom of the joint flange; this serves to retain the jointing



No. 46.—Method of securing False Face.

composition and to ensure a good joint. The cylinders are now water tested to the required pressure, usually twice the working steam pressure, and are now ready for erecting on the top of the columns. In the stuffing boxes and in the valve spindle stuffing boxes, centres are fitted from which plumb lines are suspended, and if the crank-en

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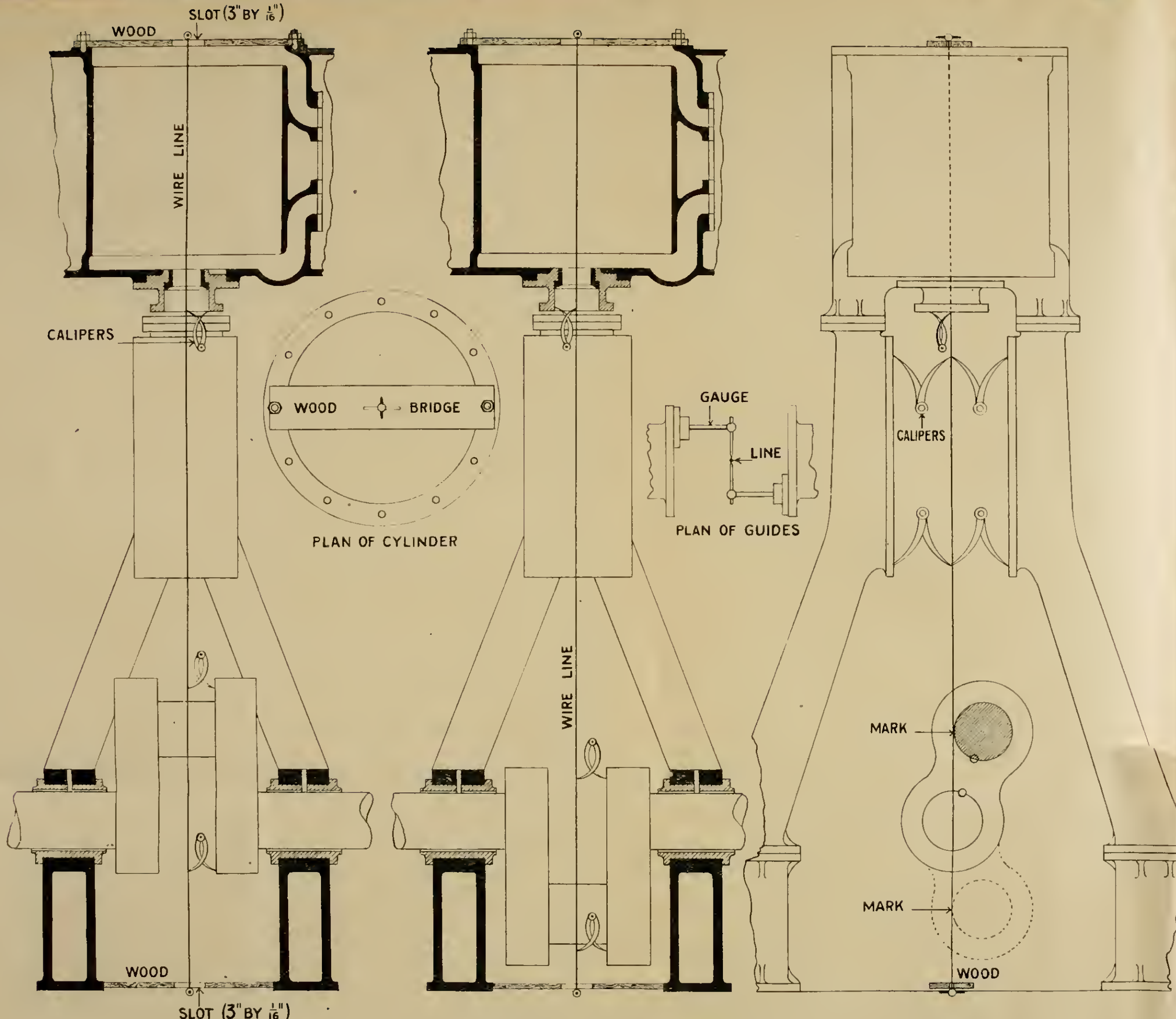
on the column heads, and the holes for bolting the cylinders to the columns are marked off. The cylinders are taken down and also the columns, which are bored on the top flanges, and any machining which is required to bring the cylinders plumb is done.

The columns are re-erected, cylinders put up and set to witness



No. 48.—Alignment of Cylinders and Shafting.

Method of Lining up Cylinders, &c.

To either set up the cylinders or to test if the cylinders and shaft centre lines are at right angles to each other (engine dismantled) proceed as follows:—

1. Take a piece of board with a slot cut as shown (about 3 inches by $\frac{1}{16}$ inch will do), and secure this by two studs to the cylinder across the centre to bring the hole in wood bridge fair.

2. Fix another slotted board in the centre of the crank-pit fore and aft with the hole dead centred as shown.

3. Tie a bolt or small bar of any kind (a file will do) to the end of a line passed down through the cylinder bridge slot, and secure the other end of this line to the crank-pit bridge slot: the bolts or bars to which the line is attached laid crosswise on the slot will allow of adjustment at top or bottom.

4. Now caliper the line at the top from the cylinder bore, and adjust it by means of the small bar as required until it is brought dead central. Repeat this at the bottom of the line, adjusting at the crank-pit bridge slot.

5. Next caliper round the line from stuffing box bore and adjust cylinder to suit if necessary; also test distance between line and crank web at top and bottom centre as shown.

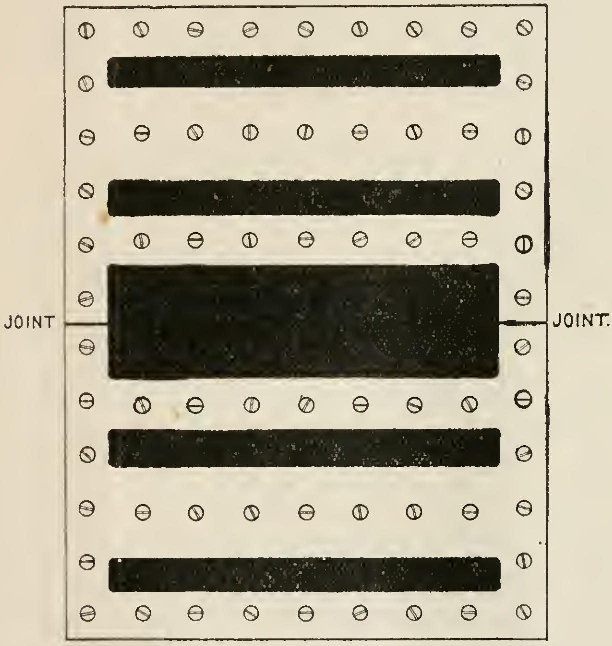
6. With crank on top mark crank-pin where line touches, then turn crank to bottom and again mark pin; now test, by calipering, if both marks are the same distance from the web.

7. To test the alignment of the guide caliper between the line and guides at top and bottom as shown in the end view.

8. To test if the guides are in line fore and aft, use a surface gauge and adjust it to touch the line when laid up against the guide forward; now try it aft, and if the point again touches the line, the guides are in line, fore and aft, if not, they are out of line and require to be canted round to square up.

[To face page 35.

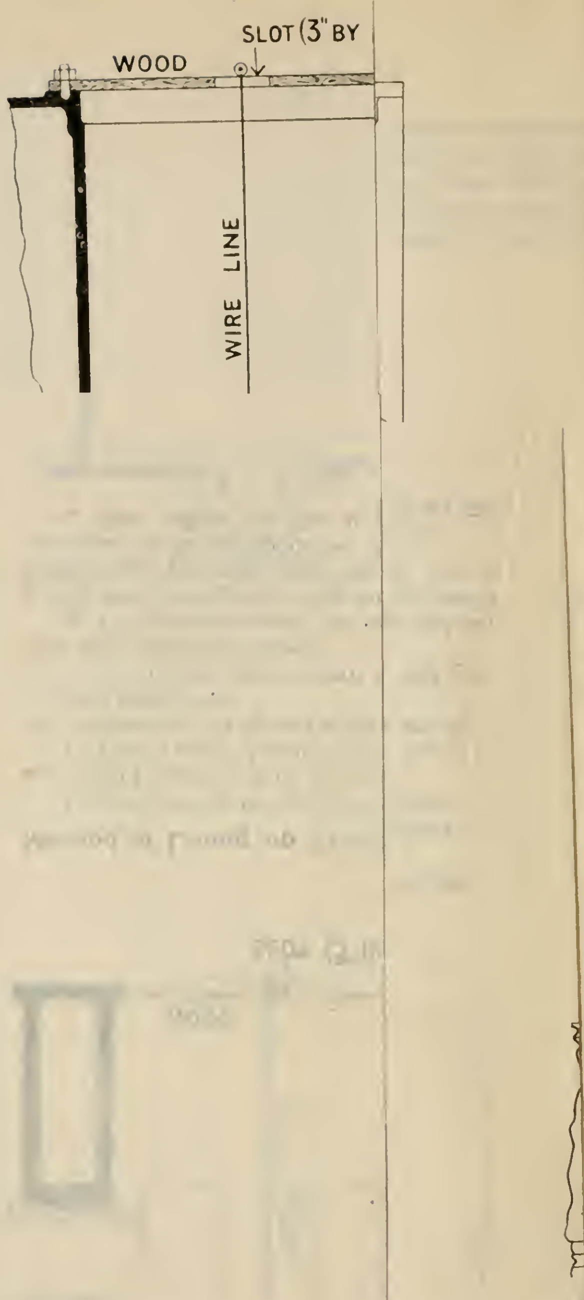
shaft is in place centres are fitted between the webs, with one edge central to the centre of the shaft, and with a line on each centre equal distance from each web. The cylinders are moved until the three lines hanging from the piston rod stuffing boxes are exactly in line with the marks on the shaft centre sticks and also in line with the centre of the shaft. The bores of the cylinders are also tested with a plumb rule, and it may be necessary to line up the feet of the cylinders to bring them plumb. If this be the case then it will entail machining or filing the head of the columns which show high, but if care has been exercised in setting up the columns it will only be a small



No. 47.—Method of securing False Face.

amount that will be required to be eased from the high point. The lines suspended from the valve spindle stuffing boxes are tested from a straight-edge, bearing on the column face, and this will show if the cylinder centres are parallel to the column faces, but it is mostly in engines having the cylinders separate that this operation is required. The cylinders being set, witness marks as before explained are chipped on the column heads, and the holes for bolting the cylinders to the columns are marked off. The cylinders are taken down and also the columns, which are bored on the top flanges, and any machining which is required to bring the cylinders plumb is done.

The columns are re-erected, cylinders put up and set to witness



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water tested to the required pressure, usually twice the working steam pressure, and are now ready for erecting on the top of the columns. In the stuffing boxes and in the valve spindle stuffing boxes, centres are fitted from which plumb lines are suspended, and if the crank-

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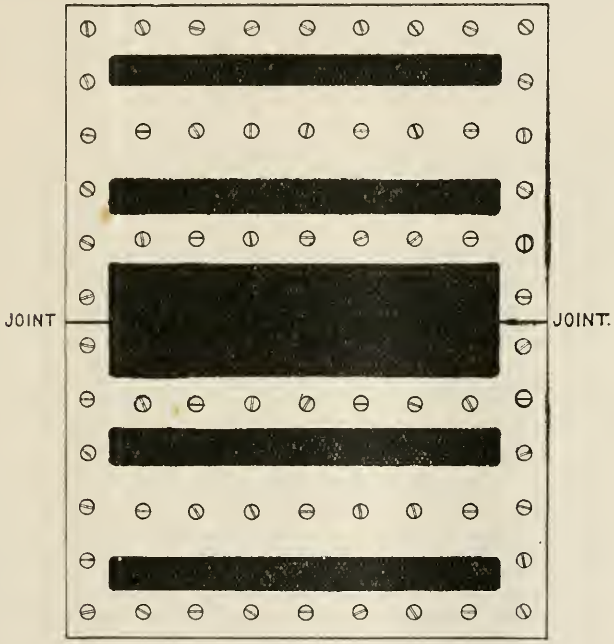

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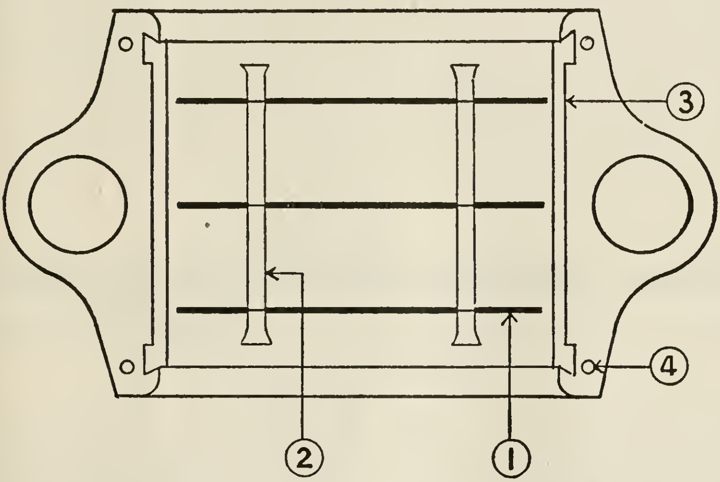
The columns are re-erected, cylinders put up and set to witness

marks, and centres retested. All being right, the holes in column heads are widened, bolts fitted. There are usually four fitted bolts in each column head, the rest being an easy fit. It is again preferable as before explained to use a fine piano wire instead of the plumb line, and also to extend the centring line up to the top of the cylinder, into which a centre is fitted, and the line set truly to this, and also to the bore of the stuffing box. As before explained every firm has its own method of carrying out these various operations, but the foregoing is a general description of the method employed in setting up the cylinders for a modern tramp steamer's engines. In many shops the cylinder feet are bedded down on the column heads, thus ensuring perfect work, but with the accurate machine work and the use of length sticks and length gauges, it is possible to get all the different parts of the machinery to come together with the minimum of hand work.

Training Connecting Rods and Fitting Running Gear.

The engines are now ready for the fitting of the gear, and the first operation is that of putting in the piston rods and pistons. The pistons are usually tried into cylinder and allowed to rest on the bottom of same. A length rod is applied from the centre of the crank-shaft up to the top of the piston and to the top of the cylinder. These points are marked off and the piston rods cut to length, allowance being made for the clearance required at each end of the stroke. The same operation is also carried out on the valve spindles. After piston rods are cut to length they are fitted into the pistons, that is the tapered part on which the piston rests is fitted up to a bearing. This operation being finished and the piston rod having been fitted into the crosshead, if the type of connecting rod is that of the double top end, the piston and rod are lifted and lowered into place in the cylinder. If the connecting rod is that of the single top end, the crosshead bearings being on the piston rod, then the piston rod requires to be lifted up through the stuffing box and through the piston, the piston rod not being passed on to the sling. The crosshead is put on to piston rod and set parallel to the column faces. Gauges are then lifted from the face of crosshead to column face and guide shoes machined and fitted. The connecting rods are now put into place, and trained. This operation is usually performed in the following manner:—The top end liners are taken out and the nuts screwed up, thus binding the top end. The piston should be lined up also so as to ensure its remaining central. The whole part, piston, piston rod, and connecting rod are now lifted up until the connecting rod butt is up clear of the crank-pin, and the distance between flange of bottom end bush and web is gauged at the four points to ascertain if the rod is hanging fair. At the same time a plumb rule is applied to the side of piston rod or crosshead and the rod brought into a true position. It is necessary while this operation is being carried out to be sure that the piston rod is exactly central in the stuffing box, and to ensure this temporary glands of wood or brass should be fitted. If the four points at the butt of the rods do not coincide, then it is necessary to ease the top end bush in such a manner that will bring butt of rod exactly fair. This being so, the bottom end of rod is now bound on the crank-pin, crank being on bottom centre, and the piston rod and crosshead are lifted up until clear of top end bushes. If the rod is true, then the crosshead will lower into place with equal clearance on each side of the top end brasses; if not, then the bottom end bush will require to be eased so as to bring rod central. In binding bottom end of rod on to crank-pin care should be taken



No. 49.—Taking "Leads" off Bottom End Bearing Bush.

1, Lead wire. 3, White metal.
2, Oil gutters or "reliefs." 4, Dowel pins to hold liners in position.

to have equal clearance on each side of bottom end bush and web. The crosshead is now bedded into top end bushes and top bushes leaded. After this operation the top end is again bound and the bottom end bush bedded on crank-pin. The crank is now turned to top centre and the crank-pin bearings leaded. The same operation takes place on the three connecting rods.

In carrying out this work on connecting rods of the single top end type, a good deal of extra work is entailed, as this type of rod is not so easy to overhaul. The crank-pin bush is bound on the crank-pin in a similar manner as before, and the piston, rod, and crosshead are lifted up; care is taken to ensure the connecting rod being central between the columns. The piston rod is lowered until the top bush is just clear of rod, and by applying a straight-edge to the inside cheek of the connecting rod, it can be seen if the connecting rod is fair to the piston rod ; if this is not so, then the bottom end bush will require to be lined up and a corresponding amount either taken off the sole of the top bottom end bush, or the bore of the bush scraped out until the rod is brought fair with the piston rod. The top end bush is bound in a similar manner as before described, and the crank-pin bush bedded upon the crank-pin. In some works it is usual to test the rods by swinging them, that is, after the top end bush has been bedded and leaded, the piston rod and connecting rod are lifted up, and the rod is swung from side to side so as to ensure that it falls back with its own weight and is in no way bound.

The crank-pin bushes are "leaded" in a similar manner as the top end, the crank is turned to the top centre, and the bottom half of the bush is lowered, lead wire, usually ·17 or ·16 B.W.G., is put in (Sketch No. 49), crossing the circumference of the bush in three places. The bush is pulled up against the crank-pin and nuts put on the bearing bolts. Usually the amount of liners required are given on the connecting rod drawing, and generally consist of one cast-iron liner from $1\frac{1}{2}$ to 2 inches thick, one $\frac{1}{8}$-inch brass liner, and three $\frac{1}{32}$-inch

No. 50.—Lead Wire as taken from Bottom End Bearings.

1, Lead wire ·02 thick at crown of bearing.
2, Lead wire heavier at sides.

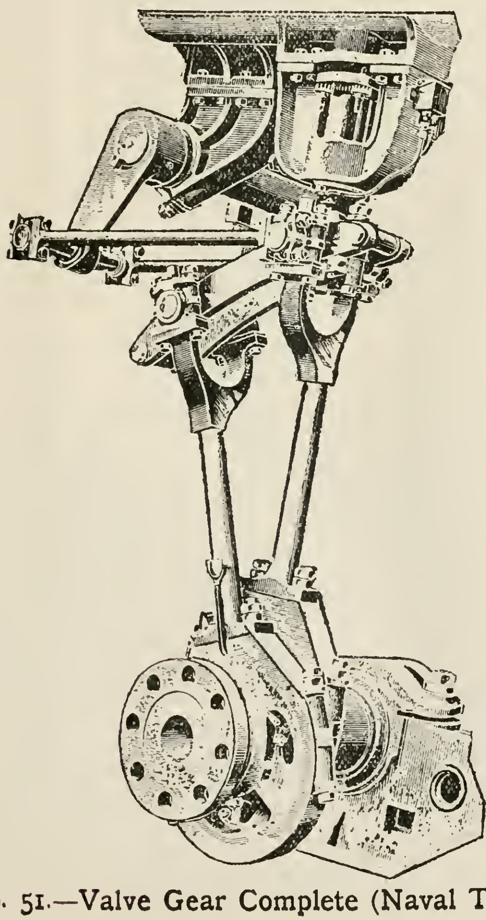
brass liners, and in some cases tin liners are fitted, but it is more general to fit all brass liners and to have the same amount of liners in all crank-pin bearings. The liners are in place while the operation of leading the bearings is being carried out, and the nuts on the bolts are hammered up, a mark being put on thimble point of bolt and on nut, care being taken to ensure both nuts being brought to an equal degree of tightness. The nuts are slackened back, and the bush lowered and lead examined. If they are not parallel and of the required thickness throughout, usually ·020 inch, then the white metal in the bush is scraped out and the operation repeated until the lead wire is of the required thickness. It is usual practice to ease the sides of the bush, so as to give a slightly heavier lead at this part, usually from ·003 to ·004 inch heavier than the lead taken from the crown of the bush. This result allows for expansion of the crank-pin when heated to a running heat, and also allows of a passage of oil (Sketch No. 50). It is not advisable to increase this clearance at the sides to too great an extent, as it takes away from the surface of the bush. The nuts on the bottom end and top end bearings are usually fitted with a lock-pin for binding the nuts, and split pins are also fitted to prevent the nuts slackening back. Oil tubes are led down the connecting rods to the crank-pin bush, with an oil cup on the fork of the rod. This oil cup is supplied with oil from a siphon box fastened to the side of the cylinders. The top end is supplied with oil in a similar manner.

The guide shoes in general practice are of cast iron faced with white metal, but in some designs only the ahead guide shoe is faced with white metal, the astern shoe being of cast iron throughout.

Reliefs are cut across the shoes (Sketch A, No. 20) and oil is supplied to the top of the column guide faces from the oil box which supplies the connecting rod bearings.

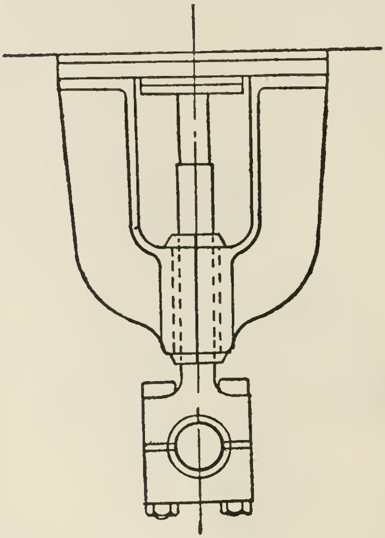
Training Valve Gear.

The valve gear is trained in a similar manner to the connecting rods, this is to ensure the gear being in line. The first operation is that of setting the valve spindle guide bracket. This bracket is



No. 51.—Valve Gear Complete (Naval Type).

fastened to the bottom of the cylinder, and has a brass guide bush either of square or round bore, through which the valve spindle passes. The spindle is put up into place and the gland put in, and the spindle supported inside the valve casing. The bracket is put in position, and bound on to the valve spindle. The spindle is set parallel to the valve face, or to the bore of the valve liner if it is an H.P., and a plumb rule tied on the body of the spindle. This being found plumb, the holes for bolting the bracket to cylinder are marked off and bracket bored. Usually the holes are widened and fitted bolts put in. The bore of the gland is tested by means of feelers to ensure



*** No. 52.—Guide Bracket and Valve Spindle.**

the spindle being central to the gland. The bracket being fixed, the quadrant is put into place and the eccentric rods and drag links.

All gear being assembled, the ahead eccentric rod is first dealt with. The bearings on the top end of the rod are bound to the pins of the quadrant, and the end of the quadrant lifted until the butt of the rod is clear of the studs on the eccentric strap. It is now lowered and the holes are tested to see that they are fair to the studs. This being so, the astern eccentric rod is dealt with in a similar manner. The eccentric straps are now bound on the pulleys by taking out the liners and screwing up the bolts, and the quadrant is lifted up out of the bearings at the top of the rod. If this is fair then the rod is true, but if it is found that the bush is not in line with the sides of the quadrant, then the butt of the rod is lined up until it is fair, and the amount of the lining is machined or filed off according to the amount.

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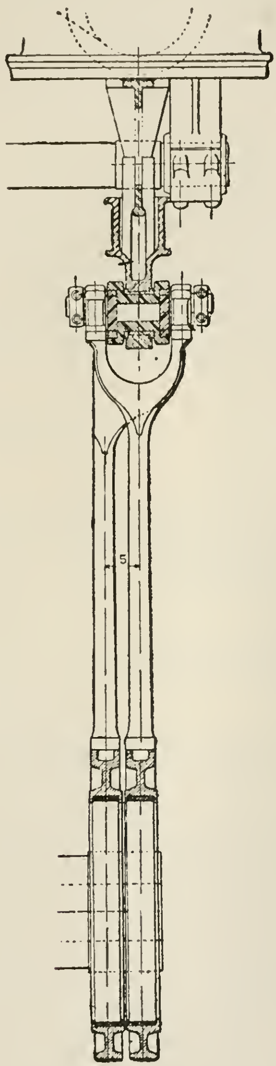
Both eccentric rods now being fair, the drag links are dealt with. These are first disconnected, and the distance from the quadrant pins

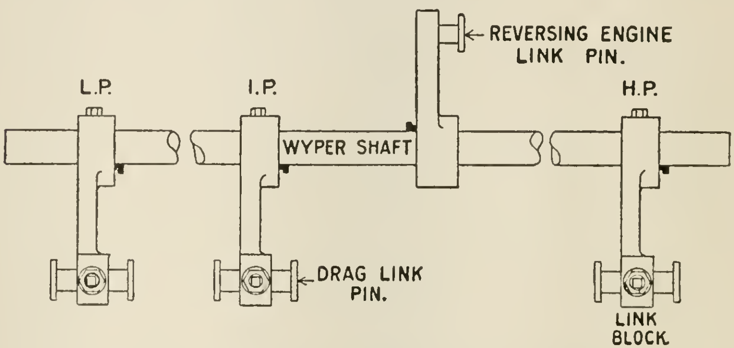


No. 53.—Eccentrics and Rods.

The rod in direct line with the valve spindle is usually the "ahead" and the one out of line the "astern."

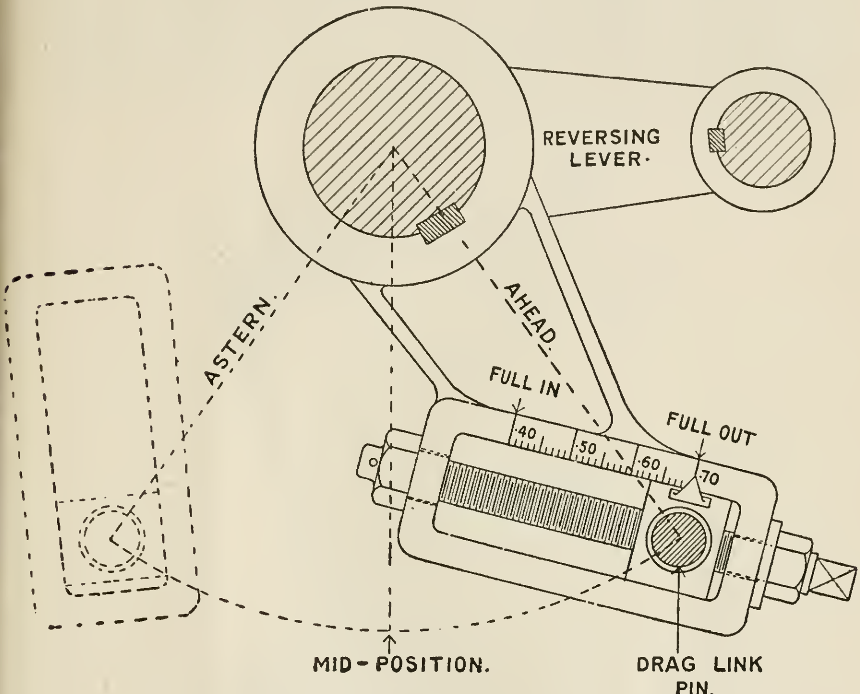
to the pins on the wyper shaft is tested, so as to ensure the quadrant being square and parallel to the wyper shaft. The drag links are now bound on the pins of the lever, and the end which couples to the quadrant pins tested for being fair. If not true, then the base of the brass is filed or machined until the link is brought in line. The same method is carried out with the other end, the quadrant end of the link being bound and the end which couples to the wyper shaft lever being brought fair. The eccentric straps are bedded on the pulleys and are usually left ·025 inch easy for same. In most of engines the eccentric straps are usually of steel lined with white metal, and are of broad surface. The wyper shaft (Sketch No. 54) is supported on brackets bolted to the columns and the bearings are usually cast iron, but in the better class of engine brass bushes are fitted. On the wyper shaft are four levers, three connecting to each of the valve gears, and the other one being connected to the reversing engines.

The three levers connecting the valve gear are fitted with adjustable bearing pins, to which the drag links are coupled. The use of these pins is for the purpose of what is usually termed "linking



No. 54.—Wyper (or Reversing) Shaft.

up," or altering the cut-off of the valve (Sketch No. 55). If the valve is arranged to cut-off at three-quarters of the stroke, then when the pins are full out, and the eccentric rod exactly in line with the valve spindle, this point of cut-off will be got. By screwing in the block in the wyper shaft lever the cut-off of the valve will be altered, as the eccentric rod is not in line with the valve spindle, and so the travel of the valve will be decreased. The link lever is usually graded to represent the different positions of cut-off as shown in Sketch No. 55. The levers on the wyper shaft are keyed on, and have also a lock pin which passes through the lever and into the body of the shaft, thus ensuring that no movement of the lever will take place. The lever to which is connected the reversing engine is usually fixed in a similar manner, and if the reversing gear is that of makers who specialise in this gear, then two levers may be fitted connected by two links to the piston rod of the reversing engine. The position of the main reversing lever is determined from the drawing, and is so fixed to give the movement of the valve gear from ahead to astern position. If the reversing gear is that of the all-round type, it is general to have the link on the top centre of the reversing wheel when gear is full ahead, that is with drag links and levers in a level position (Sketch No. 55). In some ships, tell-tale gear is fitted: this is a



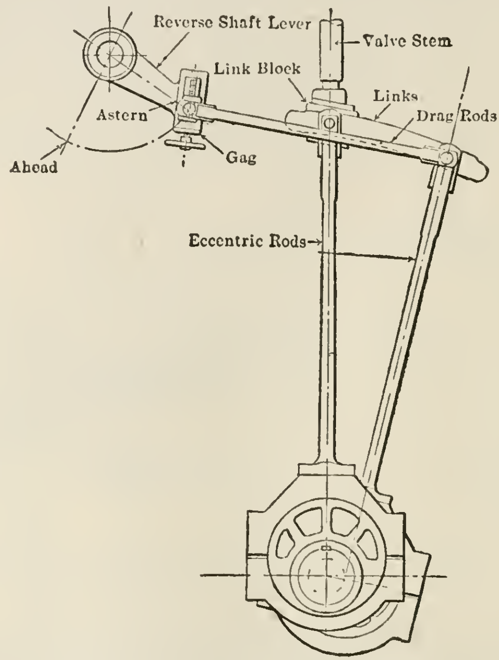
No. 55.—Reversing Gear Bell Crank.

Showing expansion link slot and block for linking up. The adjustment of the drag link pin block in the slot (by means of the screw and nut) controls the cut-off as indicated by the cut-off grades marked. The block as shown is "full out" giving a cut-off of ·70 stroke or 70 per cent.; if the block is placed in "full in" position the cut-off is then ·40 or 40 per cent. The linking up referred to is only possible when the gear is in ahead position, as when set for "astern" the slot is nearly vertical and the linking up effect then becomes inoperative.

small quadrant fixed on the wyper shaft, and from this gear is led to an index plate on the front of the forward column, and indicates which way the valve gear has been moved, and in some cases this gear is also connected to the bridge, thus showing to the officer in charge if his order transmitted by the engine-room telegraph has been properly carried out (Sketch No. 57).

Cylinder Clearances and Setting of Valves.

There are several methods of taking cylinder clearances, but the method now described is most generally employed. In taking the top clearance the crank is turned on top centre, and in this position strips of clay or putty are laid on the body of the piston and also on the top of the piston rod, and on the piston rod nut. The cover is now put on, and a few nuts screwed up to ensure that the cover is close down on the joint. The cover is now taken off, and the thickness of the clay at the various points measured, thus giving the



*** No. 56.—Valve Gear in "Astern" Position.**

Notice that the expansion slot is in a vertical position and therefore practically non-operative as regards linking up.

clearance between the cover and piston at top of stroke. While crank is in this position a mark is chipped on the guide shoe column face plate, and a fine chisel cut put across both parts (Sketch No. 58), thus showing when crank is on top centre. This position has been found previously either by plumbing the crank web, or by using a trammel from the side of the column. In finding the centre or top of stroke in this manner the method used is to turn crank up until it is nearly at top, and in this position a trammel is applied from a

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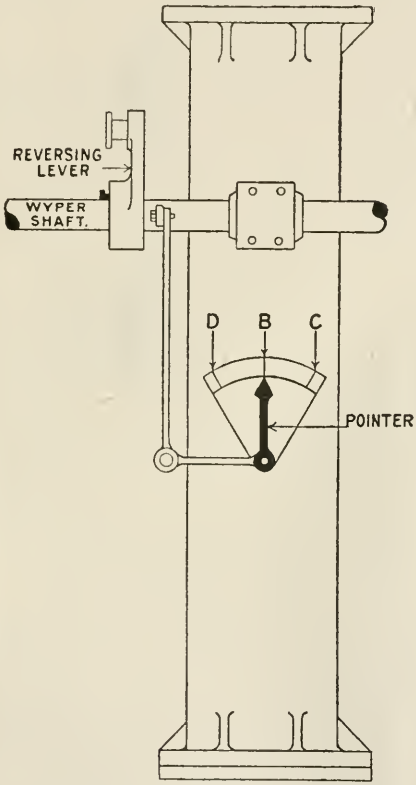
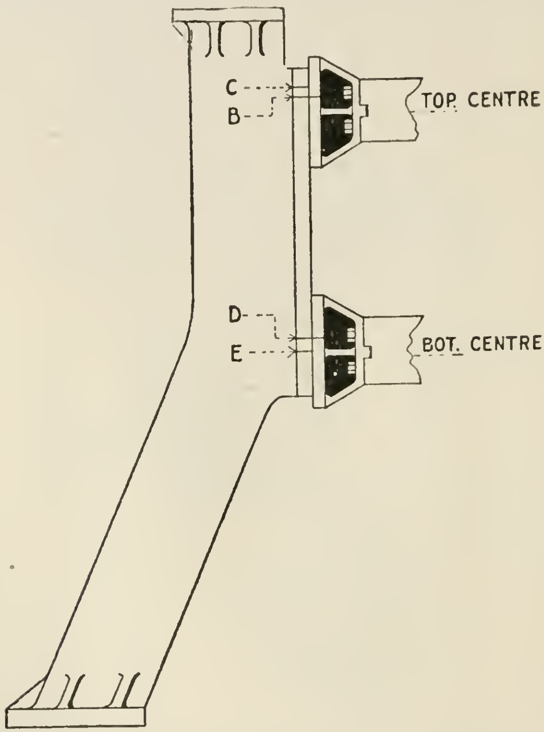
point on the column, and an arc drawn on the top of the web. At the same time a mark is drawn across the edges of guide shoe and column face. The crank is now turned over the centre, until the mark on the guide shoe and column face is again in line. In this position the crank is again marked with the trammel from the same point as before, and by bisecting the two marks a point is found, and upon turning the crank back until the trammel fits between this mark and the point on the column, the crank is on the top centre.



No. 57.—"Tell-Tale" Gear.

D, Engine running ahead. C, Engine running astern.
B, Gear in mid-position.

A mark is now put across the guide shoe and column, and the amount of clearance which has been ascertained by means of the clay or putty is marked above this mark, thus showing the amount of clearance originally allowed for. The bottom clearance is found by turning the crank to the bottom centre, and either plumbing the crank or acting as before described; the mark is also put on edge of guide shoe and column face. The top end is now disconnected, and the piston and piston rod lifted up, the bottom top end bush is taken out, and the piston and rod lowered until it rests upon the bottom of the cylinder. Another mark is now put on the column in line with the mark previously put on the guide shoe, and the difference between the two marks represents the clearance when the crank is on bottom centre. In engines of good design small brass plates are fitted on which these marks are installed, as it has



No. 58.—Method of Testing Piston Clearance.

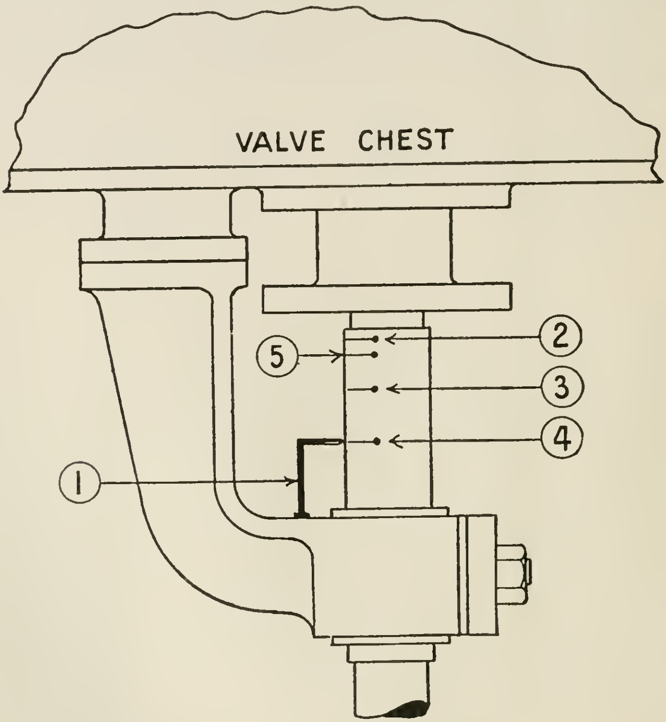
B, Piston on top centre. } Top
C, Piston touching cover. } clearance.
D, Piston on bottom centre. } Bottom
E, Piston touching cylinder bottom. } clearance.

been found that the marks on the cast iron become filled up with paint, and in some cases where there was a slight leakage of the circulating water in the column guide plate, the marks were badly corroded, making it impossible to determine the actual clearance without proving same. If it is found that the required clearance as in drawing has not been attained, it will be necessary to machine some of the parts, until this requirement is met. If the top clearance is small, and the bottom clearance large, then the piston can be let up on the taper of the piston rod, which means that the length of the piston rod will be reduced. If the clearance should be the other way, small at bottom and large at top, then the bottom of the cylinder will be examined to ensure there being no lumps on same—if so, these lumps will be chipped off; but should this not be sufficient, it will be necessary to fit a heavier or thicker bush either in the top end or bottom end of connecting rod. It may be the case that the piston would stand machining, and if so, then it can be dealt with. The clearances in reciprocating engines are usually as follows:—

H.P. top - - - -	$\frac{1}{4}$ to $\frac{3}{8}$ inch.
H.P. bottom - - - -	$\frac{3}{8}$ to $\frac{1}{2}$,,
I.P. top - - - -	$\frac{3}{8}$ to $\frac{5}{8}$,,
I.P. bottom - - - -	$\frac{5}{8}$ to $\frac{3}{4}$,,
L.P. top - - - -	$\frac{5}{8}$ to $\frac{7}{8}$,,
L.P. bottom - - - -	$\frac{3}{4}$ to 1 ,,

The above clearances allow for wear down of the connecting rod bearings.

The operation of adjusting or setting the slide valves is carried out in the following manner:—The valve gear is put in ahead position, that is, the ahead eccentric rod is brought in line with the valve spindle. It is first required to find the valve travel: this amount is given on the drawing, and to ascertain if it is correct, the engine is turned round until the eccentric pulley is at full throw on the top. This is found by having a centre line through the pulley, and also a centre line on the eccentric strap, both centre lines being vertical. The engine is then turned round until the centre line on the large part or throw of the eccentric pulley is in line with the centre line on the eccentric strap; and when in this position a line is put on valve spindle, either by using a trammel from a fixed point on the bottom of the cylinder or valve spindle bracket, or by simply drawing a line across the spindle under the gland. The engine is again turned until the centre line on the small part of the eccentric pulley is in line with the centre line on eccentric strap (Sketch No. 59). Another line is put on valve spindle as before, and the distance between is measured, representing the travel of the valve: if the distance as measured does not correspond to drawing, then the link block is moved until the required travel is arrived at, which means that the eccentric rod is exactly in line with the valve spindle usually termed "Link in line." As before explained, the moving of the link block in or out decreases or increases the travel of the valve. After travel is adjusted, the next operation is to find the leads of the valves. The H.P. engine (piston valve) is turned on to the top centre, and the space between the top inside edge of the valve and the *bottom* of the port in the valve liner is measured by means of a small wedge-shaped piece of wood. The engine is now turned to bottom centre, and the space between the *bottom* inside edge of the valve and the top edge of the bottom port is measured, and the amount obtained represents the lead of the valve, with crank on bottom. If these figures do not coincide with drawing figures, that is, if lead on top is too much and too little on bottom, then the washer on which the valves sit will require to be reduced, or if the leads were *vice*

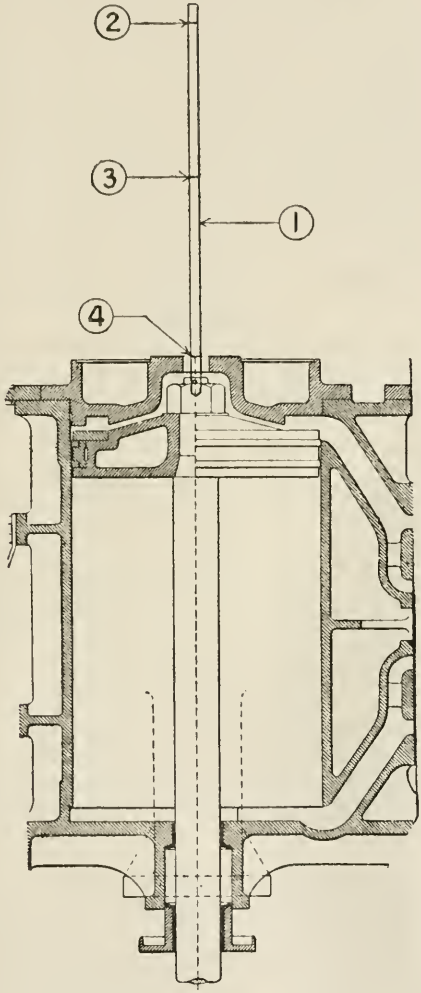


No. 59.—Valve Setting.

1, Trammel.
2, Mark for valve at "bottom."
3, Mark for valve at "mid-travel."
4, Mark for valve at "top."
5, Mark for valve at "top lead."

versa, then a thicker washer would be required, or the eccentric rod lined up. The same operation is carried out on the three engines.

Points of cut-off and port opening, and in some special cases the points of admission and compression and release, are also taken. The point of cut-off is obtained by starting with the engine on top centre, and turning in an ahead direction. The valve is watched as it goes down and opens the port to full port opening, then starts to return, and at the point where it is edge and edge with the port on the valve face the engine is stopped, and the distance it has travelled is measured from the mark on the column face to the mark on the guide shoe. That distance is the point of cut-off, and may be ·70 of the stroke, or as designed. Another and more exact method



No. 60.—To Measure the Cut-off.

1, Wood baton.
2, Mark for "bottom centre."
3, Mark for "cut off."
4, Mark for "top centre."

is by the use of batons. These batons are marked off, one for the valve face and one for the valve (Sketch No. 60), showing the ports in the valve face and the edges of the valve. To set these sticks the engine is put on top centre, and one baton is fixed to the top of the valve spindle, the other to the top of the valve casing. By setting the batons to correspond with the top lead which was found by actual measurement, then all other points will be found to be exactly correct, providing the marking-off of the batons was carefully done. A baton is also fixed on to the end of the piston rod and a line put on with the crank on top centre, and the engine turned until by observing the valve closing the port, as shown on the batons, the engine is stopped and another line put on the stick which is fastened to the piston rod, and the distance between these marks represents the point of cut-off. The other points are arrived at in the same manner, the explanation of the terms being as follows:—

Admission.—Valve just edge to edge with port in valve face or liner.

Lead.—Amount valve is open for steam with crank on centre.

Port Opening.—Greatest amount that valve opens port to steam.

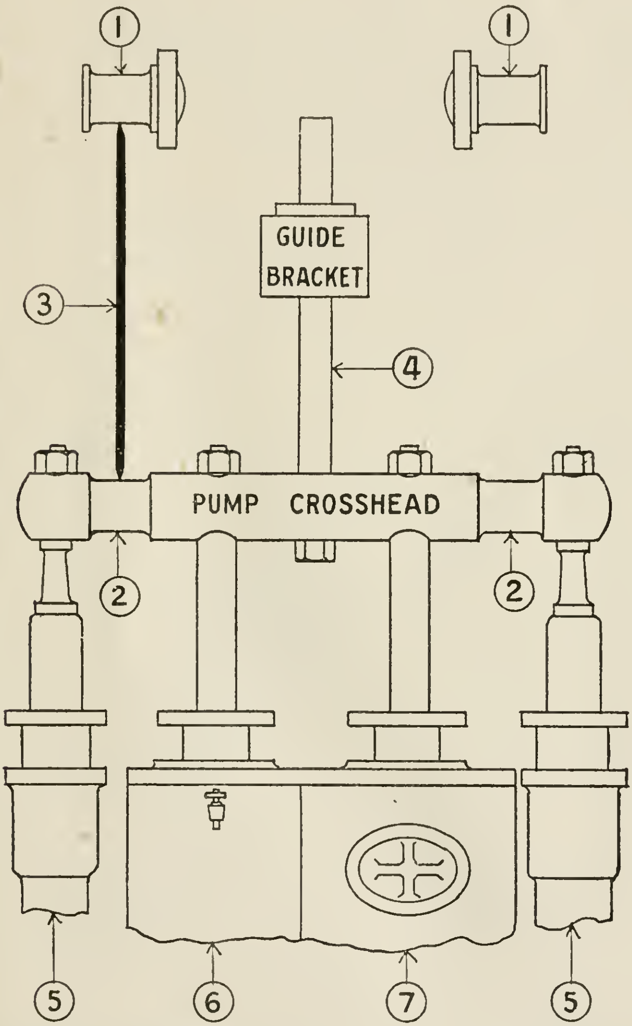
Compression.—When exhaust edge of valve is in line with edge of port closing same to exhaust.

Release.—When exhaust edge of valve is just edge to edge with port opening to exhaust.

The valve gear is now brought to astern position, that is, with the astern eccentric rod in line with the valve spindle. The usual points which are taken with astern gear are that of lead and travel, which are found in the same manner as before described. The link blocks, as before explained, are graded, and this is done by putting the eccentric rod out of line with the valve spindle and particulars taken of the valve settings, which gives decreased travel, earlier cut-off, and increased lead. These points are for the guidance of the engineer in charge, and through judicious use of this gear the most economical working of the engines may be arrived at.

Connections on Cylinders, Pumps, &c., and Dismantling Engines and Closing up Cylinders.

On each of the cylinders drain cocks are fitted, one on the cylinder and one on the valve casing, and pipes are connected to these cocks leading to the bilges, the cocks being operated by gear led down to the bottom platform in the engine-room. Indicator cocks, one on top and one on bottom of cylinder, are also fitted as well as escape valves to allow of any excessive pressure in cylinder or casing being liberated. A balance cylinder and piston is usually fitted on the L.P. valve, so as to take the weight as far as possible off the valve gear (see page 202). The bottom of the piston is open to the steam in the valve casing and the top side is in a vacuum, being connected by piping to the condenser. There are several patent balance cylinders on the market, such as Joy's, but the principle is similar to that described. On the H.P. valve casing the stop valve for admitting steam to the engine is usually fitted,



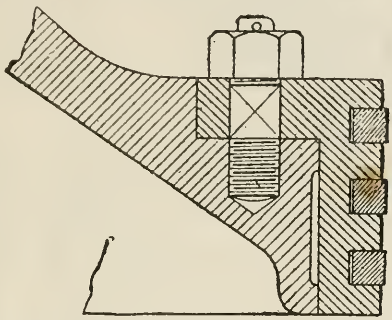
No. 61.—Trammelling Pump Links for Wear Down.

1, Lever pin.
2, Pump crosshead blocked up on cover with glands fair and free.
3, Trammel distance.
4, Guide rod of crosshead.
5, Feed pumps.
6, Circulating pump.
7, Air pump.

and connections are also led from this valve to the H.P. jacket and to the I.P. valve casing through a valve, termed an impulse valve (see page 183). This valve is used while heating up cylinders, and also for assisting in smartly moving the engines while manœuvring. The gland on the H.P. engine is usually packed with patent packing or with metallic combination packing, the I.P. gland is sometimes packed in a similar manner, and the L.P. gland with soft asbestos packing. The valve spindle glands, in the case of H.P. and I.P., are also fitted with metallic packing and the L.P. with soft packing. In the majority of engines now constructed separate pumps are fitted, but the type of engine for cargo steamers usually has the pumps worked off the main engine. These pumps consist of air pump, circulating pump, two feed pumps, and two bilge pumps. These are operated by means of levers connected to the I.P. engine by means of drag links. The levers are supported on a double bearing at the back of the I.P. column, and are usually of two steel plates with bosses between, and riveted together. The pins to which the drag links are connected are riveted into the boss on the levers. At the other end the drag links which connect to the pump crosshead are fixed. The levers are bedded down in their bearings and the covers leaded in a similar manner to that of the other bearings on the engine. The drag links are trained and bedded on to their respective pins, and the clearance of the air and circulating pumps are taken: to get the clearance, the main engines are turned to position with crank on top centre and mark put on air pump and circulating pump rod, or by applying a gauge between crosshead and pump covers. The engine is again turned to bottom centre, and another mark put on—the distance between represents the travel of the levers; and by disconnecting the drag links and lowering the pump bucket until it rests on the bottom, and putting a mark on, the difference between the travel mark and this mark will represent the bottom clearance. The pump bucket is then lifted up to the top and pulled by means of tackle until it touches the head valve, and a mark put on, and by comparing the two marks the clearance at top is arrived at. The feed and bilge pumps are tested for clearance when the pumps are at bottom stroke. The nuts on the spindles through the crosshead are slackened and the plunger lowered until it rests on the bottom, a mark put on, or gauged by callipers between the shoulder on spindle and crosshead; the distance representing the clearance will be found by putting on a mark from the same point as before when the plunger spindle is screwed up in crosshead. The usual clearance for air pumps, feed and bilge pumps is as follows:—

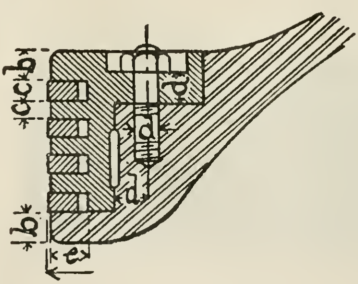
Air pump - - -	top, $\frac{1}{2}$ inch, bottom, $\frac{3}{8}$ inch.
Bilge pump - - -	bottom, 1 to $1\frac{1}{2}$ inches.
Feed pump - - -	bottom, $\frac{3}{4}$ to $1\frac{1}{4}$ inches.

It will be noticed that the air pump clearance is most on top; this is explained by the distance between lever pins and crosshead being decreased as wear on the drag links is taken up. The engines are now ready for dismantling, and the cylinders are taken down, also columns. Crank-shaft is lifted and cleaned and oiled and put back into place in the soleplate bearings. All hard bits or heavy bearings



No. 62.—Cast-Steel Piston.
With Ramsbottom rings fitted into junk ring to allow of removal.

in crank-pin bushes and top end bearings are eased, and all gear prepared for transfer to the ship. The cylinders are thoroughly examined to ensure that there is no sand or dirt in any of the ports, and this being so, they are ready for closing up. The piston rods



*** No. 63.—Cast-Steel Piston.**
Fitted with Ramsbottom rings.

For a cylinder 20 inches diameter the proportions are :—

$b = 1\frac{1}{2}$ inches. $d = 1$ inch.
$c = \frac{3}{4}$ inch. $e = 1\frac{3}{16}$ inches.
2 or 3 rings to be fitted.

are put into the pistons and hard hammered up, locking cutters or split pins fitted, and the whole is lifted and lowered into the cylinder. The interior of cylinder has been beforehand rubbed over with cylinder oil to prevent rusting during the time that machinery is stationary. The H.P. piston is usually packed with Ramsbottom

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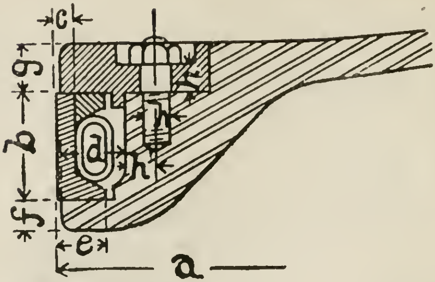
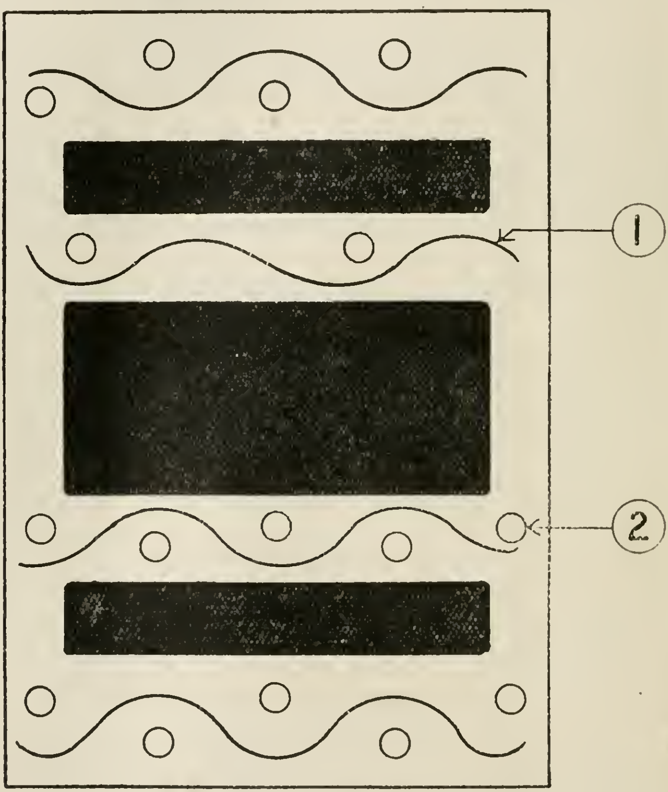


*** No. 64.—Buckley Type Piston Ring.**

For a piston 24 inches diameter the proportions are as follows:—

$b = 2\frac{1}{8}$ inches. $c = \frac{11}{16}$ inch. $d = 1\frac{11}{16}$ inches. $e = 1\frac{1}{16}$ inches. $f = 1\frac{1}{4}$ inches. $g = 2\frac{1}{4}$ inches. $h = 1$ inch.



No. 65.—Cylinder Valve Face.

Gutters and holes drilled to reduce wear.

NOTE.—The holes, 1 inch diameter, $\frac{1}{16}$ inch deep, are bored out with a flat nosed drill.

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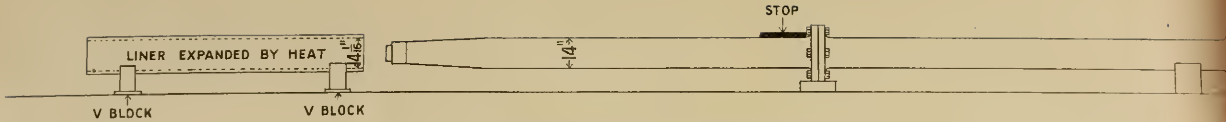
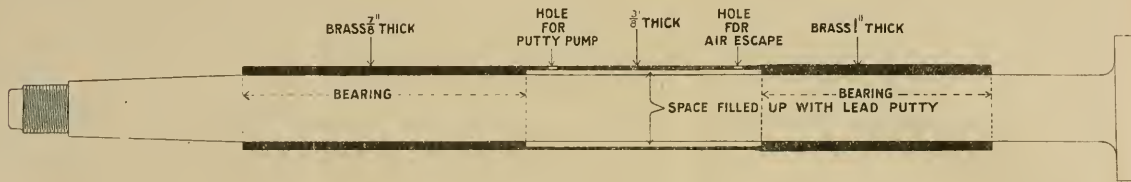
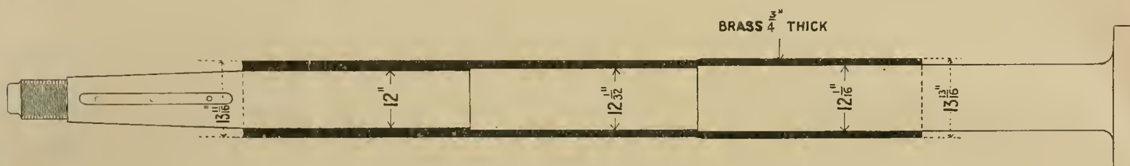


No. 66.—Method of Shrinking on Propeller Shaft Liners.



No. 67.—Propeller Shaft Continuous Liner.

(With Thickness Variation).



No. 68.—Propeller Shaft Continuous Liuer.

"Stepped" and farced on by Hydraulic Pressure (110 tons).

rings and segment packing rings (Sketch No. 63), and these are assembled and junk ring put on and nuts screwed up. The nuts on the junk ring are prevented from slackening back by means of either split pin above the nuts if square-necked studs are fitted, or by means of a guard ring which bears against nuts or pins, and is itself kept in place by being fitted on square-necked studs having split pins through the nuts. The other two cylinders are closed up in a similar manner, the I.P. piston packing rings being same as H.P., and the L.P. being either one of the patent packing rings or the packing ring with coach springs or spiral springs pressing it out against the cylinder wall. The valves are dealt with in a similar manner, the H.P. being a piston valve, I.P. a single-ported slide valve, and L.P. double-ported slide valve. Previous to putting the valves in place, oil gutters are cut on the face (Sketch No. 65); this assists to reduce friction, and in the case of the H.P. grooves are turned on the rings which join the valves. The cylinder and casing covers are jointed with asbestos joints, and in some cases asbestos tape, glands are packed, and all openings to interior of cylinders or casings closed up. The pistons and rods are supported, so that when lifting cylinders the rods will not lower to the bottom of the cylinder; the valve spindles and valves are also supported, and the whole three cylinders are now ready for transport to the ship.

Propeller Shaft Liners.

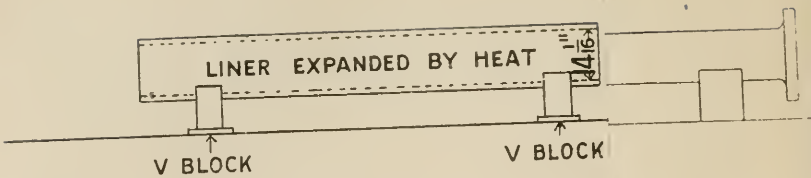
Propeller shafts are brass lined from end to end to prevent galvanic action taking place between the brass liner and steel of the shaft. The liners are fitted in two styles—

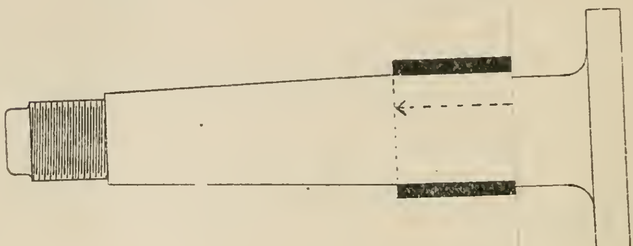
1. Shrunk on hot.
2. Forced on cold by hydraulic ram pressure.

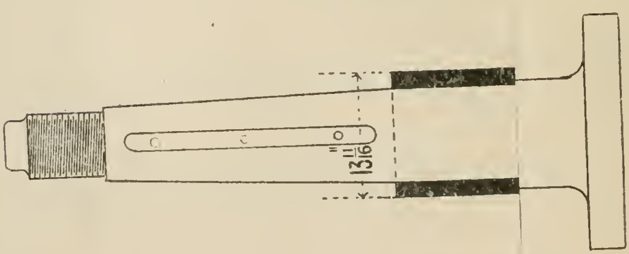
1. **Shrinking on** (Sketch No. 66).—The shaft is supported by bolting up to one of the tunnel lengths, which leaves the whole length free to receive the liner. The liner is then heated either by gas burners or by a fire built underneath, and after sufficient expansion has taken place the liner is drawn over the shaft by means of blocks and chain tackle. When the liner cools down the contraction resulting is sufficient to lock the liner to the shaft, screwed pins being seldom used in present practice, as in quite a number of cases the pins have been found to slacken back and come out of place. Before shrinking on, the liner is bored out about $\frac{1}{500}$ less in diameter than the shaft, therefore for a 12-inch shaft the inside diameter of the liner will be 12 inches $-\frac{12}{500}$ inch $= 11{\cdot}976$ inches, say $11\frac{62}{64}$ inches full.

NOTE.—If the liner sticks when being drawn on it may be forced on by pressure at the end, or expanded again by building a fire underneath.

2. **Forced on Cold** (Sketch No. 68).—In this method the liner is stepped to three diameters, the difference at each length being $\frac{1}{32}$ inch. The forward and after diameters should be a bearing fit, but the centre







page 55

rings and segment packing rings (Sketch No. 63), and these are assembled and junk ring put on and nuts screwed up. The nuts on the junk ring are prevented from slackening back by means of either split pin above the nuts if square-necked studs are fitted, or by means of a guard ring which bears against nuts or pins, and is itself kept in place by being fitted on square-necked studs having split pins through the nuts. The other two cylinders are closed up in a similar manner, the I.P. piston packing rings being same as H.P., and the L.P. being either one of the patent packing rings or the packing ring with coach springs or spiral springs pressing it out against the cylinder wall. The valves are dealt with in a similar manner, the H.P. being a piston valve, I.P. a single-ported slide valve, and L.P. double-ported slide valve. Previous to putting the valves in place, oil gutters are cut on the face (Sketch No. 65); this assists to reduce friction, and in the case of the H.P. grooves are turned on the rings which join the valves. The cylinder and casing covers are jointed with asbestos joints, and in some cases asbestos tape, glands are packed, and all openings to interior of cylinders or casings closed up. The pistons and rods are supported, so that when lifting cylinders the rods will not lower to the bottom of the cylinder; the valve spindles and valves are also supported, and the whole three cylinders are now ready for transport to the ship.

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NOTE.—If the liner sticks when being drawn on it may be forced on by pressure at the end, or expanded again by building a fire underneath.

2. **Forced on Cold** (Sketch No. 68).—In this method the liner is stepped to three diameters, the difference at each length being $\frac{1}{32}$ inch. The forward and after diameters should be a bearing fit, but the centre

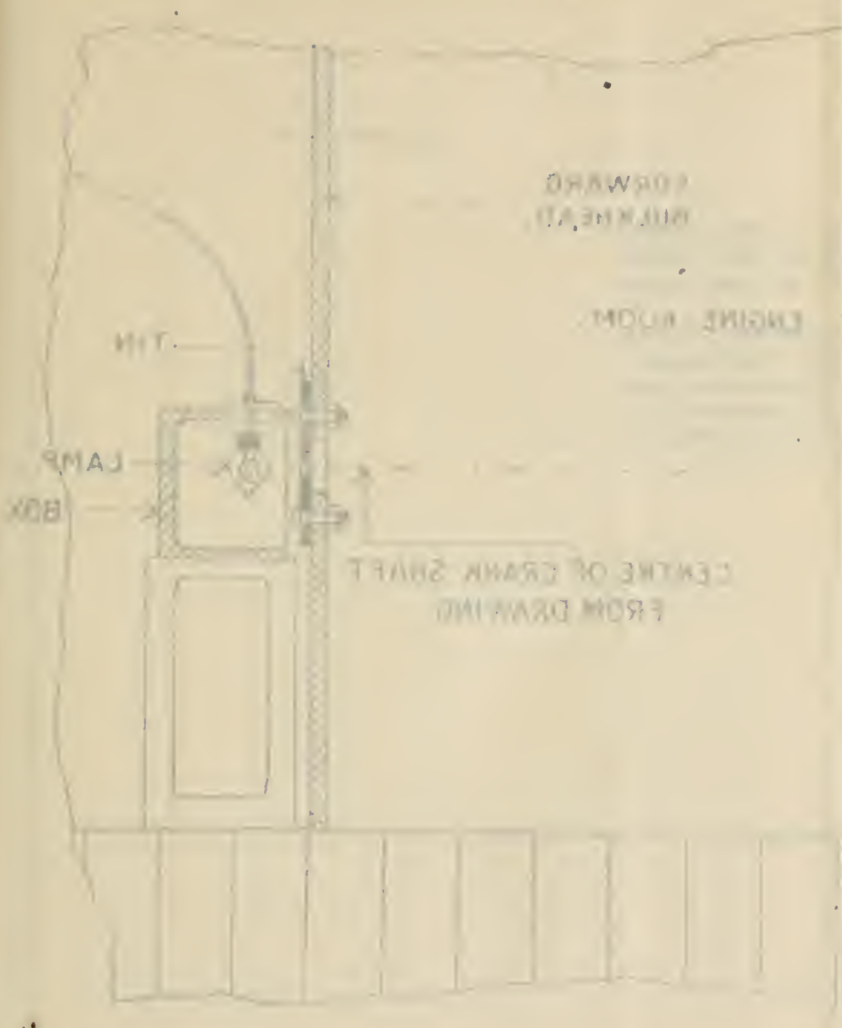
length need not be so; as in the "shrinking on" method the liner is bored out a trifle less in diameter than the shaft at each "step" and the liner is then forced on over the end of the shaft, by a hydraulic ram exerting a pressure of about 110 tons. Notice that the ram pressure only requires to be exerted for one of the stepped lengths, as the three fit simultaneously.

Variation in Liner Thickness (Sketch No. 67).

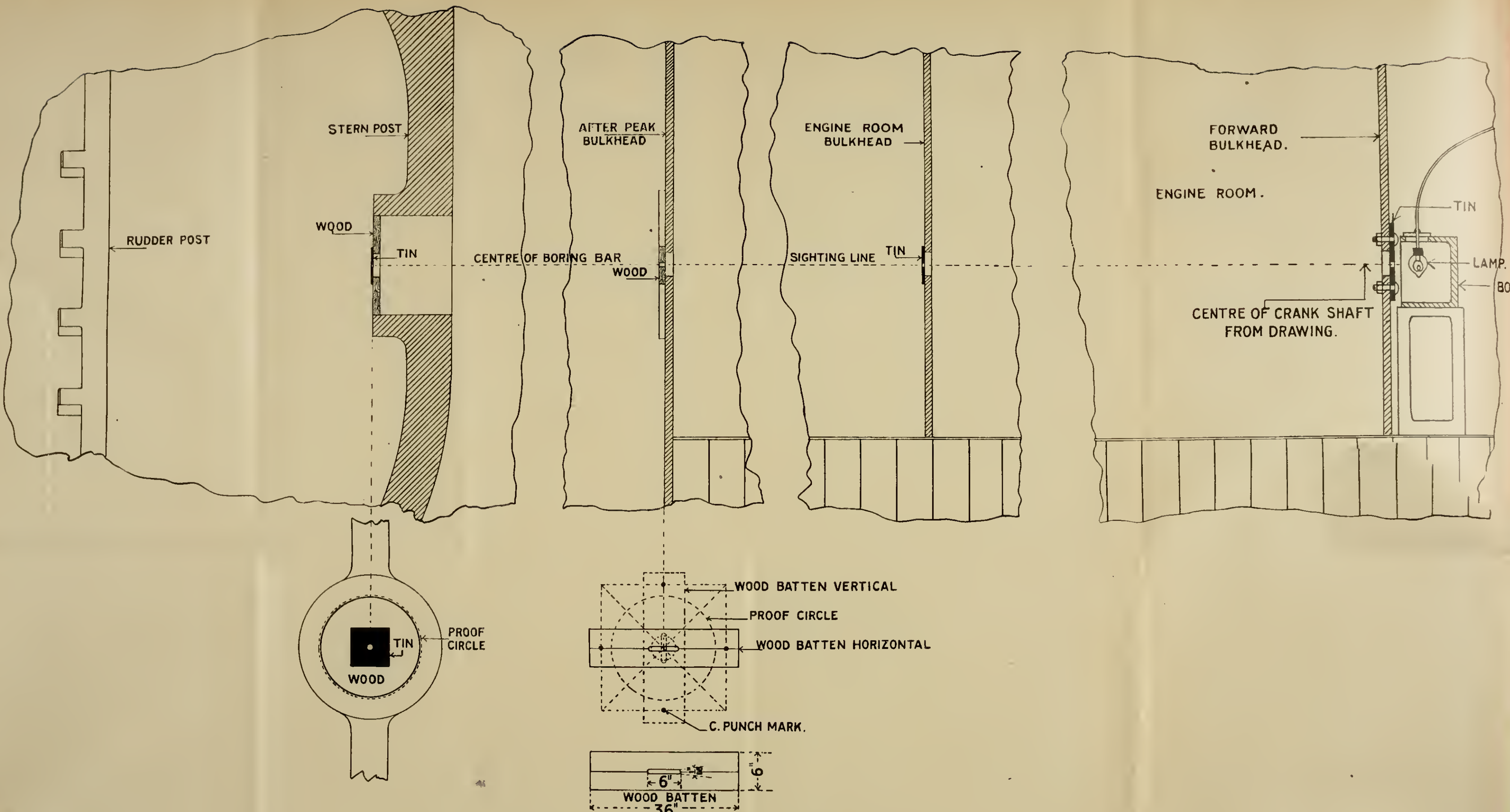
Occasionally the liner is cast in three thicknesses as shown in the sketch, being thickest at the position of the forward bearing, next at the after bearing, and least of all at the centre where the shaft does not bear at all. Two holes are bored in the liner at the centre length and putty is forced in by means of a pump through one of the holes to fill up the clearance space inside, the other allowing for the escape of air: these holes are afterwards filled up by means of screwed pins riveted over.

Marking off Ship for Boring out—Propeller Shafting and Thrust Block (Sketch No. 69).

The part of the ship's hull through which the stern tube passes is bored out to a size so as to ensure the stern tube being a good fit in same, and absolutely watertight. The method of marking off the stern post and bulkheads for boring out is as follows:—In the centre of the hole which passes through the stern bracket a wooden disc is fitted completely filling the hole. Upon this piece of wood the centre of the shaft from keel as given in drawing is marked, and the centre of the present bore of the stern frame or bracket is taken. A small hole $\frac{1}{32}$ or $\frac{1}{16}$ inch in diameter is bored through the wood at these points (Sketch No. 69). In the engine-room at the forward bulkhead, the height of centre of shaft from keel is marked off, and also the centre of the ship athwart-ships is marked. A small hole is also drilled here. At the back of this hole a lighted candle or electric lamp is fixed and the operator if by looking through from the stern to the engine-room through the small hole in the wooden disc can see the light, then that point will give the centre for boring out. If the light is not seen, then it will be necessary to shift centres, and to facilitate this, a sliding or movable centre is used, so that it can be moved about until light is visible, the light now being seen from aft end of tube bearing to engine-room. Another centre is fitted in engine-room bulkhead, and the light picked up. A similar centre is fitted in the after bulkhead in the tunnel, and light sighted. The whole is again sighted from the aft end, and light being seen, a centre is put in holes in discs and circles drawn around equal to the boring out size; proof marks are also put on so as to test boring bar. The bulkhead in forward end of tunnel is marked off in a similar manner, and is bored out for a bulkhead gland. At various distances throughout



bearing block teakwood fillers are fitted, bringing bearings up to required height (Sketch No. 70). At the forward end of the tunnel the thrust block is situated (see page 182): this block is made up of a number of shoes, as the design may require. The shoes are of cast iron, lined with white metal, having gutters cut on each side, oil being supplied from an oil box cast on each shoe. Water service connections are also made so that a water circulation takes place throughout the interior of the shoe. The block itself is of cast iron, and is rigidly bolted to the ship's frame, this part of the ship being specially



No. 69.—Method of Sighting for Line of Shafting and Boring Stern Post.

1. At distance up marked on drawing as "shaft centre" and at centre athwart-ships, cut a hole (say 1 inch diameter) in the engine-room forward bulkhead, and to this fix a piece of tin; punch a hole $\frac{1}{16}$ inch diameter at centre and place a small electric lamp in a box in the position shown.

2. Block up the stern post hole (previously bored out to *less* than the required diameter) with a wooden disc, in which cut out a 1-inch hole; on this pin a sheet of tin with a $\frac{1}{16}$-inch hole at the dead centre (see Sketch), and from the centre scribe in the proof circle for boring out.

3. Prepare two straight-edges, say 36 inches in length by 3 inches in width, and recessed at the middle, say 6 inches by $\frac{1}{32}$ inch, so that when placed together the slot so formed will be 3 inches by $\frac{1}{16}$ inch.

4. Have, say, a 2-inch hole punched in after peak bulkhead and place the sticks with slot horizontal across the hole: now move the straight-edges up and down until the light is seen when looking through from outside the stern post, then with a suitable radius measure upwards and downwards on the bulkhead, and make centre punch marks. Repeat the foregoing with the sticks and slot in a vertical position, and when the light is again picked up make centre punch marks on the bulkhead port and starboard at same radius.

5. Fill up the hole with wood and from the marks so obtained find the dead centre from which a proof circle can be set off for boring out.

6. Intermediate bulkheads are treated in the same manner as just described, and as a final test tin sheets, with $\frac{1}{16}$-inch holes in each, are pinned on to all the bulkhead openings, and the light, placed forward, should then be visible through the lot when viewed from the outside of the ship through the hole in stern post.

NOTE.—One man is placed to look through the holes and another moves the sliding sticks to find the light.

[To face page 57.

the shaft tunnel sighting sticks are erected, and from these the height of the various stools for the tunnel bearings are derived The centre on the forward end of the engine-room is used when the holding-down holes are templated and bored previous to the engines being installed. In this operation the template is laid down on the engine-room floor, and the centre line on the template is set in line to a centre line on the forward bulkhead in line with the sighting centre. If the shafting is in place then from the centre of the thrust shaft the template is set, or if shaft is not in place, then the template is set to the centre of the hole in which the bulkhead gland is fitted. The stern frame and bulkheads are bored out usually by power derived from an electric motor, or if no electric power is available then a small donkey boiler and steam-engine are erected connecting with belt to the boring bar. After boring out, the stern tube is put in place from the inside, and drawn hard up into position by means of the nut on the after end. The inner end is bolted to the bulkhead and wood liner fitted at back of same. At the outer end of stern tube a brass bush is fitted, termed the stern bush. This bush is lined with lignum vitæ, the bottom layers having the grain end on, so as to reduce the wear as much as possible. The lignum vitæ is fitted into channels in the brass bush, and is prevented from working out by a collar at the forward end of the bush, and at the aft end by means of a brass gland bolted to the flange of the bush itself (see illustration facing page 361). The space into which the wood is fitted is tapered in a fore and aft direction, and the wood is driven up into same, thus ensuring a good fit. The tail shaft is shipped into stern tube from the interior of the tunnel, and on the inside flange of the stern tube a gland is fitted, which prevents any leakage taking place into the tunnel. This gland is packed with soft rope-yarn packing soaked in tallow, and a water connection is led from the top of the stern tube to this gland, so that in the event of getting hot the gland and shaft can be cooled out. The propeller is held on shaft by means of a feather and nut. This nut is hard hammered up, and a stopper fitted. In the recess in front of the boss a rubber ring is fitted to prevent water getting in or eating away the part of the shaft which is not covered by the brass liner, and in some cases short glands are fitted on the forward side of the boss, being packed with a rubber ring (Sketch No. 69). The tunnel bearings are of cast iron lined with white metal, and are supported on built-up stool, between which and bearing block teakwood liners are fitted, bringing bearings up to required height (Sketch No. 70). At the forward end of the tunnel the thrust block is situated (see page 182): this block is made up of a number of shoes, as the design may require. The shoes are of cast iron, lined with white metal, having gutters cut on each side, oil being supplied from an oil box cast on each shoe. Water service connections are also made so that a water circulation takes place throughout the interior of the shoe. The block itself is of cast iron, and is rigidly bolted to the ship's frame, this part of the ship being specially until light is visible, the light now being seen from aft end of tube bearing to engine-room. Another centre is fitted in engine-room bulkhead, and the light picked up. A similar centre is fitted in the after bulkhead in the tunnel, and light sighted. The whole is again sighted from the aft end, and light being seen, a centre is put in holes in discs and circles drawn around equal to the boring out size; proof marks are also put on so as to test boring bar. The bulkhead in forward end of tunnel is marked off in a similar manner, and is bored out for a bulkhead gland. At various distances throughout the shaft tunnel sighting sticks are erected, and from these the height of the various stools for the tunnel bearings are derived The centre on the forward end of the engine-room is used when the holding-down holes are templated and bored previous to the engines being installed. In this operation the template is laid down on the engine-room floor, and the centre line on the template is set in line to a centre line on the forward bulkhead in line with the sighting centre. 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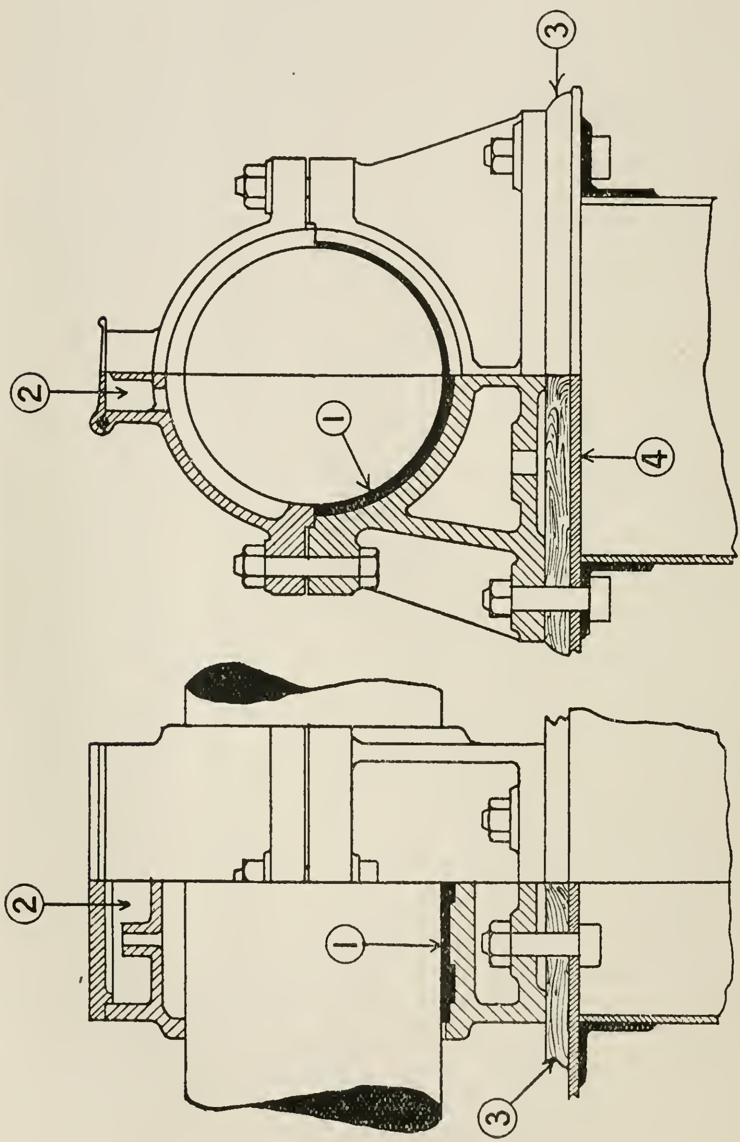
The lignum vitæ is fitted into channels in the brass bush, and is prevented from working out by a collar at the forward end of the bush, and at the aft end by means of a brass gland bolted to the flange of the bush itself (see illustration facing page 361). The space into which the wood is fitted is tapered in a fore and aft direction, and the wood is driven up into same, thus ensuring a good fit. The tail shaft is shipped into stern tube from the interior of the tunnel, and on the inside flange of the stern tube a gland is fitted, which prevents any leakage taking place into the tunnel. This gland is packed with soft rope-yarn packing soaked in tallow, and a water connection is led from the top of the stern tube to this gland, so that in the event of getting hot the gland and shaft can be cooled out. The propeller is held on shaft by means of a feather and nut. This nut is hard hammered up, and a stopper fitted. In the recess in front of the boss a rubber ring is fitted to prevent water getting in or eating away the part of the shaft which is not covered by the brass liner, and in some cases short glands are fitted on the forward side of the boss, being packed with a rubber ring (Sketch No. 69). The tunnel bearings are of cast iron lined with white metal, and are supported on built-up stool, between which and bearing block teakwood liners are fitted, bringing bearings up to required height (Sketch No. 70). At the forward end of the tunnel the thrust block is situated (see page 182): this block is made up of a number of shoes, as the design may require. The shoes are of cast iron, lined with white metal, having gutters cut on each side, oil being supplied from an oil box cast on each shoe. Water service connections are also made so that a water circulation takes place throughout the interior of the shoe. The block itself is of cast iron, and is rigidly bolted to the ship's frame, this part of the ship being specially strengthened. The interior of the block is used as a lubricating bath, being filled with fresh water and oil, through which the collars of the thrust shaft revolve, thus lubricating each face of the shoes.

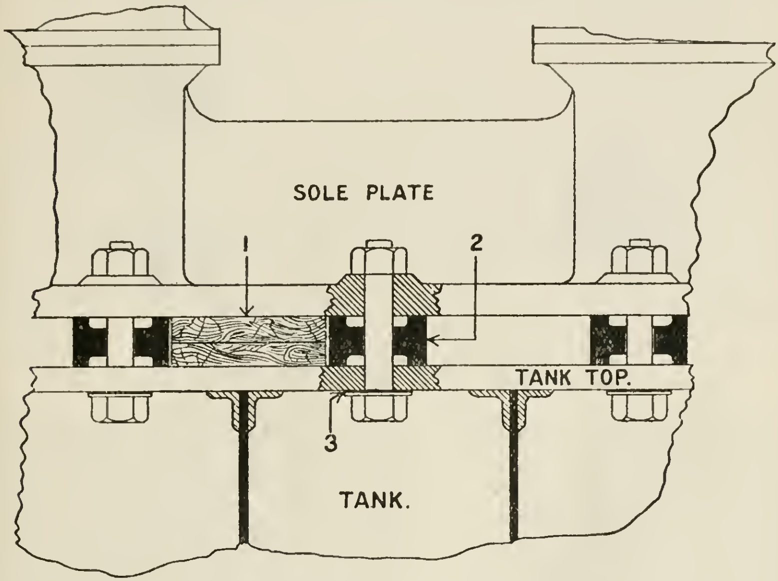


No. 70.—Tunnel Block or Bearing.

1, White metal. 2, Oil box. 3, Wood liner. 4, Built-up stool.

Erecting Machinery in Ship.

The tail shaft being shipped into place, and stern tube gland fitted and packed, the intermediate lengths of shafting are now put in and

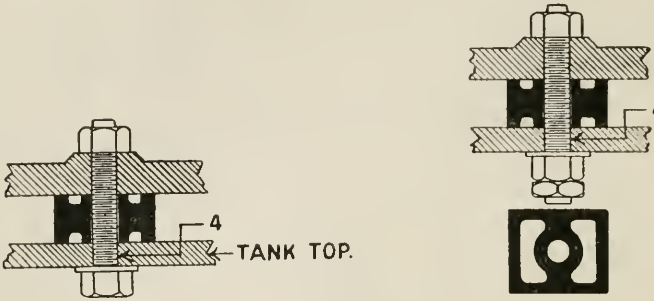


No. 71.—Cast-Iron Chocks.

1, Teakwood wedges. 2, Chock. 3, Joint.

NOTE.—The size of chock varies, but average proportions are: width 6 to 8 inches, depth 1 to $1\frac{1}{2}$ inches, fitting strips $\frac{3}{8}$ to $\frac{1}{2}$ inch.

set up in line. The shaft is blocked up and tunnel bearing block put on shaft, being bound on same by means of canvas between cover of bearings and shaft. The aft coupling of the shaft is brought fair to the coupling of the propeller shaft by means of feelers, that is, the face of the two couplings are tested to ensure that a feeler of say ·010 can be inserted at all four points; and the rim of the flange of the coupling tested also by means of a straight-edge, to ensure that shafts are in line sideways, and also for height. The space between the bottom of the bearing block and the stool is now filled up with teakwood liner. During the filling and on completion the shaft couplings are tested to ensure that they remain fair. The next length of shafting is set up in a similar manner, and so on until the complete length of shafting has been set up. At the forward end of the tunnel where the shaft passes through the bulkhead, another gland is fitted, so that in event of the tunnel being flooded, by shutting the watertight door at the entrance to the tunnel, no water would pass into the engine-room. This gland is also packed with soft rope-yarn packing. The thrust shaft is set up to the coupling of the intermediate shaft and set, and the holes for fixing block to seating are bored, and *fitted* bolts put in. The engine soleplate with crank-shaft in place is brought



No. 72.—Types of Cast-Iron Chocks.

4, Bolt screwed through tank top.

into line with coupling of thrust shaft and set up in a similar manner as that previously described, flange of couplings set fair face to face and on the rim of flanges. To arrive at this result the soleplate is made up at suitable points on iron wedges and plates, and these wedges are driven as required to bring crank-shaft coupling up to height of thrust shaft coupling. The soleplate and crank-shaft are moved bodily as required by means of screw jacks. The couplings being fair the space between the engine-room seating and the soleplate is made up by means of cast-iron chocks. These chocks are usually fitted at each bolt which binds soleplate to ship, and are chipped and filed until they are a good fit. The holes for bolts or studs are bored through the engine seating, if not previously marked off by template. In the case of there being a tank under the engine-room, screwed studs will be fitted as holding-down bolts having a nut inside the tank, jointed with washer and grummet. The soleplate being now made up and set, the columns and cylinders are lowered into position, and gear erected on engine. The pipes connecting the various parts of the engines and boilers are now fitted and jointed, the main connections on the engine being as follows:—

Main Steam Pipe.—From boilers to engine stop valve.

Main Injection Pipe.—From valve on ship's side to bottom of circulating pump.

Main Discharge Pipe.—From top of condenser to ship's side.

Feed Pump Suction.—From hot-well to suction valves on feed pump.

Feed Pump Discharge.—From feed pump discharge valves to boiler.

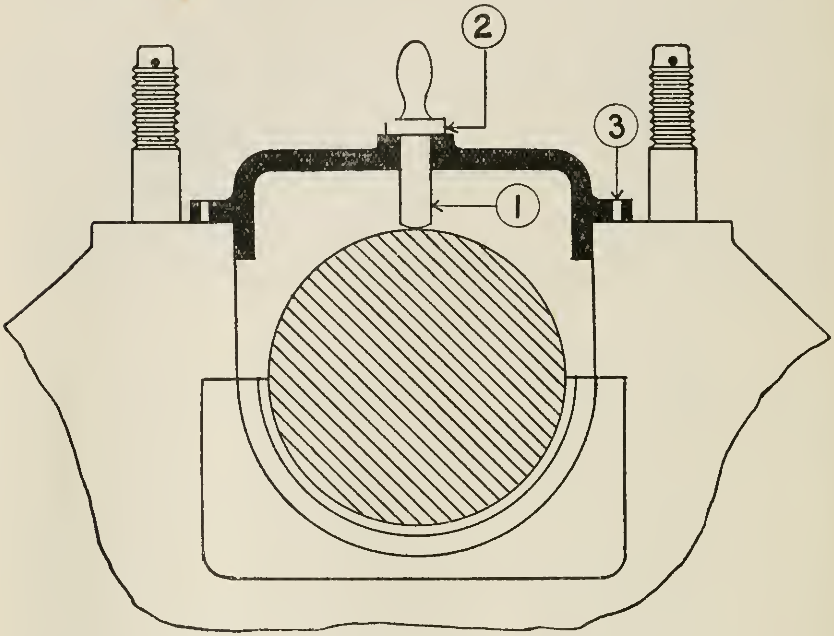
Bilge Pump Suction.—Led to distribution box in engine-room, from which various holds and wells are connected.

Bilge Pump Discharge.—From bilge pumps to discharge valve on ship's side.

The boilers are erected on stools in the boiler hold, chocks being fitted between the boiler shell and the stools (see page 156). Chocks are also fitted between the boilers, and stays are also fitted from the ship's side to the boiler, to prevent movement of the boilers in bad weather. Knees are also fitted at both ends of the boiler riveted to the tank top, and being close up to the front of the boiler at the centre; these knees are usually left from $\frac{1}{16}$ to $\frac{3}{32}$ inch clear, to allow for expansion of boiler (see page 156). The usual mountings on the boilers are as follows:—

Main stop valve Auxiliary stop valve Steam to whistle Safety valves	Usually on top of boiler.
Gauge glass connections Scum cock Auxiliary feed check Main feed check Test cocks Salinometer cock	On end of boiler.
Blow-down cock Drain cock	On bottom of boiler.

After boilers have been installed and all connections raised steam is got up and the boiler covering put on. This is usually one or other of the specialities on the market. It is put on in the form of wet pulp and dried by the heat of the boiler. Outside, a sheet-iron casing is fitted extending to the bottom quarter of the boiler, and in some ships asbestos mats are fitted round the bottom of the sheet covered with wire netting. After boilers are covered, steam is raised and safety valves set. This means that the washers between the safety valve nuts and the standards are taken out and the nuts adjusted so that the valves will lift and release the pressure on the boiler when it has reached the designed pressure. In a boiler having forced draught, an accumulation test is necessary. This means that with forced draught being maintained to the pressure required, usually $\frac{3}{4}$ inch air pressure in the ashpits, the pressure on the boiler must not rise more than 5 lbs. on the figure required, thus showing that the safety valves are of ample area to release any pressure over that which it is designed for the boiler to carry. After valves have been set, the space between the nuts and collar is gauged, and the washers fitted accurately to same. If the engines are in an advanced state of construction at this time, it is usual to have what is termed a basin trial on the same day as the safety valves are set. During a basin trial the engines are revolved at a slow speed, it not being possible to exceed this, owing to the risk of carrying away the mooring ropes.



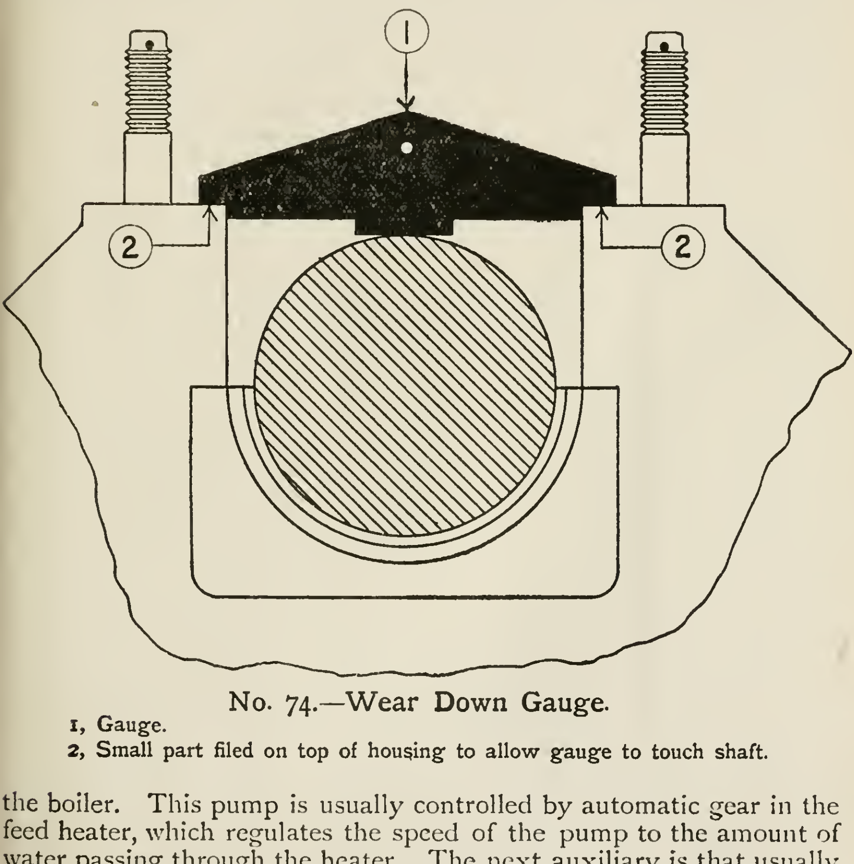
No. 73.—Wear Down Gauge.

1, Steel pin touching shaft.
2, Collar touching gauge when pin is touching shaft.
3, Holes for dowel pins.

The gauges shown in Nos. 75 and 76 are employed in testing the wear down of shafting, as when the bearings work down a clearance will be shown at the point of the bolt in No. 75 or between the projection on the gauge plate of No. 76. These clearances should be carefully noted for each bearing, and a record kept for future reference and subsequent wear down.

Auxiliary Machinery.

In describing the auxiliary machinery it will be understood that we are dealing with an installation suitable for a set of engines the construction of which has been previously described. As before explained, the feed pumps are connected and operated by the main engines. These pumps discharge the feed water to a feed heater, which is situated on the upper platform of the engine-room. The feed water passes through this heater, and becomes heated by contact with steam which is admitted into the interior. A heater of this type is termed a direct contact heater. The water falls to the bottom of the heater, and falls by gravitation to a feed pump, which is situated on the bottom platform of the engine-room. This pump delivers the water through the filter, which is one or other of the various makes described elsewhere, and then through the feed check or valve into



No. 74.—Wear Down Gauge.

1, Gauge.
2, Small part filed on top of housing to allow gauge to touch shaft.

the boiler. This pump is usually controlled by automatic gear in the feed heater, which regulates the speed of the pump to the amount of water passing through the heater. The next auxiliary is that usually termed the "General Service Donkey Pump." This pump has connections suitable to draw from the sea, hot-well, tanks, and bilges and can discharge to the boilers (through the auxiliary feed checks), tanks, and overboard.

The ballast pump is used for pumping the various ballast tanks in the ship, and has connection to all parts of the ship, also sea and bilge suction, and can discharge overboard and into the tanks (see illustration facing page 198). An evaporator is fitted, one of the various types described elsewhere, the principles of which are as follows:—Steam is passed through copper pipes of various shapes; outside of these pipes is sea water, which becomes heated, and gives off a steam vapour; this vapour is collected and led off either to the condenser or L.P. engine casing, thus adding to the amount of feed water which is pumped into the boilers. A small pump is fitted, usually operated by the main engine pump levers. This pump has a sea suction and discharge into the evaporator, and the amount of water being pumped in is adjusted so as to make up for the amount being evaporated. There are various types of feed water filters on the market, the principle of which is to extract the grease and impurities from the feed water. The filter medium usually consists of cloths of the nature of towelling held between metal perforated grids, and in some filters furnace slag is used, proving a cheap and efficient filtering material. In large ships a sanitary pump operated by the main engine levers is fitted, but in cargo steamers it is usual to have a tank on the top of the engine-room skylights; this tank is filled up every morning, supplying the necessary water for the sanitary system throughout the day. In ships having forced draught a fan engine is fitted; this engine operates the fan which supplies the air to the boiler furnaces. In Howden's system the air is carried through a trunk, and then passes around tubes situated in the boiler uptake. The air is heated by this means before passing into the furnaces. The usual pressure to carry on the air gauge or fan is from $1\frac{1}{2}$ to 2 inches; this gives a pressure in the ashpits of $\frac{3}{4}$ to $\frac{7}{8}$ inch. It will be understood that this air pressure may be altered according to conditions, such as nature of coal being used, and also as regards weather conditions. In the stokehold an ash hoist is fitted. This may be one of the specialities, such as Alley & M'Lellan's, Crompton's, or See's ash ejector. But it is more common to have a small steam winch fitted up on the top of the fiddley, with a steel wire rope led through pulleys into the ventilators, and thence to the stokehold floor. If the ship is fitted with electric light the electric engine is usually in the engine-room, steam being supplied either from the auxiliary steam pipe or direct from the boiler. This engine is generally fitted with a governor so as to ensure steady running. On deck the machinery usually consists of eight or twelve winches, steam being supplied from the main or donkey boiler, and the exhaust from these winches is led back to an auxiliary condenser situated in the engine-room. The auxiliary condenser is generally arranged so that the condensed steam flows into a tank underneath; the feed pump is connected to same, and supplies the donkey boiler with feed water from this source. The circulating water for the condenser is usually supplied by a small pump fitted for this purpose, and in up-to-date installations the condenser is supplied with circulating water from an engine which works an air circulating and feed pump together, a very compact arrangement. A steam windlass is fitted on the forecastle head, and in some cases a warping capstan is fitted on the poop or aft end of the ship. On the top platform of the engine-room the steering engine is situated: this engine has exhaust connections to the main and auxiliary condensers and also to the atmosphere. The control gear for this engine is led from the bridge, and is operated by means of the steering wheel. There are various types of steering engines on the market, but the main principles of each are similar. In some ships the steering engine is housed aft, being directly connected to the rudder head, and the connecting shafts are led along inside a casing or deck to the engine, there operating the valve on the steering engine; this valve is termed the control valve. A sketch is given showing the arrangement of shafting operating the control valve and a description of the various types of steering gears is given elsewhere. (See page 340.)

Trial Trips.

On completion of the installation of the machinery on board ship, and previous to handing the ship over to the owner's representatives, a trial trip is run. This usually consists of a series of runs over a measured course, posts or sighting points being erected on the shore, the distance between being exactly one nautical mile. After ship leaves the harbour, the compasses are adjusted, that is, the ship is slowly steamed in a circle, the compass adjusters during this time finding out and adjusting the reading of the compasses supplied to the ship. In the engine-room an engineer is told off to attend to a special part of the machinery, one attending to the boilers, regulating the water supply and seeing that the steam pressure is maintained. One man looks after an individual engine, overlooking the running of the main bearings, eccentric straps, and crank-pins. On the middle platform men are also stationed, who observe the running of the top end bearings, guide shoes and piston rods, and valve gear. The piston rods are swabbed with cylinder oil, and the other bearings on connecting rod and guide shoes are supplied with oil from siphon boxes (see page 197) on the top platform, but it is usual on trial trips to augment this supply by hand feeding. A man attends to the pumps and connections on same, overlooking the pumping of bilges, supply of circulating water, and working of feed pumps. An engineer is also in attendance in the tunnel, whose duty it is to attend to the lubrication of the tunnel bearings and thrust block. On the top platform a man is also stationed attending to the supply of oil in the siphon boxes and the working of the feed heater and steering gear, if same gear is situated in the engine-room. It is usual to proceed slowly to the measured mile so as to gradually work the bearings into good running condition. All being well and ready for the first run, draughtsmen are told off for taking indication cards and counters, and observing pressures on the various gauges connected with the machinery. It will be understood that all operations have to be smartly carried out as the time which elapses between going on the mile and coming off, even in a slow cargo tramp, is not of long duration. Immediately before coming in line with the post on the shore a warning bell is rung, "get ready," and then when in line the telegraph is rung hard signifying that the ship is "now on the mile." When the mile has been run and the ship in line with the other post or point on shore the telegraph is again rung, and the engines are slowed down, while preparations are made in the stokehold for the next run. The ship is brought round again and all ready for the return run. This is carried out five or six times, and then the course is laid for a run of three to six hours at a steady speed, thus proving that machinery is in good working order. During the running on the mile, coal is measured in the stokehold, and after indicators, cards, and particulars taken have been worked out a statement is prepared showing the result of the various runs on the mile. In large passenger ships and Admiralty ships runs of thirty hours' duration are made, the trial trips in these cases usually extending to four or five days. In Admiralty trials the feed water is measured, and the firing is carried out on a system of time firing, that is, arrangements are made to burn a certain amount of coal per square foot of fire grate per hour. The coal is measured out, and on the ringing of a gong or similar signal the firing in each stokehold takes place.

Care and Upkeep of Machinery.

In considering the care required to keep a set of triple expansion engines in good condition, we will deal with a set of new engines and consider what means are required to maintain the machinery in an efficient condition. It should be borne in mind that a loose bearing has as much chance of heating up as one that is too finely adjusted, especially when it is a connecting rod bearing that is in a slack condition. The bearing will knock on the centres ; this knocking tends to spread the white metal, with the result that ridges are formed on the side of the bearings, and also the knocking has the tendency to press out the oil which is between the bush and the pin. To ensure a bearing being in proper adjustment, after leads have been taken off and found correct, the bearings should be put together and nuts hammered up until they are at the marks which were put on when leads were in bush. By inserting a slice bar or other suitable bar between the web and the bush and testing the bush to see that it moves from side to side, this will prove that bush is not too tight, and should give good running results. The same operaton should be carried out on the top end bearing also. After ship has done outward voyage, the top main bearings should be lifted and wear down gauge applied. There are various forms of wear down gauges supplied by various builders. Sketches, Nos. 73 and 74, are given showing two forms of this gauge which are generally supplied. If wear has taken place, notes of same should be taken, and after the next run this reading should be again verified, as if the wear continues it is possible that the chocking and lining up of the soleplate has not been properly carried out. The eccentric straps are also liable to wear, especially when coming to a bearing, and to avoid the trouble of opening up valve casings and testing setting of valves, a simple method of proving same is as follows, when engines are new, and this should be done in the works if possible :—With the valve standing at full travel upwards, a mark is put on valve spindle, and from this mark a small trammel is made, touching a point either on the cylinder or valve spindle guide bracket (Sketch No. 59). By turning engine into similar position and trying trammel, any wear down that has taken place can be seen at once, the wear down having taken place either in saddle block bearings or eccentric straps. Another method is to put valve to top lead and put a trammel mark on with spindle in this position ; this is easier, as it only entails turning crank to top centre, and having gear full ahead, care being taken that link block is in same position as when trammel marks were applied. Piston rods should be carefully watched to see that engines have been carefully lined off, and that glands are true to bore of cylinder, a defect which will show up very early if such should be the case (Sketch No. 48). The working of the various pumps should be noted, and wear of pump links tested in the usual manner, that is, by disconnecting same and testing distance from pump cross-head to pins on lever by means of gauge, which should be made and kept for this purpose (Sketch No. 61). A hint may not be out of place as regards circulating pump, and that is to use as little circulating water as is necessary to ensure good vacuum. By carefully observing vacuum gauge and gradually closing down circulating inlet it will be found possible to ease the load on pump to a considerable extent. The H.P. piston valve should be kept in as tight a condition as possible, and it is not conducive to a good working piston valve to keep on lining out the rings on same. This lining only tends to wear the valve liners into an oval shape, with the result that reboring out is necessary. If the valve liners are plain, then the turning of two or three small grooves round the circumference of rings will assist to keep valve tight, as condensed steam gets into grooves and forms a film between liner and ring (see page 207). The H.P. piston, being fitted with Ramsbottom rings, should give good results if the rings have been properly manufactured, but if they show signs of having lost their elasticity, then new ones should be fitted. Another point regarding these rings is to have them as near a fit in the grooves in piston as possible, for if slack, then the constant change from one side to the other will not only wear out the rings but will inflict considerable damage on the bonnet or packing ring. The I.P. slide valve usually gives trouble, and the face of valve and valve face on cylinder should be carefully tested by means of straight-edge and feelers to ascertain that wear is taking place equally all over. Should the valve show wear round the outside, that is, the inside ports or bars show high, then it will help to equal the wear if a few short gutters are cut on the part that is wearing. Another good plan is to bore a series of holes with a flat-nosed drill (Sketch No. 65). These holes should only be about $\frac{1}{32}$ to $\frac{1}{16}$ inch deep. It is the practice now to bore out the cylinders with a bell mouth at either end, the piston travels the extreme length of the bore, with the result that no ridges are formed at top and bottom of cylinders. Condensers in new ships are sometimes a source of trouble, leaking taking place, with the result that boiler density is increased. A good test for feed water is by means of nitrate of silver, and the test consists of drawing off a small quantity of feed water in a tumbler or other transparent vessel and adding a drop of nitrate of silver; should the water become milky then salt is present, and the amount can be gauged by the resultant whiteness of the water. The use of water service on engine bearings has greatly decreased, as engineers find that a bearing will run just as well without water, always allowing that it is in line and properly adjusted. Once water is used it is not possible to run a bearing without it, so that the use of water, even in small quantities, should be avoided.

How to keep a Watch.

The engineer's life at sea is not exactly a bed of roses and the following description of his duties during the time he is on watch will emphasise this statement. The engineer whose turn it is to relieve will be called at a quarter before the hour on which he takes up duty, and promptness in relieving is most important. On entering the engine-room the first inspection should be the steering gear, bearings examined, and moving parts inspected to ensure all being in good order. The feed heater is next visited and pressure on steam gauge noted. Descending to the middle platform L.P. piston rod is felt by hand, also top end and face of columns and guide shoes. The valve gear or L.P. engine is next looked over, and the I.P. and H.P. engines dealt with in the same manner. The rocking shaft bearing is felt by hand, and the crosshead links and top end of the pump links. All being well, the bottom platform is next visited, and the bottom ends and eccentric straps and main bearings felt by hand. During the passage from one bearing to another, the L.P. and I.P. pressure gauges are glanced at, to ensure that pressure is being maintained, also the vacuum gauge. The boilers are next visited, and height of water in gauge glasses and steam pressure noted, also that the firemen are at their duties, preparing to clean fires. The pumps are next examined, care being taken to ascertain that bilge pumps are working, and also the connections examined to see what part of the ship the bilge pumps are drawing from. Port and starboard engine-room bilges are examined to ascertain the depth of water in same. The thrust block is next felt over by hand, and the interior of block examined to see that collars on shaft are immersed in the oil and water bath. The tunnel is next inspected, and each individual bearing felt by hand, also the stern tube gland. The tunnel well is examined to see that water is not excessive; then back to engine-room, and the word passed, "All right," to engineer who is going off duty.

The foregoing inspection is usually carried out in from seven to ten minutes. By this time the firemen will be well on the way cleaning fires, and the engineer will know in which boiler fires are being cleaned; this being so, the feed checks should be regulated, as the boiler on which fires are being cleaned will be at slightly lower pressure than those which are steaming full. As soon as the fires on this boiler are well away and the next fire started cleaning, it will be necessary to again regulate checks, and by the half hour after going on watch, steam should be at full working pressure and checks can then be set, so that with but little alteration the rest of the watch can be run. Should there be no greasers carried, then it will be part of the engineer's duty to attend to the oiling of the machinery. Oil is usually a precious liquid on board ship, and the engineer will only be allowed a certain amount on which to run his watch. The top cups on the side of the cylinder will first claim his attention, and should be filled up to $\frac{1}{4}$ or $\frac{1}{2}$ inch below top of tube, siphons taken out and dipped into oil and then put back. At the half hour after going on watch the first oiling round should take place. When filling up top cups on engine, the steering gear should be visited and oiled if it is situated in the engine-room, as many gears are. The main bearing cups will require filling up, and siphons redipped as was done on top, eccentric straps oiled, and it may be mentioned that very little oil, properly applied, is as much good as a canful poured on, which only runs out of the bearings and does not lubricate. It is generally the rule to supplement the siphon feed to connecting rod bearings by hand feed; usually the rule is to insert oil can into oil cup, and give what can be supplied in two or three revolutions, but this amount varies according to how the machinery runs. The valve gear is usually oiled after being on watch one hour, and then again one hour before being relieved. The pump links and rocking shaft bearings are dealt with along with the connecting rods and eccentric straps, that is every half hour. In some engines siphon fed oil cups are fitted to rocking shaft bearings and pump links, so that in this case it is only necessary to refill cups.

The usual rule for evaporators is two hours each watch, so that one hour after coming on it will be requisite to get same under weigh. Steam is turned on, and feed pump set, and vapour valve opened, and height of water maintained in gauge glass on evaporator, and steam gauge set to working pressure. On completion of two hours' evaporating, if the level of the water in the boilers is at the required height, the engineer will blow down evaporator, that is, the vapour valve and feed pump will be shut off, and a pressure raised on evaporator, the blow-down cock opened and evaporator blown out. This operation is necessary to reduce the density of the water. After blowing down, the evaporator is refilled to the required height with water, and left ready for the next watch. The thermometer on the feed pump, which is drawing from the feed heater, should be examined to see that temperature is being maintained, and that feed heater is working efficiently. Visits should be made to the stokehold and fires examined, also to the bunkers to see that trimmers are working the bunkers as directed. The tunnel should also be visited at least twice each watch, if not every hour, and the solidified oil in bearings renewed or pressed down on shaft. Attention should be given to the pumping of the bilges in the engine-room and tunnel, and mud box on bilge pump cleaned out during the watch. Temperatures of the sea water passing to circulating pump and discharge from condenser overboard should be taken and entered in the log, also temperature of feed water, and engine-room temperature, steam pressure on boilers, H.P. cylinder, I.P. cylinder, L.P. cylinder, and vacuum should be entered in log, length of times evaporator was working, and height of water in main boilers. At one bell, that is, quarter before the hour of being relieved, the engineer calls the next watch, and is again below ready to take the counter when his four hours are up. The counter is taken as eight bells strikes, and worked out from the counter of the preceding watch, and average revolutions entered in log book. The relief having gone round and passed the word, "All right," the engineer is free.

Economical Working.

To obtain the best results from the engines and boilers the following points should be attended to:—

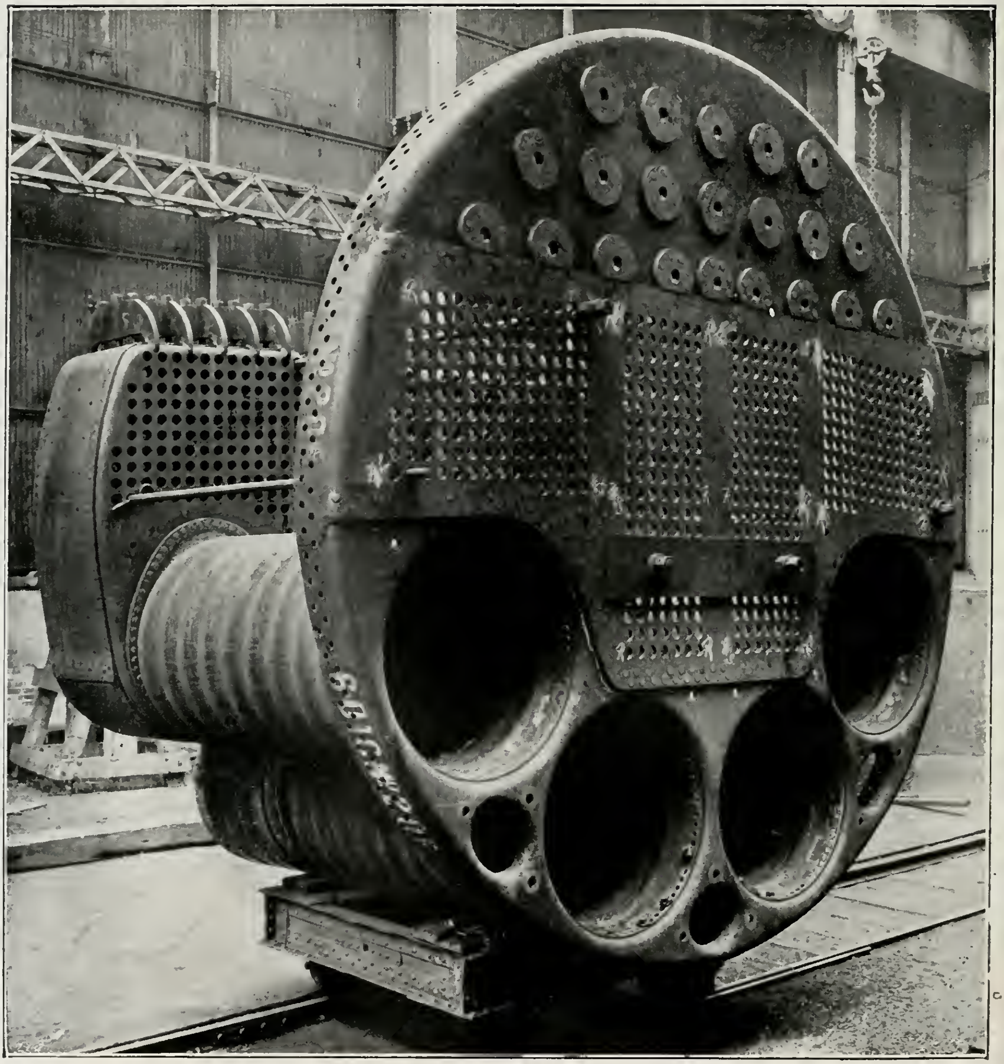
1. Keep grate surface as *short* as possible.

2. Work with stop and throttle valves well open, and expand by link gear only.

3. See that the indicator cards show good compression curves.

4. If possible, balance up the power in each cylinder (allowing extra power for engine driving pumps) by the link gear adjustment.

5. Keep pistons and valve faces tight.



VIEW OF BOILER FRONT PLATE, FURNACE, AND COMBUSTION.
(Under Construction.)

SECTION II.

BOILERS.

Tensile Strength of Plates.—The tensile strength of steel shell plates ranges from 28 tons to 32 tons per square inch; if of higher strength the metal is less ductile, and therefore less suitable for flanging or for expansion under heat. The tensile strength of combustion chamber and furnace plates ranges from 26 tons to 30 tons per square inch; if over 30 tons the plates are too brittle. The tensile strength is also known as the *ultimate* strength or breaking stress of the material.

Elastic Limit and Safety Factor.—If a tensile stress of so many tons is put on a test strip of steel the strip will become elongated, and if the load is then taken off the metal will return to its original length if the stress has been *within* the elastic limit; if, however, the stress has *exceeded* the elastic limit the metal remains elongated, as "permanent set" or fracture has then taken place.

If, therefore, the elastic limit is found by testing a number of strips the safe stress may be taken as equal to about half of this limit, and from this the Factor of Safety may be determined. Steel plates have an elastic limit ranging from 12 tons to about 14 tons per square inch.

EXAMPLE.—If $12\frac{1}{2}$ tons per square inch is found to be within the elastic limit of a steel plate, and assuming half of this as the safe working stress, determine the Factor of Safety, the tensile strength being 28 tons per square inch.

Then, $12{\cdot}5 \div 2 = 6{\cdot}25$ tons safe stress.
And, Factor of Safety $= 28 \div 6{\cdot}25 = 4{\cdot}4$.

NOTE.—The Factor of Safety for boiler shells varies from 4·4 to 4·6 according to conditions of construction.

Stresses on Shell Seams.

In cylindrical boiler shells the stress set up by the pressure on the longitudinal joints is equal to twice the stress on the circumferential joints: this is due to the difference in the end sectional area and side sectional area of the shell acted on by the pressure.

Circumferential Stress.

RULE—

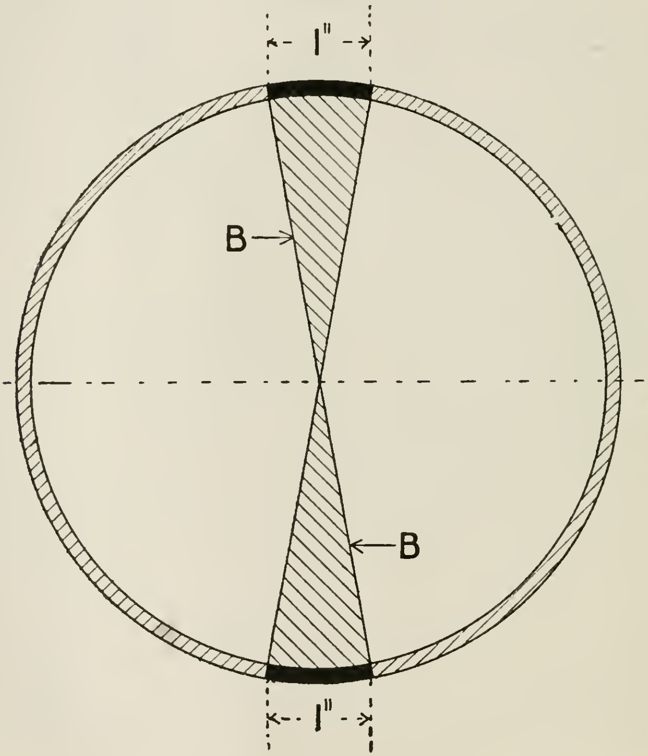
$$\frac{\text{Boiler end area} \times \text{Pressure}}{\text{Boiler circumference} \times \text{Thickness}} = \text{Stress per square inch.}$$

Longitudinal Stress.

RULE—

$$\frac{\text{Diameter} \times \text{Pressure}}{\text{Thickness} \times 2} = \text{Stress per square inch.}$$

Graphic Method of Proof for Shell Stresses.



No. 1.—Circumferential Shell Stress.

The pressure per square inch exerts a force acting from the centre on opposite sides of the diameter, and therefore on two thicknesses of the plate: this produces the stress per square inch longitudinally. The pressure per square inch also exerts a force on the boiler end area, throwing a tensile stress on the shell plate circumferentially, which produces the stress in that direction; if, then, the end area is calculated and multiplied by the pressure the result will be the total load blowing out the boiler end, and therefore resisted by the strength of the shell plate thickness circumferentially

EXAMPLE.—Determine the stress per square inch longitudinally and circumferentially on the shell of a boiler 15′ diameter, $1\frac{1}{2}''$ thick, pressure 200 lbs. per square inch.

$$\text{Longitudinal stress} = \frac{\text{Diameter} \times \text{Pressure}}{\text{Thickness} \times 2} = \frac{180 \times 200}{1 \cdot 5 \times 2} = 12000 \text{ lbs. per sq. in.}$$

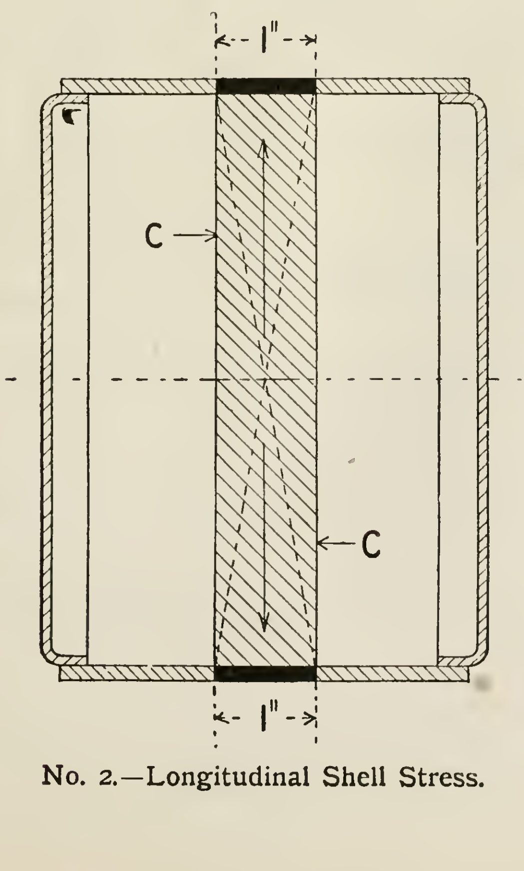
$$\left.\begin{array}{c}\text{Circumferential}\\ \text{stress}\end{array}\right\} = \frac{\text{Boiler area} \times \text{Pressure}}{\text{Boiler circum.} \times \text{Thickness}} = \frac{180^2 \times \cdot 7854 \times 200}{180 \times 3 \cdot 1416 \times 1 \cdot 5} = 6000 \text{ lbs. per sq. in}$$

NOTE.—15 ft. = 180 in.; also observe that, $\frac{180^2 \times \cdot 7854 \times 200}{180 \times 3 \cdot 1416 \times 1 \cdot 5} = \frac{180 \times 200}{4 \times 1 \cdot 5} = 6000$ lbs.

So that the circumferential stress may be expressed thus—

$$\text{Circumferential stress} = \frac{\text{Diameter} \times \text{Pressure}}{4 \times \text{Thickness}} = \text{lbs. per sq. in.}$$

Graphic Method of Proof for Shell Stresses.



No. 2.—Longitudinal Shell Stress.

It should be carefully noted that the longitudinal pressure exerts a stress on the metal *circumferentially*, also that the radial pressure exerts a stress on the metal *longitudinally*.

Observe that on two thicknesses of the shell metal circumferentially (Sketches Nos. 1 and 2) (each 1 inch wide) the pressure acts on the two areas B B to produce stress, whereas on two thicknesses of shell metal longitudinally (each 1 inch wide) the pressure acts on the two areas C C; these latter areas are therefore equal to twice B B, so that the stress longitudinally is twice that circumferentially, as previously stated. The dotted lines on C C show areas B B which are exactly half.

NOTE.—The Board of Trade require the centre circumferential shell seams to be equal to 65 per cent., and the end circumferential shell seams to be equal to 50 per cent. of the solid plate, which allows of ample strength in this direction where the smaller stress is exerted.

Strength of Shell.

The strength of a boiler shell depends, therefore, on the *Diameter* and *Thickness*, and is independent of the Length.

Boiler shells do not require stays, as circles are self-supporting: the reason for this is that the forces set up by the pressure are balanced at all positions of the circumference, or are in equilibrium.

Boiler Shell Pressure, &c.—The general equation connecting the Pressure, Diameter, and Thickness, &c., of boiler shells is as follows:—

$$28 \times 2240 \times T \times 2 \times \text{Joint} = \text{Diameter} \times \text{Factor of Safety} \times \text{Safe pressure.}$$

Where 28 tons = tensile strength of steel plates.
,, T = shell Thickness.
,, Joint = smaller result of rivet and seam section strengths.

The Factor of Safety in modern boilers varies from about 4·4 to 4·6, and represents the fraction of safe working stress as compared to breaking stress. The number 2 in the rule stands for two thicknesses of shell in one diameter—one at either side.

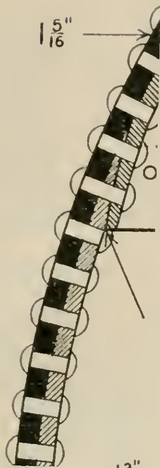
To apply the above it should be remembered that if all of the terms on one side are multiplied together and divided by all of the terms *except one* of other side then the unknown term will be brought out.

EXAMPLE 1.—Determine the required Thickness of boiler shell plate for a Pressure of 200 lbs., the Diameter being 15 feet, the Factor of safety 4·4, and the Joint strength 86 per cent.

Then, $28 \times 2240 \times T \times 2 \times \cdot 86 = 180 \text{ in.} \times 4{\cdot}4 \times 200.$

Therefore, $T = \dfrac{180 \times 4{\cdot}4 \times 200}{28 \times 2240 \times 2 \times \cdot 86} = 1{\cdot}46$ in., or say $1\frac{15}{32}$ in. thick.

NOTE.—$\frac{86}{100} = \cdot 86$, also 15 ft. = 180 in. Diameter.



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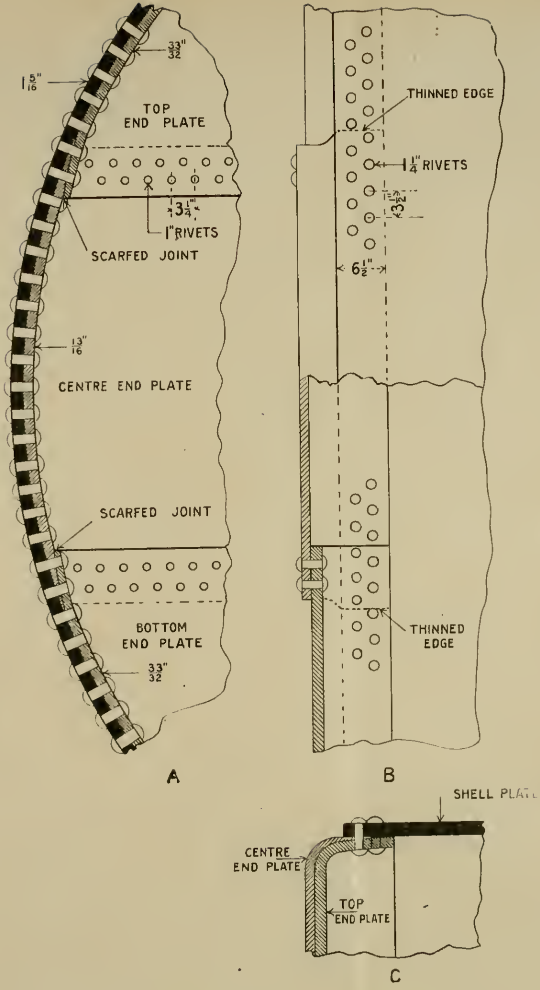

No. 3.—Joining of Shell Plates and End Plates.
(With Dimensions.)

A. End View showing Scarfed Joints of End Plates riveted to Shell.

B. Longitudinal View showing Shell lapping over End Plate Joints (half in section).

C. Plan showing Shell lapping over centre and top End Plates, which are bevelled away to form only a single thickness.

NOTE.—Where two end plates overlap one is thinned down to form an unbroken line with the other, so that the shell plate may overlap both and form a steam and water tight joint. Without this arrangement a space would be left open where the shell plate covers both end plates, and caulking up of the joint would not be possible.

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EXAMPLE 2.—Find the Safe Pressure suitable for a boiler shell 14 feet 6 inches Diameter, and $1\frac{3}{8}$ inches thick; Joint strength 85 per cent., and Safety Factor 4·5.

Then, $28 \times 2240 \times 1{\cdot}375 \times 2 \times {\cdot}85 = 174$ in. $\times 4{\cdot}5 \times$ Safe Pressure.

Therefore, Pressure $= \dfrac{28 \times 2240 \times 1{\cdot}375 \times 2 \times {\cdot}85}{174 \times 4{\cdot}5} = 187$ lbs. per square inch.

NOTE.—In the foregoing case the actual working pressure would probably be taken as 185 lbs. per square inch.

Strength of Joints.

In modern cylindrical marine boilers the average strengths of the various riveted joints are as follows:—

1. Longitudinal shell seams (D.B. straps, five rivets per pitch) = 85 per cent. of solid plate.
2. Centre circumferential shell seams (treble riveted) = 65 per cent. of solid plate.
3. End circumferential shell seams (double riveted) = 52 per cent. of solid plate.
4. End plate horizontal seams (double riveted) = 54 per cent. of solid plate.
5. Furnace and combustion chambers (double riveted) = 68 per cent. of solid plate.
6. Furnace and combustion chambers (single riveted) = 54 per cent. of solid plate.

Observe that with thin plates (furnaces and combustion chambers) the joint strength for the same type of riveting is much higher than with heavier plates, the strength of joint for similar riveting decreasing with increase of plate thickness.

Riveting.

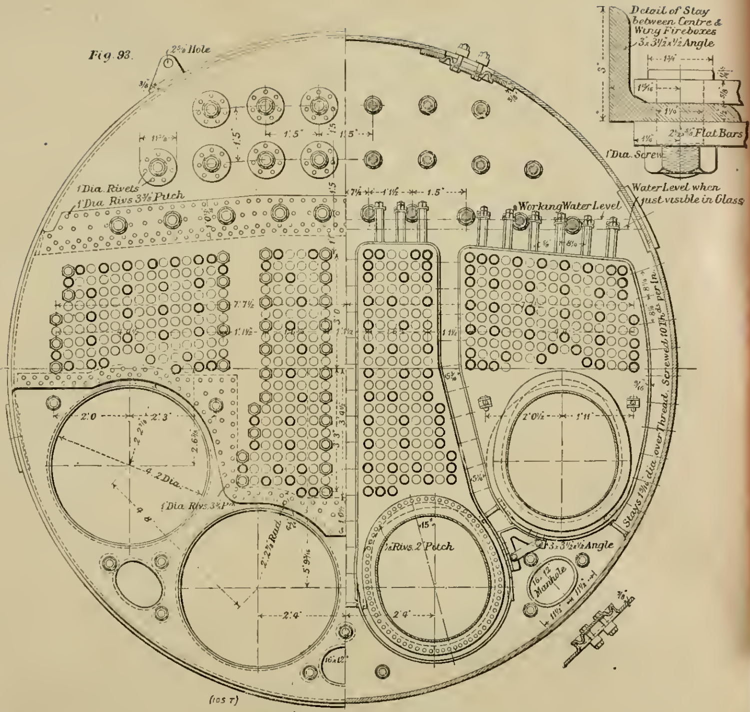
Internal parts of boilers are usually single riveted.

Circumferential seams and end plates are usually double riveted. In long boilers the *centre* circumferential seams are treble riveted to allow of the extra stress caused by barrelling when under pressure.

Longitudinal shell seams are fitted with double butt straps, and have three lines of rivets, every second rivet being omitted in the outer row (five rivets in a pitch): this is the strongest type of joint made.

In boiler joints the distance from the edge of the rivet-hole to the edge of the plate should be equal to one diameter of the rivet. Therefore the width of lap for a single riveted joint would equal three diameters of the rivet.

A joint with a great number of rivets gives a high rivet section strength, but a low plate strength; and a joint with very few rivets gives a high plate section strength, but a low rivet strength. From the above it follows that the best joint is that in which the rivet



No. 4.—End View of Double Ended Boiler, half section.

(Messrs John Brown & Co. Ltd., Clydebank.)

The difference in the diameter of the furnace mouth and back will be noticed, the left half view showing the front of the furnaces and the right half view of the back.

Notice the small clearance space allowed between the combustion chambers ($5\frac{1}{4}$ inches), also the bevelled joint of the combustion chamber bottom plates to the wrapper or side plates. The various dimensions should be carefully studied by the reader.

[To face page 75.

EXAMPLE 2.—Find the Safe Pressure suitable for a boiler shell 14 feet 6 inches Diameter, and $1\frac{3}{8}$ inches thick; Joint strength 85 per cent., and Safety Factor 4·5.

Then, $28 \times 2240 \times 1{\cdot}375 \times 2 \times {\cdot}85 = 174 \text{ in.} \times 4{\cdot}5 \times \text{Safe Pressure.}$

Therefore, $\text{Pressure} = \frac{28 \times 2240 \times 1{\cdot}375 \times 2 \times {\cdot}85}{174 \times 4{\cdot}5} = 187$ lbs. per square inch.

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Observe that with thin plates (furnaces and combustion chambers) the joint strength for the same type of riveting is much higher than with heavier plates, the strength of joint for similar riveting decreasing with increase of plate thickness.

Riveting.

Internal parts of boilers are usually single riveted.

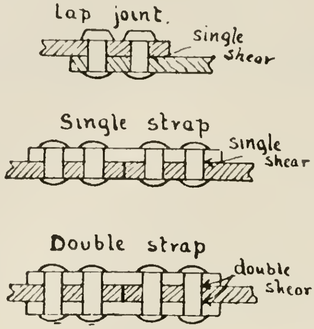
Circumferential seams and end plates are usually double riveted. In long boilers the *centre* circumferential seams are treble riveted to allow of the extra stress caused by barrelling when under pressure.

Longitudinal shell seams are fitted with double butt straps, and have three lines of rivets, every second rivet being omitted in the outer row (five rivets in a pitch): this is the strongest type of joint made.

In boiler joints the distance from the edge of the rivet-hole to the edge of the plate should be equal to one diameter of the rivet. Therefore the width of lap for a single riveted joint would equal three diameters of the rivet.

A joint with a great number of rivets gives a high rivet section strength, but a low plate strength; and a joint with very few rivets gives a high plate section strength, but a low rivet strength. From the above it follows that the best joint is that in which the rivet section and plate section strengths are about equal, hence the reason for omitting every alternate rivet in the outer row in the usual type of D.B. strap joint riveting.

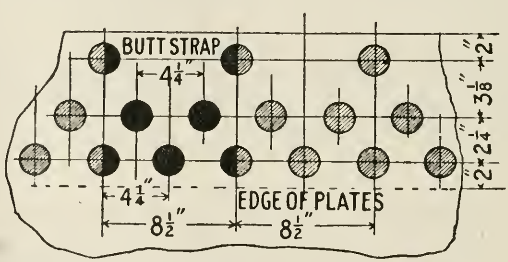
In a lap joint and a single butt strap joint the rivets are in single shear (see sketches).



No. 5.

A double butt strap joint has the rivets in double shear, which increases the strength of the rivet section 1·875 times (see sketch).

To count the Number of rivets in a pitch (N) of a joint, take the greatest pitch and count the complete number of rivets *enclosed* within it. The result is taken as the Number of rivets in a pitch (see sketch).



No. 6.—Double Butt Strap Joint Type of Riveting (Five Rivets in a Pitch).

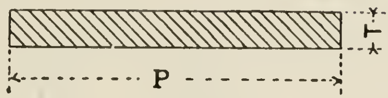
Plate $1\frac{5}{16}$ inches thick; Straps $\frac{7}{8}$ inch thick; Rivets $1\frac{11}{32}$ inches diameter. Joint strength, 84 per cent.

Observe that the shaded rivets and parts of rivets give the number enclosed *within* the greatest pitch.

NOTE.—This type of joint and riveting is only employed on longitudinal shell seams.

Seam Section Strength.

A piece of solid plate represents absolute strength, or 100 per cent.; if then rivet holes are drilled out, the metal is now less in area and the strength will be under 100 per cent. To find the strength, then, of the plate after the holes are made, and which is called the "plate at seam strength."



Solid Plate.

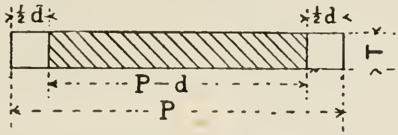


Plate cut for Holes.

No. 7.

	Solid plate	Plate left		Per cent.	
Then, as	$P \times T$:	$(P-d) \times T$	: :	100 :	Seam strength compared with solid plate.

The thickness T appearing on both sides can be omitted altogether.

Therefore, $$\frac{(P-d)\times T\times 100}{P\times T}=\frac{(P-d)\times 100}{P}=\text{Seam strength.}$$

Notice that we have now one half rivet hole taken away at either end of the pitch distance, or one rivet diameter in all, so that the solid metal section is reduced by one rivet diameter in each pitch distance. Also notice that the plate thickness T cancels out top and bottom.

Rivet Section Strength.

Rivets are fitted into the holes and closed up tight either by hydraulic or hand riveting (depending on the position of the seam). The rivets thus placed in position are intended to act as a substitute for the metal taken away. To find the strength, then, of the rivet section as compared with the solid plate.

	Solid plate	Rivet section		Per cent.	
Then, as	$P \times T$:	$d^2 \times \cdot 7854 \times \text{No.}$	: :	100 :	$\left\{\text{Rivet section strength compared with solid plate.}\right.$

Or, $$\frac{d^2\times\cdot 7854\times \text{No.}\times 100}{P\times T}=\text{Rivet section strength.}$$

Observe that the solid plate section for one pitch is equal to "pitch × thickness," and that the rivet area section for one pitch is equal to "rivet area × number of rivets in one pitch."

NOTE.—The foregoing assumes that the plate section and rivet section are of equal strength, but in the case of steel plates and steel rivets the shearing strength of the rivet section is to be taken as only 23 tons per square inch, and the tensile strength of the plate 28 tons per square inch. Therefore the formula would then read—

$$\frac{d^2\times\cdot 7854\times \text{No.}\times 100\times 23}{P\times T\times 28}=\text{Rivet section strength.}$$

Again, if double butt straps are fitted, as in longitudinal shell joints, the rivet section strength is doubled, as the rivets are now in "double shear," but as a margin of safety, 6¼ per cent. of this is deducted and the increase of strength taken as only 1·875 times that of single shear.

Joints and Riveting.

Single Riveted lap joints are employed in furnaces and combustion chambers, the strength of joint varying from 50 to 55 per cent. of the solid plate.

Double Riveted lap joints are employed in boiler end plates and circumferential shell seams, the strength of joint varying from 50 to 54 per cent. of the solid plate.

Treble Riveted lap joints are employed in the centre circumferential shell seams of long boilers (double-ended type), the strength of joint varying from 60 to 65 per cent. of the solid plate.

Double Butt strap joints with five rivets in the greatest pitch are only employed on the longitudinal shell seams, the strength of joint varying from 83 to 86 per cent. of the solid plate.

NOTE.—The "joint" strength is always taken as the smaller of the "rivet section," and the "plate section at seam" strength results, as the weaker section limits the strength. This is shown in the various worked out examples which follow.

Types of Joints with Dimensions (suitable for Patches).

No. 1 (Sketch No. 8). Plate $\frac{3}{8}''$ thick, single riveted lap joint to be applied.

Then, Rivet diam. $=1\cdot2\times\sqrt{T}=1\cdot2\times\sqrt{\cdot375}=\cdot734''$, say $\cdot75''$ diam. of rivet.

Again, Rivet Pitch $=\dfrac{100\times\text{Rivet diam.}}{100-\text{joint}}=\dfrac{100\times\cdot75}{100-55}=1\cdot66''$, say $1\frac{11}{16}''$ pitch.

NOTE.—T=Plate thickness; joint=55 per cent. for single riveting. The width of lap $=\cdot75\times3=2\cdot25''$.

Required Pitch of Rivets for Equal Strength in Seam Section and in Rivet Section.

For equal strength—

(1) $\dfrac{P-d}{P}$ is to be equal to $\dfrac{\text{Rivet area}\times\text{No.}\times C\times 23}{P\times t\times 28}$.

(2) Cancel out P under the line on each side.

Then, $$P-d=\frac{\text{Rivet area}\times\text{No.}\times C\times 23}{t\times 28}.$$

(3) Transpose and change over signs, minus to plus.

So that, $$P=\frac{\text{Rivet area}\times\text{No.}\times C\times 23}{t\times 28}+d.$$

EXAMPLE.—Find required pitch of rivets to give equal strength for seam and rivet section in a double-riveted lap joint, the diameter of rivets being $1''$, and the plate thickness $\frac{5}{8}''$.

NOTE.—$C=1$ for lap joint.
$C=1\cdot875$ for D.B.S. joints.

Then, $$\frac{P-1}{P}=\frac{1^2\times\cdot7854\times2\times23}{P\times\cdot625\times28}.$$

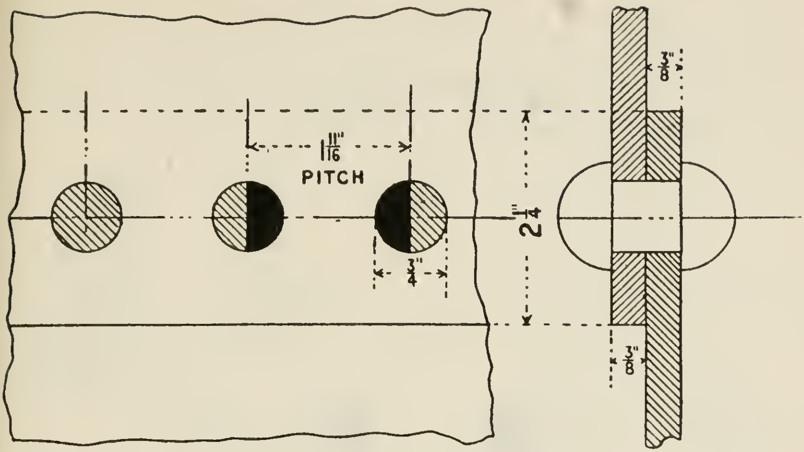
Cancel out P on bottom line of each side.

Therefore, $$P-1=\frac{1^2\times\cdot7854\times2\times23}{\cdot625\times28}=2\cdot06''.$$

Transpose and change signs.

So that, $$P=\frac{1^2\times\cdot7854\times2\times23}{\cdot625\times28}+1''=3\cdot06\text{, say }3''.$$

NOTE.—As the value of C is 1 for lap joints it may be omitted altogether in the present calculation.



No. 8.—Single Riveting.
(Furnace or Combustion Chamber.)

Diameter and Pitch of Rivets, &c.

RULE—

$$\text{Rivet diameter} = 1{\cdot}2 \times \sqrt{T}.$$

Therefore, Rivet diameter $= 1{\cdot}2 \times \sqrt{{\cdot}375} = 1{\cdot}2 \times {\cdot}612 = {\cdot}734$ in., say $\frac{3}{4}$ in. rivet.

NOTE.—T = plate thickness.

RULE—

$$\text{Rivet Pitch} = \frac{100 \times \text{Rivet diameter}}{100 - \text{joint}}.$$

Therefore, Pitch $= \frac{100 \times {\cdot}75}{100 - 54} = 1{\cdot}65$ in., say $1\frac{11}{16}$ in.

RULE — Distance from edge of plate to Rivet centre = Rivet diameter × 1·5.

Therefore, $\cdot 75 \times 1{\cdot}5 = 1\frac{1}{8}$ in.

The width of lap is in this case equal to

$1\frac{1}{8}$ in. $\times 2 = 2\frac{1}{4}$ in. (single riveting).

NOTE.—The joint strength for single riveting with thin plates is taken as about 54 per cent. of the solid plate.

To prove joint strength.

$$\text{Seam} = \frac{(p-d) \times 100}{P} = \frac{(1{\cdot}6875 - {\cdot}75) \times 100}{1{\cdot}6875} = 55{\cdot}5 \text{ per cent.}$$

$$\text{Rivets} = \frac{d^2 \times {\cdot}7854 \times \text{No.} \times 23 \times 100}{P \times T \times 28} = \frac{{\cdot}75^2 \times {\cdot}7854 \times 1 \times 23 \times 100}{1{\cdot}6875 \times {\cdot}375 \times 28} = 57{\cdot}3.$$

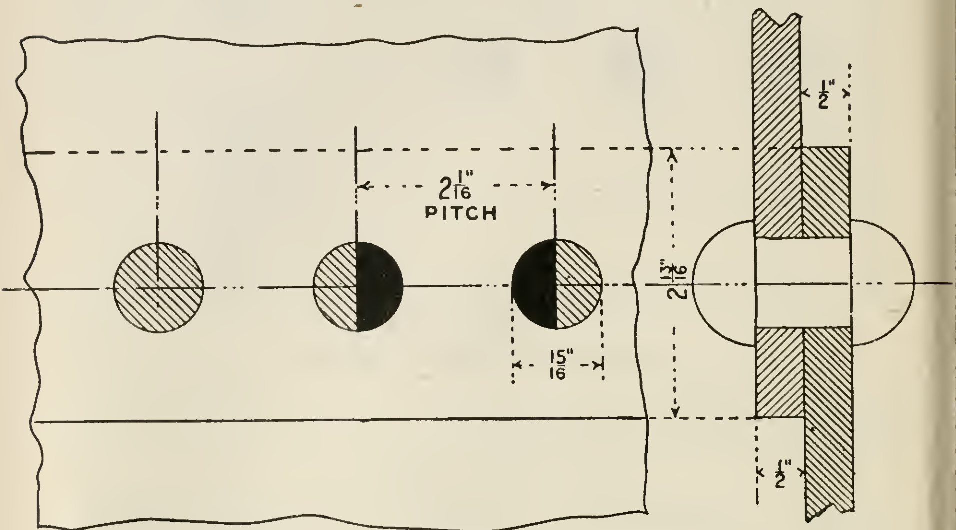
The joint strength is therefore equal to 55·5 per cent. of the solid plate.

No. 2 (Sketch No. 9). Plate $\frac{1}{2}$ inch thick, single riveted lap joint.

$$\text{Rivet diameter} = 1{\cdot}2 \times \sqrt{T} = 1{\cdot}2 \times \sqrt{\cdot 5} = {\cdot}848 \text{ in., say } \tfrac{15}{16} \text{ in. Diameter of rivet.}$$

$$\text{Rivet Pitch} = \frac{100 \times \text{Rivet diameter}}{100 - \text{joint}} = \frac{100 \times {\cdot}9375}{100 - 55} = 2{\cdot}08 \text{ in., say } 2\tfrac{1}{16} \text{ in. pitch.}$$

$$\text{Width of Lap} = {\cdot}9375 \times 3 = 2{\cdot}8125 \text{ in.} = 2\tfrac{13}{16} \text{ in.}$$



No. 9.—Single Riveting.
(Furnace or Combustion Chamber.)

Strength of Joint.

$$\text{Seam} = \frac{(p-d) \times 100}{P} = \frac{(2{\cdot}0625 - {\cdot}9375) \times 100}{2{\cdot}0625} = 54{\cdot}5 \text{ per cent.}$$

$$\text{Rivets} = \frac{d^2 \times {\cdot}7854 \times \text{No.} \times 23 \times 100}{P \times T \times 28} = \frac{{\cdot}9375^2 \times {\cdot}7854 \times 1 \times 23 \times 100}{2{\cdot}0625 \times {\cdot}5 \times 28} = 54{\cdot}9 \text{ per cent.}$$

Joint strength (smaller) = 54·5 per cent. of solid plate.
Width of lap = ·9375 × 3 = 2·8125 in., or $2\frac{13}{16}$ in.

No. 3 (Sketch No. 10). Plate $\frac{1}{2}$ inch thick, double riveted lap joint (zig-zag).

$$\text{Rivet diameter} = 1{\cdot}2 \times \sqrt{T} = 1{\cdot}2 \times \sqrt{\cdot 5} = {\cdot}848, \text{ say } \tfrac{15}{16} \text{ in. diameter of rivet.}$$

$$\text{Rivet Pitch} = \frac{100 \times \text{Rivet diameter}}{100 - \text{joint}} = \frac{100 \times {\cdot}9375}{100 - 70} = 3{\cdot}125 \text{ pitch } (3\tfrac{1}{8} \text{ in.}).$$

To prove strength of joint.

$$\text{Plate at Seam} = \frac{3{\cdot}125 - {\cdot}9375}{3{\cdot}125} \times 100 = 70 \text{ per cent.}$$

$$\text{Rivet Section} = \frac{{\cdot}9375^2 \times {\cdot}7854 \times 100 \times 2 \times 23}{3{\cdot}125 \times {\cdot}5 \times 28} = 72{\cdot}4 \text{ per cent.}$$

Diameter an

NOTE.—In
$\frac{15}{16}$ in. is fixed c

Cent

Distance be

RULE—
$V = \frac{\sqrt{(1}}{}$

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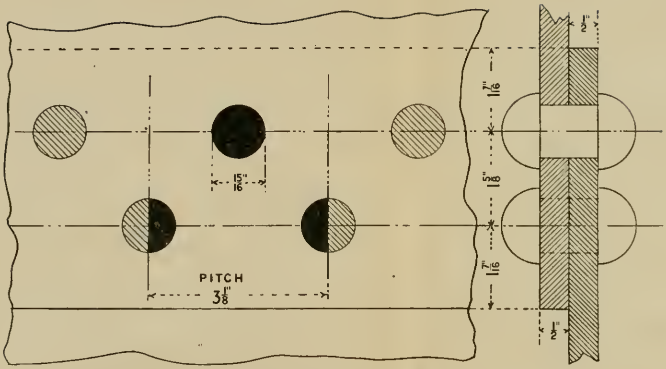


No. 10.—Double Riveting (Zig-zag).

(Furnace or Combustion Diameter.)

Diameter and Pitch of Rivets, &c.

Diameter of Rivets $= 1{\cdot}2 \times \sqrt{\cdot 5} = 1{\cdot}2 \times {\cdot}707 = {\cdot}848$ in., say $\frac{7}{8}$ in., or $\frac{15}{16}$ in.

NOTE.—In certain cases it is advisable to make the rivet fully the size found by the rule. In this case $\frac{15}{16}$ in. is fixed on as the diameter.

$$\text{Pitch} = \frac{100 \times \text{rivet diameter}}{100 - \text{joint}} = \frac{100 \times {\cdot}9375}{100 - 70} = 3{\cdot}125 \text{ in., or } 3\tfrac{1}{8} \text{ io.}$$

Centre of Rivet to edge of plate = Rivet diameter $\times 1{\cdot}5 = {\cdot}9375 \times 1{\cdot}5 = 1{\cdot}40625$ in., or $1\frac{7}{16}$ in.

Distance between rivet rows (V) (Zig-zag Riveting).

RULE—

$$V = \frac{\sqrt{(11 \times p + 4 \times d) \times (p + 4 \times d)}}{10} = \frac{\sqrt{(11 \times 3{\cdot}125 + 4 \times {\cdot}9375) \times (3{\cdot}125 + 4 \times {\cdot}9375)}}{10} =$$

$1{\cdot}61$ in., say $1\frac{5}{8}$ in., between rivet rows.

NOTE.—The average strength of double riveted joints for thin plates = 70 per cent. of solid plate.

[To face page 80.

The joint strength is therefore equal to 70 per cent. of the solid plate.

It should be remembered that the shearing strength of steel rivets is only to be taken as 23 tons per square inch, whereas the tensile strength of the steel plate is taken as 28 tons per square inch.

The distance between the rows of rivets can be calculated as follows:—

$$\text{Distance between Rivet Rows (chain riveting)} = \frac{4 \times \text{diameter of rivet} + 1}{2} =$$

$$\frac{4 \times \cdot 9375 + 1}{2} = \frac{3 \cdot 7500 + 1}{2} = 2 \cdot 375 \text{ in. } (2\tfrac{3}{8} \text{ in.}).$$

$$\text{Distance between Rivet Rows (zig-zag riveting)} =$$

$$\frac{\sqrt{(11 \times \text{pitch} + 4 \times \text{Rivet diameter}) \times (\text{pitch} + 4 \times \text{Rivet diameter})}}{10} =$$

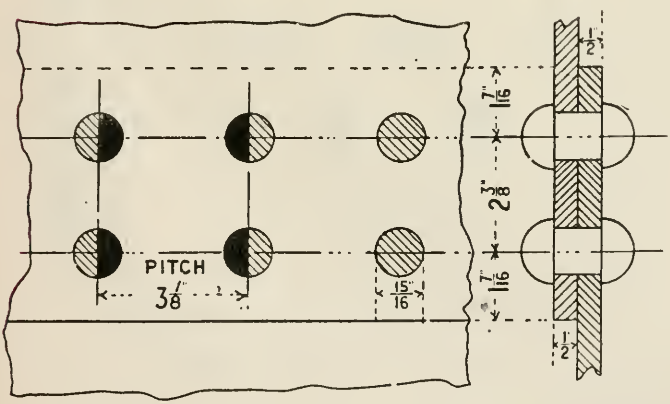
$$\frac{\sqrt{(11 \times 3 \cdot 125 + 4 \times \cdot 9375) \times (3 \cdot 125 + 4 \times \cdot 9375)}}{10} =$$

$$\frac{\sqrt{(34 \cdot 375 + 37 \cdot 5) \times (3 \cdot 125 + 3 \cdot 75)}}{10} =$$

$$\frac{\sqrt{38 \cdot 125 \times 6 \cdot 875}}{10} =$$

$$\frac{\sqrt{262 \cdot 109375}}{10} =$$

$$\frac{16 \cdot 19}{10} = 1 \cdot 619 \text{ in., say } 1\tfrac{5}{8} \text{ in.}$$



No. 11.—Double Riveting (Chain).

(Furnace or Combustion Chamber.)

In all cases the distance from edge of plate to centre of rivet = Rivet diameter × 1·5.

Therefore, $\frac{15}{16}$ in. × 1·5 = 1·406 in., say $1\frac{7}{16}$ in.

The joint strength is therefore equal to 70 per cent. of the solid plate.

It should be remembered that the shearing strength of steel rivets is only to be taken as 23 tons per square inch, whereas the tensile strength of the steel plate is taken as 28 tons per square inch.

The distance between the rows of rivets can be calculated as follows :—

$$\text{Distance between Rivet Rows (chain riveting)} = \frac{4 \times \text{diameter of rivet} + 1}{2} =$$

$$\frac{4 \times \cdot 9375 + 1}{2} = \frac{3\cdot 7500 + 1}{2} = 2\cdot 375 \text{ in. } (2\tfrac{3}{8} \text{ in.}).$$

Distance between Rivet Rows (zig-zag riveting) =

$$\frac{\sqrt{(11 \times \text{pitch} + 4 \times \text{Rivet diameter}) \times (\text{pitch} + 4 \times \text{Rivet diameter})}}{10} =$$

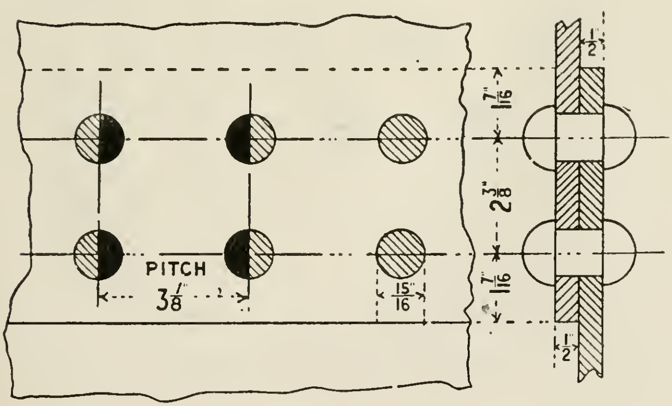
$$\frac{\sqrt{(11 \times 3\cdot 125 + 4 \times \cdot 9375) \times (3\cdot 125 + 4 \times \cdot 9375)}}{10} =$$

$$\frac{\sqrt{(34\cdot 375 + 37\cdot 5) \times (3\cdot 125 + 3\cdot 75)}}{10} =$$

$$\frac{\sqrt{38\cdot 125 \times 6\cdot 875}}{10} =$$

$$\frac{\sqrt{262\cdot 109375}}{10} =$$

$$\frac{16\cdot 19}{10} = 1\cdot 619 \text{ in., say } 1\tfrac{5}{8} \text{ in.}$$



No. 11.—Double Riveting (Chain).

(Furnace or Combustion Chamber.)

In all cases the distance from edge of plate to centre of rivet = Rivet diameter × 1·5.

Therefore, $\tfrac{15}{16}$ in. $\times 1\cdot 5 = 1\cdot 406$ in., say $1\tfrac{7}{16}$ in.

Joint strength (Sketch No. 11).

$$\text{Seam} = \frac{(p-d)\times 100}{P} = \frac{(3{\cdot}125 - {\cdot}9375)\times 100}{3{\cdot}125} = 70 \text{ per cent.}$$

$$\text{Rivets} = \frac{d^2 \times {\cdot}7854 \times \text{No.} \times 23 \times 100}{P \times T \times 28} = \frac{{\cdot}9375^2 \times {\cdot}7854 \times 2 \times 23 \times 100}{3{\cdot}125 \times {\cdot}5 \times 28} = 72{\cdot}4 \text{ per cent.}$$

Joint strength $=70$ per cent. of solid plate.

Distance between Rivet Rows (V) (chain riveting).

RULE—

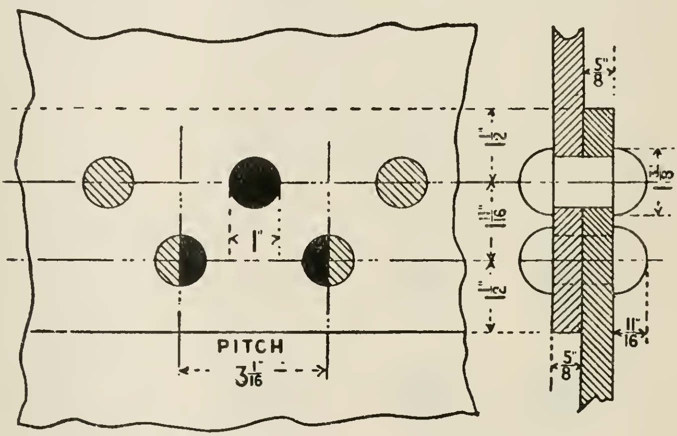
$$V = \frac{4 \times d + 1}{2} = \frac{4 \times {\cdot}9375 + 1}{2} = 2{\cdot}375 \text{ in., or } 2\tfrac{3}{8} \text{ in.}$$

No. 4 (Sketch No. 12). Plate $\frac{5}{8}$ inch thick, double riveted lap joint (zig-zag).

Rive diameter $= 1{\cdot}2 \times \sqrt{T} = 1{\cdot}2 \times \sqrt{{\cdot}625} = {\cdot}948$ in., say 1 in., diameter.

$$\text{Rivet Pitch} = \frac{100 \times \text{Rivet diameter}}{100 - \text{joint}} = \frac{100 \times 1}{100 - 67} = 3{\cdot}03 \text{ in., say } 3\tfrac{1}{16} \text{ in. pitch.}$$

NOTE.—Take 67 per cent. as average strength of double riveted joints.



No. 12.—Double Riveting (Zig-zag).

(Furnace or Combustion Chamber.)

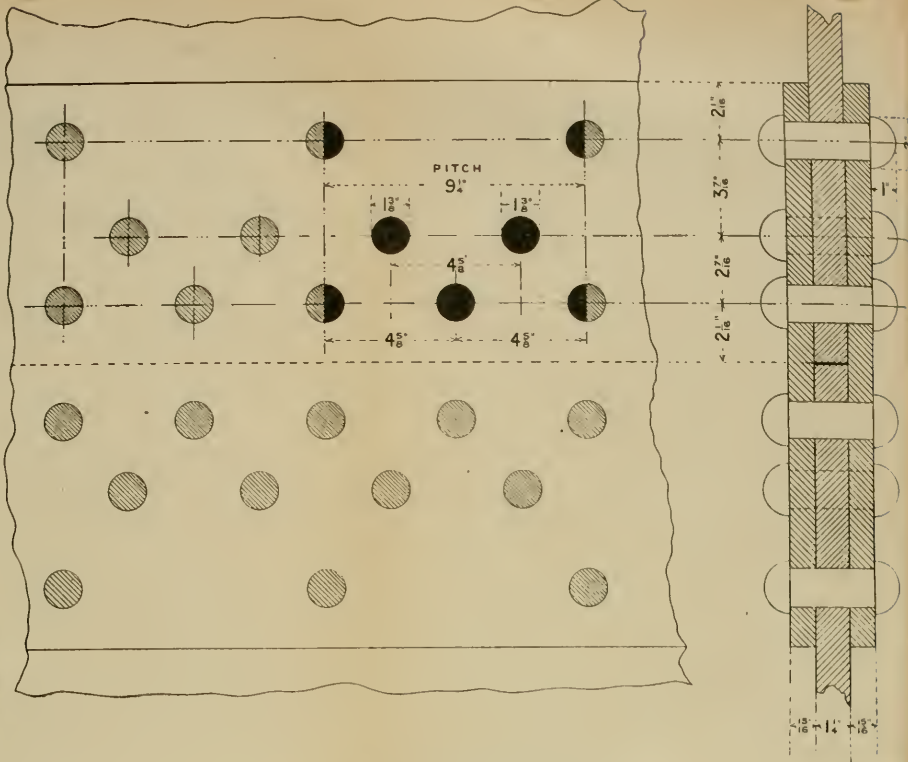
Joint strength.

$$\text{Seam} = \frac{(p-d)\times 100}{P} = \frac{(3{\cdot}0625 - 1)\times 100}{3{\cdot}0625} = 67{\cdot}3 \text{ per cent.}$$

$$\text{Rivets} = \frac{d^2 \times {\cdot}7854 \times 2 \times 23 \times 100}{P \times T \times 28} = \frac{1^2 \times {\cdot}7854 \times 2 \times 23 \times 100}{3{\cdot}0625 \times {\cdot}625 \times 28} = 67{\cdot}4 \text{ per cent.}$$

Joint strength (smaller) $= 67{\cdot}3$ per cent. of solid plate.

NOTE.—As the plate thickness increases the joint strength of single and double riveting decreases.



No. 13.—Double Butt Strap Joint.

(Longitudinal Shell Seams.)

Five Rivets in a pitch are in double shear, which increases the rivet section strength 1·875 times over that of single shear.

Diameter and Pitch of Rivets, &c.

Rivet diameter $= 1{\cdot}2 \times \sqrt{1{\cdot}25} = 1{\cdot}33$ in., say $1\frac{3}{8}$ in.

Rivet Pitch $= \frac{100 \times 1{\cdot}375}{100 - 85} = 9{\cdot}16$ in., say $9\frac{1}{4}$ in.

NOTE.—Strength of double butt strap joints averages 85 per cent. of solid plate.

Distance between outer rivet rows (V).

RULE—

$$V = \sqrt{\left(\frac{11 \times p}{20} + d\right) \times \left(\frac{p}{20} + d\right)} = \sqrt{\left(\frac{11 \times 9{\cdot}25}{20} + 1{\cdot}375\right) \times \left(\frac{9{\cdot}25}{20} + d\right)} = 3{\cdot}44 \text{ in., say } 3\tfrac{7}{16} \text{ in.}$$

Distance between inner rivet rows (V_1).

RULE—

$$V_1 = \frac{\sqrt{(11 \times p + 8 \times d) \times (p + 8 \times d)}}{20} = \frac{\sqrt{(11 \times 9{\cdot}25 + 8 \times 1{\cdot}375) \times (9{\cdot}25 + 8 \times 1{\cdot}375)}}{20} \; 2{\cdot}38 \text{ in., say } 2\tfrac{7}{16} \text{ in.}$$

Rivet centre to edge of plate $= 1{\cdot}5 \times 1{\cdot}375 = 2{\cdot}0625$ in., or $2\frac{1}{16}$ in.

To prove joint strength.

$$\text{Seam} = \frac{(p - d) \times 100}{P} = \frac{(9{\cdot}25 - 1{\cdot}375) \times 100}{9{\cdot}25} = 85{\cdot}1 \text{ per cent.}$$

$$\text{Rivets} = \frac{d^2 \times {\cdot}7854 \times \text{No.} \times 23 \times 1{\cdot}875 \times 100}{P \times T \times 28} = \frac{1{\cdot}375^2 \times {\cdot}7854 \times 5 \times 23 \times 1{\cdot}875 \times 100}{9{\cdot}25 \times 1{\cdot}25 \times 28} = 97{\cdot}6 \text{ per cent.}$$

NOTE.—The number of rivets in a pitch is five, as will be seen from the shaded rivet sections shown in the sketch; notice that the half rivet sections enclosed within the pitch limit require to be counted in.

Combined Rivet and Seam Section.

RULE—

$$\frac{(p - (d \times 2)) \times 100}{P} + \frac{\text{rivet section strength}}{\text{No. in a pitch}} = \frac{(9{\cdot}25 - (1{\cdot}375 \times 2)) \times 100}{9{\cdot}25} + \frac{97{\cdot}6}{5} = 89{\cdot}8 \text{ per cent.}$$

The joint strength (smaller) is, therefore, equal to 85·1 per cent. of the solid plate.

NOTE.—For double shear on the rivets, as in this class of joint, the Board of Trade now allow 1·875 instead of 1·75 as formerly.

$$\text{Butt Strap Thickness} = \frac{5 \times \text{plate thickness} \times (\text{Pitch} - \text{Rivet diameter})}{8 \times (\text{Pitch} - \text{twice Rivet diameter})} =$$

$$\frac{5 \times 1{\cdot}25 \times (9{\cdot}25 - 1{\cdot}375)}{8 \times (9{\cdot}25 - 2{\cdot}75)} = {\cdot}94 \text{ in., say } \tfrac{15}{16} \text{ in}$$

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Distance between Rivet Rows (V) (Zig-zag Riveting).

RULE—

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

$$\frac{\sqrt{(11\times 3{\cdot}0625+4\times 1)\times(3{\cdot}0625+4\times 1)}}{10}=1{\cdot}63\text{ in., say }1\tfrac{5}{8}\text{ in. between rows.}$$

Rivet centre to edge of plate $=1\times 1{\cdot}5=1\tfrac{1}{2}$ in.

To prove strength of joint.

$$\text{Plate at Seam}=\frac{3{\cdot}0625-1}{3{\cdot}0625}\times 100=67\text{ per cent.}$$

$$\text{Rivet Section}=\frac{1^2\times{\cdot}7854\times 100\times 23\times 2}{3{\cdot}0625\times{\cdot}625\times 28}=67\text{ per cent.}$$

The width of lap can, if required, be calculated by the method shown in example No. 3.

Distance from centre of rivet to edge of plate $=1$ inch $\times 1{\cdot}5=1{\cdot}5$ inches.

No. 5. Double Butt Strap Joints (Sketch No. 13).—As before stated, this type of joint is only fitted in the longitudinal shell seams of boilers.

Shell plate $1\tfrac{1}{4}$ inches thick, joint strength to be taken as 85 per cent.

Then,

Rivet diameter $=1{\cdot}2\times\sqrt{T}=1{\cdot}2\times\sqrt{1{\cdot}25}=1{\cdot}32$ in., say $1\tfrac{3}{8}$ in. diameter.

$$\text{Rivet Pitch}=\frac{100\times\text{Rivet diameter}}{100-\text{joint}}=\frac{100\times 1{\cdot}375}{100-85}=9{\cdot}16\text{ in., say }9\tfrac{1}{4}\text{ in. pitch.}$$

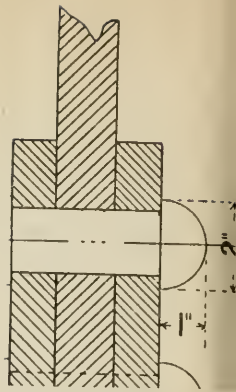
To prove strength of joint.

$$\text{Plate at Seam}=\frac{9{\cdot}25-1{\cdot}375}{9{\cdot}25}\times 100=85\text{ per cent.}$$

$$\text{Rivet Section}=\frac{1{\cdot}375^2\times{\cdot}7854\times 23\times 5\times 1{\cdot}875\times 100}{9{\cdot}25\times 1{\cdot}25\times 28}=97{\cdot}6\text{ per cent.}$$

Referring to Sketch No. 13, the joint strength or smaller result is therefore equal to 85 per cent. of the solid plate.

It will be noticed that in this type of joint the rivets are unequally distributed, the outer row having every second rivet omitted. The



Distance between Rivet Rows (V) (Zig-zag Riveting).

RULE—

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

$$\frac{\sqrt{(11 \times 3{\cdot}0625 + 4 \times 1) \times (3{\cdot}0625 + 4 \times 1)}}{10} = 1{\cdot}63 \text{ in., say } 1\tfrac{11}{16} \text{ in. between rows.}$$

Rivet centre to edge of plate $= 1 \times 1{\cdot}5 = 1\tfrac{1}{2}$ in.

To prove strength of joint.

$$\text{Plate at Seam} = \frac{3{\cdot}0625 - 1}{3{\cdot}0625} \times 100 = 67 \text{ per cent.}$$

$$\text{Rivet Section} = \frac{1^2 \times {\cdot}7854 \times 100 \times 23 \times 2}{3{\cdot}0625 \times {\cdot}625 \times 28} = 67 \text{ per cent.}$$

The width of lap can, if required, be calculated by the method shown in example No. 3.

Distance from centre of rivet to edge of plate $= 1$ inch $\times 1{\cdot}5 = 1{\cdot}5$ inches.

No. 5. Double Butt Strap Joints (Sketch No. 13).—As before stated, this type of joint is only fitted in the longitudinal shell seams of boilers.

Shell plate $1\tfrac{1}{4}$ inches thick, joint strength to be taken as 85 per cent.

Then,

Rivet diameter $= 1{\cdot}2 \times \sqrt{T} = 1{\cdot}2 \times \sqrt{1{\cdot}25} = 1{\cdot}32$ in., say $1\tfrac{3}{8}$ in. diameter.

$$\text{Rivet Pitch} = \frac{100 \times \text{Rivet diameter}}{100 - \text{joint}} = \frac{100 \times 1{\cdot}375}{100 - 85} = 9{\cdot}16 \text{ in., say } 9\tfrac{1}{4} \text{ in. pitch.}$$

To prove strength of joint.

$$\text{Plate at Seam} = \frac{9{\cdot}25 - 1{\cdot}375}{9{\cdot}25} \times 100 = 85 \text{ per cent.}$$

$$\text{Rivet Section} = \frac{1{\cdot}375^2 \times {\cdot}7854 \times 23 \times 5 \times 1{\cdot}875 \times 100}{9{\cdot}25 \times 1{\cdot}25 \times 28} = 97{\cdot}6 \text{ per cent.}$$

Referring to Sketch No. 13, the joint strength or smaller result is therefore equal to 85 per cent. of the solid plate.

It will be noticed that in this type of joint the rivets are unequally distributed, the outer row having every second rivet omitted. The omission of these rivets has the effect of raising up the joint strength, for, were the rivet included, we should then have a very strong rivet section strength, but a very weak plate section strength, and, as the smaller section strength must in all cases be taken as the *joint strength*, the joint would be weak. By omitting every second rivet in the outer row the rivet section strength is decreased, and the plate section increased; therefore the smaller result being now higher than before, the joint strength is proportionally greater.

There is no benefit to be got by having a high rivet section strength, and a low plate section strength, as the smaller or joint strength governs the safe working pressure to be allowed on the plates.

The result on the joint strength by giving equal rivets in each row will now be shown.

Observe that the greatest (only) pitch is now $4\frac{5}{8}$ inches, and that the number of rivets included in *one* pitch is *three* instead of five as previously.

$$\text{Therefore, Plate at seam} = \frac{4 \cdot 625 - 1 \cdot 375}{4 \cdot 625} \times 100 = 70 \cdot 2 \text{ per cent.}$$

$$\text{Rivet section} = \frac{1 \cdot 375^2 \times \cdot 7854 \times 23 \times 3 \times 1 \cdot 875 \times 100}{4 \cdot 625 \times 1 \cdot 25 \times 28} = 118 \text{ per cent.}$$

1. As shown above, the joint strength has dropped from 85 per cent. to 70·2 per cent., which proves that the omission of every alternate rivet in the outer row as actually carried out in practice has the effect of raising up the joint strength to a maximum.

2. The rivet section strength is *increased* from 97·6 per cent. to 118 per cent., but, as the *smaller* result only must always be taken, this simply represents rivet section strength wasted. The re-adjustment of rivet section with every second rivet omitted in the outer row takes away from the rivet section and gives to the plate section, or, what the rivet section loses the plate section gains, and the joint strength is proportionately increased.

Circumferential Riveting (Sketch No. 14).

$$\text{Seam strength} = \frac{(p - d) \times 100}{P} = \frac{(3 \cdot 5 - 1 \cdot 375) \times 100}{3 \cdot 5} = 60 \cdot 7 \text{ per cent.}$$

$$\text{Rivet strength} = \frac{d^2 \times \cdot 7854 \times \text{No.} \times 23 \times 100}{P \times T \times 28} = \frac{1 \cdot 375^2 \times \cdot 7854 \times 2 \times 23 \times 100}{3 \cdot 5 \times 1 \cdot 25 \times 28} = 55 \cdot 7 \text{ per cent.}$$

NOTE.—The shell plates are $1\frac{1}{4}$ in. thick.

Joint strength = 55·7 per cent. of solid plate.

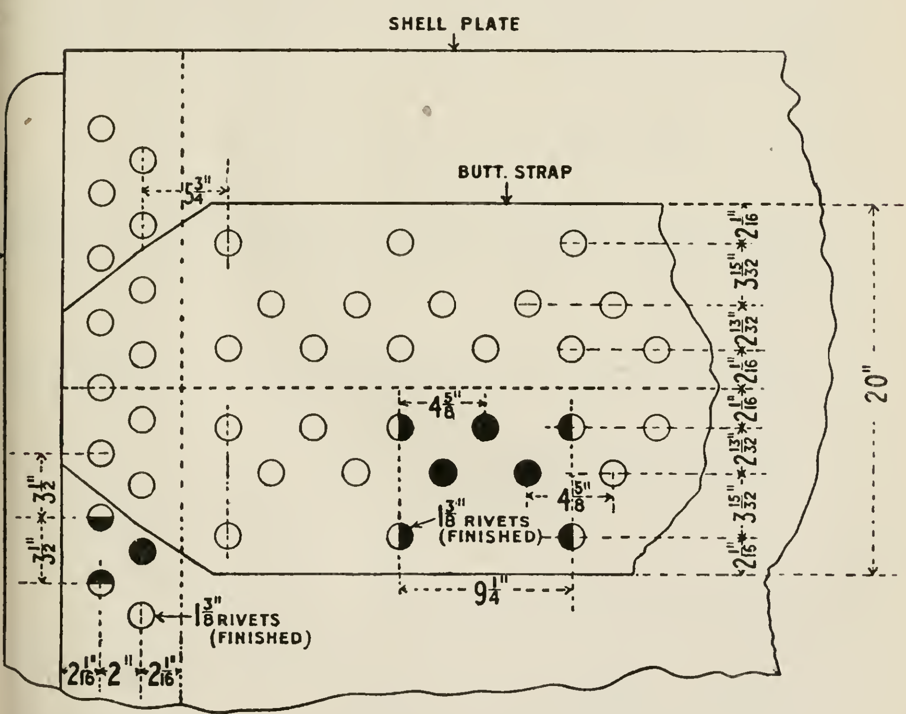
This is sufficient for the end seams, which according to Board of Trade requirements must not be less in strength than *half that* of the longitudinal seams, which in this case is exceeded by a fair margin.

Distance between Rivet Rows (V).

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

$$\frac{\sqrt{(11 \times 3{\cdot}5 + 4 \times 1{\cdot}375) \times (3{\cdot}5 + 4 \times 1{\cdot}375)}}{10} = 1{\cdot}99 \text{ in., say } 2 \text{ in.}$$

Distance from rivet centre to edge of plate $= 1{\cdot}5 \times 1{\cdot}375 = 2{\cdot}0625$ in., or $2\frac{1}{16}$ in.



No. 14.—End Circumferential and Longitudinal Shell Riveting.

Observe that there are five rivets in a pitch (sectioned black) longitudinally, and two rivets in a pitch circumferentially.

For working out of longitudinal shell riveting see Sketch No. 13.

Combined Strength of Seam and Rivets.

In double butt strap joints the combined strength of seam and rivets exceeds that of either the seam strength or rivet section strength, as can easily be proved by the following rule. It should be noted that at the outer rivet rows *one* rivet section is taken out of the plate metal at the large pitch (see No. 13), whereas at the inner rivet rows *two* rivet sections are cut out of the same pitch length: the seam section is therefore weakest at the inner rows, as the formula would then be, $\frac{p-(d \times 2)}{P}$ instead of $\frac{p-d}{P}$ as for the outer row: but as a counterbalance to this it should be noted that before the plate between the inner rivet rows would fracture, one rivet section of the outer row would also require to be sheared, so that we have the shearing strength of one rivet added to the plate strength at seam. To obtain the strength of one single rivet, merely divide the rivet section strength as found in the usual way by the number of rivets in a pitch, which, as in the case of double butt strap joints, is five.

$$\text{Therefore, Combined strength} = \frac{p-(2 \times d)}{P} \times 100 + \frac{\text{Rivet section strength}}{5}.$$

As this result is in every case in excess of either rivet or seam section strengths, it follows that it is correct to take the smaller of the latter two as the joint strength, which is always done in actual practice.

EXAMPLE.—Work out the combined strength of rivet and seam section for the riveting shown in Sketch No. 13.

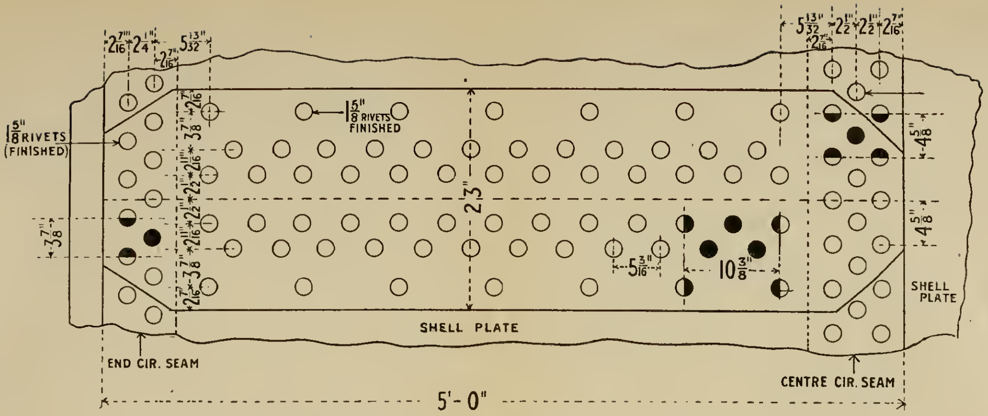
$$\text{Combined strength} = \frac{9{\cdot}25-(1{\cdot}375 \times 2)}{9{\cdot}25} \times 100 + \frac{97{\cdot}6}{5} =$$

$$\frac{9{\cdot}25-2{\cdot}75}{9{\cdot}25} \times 100 + 20{\cdot}5 = 89{\cdot}8 \text{ per cent.}$$

So that the results come out as follows :—

Strengths	Seam Section	= 85·1	per cent. of solid plate (joint).
	Rivet ,,	= 97·6	,, ,,
	Combined ,,	= 89·8	,, ,,

Notice that the joint is equal to the smallest result of the three.



No. 15.—Longitudinal, End Circumferential, and Centre Circumferential Riveting.

NOTE.—The limit rivet pitch allowed by the Board of Trade is $10\frac{1}{2}$ inches. The darkened rivets indicate the number in a pitch for each joint shown.

Joint Strengths, &c. Longitudinal Shell Seams (plate $1\frac{5}{8}$ inches thick).

$$\text{Seam strength} = \frac{(p-d)\times 100}{P} = \frac{(10\cdot375 - 1\cdot625)\times 100}{10\cdot375} = 84\cdot3 \text{ per cent.}$$

$$\text{Rivet strength} = \frac{d^2\times\cdot7854\times\text{No.}\times 23\times 1\cdot875\times 100}{P\times T\times 28} = \frac{1\cdot625^2\times\cdot7854\times 5\times 23\times 1\cdot875\times 100}{10\cdot375\times 1\cdot625\times 28} = 94\cdot7 \text{ per cent.}$$

Joint strength (smaller) = 84·3 per cent. of solid plate.

It will be obvious that the joint strength would be improved by reducing the rivet section strength, as this would result in raising the seam section strength and therefore the joint strength.

NOTE.—In this example the rivet diameter and plate thickness are of the same size.

Distance between Outer Rivet Rows (V).

$$V = \sqrt{\left(\frac{11\times p}{20}+d\right)\times\left(\frac{P}{20}+d\right)} = \sqrt{\left(\frac{11\times 10\cdot375}{20}+1\cdot625\right)\times\left(\frac{10\cdot375}{20}+1\cdot625\right)} = 3\cdot9 \text{ in., say } 3\tfrac{7}{8} \text{ in.}$$

Distance between Inner Rivet Rows (V_1).

$$V_1 = \frac{\sqrt{(11\times p+8\times d)\times(p+8\times d)}}{20} = \frac{\sqrt{(11\times 10\cdot375+8\times 1\cdot625)\times(10\cdot375+8\times 1\cdot625)}}{20} = 2\cdot7 \text{ in., say } 2\tfrac{11}{16} \text{ in.}$$

Rivet centre to edge of plate $= 1\cdot5\times 1\cdot625 = 2\cdot4375$ in., or $2\frac{7}{16}$ in.

Circumferential End Shell Seams.

$$\text{Seam section} = \frac{(p-d)\times 100}{P} = \frac{(3\cdot875-1\cdot625)\times 100}{3\cdot875} = 58 \text{ per cent.}$$

$$\text{Rivet section} = \frac{d^2\times\cdot7854\times\text{No.}\times 23\times 100}{P\times T\times 28} = \frac{1\cdot625^2\times\cdot7854\times 2\times 23\times 100}{3\cdot875\times 1\cdot625\times 28} = 54\cdot1 \text{ per cent.}$$

Joint strength (smaller) = 54·1 per cent. of solid strength.

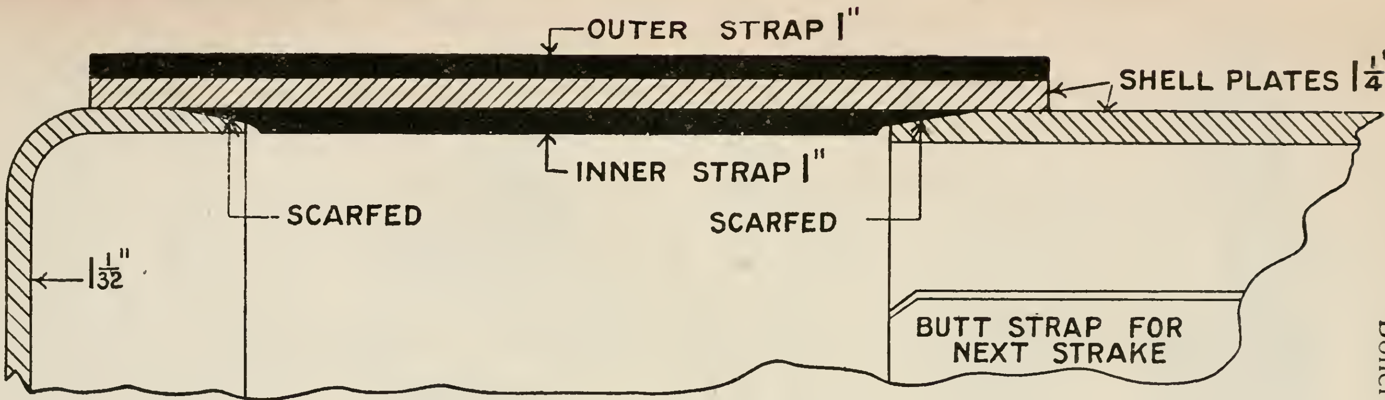
Circumferential Centre Shell Seams.—The Board of Trade limit strength for this joint is 65 per cent., which usually necessitates three rivets in a pitch; this additional strength is required to meet the barrelling tendency of the boiler under pressure, as the centre seams, unlike the end seams, are unsupported, whereas the end seams are stayed by the boiler end plates.

$$\text{Seam section} = \frac{(p-d)\times 100}{P} = \frac{(4\cdot625-1\cdot625)\times 100}{4\cdot625} = 65 \text{ per cent. (nearly).}$$

$$\text{Rivet section} = \frac{d^2\times\cdot7854\times\text{No.}\times 23\times 100}{P\times T\times 28} = \frac{1\cdot625^2\times\cdot7854\times 3\times 23\times 100}{4\cdot625\times 1\cdot625\times 28} = 68 \text{ per cent. (nearly).}$$

Distance between Rivet Rows (V).

$$V = \frac{\sqrt{(11\times p+4\times d)\times(p+4\times d)}}{10} = \frac{\sqrt{(11\times 4\cdot625+4\times 1\cdot625)\times(4\cdot625+4\times 1\cdot625)}}{10} = 2\cdot5 \text{ in., or } 2\tfrac{1}{2} \text{ in.}$$



No. 16.—Double Butt Straps.

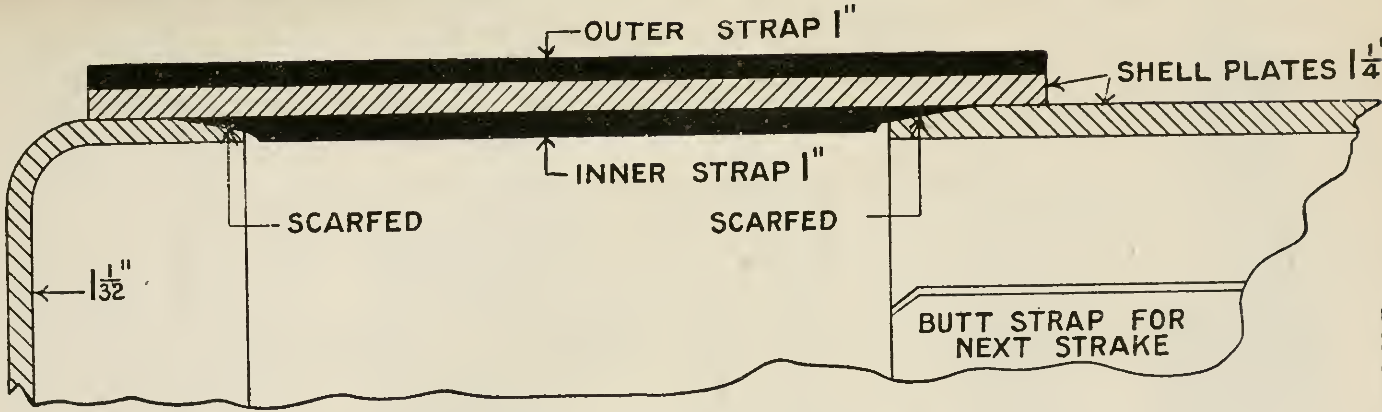
(Longitudinal Sectional View.)

Notice how the inner strap is bevelled away at the ends so that it may fit into the end plate and centre shell plate.

To Find Thickness of Straps (assume pitch of rivets $9\frac{1}{2}$ inches, and diameter of rivets $1\frac{3}{8}$ inches).

RULE—

$$\text{Thickness} = \frac{5 \times T \times (p - d)}{8 \times (p - 2 \times d)} = \frac{5 \times 1{\cdot}25 \times (9{\cdot}5 - 1{\cdot}375)}{8 \times (9{\cdot}5 - 1{\cdot}375 \times 2)} = {\cdot}94 \text{ in., say 1 in. thick (each strap).}$$



No. 16.—Double Butt Straps.

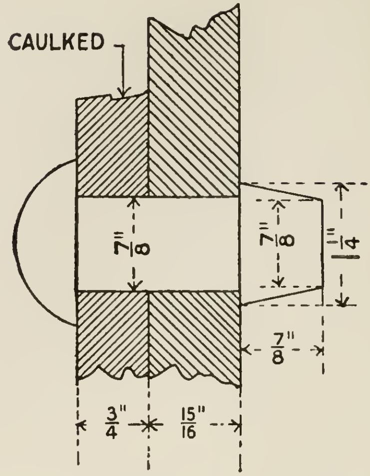
(Longitudinal Sectional View.)

Notice how the inner strap is bevelled away at the ends so that it may fit into the end plate and centre shell plate.

To Find Thickness of Straps (assume pitch of rivets $9\frac{1}{2}$ inches, and diameter of rivets $1\frac{3}{8}$ inches).

RULE—

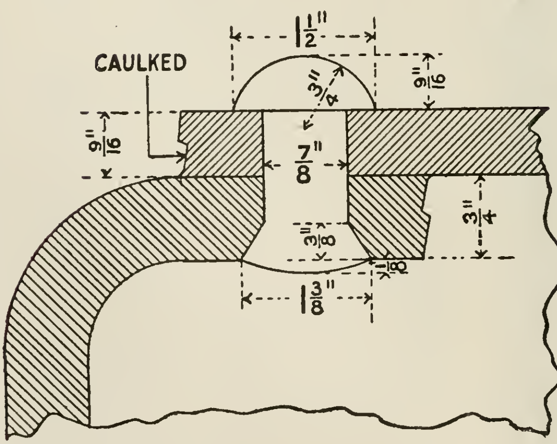
$$\text{Thickness} = \frac{5 \times T \times (p - d)}{8 \times (p - 2 \times d)} = \frac{5 \times 1{\cdot}25 \times (9{\cdot}5 - 1{\cdot}375)}{8 \times (9{\cdot}5 - 1{\cdot}375 \times 2)} = {\cdot}94 \text{ in., say 1 in. thick (each strap).}$$



No. 17.—Machine Riveting.

(With Dimensions.)

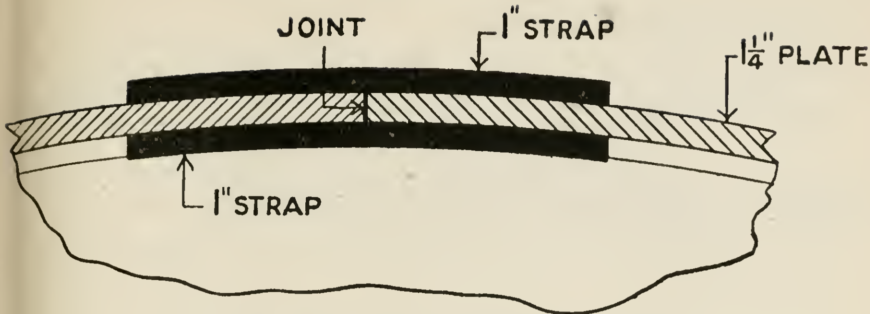
The dimensions should be carefully studied and compared.



No. 18.—Riveting and Caulking of Combustion Chamber Top.

(With Dimensions.)

Hand riveting is usually as shown, that is, the rivet is countersunk on one side, and snap or flat headed on the other side. Compare the plate thickness and rivet diameter, &c.



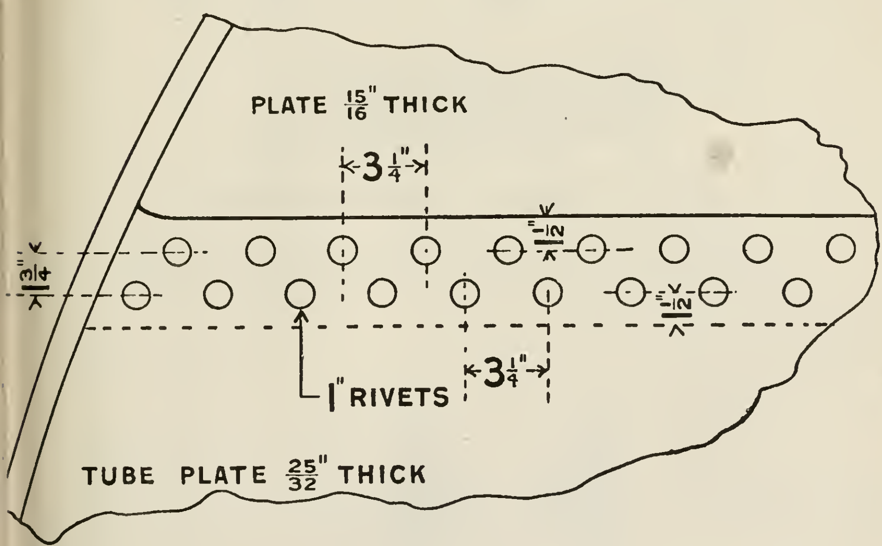
No. 19.—Shell Plates and Butt Straps.

This view, a fore and aft section, of the shell shows clearly how the butt straps sit in position.

To find Butt Strap Thickness.—Rivets $1\frac{3}{8}$ inches diameter, pitch 9 inches.

RULE—

$$\text{Thickness} = \frac{5 \times T \times (p - d)}{8 \times (p - (d \times 2))} = \frac{5 \times 1{\cdot}25 \times (9 - 1{\cdot}375)}{8 \times (9 - (1{\cdot}375 \times 2))} = 1 \text{ inch thick.}$$



No. 20.—View of Boiler End near Top.

(Showing Riveting of Tube Plate and Top End Plate.)

Joint strength.

$$\text{Seam Section} = \frac{p - d}{P} \times 100 = \frac{3{\cdot}25 - 1}{3{\cdot}25} \times 100 = 69{\cdot}2 \text{ per cent.}$$

$$\text{Rivet Section} = \frac{A \times N \times 100 \times 23}{P \times T \times 28} = \frac{1^2 \times {\cdot}7854 \times 2 \times 100 \times 23}{3{\cdot}25 \times {\cdot}78 \times 28} = 50{\cdot}8 \text{ per cent.}$$

The joint strength is, therefore, 50·8 per cent. of the solid plate.

Distance between Rivet Rows.

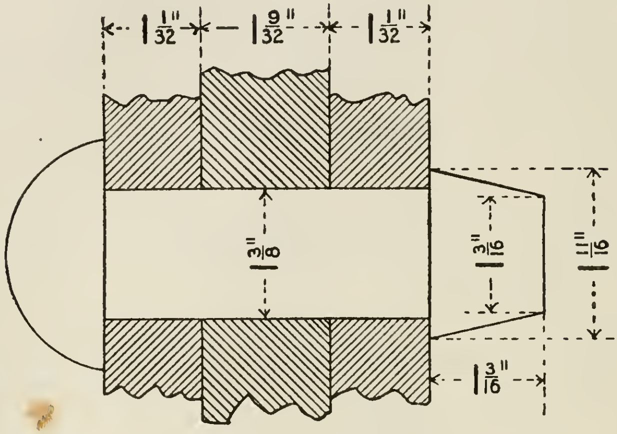
RULE—

$$\frac{\sqrt{(11 \times p + 4 \times d) \times (p + 4 \times d)}}{10} = \frac{\sqrt{(11 \times 3{\cdot}25 + 4 \times 1) \times (3{\cdot}25 + 4 \times 1)}}{10} = 1{\cdot}7 \text{ in., say } 1\tfrac{3}{4} \text{ in.}$$

Distance between edge of hole and edge of plate.

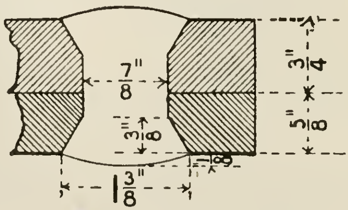
RULE—

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



No. 21.—Shell Plate Machine Riveting.

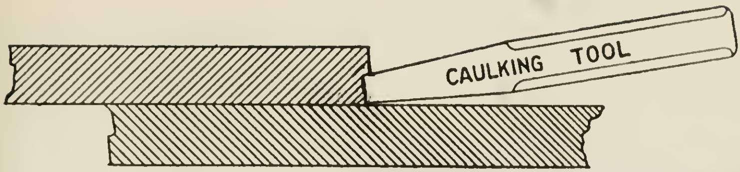
This sketch shows the shell plate and the two butt straps in section with the rivet in position; the dimensions are marked, and should be noted.



No. 22.—Hand Riveting.

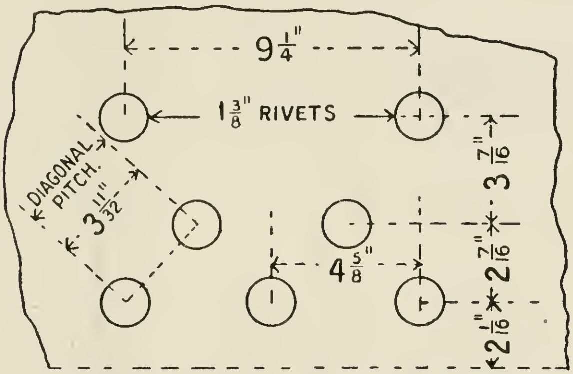
(Countersunk.)

The dimensions indicate the proportions of countersink, &c. adopted in hand riveting as employed in patching.



No. 23.—Caulking Tool for Boiler Plates.

The tool is shown in position for caulking a thick plate.



No. 24.—Diagonal Pitch of Rivets.

For the type of joint shown above the diagonal pitch of the rivets should not be less than that found by the following rule.

RULE—

$$\text{Diagonal Pitch} = \frac{3 \times \text{Pitch} + 4 \times \text{diameter}}{10},$$

$$\text{Therefore, Diagonal Pitch} = \frac{3 \times 9{\cdot}25 + 4 \times 1{\cdot}375}{10} = 3{\cdot}326 \text{ in., or say } 3\tfrac{11}{32} \text{ in.}$$

Steam Space Stays.

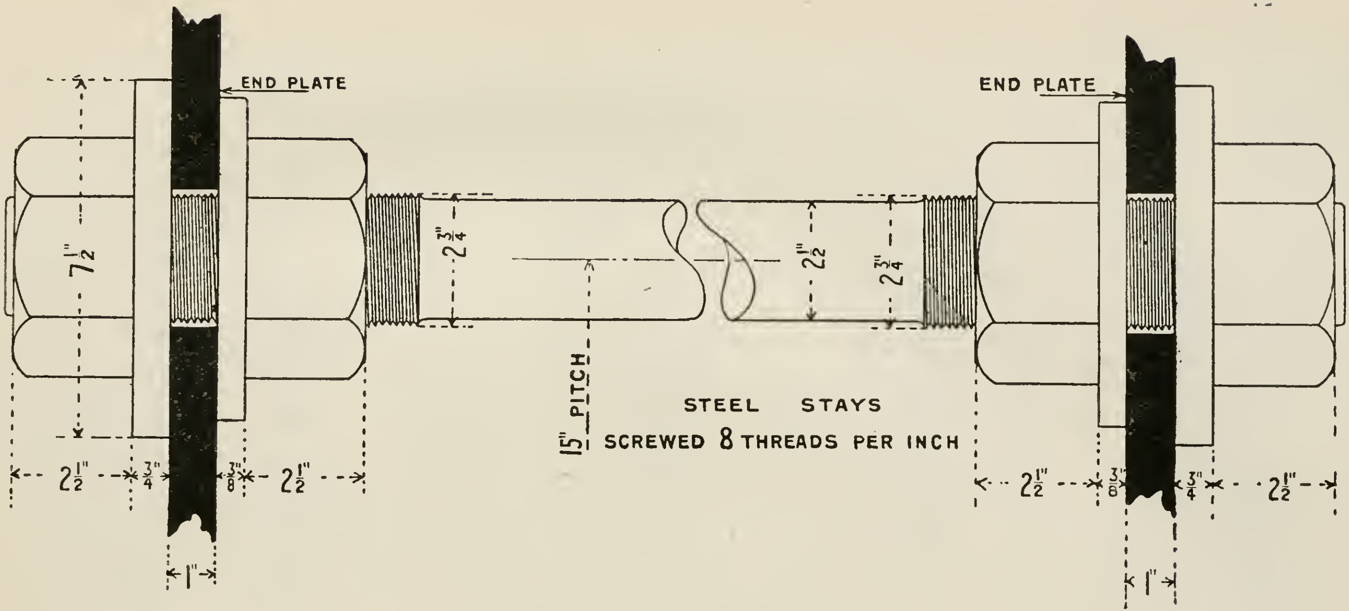
These steel stays range in diameter from $2\frac{1}{2}$ to $3\frac{1}{2}$ inches diameter, with a normal pitch of 16 inches for pressures of from 160 to 210 lbs. per square inch. The stays are secured to the plates by one of the following methods.

1. Holes cut in both plates and stays held in place by nuts and washers both inside and outside the plate at either end.

2. Stays screwed through both end plates and fitted with single nuts and washers on outside of plates only.

3. Holes cut in both end plates and stays held in place by means of a thin nut inside and a thick nut outside.

Occasionally the nuts are bevelled off as shown in the sketch, and



No. 25.—Steam Space Stay.
(With Double Nuts and Washers.)

The large steam space stays are often fixed as shown, and the Board of Trade Constant for this arrangement is 165.

To find Pitch of Stays (pressure 210 lbs.),

RULE—

$$C \times (T+1)^2 = (S-6) \times \text{Pressure.}$$

Therefore, $165 \times (16+1)^2 = (S-6) \times 210.$

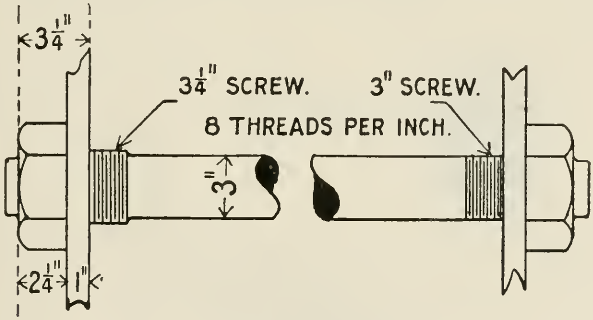
Then, $S = \dfrac{165 \times (16+1)^2}{210} + 6 = 233.$

So that, Pitch $= \sqrt{233} = 15{\cdot}2$ in., say 15 in., Pitch.

a layer of mastic cement filled in as a joint: in addition to this a serving of mastic cement is placed between the nut and washer, and between the washer and plate for the same purpose.

In the case of No. 1 arrangement, the outside washers are sometimes riveted to the plates and are then made of a diameter equal to three times that of the stay at thread. The stays are during manufacture staved up $\frac{1}{4}$ or $\frac{1}{2}$ inch at the ends (the stay being afterwards annealed) to allow of the cutting of the screw, which at bottom of thread should not be less in diameter than the body of the stay. Under working conditions these stays are subjected to a tensile stress, and should therefore be made out of the solid, as, if welded, the stress conditions mentioned would tend to open up the welded portion of the stay. The usual number of screw threads cut in the stay ends is from 6 to 8 per inch. These stays occasionally show signs of corrosion at or near the ends, the cause of which is most likely magnesium chloride gas contained in sea water feed and set free by the effects of heat.

When the stays are screwed into the plates, the plates are afterwards caulked round the stays to ensure tightness of joint.



No. 26.—Steam Space Stay.

(Pressure, 200 lbs. Pitch, 16 inches.)

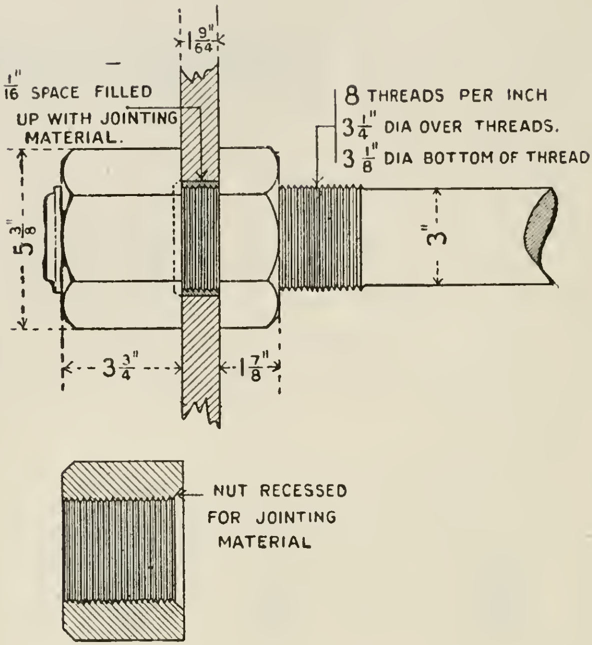
Stay diameter at smallest part = $2\frac{3}{4}$ inches.

Then, $$\frac{2{\cdot}75^2 \times {\cdot}7854 \times 9000}{16 \times 16} = 208 \text{ lbs. safe pressure.}$$

Observe that the stay being screwed into the plates is larger in diameter at one end than the other, the screwed portions being $3\frac{1}{4}$ inches diameter and 3 inches diameter.

The plates are carefully caulked round the stay at both ends and a touch of lead putty put on the nuts before screwing up tight.

NOTE.—The depth of nut plus plate thickness is equal to the diameter of stay over threads, or $2\frac{1}{4}$ inches + 1 inch = $3\frac{1}{4}$ inches.



No. 27.—Steam Space Stay.

(With Dimensions.)

(Pressure, 220 lbs. Pitch, 17 inches.

In this type of stay, the holes in the end plates are cut clear, and nuts are screwed on the stay ends inside and outside, the ends being staved up for the cutting of the screws (plus thread) Mastic cement or lead putty is filled into the clearance ($\frac{1}{16}$ inch).

NOTE.—If no washers are fitted on the nuts the depth of the outside nut must be increased to make up.

Flat Surfaces and Stays.

The flat (or nearly flat) surfaces of boilers are stayed as follows:—

1. Boiler end plates, by large stays in steam and water spaces.
2. Combustion chamber sides and backs, by small screwed stays.
3. „ „ tops, by girders and stays.
4. „ „ bottoms are self-staying, being semicircular.
5. End plates between tube nests, by doubling plates riveted on.

The strength of flat surfaces vary as the Surface supported (pitch of stays squared) and the Thickness of plate squared: this can be seen by examining the Rule, which is as follows:—

RULE—

$$C \times (T+1)^2 = (S-6) \times \text{Safe Pressure.}$$

$C = 100$ to 168, depending on conditions of construction.
$T =$ Plate Thickness in sixteenths.
$S =$ Surface supported (usually Pitch squared).

NOTE.—In ordinary practice C may be taken as 100 for combustion chamber plates, and 150 for boiler end plates.

If, then, the constants C, 6 and 1 are deleted, we have left T^2 and S, therefore the strength varies as these two terms.

TO TRANSPOSE THE RULE—

The following shows how the rule may be transposed to find either the Safe Pressure, the plate Thickness, or the Pitch of stays.

$$\text{1. Safe Pressure} = \frac{C \times (T+1)^2}{S-6} = \text{lbs.}$$

$$\text{2. Surface supported} = \frac{C \times (T+1)^2}{\text{Safe Pressure}} + 6 = S.$$

$$\text{Then,} \qquad \text{Pitch} = \sqrt{S}.$$

Notice that the **square root** of the surface supported is equal to the pitch of stays required.

$$\text{3. Thickness of Plate} = \sqrt{\frac{(S-6) \times \text{Pressure}}{C}} = (T+1).$$

$$\text{Then,} \qquad (T+1) - 1 = T \text{ (in sixteenths).}$$

Observe that the result brought out by the square root is $T+1$, that is *one more* than T, therefore 1 requires to be subtracted to obtain T (in sixteenths of an inch).

Data for End Plates and Stays (Sketch No. 27).—The safe working pressure is 220 lbs. per square inch, and the pitch of stays 17 inches; find the required thickness of end plate if the Constant is 168: also find the required diameter of stay.

Plate Thickness.

RULE—

$$C \times (T+1)^2 = (S-6) \times \text{Safe Pressure.}$$

Where, C = Constant, T = 16th in plate, S = Surface supported or Pitch².

$$\text{Then,} \quad 168 \times (T+1)^2 = (17^2 - 6) \times 220.$$

$$\text{Therefore,} \quad T = \sqrt{\frac{(17^2-6) \times 220}{168}} - 1 = \sqrt{\frac{(289-6) \times 220}{168}} - 1 = 18{\cdot}26.$$

$$\text{So that,} \quad \frac{18{\cdot}26}{16} = 1\tfrac{9}{64} \text{ in. thick.}$$

Observe that the result comes out in *sixteenths* of an inch, therefore 18·26 sixteenths is equal to $1\frac{9}{64}$ inch.

Stay Diameter.

RULE—

$$\text{Surface} \times \text{Safe Pressure} = \text{Stay area} \times 9000.$$

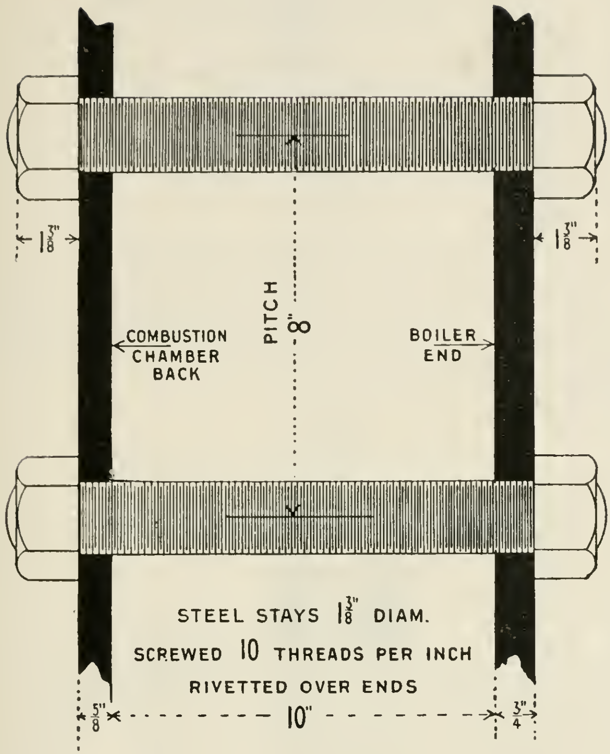
$$\text{Then,} \quad 17^2 \times 220 = \text{Stay area} \times 9000.$$

Therefore,

$$\frac{17^2 \times 220}{9000} = \overset{\text{Stay area}}{7{\cdot}064}, \text{ and, } \sqrt{\frac{\overset{\text{Stay area}}{7{\cdot}064}}{{\cdot}7854}} = 2{\cdot}99 \text{ in., say 3 in., Diameter of stay.}$$

Water Space Stays.

The combustion chamber stays usually range in diameter from about $1\frac{1}{4}$ to $1\frac{3}{4}$ inches, with a pitch of from $7\frac{1}{2}$ to $8\frac{1}{2}$ inches. These stays are screwed through the plates to be supported, and are secured by means of two nuts in all, one outside the boiler and the other inside the combustion chamber, the stay ends being riveted over the nuts. The plate is usually caulked round the stay to keep the joint tight. Occasionally the outside end of the stay is riveted over only, and the plate caulked round the stays (see sketch). These stays are, under ordinary conditions, subject to a tensile stress, but this is usually augmented by a bending stress produced by the floating tendency of the combustion chamber, which, it will be observed, is a hollow chamber immersed in water. This lifting tendency strains the stays often to the point of fracture, and in consequence permitting corrosion to take place at a more rapid rate.



No. 28.—Combustion Chamber Stays.

These stays are of steel, and are allowed a working tensile stress of 9000 lbs. per square inch. The plates are tapped and the stays screwed in. Occasionally the outer end is merely riveted over, the nut being omitted.

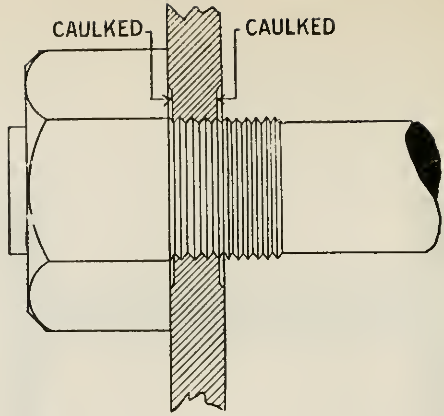
To Find Pitch of Stays (assume pressure as 200 lbs.).

RULE—

$$\text{Pitch}^2 \times \text{Pressure} = \text{Stay area} \times 9000.$$

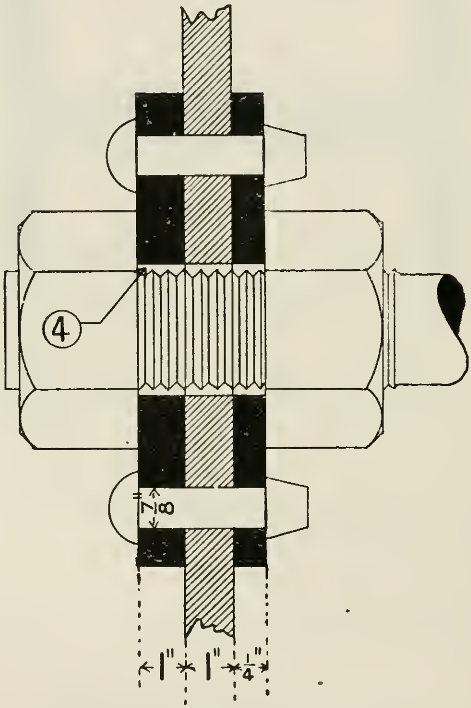
Then, $$\frac{\text{Stay area} \times 9000}{\text{Pressure}} = P^2 = \frac{1 \cdot 375^2 \times \cdot 7854 \times 9000}{200} = 66 \cdot 7.$$

Therefore, $\sqrt{66} = 8 \cdot 1$ in., say 8 in., pitch.



No. 29.—Caulking of Stays.

When the stays are screwed into the plates the latter are caulked round the stays, to ensure tightness, a touch of mastic cement or red lead putty being placed on the face of the nut before screwing up.



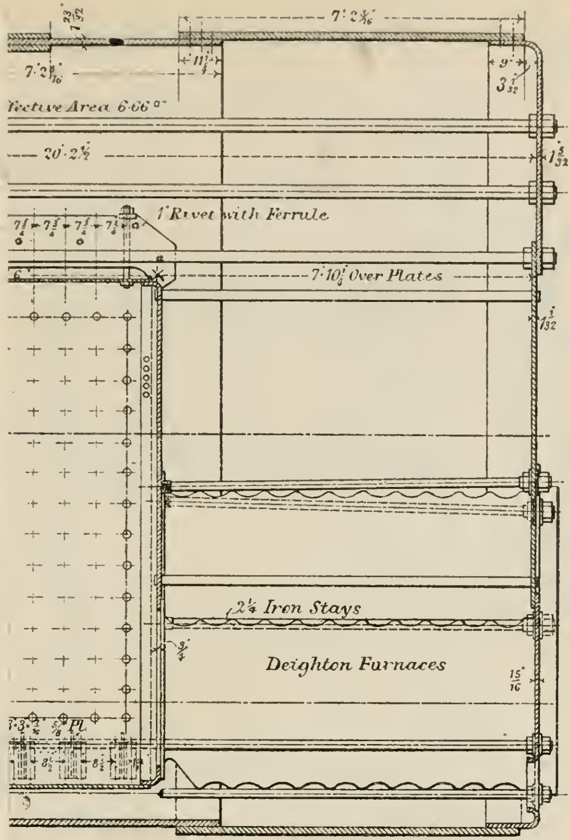
No. 30.—Steam Space Stay.

(With Double Nuts and Riveted Washers.)

The space 4 (about $\frac{1}{32}$ inch) is filled up with lead putty or mastic cement

In this type of stay the outside washers, which are riveted to the end plates, require to be of the same thickness as the plates, and of a diameter equal to two-thirds of the stay pitch.

The Constant allowed in this case is 168 for iron and 210 for steel.



n.
n.
½ in.

ıs	-	Double riveted.
-	-	Double riveted.
ams	-	Treble riveted.
-	-	D.B. strap with five rivets in a pitch.
mbers	-	Single riveted.

ers (front to back) = 4 ft. 6 in.
chambers = 10 ft. 6 in.

= $1\frac{1}{8}$ in. each.
ix stay bolts = each $1\frac{1}{2}$ in. diameter.

ıspension stays of any kind are fitted in this D.-E.
bered that in many boilers of this type such stays
also that the four separate combustion chambers are
me supported, by means of angle-plate stools riveted
chambers.

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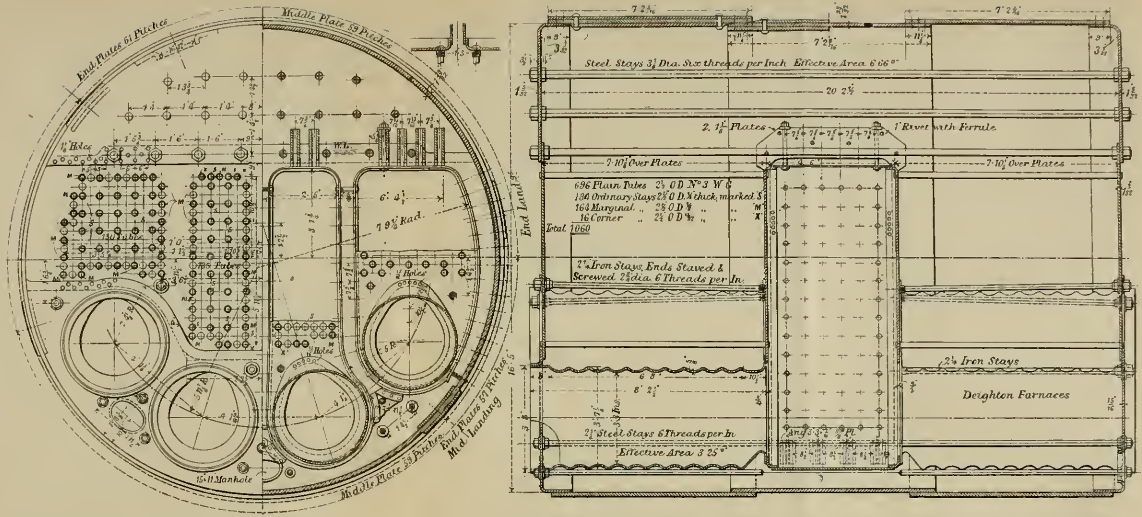

No. 30 A.—Double-Ended Boiler.

Data— Pressure = 215 lbs. (gauge).
Diameter = 16 ft. 6 in.
Length = 20 ft. $2\frac{1}{2}$ in.
Rates of heating surface to grate surface = 45 : 1.
Tensile strength of steel = From 30 to 32 tons per square inch.
Pitch of rivets = 10 in.
Strength of plate = 82·8 per cent. (joint strength).
Strength of rivets = 90·5 per cent.

Factor of Safety.—Given the above data the Factor of Safety can be calculated thus:—

$$\text{Tons} \times 2240 \times \text{T in.} \times 2 \times \text{Joint} = \text{Diameter} \times \text{Pressure} \times \text{Factor}.$$

Then, $$30 \times 2240 \times 1\tfrac{23}{32} \text{ in.} \times 2 \times \cdot 828 = 198 \text{ in.} \times 215 \times \text{Factor}.$$

So that, $$\text{Factor} = \frac{30 \times 2240 \times 1\frac{23}{32} \times 2 \times \cdot 828}{198 \text{ in.} \times 215} = 4\cdot 5 \text{ (nearly)}.$$

Note. 30 tons is taken, being the ***minimum*** tensile strength of the plates.
16 ft. 6 in. = 198 in.
82·8 per cent. = ·828.

Shell plate thickness = $1\frac{23}{32}$ in.
Butt strap thickness = $1\frac{7}{16}$ in.
Diameter of rivet holes = $1\frac{13}{16}$ in.

Riveting—

End circumferential shell seams	Double riveted.
End plate seams	Double riveted.
Centre circumferential shell seams	Treble riveted.
Longitudinal shell seams	D.B. strap with five rivets in a pitch.
Furnaces and combustion chambers	Single riveted.

Depth of combustion chambers (front to back) = 4 ft. 6 in.
Height of centre combustion chambers = 10 ft. 6 in.
Depth of girders = 13 in.
Thickness of girder plates = $1\frac{1}{8}$ in. each.
Each girder is fitted with six stay bolts = each $1\frac{1}{2}$ in. diameter.

It will be noticed that no suspension stays of any kind are fitted in this D.-E. boiler, but it should be remembered that in many boilers of this type such stays are fitted (see p. 99). Observe also that the four separate combustion chambers are held in place, and at the same time supported, by means of angle-plate stools riveted on to the adjoining combustion chambers.

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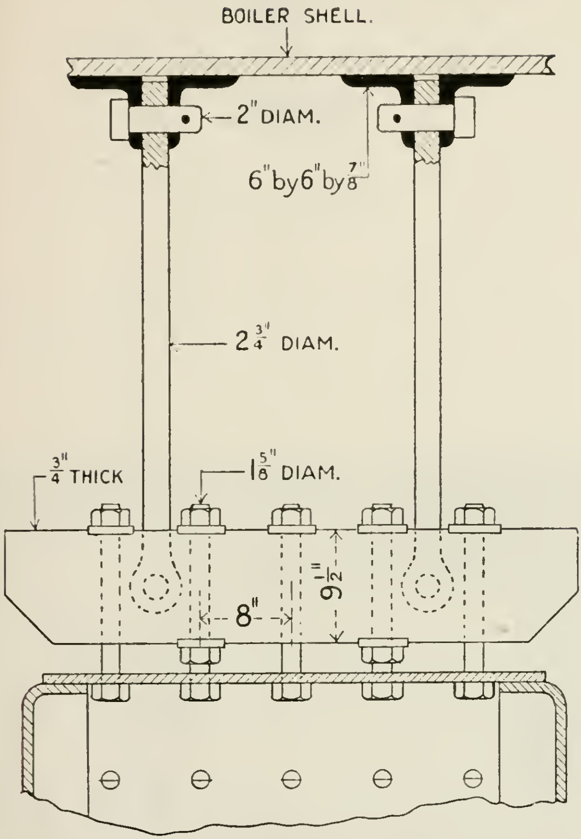
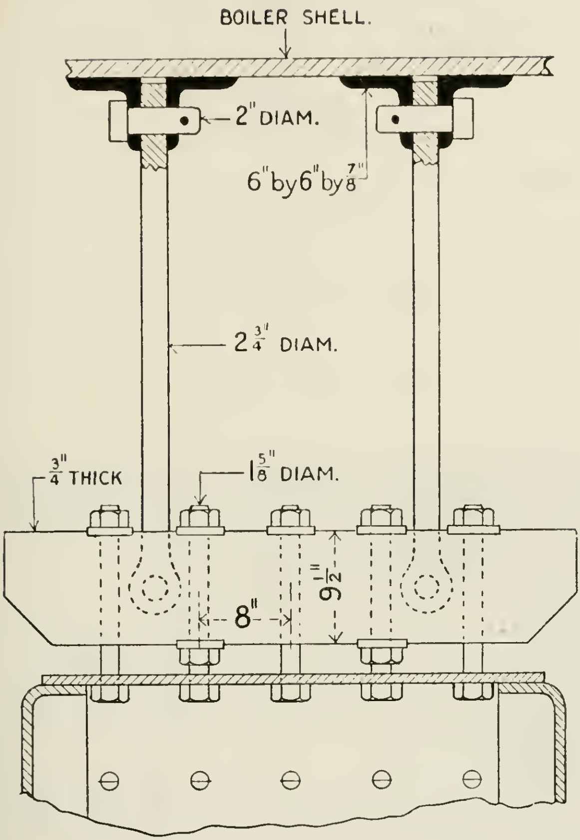


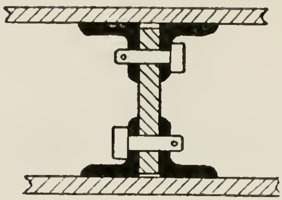
No. 31.—Method of supporting Combustion Chambers in Double Ended Boilers.

In modern high pressure double-ended boilers, the combustion chambers are often supported as shown above. Observe that the girders do not rest or bear on the combustion chamber plates as ordinarily, but are quite clear, or "floating," the necessary support being supplied by the two large suspension stays, which are secured by pins to double angle irons riveted to the shell. The pins are of turned steel. The combustion chamber bottoms are usually held rigidly in place by angle or plate stays to the boiler shell.



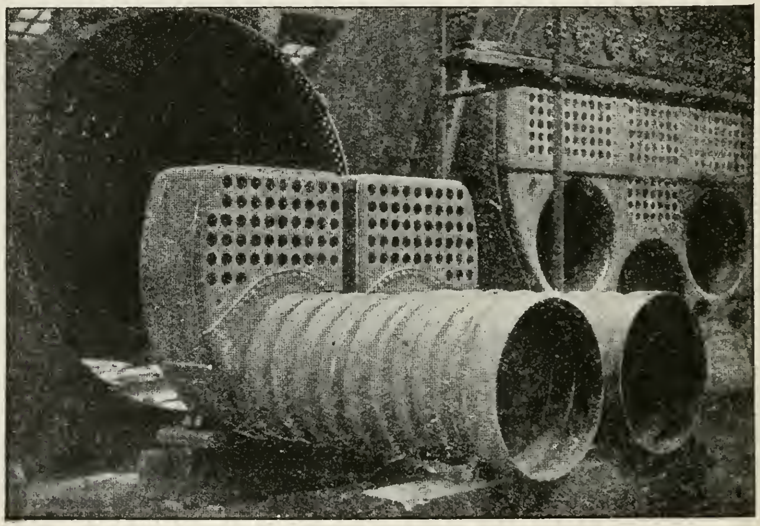
No. 31.—Method of supporting Combustion Chambers in Double Ended Boilers.

In modern high pressure double-ended boilers, the combustion chambers are often supported as shown above. Observe that the girders do not rest or bear on the combustion chamber plates as ordinarily, but are quite clear, or "floating," the necessary support being supplied by the two large suspension stays, which are secured by pins to double angle irons riveted to the shell. The pins are of turned steel. The combustion chamber bottoms are usually held rigidly in place by angle or plate stays to the boiler shell.



No. 32.—Method of Connecting Wing and Centre Combustion Chamber Plates.

In modern high pressure boilers the wing and centre combustion chambers are stayed together as shown in the sketch. A plate form of stay is employed, and is connected by means of pins through the plate and through angle irons, riveted to the combustion chamber plates. This type of stay is fitted at the bottom of the combustion chambers, forming a segment round the lower parts.

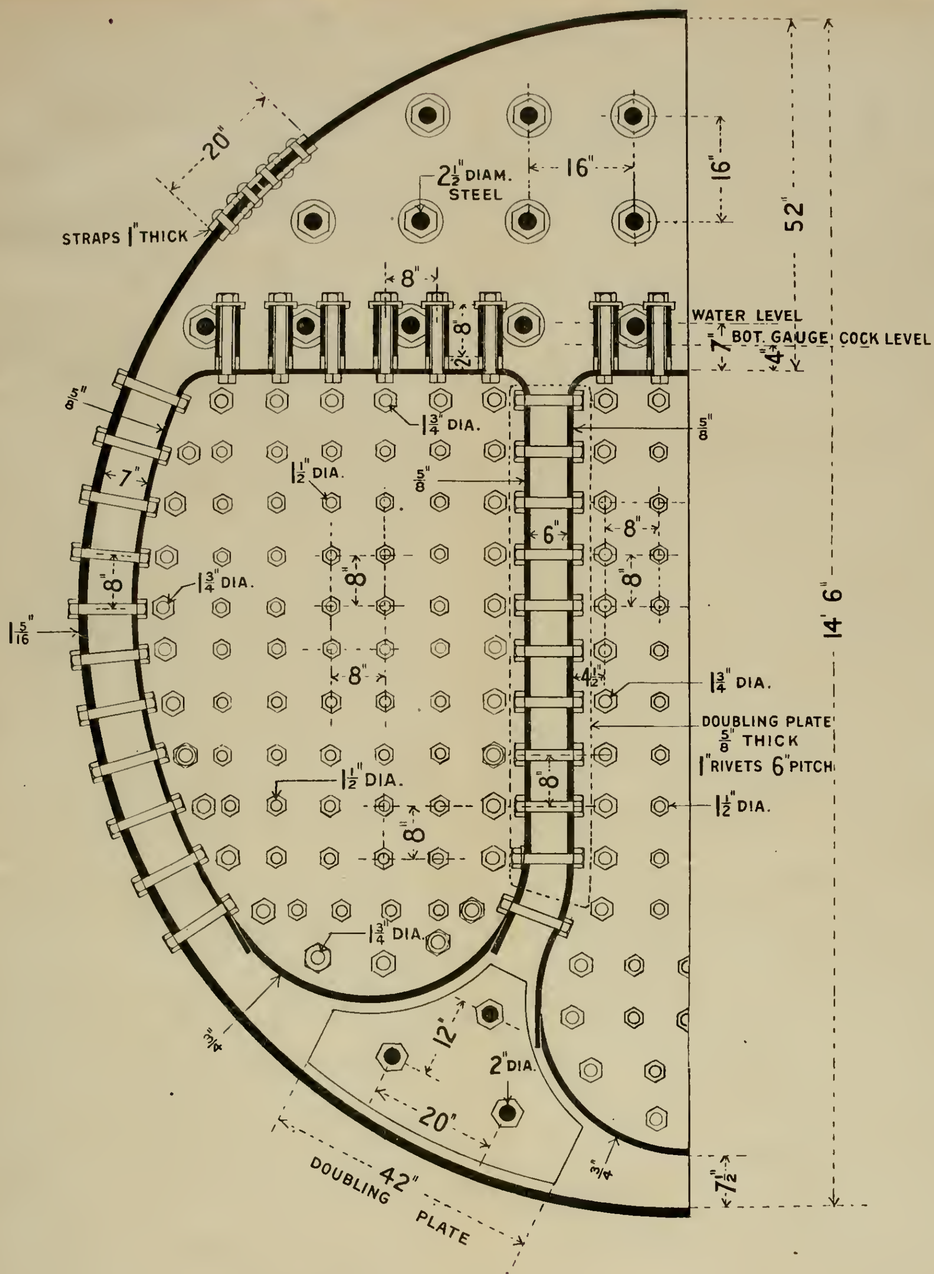


No. 34.—View of Wing Combustion Chambers and Furnaces.

Note the method of connecting the Furnace Flange to the Combustion Chamber, also the absence of a Flange at the Furnace Mouth.

Combustion Chambers.

The bottoms of combustion chambers are often stiffened by means of angle or **T** steel segments, which are riveted on, and extend from the bottom for some distance up the sides. Certain makers also fit plate stays, as described above, between these angle segments of adjacent combustion chambers to keep them in position.



No. 33.—Sectional View of Boiler showing Combustion Chamber Staying.

The student should carefully examine the above sketch and note the pitch and diameter of stays, the shape and size of the girders, the thickness of the plates, the distance between the combustion chambers and shell, the water level, bottom gauge cock level, the doubling plate between the tube nests, the distance between the combustion chamber top and boiler shell at top, &c. &c., as any of these dimensions may be asked for at the examination. It is also good practice to copy out parts of the above sketch and fill in the dimensions.

To find Pitch of Stays (Plate $\frac{5}{8}$ inch thick, pressure 200 lbs.).

RULE—

$$C \times (T+1)^2 = (S-6) \times \text{Working Pressure.}$$

Where C = Constant = 100 for Combustion chamber backs and sides.

" T = Plate thickness in sixteenths.

" S = Surface supported by one stay (pitch squared).

Then, $100 \times (10+1)^2 = (S-6) \times 200$.

Therefore, $S = \frac{100 \times (10+1)^2}{200} + 6 = 66{\cdot}5$, and, $\sqrt{66{\cdot}5} = 8{\cdot}1$ in., say 8 in., Pitch.

Notice that the Surface = Square of pitch, so that to convert Surface into Pitch, the square root requires to be extracted.

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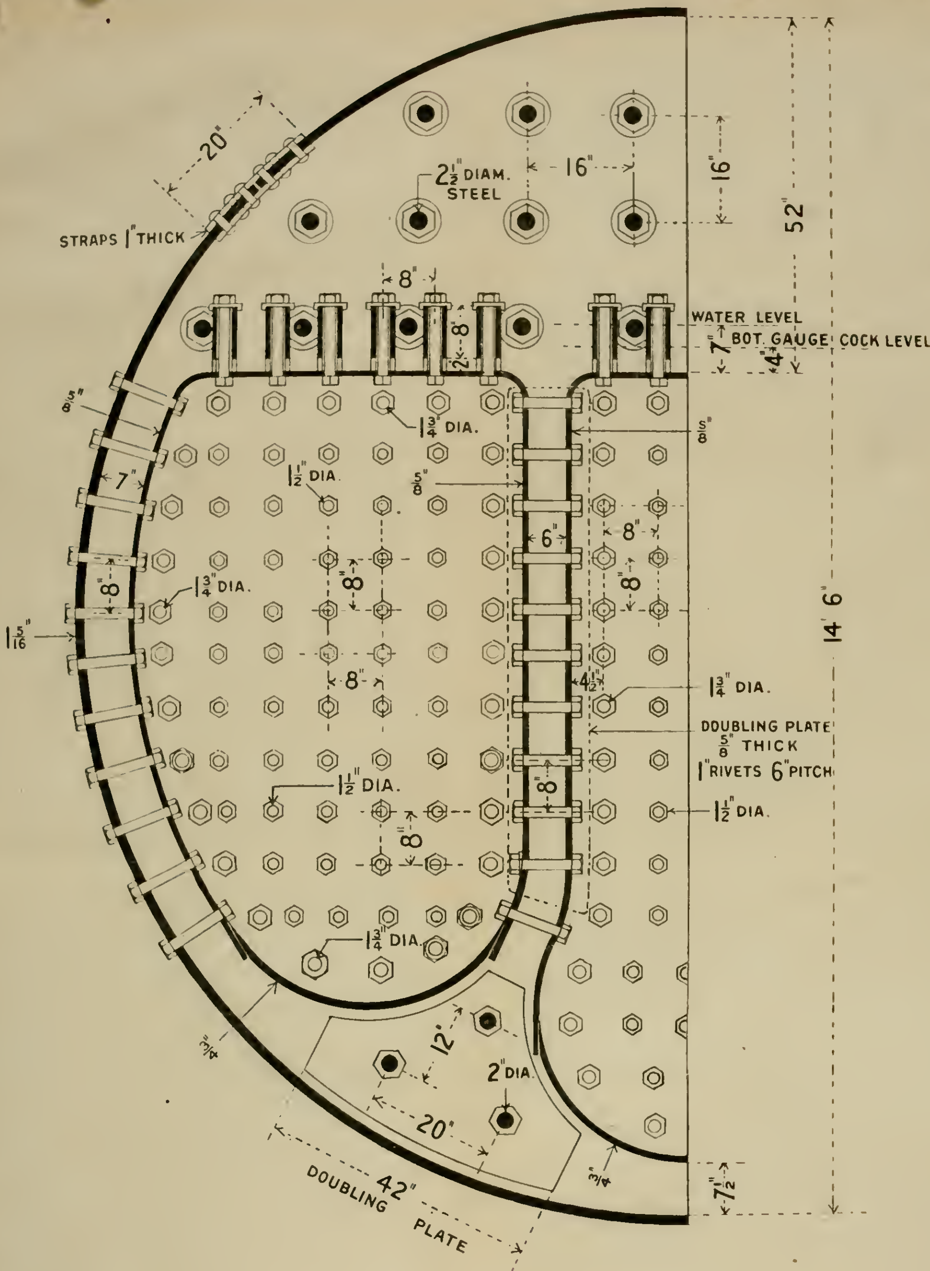

No. 33.—Sectional View of Boiler showing Combustion Chamber Staying.

The student should carefully examine the above sketch and note the pitch and diameter of stays, the shape and size of the girders, the thickness of the plates, the distance between the combustion chambers and shell, the water level, bottom gauge cock level, the doubling plate between the tube nests, the distance between the combustion chamber top and boiler shell at top, &c. &c., as any of these dimensions may be asked for at the examination. It is also good practice to copy out parts of the above sketch and fill in the dimensions.

To find Pitch of Stays (Plate $\frac{5}{8}$ inch thick, pressure 200 lbs.).

RULE—

$$C \times (T+1)^2 = (S-6) \times \text{Working Pressure.}$$

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„ T = Plate thickness in sixteenths.

„ S = Surface supported by one stay (pitch squared).

Then, $100 \times (10+1)^2 = (S-6) \times 200.$

Therefore, $S = \frac{100 \times (10+1)^2}{200} + 6 = 66{\cdot}5$, and, $\sqrt{66{\cdot}5} = 8{\cdot}1$ in., say 8 in., Pitch.

Notice that the Surface = Square of pitch, so that to convert Surface into Pitch, the square root requires to be extracted.

Combustion Chamber Stays.

It will generally be found that the marginal stays of combustion chambers are first affected by wear, and particularly the top row. Often the stays in this row develop minute cracks which may, however, on testing, be found to extend right into the body of the stay metal. The floating tendency of the combustion chamber (which, it should be observed, is simply a hollow vessel immersed in water) throws severe stresses on the stays, and generally, through time, results in straining the metal of the stays as described, due to the bending stresses set up. The marginal stays having to support more surface than the inner stays are of larger section (often $1\frac{1}{2}$ to $1\frac{3}{4}$ inches diameter).

Combustion Chamber Bottoms.—The following rule brings out the safe working pressure for combustion chamber bottom plates, which, it should be observed, form semicircular surfaces of plain section.

RULE—

$$\frac{9900 \times T}{3 \times D} \times \left(5 - \frac{L + 12}{40 \times T}\right) = \text{Safe Pressure.}$$

Where T = Plate Thickness.
„ L = Length between furnace back end and back of combustion chamber.
„ D = Radius of combustion chamber bottom × 2.

EXAMPLE.—Find the safe working pressure for the bottom plate of a combustion chamber $\frac{3}{4}$ inch thick: length of chamber from front to back 32 inches, and outside radius of bottom 23 inches.

Then, $$\text{Safe Pressure} = \left(\frac{9900 \times \cdot 75}{3 \times (23 \times 2)}\right) \times \left(5 - \frac{32 + 12}{60 \times \cdot 75}\right) =$$

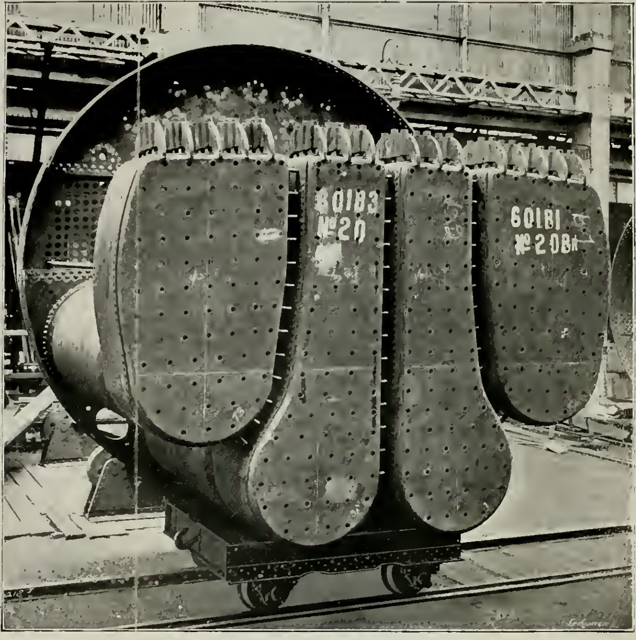
$$\frac{7425}{138} \times (5 - \cdot 977) = 53\cdot 8 \times 4 = 215\cdot 2 \text{ lbs. per sq. in.}$$

Combustion Chamber Girders.

The combustion chamber top of single ended, and in some cases double ended, boilers is supported against collapse as shown in the sketch.

The girders are formed of two vertical steel plates each about $\frac{3}{4}$ or $\frac{7}{8}$ inch thick, and between these plates stays are passed down and screwed into the top plate; nuts and overlapping washers are fitted above. The plates forming the girders are held together by rivets fitting through thimbles.

The girder plates are often made overlapping on to the top plate at the sides, but some firms have the plates cut clear altogether (see No. 39). Under working conditions the girder plates are subjected to a compressive and bending stress, the compression being at the ends where the load is taken up by the back tube plate and combustion chamber plate, and the tensile stress acts at and near the centre of the girder.



No. 35. Boiler under Construction.

View showing combustion chamber boxes, girders, stays, and front end plate. The close staying of the combustion chamber back plates should be carefully noted.

NOTE.—The girders shown are all of the single plate type, bossed out where the stays pass through. The bevelled joints of the combustion chamber bottom plate and side plates should also be noted.

Combustion Chamber Stays.

It will generally be found that the marginal stays of combustion chambers are first affected by wear, and particularly the top row. Often the stays in this row develop minute cracks which may, however, on testing, be found to extend right into the body of the stay metal. The floating tendency of the combustion chamber (which, it should be observed, is simply a hollow vessel immersed in water) throws severe stresses on the stays, and generally, through time, results in straining the metal of the stays as described, due to the bending stresses set up. The marginal stays having to support more surface than the inner stays are of larger section (often $1\frac{1}{2}$ to $1\frac{3}{4}$ inches diameter).

Combustion Chamber Bottoms.—The following rule brings out the safe working pressure for combustion chamber bottom plates, which, it should be observed, form semicircular surfaces of plain section.

RULE—

$$\frac{9900 \times T}{3 \times D} \times \left(5 - \frac{L + 12}{40 \times T}\right) = \text{Safe Pressure.}$$

Where T = Plate Thickness.
,, L = Length between furnace back end and back of combustion chamber.
,, D = Radius of combustion chamber bottom × 2.

EXAMPLE.—Find the safe working pressure for the bottom plate of a combustion chamber $\frac{3}{4}$ inch thick: length of chamber from front to back 32 inches, and outside radius of bottom 23 inches.

Then, Safe Pressure $= \left(\frac{9900 \times \cdot75}{3 \times (23 \times 2)}\right) \times \left(5 - \frac{32 + 12}{60 \times \cdot75}\right) =$

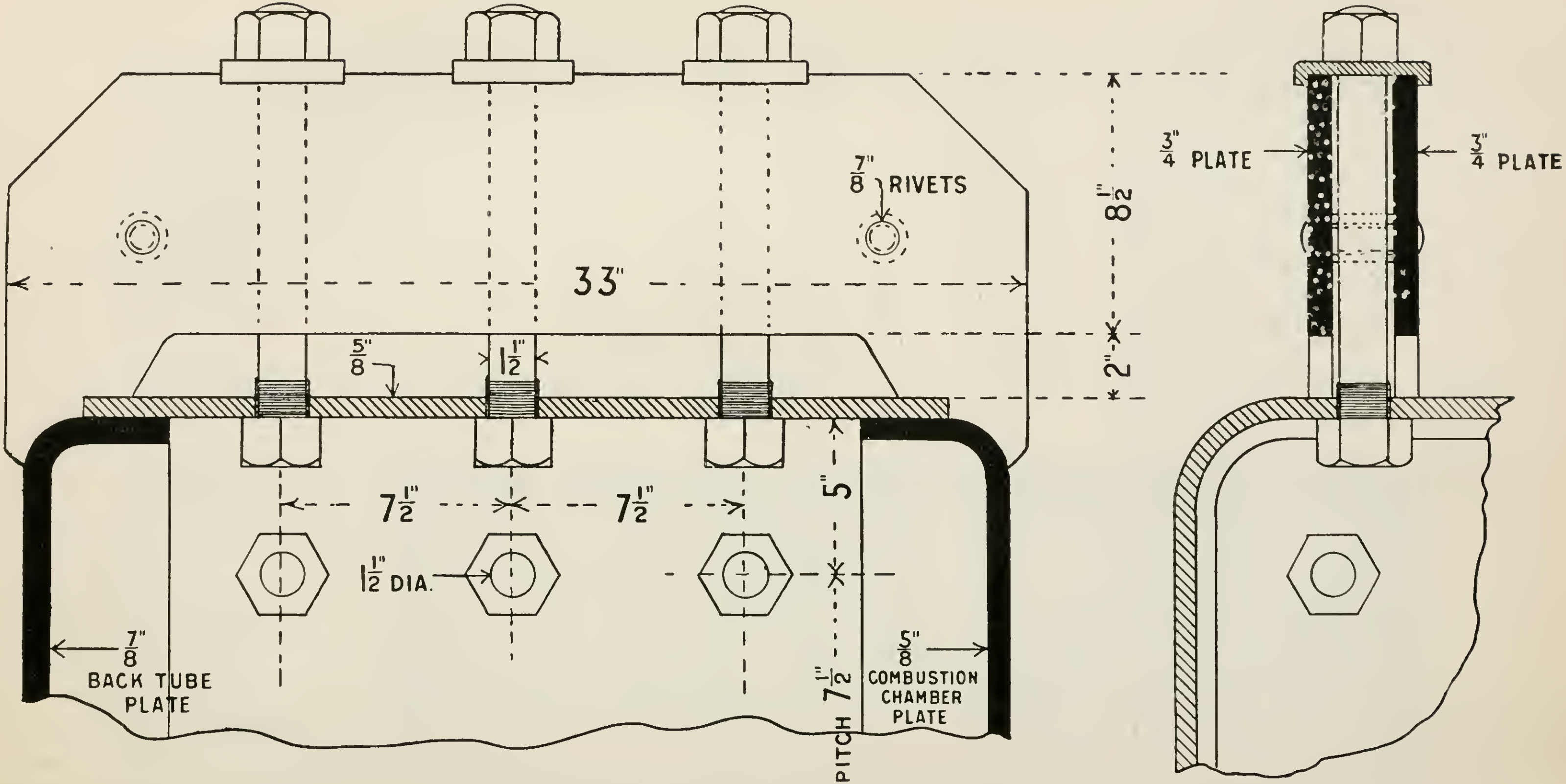
$$\frac{7425}{138} \times (5 - \cdot977) = 53\cdot8 \times 4 = 215\cdot2 \text{ lbs. per sq. in.}$$

Combustion Chamber Girders.

The combustion chamber top of single ended, and in some cases double ended, boilers is supported against collapse as shown in the sketch.

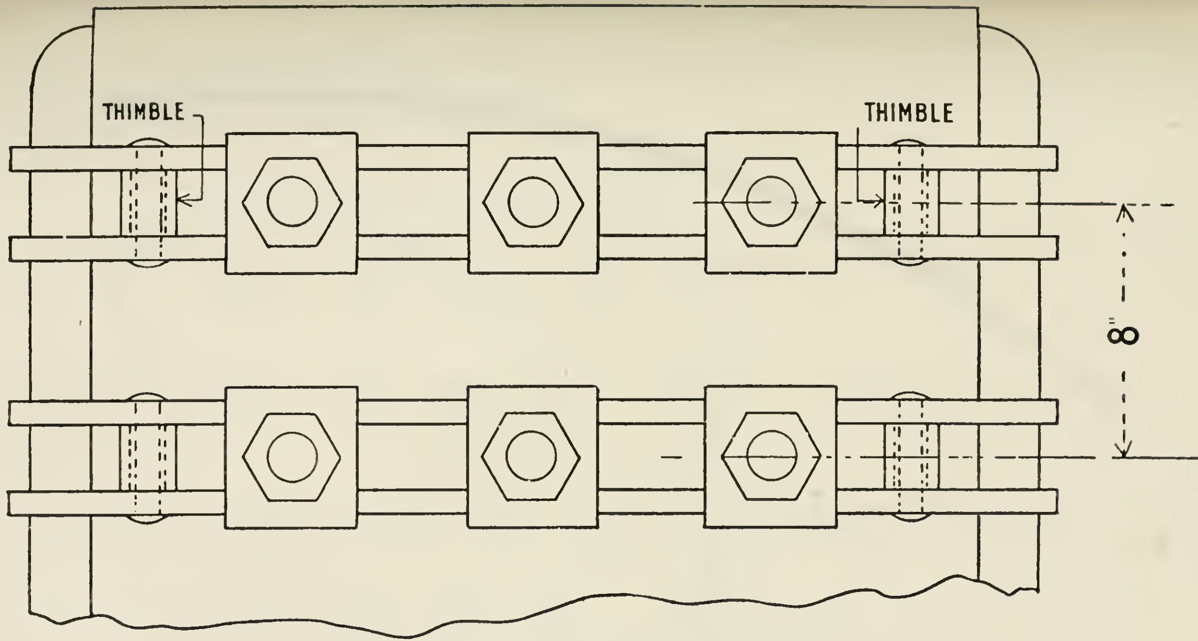
The girders are formed of two vertical steel plates each about $\frac{3}{4}$ or $\frac{7}{8}$ inch thick, and between these plates stays are passed down and screwed into the top plate; nuts and overlapping washers are fitted above. The plates forming the girders are held together by rivets fitting through thimbles.

The girder plates are often made overlapping on to the top plate at the sides, but some firms have the plates cut clear altogether (see No. 39). Under working conditions the girder plates are subjected to a compressive and bending stress, the compression being at the ends where the load is taken up by the back tube plate and combustion chamber plate, and the tensile stress acts at and near the centre of the girder.



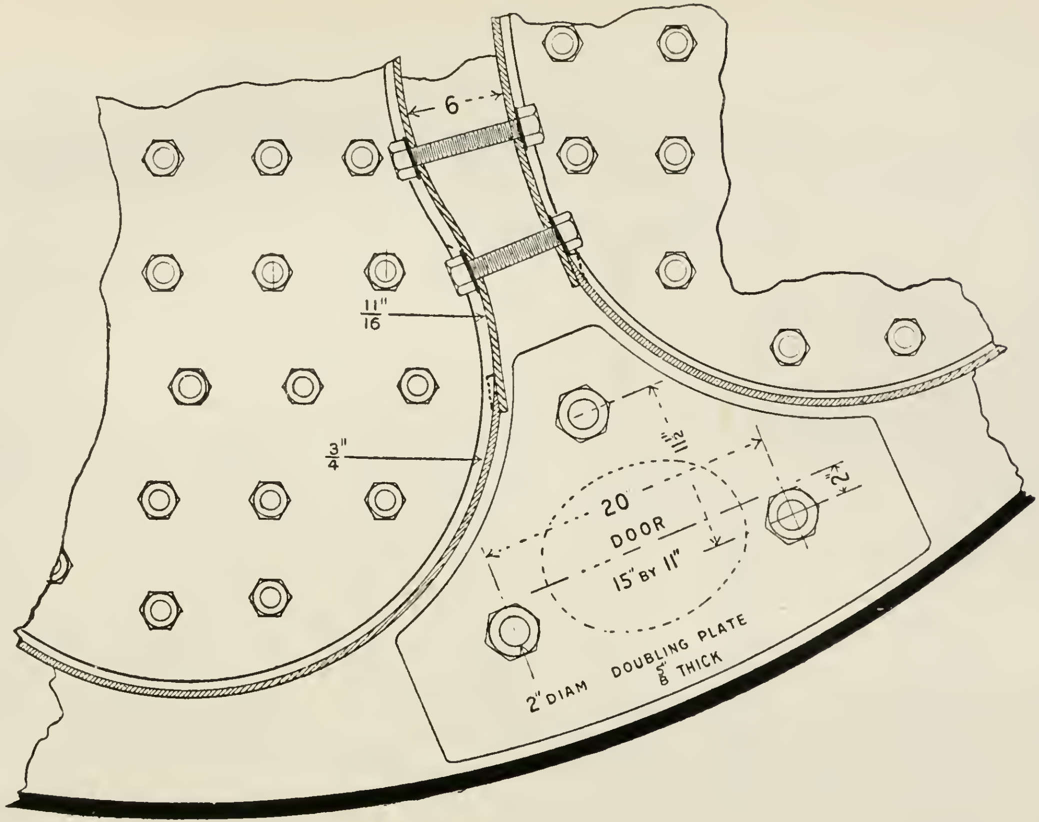
No. 36.—Combustion Chamber Girders.

(With Dimensions.)



No. 37.—Plan of Combustion Chamber Top.

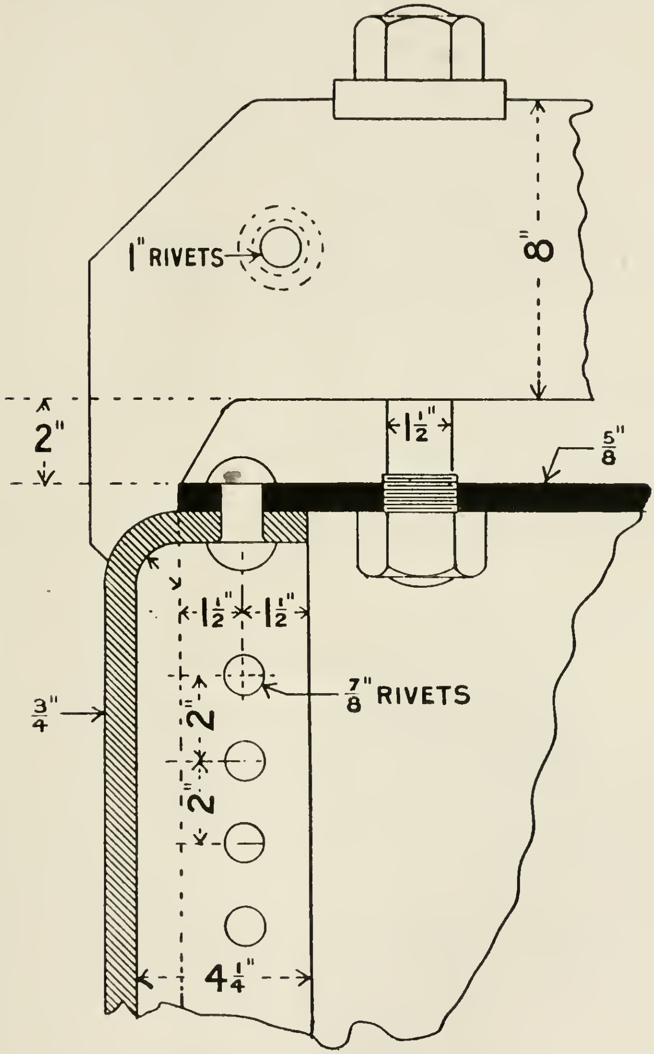
This view shows the top of a combustion chamber as seen looking from above. The girder plates, overlapping washers, thimbles, and rivets are all in view, and a good idea of the general arrangement should be obtained from the sketch.



No. 38.—Staying of Centre and Wing Combustion Chambers.

(Also Doubling Plate of End Manhole.)

The arrangement of the stays to cover effectively the areas to be supported, and the plate flanges, &c., should be carefully studied. Also the manhole, stays, and doubling plate with their respective dimensions.



No. 39.—Combustion Chamber Top and Girder.

Some firms arrange the girders cut clear of the top plate as shown. Attention should be given to the dimensions of the plates, rivets, and stays.

The strength of girders varies (as in the case of a beam) directly as the Depth^2 and the Thickness, and inversely as the Length.

Or, Strength varies as, $\frac{D^2 \times T}{L}$.

RULE—

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

Where, C = 990 for three stays, and 1100 for four stays.
,, d = Depth (inches).
,, T = Combined thickness of girder plates.
,, W = Width of combustion chamber (inches).
,, P = Pitch of stays (inches).
,, D = Distance between girder centres (inches).
,, L = Length of girders (feet).

Application.—To apply the rule to the case above (Sketch 36).

Then, $\text{Working Pressure} = \frac{990 \times 8{\cdot}5^2 \text{ in.} \times 1{\cdot}5 \text{ in.}}{(31 \text{ in.} - 7{\cdot}5 \text{ in.}) \times 8 \text{ in.} \times 2{\cdot}75} = 207$ lbs. per sq. in.

NOTE.—The width of combustion chamber is about 31 inches, and assume the distance between the girders as 8 inches. Again notice that 33 inches = 2·75 feet.

Depth of Girders (Sketch 39).

C = 990 for three stays through girders.
d = Depth of girder.
T = Thickness of girder plates (combined) = $1\frac{1}{2}$ inches.
W = Combustion chamber width = 31 inches
P = Pitch of stays = $8\frac{1}{2}$ inches.
D = Distance between girders = 8 inches.
L = Length of girders in feet = 2·75 feet.
Pressure = 190 lbs. per square inch.

RULE—

$$C \times d \times T = (W - P) \times D \times L \times \text{Pressure}.$$

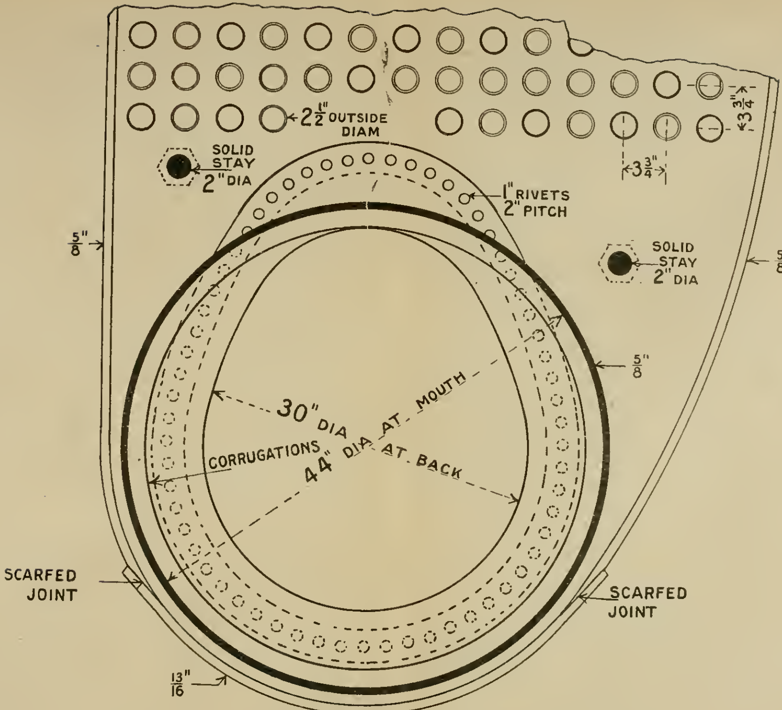
Therefore, $$d^2 = \frac{(W - P) \times D \times L \times \text{Pressure}}{C \times T} =$$

$$\frac{(31 - 8{\cdot}5) \times 8 \times 2{\cdot}75 \times 190}{990 \times 1{\cdot}5} = 63{\cdot}3.$$

And, $\sqrt{63{\cdot}3} = 7{\cdot}9$ in., say 8 in., depth of girders.

Tubes.

The smoke tubes are generally made 3 or $3\frac{1}{2}$ inches outside diameter with natural draught and $2\frac{1}{2}$ inches outside diameter with forced draught, the thickness ranging from $\frac{1}{8}$ or $\frac{3}{16}$ inch in the case of ordinary tubes, to $\frac{5}{16}$ or $\frac{1}{2}$ inch in the case of stay tubes. The



No. 40.—Front View of Goose-Neck Type (Gourley-Stephen) Furnace.
(Front Tube Plate removed.)

This sketch shows clearly the difference in the section of the furnace at back and front, the back being constricted to an oval section (30 inches diameter). The flange at the back is also much higher in position than the circular portion of the furnace and is riveted all round to the back tube plate.

Also observe the scarfed joint of the combustion chamber plating at the sides.

The side wrapper plates are $\frac{5}{8}$ inch thick, but the bottom is heavier, being $\frac{13}{16}$ inch thick.

Thickness of Combustion Chamber Bottom Plate.

RULE—

$$\left(\frac{9900 \times T}{3 \times D}\right) \times \left(5 - \frac{L + 12}{60 \times T}\right) = \text{Working Pressure}$$

Where T = Thickness = $\frac{13}{16}$ = ·8125.
„ D = Diameter outside = 46 inches.
„ L = Length (front to back) = 33 inches.

Then, $$\text{Safe Pressure} = \left(\frac{9900 \times \cdot 8125}{3 \times 46}\right) \times \left(5 - \frac{33 + 12}{60 \times \cdot 8125}\right) = 232\cdot 8 \text{ lbs.}$$

This proves that the bottom plate is of ample strength, as less thickness would be sufficient, but the extra thickness given to the plate allows a margin for corrosion.

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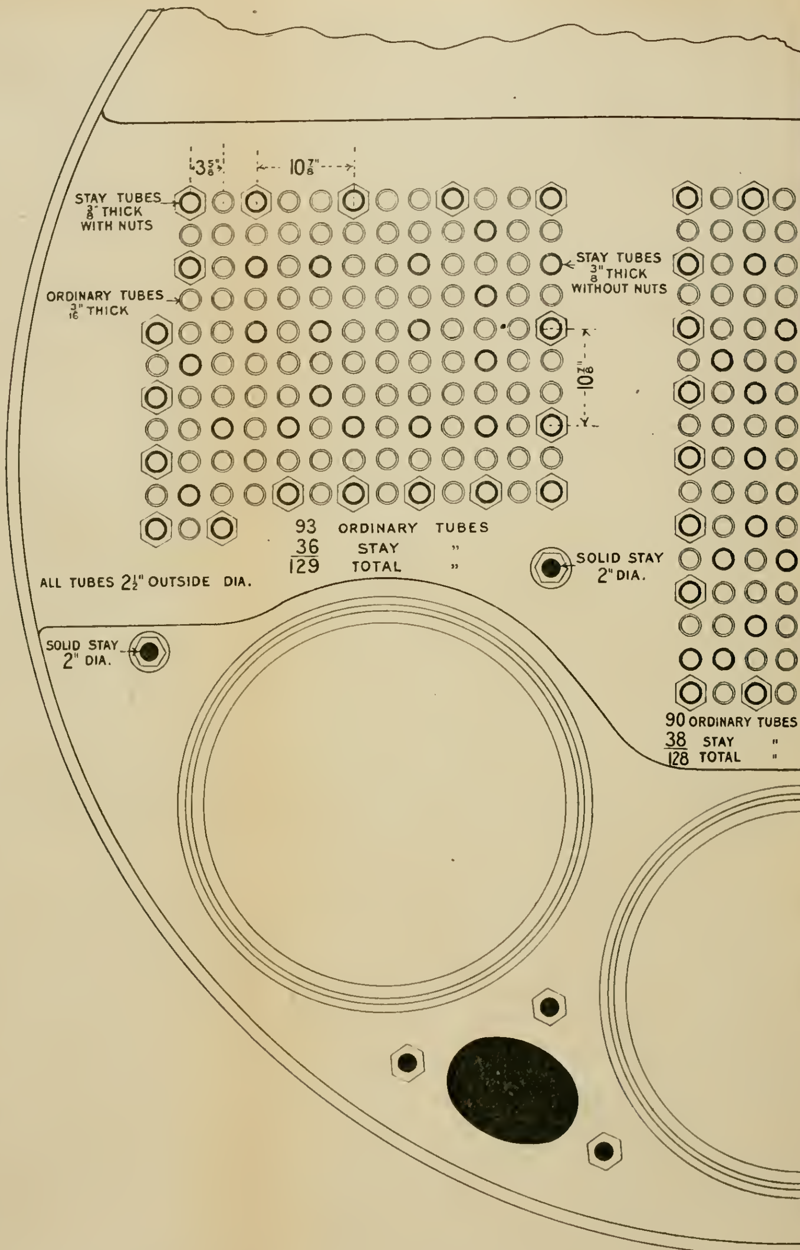
diameter with natural draught and $2\frac{1}{2}$ inches outside diameter with forced draught, the thickness ranging from $\frac{1}{8}$ or $\frac{3}{16}$ inch in the case of ordinary tubes, to $\frac{5}{16}$ or $\frac{1}{2}$ inch in the case of stay tubes. The



No. 41.—Tube Boxes, showing dimensions and number of Stay Tubes and Ordinary Tubes.

The pitch of the tubes is $3\frac{5}{8}$ inches, and the maximum pitch of the stay tubes $10\frac{7}{8}$ inches.

Notice that the marginal or bounding stay tubes are the heaviest fitted, and are secured by flat nuts at front tube plate. Also note the tube plate solid stays 2 inches diameter. The clear space between the tube nests is 11 or 12 inches, as this is required to allow of access to the lower parts of the boiler. As stays cannot be fitted on this plate area, the end plates are strengthened by means of *doubling plates* riveted on, which take the place of solid stays (see No. 33).

Thickness of Tube Plates—

RULE—

$$(D - d) \times T \times 28000 = W \times D \times \text{Pressure}.$$

Where D = Pitch of Tubes. Where T = Thickness of Tube Plates.
" d = Inside diameter of tubes. " W = Width of combustion chamber.

EXAMPLE.—Determine the required thickness of tube plates for a pressure of 200 lbs., the pitch of tubes being $3\frac{5}{8}$ inches, the inside diameter of tubes $2\frac{1}{8}$ inches, and the width of combustion chamber 33 inches.

Then, $$T = \frac{W \times D \times \text{Pressure}}{(D - d) \times 28000} = \frac{33 \times 3{\cdot}625 \times 200}{(3{\cdot}625 - 2{\cdot}125) \times 28000} = {\cdot}56 \text{ in., say } \frac{9}{16} \text{ in.}$$

NOTE.—The back tube plate is seldom made less than $\frac{3}{4}$ inch thick, notwithstanding the calculated thickness by rule.

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front end of all the tubes is slightly larger in diameter than the back end to allow of easy insertion or extraction. In the case of the tubes the ends are expanded out to the holes in the plate by an appliance known as a "tube expander" (see sketch), which operation is considered sufficient to ensure tightness. The stay tubes represent solid bar stays and are required to support the flat surfaces formed by both tube plates. The sectional area of the stay tube metal should therefore be equal to that of a solid stay. The required thickness can be found as follows:—

EXAMPLE.—Determine the required thickness of steel stay tubes $2\frac{1}{2}$ inches outside diameter to be equal in strength to a solid stay, the maximum pitch of the stay tubes being $10\frac{7}{8}$ inches and the pressure 180 lbs. per square inch.

Then, Diameter of solid stay $=\sqrt{\dfrac{10 \cdot 875^2 \times 180}{9000 \times \cdot 7854}} = 1 \cdot 74$, say $1\frac{3}{4}$ in., diameter.

The cross sectional area of metal of the stay tubes must also be equal to the solid stay area.

Therefore, $(2 \cdot 5^2 - d^2) = 1 \cdot 75^2 = (6 \cdot 25 - d^2) = 3 \cdot 0625.$

Therefore, $6 \cdot 25 - 3 \cdot 0625 = d^2 = 3 \cdot 1875.$

Therefore, Inner diameter of tube $= \sqrt{3 \cdot 1875} = 1 \cdot 78$ in., say $1\frac{3}{4}$, in.

And Tube thickness $= \dfrac{2 \cdot 5 - 1 \cdot 75}{2} = \cdot 375$ in. ($\frac{3}{8}$ in.).

The ends of the tubes are staved up to slightly increase the diameter and so allow for the cutting of the screw, and it should be noted that the external diameter at the front end exceeds that of the back end by about $\frac{1}{4}$ inch. The ordinary tubes are subject to a compressive stress and the stay tubes to an additional tensile stress.

In general practice about one-third or more of the total tubes are arranged as stay tubes, and these tubes are usually fixed by one of the following methods:—

1. Both plates tapped and tubes screwed in, then expanded into the plates.

2. Both plates tapped and tubes screwed in, expanded into the plates, and a nut $\frac{3}{4}$ inch thick fitted outside the front tube plate only.

The plates are caulked round the tubes and the back end of the tubes are also beaded over as shown in sketch.

It is now the common practice to fit nuts only on the marginal or bounding stay tubes, the others being merely screwed into the plates, expanded in the screws, caulked and beaded over. The sketch on page 108 illustrates the foregoing.

Often three thicknesses of stay tubes are employed in one boiler to meet the requirements of the differently supported areas of the tube plates, the heaviest type $\frac{1}{2}$ inch thick being used for the marginal stay tubes, and the others, varying in thickness from $\frac{3}{8}$ to $\frac{5}{16}$ inch, are fitted inside the marginal area of the plates.

front end of all the tubes is slightly larger in diameter than the back end to allow of easy insertion or extraction. In the case of the tubes the ends are expanded out to the holes in the plate by an appliance known as a "tube expander" (see sketch), which operation is considered sufficient to ensure tightness. The stay tubes represent solid bar stays and are required to support the flat surfaces formed by both tube plates. The sectional area of the stay tube metal should therefore be equal to that of a solid stay. The required thickness can be found as follows:—

EXAMPLE.—Determine the required thickness of steel stay tubes $2\frac{1}{2}$ inches outside diameter to be equal in strength to a solid stay, the maximum pitch of the stay tubes being $10\frac{7}{8}$ inches and the pressure 180 lbs. per square inch.

Then, Diameter of solid stay $= \sqrt{\dfrac{10{\cdot}875^2 \times 180}{9000 \times {\cdot}7854}} = 1{\cdot}74$, say $1\frac{3}{4}$ in., diameter.

The cross sectional area of metal of the stay tubes must also be equal to the solid stay area.

Therefore, $(2{\cdot}5^2 - d^2) = 1{\cdot}75^2 = (6{\cdot}25 - d^2) = 3{\cdot}0625.$

Therefore, $6{\cdot}25 - 3{\cdot}0625 = d^2 = 3{\cdot}1875.$

Therefore, Inner diameter of tube $= \sqrt{3{\cdot}1875} = 1{\cdot}78$ in., say $1\frac{3}{4}$, in.

And Tube thickness $= \dfrac{2{\cdot}5 - 1{\cdot}75}{2} = {\cdot}375$ in. ($\frac{3}{8}$ in.).

The ends of the tubes are staved up to slightly increase the diameter and so allow for the cutting of the screw, and it should be noted that the external diameter at the front end exceeds that of the back end by about $\frac{1}{4}$ inch. The ordinary tubes are subject to a compressive stress and the stay tubes to an additional tensile stress.

In general practice about one-third or more of the total tubes are arranged as stay tubes, and these tubes are usually fixed by one of the following methods:—

1. Both plates tapped and tubes screwed in, then expanded into the plates.

2. Both plates tapped and tubes screwed in, expanded into the plates, and a nut $\frac{3}{4}$ inch thick fitted outside the front tube plate only.

The plates are caulked round the tubes and the back end of the tubes are also beaded over as shown in sketch.

It is now the common practice to fit nuts only on the marginal or bounding stay tubes, the others being merely screwed into the plates, expanded in the screws, caulked and beaded over. The sketch on page 108 illustrates the foregoing.

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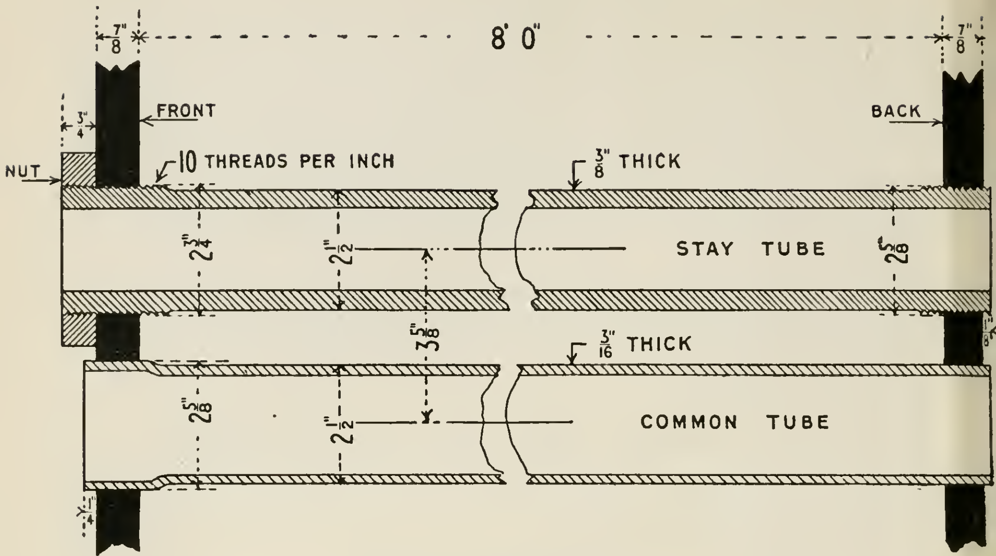


No. 42.—Boiler Tubes.

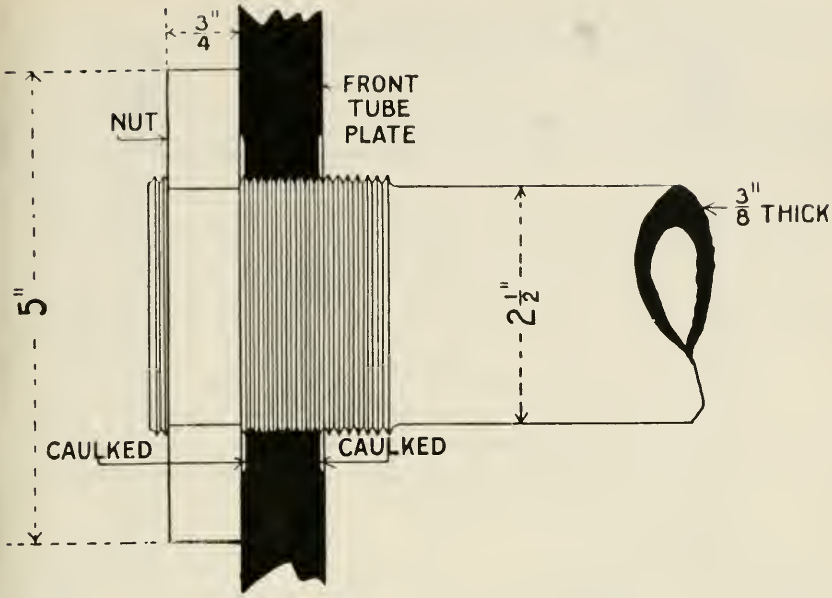
The stay tubes are $\frac{3}{8}$ inch thick, and are screwed into both plates and caulked in place; in addition, a flat nut is screwed over the front end. This type of stay tube is used round the boundary of a tube nest. Notice the difference in diameter at either end, the front being the larger. About one-third of the total tubes are fitted as stay tubes, the thickness ranging from $\frac{1}{4}$ inch for the inner stay tubes to $\frac{3}{8}$ inch for the outer or marginal stay tubes, which are nutted.

The ordinary tube shown is $\frac{3}{16}$ inch in thickness, and like the stay tube is swelled out at the front end. These tubes are expanded in place by an ordinary three-roller tube expander.

Stay Tubes.—The light pattern stay tubes (internal) are expanded into both tube plates, and beaded over; the heavy pattern of stay tubes (marginal) are caulked in position both inside and outside at the front, and inside at the back end, also beaded over, as is usual with all tubes.

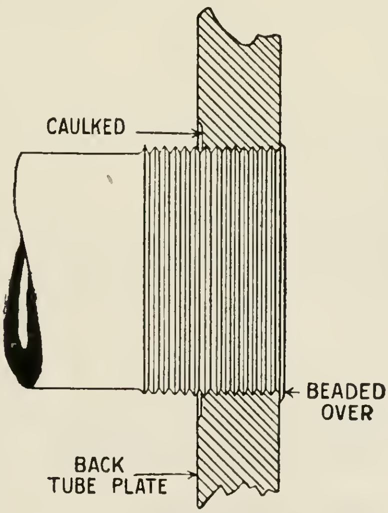
NOTE.—The ratio of tube heating surface to grate surface is as 25 is to 1 in most cases.

"Serve" Tubes.—The ribs inside this type of tube increases the effective area of the tube, and thus extracts more heat from the waste gases as they pass through. This results in increased evaporation and economy.

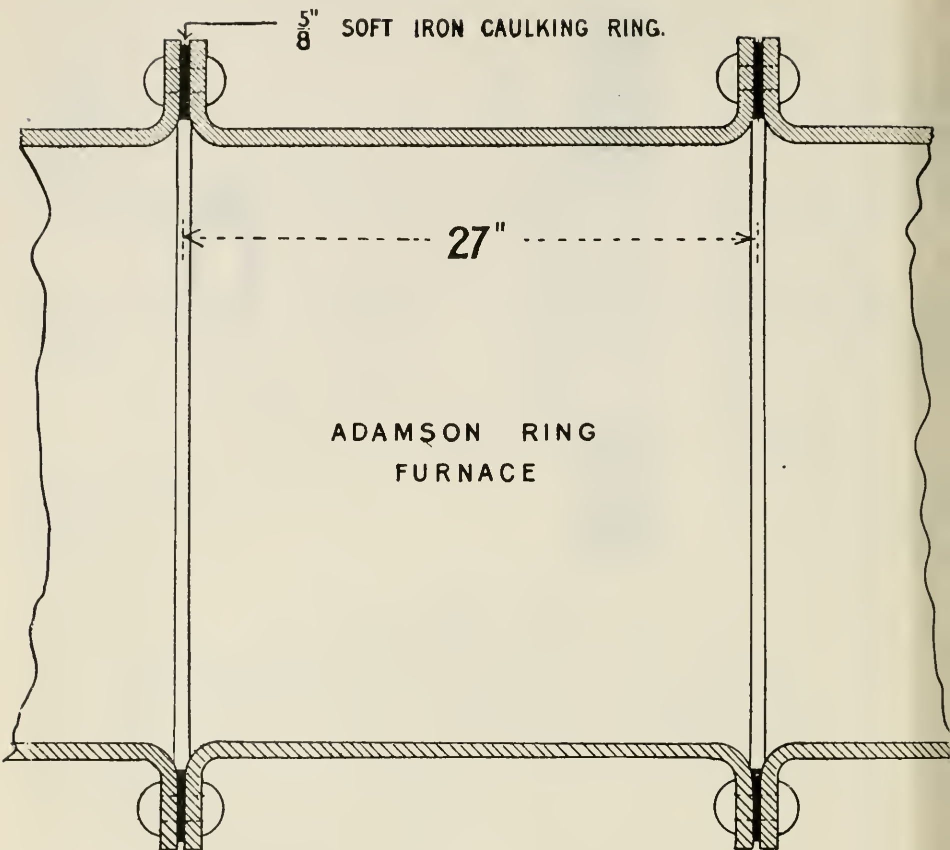


No. 43.—Heavy Pattern Stay Tube.
(With Flat Nut in Front Tube Plate.)

These tubes being too heavy for expanding, the plates are caulked round the tube instead. Tubes of this type are placed on the margins of a tube nest.



No. 44.—Back End of Stay Tube.
(Showing Screwing in, Caulking, and Beading over.)



No. 45.—Adamson Ring Furnace.

As with the Bowling Hoop furnace the different lengths are riveted together through the flanges and soft iron caulking rings. The flanges stiffen the furnace and allow for expansion, while tightness of joint is obtained by close caulking up of the soft metal rings.

Safe Pressure.—For a plain steel furnace the rule is as follows :—

$$\text{Working pressure} = \frac{99000 \times T^2}{(L+1) \times D}.$$

Where T = Thickness.
,, L = Length in feet.
,, D = Diameter in inches.

If, however, the furnace is fitted with stiffening rings as shown above, then L = *Length between the rings* subject to a limit pressure found as follows :—

$$\text{Limit pressure} = \frac{9900 \times T}{D}.$$

So that the actual working pressure is to be the smaller of the two results obtained.

EXAMPLE.—Determine the safe working pressure for the furnace shown in Sketch No. 45, which is $\frac{3}{4}$ inch thick and 42 inches diameter.

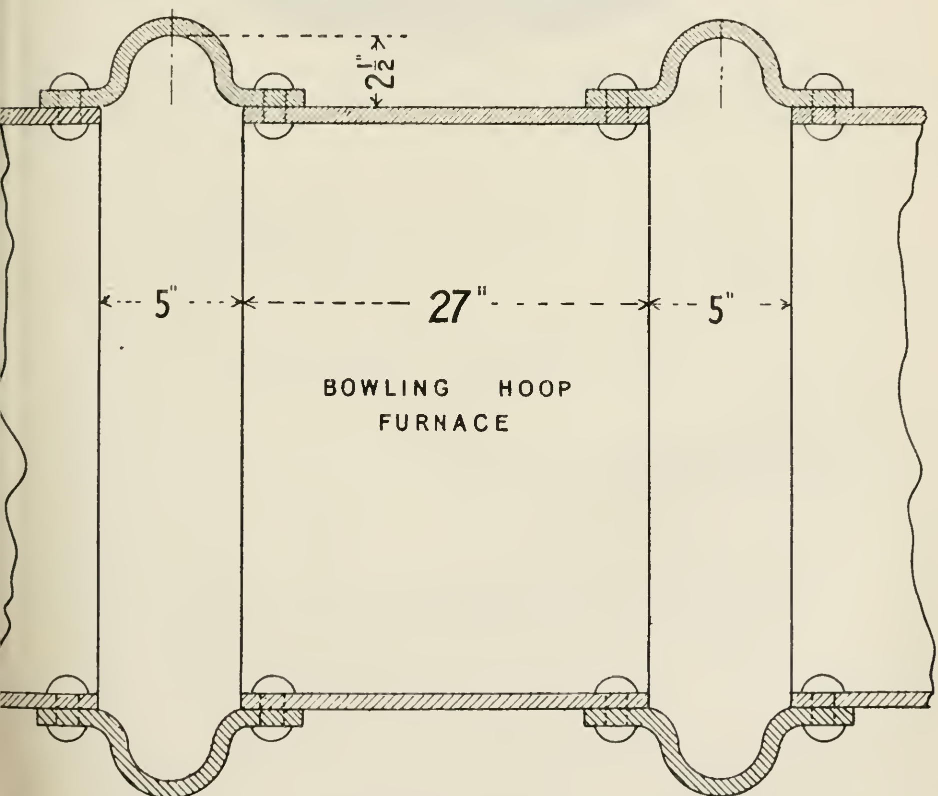
$$\begin{array}{c}\text{Working Pressure}\\ \text{(If less than limit pressure)}\end{array} = \frac{99000 \times \cdot 75^2}{(2 \cdot 25 + 1) \times 42 \text{ in.}} = 407 \text{ lbs.}$$

NOTE.—27 inches = 2·25 feet.

$$\text{Limit Pressure} = \frac{9900 \times \cdot 75}{42} = 176 \text{ lbs.}$$

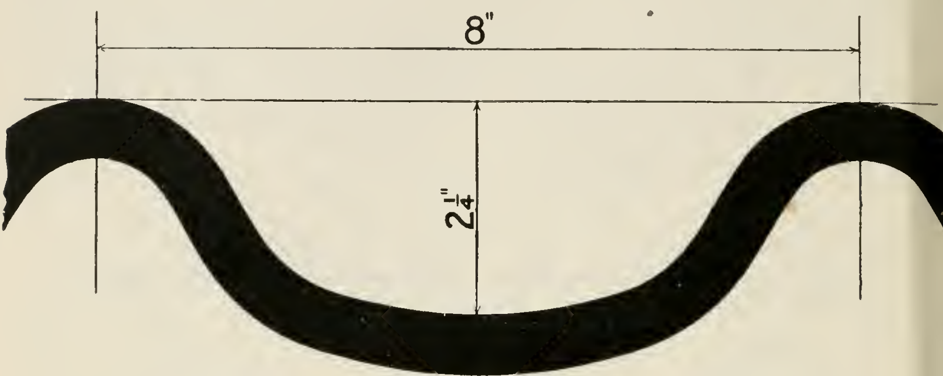
The safe pressure to be carried is therefore 176 lbs.

NOTE. — The strength of a plain furnace depends on the Length, Diameter, and Thickness squared.

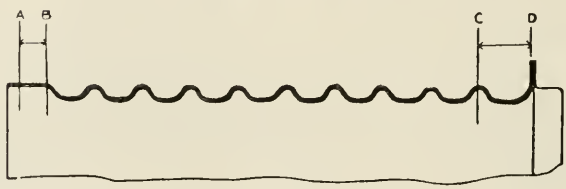


No. 46.—Bowling Hoop Furnace.

The flanged rings shown are known as "Bowling Hoops," and when fitted to a plain furnace increase the strength and at the same time allow for expansion. As will be understood the furnace length is divided up into two or three sections, and these are joined by the hoops.

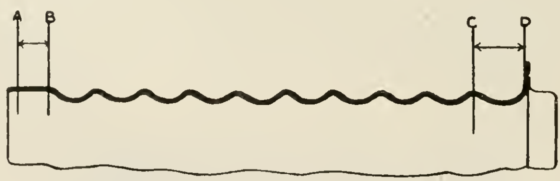


No. 48.—Suspension Bulb Furnace Corrugation.



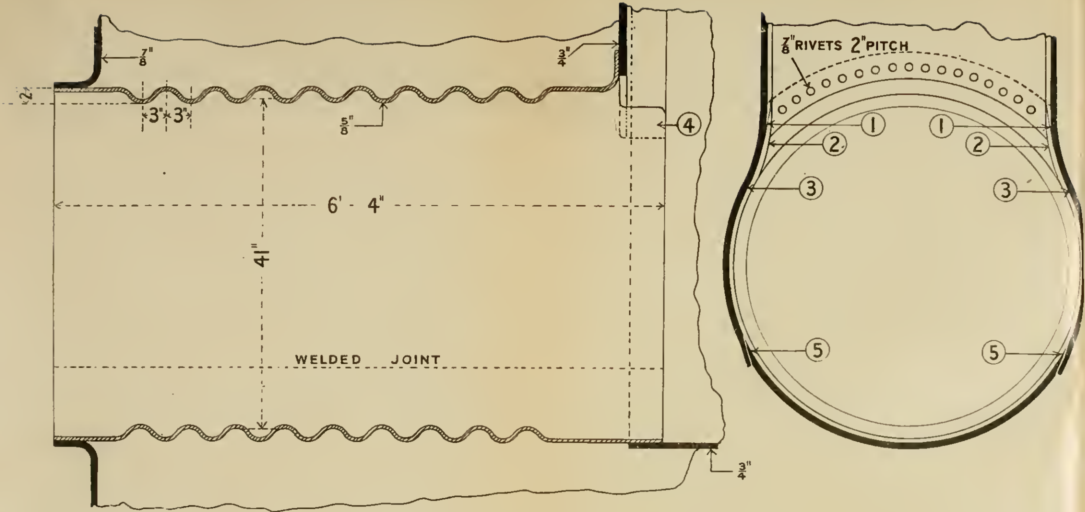
No. 49.—Suspension Bulb Furnace.

The lengths marked A B at front and C D at back are required by Board of Trade not to exceed 9 inches.



No. 50.—Morison Type Furnace.

The lengths marked A B and C D are required by Board of Trade not to exceed 9 inches.



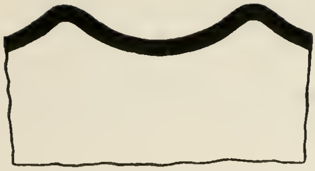
No. 47.—Fox Corrugated Furnace (ordinary type).

(With Details and Dimensions.)

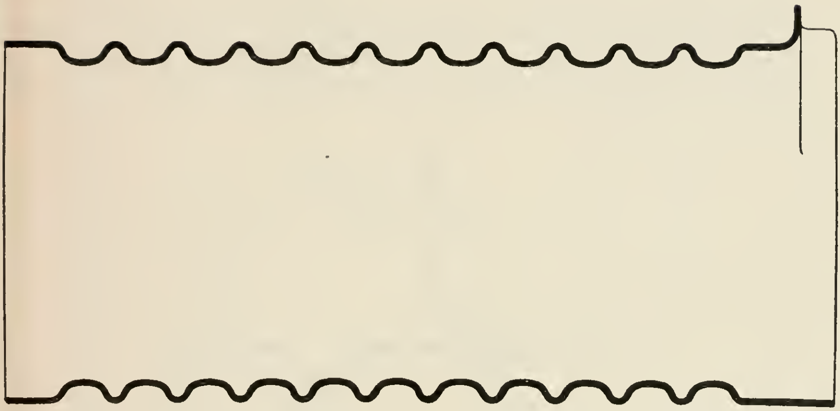
1, Thinned edge of back tube plate.
2, Thinned edge of furnace.
3, Wrapper plate of combustion chamber.
4, Position of three plate overlap (two plates of which are thinned away as described).
5, Scarfed joint of wrapper plate and bottom plate.

In this type of furnace, *three plates* overlap at the area marked 4, the plates being furnace flange, back tube plate, and combustion chamber side or wrapper plate. Two of these plates are, however, thinned away to form only one thickness as shown in the end view at 1 and 2. Owing to unequal expansion and difficulty in accurate fitting, this part of the furnace (known as the "saddle") frequently causes trouble by leakage and corrosion.

[*To face page* 113.



No. 51.—Morison Type Corrugation.



No. 52.—Morison Suspension Bulb Type Furnace.

RULE—

$$15000 \times T = D \times W.P.$$

Therefore, $$\frac{15000 \times T}{D} = W.P.$$

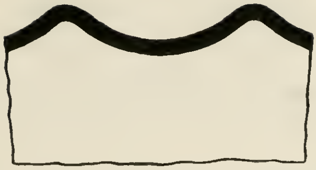
Or, $$\frac{D \times W.P.}{15000} = T.$$

And, $$\frac{15000 \times T}{W.P.} = D.$$

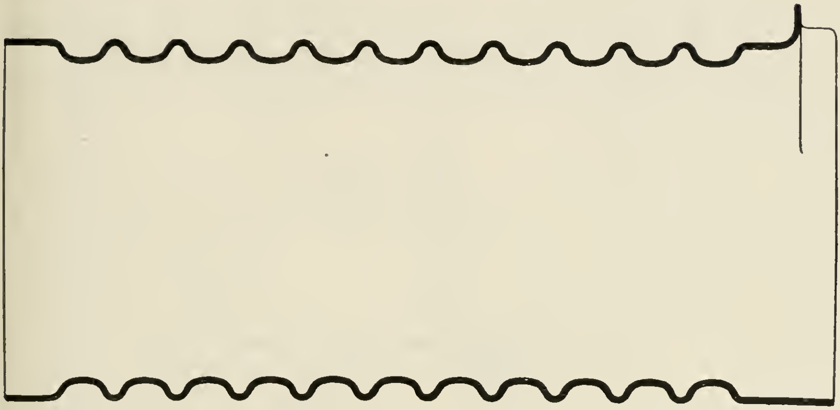
Where, T = Thickness.
,, D = Least outside diameter in inches.
,, W.P. = Working Pressure per square inch.

This type of furnace is allowed a higher Constant than any other type.

Pitch of bulbs = 8 inches.
Depth of bulbs = $2\frac{1}{4}$ inches (from top to least outside diameter).



No. 51.—Morison Type Corrugation.



No. 52.—Morison Suspension Bulb Type Furnace.

RULE—

$$15000 \times T = D \times W.P.$$

Therefore, $$\frac{15000 \times T}{D} = W.P.$$

Or, $$\frac{D \times W.P.}{15000} = T.$$

And, $$\frac{15000 \times T}{W.P.} = D.$$

Where, T = Thickness.
,, D = Least outside diameter in inches.
,, W.P. = Working Pressure per square inch.

This type of furnace is allowed a higher Constant than any other type.

Pitch of bulbs = 8 inches.
Depth of bulbs = $2\frac{1}{4}$ inches (from top to least outside diameter).

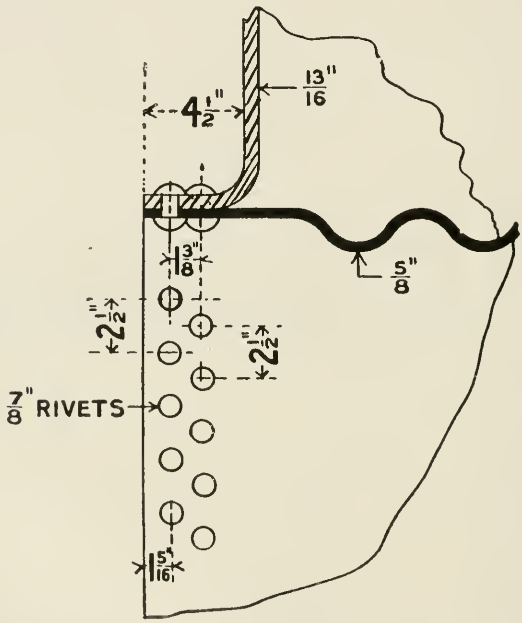
9

Repair for Leaky Telescope Furnace Joint.

Occasionally part of a corroded furnace is cut out circumferentially, and removed, a new length is then telescoped into the part remaining, and double riveting (zig-zag) employed on the joint. If, after the heat is applied, the joint proves leaky, the following remedies may be tried :—

1. Caulk the edges of the plates at joint.
2. Set up each rivet when cold with a hammer.
3. Build up an arch of fire brick on bars at position of joint to shield it from the heat.
4. Bore out, say, $\frac{5}{8}''$ diameter holes between rivets, and near edge of joint landing, tap, and screw in pins, which afterwards rivet over cold.

NOTE.—To form a driving fit, the telescope length should be $\frac{3}{32}''$ less in *circumference* than the part it fits into.



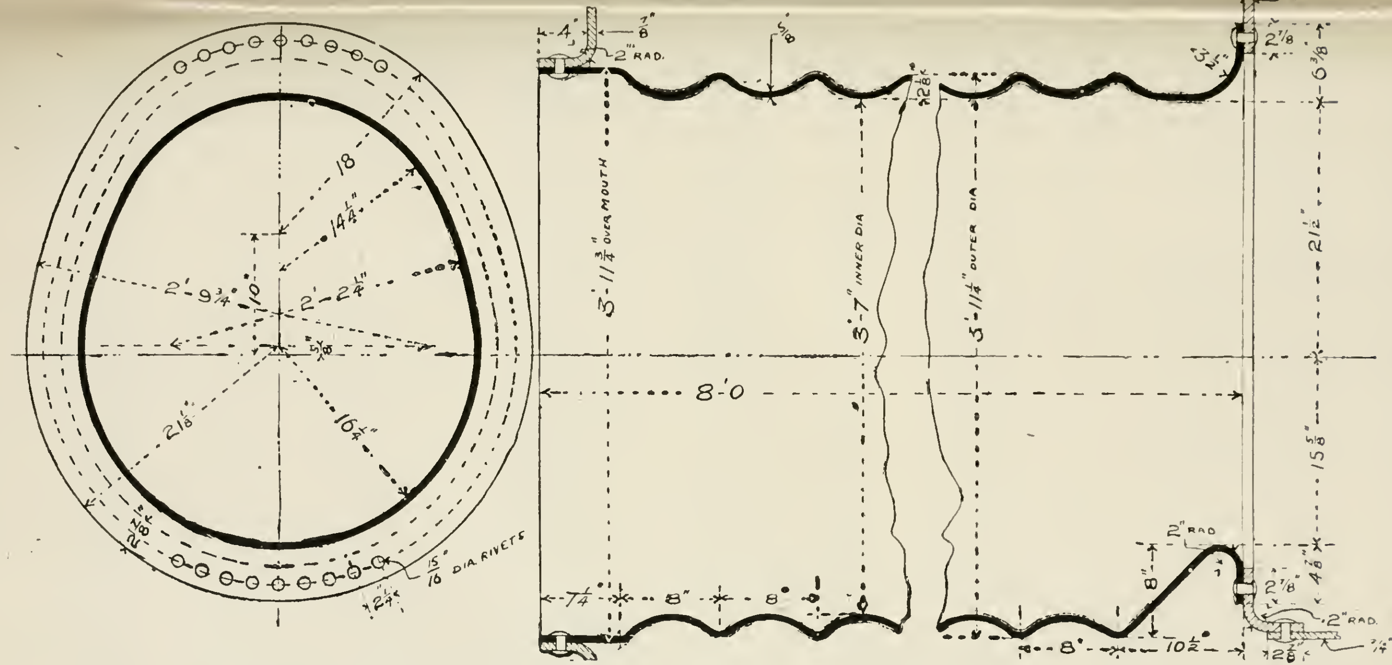
No. 53.—Furnace Front Riveting.

When double riveting is employed on the furnace front, the proportions of the joint are as shown, and the strength works out as follows :—

$$\text{Seam Section} = \frac{p-d}{P} \times 100 = \frac{2{\cdot}5 - {\cdot}875}{2{\cdot}5} \times 100 = 65 \text{ per cent.}$$

$$\text{Rivet Section} = \frac{A \times N \times 23 \times 100}{P \times T \times 28} = \frac{{\cdot}875^2 \times {\cdot}7854 \times 2 \times 23 \times 100}{2{\cdot}5 \times {\cdot}625 \times 28} = 63 \text{ per cent.}$$

The joint strength = 63 per cent.



No. 53a.—Morison Type Corrugated Furnace (With dimensions complete.)

View on left shows section at "goose-neck" position.

A careful study of the various sizes, flanging, &c., will be found of value to the student. It should be noted that the diameter at the front end is slightly in excess ($\frac{1}{2}''$) of that over the corrugations to allow for insertion or withdrawal of the furnace tube.

The pitch of the corrugations is 8″, and the depth of corrugation $2\frac{1}{8}''$. The "goose-neck" construction at the back allows of the furnace being taken out and a new one put in, as the tube can be drawn forward and then canted up by the lifting gear, the slope of the goose-neck at bottom allowing of this movement being possible.

NOTES ON BOILER UPKEEP AND REPAIR, ETC.

Boilers Laid Up.

Boilers not in use should be filled up full with fresh water from which the air has been expelled by boiling, the water to be kept slightly alkaline in condition by the addition of soda or lime.

If not convenient to fill up with fresh water, then dry the boilers with airing stoves, and place inside, on perforated trays, burning charcoal or coke, afterwards closing up the boilers air-tight. The coke absorbs the oxygen and thus prevents oxidation.

Zinc Plate Allowance.

Allow about 3 square inches of zinc, 1″ thick, per square foot of tube surface.

Lime and Soda.

If lime is used in boilers, allow about $1\frac{1}{2}$ lbs. to 2 lbs. per 1000 I.H.P. per 24 hours. The same proportion holds good for soda.

The lime to be dissolved in buckets, strained, and poured into the lime tanks or feed tanks.

Emptying Boilers.

Whenever possible, boilers should not be blown down, but ought to be allowed to cool down before emptying.

Hydrogen Gas in Boilers.

Hydrogen gas may be present in boilers newly emptied, and, if mixed with certain proportions of air, will produce an explosive mixture. Boilers should, therefore, be ventilated after blowing down or pumping out.

Leakage in Boilers.

In the event of serious leakage the ash-pit and furnace doors should be kept closed, and the steam pressure at once reduced by driving the main engines at maximum speed, by blowing steam into the condenser (silent blow-off), and by increasing the feed supply. The safety valves should be eased, and the fires quenched as soon as possible.

Buckled Plates.

For slight bulges in combustion chamber plates, clean the surfaces and leave them as they are, as if forced back by hydraulic jack pressure the treatment tends to weaken the plates.

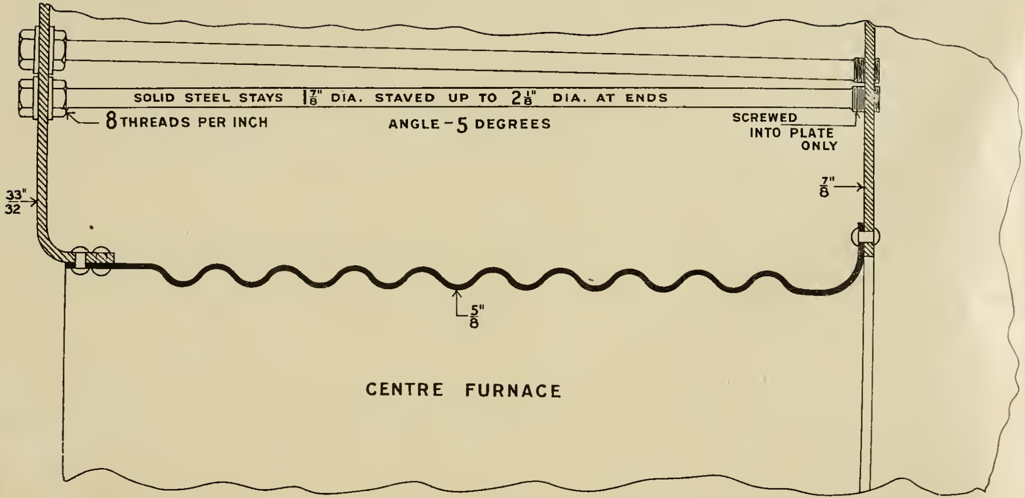
It is also necessary to renew the surrounding stays, making them, say, $\frac{1}{8}''$ or $\frac{1}{4}''$ larger in diameter.

Parts buckled and exposed to heat may be protected by means of a firebrick shield, or by covering over with fireclay.

The bulging referred to is, in most cases, caused by oil or scale deposits, which result in overheating of the plate.

Corrosion.

For parts showing symptoms of corrosion, clean and scrape carefully, then apply lime wash, or black lead polished up into the surfaces. A mixture of paraffin and zinc is often used for the same purpose.



No. 56.—Furnace and Tube Plate Solid Plates.

These stays are fitted to support the front and back tube plates on either side of the furnaces (Sketch No. 41).

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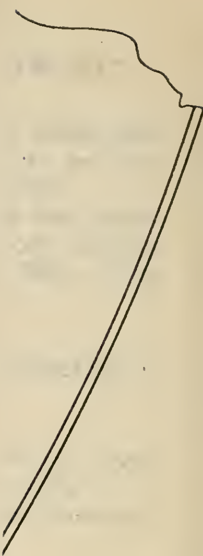

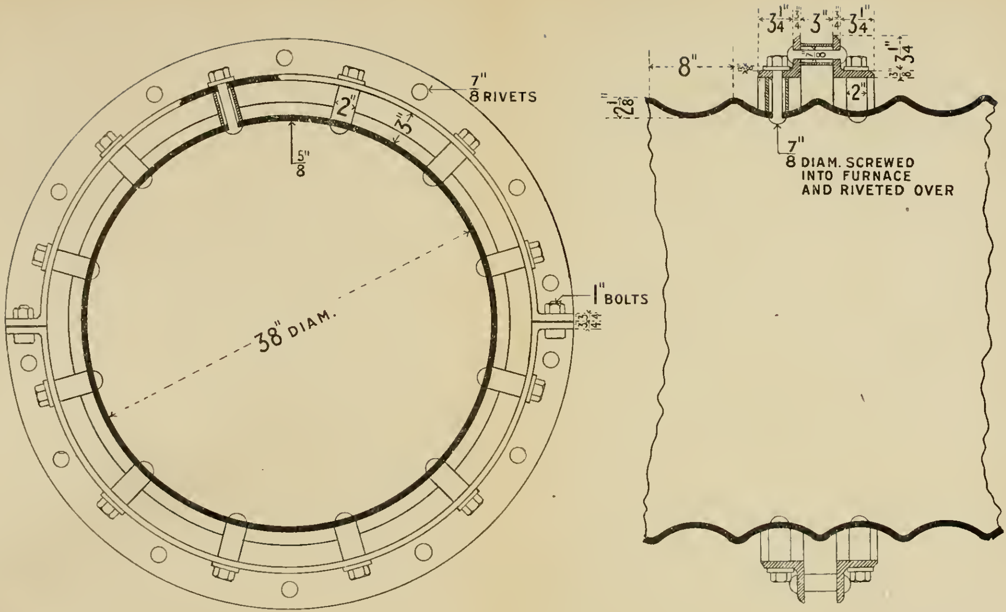

No. 57.—Method of Repair for a Weak or Collapsed Furnace.

As will be seen from the sketch, the repair consists of two angle irons riveted together through thimbles, and forming two half rings which are bolted together as shown. Pins $\frac{7}{8}$ inch diameter are tapped into the furnace and are riveted over inside, with nuts and washers outside, the pitch of these being about 12 inches. This arrangement stiffens the furnace, and is equally suitable for either plain or corrugated.

NOTE.—The ring is kept 3 inches clear of the furnace metal to allow of free circulation of the water.

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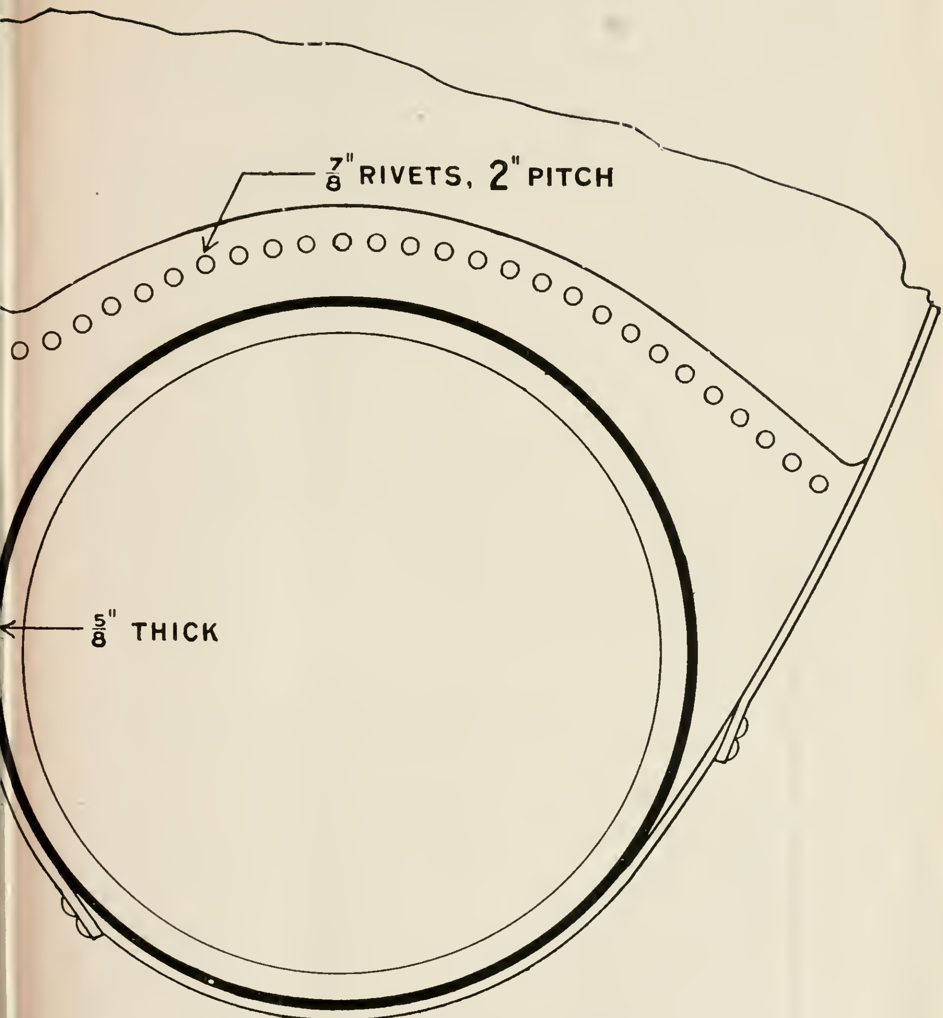


No. 54.—Wing Furnace Flanging.

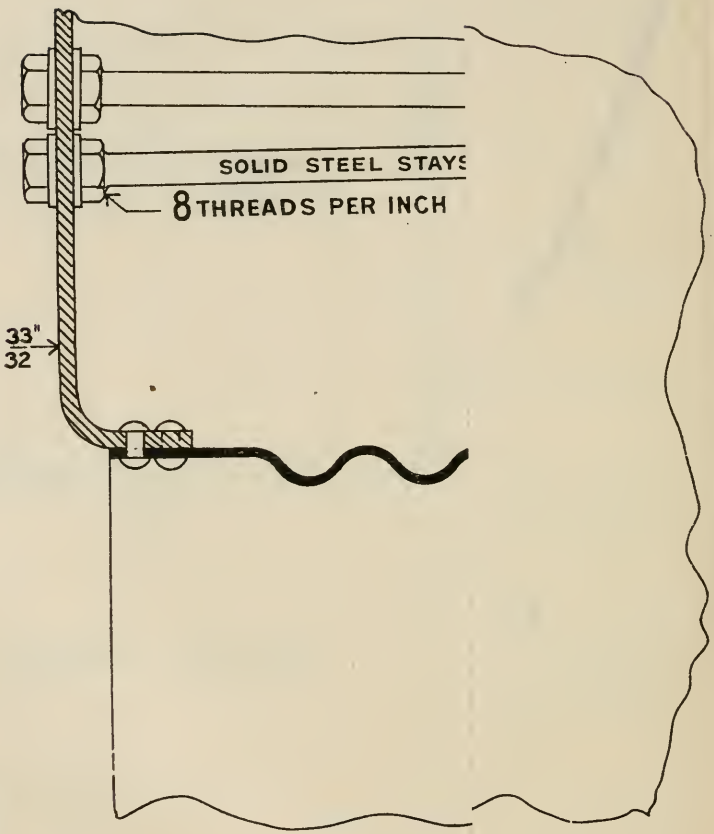
Notice that the flange is extended out to fit the shape of the combustion chamber.

To find Thickness of Furnace (Diameter, D = 44 inches, Pressure = 200 lbs.).

RULE—

$$14000 \times T \text{ in.} = D \text{ in.} \times \text{Pressure.}$$

Therefore, $$T \text{ in.} = \frac{D \text{ in.} \times \text{Pressure}}{14000} = \frac{44 \times 200}{14000} = \cdot 62 \text{ in., say } \tfrac{5}{8} \text{ in.}$$



These stays are fit

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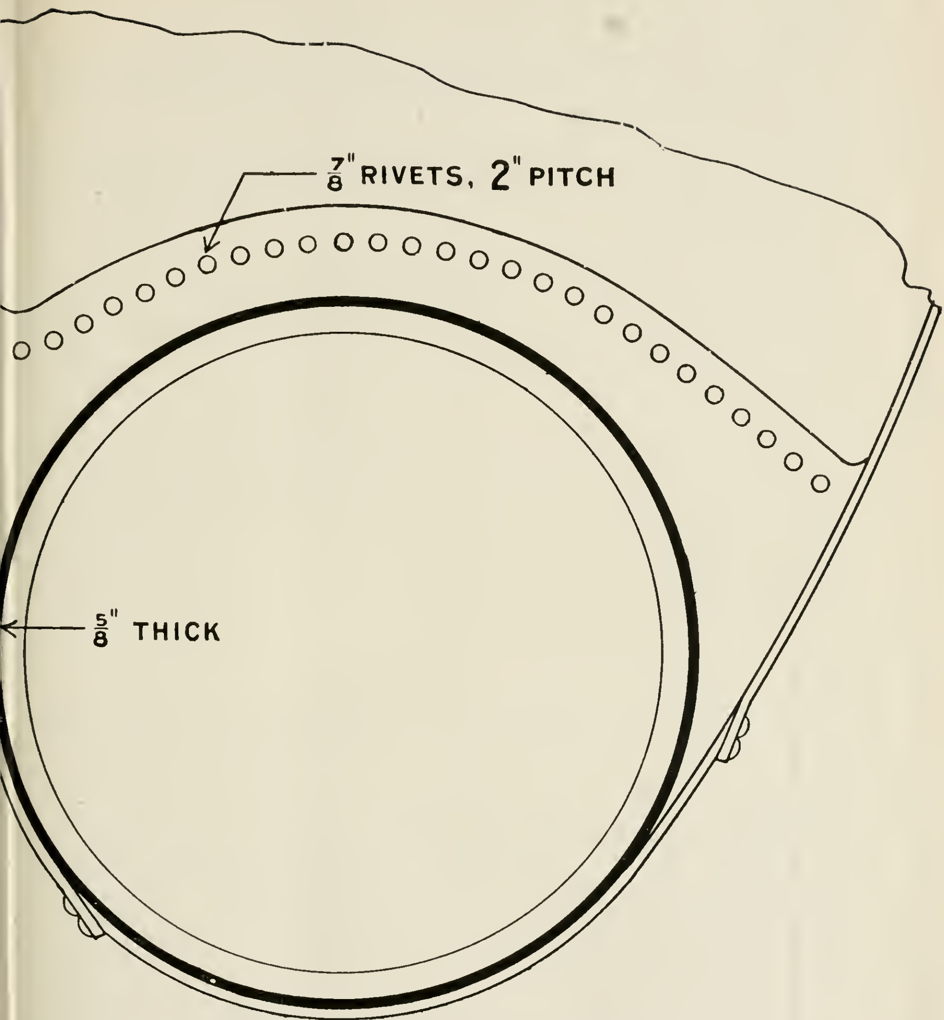

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Therefore, $$T \text{ in.} = \frac{D \text{ in.} \times \text{Pressure}}{14000} = \frac{44 \times 200}{14000} = \cdot 62 \text{ in., say } \tfrac{5}{8} \text{ in.}$$

Furnaces.

Furnaces are usually made from $\frac{1}{2}$ to $\frac{3}{4}$ inch in thickness.

The strength of a plain furnace depends on the Length, Diameter, and Thickness squared.

Method of Strengthening a Weak or Collapsed Furnace (with Dimensions).

The Sketch No. 57 illustrates the best method of repair for a weak furnace, or for a collapsed furnace which cannot be set up.

Observe that the $\frac{7}{8}$-inch studs are screwed into the furnace metal and riveted over, also that thin thimbles or distance pieces are fitted between the half-round strengthening ring and the furnace, to allow of free circulation of the water. The pitch of the studs varies from 10 to 13 inches.

As in a plain furnace the strength depends upon the length, diameter, and thickness squared, if we reduce the length by one-half we double the strength; therefore the ring round the centre practically shortens the length, and correspondingly increases the strength (always subject to Limit Pressure Rule result).

NOTE.—Thimbles must be fitted between the furnace and the ring to allow of water circulation between.

Types of Furnaces.

The Sketches No. 58 illustrate the corrugations of the three most important types of furnaces in use at the present time.

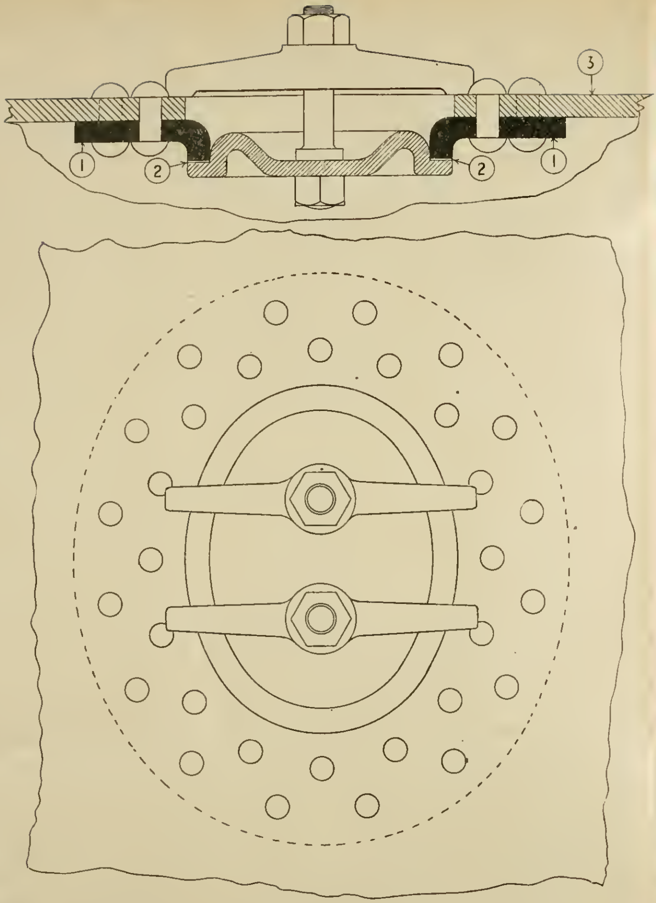
Corrugated Furnace Manufacture.

The process of manufacture of furnaces is as follows:—The plate is rolled with one edge thicker than the rest of the plate, so as to allow for the thinning which takes place in flanging. The plate is then sheared to size, and bent to a cylindrical shape in the bending rolls. It is then passed to the electrical welding apparatus, where it is welded up, after which the furnace is taken to the corrugating mill to be corrugated, and to the various hydraulic machines for flanging. The furnace is then machined to correct sizes, and set true to template. Finally, each furnace is carefully annealed before despatch.

Advantages of Corrugated Furnaces over Plain Furnaces.

1. Stronger than a plain furnace of the same dimensions.
2. Better expansion allowance by means of the corrugations or ribs.
3. More surface for the same length, and therefore better evaporation is obtained.

NOTE.—For a plain furnace the limit pressure is equal to $\frac{9900 \times T}{D}$.



No. 61.—Boiler Shell Manhole (16 inches by 12 inches).

1, Compensation Ring. 2, Joint of Door. 3, Boiler Shell.

NOTE. The upper view (section) is taken in the longitudinal direction, and shows the short diameter of the manhole (12 inches).

The compensation ring is of the same thickness as the shell plate.

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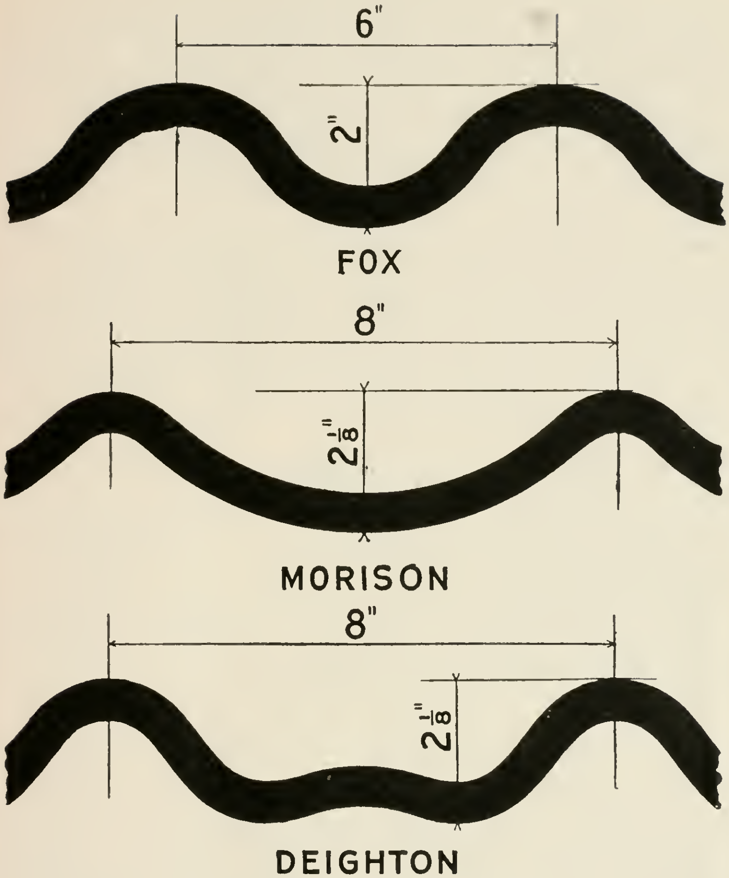


No. 58.—Types of Furnaces.

(With Pitch and Depth of Corrugations.)

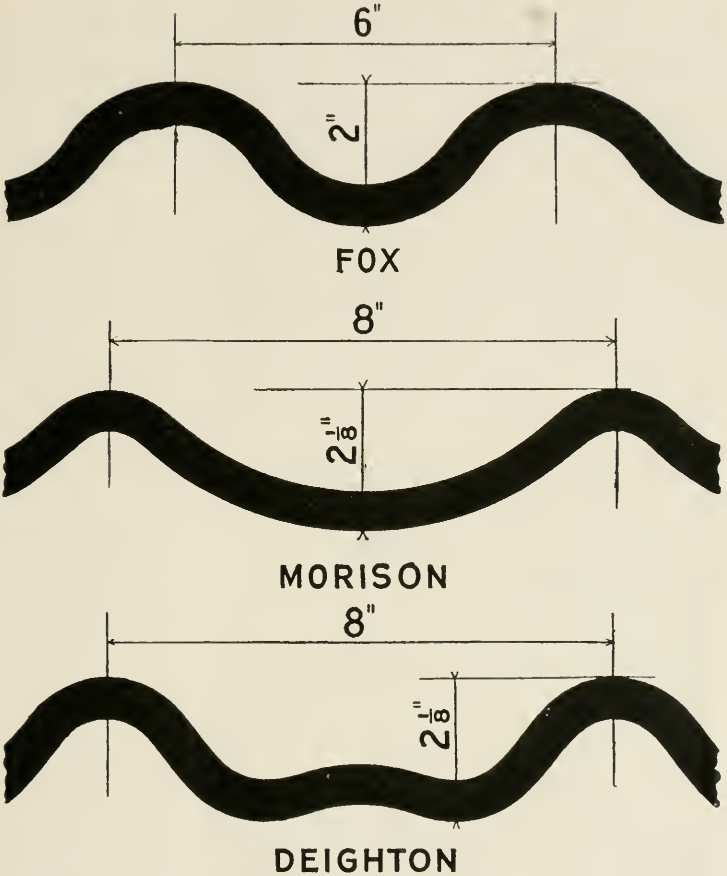
A corrugated furnace of the same dimensions is about half as strong again as a plain furnace of average proportions.

The strength of a corrugated furnace depends upon the Diameter and Thickness.

Collapse of Furnaces.

Furnaces may be collapsed by any of the following causes:—

1. Excessive scale. The scale keeps the water from being in direct contact with the plate, and overheating takes place (about 600° Fahr.), with the result that as the plate is weakened part of the furnace bulges in.



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(With Pitch and Depth of Corrugations.)

A corrugated furnace of the same dimensions is about half as strong again as a plain furnace of average proportions.

The strength of a corrugated furnace depends upon the Diameter and Thickness.

Collapse of Furnaces.

Furnaces may be collapsed by any of the following causes:—

1. Excessive scale. The scale keeps the water from being in direct contact with the plate, and overheating takes place (about 600° Fahr.), with the result that as the plate is weakened part of the furnace bulges in.

2. Oil deposits adhering to the furnace. Oil deposited on the furnace top or sides, for a given thickness, is more serious than ordinary scale, as greater overheating of the furnace plates will ensue.

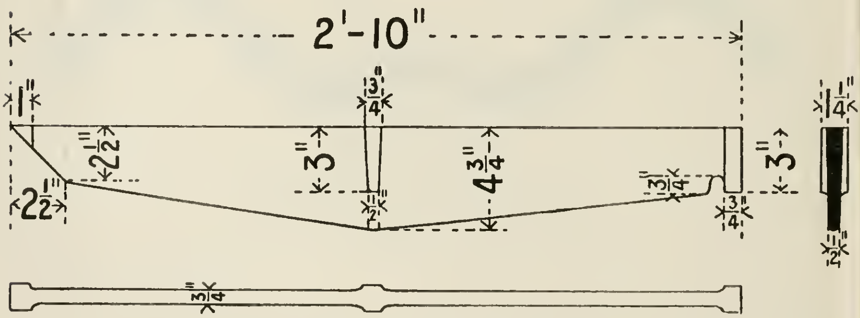
3. Saturation or salt deposits. The boiler density rising above $\frac{7}{32}$, the surplus salt in the water deposits, and the water not being in contact with the metal, overheating of the plates takes place and collapse follows.

Cases have been observed where furnaces have come down when lying under banked fires. One of the most reasonable theories put forward to account for this is the lack of circulation existing (especially with a high density), causing a layer of steam to form between the water and the metal of the furnace, with consequent overheating and collapse; but opinion is somewhat divided on this point, and the true cause is not definitely known.

Furnace Temperatures, &c. (approximate).

Furnace temperature - - - -	about	2600°	Fahr.
Combustion chamber temperature - -	,,	1500°	,,
Uptake temperature - - - -	,,	750°	,,
Funnel temperature - - - -	,,	600°	,,

NOTE.—The above temperatures (measured by a pyrometer) vary under different conditions, but these may be taken as average.

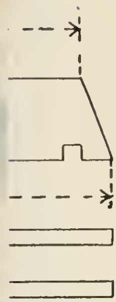


No. 59.—Fire Bar.

(With Dimensions.)

The air spaces are formed by small projections cast on the sides of the bars, which ensures the required air clearance.

When the bars are fitted in two lengths, the dimensions are usually as marked. Notice that one end of the bar is bevelled away to allow for expansion under heat, and the other end is "hooked" to grip the edge of the centre bearer.



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The manholes in the boiler ends are often arranged with the end plates flanged in to give strength, instead of fitting a compensation plate; in addition to this three large stays are passed through from end to end of the boiler, and secured to the end plates by nuts, as shown in the sketch. These stays support the plate round the portion weakened by the cutting of the holes.

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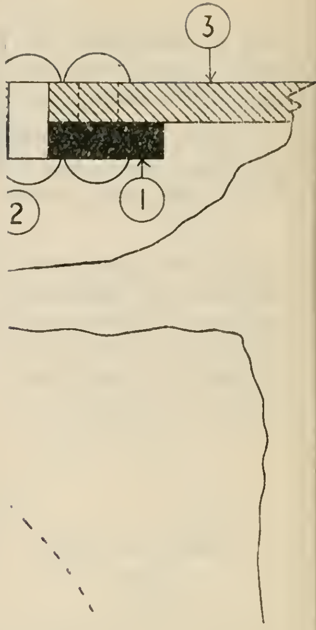

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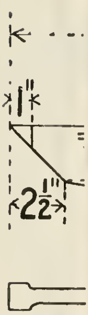
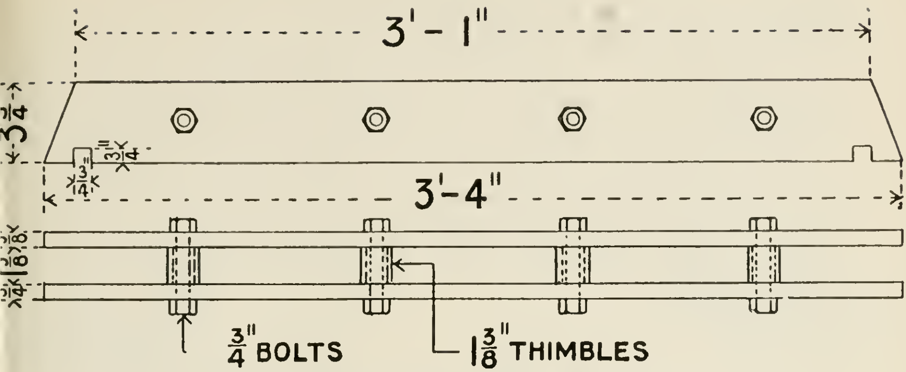


The air spaces are formed by small projections cast on the sides of the bars, which ensures the required air clearance.

When the bars are fitted in two lengths, the dimensions are usually as marked. Notice that one end of the bar is bevelled away to allow for expansion under heat, and the other end is "hooked" to grip the edge of the centre bearer.



No. 60.—Centre Bearer for Fire Bars.

When the bars are arranged in two lengths, the centre bearer is fitted as shown; the bearer sits on an angle knee plate at either side of the furnace, the knees being held in position by studs screwed through the furnace, and the bars hook on to the edge of the bearer plates.

Manholes.

The standard size for manholes is 16 inches by 12 inches, but sometimes those on the end plates between the furnace openings are less, being 15 inches by 11 inches, or thereabout.

Shell manholes are cut with the long diameter circumferentially and the short diameter *longitudinally*, as the strength of the plate is least in this direction.

Shell manholes have compensation rings riveted to the shell and of equal thickness, the area of ring to be not less than that of the metal cut out plus the rivet hole areas. The compensation ring also forms the joint for the door (Sketch No. 61).

The clearance round the door and opening should be as nearly as possible $\frac{1}{16}$ inch a side, as excessive clearance is often the cause of the doors blowing out.

After blowing down the boilers, and before taking off the manhole doors, the drain cock of the water gauge should be opened so that air may enter the boiler and destroy the possible vacuum left by blowing down. Neglect of this may lead to an accident, as the doors might be blown in by atmospheric pressure when the nuts are eased back.

The manholes in the boiler ends are often arranged with the end plates flanged in to give strength, instead of fitting a compensation plate; in addition to this three large stays are passed through from end to end of the boiler, and secured to the end plates by nuts, as shown in the sketch. These stays support the plate round the portion weakened by the cutting of the holes.

Compensating Ring of Manhole.

As the stress on the boiler shell longitudinally is twice that circumferentially, it is sufficient to calculate the width of ring for the longitudinal direction only.

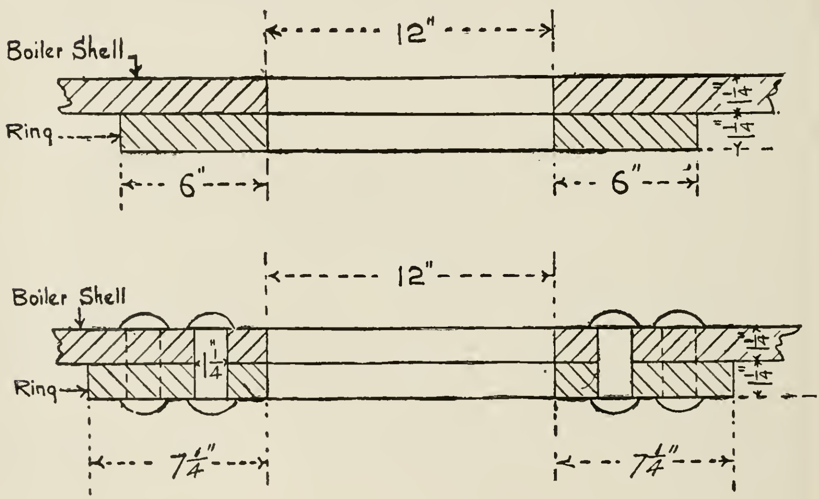
Suppose manhole opening to be 16″ by 12″, then the 12″ cut is longitudinally, and if the shell is, say, $1\frac{1}{4}''$ thick, the section removed is equal to $12 \times 1{\cdot}25 = 15$ □″.

The compensating ring section must be at least equal to this, and assuming equal thickness of shell and ring, and neglecting rivets, the rule becomes :—

$$12'' \times 1{\cdot}25'' = W'' \times 1{\cdot}25'' \times 2$$

W = width of ring.
2 = two widths (one on either side).

Then, $W = \dfrac{12 \times 1{\cdot}25}{1{\cdot}25 \times 2} = 6''$ width of ring (rivets neglected).

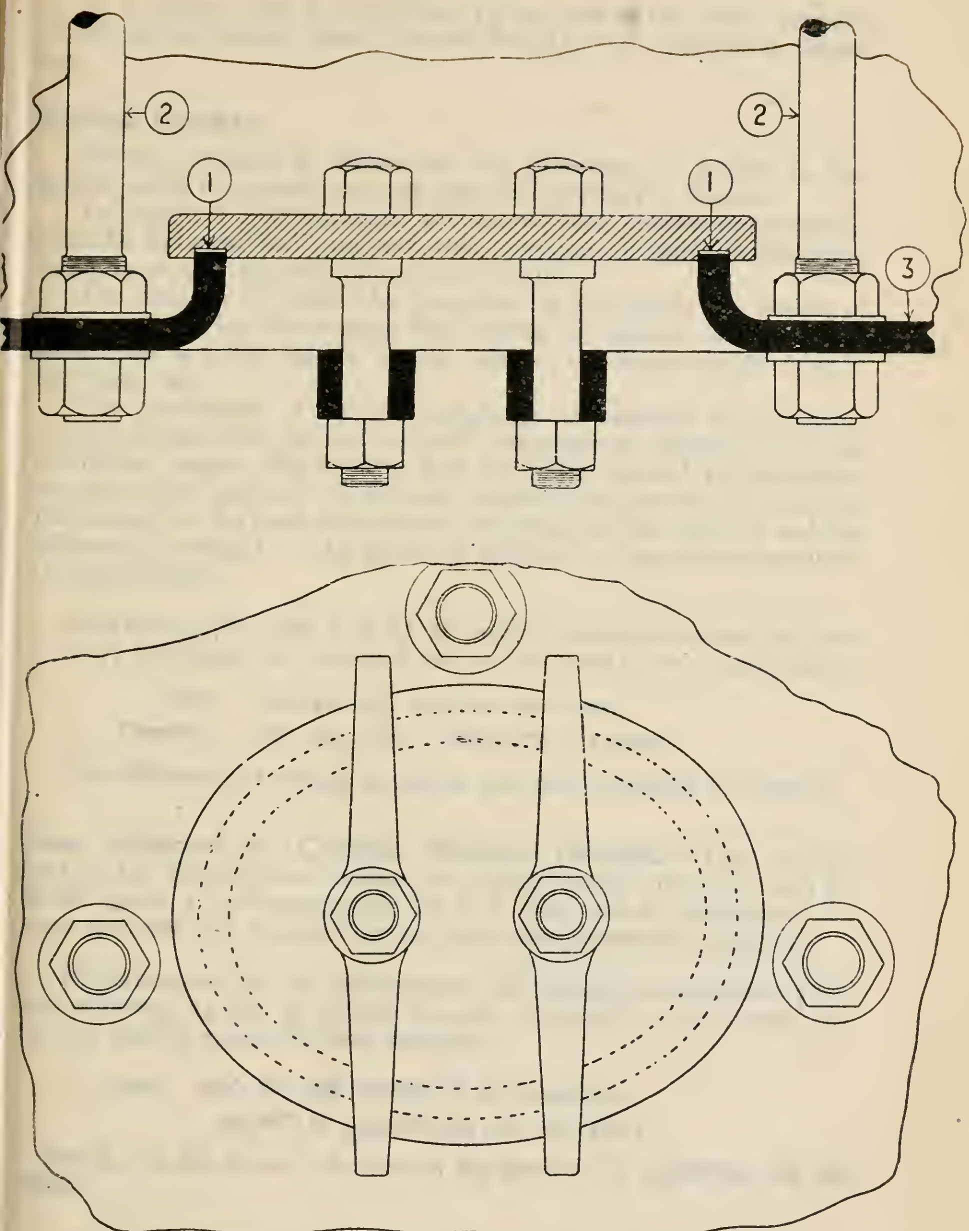


If, say, the rivets are $1\frac{1}{4}''$ diameter, then W is found as follows :—

$$12 \times 1{\cdot}25 = (W - 1{\cdot}25) \times 1{\cdot}25 \times 2.$$

Then, $W = \dfrac{12 \times 1{\cdot}25}{1{\cdot}25 \times 2} + 1{\cdot}25 = 7\frac{1}{4}''$ width of ring (rivets allowed for).

As the rivet holes weaken the ring, the width of it must be increased in proportion to keep up the strength.



No. 62.—Boiler End Plate Manhole (15 inches by 11 inches).

1, Joint of door. 2, Stays. 3, End plate flanged in.

[*To face page* 120.

A doubling plate is also riveted to the *back* of the boiler opposite to the manhole in the front (Sketch No. 33), and covering a similar area.

Natural Draught.

Natural draught is caused by the difference of weight in the heated air of the uptake and the cold air entering the furnace.

To obtain a good draught the funnel and uptake temperatures must be between 600° and 700°, this temperature being necessary to bring about the required difference of weight.

The draught can often be improved by increasing the length of the funnel, as by this means the column of heated, and therefore lighter, air is made less in weight, against the same weight of cold and heavy air.

The production of natural draught is an example of heat convection, as the cold air at, say, 60° temperature, passing down the ventilators, enters the furnace and becoming heated expands and therefore rises, passing off by way of the tubes, uptake, and funnel. The weight of the heated air is less than that of the cold air, and the difference in weight can be found by taking the absolute temperature as shown below.

EXAMPLE.—A cubic foot of air at 62° temperature weighs ·076 of a lb.; determine the weight if the air is heated to 600° temperature.

Then, $62+461=523$, and $600+461=1,061$.

Therefore, As $1061 : 523 :: \cdot076 = \cdot0374$ of a pound.

The difference of weight as shown produces a current or draught.

Heat Absorbed in Creating Natural Draught.—The specific heat of the funnel gases is about ·23, which means that to raise 1 lb. of the gases 1° in temperature ·23 of a heat unit is necessary. To show then the loss incurred by the generation of natural draught:—

EXAMPLE.—Cold air temperature 62°, uptake temperature 700°, and allowing 24 lbs. of air per lb. coal, calculate the heat units per lb. coal used in producing the draught.

Then, $700° - 62° = 638°$ increase of air temperature,

and B.T.U. required $= 638 \times 25 \times \cdot23 = 3668\cdot5$.

NOTE.—24 lbs. of air + 1 lb. coal = 25 lbs. gases in all (neglecting ash and clinker).

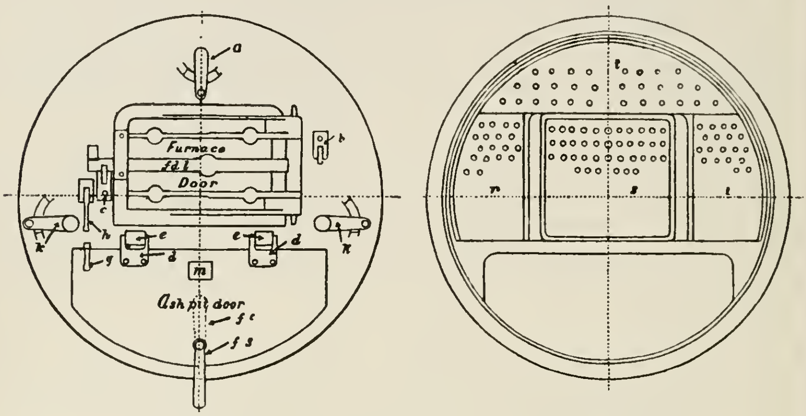
As a Percentage.—Assuming that 1 lb. coal contains 14,500 Heat Units,

Then, As $14500 : 3668\cdot5 :: 100 : 25$ per cent.

So that 25 per cent. of the heat units in each lb. of coal are used up in producing the necessary difference in temperature of the funnel gases required to form a draught by difference of weight.

Howden's Forced Draught.

In this well-known system of artificial draught, a large fan driven by means of a small engine, usually placed in the engine-room, forces the air along the air trunk into the air heating box



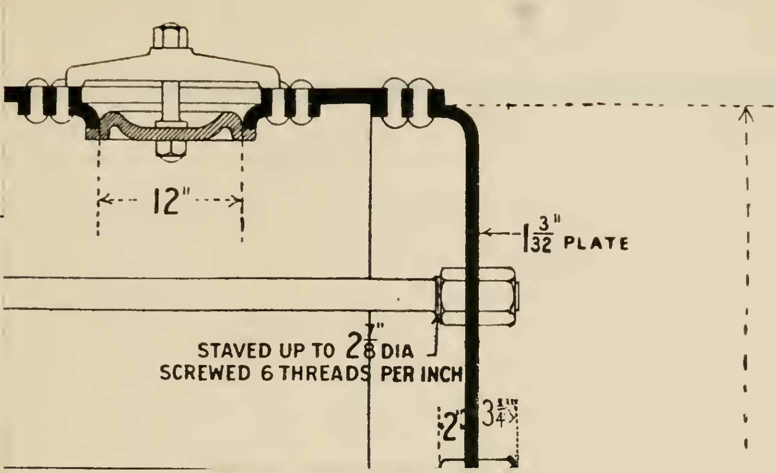
Front View. Inside View.

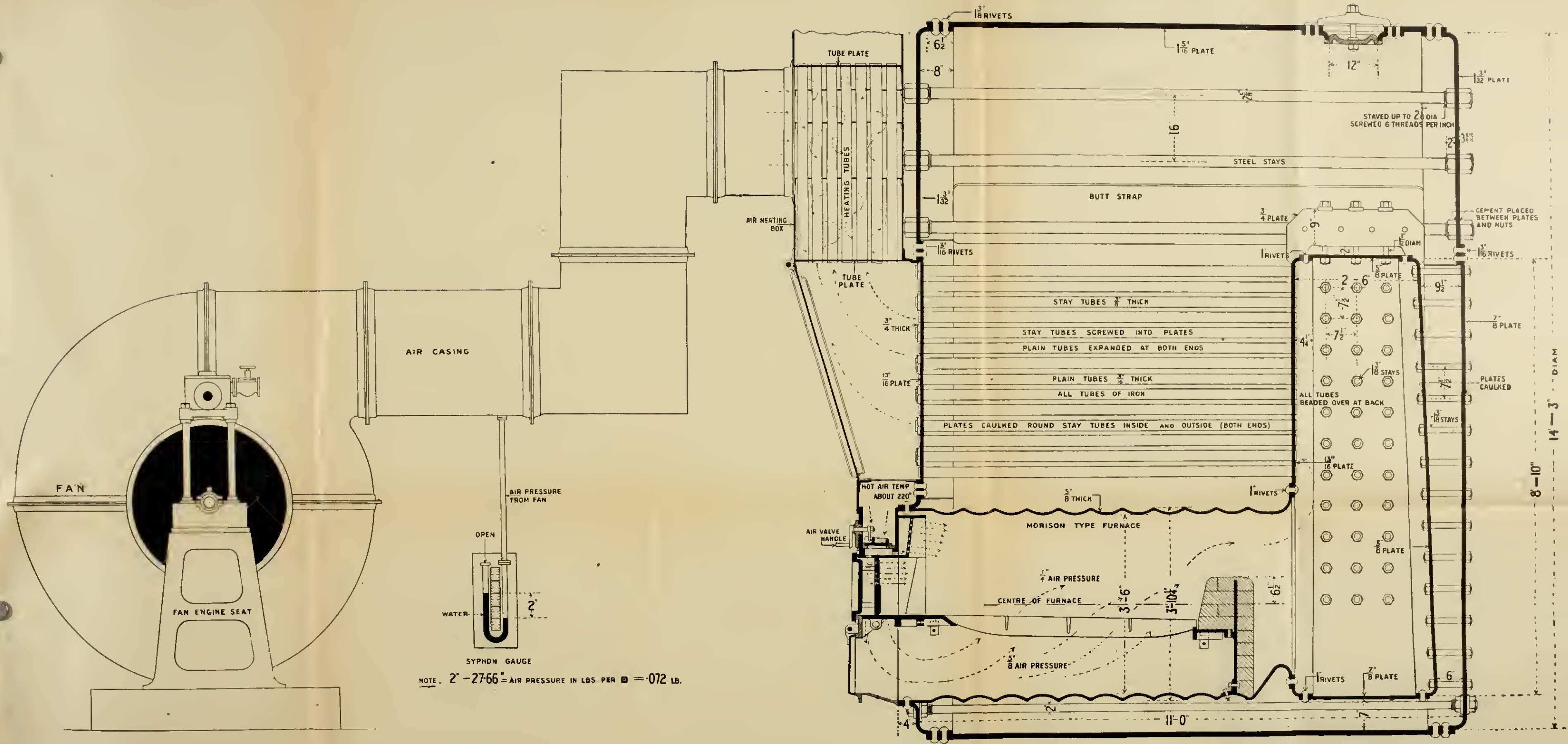
No. 64.—Howden's Forced Draught Furnace Mountings.

a, Top Valve Handle.
b, Back Catch.
c, Front Catch.
d, Hinge Flat.
e, Hinge Centre.
fc, Ashpit Door Handle for Centre Front.
fs, Ashpit Door Handle for Side Front.
g, Catch for Ashpit Door.
h, Hanger for Ashpit Door.
k, Side Valve Handle.
l, Left-hand Baffle Plate (or Air Box).
m, Mica Plate.
r, Right-hand Baffle Plate (or Air Box).
s, Door Baffle Plate (or Air Box).
t, Top Baffle Plate (or Air Box).
u, Furnace Door.
x, Ashpit Door.

situated in the uptake, and consisting of two tube plates with a series of short vertical tubes between. The hot gases from the smoke tubes pass up through the sets of short tubes in the box, and the air, being outside of these, is heated by the otherwise waste gases, and passes by openings in the sides through a casing on the boiler end down to the hot air receiver above the furnaces (see sketch).

The fronts of the furnaces are closed, and have three metal valves fitted, by which the air supply can be regulated to the furnaces, both above and below the fire-bars, one valve admitting the air


12"
$1\frac{3}{32}$" PLATE
STAVED UP TO $2\frac{7}{8}$" DIA
SCREWED 6 THREADS PER INCH
$3\frac{1}{4}$"



No. 63.—Sectional View of Boiler with Howden's Forced Draught.

The draught is generated by the fan shown, and the cold air (say at 60°) is delivered, by way of the air casing, into the air heating box which is situated in the uptake. The air circulates *outside* the heating tubes, and becomes raised in temperature by the waste gases to 200° or 220° Fahr., at which temperature it enters the furnaces, being admitted by one valve above the fire-bars, and by two valves (both operated by one handle) to below the bars. A suitable air pressure at the furnace is $\frac{1}{4}$ inch pressure above the bars and $\frac{3}{8}$ inch pressure below the bars. This pressure can only be measured by a portable U tube similar to the one shown in the drawing.

NOTE.—The various details shown above should be carefully studied by the student, the data given being that of the most recent practice. It should be observed that the water in the U tube of the hydrostatic gauge rises higher in the leg open to the atmosphere, as the pressure of the draught is slightly in excess of the atmospheric pressure. With induced draught the water would rise highest in the opposite leg of the tube.

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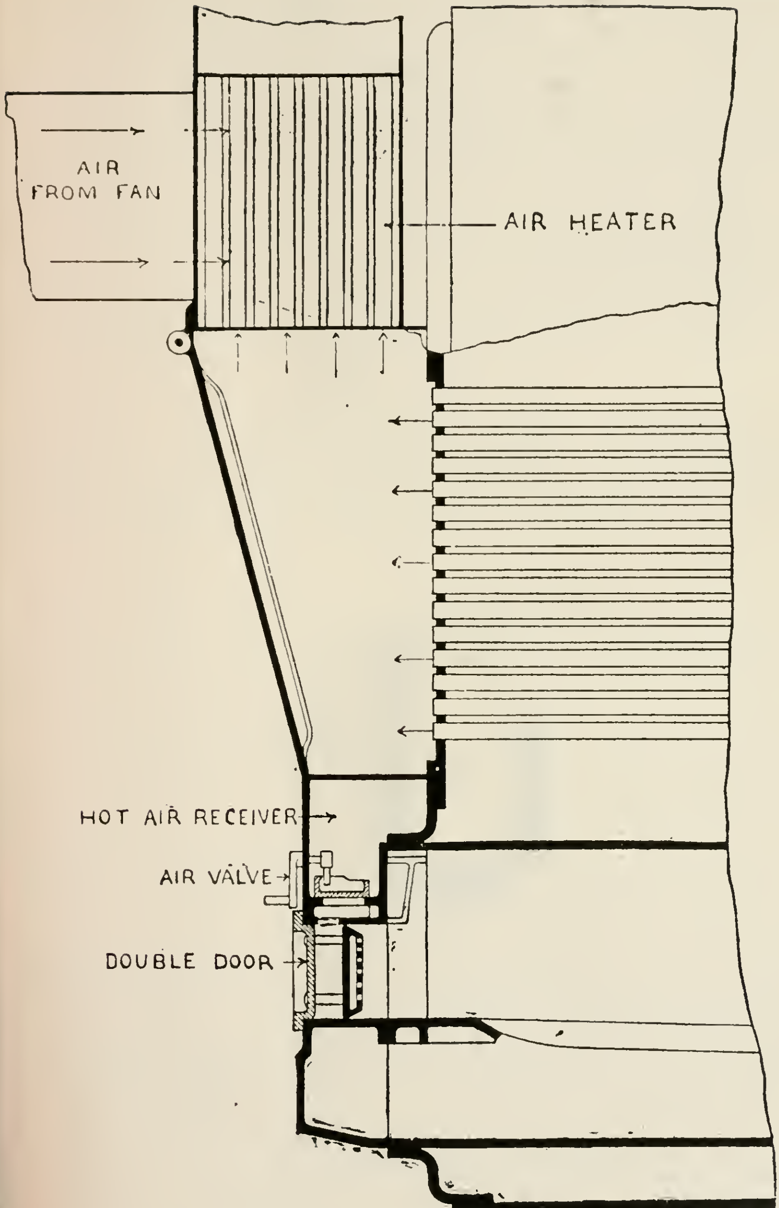
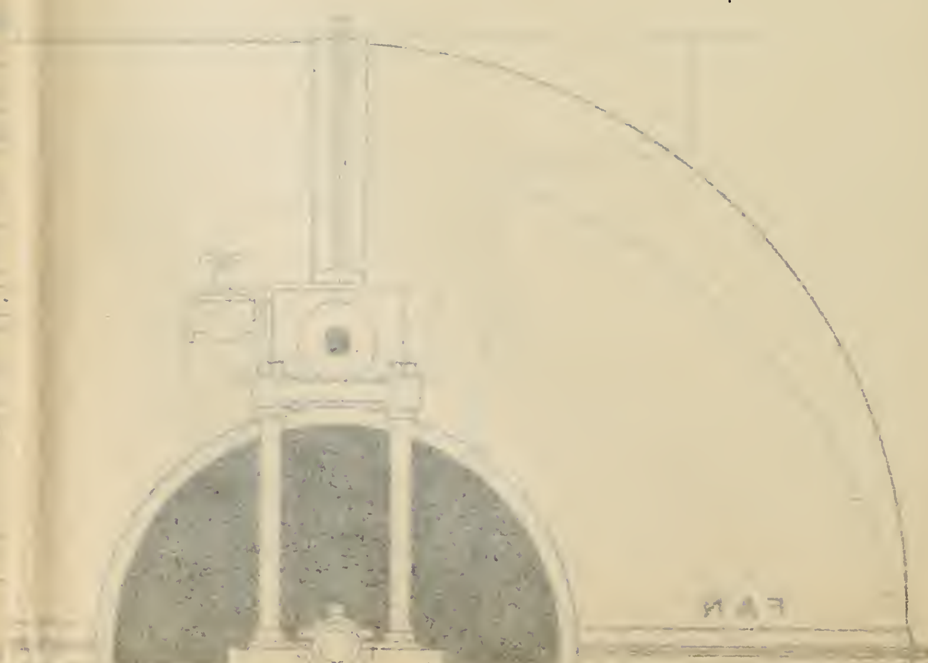
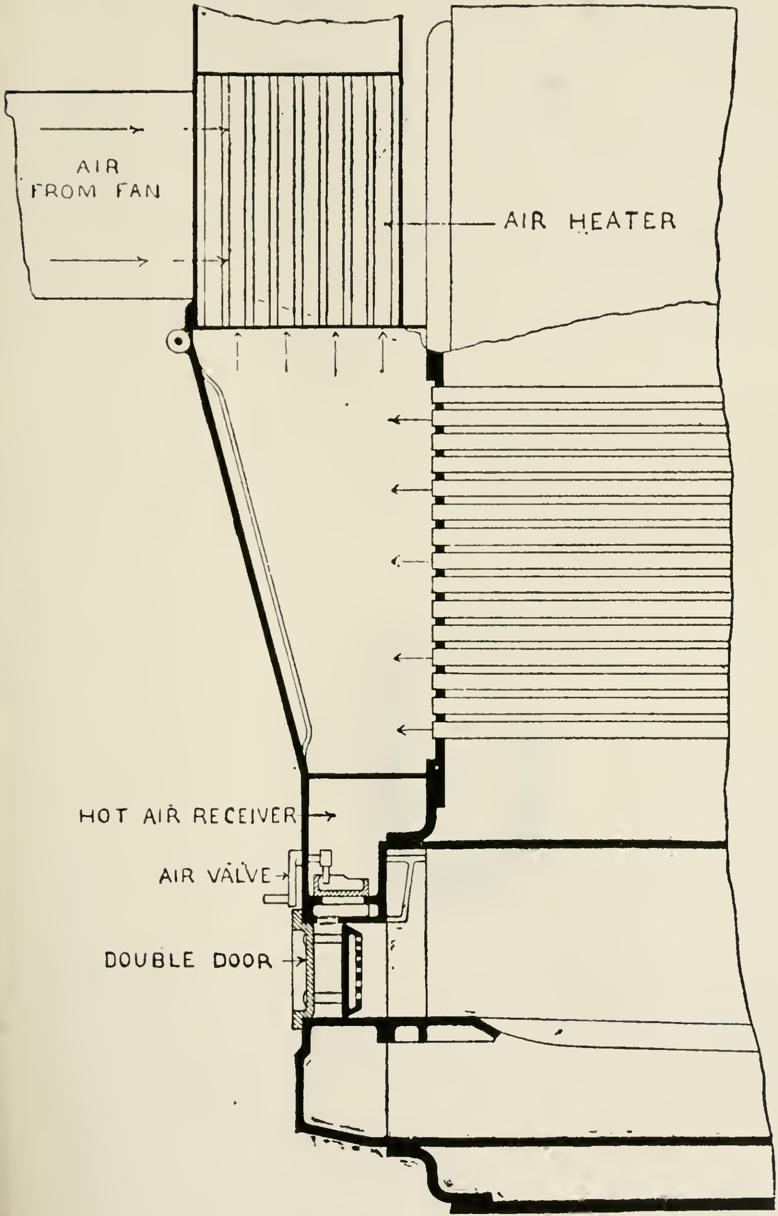


No. 65.—Howden's Forced Draught.

The air is heated by the otherwise waste gases to about 220° temperature before entering the furnaces.



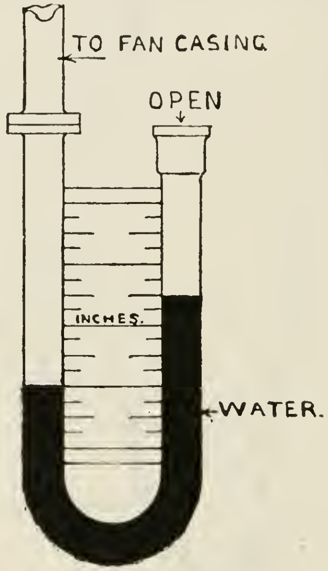


No. 65.—Howden's Forced Draught.

The air is heated by the otherwise waste gases to about 220° temperature before entering the furnaces.

above the bars, and the other two admitting it below. The furnace doors are made double, the outer half being airtight, and the inner half being perforated with small holes for the jets of air to pass through. It should be mentioned that with forced draught the smoke tubes are made smaller than usual, generally from $2\frac{1}{4}$ to $2\frac{1}{2}$ inches in diameter outside, and that strips of twisted metal, called "retarders," are often fitted inside of them to increase their heating power by retarding or keeping back the hot gases, so that as much of the heat as possible is given up to the water before the gases leave the tubes.

The force or intensity of the draught is measured by a U-shaped glass tube containing water, one end of the tube being connected



No. 66.

to the air trunk and the other end left open to the atmosphere. The air pressure in the trunk forces the water higher in the leg of the tube which is open to the atmosphere and lower in the leg open to the air casing, and the difference of the two water levels is called the air pressure, and is expressed in inches of water.

In practice from $1\frac{1}{2}$ to 3 inches of water is the amount carried.

If the water gauge shows, say, 3 inches of water, to find the pressure of the draught divide this by 27·66 inches.

NOTE.—A column of water 27·66 inches in height weighs 1 lb. per square inch.

Thus, $\frac{3}{27{\cdot}66} = {\cdot}108$ lb. per sq. in.

NOTE.—If the water gauge for the draught indicates about $2\frac{1}{2}$ or 3 inches at the fan, the pressure under the fire-bars will only be equal to about $\frac{5}{8}$ inch or thereabout.

Gains of Forced Draught.

1. Smaller boilers for the same power, as the consumption is more per square foot of grate.
2. Hot air enters the furnace in place of cold air.
3. Better steaming of power boilers.
4. Better control of fires, as the draught is independent of weather conditions.

With forced draught the air is partly heated by the waste gases before entering the furnaces, which means that less heat requires to be taken from the coal to heat it, and, as the consumption per square foot of grate surface is more than with natural draught, more evaporation will be the result, and therefore a smaller boiler will supply the same amount of steam: in addition to this the boilers generally steam much easier with forced draught.

NOTE.—As part of the heat of the waste gases is absorbed by the air in the heating box, the temperature of the funnel gases is less, being somewhat between 450° and 550° in general practice. The safety valves are made larger when forced draught is fitted, to allow for the increased evaporation per square foot of heating surface and resultant increase of steam generation.

Notes on Forced Draught.

About 20 lbs. of air per pound of coal are required for combustion, instead of 24 lbs. as with natural draught. The air being heated to about 210° by the waste products of combustion, requires less heat from the coal to effect combustion.

The consumption of coal may range from 25 to 40 lbs. per square foot of grate per hour according to the force of the draught carried. This results in reduced size of boilers being sufficient for a given power.

The air being forced into the furnaces is better distributed and more effective, which results in higher furnace temperatures.

With a water pressure of $2\frac{1}{2}$ inches at the fan, the pressure under the bars should be equal to about $\frac{7}{8}$ inch water, and above the bars equal to about $\frac{1}{4}$ inch water.

The chief disadvantages of forced draught are:—

1. Greater risk of collapsed furnaces if coated with scale or oil deposit, owing to higher temperature of furnace.
2. Greater tendency to leaky tubes and seams, owing to higher temperature of gases.
3. Trouble with tubes choking up with soot, if not cleaned often and regularly.

Heat Saved.

The approximate number of heat units saved by forced draught per pound of coal may be calculated as follows:—

Natural Draught.—Assume 24 lbs. of air per pound coal, cold air temperature 62°, funnel gases temperature 650°, specific heat of gases ·23.

Then, $650° - 62° = 588°$ rise of air temperature.

And, $588 \times 25 \times \cdot 23 = 3{,}381$ Heat units required per pound of coal.

NOTE.—24 lbs. of air + 1 lb. of coal = 25 lbs. weight of gases in all.

Forced Draught.—Assume 20 lbs. of air per pound coal, heated air temperature 200°, funnel gases temperature 550°.

Then, $550° - 200° = 350°$ rise of air temperature.

And, $350 \times 21 \times \cdot 23 = 1690{\cdot}5$ Heat units required per pound of coal.

NOTE.—20 lbs. of air + 1 lb. of coal = 21 lbs. of gases in all.

So that $3381 - 1690{\cdot}5 = 1690{\cdot}5$ Heat units saved per pound of coal burnt.

Howden's Forced Draught.—As a general rule the best results are obtained when the air pressure below the bars is equal to 1 inch water, and above the bars $\frac{5}{8}$ inch water, giving a difference of $\frac{3}{8}$ inch, and to obtain this it may be found necessary to build up the bridge by, say, two brick thickness more than that arranged for originally by the makers. If this alteration is made, the results will, in most cases, be found to be the best possible, both as regards combustion and the life of the boilers.

Pitting and Corrosion.—In boilers the general causes of corrosion are:—

1. Fatty acids from animal or vegetable oils, which are set free when the oil is decomposed by the heat. 2. Oxygen and CO_2 from air brought in with the feed water, and set free by the heat. 3. Galvanic action, due to the difference in composition of the metals used in the construction of the boiler, such as iron and steel, and to other similar causes.

The best oil for internal lubrication, or for rods, is mineral oil, which is a pure hydrocarbon, and free from acids.

Most of the oil used for internal lubrication of the engines finds its way to the boilers, by being brought with the steam into the condenser, and afterwards pumped into the boilers by the feed pumps.

Places Pitted.—The parts of a boiler where pitting occurs vary a great deal in different boilers, but the most common places are—(1) About the line of the fire-bars on the water side of the furnaces; (2) at the sides, bottom, and back of the combustion chamber; (3) at the back ends of the tubes, and at the combustion chamber end of the small stays, which are exposed to the high temperature gases.

Furnace Collapse by Retention of Gases.

In some cases the collapse of furnaces may be brought about as follows :—

If with forced draught the tubes become badly choked up with soot the outlets for the products of combustion are correspondingly reduced, and the gases being thus retained, may rise in temperature to a serious degree, and overheat the furnace metal ; if the temperature of the plates exceeds 600°, collapse of the crown will likely take place. Retarders placed in the tubes unfortunately tend to produce the choking up referred to, as also do the "Serve" type of boiler tube.

Collapse of Furnaces.—When a furnace crown is brought down by oil deposits, it often happens that after the boiler has been blown down and the furnace examined inside, no trace whatever can be discovered of the cause of collapse. This is accounted for by the fact that when overheating of the plate takes place, and consequent buckling, the intense heat resulting burns completely away the layer of oil, thus leaving no trace. The only clue to the true cause of the collapse lies in the fact that generally the metal is *cleaner* at the place where the oil had formerly lain than on other parts of the furnace metal.

Heating Surface and Grate Surface.

In cylindrical marine boilers the ratio of Heating to Grate surface is about 30 or 35 to 1, and in water-tube boilers from 40 to 1 upwards.

Boiler Repairs, Parts Corroded, &c.

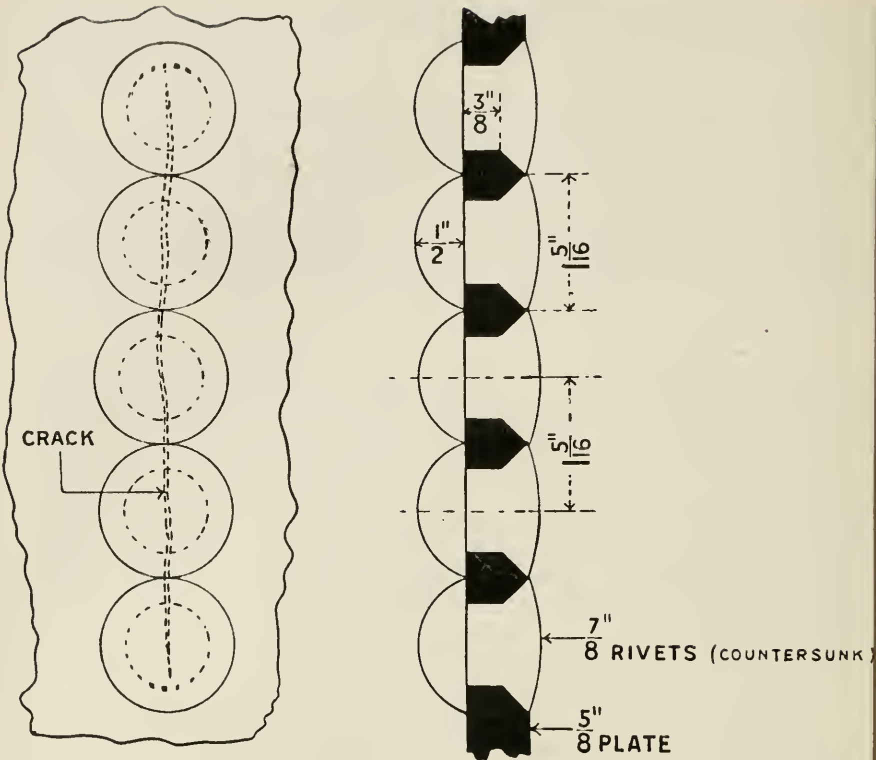
Chain Patch.—Cracks in furnaces and combustion chambers are often repaired by means of a chain patch, consisting of a series of pins tapped into the crack and into each other, the ends being riveted over as shown in the sketch, this method of repair being handy and suitable for small cracks.

Patches.—For a badly corroded section of a furnace or combustion chamber a riveted patch may be found necessary and should be arranged as follows :—

1. First cut out the defective piece of plate.
2. Shape the patch to template, and of a thickness about $\frac{1}{16}$ inch less than the furnace or combustion chamber it is to be fitted on to.
3. Employ rivets of a diameter determined as follows :—

RULE—

$$1{\cdot}2 \times \sqrt{\text{plate Thickness}} = \text{rivet diameter.}$$



No. 67.—Method of Repair for a Crack.

The repair consists of a series of rivets countersunk on one side, and just touching each other as shown. This method of repair for a longitudinal crack in a furnace or combustion chamber will be found, in most cases, to stand better than the chain patch obtained by screwed pins tapped half into each other and riveted over.

And of a pitch found as follows :—

RULE—

$$\frac{100 \times \text{rivet diameter}}{100 - \text{joint}} = \text{Pitch of rivets.}$$

It must be remembered that the joint strength referred to is equal to about 52 per cent. if single riveting is employed, and 68 to 70 per cent. if double riveting.

4. Rivet the patch on to the fire side of plate so that the effect of the heat will take place on the edges of the patch in place of the

edges of the furnace or combustion chamber metal. This will also be found more convenient for caulking.

5. The patch should form a metal to metal joint without any other jointing material.

6. The distance from edge of rivet hole to edge of plate must be equal to one rivet diameter, and the width of lap for single riveting will therefore be equal to three rivet diameters.

General Repairs.

If a combustion chamber shows a buckle between some of the stays, it is probably due to defective circulation, oil or scale deposits adhering to the plate and causing overheating. If the buckle is bad, tap a stay through it and the boiler plate between the other stays.

If a furnace or combustion chamber plate develops a blister, it is usually caused by the plate being laminated, which means that some dirt or sand has been rolled up with the metal during manufacture; the plate, not being solid throughout, blisters when heated.

If a piece of a furnace requires to be patched, first cut out the defective piece of the plate and then rivet on the patch (metal to metal) on the fire side of the plate. The thickness of patch should be from $\frac{3}{8}$ inch to $\frac{1}{2}$ inch, and the diameter of rivets about $\frac{7}{8}$ inch.

If a combustion chamber stay leaks badly, and cannot be kept tight, take out the stay and tap the holes to a larger diameter, then screw in a new stay.

NOTE.—If the plates cannot be tapped, a distance piece or thimble is necessary to form a joint, when a bolt is used instead of a screwed stay.

Stay Repair.—The following method of repair for a leaky combustion chamber riveted stay may be applied when a new stay of a slightly larger size cannot be obtained. Chip off the riveted head flush with the plate, bore a hole in the stay, say $\frac{3}{4}$ inch diameter, then drift out the hole to tighten up the threads in the plate, next tap the hole $\frac{7}{8}$ inch diameter and screw in a pin of that size, fitting a $\frac{3}{4}$-inch thick washer, with a joint of asbestos and red lead.

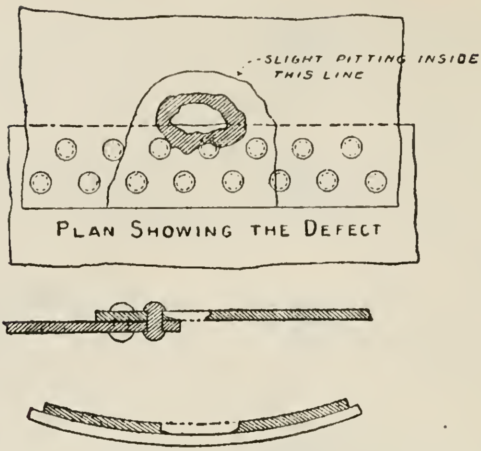
NOTE.—As the $\frac{7}{8}$-inch pin now takes the place of the stay, theoretically, the pressure should be reduced by rule to this size. In the proportion of the respective stay areas assume original stay to be $1\frac{1}{4}$ inches diameter and boiler pressure 180 lbs., then,

As, $1{\cdot}25^2 \times {\cdot}7854 : {\cdot}875^2 \times {\cdot}7854 :: 180$ lbs.

Or, as, $1{\cdot}25^2 : {\cdot}875^2 :: 180$ lbs.

Therefore, $\frac{{\cdot}875^2 \times 180}{1{\cdot}25^2} = 88$ lbs. Safe pressure.

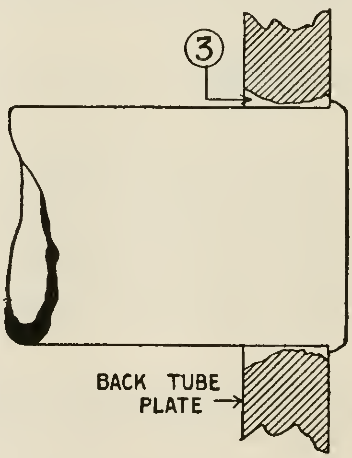
This of course neglects the holding power of the stay obtained by the expanding out of the metal, which may more or less make up for the loss of area, and permit of the original pressure being carried.



No. 68.—Pitting of Boiler Shell Plate.

The above sketch shows the effects produced by pitting on a boiler shell plate $\frac{31}{32}$ inch thick, and which resulted in an explosion.

The pitting appears to have originated in local galvanic action on a small area of the plate, due to impurity in manufacture, and this afterwards extended as shown above and blew out.



No. 69.—Tube Plate Corrosion.

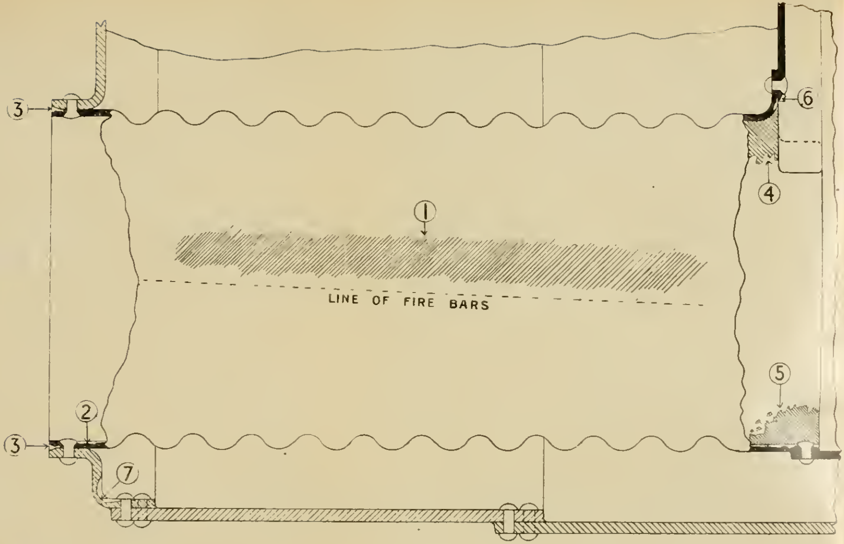
If a boiler tube leaks and the leakage is not checked at once by re-expanding, the tube plate is in danger of corroding as shown in the sketch.

This corrosion is caused by small quantities of water leaking through, which, becoming decomposed by the heat, sets free oxygen gas: the oxygen gas combines chemically with the tube metal to form iron oxide, which results in wasting of the metal.

No. 71.—Furnace Corrosion.

1, Corrosion due to heat setting free oxygen gas.

2, Corrosion due to moisture such as from wet ashes, &c. The heat sets free oxygen and CO as, which in combination are highly corrosive.



No. 72.—Corrosion on Furnaces, &c.

1, Corrosion above line of bars due to high temperature and liberation of free oxygen gas.
2, Corrosion due to moisture, such as from wet ashes.
3. Corrosion due to expansion and leakage, also repeated caulking of the plate edges.
4, Corrosion due to unequal expansion, straining, and leakage.
5, Corrosion due to straining of plates and faulty circulation.
6, Corrosion due to straining of plates, intense heat, and leakage.
7, Corrosion due to grease, and other deposits.

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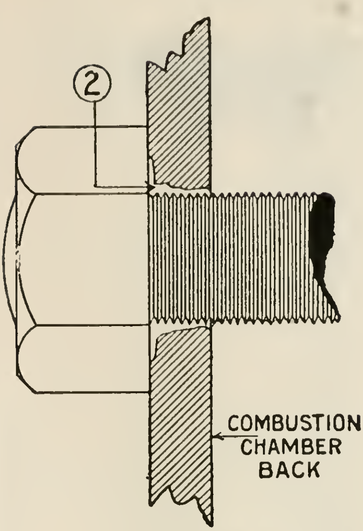
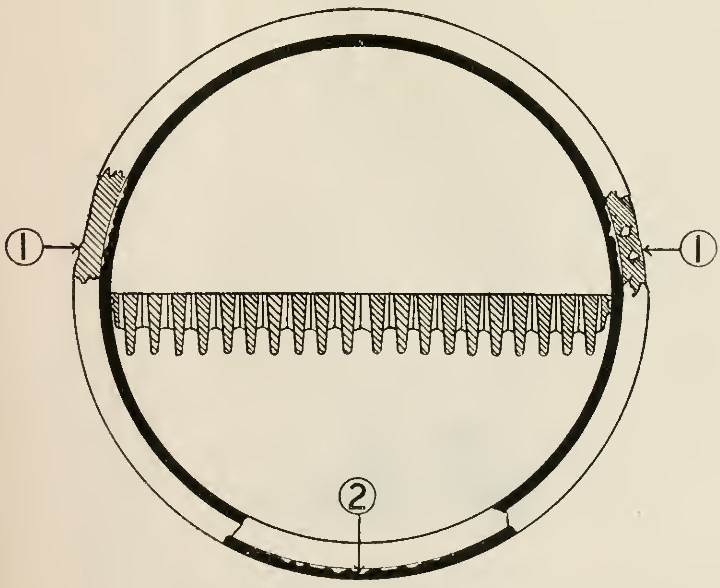


No. 70.—Combustion Chamber Plate Corrosion

If a stay leaks at the back end, corrosion may follow by the oxygen gas set free by the heat. This is shown at 2 in the sketch, and the repair would be to take out the stay and re-tap the holes to a larger size, screwing in a heavier stay.



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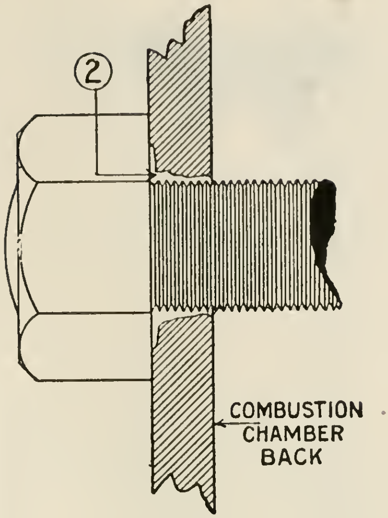

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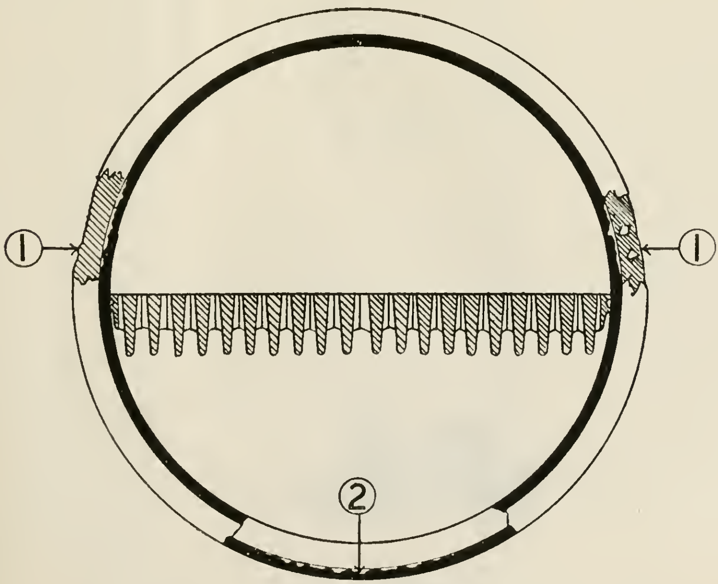

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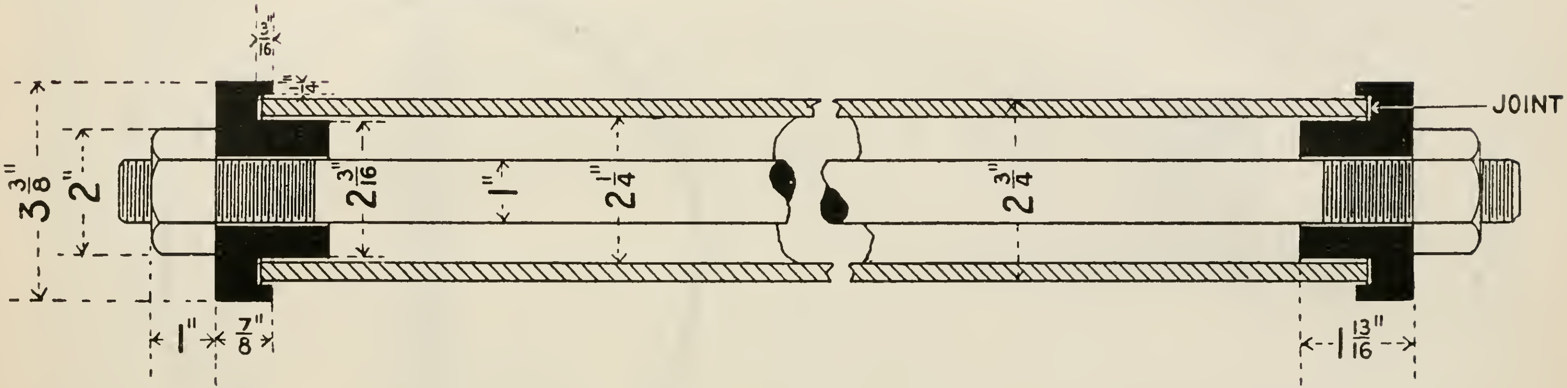
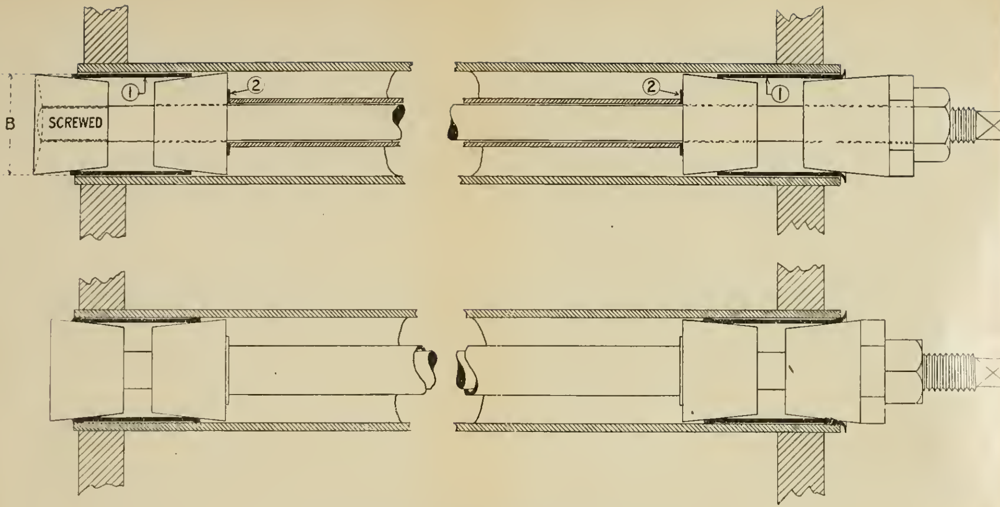


No. 73.—Permanent Type Tube Stopper.

The rod shown is screwed at the ends, and the caps are screwed up hard against the tube ends, a joint being made between the two. The objection to this pattern of stopper is that a man requires to enter the combustion chamber to fit and screw up the back end cap.



No. 74.—The Bagguley Patent "All-Metal" Tube Stopper.

1, 1, Soft metal sleeves. 2, 2, Copper washers.
B, Diameter of cones is equal to <u>outside</u> diameter of sleeves

When the nut is tightened up the cones press out the ends of the sleeves, thus forming four metal-to-metal joints.

The patent stopper shown above makes four metal-to-metal joints in the faulty tube, two at each end, and the whole operation is performed by means of screwing up a single nut at the front end of the tube, the stopper being placed in position from the front end.

The patent stopper consists of a long bolt which passes right through the boiler tube to be stopped; this bolt has a tapered head on the back end which exactly fits the tube. Three conical sleeves slip on to this bolt, making a tight fit externally on the bolt, and internally on the boiler tube. The front end of the bolt is screwed and fitted with a feather-way, and on this end a hexagon washer or nut is fitted with a feather to fit the above-mentioned feather-way. A screwed nut is then fitted to the bolt, and a malleable-iron tube is passed over the bolt, which keeps the cone pieces in position, and enables them to be tightened up simultaneously. Two soft metal sleeves, 1, 1, complete the arrangement.

In stopping a faulty tube the whole apparatus is passed through from the smoke-box end as one piece, and by holding the hexagon washer by means of a spanner and screwing up the nut the four taper pieces are screwed up simultaneously, expanding the soft metal sleeves and thus effectively stopping up the faulty tube, and cutting it out of action for any length of time.

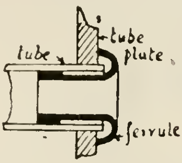
[*To face page* 133.

Iron Tubes and Stays.

In steel boilers the tubes and small stays are often made of iron, for the reason that iron corrodes less than steel under corrosive influences.

Leaky Tubes.

Owing to the tubes and tube plate expanding at different rates, the back ends of the tubes often leak. This is remedied by re-expanding, or by fitting into the tube at the back end a capped ferrule as used in the Navy, to keep the heat off the tube end and the plate (Sketch No. 75). Scale and soot on the tubes and plates tend to increase the leakage, and with forced draught, as the heat is more intense, the leakage is still further increased.



No. 75.—Capped Ferrule for Leaky Back Ends.

Collapsed Tubes.

For a collapsed tube the best repair is a permanent stopper (Sketch No. 73), formed of a long rod screwed at the ends, and cap washers fitting over the ends of the tube and screwed up tight with a joint between the tube and the washers. Patent stoppers are also employed to close up a cracked tube, the "Bagguley" type being shown in Sketch No. 74.

Safety Valves.

To find the Load on any valve, multiply the valve Area by the Pressure per square inch.

To find the Pressure per square inch, divide the Load on the valve by the valve Area.

EXAMPLE.—The Pressure is to be 40 lbs. per square inch and the valve is 5 inches diameter, find the Load.

$$5^2 \times \cdot 7854 \times 40 = 785 \cdot 4 \text{ lbs. load.}$$

EXAMPLE.—The Load on a dead-weight valve is 1000 lbs., and the valve is 3 inches diameter; find the blowing-off Pressure per square inch.

$$\frac{1000}{3^2 \times \cdot 7854} = 141 \cdot 4 \text{ lbs. pressure per sq. in.}$$

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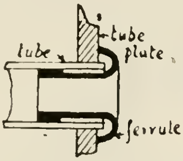

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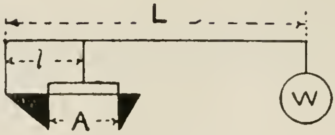
$$\frac{1000}{3^2 \times \cdot 7854} = 141 \cdot 4 \text{ lbs. pressure per sq. in.}$$

Lever Safety Valve.



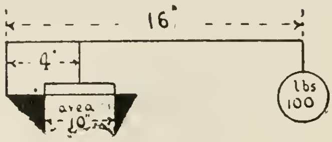
No. 76.

NOTE.—A = valve Area, load = A × boiler pressure.

Then, $L \times W = l \times \text{load}$.

Therefore, $\frac{L \times W}{l} = \text{load}$, and, $\frac{\text{load}}{A} = \text{pressure per sq. in.}$

Again, $\frac{l \times \text{load}}{L} = W$; or, $\frac{l \times \text{load}}{W} = L$.



No. 77.

EXAMPLE 1.—

RULE—

$$16 \text{ in.} \times 100 \text{ lbs.} = 4 \text{ in.} \times \text{Load.}$$

Then, $\frac{16 \text{ in.} \times 100 \text{ lbs.}}{4 \text{ in.}} = 400$ lbs. Load,

and Pressure per sq. in = Load ÷ Valve Area = 400 ÷ 10 = 40 lbs. per sq. in.

NOTE.—Valve area = 10 square inches.

In the foregoing case, the weight of the lever, and of the valve and spindle, are neglected. If they are to be allowed for, the following data must be given:—

1. Centre of gravity of the lever.
2. Weight of the lever.
3. Weight of the valve and spindle.

EXAMPLE 2.—In the previous case the lever weighs 7 lbs., its centre of gravity is 12 inches from the fulcrum, and the valve and spindle weigh 5 lbs.; find the load on the valve, and the pressure per square inch, allowing for the lever and the valve and spindle.

$\frac{12 \text{ in.} \times 7 \text{ lbs.}}{4 \text{ in.}} = 21$ lbs. more due to lever.

Then, $400 + 21 + 5 = 426$ lbs. load,

and, $\frac{426}{10 \text{ in.}} = 42{\cdot}6$ lbs. pressure per sq. in.

NOTE.—The centre of gravity is the point at which the lever balances if placed on a knife edge. As the weight of the valve and spindle is direct weight, it is simply added to the other two loads.

EXAMPLE 3.—Fulcrum to Weight 30 inches, fulcrum to Valve 5 inches, Load on valve 300 lbs.; find Weight.

Then, $L \times W = l \times \text{load} = 30 \times W = 5 \times 300.$

Therefore, $W = \frac{5 \times 300}{30} = 50 \text{ lbs.} = W.$

EXAMPLE 4.—Fulcrum to valve 6 inches, Weight 20 lbs., boiler pressure 30 lbs. per square inch, valve area 4 square inches; find the Length from fulcrum to Weight.

Then, $L \times W = l \times \text{load} = L \times 20 = 6 \times 120.$

NOTE.—$4 \times 30 = 120$ lbs. load on valve.

Therefore, $L = \frac{6 \times 120}{20} = 36$ in.

EXAMPLE 5.—Fulcrum to valve 6 inches, fulcrum to Weight 32 inches, Weight 25 lbs.; find the boiler pressure per square inch if the valve is 3 inches diameter.

Then, $L \times W = l \times \text{load} = 32 \times 25 = 6 \times \text{load}.$

Therefore, $\text{load} = \frac{32 \times 25}{6} = 133{\cdot}3$ lbs.

Then, Pressure = load ÷ valve area $= 133{\cdot}3 \div 3^2 \times {\cdot}7854 = 18{\cdot}8$ lbs. per sq. in.

Spring Safety Valves.

At 60 lbs. gauge pressure, the Board of Trade allowance of safety valve area is $\frac{1}{2}$ square inch per square foot of fire-grate surface.

At higher pressures the valve area required is less, because high-pressure steam has less volume than low-pressure steam; and at lower pressures the valve area required is more.

To find the valve area per square foot of grate for, say, 160 lbs. gauge pressure, $60 + 15 = 75$ lbs. gross, and $160 + 15 = 175$ lbs. gross.

Then, as 175 lbs. : 75 lbs. : : ·5 in. = ·214 of a square inch.

NOTE.—In working out valve areas, the gross pressure must be taken.

The lip cast round the safety valve face is to give an increase of valve surface when the valve lifts, so that the extra compression of the spring, due to the lift, may be neutralised. Without this fitting, the boiler pressure would increase with the valve lift.

To find the Compression.

RULE—

$$\frac{\text{Load on valve} \times \text{Spring mean diameter}^3 \times \text{Number of coils}}{2000000 \times \text{steel}} = \text{Compression.}$$

EXAMPLE.—Find the compression required for a 4-inch safety valve, pressure 160 lbs., and mean diameter of spring $3\frac{1}{2}$ inches: there are thirteen coils of square steel of $\frac{3}{4}$ inch side.

$$\frac{4^2 \times \cdot 7854 \times 160 \times 3\cdot 5 \times 3\cdot 5 \times 3\cdot 5 \times 13}{2000000 \times \cdot 75 \times \cdot 75 \times \cdot 75 \times \cdot 75} = 1\cdot 77 \text{ in. compression.}$$

The pressure per square inch varies as the compression of the spring.

EXAMPLE.—The pressure is 160 lbs. per square inch, and the compression is 2 inches; find the compression and thickness of the washer required to be put in to have the valve blowing off at 150 lbs. per square inch.

As 160 : 2 in. : : 150 = 1·875 in. compression.

Then, 2 in. − 1·875 in. = ·125 of an inch thickness of washer to go in under the compression nut.

NOTE.—The compression varies directly as the pressure.

To find the Diameter of Safety Valve.

$$\sqrt{\frac{\text{Square feet of grate} \times 37\cdot 5}{\text{gross pressure} \times \cdot 7854}} = \text{diameter of valve.}$$

NOTE.—If for forced draught allow about 25 per cent. more area of valve.

The Constant 37·5 is obtained by multiplying the valve area allowed at 60 lbs. gauge pressure by the gross pressure corresponding to it.

Thus, 60 + 15 = ·75, and 75 × ·5 = 37·5.

Superheated Steam.

Of late years a decided reaction has set in among marine engineers in favour of superheated steam, which, as proved conclusively by recent exhaustive experiments, possesses undoubted advantages, and the use of which results in considerable economy. It has been found that with superheating to the extent of 50° the gain is about 8 per cent., and if the superheat is increased to 200° above the natural pressure temperature of the steam, then the resulting economy is about 30 per cent.

It should be noted that if saturated steam—that is, steam drawn from a boiler—is raised in temperature and the *pressure* kept constant, the volume increases; this means a larger volume of steam produced for the same amount of water evaporated, and therefore less boiler space required. Again, by superheating steam which originally contains water, the water is evaporated, thus giving drier steam, which results in less cylinder condensation losses, and less transfer of heat to the cylinder walls. It will thus be seen that the advantages of superheated steam are undoubted.

NOTE.—The volume varies with the absolute temperature if the pressure is kept constant.

EXAMPLE.—Boiler steam (saturated) at a pressure of 180 lbs. gauge pressure, temperature of 380°, and specific volume 2·31 cubic feet; find the volume if the steam is superheated 100° Fahr.

Then, $380° + 100 = 480°$ steam temperature when superheated.

And, $461 =$ absolute temperature constant.

Therefore, as, $(380° + 461°) : (480° + 461°) :: 2·31 = 2·58$ cubic feet volume.

So that the volume of the steam per pound is now increased by $2·58 - 2·31 = ·27$ of a cubic foot.

In addition to this it must be remembered that the temperature is higher, and the steam of a drier condition.

Advantages of Superheated Steam.

(1.) Increase of steam volume.
(2.) Dry steam enters the cylinders, and less condensation losses result.
(3.) Less leakage of steam past valves and pistons.
(4.) Less danger from water hammer in main steam pipes or chests.

Against these gains there are, however, certain disadvantages, which also require to be taken into account.

Disadvantages of Superheated Steam.

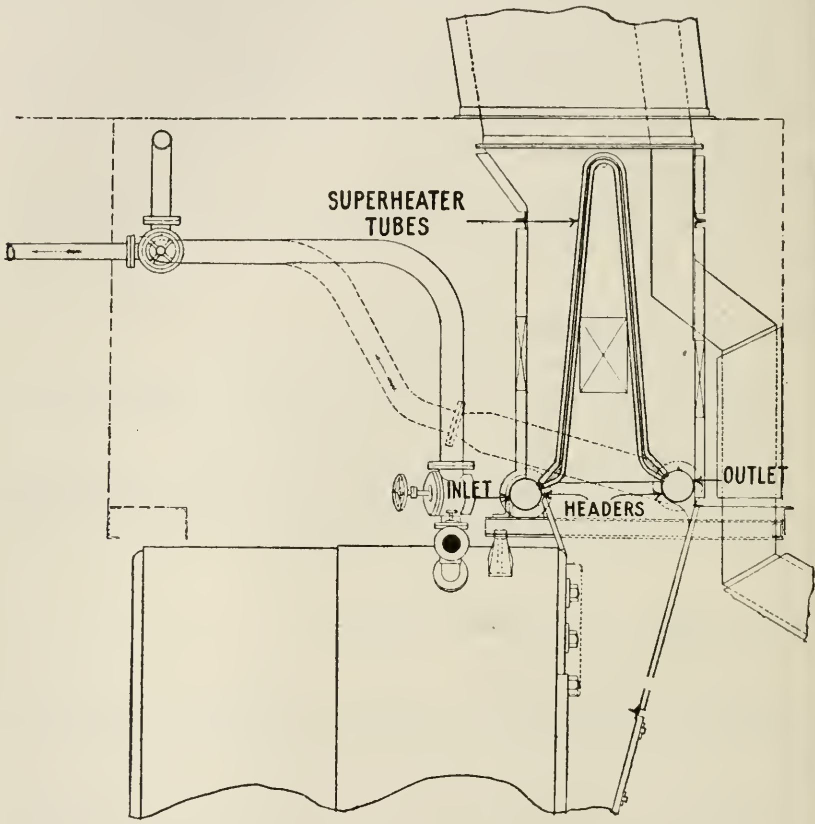
(1.) Difficulty of lubrication.
(2.) Piston rings more easily broken.
(3.) Trouble experienced in keeping superheater coils or tubes tight and in good working order.

Methods of Superheating.—In marine practice the ordinary method has been to utilise the funnel gases in superheating the steam, although the independently fired type as devised by Professor Watkinson has also been fitted in some steamers.

The Watkinson superheater (Sketch No. 78) consists of a series of U-shaped mild steel tubes, expanded into large headers similar to those employed in water-tube boilers. This nest of tubes is fitted in the uptake, and the steam from the boiler enters one of the headers, and passing through the U-shaped tubes becomes superheated by the furnace gases which are flowing over the tubes. The steam then enters the other header and flows along the steam pipe to the engine.

Suitable drainage arrangements are fitted to keep the drums and tubes clear of water, and by suitable bye-pass valves and pipes the steam may, if required, pass direct from the boilers to the engine without entering the superheater coils.

When it is stated that the average loss due to initial cylinder condensation is about 15 per cent. with saturated steam, the advantage of superheated or dry steam will be apparent, as less water being present



No. 78.—"Watkinson" Type Marine Superheater.

in the steam the condensation losses are greatly reduced, and may be practically eliminated.

To overcome the lubrication difficulty, special high temperature mineral oils are now manufactured by the various oil companies, which are said to resist the disintegrating effects of the superheated steam, and allow of suitable lubrication of rods, pistons, and valves.

Regarding the resulting increase of volume due to superheating Professor Watkinson says: "During superheating, although the pressure of the steam remains constant, its volume is greatly increased. The amount of heat required to superheat 1 lb. of steam by 150° Fahr. is 72 British heat units, which is only about 6 per cent. of the heat required to generate 1 lb. of dry saturated steam. The increase in volume due to this additional 6 per cent. of heat averages about 30 per cent."

Steam Pipes.

Steam pipes are made of the following materials :—

1. Copper, seamless or brazed.
2. Wrought iron, generally lap welded.
3. Steel, also lap welded.

Sometimes in steel pipes a riveted butt strap is fitted covering the weld, and copper pipes are further strengthened by being covered with wrappings of wire rope, or by iron bands secured at short distances along the pipe. Cast iron also, in a few cases, has been used for steam pipes.

The three principal causes of recent accidents to steam pipes were—(1) Insufficient allowance for expansion; (2) defective drainage arrangements; and (3) vibration.

Before opening the main stop-valves the drains on the pipes or chests should be opened to clear away all water which may have collected in them.

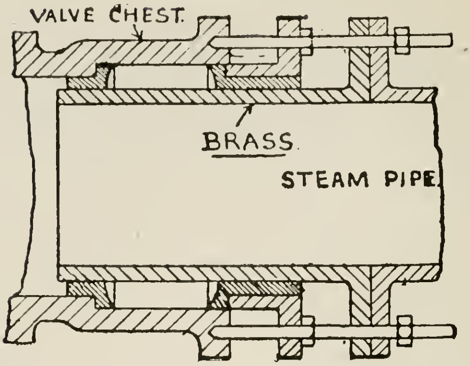
When steam pressure strikes a body of water it imparts to the water a velocity nearly equal to its own, and the resulting force acquired is so great that the chest or pipe may be burst: this is known as "water hammer," and accounts for many of the accidents to steam pipes and stop-valve chests; hence the necessity for drain cocks being fitted and used.

Water Hammer in Steam Pipes.—Whenever possible water lodging in steam pipes should be drained out when the steam pressure is off, otherwise the draining out of the water may result in the setting up of water hammer action with danger of bursting the pipe or valve chest.

According to Mr C. E. Stromeyer of the Manchester Steam Users Association, the conditions favourable to water hammer are as follows :—

1. Water in contact with steam.
2. Rapid condensation in pipes or chests.
3. Agitated water surfaces.
4. Steam pressure at one part of a pipe and vacuum at another part.

Under ordinary practical conditions the pressure per square inch on a pipe produced by water hammer may range from 250 lbs. to 300 lbs., or even more.



No. 79.—Steam Pipe Expansion Joint.

NOTE.—When no expansion joints are fitted on steam pipes, bends are formed on the pipes to allow for expansion.

Circulation and Priming.

Circulation in a boiler is the rising of the heated and expanded water, and the sinking of the colder and heavier water to take its place, resulting in a continuous current passing from the bottom upwards, and from the top downwards.

When water is heated it becomes lighter, and expanding, rises to the top in the form of small steam bubbles, which, on reaching the surface, burst and give off a small amount of steam, and, if the boiler is properly designed, the colder water being heavy falls, and in its turn becomes heated.

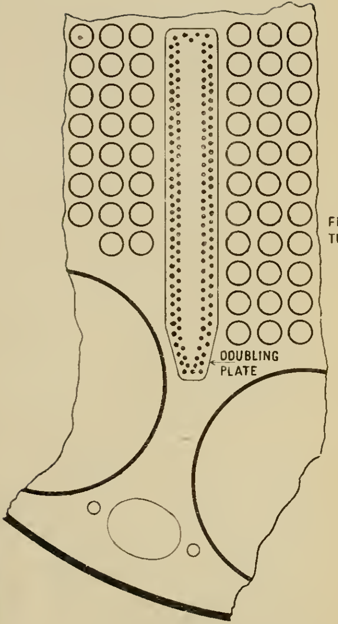
Should insufficient allowance be made for the circulation, or anything occur to check it, priming will most likely begin, as priming is caused by bad circulation.

Defective circulation may be caused by:—

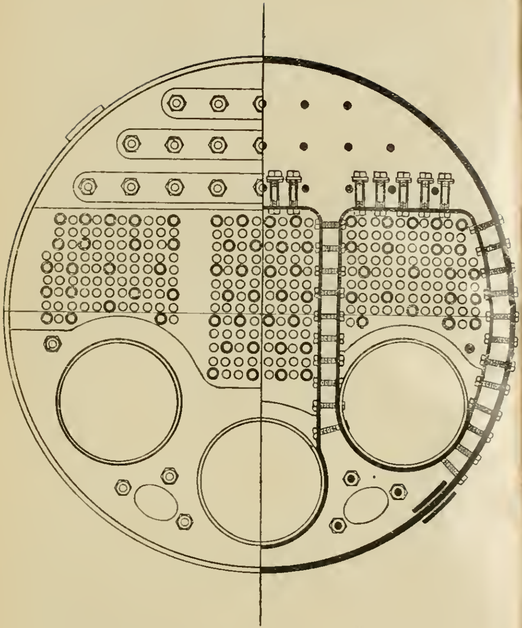
1. Close arrangement of tube nests.
2. Small steam space.
3. Dirty water.
4. Bad firing.

Observe that any of the foregoing causes bring about bad circulation, and consequently priming.

The bottom seams of the boiler sometimes leak owing to the difference of temperature existing between the top and bottom caused by defective circulation. The top plates being hotter, and expanding more in proportion, tend to drag open the bottom seams, and leakage is the result.



No. 81.—"Doubling" Plate.



No. 82.—End View of Boiler Half in Section.

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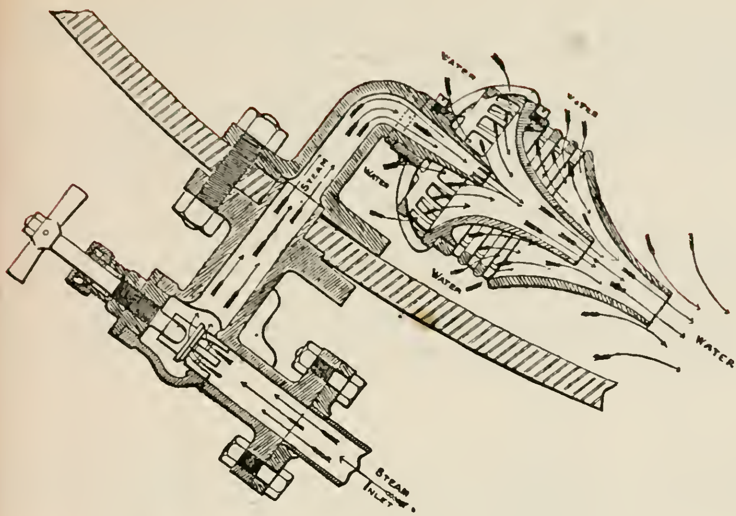


No. 80.— Weir's Patent Hydrokineter.

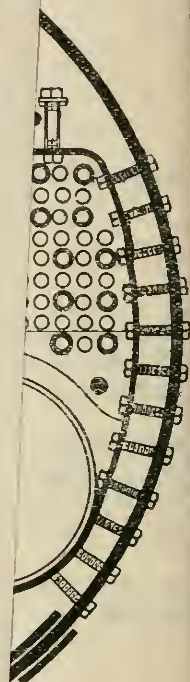
In getting up steam, circulation is assisted by means of a hydrokineter (Sketch No. 80), or by the donkey connection for pumping the water from one part of the boiler to another (see page 151).

Doubling Plates.

Doubling plates, as shown in Sketch No. 81, are sometimes fitted to boilers to increase the strength of flat surfaces in places where ordinary stays or stay tubes cannot be fitted. The plate shown is about $\frac{7}{8}$ inch thick, and is riveted to the boiler end plates in the spaces between the tubes. Stays are not admissible at this part of the boiler owing to the necessity for keeping clear the spaces between the tubes to allow of the furnaces being examined, cleaned, or repaired, so that the only alternative method of support is by means of doubling plates.

End View (Sketch No. 82).

The sketch shows the general construction of a modern high-pressure boiler, as seen half in section from the end. Observe how the combustion chamber tops are supported by the "dogs," or girders, and stays passing through which are *tapped* into the top plates.



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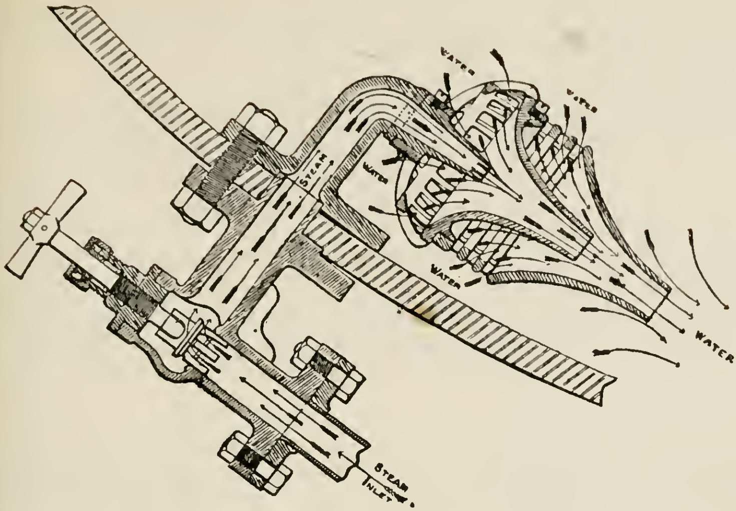

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The following are the principal dimensions to be carefully noted :—

Pitch of girders, about 8 inches.
Depth ,, ,, 9 inches.
Distance between combustion chamber plates, about 7 inches.
,, ,, combustion chamber and boiler shell, about 7 inches.
,, ,, furnace and bottom of boiler, about 7 inches.
,, ,, bottom row of tubes and furnace top, about 10½ inches.
,, ,, each nest of tubes, about 11 inches.

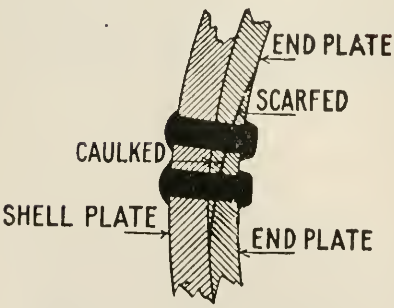
Observe how the plate at the sludge hole openings is strengthened by means of the stays shown, which pass from end to end of the boiler, the top one being heavier than the lower two.

These stays are usually about 1¾ inches or 2 inches diameter.

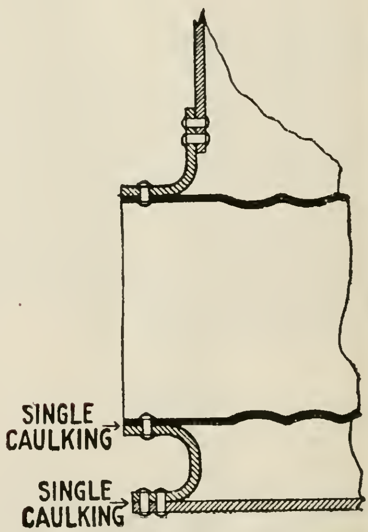
Also note the arrangement of the stay tubes, which are indicated by dark circles.

NOTE.—The manhole opening in the shell (about 16 inches × 12 inches) is strengthened by means of a doubling or "compensating" ring riveted to the plate round the hole. The effective surface area of this ring should not be less than the area of the metal cut away to form the opening.

Scarfed Joints.—This type of joint is used in boilers where three plates overlap each other, as for example when two end plates and the shell overlap. Two of the plates are scarfed or thinned down, as shown in the sketch, and the third plate covers both ; this reduces the thickness to that of only two plates. Extra heavy caulking is required to keep the joint tight.



No. 83.—Scarfed Joint.



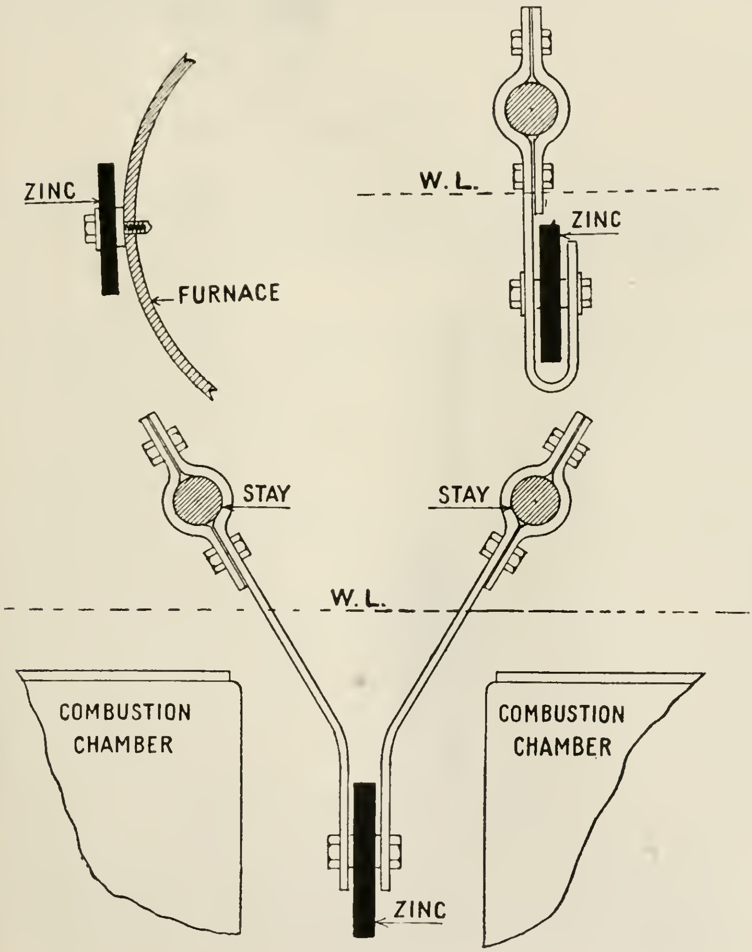
No. 84.—Flanged-out Plates.

Flanged-out Plates.—Sometimes one end of the boiler is flanged outwards, as shown, to allow of the convenience of the hydraulic riveting machine. The disadvantage of this arrangement lies in the fact that only single caulking is possible, whereas with the plates flanged inwards double caulking can be employed.

Zinc Plates.

The sketches show the usual methods of connecting the zinc slabs to the boiler metal, and it is important that the following points be attended to:—

(1) Proper metallic contact between the zinc and the boiler, and to ensure this the surfaces in contact should be filed up bright.



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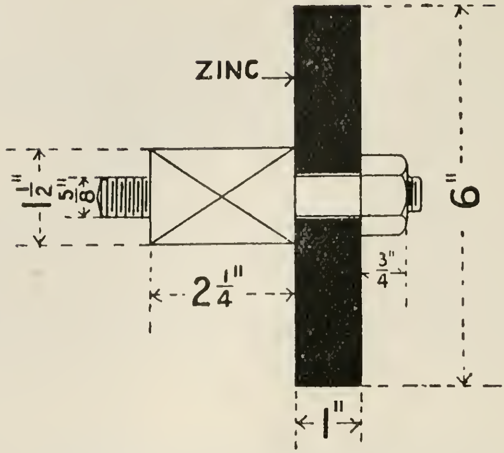
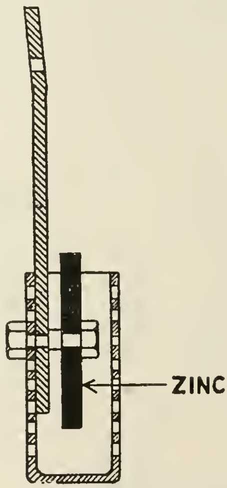


No. 86.—Zinc Block and Stud.

The zinc plates fitted into boilers to set up galvanic action and reduce corrosion are usually fixed to the furnaces, combustion chambers, and end plates as shown, the stud being screwed in to form effective metallic contact.



No. 87.—Zinc Plate in Box.

In water-tube boilers the zinc plates are often carried in perforated boxes, the idea being to prevent the oxide of zinc (produced by the galvanic action) from being distributed over the boiler. The oxide drops to the bottom of the box, and is thus kept by itself.

(2) When convenient the removal of scale or oily deposits from the zinc plate connections, as these substances insulate the parts and interrupt the flow of the current.

(3) The renewal of the zinc slabs whenever they are found to have become spongy, as the galvanic action is then practically exhausted.

When the zinc is suitably connected as above described, the chemical action which occurs results in the formation of Oxide of Zinc and the liberation of bubbles of Hydrogen Gas. At the zinc plate Oxygen is set free and combines with the zinc, and at the boiler metal Hydrogen Gas is set free.

NOTE.—As the proportion of zinc used is very small compared with the amount of boiler metal to be protected, the corrosion is merely reduced in extent, and is not prevented altogether.

The usual allowance of zinc is about one square foot, inch thick per each 80 I.H.P.

Water Gauge.

In Fig. 1 the column shown is hollow cast, so that the water or the steam could pass through it: the test cocks show this.

NOTE.—Open the drain and blow through, then close the drain and see if the water rises to the working level; if so, the connections are all clear; if, however, no water shows, then either C or D is choked or the water is too low; if, on the other hand, the glass shows full, then either A or B is choked or the water is too high.

To test if the steam connections are clear, shut cocks C and D, and have open cocks A, B, and the drain cock E. If steam blows through the cocks are clear.

To test if the water connections are clear, shut cocks A and B, and have open cocks D, C, and the drain cock E. If water blows through, the cocks are clear.

If cock A or cock B is choked, the glass will show full up. It will show the same if the pipe between A and B is choked.

If cock C or cock D is choked, the glass will continue to show what the water level was at the time the cocks stuck, as the water will be shut off from the boiler altogether. If the drain is opened and shut, the glass will show empty as long as the cocks remain choked. The same thing will happen if the pipe between C and D is choked.

If the glass is showing full water, due to the cock A, or the cock B, having got choked, to test if it is A, shut D and B and blow through A, C, and E; if steam blows out A is clear, if not, A is stuck. To test B, shut A and C and blow through D, B, and E; if water blows out, B is clear, if not, B is choked.

If one of the two cocks C and D is choked, to find which it is, shut A and C and blow through D B and the drain E; if water blows out,

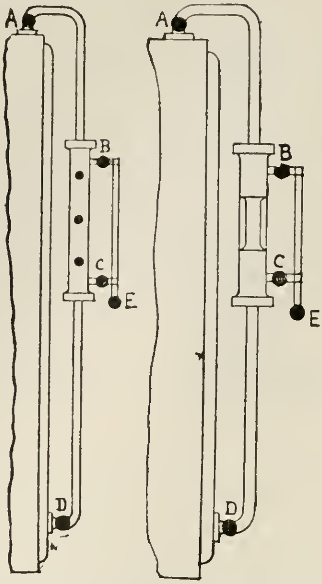


Fig. 1. Fig. 2.

No. 88.

D is clear, if not, D is choked. To test C, shut D and B and blow through A, C, and E; if steam blows out, C is clear, if not, C is choked.

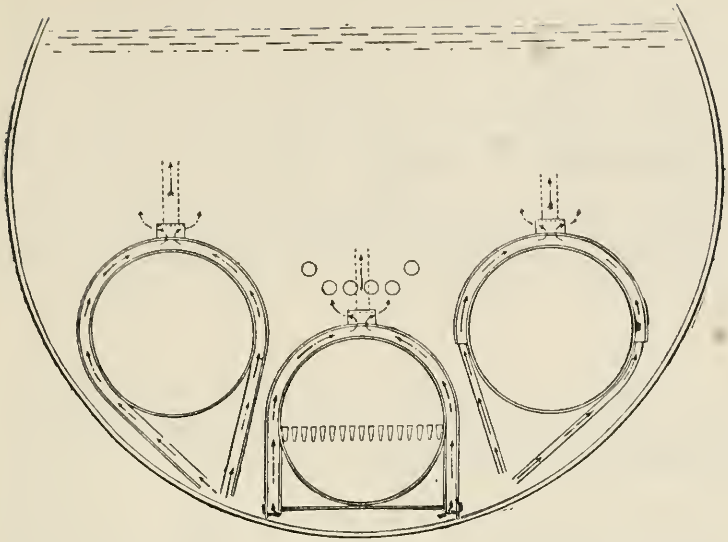
NOTE.—In testing, before closing any of the cocks A, B, C, or D, it is advisable to first open the drain cock E, so that shocks on the pipes and glass may be avoided.

If cocks A and B are closed and the others left open and the glass blown through, if water comes, the water connections are clear: if, next, cocks C and D are closed and the glass blown through with cocks A and B open so that steam comes, the steam connections are clear; but if, on shutting the drain cock, no water shows in the glass, this proves that the water level in the boiler is too low, as it must be lower than the bottom nut of the gauge glass, otherwise the glass would show water.

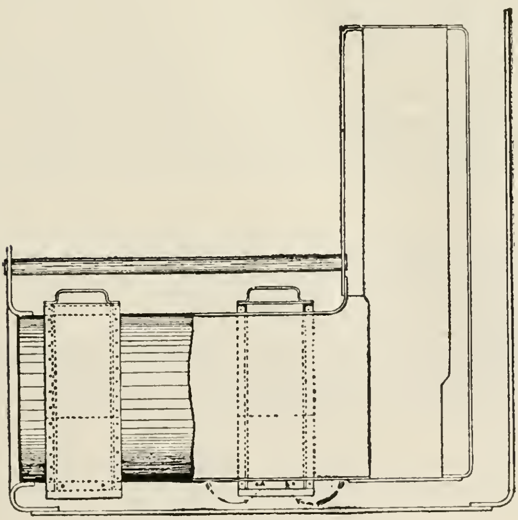
If the gauge column is cast solid as shown in Fig. 2, then the water or steam could not pass through it, and to test the water and the steam, single shutting off on the column is sufficient.

To test the steam side, shut cock C, and leave cocks A, B, and the drain cock E open; if steam blows out, the connection is clear. To test the water side, shut cock B and blow through cocks D, C, and the drain cock E; if water blows out, the connection is clear.

The glass usually shows from $1\frac{1}{2}$ inches to 2 inches less than the boiler level, the reason for this being that the water in the glass is colder than the water in the boiler, and, as water contracts in cooling, the level is lower in proportion.



No. 88a.



No. 88b.

Tait's Patent Water Circulator, showing its Application to Marine Boilers.

Tait's Patent Water Circulator.

The inner side of each plate is constructed with projections which fit closely into the corrugations of the furnace, so as to be as nearly as possible water-tight, and thus secure the maximum of rapid heating.

The lower part of the circulator projects beyond the body of the boiler as low as practicable, in order to reach the water at its coldest temperature in the bottom.

Over the top of the furnace the two sides of the circulator are held in position by a plate which is raised some 3″ above, and serves to break the rise of the superheated water, throwing it out on both sides.

The use of this invention secures a steady and continuous circulation of the water from the bottom of boilers when under fire. The result is attained by controlling the rapid convection currents which are created by the superheated water, so that a continuous inflow is being drawn up from the coldest level of the water, below the furnace, instead of being allowed to flow in from the nearest point.

The circulator really forms a saddle boiler, which is placed over the boiler at the hottest part of the furnace.

Its width is limited by the size of the manhole, generally running from 11″ to 15″ over all, according to circumstances and the type of furnace.

Boiling Points and Steam Temperatures.

Fresh water boils at a temperature of 212° under the atmospheric pressure.

In a good vacuum water will boil at a temperature of 90°, while under a pressure of 160 lbs. gross, the boiling point of fresh water would be 370°. Therefore the boiling point depends on the pressure on the surface of the water and varies accordingly, being high for high pressures, and low for low pressures.

At	15 lbs. atmospheric pressure		the temperature		is	212°	Fahr.
,,	60 lbs. gauge		,,	,,	,,	307°	,,
,,	80	,,	,,	,,	,,	324°	,,
,,	100	,,	,,	,,	,,	338°	,,
,,	150	,,	,,	,,	,,	366°	,,
,,	160	,,	,,	,,	,,	370°	,,
,,	180	,,	,,	,,	,,	380°	,,
,,	200	,,	,,	,,	,,	388°	,,
,,	250	,,	,,	,,	,,	405°	,,

NOTE.—The temperature of the water is the same as that of the steam.

Salinometer, Density, &c.

The salinometer measures the *density* of the boiler water.

The salinometer does not in all cases measure the *amount* of salt in the water.

If we take 32 lbs. of sea water and boil off all the water, about 1 lb. of solid matter will be left behind, hence the figure $\frac{1}{32}$.

A gallon of water weighs about 10 lbs., and 10 lbs. $\times$ 16 oz. $=$ 160 oz., therefore $160 \div 32 = 5$ oz. of solid matter per gallon.

Of the 5 oz. of solid matter in 1 gal. of sea water, fully 4 oz. are salt and the rest lime, &c.

The temperature marked on the salinometer is 200° Fahr., and if the water cools down below this, the salinometer will show more density in proportion to the drop of temperature, because water contracts in volume when cooling down to 39° Fahr.

For every 10° less temperature, allow the salinometer to be showing $\frac{3}{4}$ oz. more than the actual density.

Thus, suppose the water cools to 160°, that is 40° lower than it should be, then the salinometer would indicate about 3 oz. in excess of the real density, which amount would require to be subtracted from that shown by the salinometer to obtain the correct density.

Water is at its least volume at 39°, and expands if heated above this temperature, or if cooled below it.

Ordinary sea water at a temperature of 50° will show on the salinometer as fully 13 oz. density.

NOTE.—Salt and sugar will show on the salinometer as density, as these substances enter into solution.

To make a rough salinometer, take a long, narrow bottle, weight it and cork it; take a gallon of fresh water and heat it to 200°; put in the bottle and mark where it floats 0, or fresh water: next take a gallon of sea water, heat it to 200°, and mark where the bottle floats $\frac{1}{32}$, boil down the water to half a gallon, and when it cools down to 200° put in the bottle a third time, and mark where it floats $\frac{2}{32}$, or 10 oz.; other densities may be set off by evaporating the water away to one-third and one-fourth the original quantity.

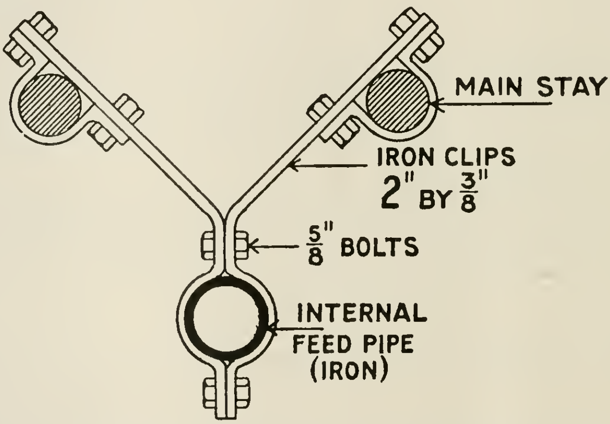
To test the density without a salinometer, draw off some of the boiler water and boil it over a fire; when boiling, put in the thermometer and observe the temperature; the density may then be roughly found by means of the following table:—

Fresh water boils at		212°
5 oz. density	,,	213·2°
10 ,, ,,	,,	214·4°
15 ,, ,,	,,	215·6°
20 ,, ,,	,,	217·8°

Allow 216° as the limit temperature, corresponding to about 15 oz. density: should the boiling point exceed this, surface the boiler to keep down the density. *

NOTE.—The above boiling points correspond to a barometer height of 30 inches. The boiling point varies with the barometer, up and down; for every inch difference in the barometer, the boiling point varies 1·5° in temperature.

If the boiler retains a density of 35 oz. (saturation point) and more feed water containing salt is supplied, then when the water fed in is evaporated, the salt contained in it deposits.

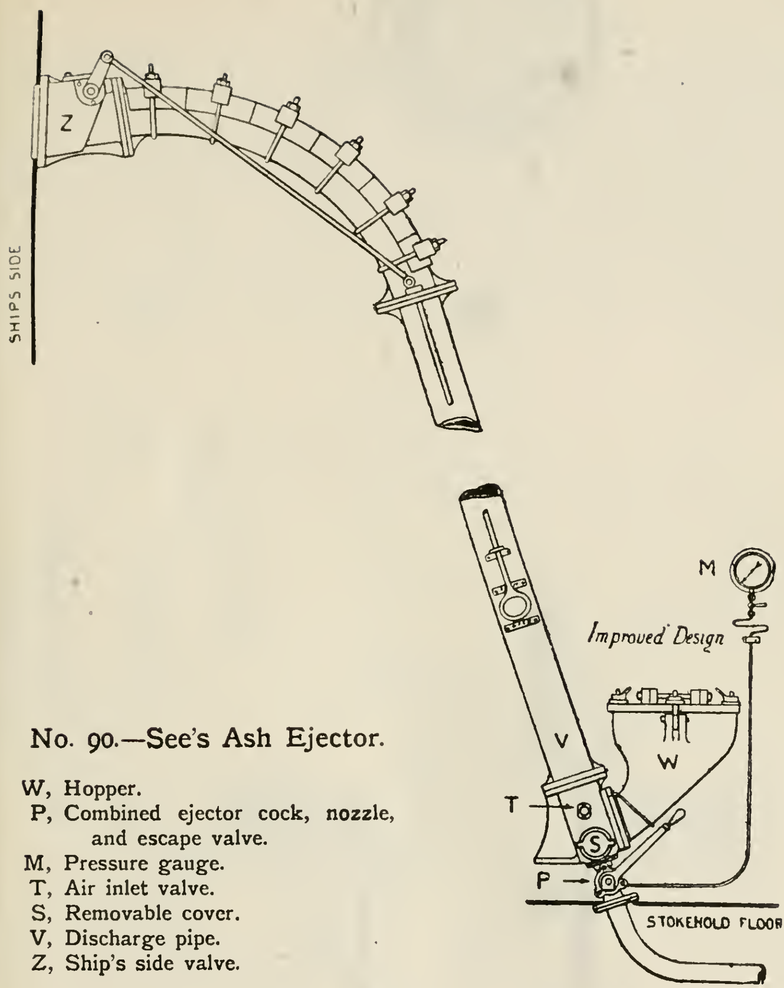


No. 89.—**Method of Supporting Internal Feed Pipes.**

See's Ash Ejector.

See's ash ejector consists of a cast-iron hopper with a grating for limiting the size of the ash or clinker put in, a water nozzle and pipe led up to the ship's side above the water level, and a cock to regulate the flow. A high-pressure donkey pump connection is also required

*** NOTE.—For accuracy, it is advisable to first boil fresh water and note the boiling point, then allow 1·2 degrees per 5 oz. for the boiler water boiling point excess.**



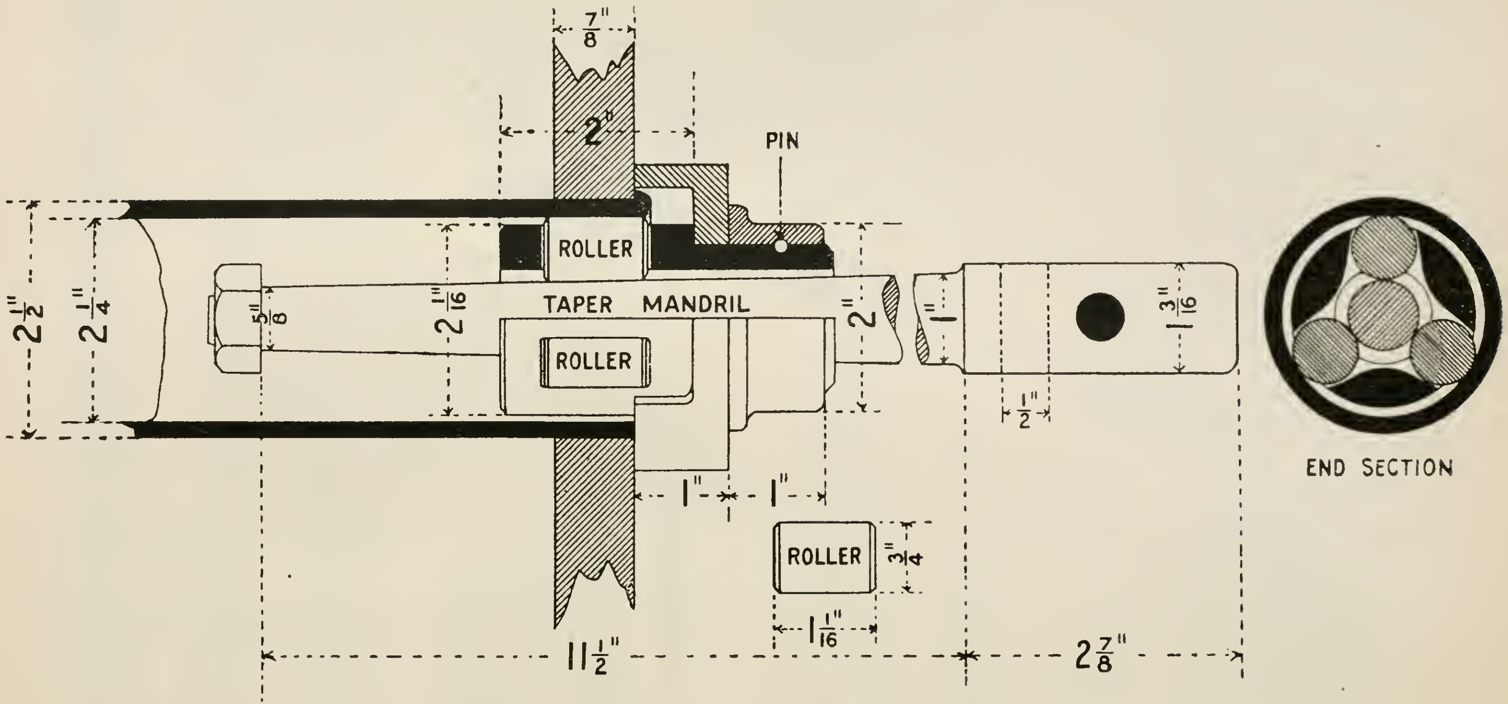
No. 90.—See's Ash Ejector.

W, Hopper.
P, Combined ejector cock, nozzle, and escape valve.
M, Pressure gauge.
T, Air inlet valve.
S, Removable cover.
V, Discharge pipe.
Z, Ship's side valve.

to obtain the necessary force of water pressure. Before opening the ejection cock it is necessary that the water pressure of the pump be not less than 200 lbs. per square inch, as shown by the gauge, also that the valve on the ship's side is full open. When the cock is opened the rush of water at high pressure past the grating carries with it the ashes shovelled through, and discharges them overboard. A small air valve is fitted on the pipe, and this must be kept open when working the ejector.

Tube Expander (Sketch No. 91).

The expander consists of a built-up case containing three rollers which project through corresponding openings cut in the shell of the

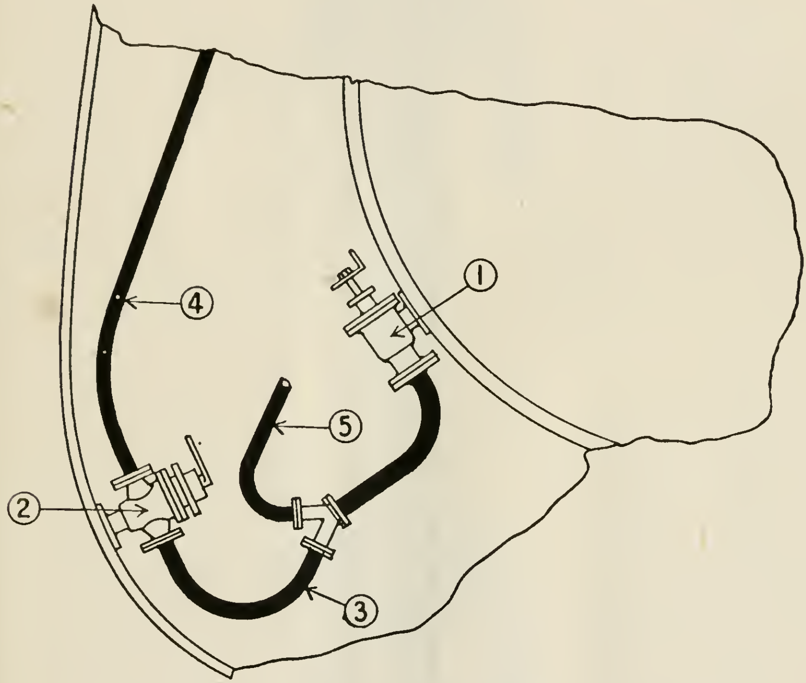


No. 91.—Tube Expander.

(With Dimensions.)

case. A taper mandril fits into the centre space of the rollers, and, on being knocked in with a hammer and revolved by a bar at the end, forces out the rollers against the tube and tube plate, thus forming a steam and water tight joint. In adjusting the position of the expander care should be taken that the rollers are in line with the tube plate, as otherwise the tube may leak, and, in addition, fracture may result owing to the tube being unduly stressed by the rollers acting at the wrong position.

All boiler tubes are expanded as described, with the exception of the heaviest pattern of stay tubes, which are caulked in, the expander being in this case too light for the work.



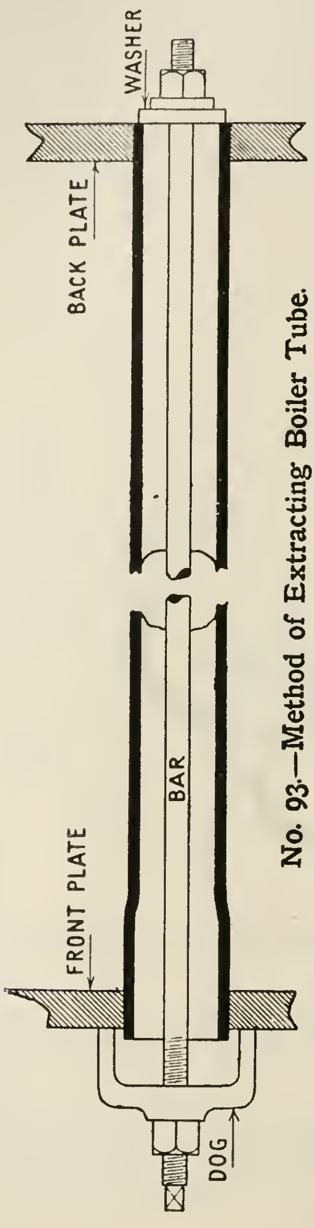
No. 92.—Blow-off and Circulating Connections.

1, Bottom blow-off valve.
2, Ship's side blow-off cock (two-way).
3, Bottom blow-off pipe.
4, Surface blow-off pipe.
5, Donkey suction pipe for circulating.

The above is a common arrangement of surface and bottom blow-off combined with the donkey circulating connection. Pipe 5 leads to the donkey pump suction valve, and the colder water lying at the bottom of the boiler can thus be pumped out, and discharged back into the boiler through the check valve.

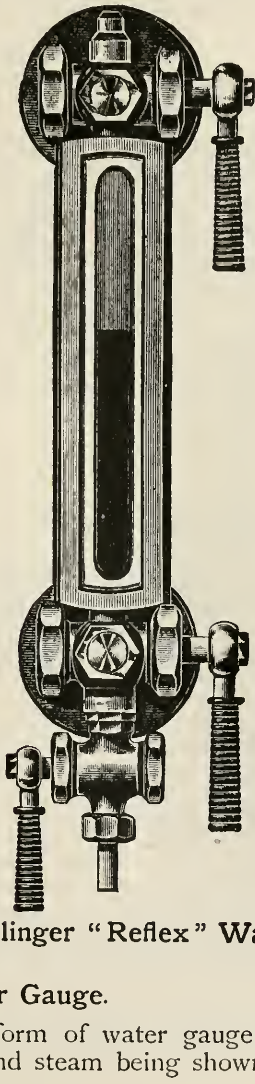
To Cut out a Boiler Tube.

The usual method of removing a defective tube is illustrated in the sketch, and may be described as follows:—The beading at the



No. 93.—Method of Extracting Boiler Tube.

back end of the tube is first cut off flush with the plate, and the tube end cut or ripped in three or four places. The end thus cut up is then hammered inwards, and the bar or rod passed through the tube, with a strong washer fitted in position over the tube end, the washer diameter being of course less than the diameter of the hole. At the front end a dog is fitted as shown and a screwing-up nut; if the bar is then held by the square on the end, and the nut tightened up, the tube will, in most cases, be started and finally drawn out.



No. 94.—Klinger "Reflex" Water Gauge.

Klinger Patent Water Gauge.

In this admirable form of water gauge the chief improvement consists in the water and steam being shown in striking contrast to each, the water in the glass appearing dark and the steam light; the water level can thus be read at some distance away from the gauge.

In the "Klinger" water gauge a metal casing connects the top and bottom cocks, and a window of thick and strong glass inserted in the casing allows of the reading of the water level. This window takes the place of the fragile glass tube commonly fitted.

The observation glass or window being corrugated vertically at the back, reflects the light in that part of the gauge which contains the steam, whereby this part of the glass becomes opaque and of a bright lustre.

In that part of the gauge containing the water the light is not reflected, but passes in a slight deflection to the rear of the gauge.

The glass being thus transparent in this part of the gauge, the water will appear of the dark colour of the background of the casing, in other words the water appears black, while the steam shines with a silvery lustre. This, it must be admitted, is a vast improvement over the ordinary glass, in which if any distance away it is often very difficult to determine whether full of water or empty altogether.

A further advantage of the "Klinger" type of water gauge is the elimination of stuffing boxes at the top and bottom of the glass, which is in itself a consideration of some importance to practical men.

Auld's Patent Steam Reducing Valve.

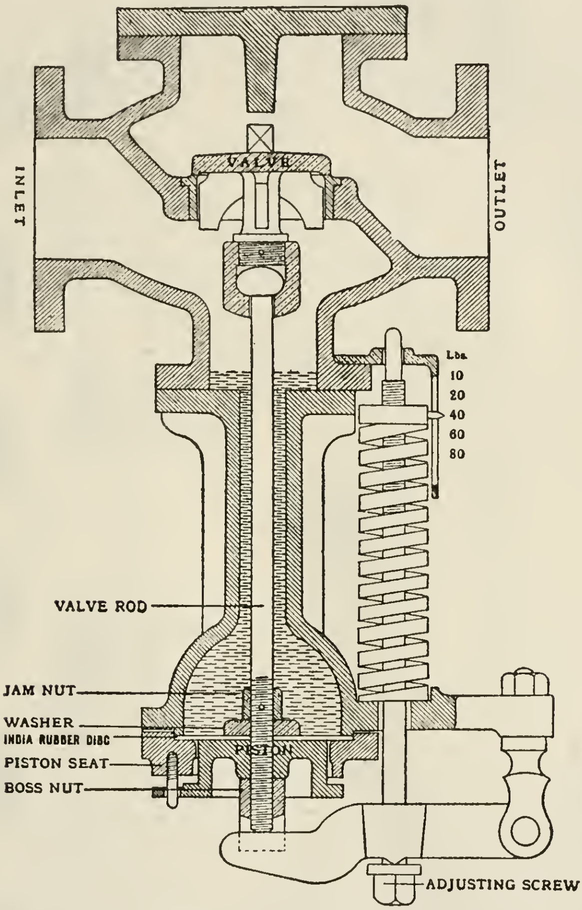
The inflowing steam is admitted between the valve and a piston, covered by an elastic disc.

The piston and the valve are in equilibrium on the high-pressure side.

The reduced pressure on the outlet side is obtained by compressing or relaxing the spring by the adjusting screw until the pointer or spring cap is opposite the figure on index plate representing pressure wanted. The valve being thus opened, the steam flows through valve to low-pressure side of same, until the pressure on reduced side balances the load on the spring by pressing on top side of valve, and so regulating the reduced pressure to the point desired.

A column of water of condensation, shown by dotted lines, is interposed between the steam and the elastic disc.

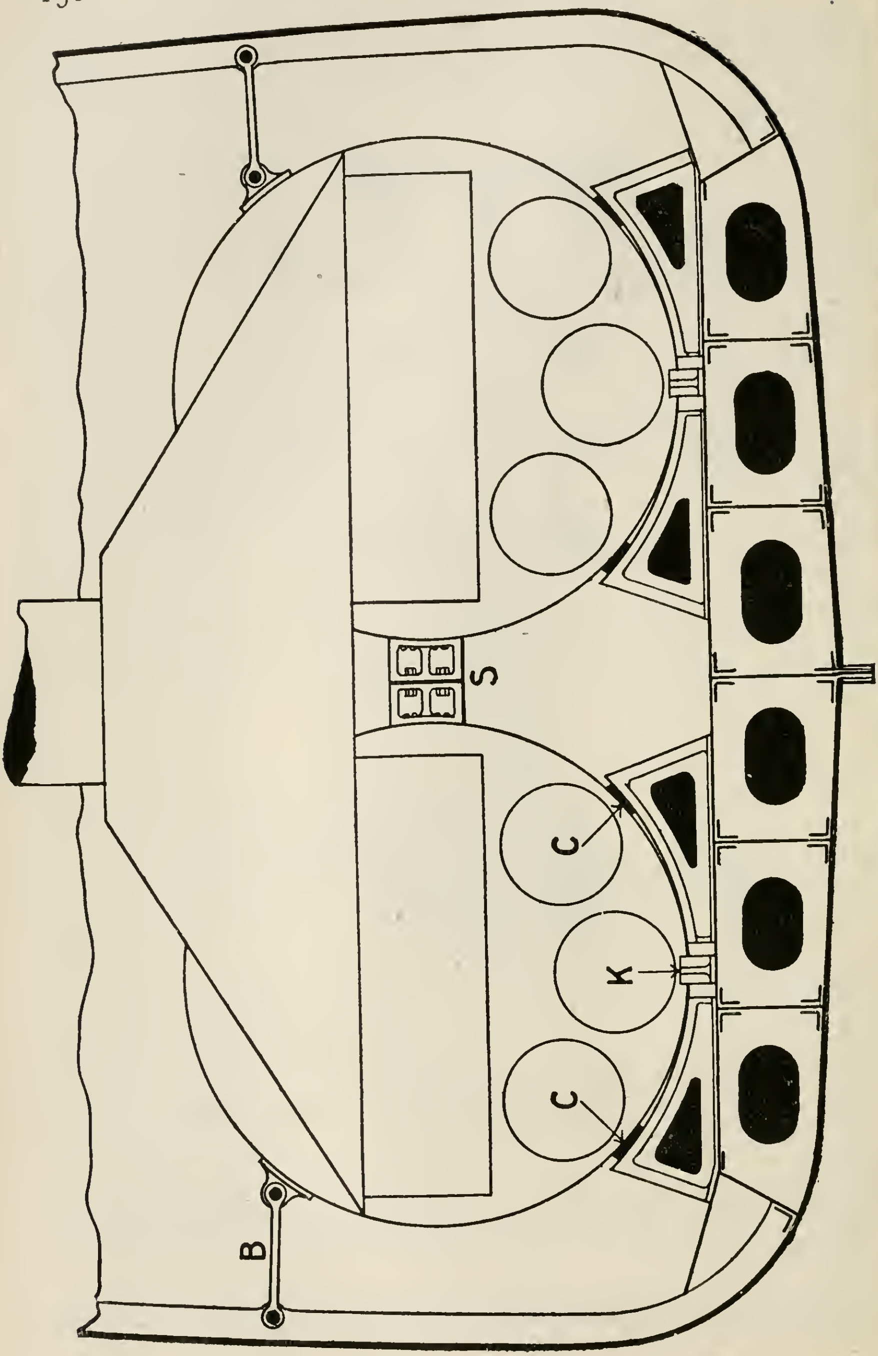
Should it be desired to obtain steam (the reverse way through reducing valve) from donkey boiler for deck or engine-room machinery, when there is no steam in main boilers, open up the reducing valve by compressing the spring by means of the adjusting



No. 95.—Auld's Patent Steam Reducing Valve.

screw till the reducing valve opens; of course, reset the reducing valve to its usual working pressure on reduced side before getting up steam in the main boilers.

NOTE.—A reducing valve will pass steam of lower pressure than it is set for, but will not pass steam of higher pressure.



No. 96.—Method of Staying Boilers in Position

Boilers Secured in Position (Sketch No. 96).

The boilers rest on stools (which, if required, are wedged as shown at C, C), and are prevented from moving longitudinally by knees K, which are riveted or bolted to the ship's frames, clearance being left (about $\frac{1}{8}$ inch) between the boiler ends and the knees for fore and aft expansion.

Side stays B are fixed by eyes and pins to brackets riveted to the boiler shell, and block stays S are often fitted in between the boilers at the centre. The pins and eyes of the side stays B allow for expansion under heat.

Table giving Results of a few Experiments with Auld's Patent Steam Reducing Valve, showing how much the Reduced Pressure Steam is Superheated in passing through the Valve.

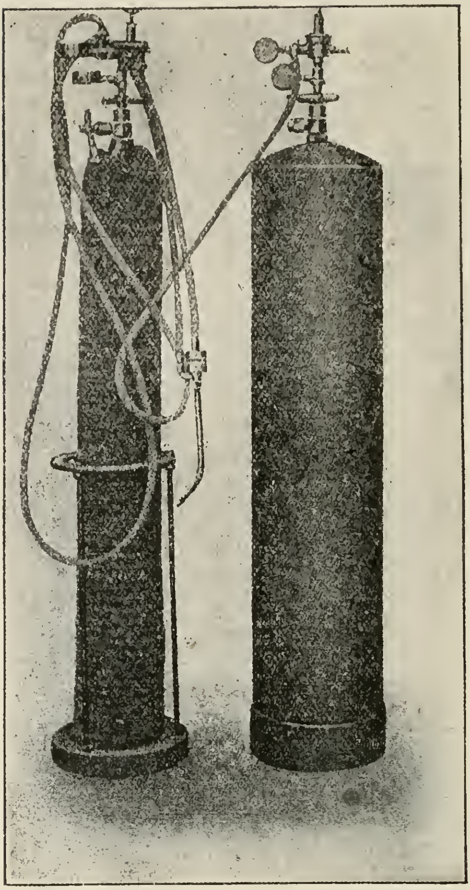
Gauge Pressure.	Temperature.	Reduced Pressure.	Temperature.	Superheat.
160 lbs.	370°	100 lbs.	349°	12°
160 ,,	370°	40 ,,	338°	52°
200 ,,	384°	120 ,,	369°	20°
200 ,,	384°	40 ,,	354°	68°
200 ,,	384°	10 ,,	341°	128°

Autogenous Welding.

The writer is indebted to the British Autogenous Welding Company, Ltd., for the following description of their system of welding:—

The main application of this new method of repair, known under the name of Autogenous Welding, by means of the oxy-acetylene blowpipe, is in repairing cracks and corrosions in marine boilers, doing away with the old and unsatisfactory patching, and, in a great many cases, saving the whole boiler from being scrapped; as, if the boilers are systematically kept in repair by this process, they will remain in good condition for a number of years longer than would be the case otherwise: stems, stern posts, rudder posts, and rudders can be repaired by the same method.

The plant employed to carry out these repairs, known as a high pressure plant, consists of a cylinder of acetylene and one of oxygen, a regulator and length of tubing for each of the gases, and an oxy-acetylene blowpipe as shown in Sketch No. 97.



No. 97.—Autogenous Welding Apparatus.

For this class of work this plant has great advantages over any other plant, which would, of necessity, have to be of the low pressure type in which an acetylene generator is used, when there would be risk of leakage caused through too fast generation or upsetting.

The high pressure plant, when work is being done inside a furnace, is taken into that furnace, thus, being out of the way of any one working in the stokehold, this is a decided advantage when a ship is in port for a short time only and a lot of work has to be got through.

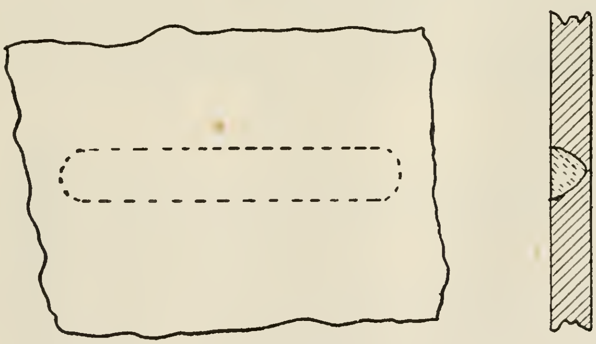
Besides the above-mentioned advantage over the low pressure system there are several others, such as safety, high efficiency caused through having both gases under about the same pressure, thus getting a very intimate mixture and the gas being better purified, easy adjustment of the flame, and the simple construction of the blowpipe, there being nothing to go wrong inside it, and the lightness and balance of the same, which is very important when it is considered that a man has frequently to work in a very strained position, the work being done on the horizontal, vertical, or overhead, it being immaterial, provided he can get his blowpipe flame to play on the correct place ; in some cases, in fact, a man cannot see directly what he is doing but has to work more or less by feel, and makes as good a job as if it was perfectly easy to get at.

For this process of repair the men employed must, when not actually at work on board ship, be kept practising in the shops on an old boiler, and no new man should be put to work on a ship before he has had some months' tuition and practice in the shops ; practice on lighter work is of no use whatever, in fact, if anything, it is detrimental to good work on heavy work, and all ship work comes under this class.

As mentioned earlier, the main faults in boilers are found in cracks and corrosions : we will discuss the former first, giving a few sketches and showing the method of repair.

Cracks are usually found in the furnaces on a belt of from 4 to 8 inches wide, a short distance above the fire-bars, and running the whole length of the furnace.

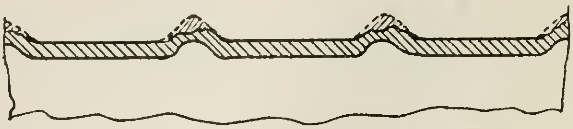
If there are only one or two, each one is cut out, as shown in No. 98, to a V shape, new metal is then added from a rod of Swedish



No. 98.

iron by the welder, who holds his blowpipe in one hand and this iron in the other, adding it drop by drop to the molten mass at the tip of the flame ; if he does not get the original plate properly molten before adding new metal he will not make a weld, and the crack will open on cooling. Other positions in which cracks are to be frequently found are in the landing edges of the furnaces and combustion chamber plates running inwards from the edge of the plate into the rivet holes and sometimes beyond; these cracks are repaired in the same way as furnace cracks, only great care must be taken not to weld the top plate to the one underneath, if this is done endless trouble will be caused.

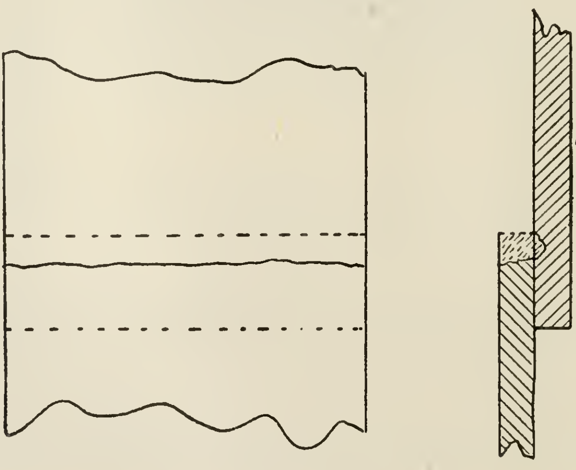
In some cases there are a number of cracks within a small area; it is then advisable to cut out the whole of the affected part and weld in an entirely new piece of plate. In case of a furnace with thickened ribs it is an easy matter to build up the new plate to the correct shape as shown in No. 99, the new metal being shown by



No. 99.

the dotted lines and shaded portions. Corrosion usually takes place in the same region in a furnace as cracks, that is, in a belt 4 or 8 inches wide, a short distance above the fire-bars, extending the whole length of the furnace; to repair this all scale and dirt has to be carefully removed from the corrosion which is then built up to the original thickness of the furnace; the method employed is exactly the same as that for cracks, new metal being added little by little as the plate is brought to a molten state.

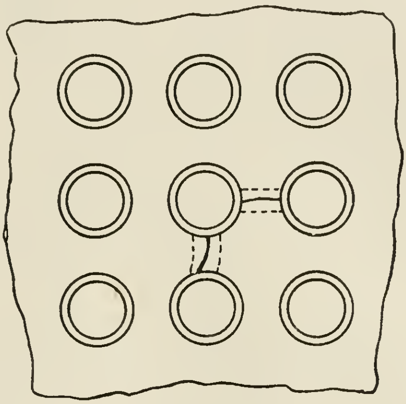
Corrosions very frequently occur at the landing edges of the furnaces and combustion chamber plates; when this is so it is usually found that the back plate is also corroded as shown in No. 100.



No. 100.

To repair, the corroded places are first thoroughly cleaned, the corrosion C is then first made good, the landing of the front plate is then built up to the required amount, the same care being taken in this case not to weld the front to the back plate as in the case of landing edge cracks.

Besides cracks and corrosions in furnaces and combustion chambers, the same defects sometimes arise in the tube plates: cracks extend from one tube hole to another, and are very difficult to repair; as this is the least section of the metal (see No. 101), and is subject to great strain, fortunately these cracks do not often occur. To repair, the crack is cut out as previously explained, and welded; a sheet of iron should be put over the further end of the tubes to prevent a draught being set up.



No. 101.

Corrosion in the tube plates occurs when there is a leak between a tube and the plate; this can be easily repaired by cleaning and adding the necessary new metal.

Another frequent defect in a boiler is corrosion round the mud hole flanges; this can be repaired, if not too far gone, by building up as previously explained. If, however, a flange is too far gone to be repaired by this means, a new flange can be welded in as shown in No. 102, and the joint of the cover can be made at the surface.

No. 102.

With regard to repairs to hulls of vessels these are far more limited than those of boilers; this is to a great extent due to the inferior metal used in their construction as compared to that used in boilers, thus it is impossible to weld a piece into the middle of a ship's side plate.

Welding of frames does not always succeed unless they are set free over a great length.

No difficulty is met with in repairing stems as they are of comparatively small thickness. The repair is carried out as follows:—The crack is cut to a V both sides, the bottoms of the two incisions meeting in the centre, the welding is then done from both sides simultaneously.

Stern posts and rudder posts are much more difficult to repair owing to their thickness. Before repairing, the work must be brought to a red heat by means of a forge or some other method, the blowpipes are then brought into use and the work proceeds in the usual way, only, when once started, the job must not be left till finished.

Besides the above-mentioned repairs, cutting away of furnaces, ship's plates, rivets, &c., can be done with a specially-constructed blowpipe in which an oxy-acetylene flame is used to heat the part to be cut, and an oxygen jet is used to do the cutting; with this blowpipe a great deal of time can be saved, for example, a stem can be cut in four minutes, and the largest stern post in ten minutes, or a furnace can be cut out in about one and a half hours if it is desired to replace it.

Autogenous welding is the uniting of metals by means of heat alone, without the intervention of any different metal.

The heat is obtained by the combustion of acetylene with pure oxygen, which gives a temperature of about 6500° Fahr.

Cutting by Oxygen Jet.—In cutting plates a small surface spot is first heated up by the acetylene flame, and a jet of oxygen is projected on to this hot plate from a separate orifice, the metal being burnt or oxidised away where the jet strikes, with the result that a cut is made similar to that produced by a saw.

General.

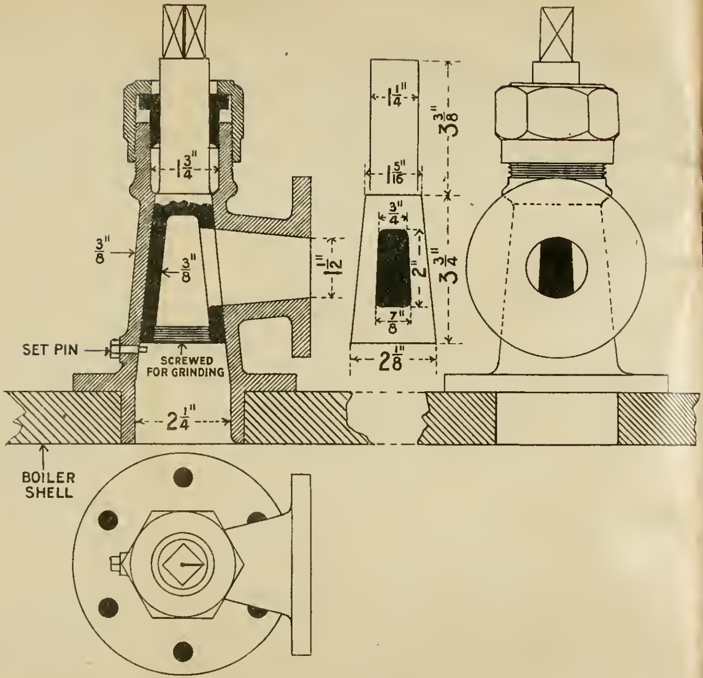
Acetylene (C_2H_2) is a gas of nearly the same weight as atmospheric air.

Each cubic foot of gas generates about 1450 B.T.U.

During combustion of C_2H_2 carbonic acid gas is formed (CO_2), also water vapour (H_2O).

The flame formed by the blowpipe consists for the most part of CO gas, but at the point of CO_2.

The welding flame is surrounded by a film of Hydrogen gas which prevents oxidation from taking place, and thus allows of an efficient weld being formed without the use of a flux.



No. 103.—Water Gauge Cock on Boiler.

This type of cock has the plug taper reversed, thus reducing risk of the plug blowing out. In removing the plug from the shell, it must first be knocked down into the boiler.

A small set pin screwed through the shell of the cock prevents the plug from dropping out of place when in use.

The objection to this type of plug is the tendency for the hole in the plug to become choked up with grease and dirt, and thus reduce the opening of the port.

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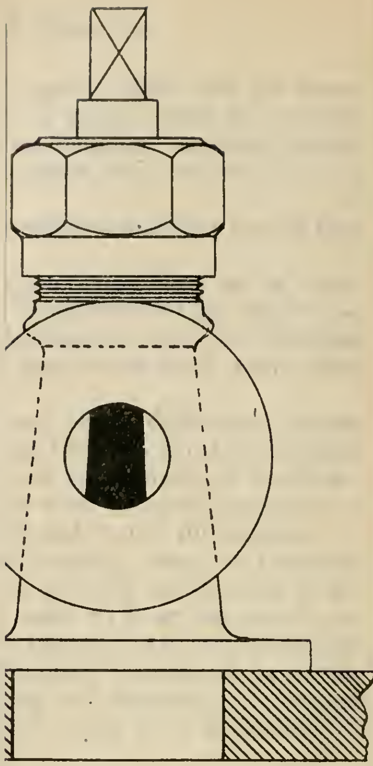

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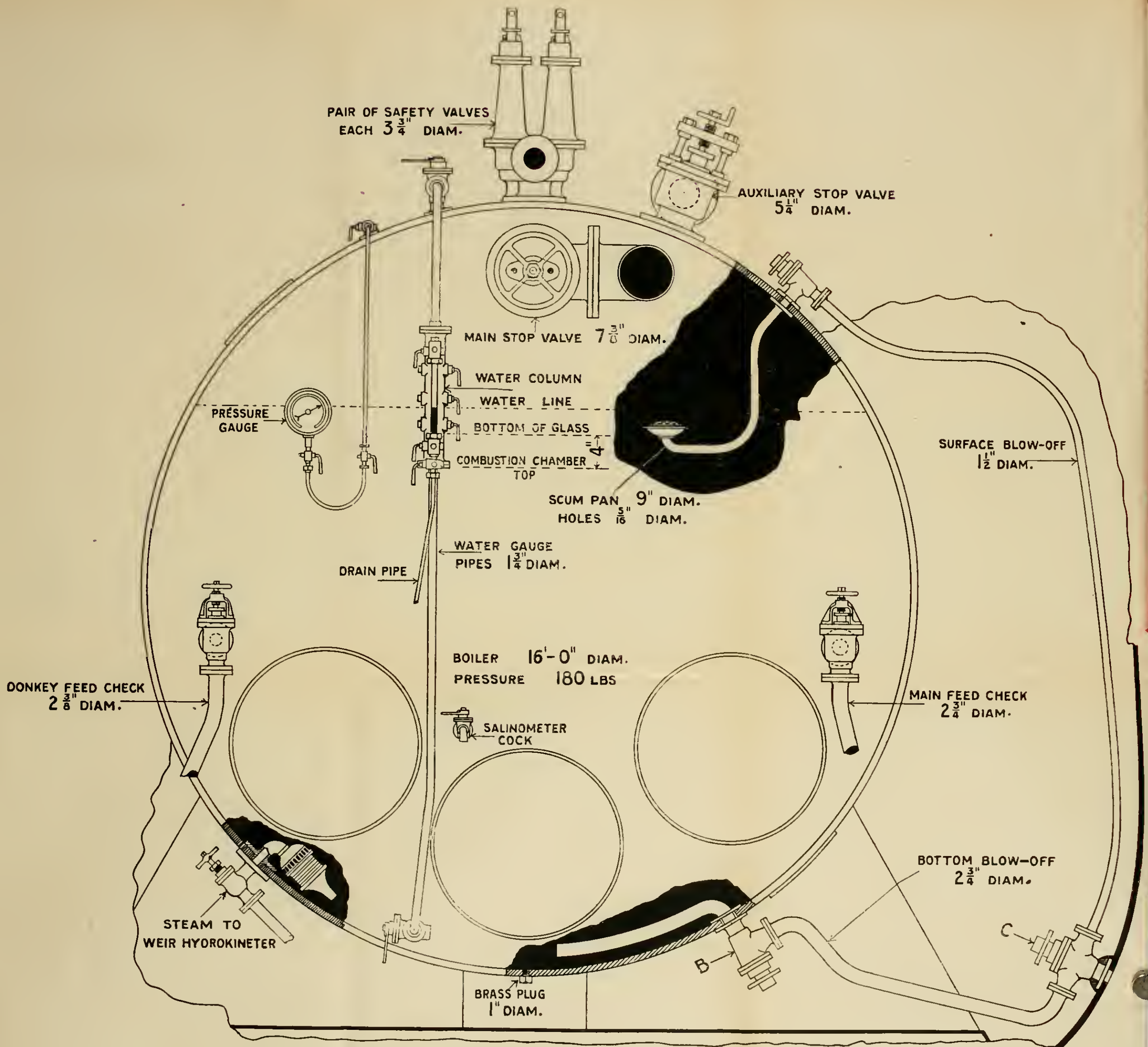

No. 104.—Boiler Mountings.

The principal boiler mountings are shown in the drawing, and are dimensioned for a boiler 16 feet diameter, with a working pressure of 180 lbs. per square inch. The different uses of the mountings are as follows:—

1. Safety valves.—To relieve the pressure when it exceeds the safe limit.
2. Main stop-valves.—To admit the steam into the main steam pipe, thence to the engines.
3. Auxiliary stop-valve.—To give steam to the auxiliary engines, &c.
4. Main Feed check valve.—To admit and control the feed water from the main feed pumps into the boiler.
5. Donkey Feed check valve.—To admit and control the feed water from the donkey feed pumps into the boiler.
6. Surface Blow-off.—To scum the greasy matter off the surface of the water.
7. Bottom Blow-off.—To blow out the boiler water and reduce the density if required.
8. Water gauge.—To show the correct water level in the boiler.
9. Hydrokineter.—To circulate the water in the boiler when getting up steam by a jet of steam from the donkey boiler.
10. Pressure gauge.—To indicate the pressure of the steam (above the atmosphere) in the boiler.

The surface blow-off internal pipe should always be *led down* from the top of the boiler, as shown, and not up from the bottom, for should the pipe break off or crack, danger of inadvertently lowering the water to a dangerous extent will be avoided.

NOTE.—Notice that the bottom cock of the water gauge is 4 inches above the level of the combustion chamber top.

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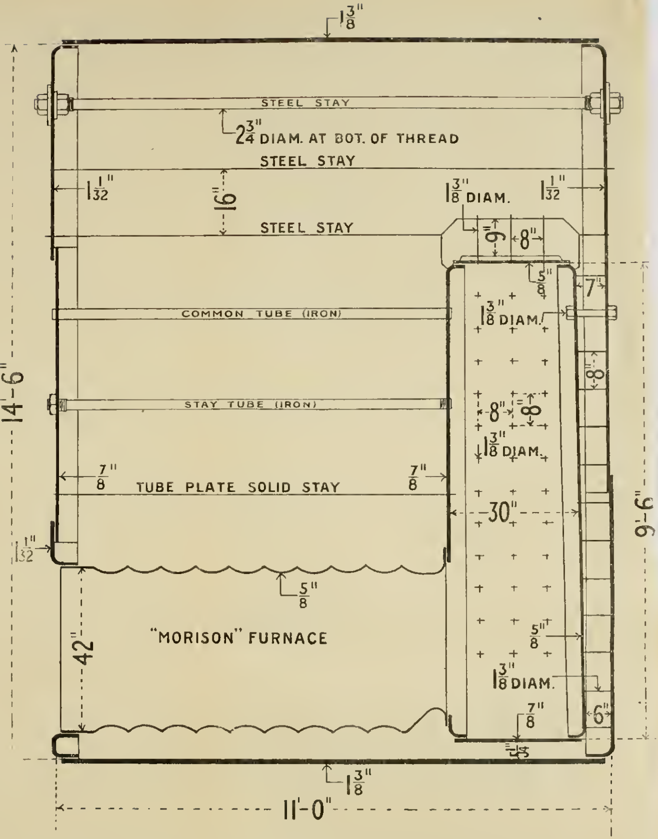

No. 105.—Sketch of Marine Boiler with Principal Dimensions (Longitudinal section).

Students preparing for the First Class Examination should practise drawing the above sketch from memory, noting carefully the dimensions, and the flanging of the plates, &c.

Notice that the furnace shown is of the Gourley-Stephen withdrawable type, as the slightly elevated position of the flange at the back allows of the furnace being canted up and withdrawn from the front end opening.

Shell Thickness.—To find the required shell thickness if the pressure is to be 180 lbs. per square inch, joint 84 per cent., and Factor of Safety 4·6.

RULE—

$$28 \times 2240 \times T \times 2 \times \text{joint} = \text{Factor} \times D \text{ in.} \times \text{Safe Pressure.}$$

Therefore, $T = \dfrac{\text{Factor} \times D \text{ in.} \times \text{Safe Pressure}}{28 \times 2240 \times 2 \times \text{joint}} = \dfrac{4{\cdot}6 \times 174 \text{ in.} \times 180}{28 \times 2240 \times 2 \times {\cdot}84} = 1{\cdot}36$ in., say $1\frac{3}{8}$ in.

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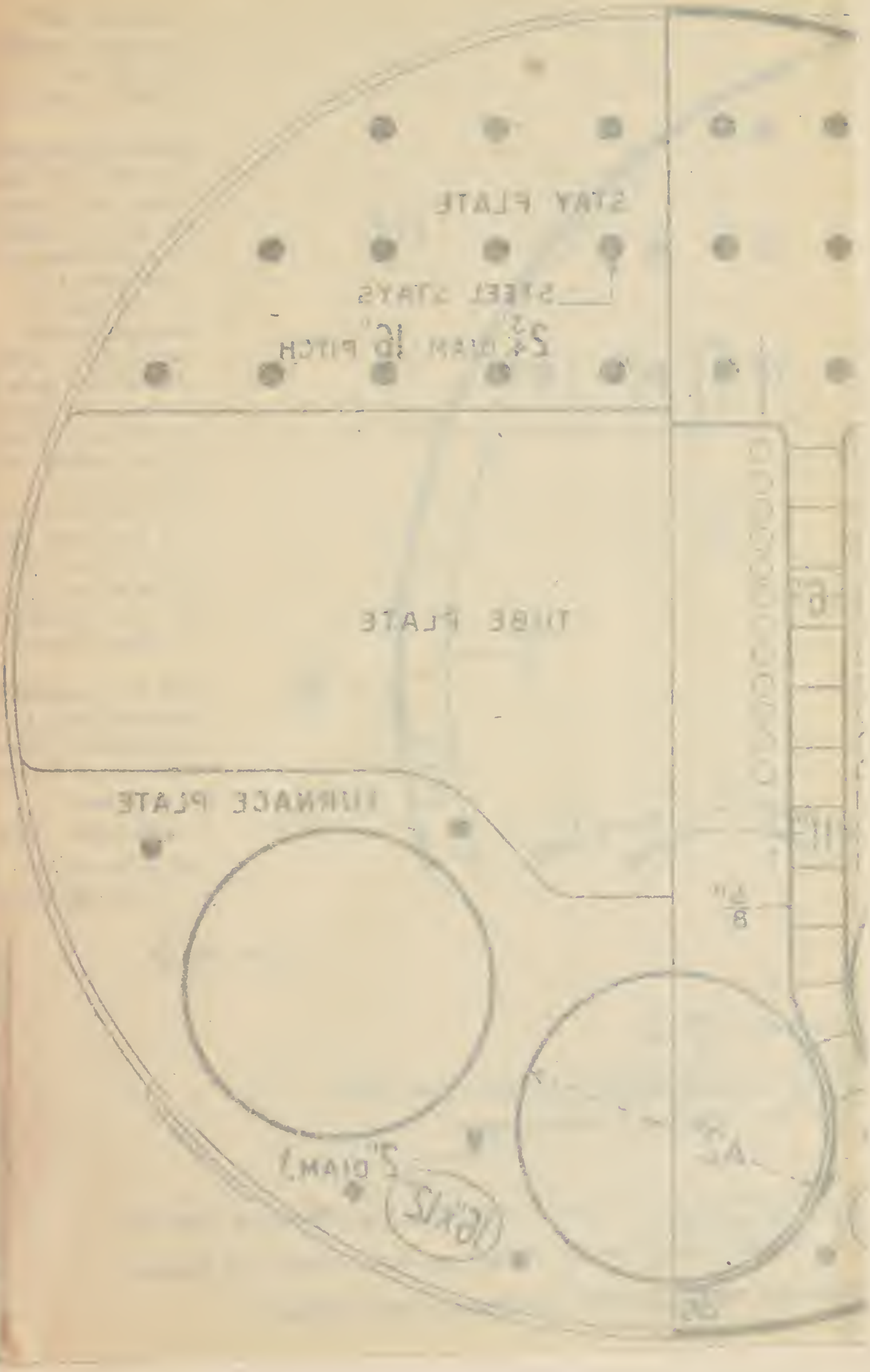

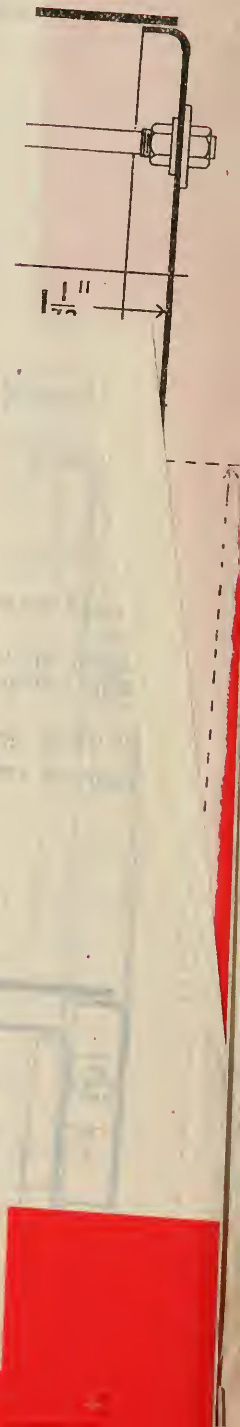

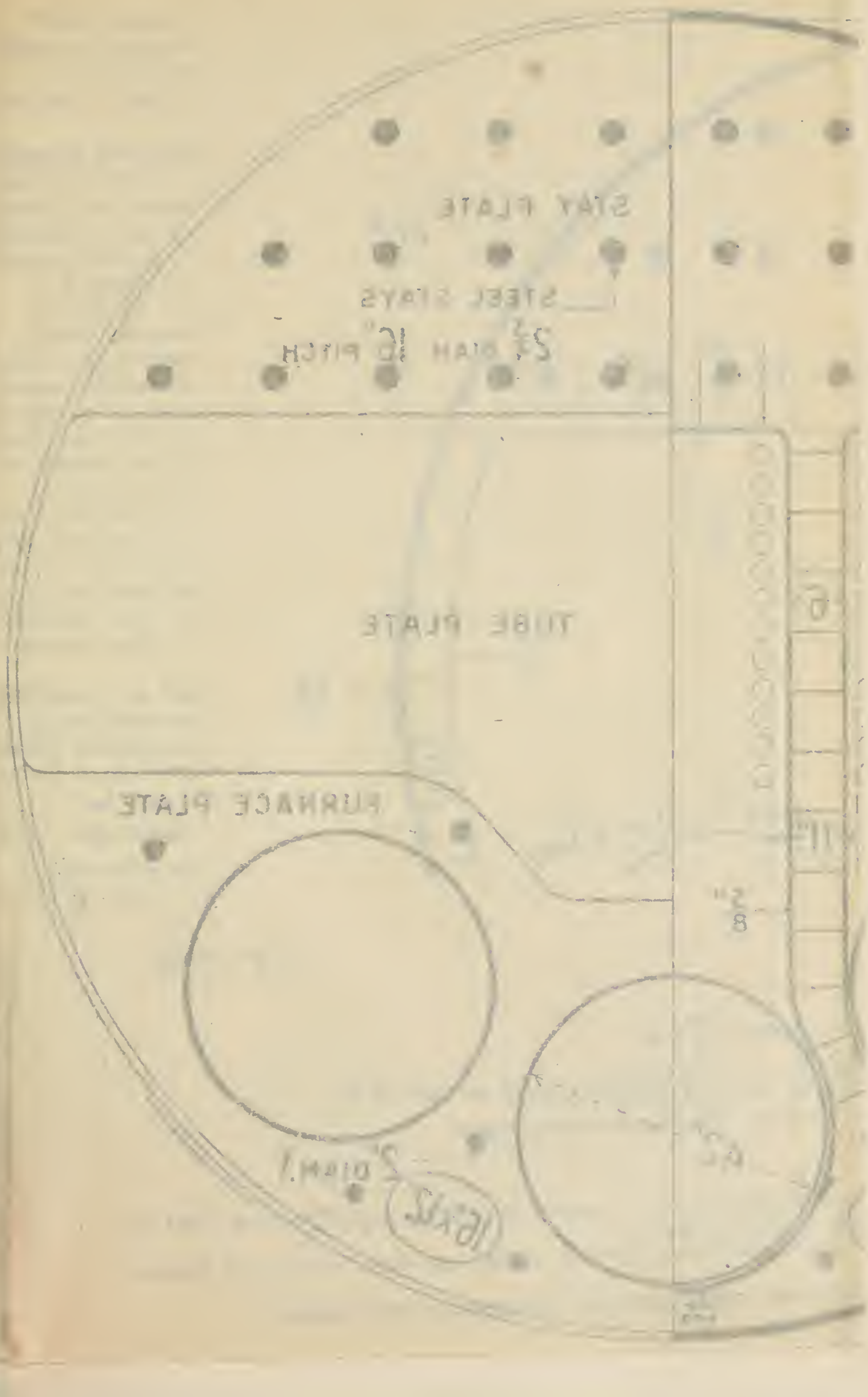

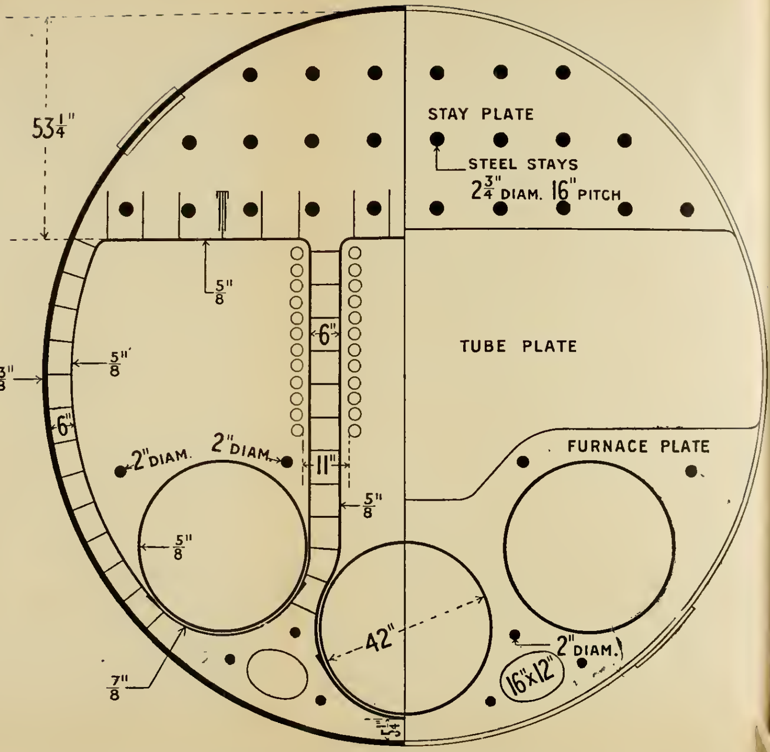

No. 106.—Sketch of Marine Boiler, with Principal Dimensions (half section).

Students preparing for the First Class Examination should practise drawing the above sketch from memory, noting carefully the dimensions and the flanging of the plate, &c.

Safe Pressure.—To find the required Safe Pressure, if the Diameter is 14 feet 6 inches, Factor of Safety 4·6, and the joint strength 84 per cent.

RULE—

$$28 \times 2240 \times T \times 2 \times \text{joint} = \text{Factor} \times D \text{ in.} \times \text{Safe Pressure.}$$

Therefore, $$\text{Pressure} = \frac{28 \times 2240 \times T \times 2 \times \text{joint}}{\text{Factor} \times \text{Diameter in.}} = \frac{28 \times 2240 \times 1{\cdot}375 \times 2 \times {\cdot}84}{4{\cdot}6 \times 174 \text{ in.}} = 181 \text{ lbs., say } 180 \text{ lbs.}$$

NOTE.—1⅜ inches $= 1{\cdot}375$; 14 feet 6 inches $= 174$ inches; $\frac{84}{100} = {\cdot}84$.

The pressure of oxygen supply to the blowpipe varies (with different sizes) from 10 lbs. to 25 lbs.

In cutting plates by means of an oxygen jet the oxygen ignites the plate, which burns away as oxide of iron (similar to rust).

Bottom Blow-off.—In the opinion of the writer the boiler bottom blow-off cock to the ship's side might well be dispensed with altogether, as, unless carefully handled, it undoubtedly constitutes a danger, owing to the possibility of lowering the water level below the combustion chamber tops, which may result in :—

1. Collapse of combustion chamber tops.
2. Deposit of greasy scum matter on top of combustion chambers.

Many engineers are now in favour of discarding the bottom blow-off as commonly fitted, and connecting the bottom pipe as a donkey pump suction, which arrangement allows of either pumping out the boiler, or of circulating when getting up steam by drawing out the colder water at the bottom and pumping it back into the boiler through the donkey feed check higher up.

The surface blow-off (Sketch No. 104), if properly fitted as shown, when opened, only allows the water to be lowered down to the scum pan level, and by its use the oily scum floating on the surface can be blown out, whereas, if the bottom cock is employed, the scum referred to is apt to settle down on the combustion chamber tops, and once deposited will be found extremely difficult to remove except by thorough washing out of the boiler.

Efficiency of Boiler.—The average heating value of 1 lb. of coal is 14500 heat units, and with a feed temperature of, say, 140°, and steam temperature of 380° (180 lbs. gauge pressure), the evaporation would be as follows if no loss of heat occurred :—

$$\text{Heat Units of Evaporation} = 1115 + \cdot 3 \times 380^\circ - 140^\circ = 1089 \text{ Heat Units.}$$

And, $14500 \div 1089 = 13\cdot 3$ lbs. of water evaporated into steam per pound of coal.

If, however, the actual evaporation as measured is only $9\frac{1}{2}$ lbs. of water per pound of coal,

Then, $\text{Boiler Efficiency} = 9\cdot 5 \div 13\cdot 3 = \cdot 714$ or 71·4 per cent.

Equivalent Evaporation.—As a standard of comparison the evaporation obtained by a fuel "from and at" 212° is usually taken, that is, the feed is assumed as being at 212° temperature, and the steam at 212° temperature.

EXAMPLE.—Feed temperature 140°, steam pressure 180 lbs., and temperature 380° ; if the actual evaporation per pound of coal as tested is 10 lbs. of water, find the equivalent evaporation.

Then, $\text{Heat Units for evaporation} = 1115 + \cdot 3 \times T^\circ - t^\circ = 1115 + \cdot 3 \times 380 - 140$
$= 1089$ Heat Units per pound water.

Therefore, $$\frac{1089 \times 10}{966} = 11\cdot 27 \text{ lbs.}$$


LATE

The pressure of oxygen supply to the blowpipe varies (with different sizes) from 10 lbs. to 25 lbs.

In cutting plates by means of an oxygen jet the oxygen ignites the plate, which burns away as oxide of iron (similar to rust).

Bottom Blow-off.—In the opinion of the writer the boiler bottom blow-off cock to the ship's side might well be dispensed with altogether, as, unless carefully handled, it undoubtedly constitutes a danger, owing to the possibility of lowering the water level below the combustion chamber tops, which may result in :—

1. Collapse of combustion chamber tops.
2. Deposit of greasy scum matter on top of combustion chambers.

Many engineers are now in favour of discarding the bottom blow-off as commonly fitted, and connecting the bottom pipe as a donkey pump suction, which arrangement allows of either pumping out the boiler, or of circulating when getting up steam by drawing out the colder water at the bottom and pumping it back into the boiler through the donkey feed check higher up.

The surface blow-off (Sketch No. 104), if properly fitted as shown, when opened, only allows the water to be lowered down to the scum pan level, and by its use the oily scum floating on the surface can be blown out, whereas, if the bottom cock is employed, the scum referred to is apt to settle down on the combustion chamber tops, and once deposited will be found extremely difficult to remove except by thorough washing out of the boiler.

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$$\text{Heat Units of Evaporation} = 1115 + \cdot 3 \times 380^\circ - 140^\circ = 1089 \text{ Heat Units.}$$

And, $14500 \div 1089 = 13\cdot 3$ lbs. of water evaporated into steam per pound of coal.

If, however, the actual evaporation as measured is only $9\frac{1}{2}$ lbs. of water per pound of coal,

Then, $\text{Boiler Efficiency} = 9\cdot 5 \div 13\cdot 3 = \cdot 714$ or $71\cdot 4$ per cent.

Equivalent Evaporation.—As a standard of comparison the evaporation obtained by a fuel "from and at" 212° is usually taken, that is, the feed is assumed as being at 212° temperature, and the steam at 212° temperature.

Example.—Feed temperature 140°, steam pressure 180 lbs., and temperature 380°; if the actual evaporation per pound of coal as tested is 10 lbs. of water, find the equivalent evaporation.

Then, $\text{Heat Units for evaporation} = 1115 + \cdot 3 \times T^\circ - t^\circ = 1115 + \cdot 3 \times 380 - 140$
$= 1089$ Heat Units per pound water.

Therefore, $$\frac{1089 \times 10}{966} = 11\cdot 27 \text{ lbs.}$$

One pound of the coal referred to can therefore evaporate 11·27 lbs. of water into steam "from and at" a temperature of 212°.

Weight of Gases passing up Funnel.—The weight of the waste gases which pass up the funnel can be estimated as follows :—

EXAMPLE.—If the consumption is 25 tons of coal per twenty-four hours, determine the total weight of the products of combustion passing up the funnel during that period.

NOTE.—Each pound of coal requires about 24 lbs. of air for complete combustion.

Therefore, Weight of air required $= 25 \times 24 = 600$ tons.

But (neglecting ash and clinker) the 25 tons of coal also pass off in the form of gas.

So that, total weight of gases $= 600 + 25 = 625$ tons per twenty-four hours.

The actual weight of the gases would be perhaps about 10 per cent. less than the above, if the residue of ash and clinker were deducted.

Shortness of Water.—If a boiler runs short of water through faulty check valve, or other causes, the first thing to do is to pump in more feed water, either hot or cold, preferably hot of course, but even cold water will in most cases result in no serious injury or danger if the water is not actually below the crowns. The seams or tubes may afterwards leak slightly, but nothing more is likely to happen.

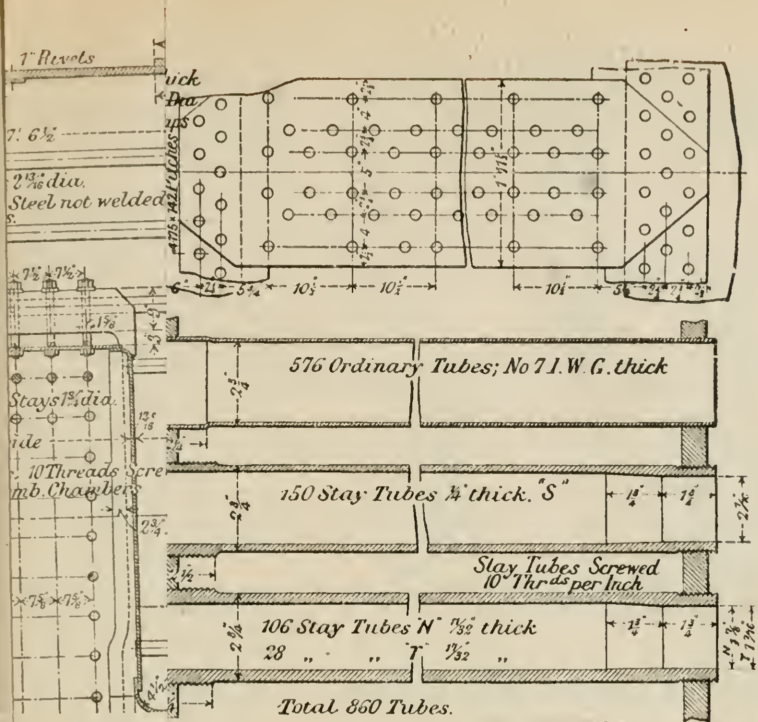
If, then, one boiler out of, say, a set of four boilers shows a low water level, the engineer on watch should at once open up the check valve of that boiler, and reduce the lift of the check valves on the other three boilers: in addition to this it may become necessary to put on an extra feed pump (if one is available) to the boiler which shows no water in the glass. The point to be remembered is to *get water* into the boiler as soon as possible, and, as before stated, even cold feed water may be pumped in without risk of accident if the boiler is merely short of water.

Velocity of Gases.

The mean velocity of the gases passing through the uptake and funnel is about 13 feet per second; the mean velocity of the gases passing over the furnace bridge is about 75 feet per second; the mean velocity of the gases passing through the tubes is about 60 feet per second.

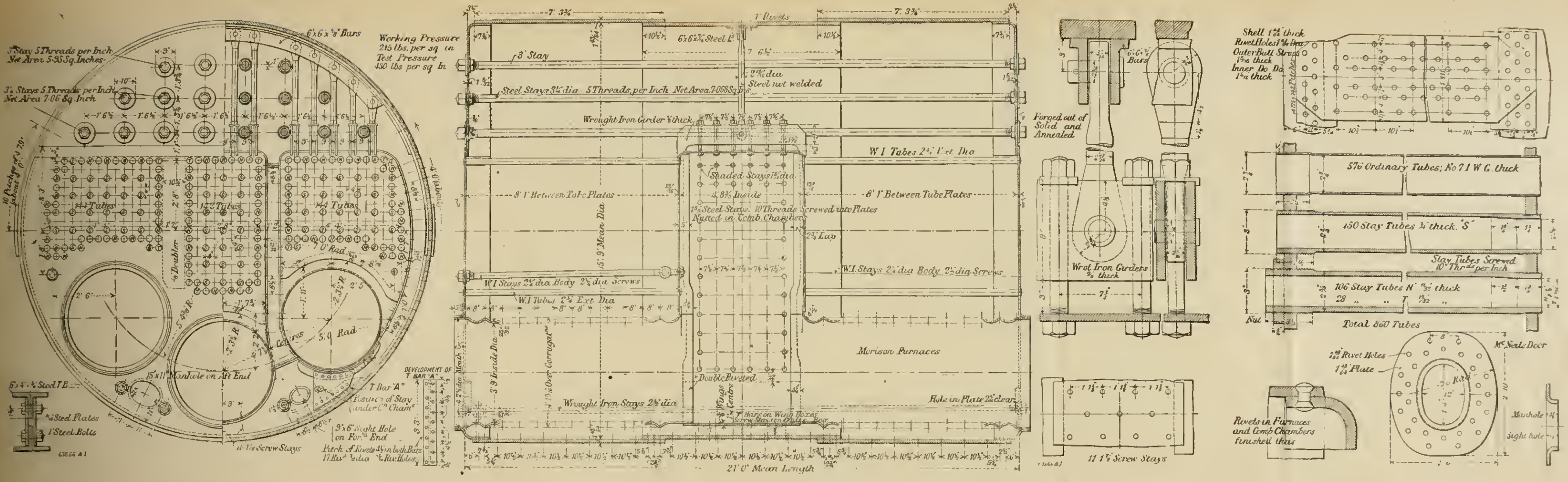
Evaporation per Pound of Coal.

One pound of average coal gives out about 9000 units of heat (B.T.U.).



Double-butt strap

Rivet diameter - - - -	$1\frac{1}{4}$ inches (holes $1\frac{5}{16}$ inches).
Maximum Pitch of Rivets (five rivets per pitch) - - - -	$8\frac{1}{16}$ „
Distance between inner rows - -	$2\frac{5}{16}$ „
„ „ outer „ - -	$3\frac{5}{16}$ „
Rivet section strength - - -	93·2 per cent.
Plate „ „ - - -	83·7 „



Double Ended Boiler of White Star Liner "Britannic."

(Reproduced by permission from "Engineering," Feb. 27, 1914.

White Star Liner "Britannic."

(By Messrs Harland & Wolff Ltd.)

GENERAL DATA—

Length over all	900 ft.
Breadth	94 ,,
Depth, moulded	64 ,, 3 in.
Height from keel to bridge	104 ,, 6 ,,
Gross tonnage	50000 tons
Load draught	34 ft 7 in.
Displacement at load draught	53000 tons.
Combined I H P. of wing reciprocating engines (ahead and astern)	32000 ,,
Shaft horse power of centre turbine (ahead only)	18000 ,,
Sea speed	21 knots

BOILER DATA—

Number of double ended boilers	24
,, ,, single ,,	5
Diameter of all boilers	15 ft 9 in.
Length of double ended boilers	21 ,, 0 ,,
,, ,, single ,,	11 ,, 9 ,,
Number of furnaces on each double ended boiler	6
,, ,, ,, single ,, ,,	3
Total heating surface per double ended boiler	5702 sq. ft.
,, grate ,, ,, ,, ,,	130·8 ,, ,,
,, heating ,, ,, single-ended ,,	2822 ,, ,,
,, grate ,, ,, ,, ,,	65·4 ,, ,,
Ratio of heating surface to grate surface	As 43 is to 1
Inside diameter of furnace corrugations	3 ft 9 in
Type of corrugation	Morison.
Total number of furnaces	159.
Working boiler pressure	215 lbs. (gauge).
Test pressure	430 ,, ,,

It will be noted that each combustion chamber is suspended from the shell by a large central stay, secured top and bottom by eyes and pins to angle irons above and to the girder below. The large scale drawing on right shows clearly the details and dimensions of the suspension stay. It should also be carefully noted that when suspension stays are fitted the combustion chambers are also anchored to the shell at the bottom, either by screwed stays (centre furnace) or by plate stays (wing furnaces).

[To face page 164.

To find the units of heat required to evaporate 1 lb. of water into steam, the rule is as follows:—

$$1115 + \cdot 3 \times T - t = \text{Units of heat.}$$
T = Steam temperature. t = Feed temperature.

EXAMPLE.—The steam pressure is 160 lbs. or 370° temperature, and the feed water temperature is 140°; find the units of heat required to evaporate 1 lb. of water into steam, and the number of pounds of water evaporated by 1 lb. of coal.

Then, $1115 + \cdot 3 \times 370° - 140° = 1086$ units of heat required per pound of coal.

Therefore, $\frac{9000}{1086} = 8\cdot 28$ lbs. of water evaporated per pound of coal.

NOTE.—To evaporate 1 lb. of water at 212° temperature into a pound of steam at atmospheric pressure requires 966 units of latent heat.

Boiler Dimensions.

The following data refer to a modern type double-ended boiler carrying a pressure of 170 lbs. per square inch, and the various plate thicknesses and details of riveting should be carefully noted.

Boiler Data.

Pressure	170 lbs. per square inch.
Diameter	13 feet 10 inches.
Length (double ended)	20 ,, $1\frac{1}{2}$,,
Number of furnaces	6.
,, combustion chambers	6.
Grate length	6 feet 10 inches.
,, width	3 ,, $4\frac{1}{2}$,,
Total grate area	138 square feet.
Heating surface of tubes	2840 ,,
,, ,, furnaces	242 ,,
,, ,, combustion chambers	444 ,,
Total heating surface	3664 ,,
Heating surface to grate	26·5 is to 1.
Area over bridge	3·11 square feet.

Shell consists of three courses of plates, each course having three plates; the centre course is outside of the end courses.

Shell thickness	$1\frac{1}{4}$ inches.

Longitudinal Shell Seams.

Double-butt strap thickness	1 inch.
Rivet diameter	$1\frac{1}{4}$ inches (holes $1\frac{5}{16}$ inches).
Maximum Pitch of Rivets (five rivets per pitch)	$8\frac{1}{16}$,,
Distance between inner rows	$2\frac{5}{16}$,,
,, ,, outer ,,	$3\frac{5}{16}$,,
Rivet section strength	93·2 per cent.
Plate ,, ,,	83·7 ,,

ve[illegible] gases passing through the tubes is about 60 feet per second.

Evaporation per Pound of Coal.

One pound of average coal gives out about 9000 units of heat (B.T.U.).

To find the units of heat required to evaporate 1 lb. of water into steam, the rule is as follows :—

$$1115 + \cdot 3 \times T - t = \text{Units of heat.}$$
T = Steam temperature. t = Feed temperature.

EXAMPLE.—The steam pressure is 160 lbs. or 370° temperature, and the feed water temperature is 140° ; find the units of heat required to evaporate 1 lb. of water into steam, and the number of pounds of water evaporated by 1 lb. of coal.

Then, $1115 + \cdot 3 \times 370° - 140° = 1086$ units of heat required per pound of coal.

Therefore, $\frac{9000}{1086} = 8 \cdot 28$ lbs. of water evaporated per pound of coal.

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Boiler Dimensions.

The following data refer to a modern type double-ended boiler carrying a pressure of 170 lbs. per square inch, and the various plate thicknesses and details of riveting should be carefully noted.

Boiler Data.

Pressure - - - -	170 lbs. per square inch.
Diameter - - - -	13 feet 10 inches.
Length (double ended) - - -	20 ,, $1\frac{1}{2}$,,
Number of furnaces - - -	6.
,, combustion chambers -	6.
Grate length - - - -	6 feet 10 inches.
,, width - - - -	3 ,, $4\frac{1}{2}$,,
Total grate area - - - -	138 square feet.
Heating surface of tubes - -	2840 ,,
,, ,, furnaces - -	242 ,,
,, ,, combustion chambers	444 ,,
Total heating surface - - -	3664 ,,
Heating surface to grate - -	26·5 is to 1.
Area over bridge - - -	3·11 square feet.

Shell consists of three courses of plates, each course having three plates ; the centre course is outside of the end courses.

Shell thickness - - - -	$1\frac{1}{4}$ inches.

Longitudinal Shell Seams.

Double-butt strap thickness - -	1 inch.
Rivet diameter - - - -	$1\frac{1}{4}$ inches (holes $1\frac{5}{16}$ inches).
Maximum Pitch of Rivets (five rivets per pitch) - - - -	$8\frac{1}{16}$,,
Distance between inner rows - -	$2\frac{5}{16}$,,
,, ,, outer ,, - -	$3\frac{5}{16}$,,
Rivet section strength - - -	93·2 per cent.
Plate ,, ,, - - -	83·7 ,,

Centre Circumferential Shell Seams.

Rivet diameter - - - -	$1\frac{5}{16}$ inches (holes $1\frac{3}{8}$ inches).
,, pitch (three rivets per pitch) -	$4\frac{1}{8}$,,
,, section strength - - -	68·6 per cent.
Plate ,, ,, - - -	66·6 ,,

End Circumferential Seams.

Rivet diameter - - - -	$1\frac{1}{4}$ inches (holes $1\frac{5}{16}$ inches).
,, pitch (two rivets per pitch) -	$3\frac{1}{2}$,,
,, section strength - - -	70·15 per cent.
Plate ,, ,, - - -	62·5 ,,

End Plates.

Top end plate thickness - -	$1\frac{1}{8}$ inches.
Centre ,, ,, (front tube plate)	$\frac{3}{4}$ inch.
Bottom ,, ,, (furnace plate) -	$1\frac{1}{8}$ inches.
Furnace length - - - -	7 feet 8 inches.
,, diameter - - -	3 ,, 7 ,,
,, thickness - - -	$\frac{1}{2}$ inch.
Combustion chamber width - -	2 feet 6 inches.
Back tube plate thickness - -	$\frac{3}{4}$ inch.
Thickness of combustion chamber plates	$\frac{9}{16}$,,
Diameter of combustion chamber stays -	$1\frac{1}{4}$ inches (bottom of thread).
Pitch ,, ,, ,,	from $6\frac{1}{2}$ to $8\frac{1}{2}$ inches.
Length of combustion chamber girders -	2 feet $6\frac{1}{2}$ inches.
Depth ,, ,, ,,	$7\frac{3}{4}$ inches.
Thickness ,, ,, ,, (two plates each) - - -	$\frac{3}{4}$ inch.

Each girder fitted with three $1\frac{1}{2}$-inch diameter bolts.

Tubes.

Diameter of tubes - - -	3 inches outside.
Length ,, - - -	7 feet $4\frac{1}{2}$ inches.
Pitch of tubes - - - -	$4\frac{1}{4}$ inches horizontally.
,, ,, - - - -	4 ,, vertically.
Number of plain tubes per boiler -	324 (No. 8 B.W.G.).
,, stay ,, ,, -	180 (,, 6 ,,).
Diameter of main stays (steel) - -	$2\frac{1}{2}$ inches.
Manhole - - - -	16 × 12 inches.
Mudholes (five in number) - -	15 × 11 ,,

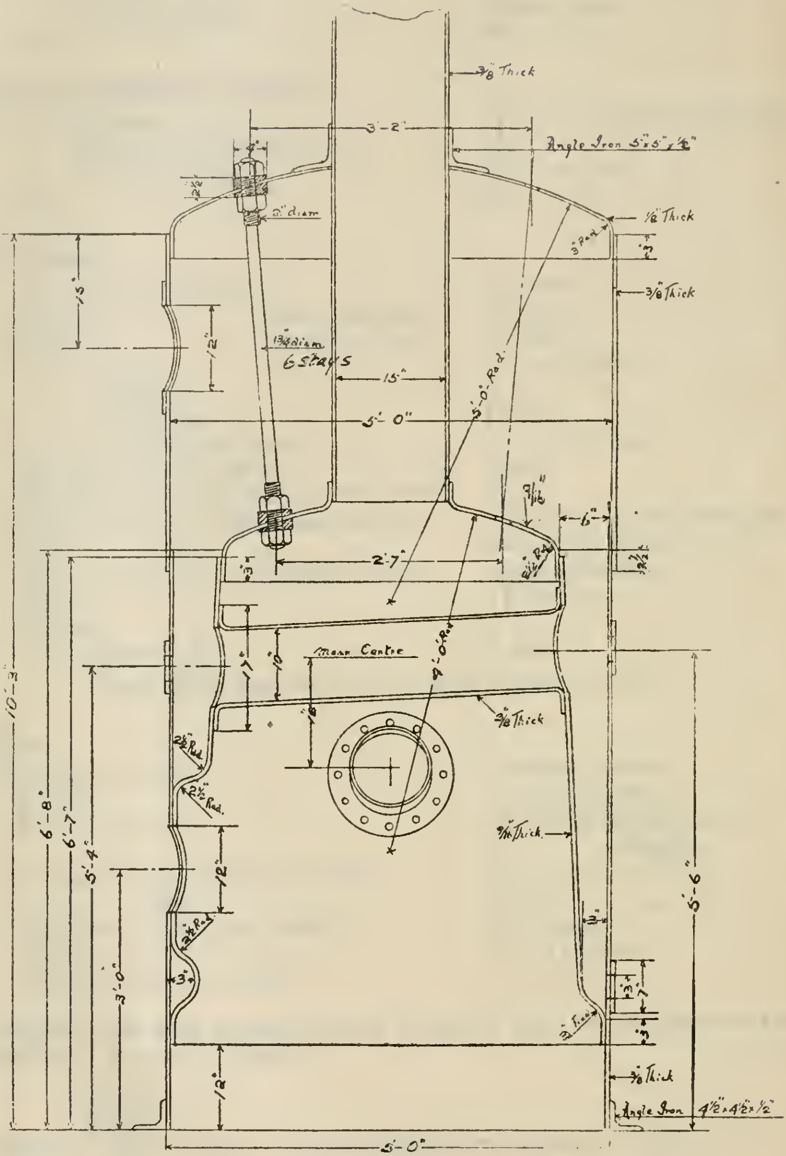
NOTE.—The grate surface for each furnace is equal to the length of bars multiplied by diameter of furnace.

Therefore, $\dfrac{\overset{\overset{6'\,10''}{12}}{82} \times \overset{\overset{3'\,4\frac{1}{2}''}{12}}{40\cdot5}}{144} = 23$ sq. ft., and 23 × six furnaces = 138 sq. ft. (total).

And, Total Heating Surface ÷ Grate Surface = Ratio.

Then, 3664 ÷ 138 = 26·5 to 1.

Vertical Donkey Boiler



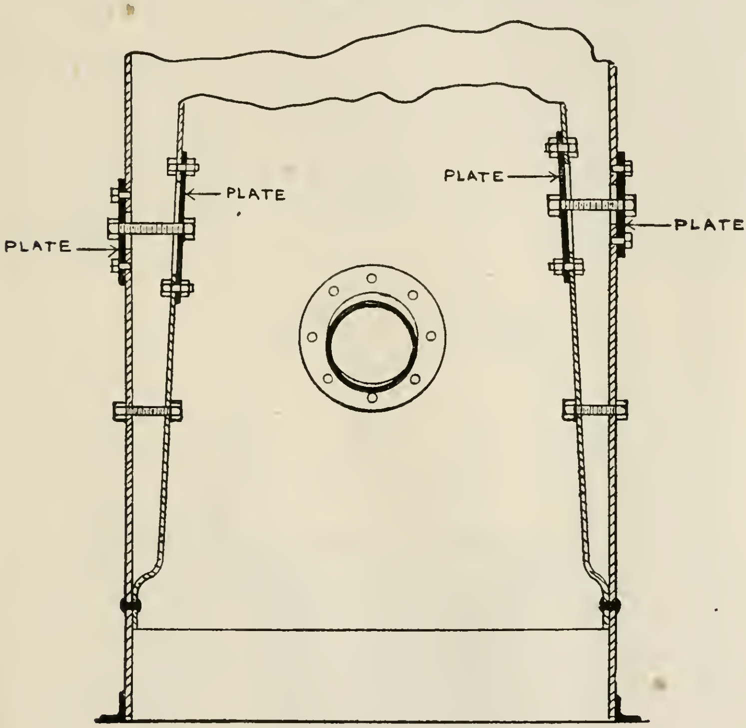
No. 107.—Vertical Type Donkey Boiler.
Pressure, 80 lbs. (gauge).

Referring to the joint strengths of each seam, it must be remembered that the smaller per cent. of rivet section and plate section at seam is the joint strength, therefore,

Joint strength for	longitudinal shell seams		=83·7 per cent.
,, ,,	centre circumferential shell seams		=66·6 ,,
,, ,,	end ,, ,,	,,	=62·5 ,,

The screwed portions of the stay tubes and combustion chamber stays have twelve threads per inch.

Repair for Broken Cross-Water Tube in Vertical Donkey Boiler.



No. 106a.—Donkey Boiler Repairs.

Cut out the broken tube, and fit blind flanges over the openings of the fire box (inside) which either rivet or bolt as shown in sketch.

Remove small dogs and inspection doors on shell and pin on small plates, then screw in stays as shown to support fire box. Stays of about $1\frac{5}{8}''$ diameter would be found sufficient.

Repair for Weak (Corroded) Fire Box.

Tap in a ring of stays through shell and fire box, and either rivet over or fit on nuts. The stays will give additional strength to the fire box.

Sketch 107 shows clearly the construction of this kind of boiler. The cross water tubes are for improving the circulation and increasing the heating surface, and it is to be noted that a handhole is fitted opposite each tube for cleaning purposes.

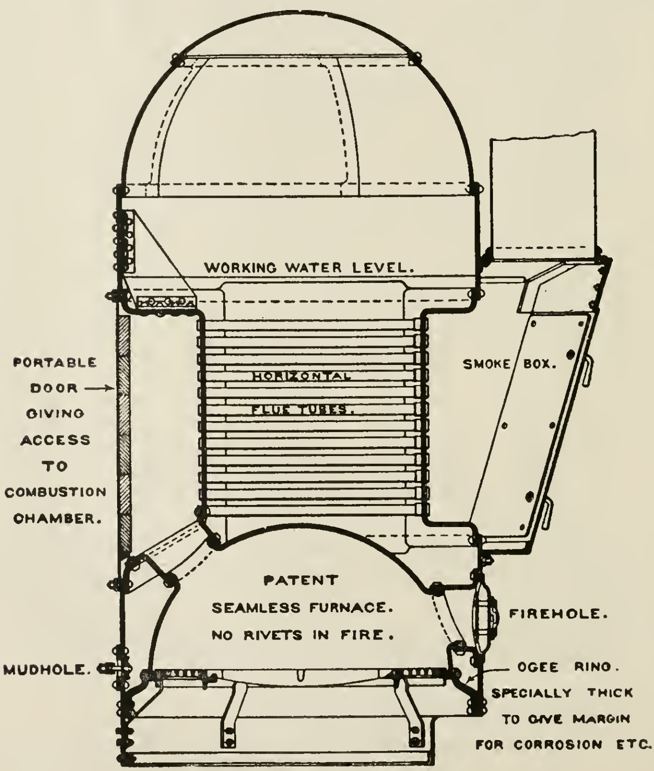
There are usually from four to six stays of about 2 inches diameter for supporting the fire-box.

This boiler has a wet uptake, and corrosion takes place at the water level; corrosion also goes on at the bottom, between the shell and the fire-box, and is caused by what is called "grooving," that is, the upper part of fire-box expands by the heat, but the lower part at the bottom being riveted to the shell is kept rigid; therefore the skin of the metal cracks slightly at the bend, and allows of corrosion taking place. The corrosion is also increased by the want of circulation at this part of the boiler.

As the fire-bars are low down, and the ash-pit immediately underneath, the boiler is a dry bottomed one. For ordinary proportions, the shell is about $\frac{3}{8}$ inch thick and the fire-box $\frac{5}{8}$ inch thick.

The average dimensions for this type of boiler are as follows:—

Pressure	80 lbs. gauge.	Cross tubes	10 inches diameter.
Diameter	5 feet.	Uptake	15 ,, ,,
Height	10 ,,	Water space round fire-box	6 inches to 3 inches.
Shell plates	$\frac{3}{8}$ to $\frac{1}{2}$ inch thick.	Vertical stays	2 inches diameter (six in number).
Fire-box plates	$\frac{9}{16}$ to $\frac{5}{8}$,,		



No. 108.—Cochran Patent Vertical Type Multitubular Boiler. For Marine Use.

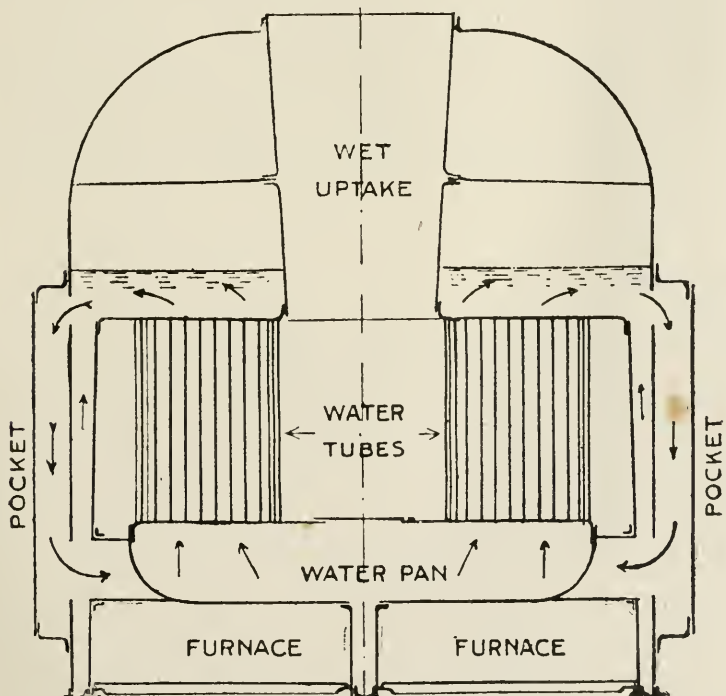
The chief advantages of this type of donkey boiler as compared with the ordinary type are :—

1. Greater steaming capacity, with same space occupied.
2. Increase of heating surface as compared with grate surface.
3. Improved construction of parts, including new patent seamless furnace, having neither riveted nor welded seam exposed to flame, which is a matter of considerable practical importance.

Haystack Boiler.

This type of water tube boiler is fitted with four furnaces, and two circulation pockets, as shown. The water circulates down the pockets, and enters the pan by means of large pipes, and after evaporating rises up through the tubes into the steam space.

The Haystack boiler is of large capacity, and is a fairly quick steam raiser, but in a number of cases a serious disadvantage exists in the difficulty experienced in locating the true water level, as the glass often indicates incorrectly. In a number of boilers, which have come under the writer's observation, the water in the gauge glass



No. 109.—Haystack Boiler.

behaved in a most erratic manner, when under weigh and when stopped, indicating falsely under both conditions, as sometimes the glass would show full when the engines were running, and empty when stopped. This is evidently due to the position of the tubes and the shell, and the peculiar effect of the rising circulation currents.

This boiler has a dry bottom, and a wet uptake.

Water Tube Boilers.

This class of boiler has not as yet been adopted to any considerable extent in the Merchant Service, comparatively few steamers having been fitted with them ; from this it would appear that shipowners are awaiting the thorough testing, in the Navy, of this type of steam generator before seriously considering the advisability of having it included in the specifications of new steamers. It should, however, be borne in mind that a certain class of boiler may suit Naval practice, but not be so well adapted for the Merchant Service, owing to the different requirements existing in each particular case.

The cylindrical, or, as it is usually termed, "Scotch" boiler which has for so long a period done its duty satisfactorily as an effective steam generator, is still the favourite boiler in use for passenger and cargo steamers ; but should the demand for a still further increase of pressure become general, as seems probable, it is almost certain that engineers will require to turn towards water tube boilers of one type or another to obtain the requisite pressure, compatible with safety and convenience of manufacture, as, for a pressure of, say, 300 lbs. per square inch, the diameter of a cylindrical boiler would require to be very small, or the shell thickness very great, to allow of this being safely carried.

Yarrow Boiler.

Perhaps the two best known types of water tube boilers in general marine practice are the Yarrow and the Babcock & Wilcox. The Yarrow boiler is of simple construction, consisting of two bottom water and mud drums, and one top steam drum with straight tubes connecting the top drum to the bottom drums. The top drum is circular and the bottom drums of oval shape. The tubes are expanded into the drums, and also bell-mouthed (an extra precaution) to prevent drawing out of tubes. The tubes are easily cleaned and do not silt up so easily as the curved tubes in other types of boilers. Two large tubes are led from the top drum to the bottom drums at each side ; these tubes are termed downcast tubes, and serve as a return connection from the top to the bottom drums when circulation is going on. These tubes, it may be mentioned, are outside of the boiler casing and are not in contact with heat. A large casing, built up of asbestos lined plates, is fitted outside the tubes, and doors are fitted to allow of easv cleaning out of soot which gathers at the bottom of the casing (Sketch of Boiler, No. 110). A large grate and combustion chamber is one of the features of the Yarrow boiler. Zinc plates are fitted, as in Scotch boilers, in the upper or steam drum, and also in each of the bottom drums. Yarrow boilers are usually installed in close stokeholds, fans being driven to give air pressure required. It is found that water tube boilers give the best results with an even fire of 6 to 7 inches thick, and level firing. In the upkeep of Yarrow boilers it is necessary to clean out soot chambers at least every second day, and the space between tubes also requires cleaning as often as is possible, but the soot chambers can be cleaned while boiler is steaming. Furnace doors and ashpit doors are so arranged that, in event of boiler tube bursting, all doors will close, thus confining escaping steam as much as possible. The tubes vary in diameter from $1\frac{1}{8}$ to $1\frac{1}{4}$ inches.

Babcock Boiler.

The Babcock & Wilcox boiler is of different construction, the tubes being expanded into boxes usually termed headers. These headers are fitted with small doors in line with the tubes to allow of cleaning same. The headers or sections are again connected to the top or steam drum. This boiler has also a large grate surface, and is best fired on the same principle as the Yarrow, that is, a level fire of 6 or 7 inches thick. This boiler has the advantage of working on natural draught, and is a good steaming boiler under those conditions. As in the Yarrow boiler the circulation takes place in a similar manner, downcast tubes being fitted leading back to the mud drum or bottom header. Blow-off cocks are fitted on this header for cleaning out header. The furnace in water tube boilers is lined with fire-brick throughout, and the ashpits are kept supplied with water so as to avoid damage to the fire-bars. Superheaters are being fitted in the Babcock & Wilcox boiler, superheating the steam to 100° to 150°. The superheater is fitted athwart-ships, and consists of a series of U-shaped tubes connected to headers in a similar manner to the main boiler tubes. After steam is generated in the main boiler it is passed through the superheater before passing to the machinery. Steam connections for cleaning tubes of soot are fitted, access being obtained by doors on each end of boiler. Zinc plates in perforated holders are fitted in the steam and water drum, and also in the bottom or mud drum. These boilers are possessed of several advantages, steam being quickly raised, but to maintain this efficiency it is necessary to keep boiler in a clean condition, which necessitates the cleaning of the outside of the tubes as often as required. This operation can be carried out while boiler is steaming, as in the Yarrow boiler.

Water tube boilers require to be kept as clean as possible owing to the possibility of the tubes silting up, and require constant attention as regards treatment with lime, &c.

The lower two rows of tubes are about 4 inches diameter, and the others $2\frac{1}{4}$ inches diameter; the larger size of the lower rows reduces the tendency to upward bending, due to the intense heat, a fault common to the lower tube rows in all water tube boilers, the severe expansion produced by the high temperature acting to bend the tubes *away* from the heat.

The larger sized lower tubes also allow better for the scale deposit, which, if the feed is of any appreciable density, quickly forms on these tubes.

Both of the above described boilers are specially suited for the burning of oil fuel, which is now in general practice in the Navy (see page 609).

The special advantages of water tube boilers as compared with the ordinary cylindrical type are as follows:—

Advantages.

1. Suitability for high pressures (often 300 lbs. per square inch).
2. Less weight for the same power.
3. Greater safety in event of accident owing to the smaller amount of water carried.
4. Quicker raising of steam (about one hour is the time usually required).

As a set-off against the above stated advantages it should be mentioned that the water tube boiler requires more skilful firing than ordinary, also careful attention to the feeding is necessary, the amount of water carried being so small that should anything temporarily check the feed supply the water might all evaporate in a very short time and the boiler become empty, with the consequent danger of explosion. In some of the types of water tube boilers in use, the lower sets of tubes are liable to become overheated and damaged by oil or scale deposits, and for this reason the feed water has to be kept as pure as possible, and the boilers run at a very low density.

Disadvantages.

1. More skilful firing required.
2. Regular feeding.
3. Pure feed water necessary.
4. Large number of joints to be kept tight (in certain types).
5. Small amount of water carried, which would quickly evaporate if feed supply is temporarily checked.
6. Difficulty in cleaning the tubes of scale deposit.

General Construction.

The construction of a water tube boiler consists, in general, of a steam drum at the top, connected by means of straight or curved tubes to the water and mud drums at the bottom ; in some cases the upper ends of the tubes open into the steam space of the drum, and in others into the water space.

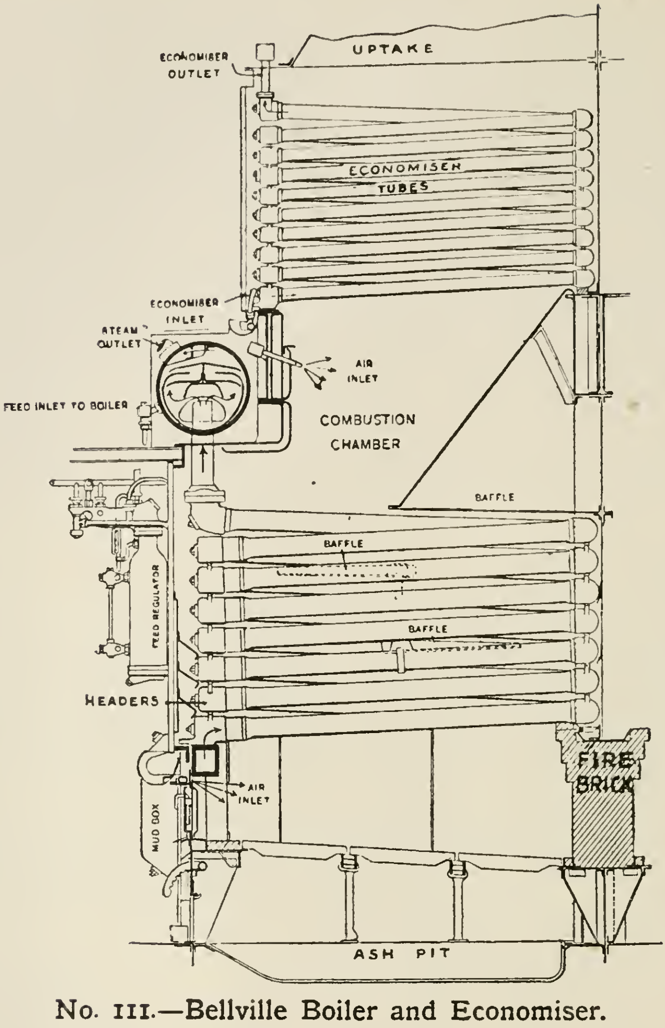
The feed check valve is placed on the top or steam drum, and as the water enters the drum it falls through the down-take pipes to the sediment collector at the bottom, where the dirt is deposited and afterwards blown off. The cold water becoming heated and evaporating rises up through the sets of tubes or "elements" as they are termed, and passes into the drum at the top in the form of steam. The mountings of water tube boilers are similar to those of the cylindrical type, and it should be stated that all the various parts forming the boiler are covered in by an iron casing. In some types, such as the "Bellville," feed water heaters or "economisers" are fitted in the uptake, and usually consist of a series of tubes through which the feed water passes before entering the steam drum, and is consequently raised in temperature by the otherwise waste gases of combustion. This heating of the feed water increases the steaming power of the boiler and reduces the consumption.

"Bellville" Boiler.

In the "Bellville" type of water tube boiler the tubes are straight, but lie at a slight angle, the ends being connected, and so form an "element." Each "element" consists of a set of tubes forming a zigzag from the water drum below to the steam drum above. The front ends of the tubes are fixed into "headers," and the back ends, as before stated, are connected to form the spiral arrangement. Doors are fitted at the ends of the tubes for purposes of examination and cleaning.

The tubes, which are made of good iron or mild steel, vary in thickness according to the position they occupy, the lower sets being made thicker than the upper ones, to withstand the intense heat to which they are subjected, and which has the effect, in some cases, of causing them to become bent ; this is most likely to happen when deposits of oil or scale form in the tubes.

The mountings, as before stated, are similar to those of the ordinary marine type of boiler, with the exception perhaps of the reducing valve which is fitted to the "Bellville" type, as the boiler pressure carried is usually in excess of that required in the engines—often 250 lbs. pressure in the boiler, which is reduced to 200 or 180 for the H.P. valve chest. The "Bellville" boiler is supplied with a special feed regulator, consisting of a chamber containing a float in connection with the water level in the boiler ; the float connects with a system of levers, which in turn are in connection with the feed regulation valve, and as the float rises and falls with the amount of water contained in the upper drum, the levers open or shut the feed valve, and so regulate the water supply to the requirements of the boiler.



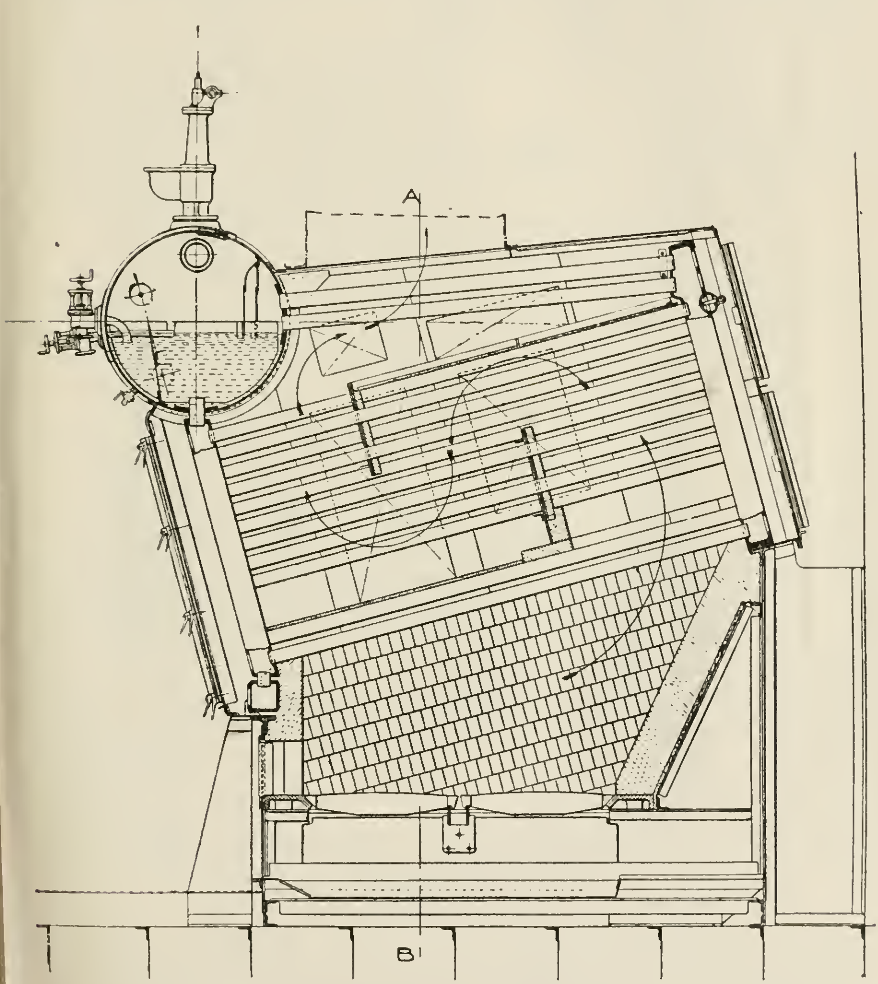
No. 111.—Bellville Boiler and Economiser.

As mentioned previously, careful feed regulation is one of the most important points to be attended to for the successful working of this class of boiler.

"Babcock & Wilcox" Boiler.

A general idea of the construction of the "Babcock & Wilcox" water tube marine boiler will be obtained by referring to the illustrations, which clearly show the various parts.

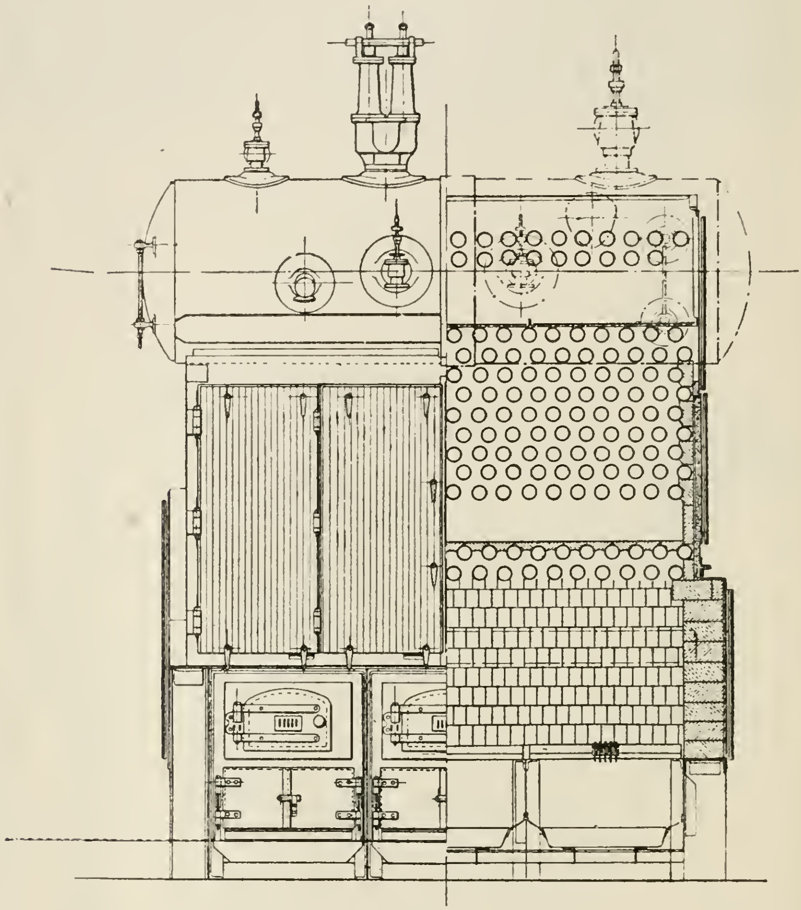
The boiler is constructed entirely of wrought steel, and consists of a series of straight water tubes placed in an inclined position, under which the furnace is situated; the tubes are expanded at each end into boxes of sinuous shape called "headers." Opposite each tube there is a separate handhole in the "headers" for inspection and cleaning of the tube, and it is peculiar to note that no stay tubes are fitted. The upward and downward headers are in communication at the top end with the steam and water drum, the downward headers being connected with a mud drum at the bottom end, which is fitted with blow-off cocks for clearing out the sediment which collects there.



LONGITUDINAL SECTION

No. 112.

At the sides the boiler has sets of inclined tubes arranged slightly different from the centre series, but forming with them the effective heating surfaces of the boiler. The furnace sides are lined with fire-bricks, and the boiler itself is covered in by a light wrought-iron casing



FRONT ELEVATION SECTION AT A B

No. 113.

which can be removed when it is necessary to obtain access to the tubes, &c., for repair or cleaning. In this type of boiler the circulation is particularly well provided for, the water rising up through the inclined tubes, past the uptake headers, and into the steam and water drum, and returning by means of the downward headers. The mud drum at the bottom traps the impurities, such as sediment, &c., and these are blown out of the collector by the cocks fitted for that purpose.

The joints are all metal to metal, and in the case of the tubes the ends are simply expanded into the plates, no screwed joints being used.

Expansion of the boiler under heat is allowed for by the manner in which the mud drum is held down to the foundations.

Schmidt Type Superheater.

For marine practice this type of smokebox superheater has proved fairly satisfactory, and has recently been fitted in the boilers of quite a large number of new vessels, including many supplied with geared-down turbines, in which a moderate degree of superheat (say from 100° to 150°) is found sufficient.

The following data of superheat working is taken from a large quadruple expansion engine set, and the results obtained in this case were very satisfactory indeed; the economy of superheated steam over saturated steam showing clearly on the coal consumption.

Superheat Data.

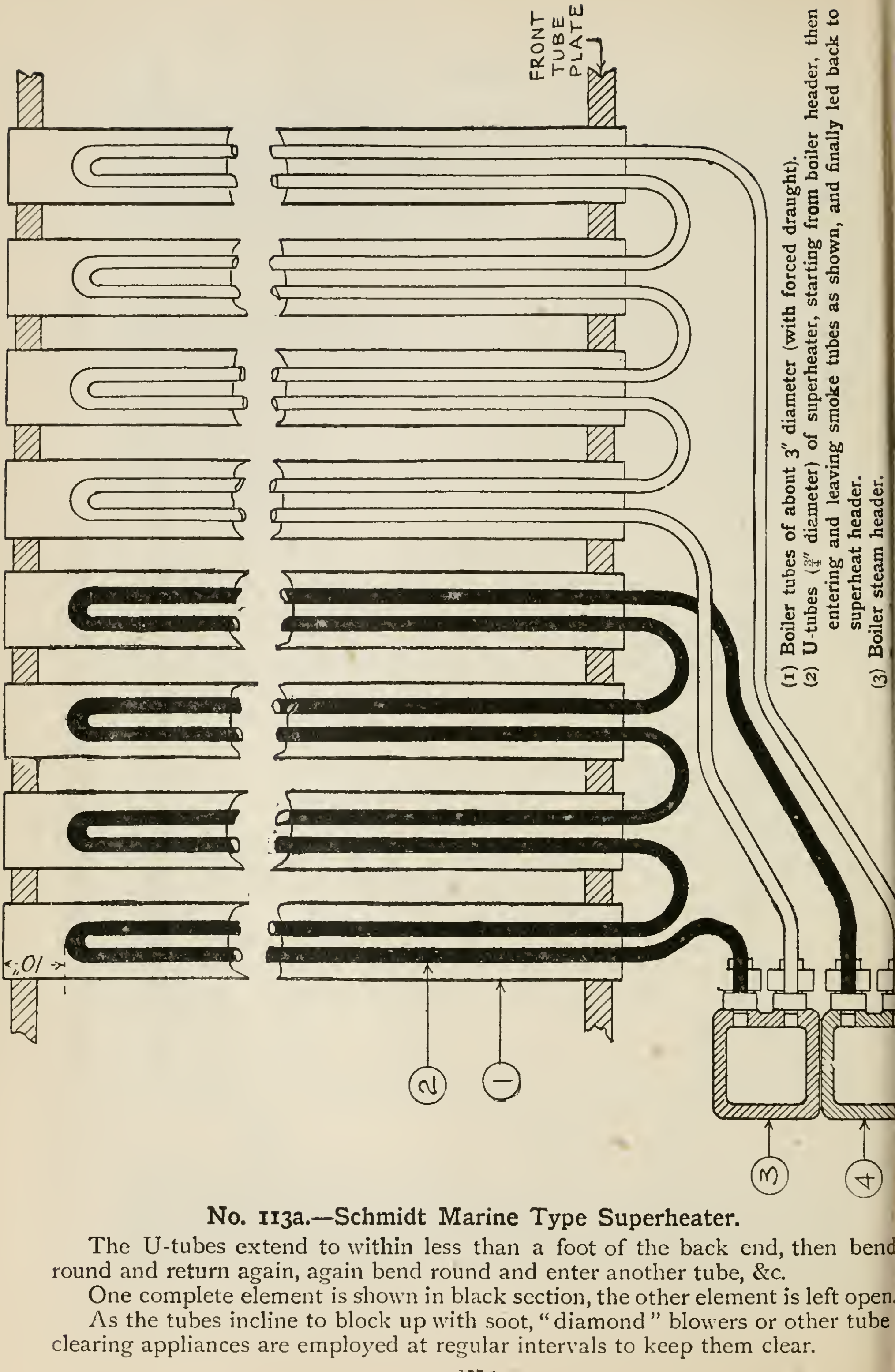
Type of engines - - - -	Quadruple expansion.
I.H.P. - - - - -	4200
Boilers - - - - -	4 single-ended.
Diameter of smoke tubes - - -	3″
Boiler pressure - - - -	220 lbs. gauge.
Diameter of superheater tubes - -	¾″
Draught - - - - -	Howden forced.
Air pressure at fan - - -	2¾″
,, under bars - - -	¾″
,, above ,, - - -	⅜″
Revolutions of fan - - -	260 p.m.

Data of Pressures and Temperatures with Superheat.

Position.	Gauge Pressure in Lbs. per Sq. In.	Temperature for Pressure.	Temperature as Tested.	Degree of Superheat.
Boiler steam - -	220	396°	590°	194°
H.P. steam - -	210	392°	580°	188°
1st I.P. steam -	110	344°	440°	96°
2nd I.P. steam -	47	295°	295°	0°
L.P. steam - -	9·5	236°	230°	– 6°

Smoke box temperature - -	385°
Uptake temperature - - -	370°
Funnel ,, - - -	355°

It should be noted that the superheating of the steam by the waste gases extracts the heat of the latter, and lowers the gas temperature, as shown by the recorded results; this again has the effect of reducing the amount of heat available for the air heating tubes of the forced draught system.

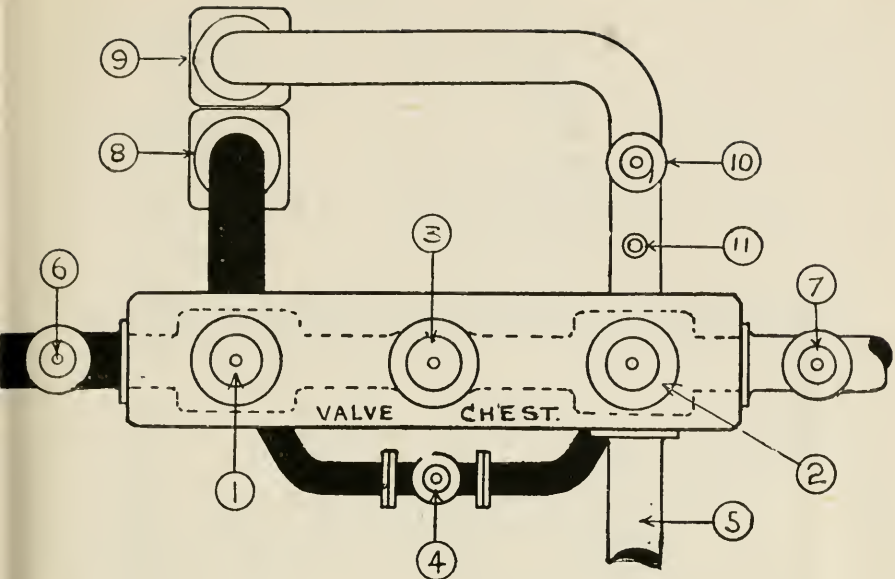


No. 113a.—Schmidt Marine Type Superheater.

The U-tubes extend to within less than a foot of the back end, then bend round and return again, again bend round and enter another tube, &c.

One complete element is shown in black section, the other element is left open.

As the tubes incline to block up with soot, "diamond" blowers or other tube clearing appliances are employed at regular intervals to keep them clear.



No. 113b.—Schmidt Superheater Valve Chest.
(Fitted on Top of Boilers.)

(1) Main stop valve giving steam from boilers either to superheater header box, or to engines direct through bye-pass valve (3).

(2) Superheater steam stop valve to engines.

(3) Bye-pass valve, for direct boiler steam to engines, through (1), (3), (2), and (5).

(4) Mixing valve for giving a mixture of boiler steam and superheated steam as may be found necessary.

(5) Main steam pipe to engines (either boiler steam or superheated steam).

(6) Auxiliary boiler steam stop valve.

(7) ,, superheated ,, ,, ,,

(8) Boiler steam header (before superheat).

(9) Superheat ,, ,, (after ,,).

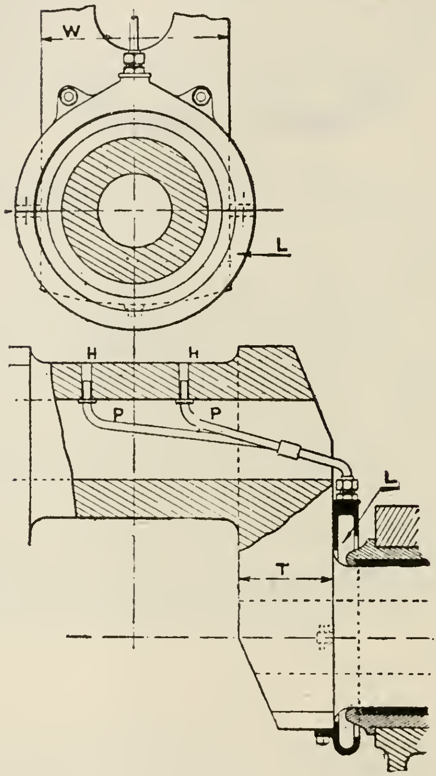
(10) Safety valve (single) which may be required if valve (2) is shut and steam is being passed through valve (1) and bye-pass (3), that is, the superheater shut off from boilers.

(11) Fitting for connecting up pyrometer (high temperature indicator).

NOTE.—With superheater off when working bye-pass, it is advisable to have main stop valve (1) eased off seat to allow a little moisture to be present in the U-tubes of the superheater, otherwise damage to the tubes, &c., by dry steam at high temperature is likely to take place.

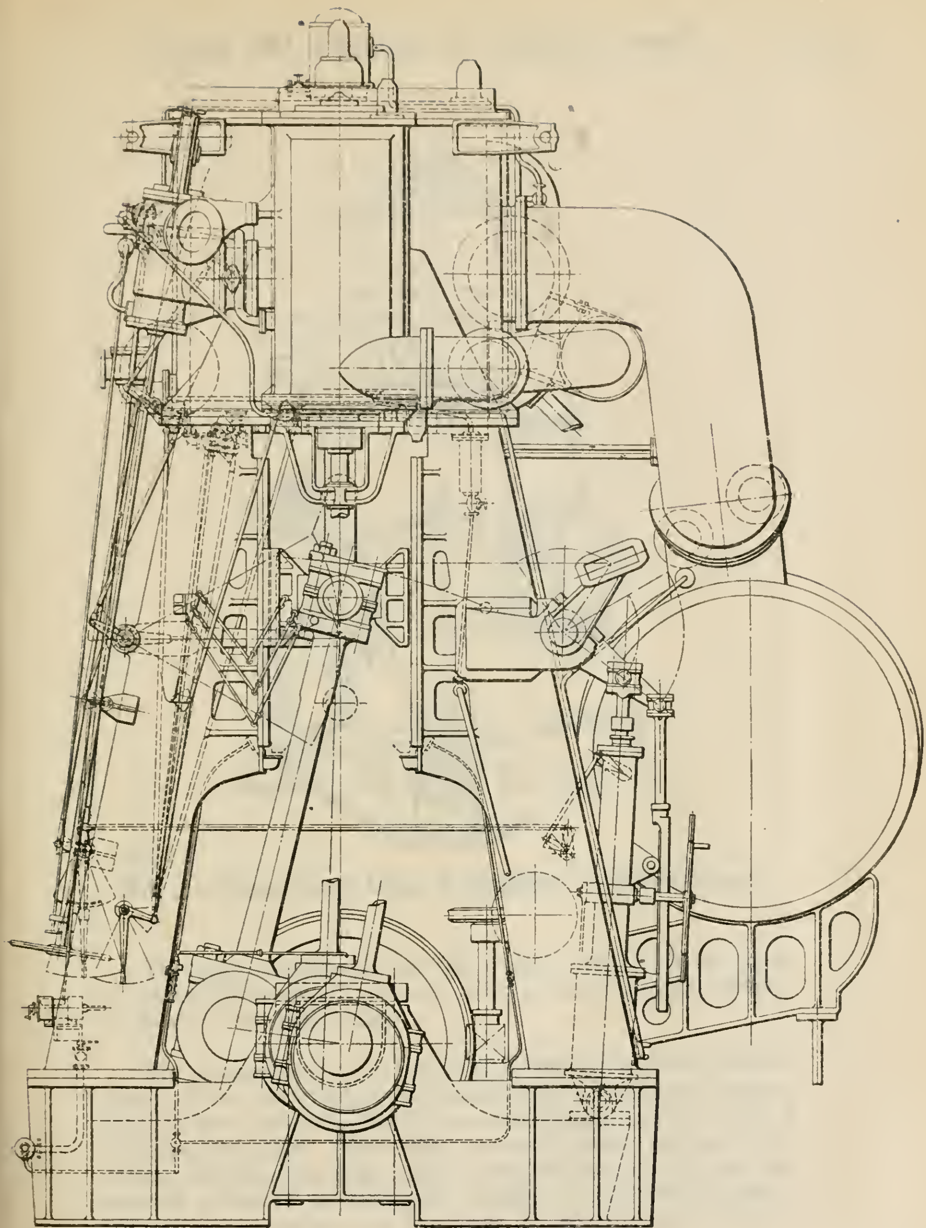
SECTION III.

NOTES AND SKETCHES OF VARIOUS DETAILS.



No. 1.—Crank-Pin Centrifugal Lubrication System (Naval Practice).

Oil is fed into the container L, and by the action of centrifugal force is delivered to the crank-pin bearing surfaces H, H through the pipes P, P shown. Radial holes are cut through from the inside of the crank-pin.

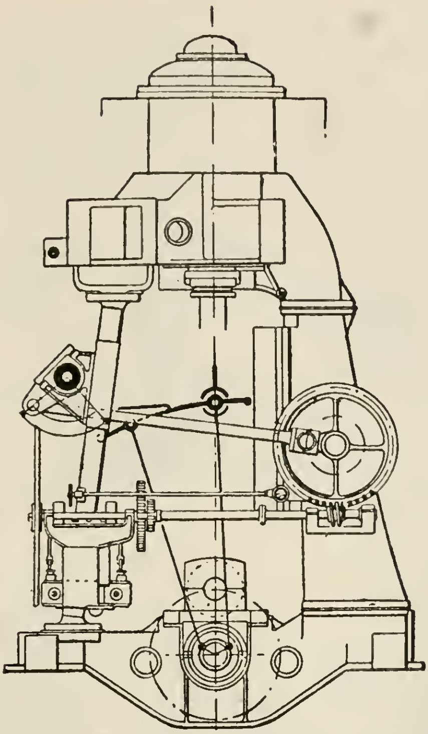


No. I A.—Triple Expansion Engines.

View from H.P. end, looking Aft.

Observe that the expansion slot of the reversing crank arm is in a slightly inclined position for "ahead," but when run over to "astern" position (as shown by the dotted arc) the slot will be vertical, thus ensuring that full-gear conditions are obtained when going astern, no matter how much the gear was shut in when running ahead.

[*To face page* 178.

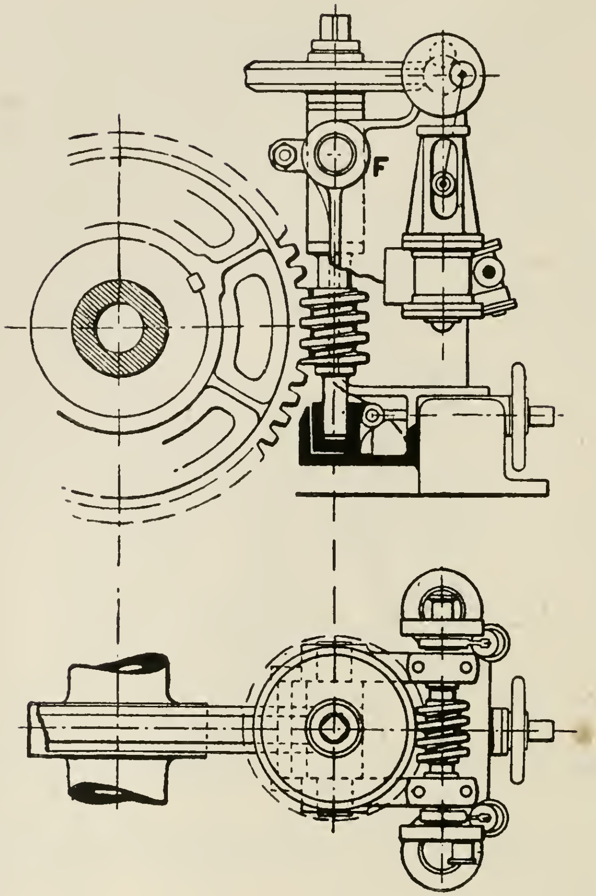


No. 2.—Reversing Gear Complete (Cruiser Type).

This gear is known as the "all round" type, as if the gear is "missed," the wheel continues moving round without damage or shock to the link motion.

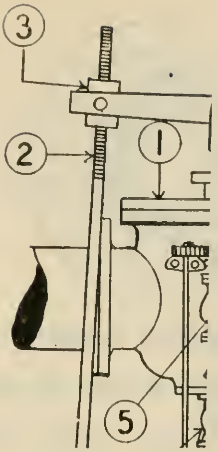
For an engine of 1200 I.H.P. the reversing engine cylinders (two) will be about $4\frac{1}{2}$ inches diameter by 4-inches stroke, and from 20 to 25 revolutions of the reversing engine will be required to reverse the gear. The engine is usually made reversible by means of a hand operated piston valve, similar in design to the control valves of steering gear engines, which admits steam either to the centre or to the ends of the cylinder valves as required.

The reversing engine shown on left drives a worm shaft geared into a large worm-wheel, to which is attached the link from the bell crank. Hand-wheel gear is also shown, and a brake strap connection to the worm-wheel, to increase the control.



No. 3.—Twin Cylinder Turning Engine with Double Worm Gear.

The lower end of the main worm can be drawn out of gear by means of the wheel shown on right. The turning engine runs at about 300 revolutions per minute, and travels about 2000 revolutions for one revolution of the main engine, thus requiring $6\frac{1}{2}$ minutes to turn the main engines once round, as $2000 \div 300 = 6{\cdot}6$ minutes. For engines of, say, 1500 I.H.P. the turning engine cylinders will be about $3\frac{1}{2}$ inches diameter and 4 inches stroke.



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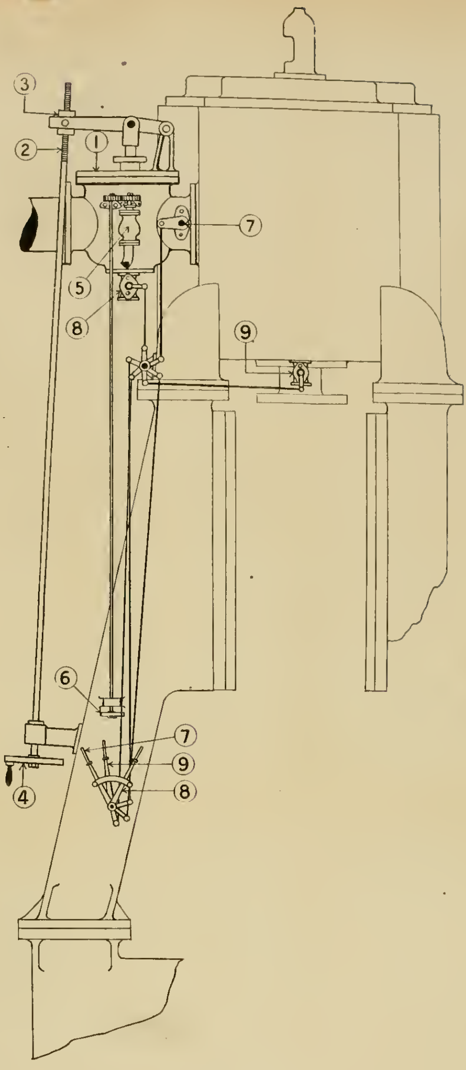
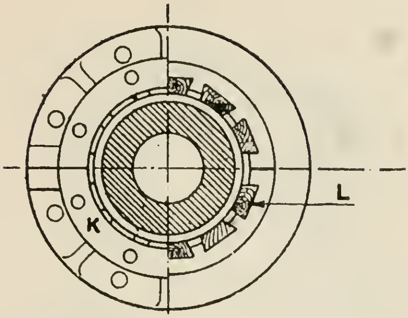


No. 4.—H.P. Main Steam Connections, &c.

1, Engine-room stop valve.
2, Screwed rod to wheel below.
3. Nut on stop valve lever.
4. Wheel actuating stop-valve lever by means of nut 3.
5, Heating up steam valve (used before starting).
6, Wheel for operating heating-up valve.
7, Throttle valve.
8, Drain on stop valve.
9, Drain on cylinder bottom.

NOTE.—The handles shown below for working the throttle valve, stop-valve drain, and cylinder drain are numbered similarly to the connections named.

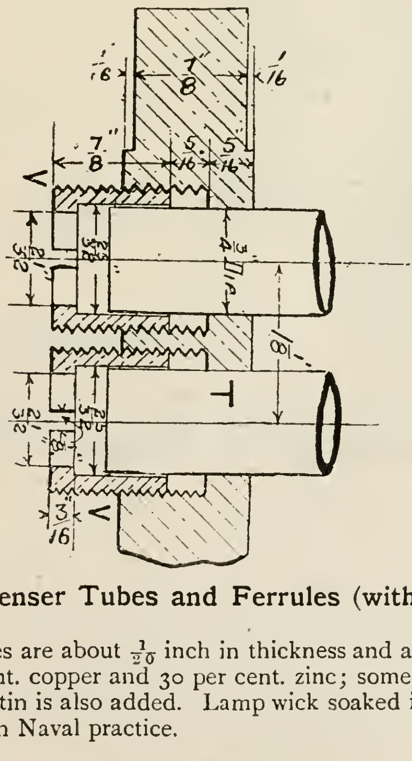
[To face page 180.



No. 5.—Section of Stern Tube showing Lignum Vitæ Strips.

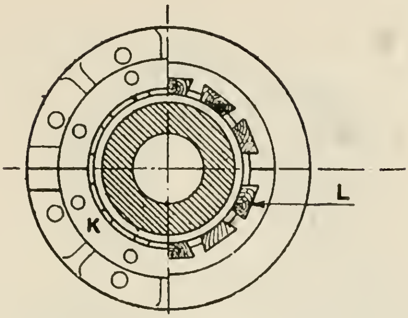
K, Check plate to keep strips in position aft. L, Lignum vitæ strip.

The shaft (hollow) is shown in section. The clearance spaces between the strips is to allow of the admission of water for lubrication inside the tube. Lignum vitæ wears better than brass when grit or sand is present in the water.



No. 6.—Condenser Tubes and Ferrules (with dimensions).

The tubes are about $\frac{1}{20}$ inch in thickness and are composed of 70 per cent. copper and 30 per cent. zinc; sometimes a small per cent. of tin is also added. Lamp wick soaked in oil is used as packing in Naval practice.



No. 5.—Section of Stern Tube showing Lignum Vitæ Strips.

K, Check plate to keep strips in position aft. L, Lignum vitæ strip.

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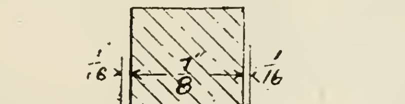

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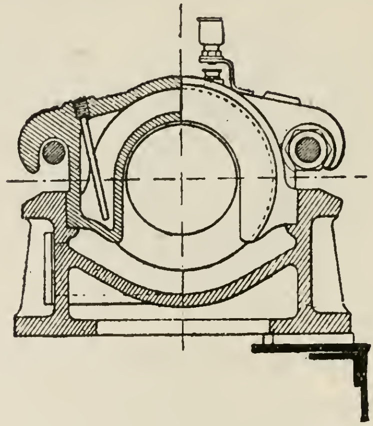


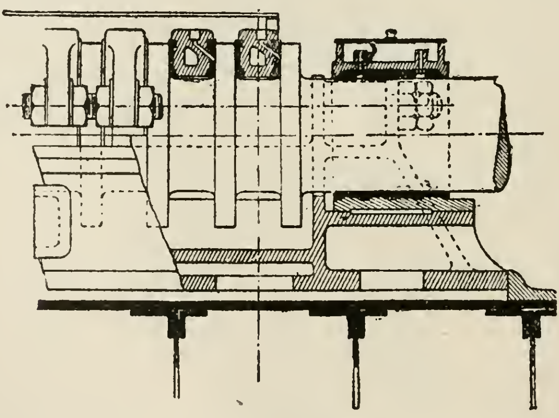
No. 7.—End View of Thrust Block Shoe.

The right half shows the white metal bearing surface of the shoe, and the left half the hollow cast interior with cooling water connection. The seating of the block for bolting down is also shown on the right.

The pressure on the shoes is usually about 50 lbs. per square inch, and the total pressure on the thrust is estimated as follows:—

$$\frac{\text{I.H.P.} \times 33000 \times \frac{2}{3}}{\frac{\text{ship knots} \times 6080}{60}} = \text{total lbs. on thrust.}$$

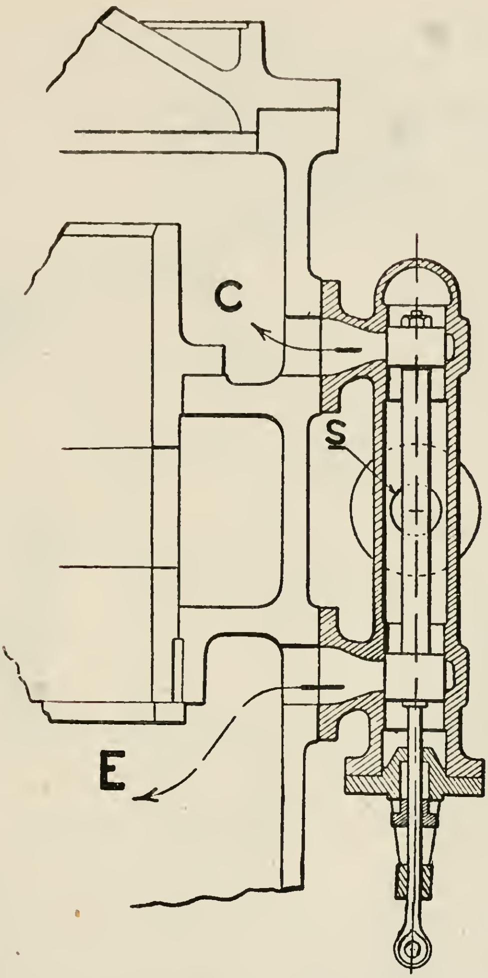
NOTE.—Two-thirds of the total I.H.P. is assumed as the effective power applied to the thrust block.



No. 8.—Thrust Block (Part Section) with End Bearing.

The thrust seating is clearly shown, and the check angle plates fitted at the ends to secure the block in position. The oil service to the white metal surfaces of the shoes is also shown. Each shoe has separate adjustment by means of the two horizontal studs and double nuts shown.

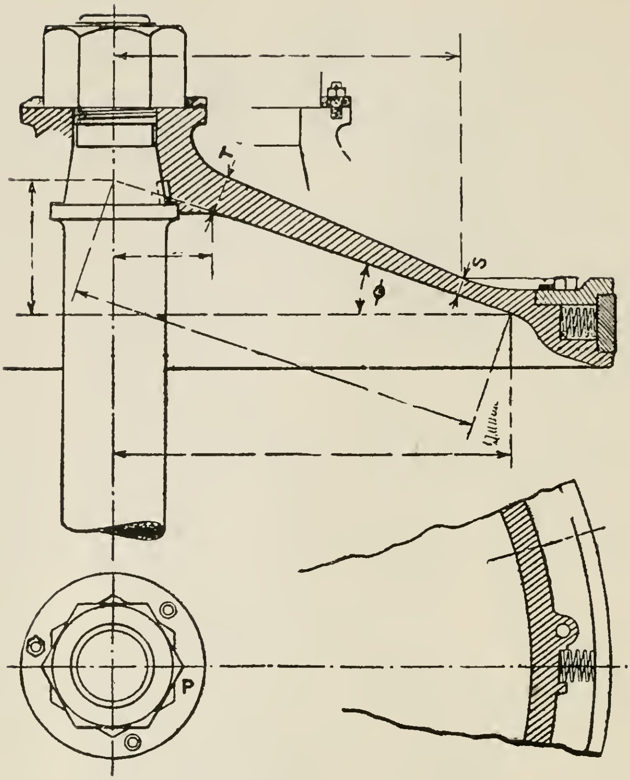
NOTE.—With engines running ahead the pressure is on the after surface of the thrust rings for either a right or left hand propeller.



No. 9.—I.P. Cylinder Starting Valve.

S, Steam to valve. C, Steam to I.P. receiver.
E, Steam to L.P. receiver.

By means of the valve shown, the live steam can be given to either the I.P. or L.P. receiver to assist the starting of the engines. The I.P. cylinder illustrated is fitted with a piston valve (outside steam), the liner for the top end only being shown in the sketch.



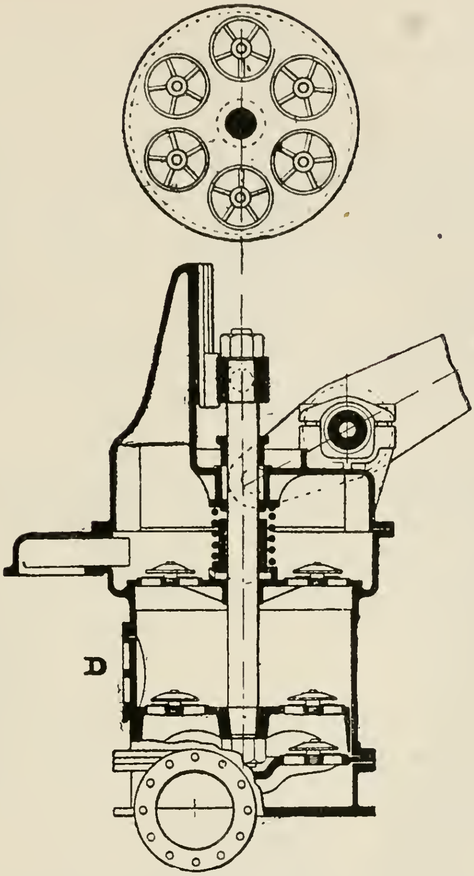
No. 10.—L.P. Piston (Naval Type).

P, Ring plate, to hold piston rod nut in place against screwing back.

The piston ring is forced outwards by a number of small spiral springs fitted in recesses in the piston body, the pressure exerted being about 2 lbs. per square inch. The junk ring and piston flange are checked to prevent the piston ring from coming in.

The junk ring J is secured to the piston by steel collar studs with gun-metal nuts, which are held in place by a steel guard ring G. The guard ring G is again secured by square-necked studs with nuts and split pins.

For an L.P. piston $81\frac{1}{2}$ inches diameter, the piston thickness $T = 2\frac{3}{8}$ inches, and $S = 1\frac{1}{4}$ inches.

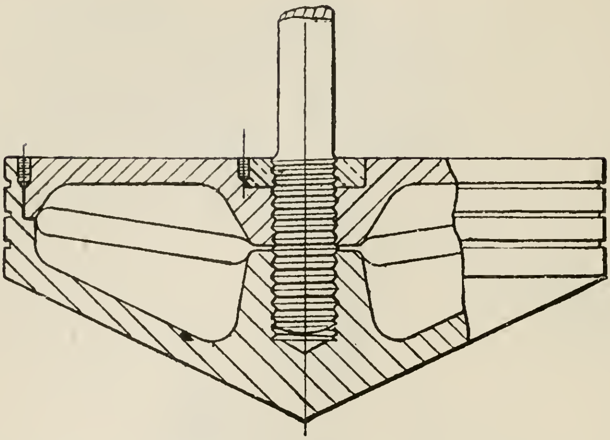


No. 11.—Air Pump Driven by Separate Lever (Naval Type).

D, Inspection Door.

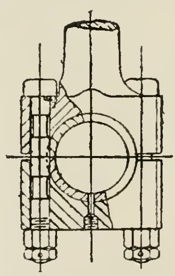
On up stroke, foot valves and head valves open.
On down stroke, bucket valves open.

The air and vapour together with the condensed water is removed by the air pump, thus reducing the pressure in the condenser below that of the atmosphere, the result of which is to increase the M.E. pressure on the L.P. piston and the work done by the engine.



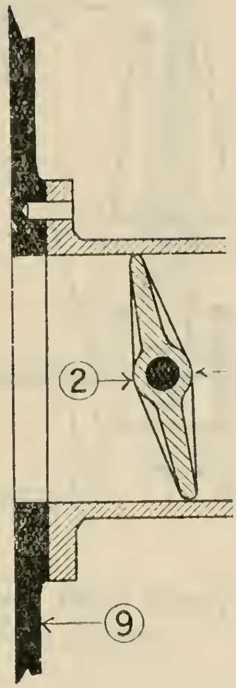
No. 12.—Displacement Type Air Pump ("Edwards" Type) Bucket with Water Packing Grooves.

Air pump buckets are often packed as shown, in place of the usual rope packing.



No. 13.—Valve Spindle Eye Bush.

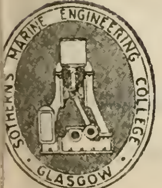
The bolts shown are reduced in diameter between the bearing parts similar to those fitted in connecting rod bottom ends.

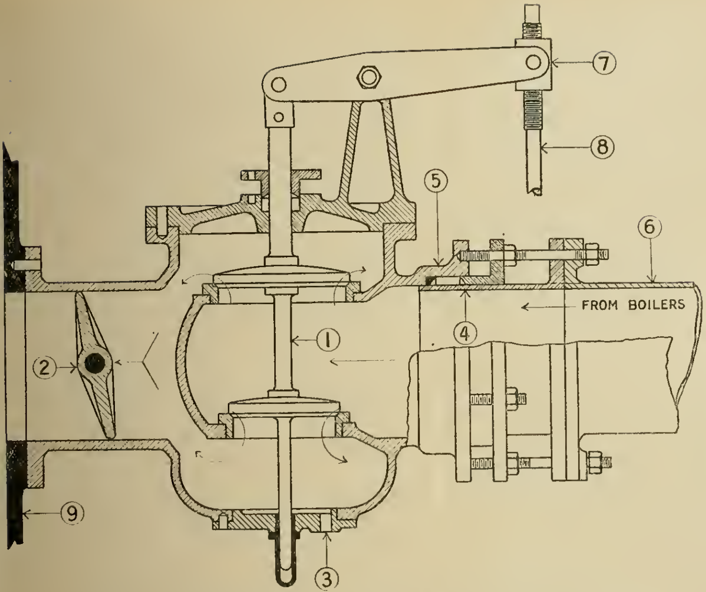


No. 14

1, Double-beat or
2, Throttle or butt
3, Drain.

The valve sl
the steam enteri
arrangement allc
at the top valv
the valve.
The expans
long studs nutte
check collar, stu
steam on, say, a
The drain i
accident when s





No. 14.—Double-Beat Valve, Throttle Valve, and Expansion Joint.

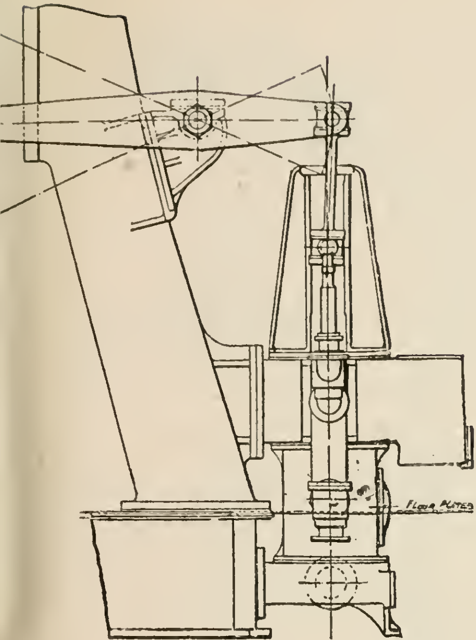
1, Double-beat or equilibrium valve.
2, Throttle or butterfly valve.
3, Drain.
4, Brass internal pipe.
5, Cast iron.
6, Copper or steel pipe.
7, Nut.
8, Spindle down to hand-whee.
9, H.P. chest.

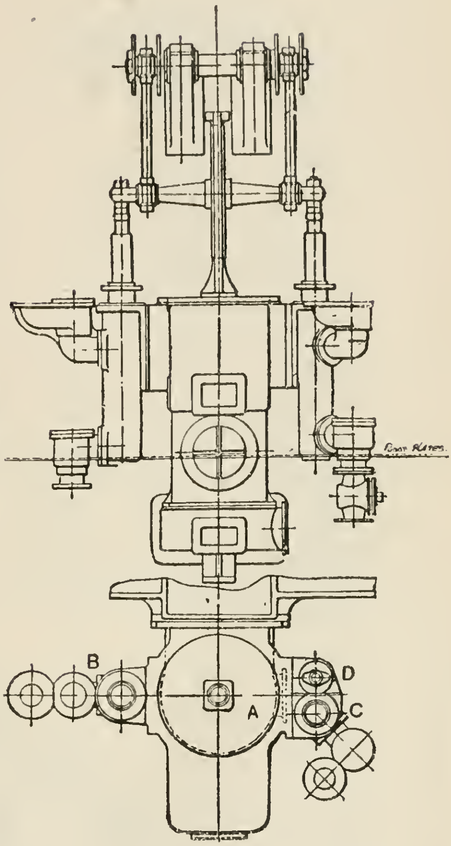
The valve shown is commonly fitted as an engine-room stop-valve, and is of the balanced type. the steam entering from the centre and flowing out by means of the lower and upper valves; this arrangement allowing of easy manipulation. The chief drawback is the tendency to leakage at the top valve, due to unequal expansion of the brass spindle and the cast-iron chest of the valve.

The expansion joint consists of a small stuffing-box gland and safety collar with at least two long studs nutted as shown; the internal portion of the steam pipe is separate and of brass. The check collar, studs, and nuts shown prevent the pipe from being blown out by the action of the steam on, say, a bend of the pipe.

The drain is an important fitting, as by neglect of its use water may accumulate and cause accident when steam is turned on by the action known as "water hammer."

[To face page 186.





No. 15*—Column, Pump Lever, Air and Feed Pumps.

The sketch shows the usual arrangements of the pumps when driven by levers, the circulating pump being independent and of the centrifugal type. Observe the guide for the pump crosshead, also the heavy links from pump lever to crosshead. The average sizes of pumps for an engine of 1200 I.H.P. would work out as follows :—

Cylinders, 24″, 40″, 66″; stroke, 42″.

Pump strokes = 21″.

Then, Air pump volume = $\frac{1}{16}$ of L.P. cylinder volume.

And, Feed pump volume (each) = $\frac{1}{700}$,, ,,

1, Diameter of air pump $= \sqrt{\dfrac{66^2 \times 42}{16 \times 21}} = 24''$ (nearly).

2, Diameter of each feed pump $= \sqrt{\dfrac{66^2 \times 42}{700 \times 21}} = 3{\cdot}5''$.

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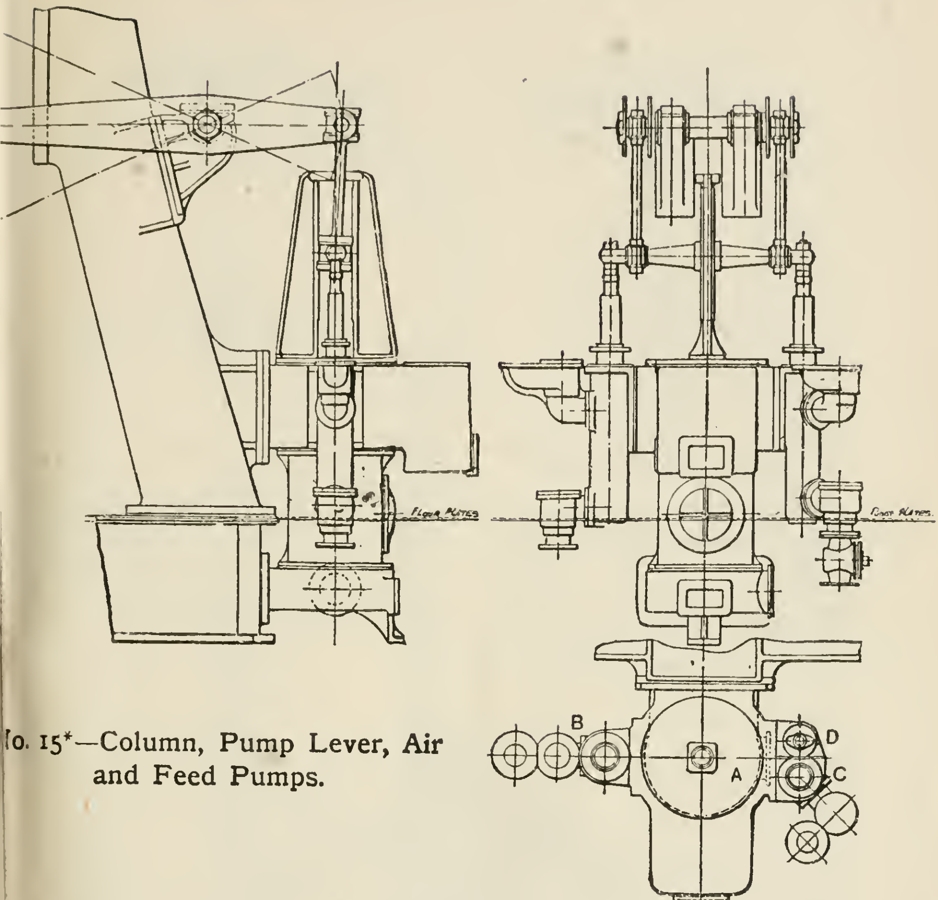

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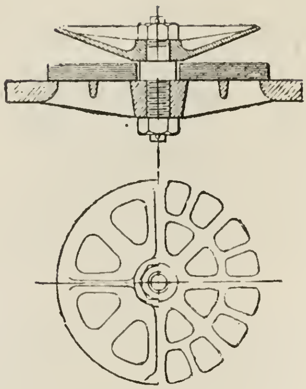
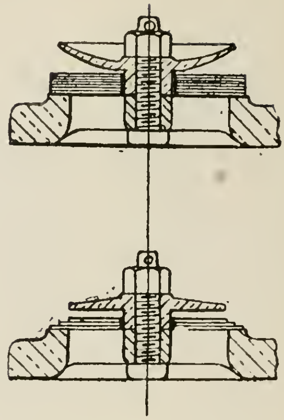


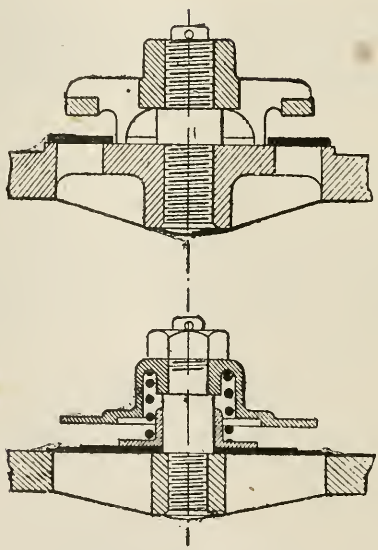
No. 16.—Air Pump Valve (Indiarubber Type).

The plan shows the holes formed in the valve guard (left half) through which the air pressure forces the valves back on to their seats. The right half of the plan shows the holes in the grating through which the vapour and water passes.



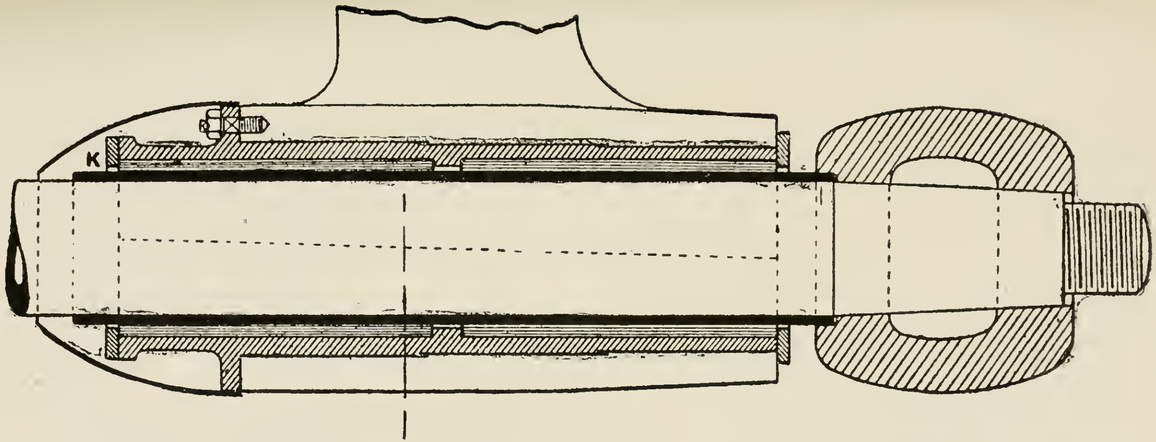
No. 17.—Air Pump Valves.

The upper valve is of rubber, the lower one is metallic ("Kinghorn"), the lift of the latter being about $\frac{1}{8}$ or $\frac{3}{16}$ inch and the lift of the former about $\frac{3}{4}$ inch at outer circumference of valve.



No. 18.—Air Pump Valves (Metallic Type).

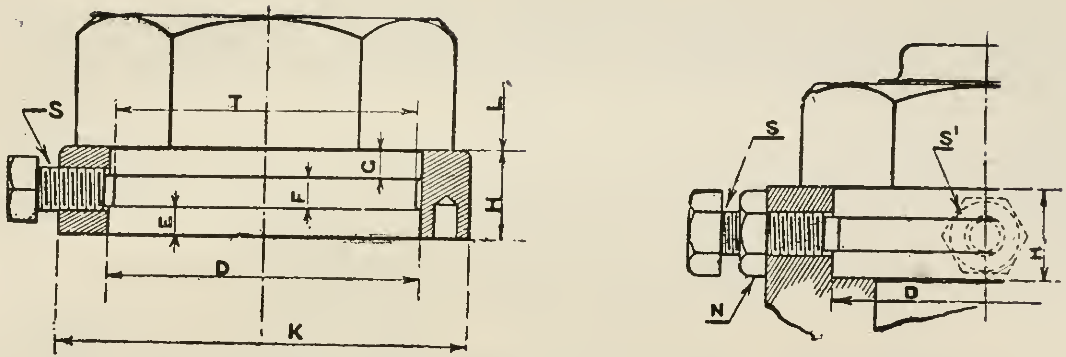
The lower valve shown is filled with a light spring.



No. 19.—"A" Bracket Bearing of Twin or Triple Screw Steamer.

A small tube is fitted through the "A" bracket and lined with lignum vitæ (in two lengths), check plates K being pinned on either end; the tube is secured by means of a flange and studs to the bracket.

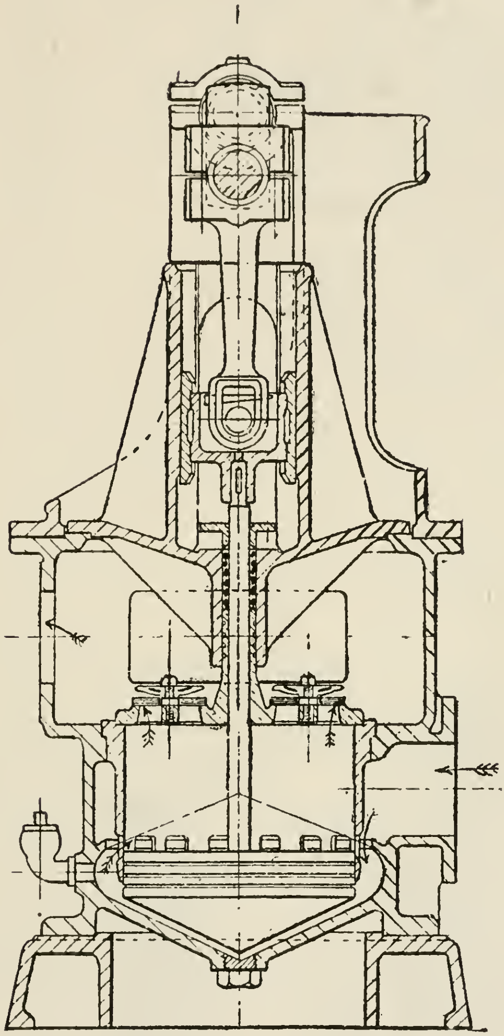
NOTE.—As brackets of this design were found to affect the speed of larger steamers by the resistance set up, the latest system is that of "bossing" or carrying out the plating of the lines aft to meet the bracket; this arrangement reduces the resistance and proportionally increases the speed for a given power.



No. 20.—Dimensions of Connecting Rod Bolts and Nuts.

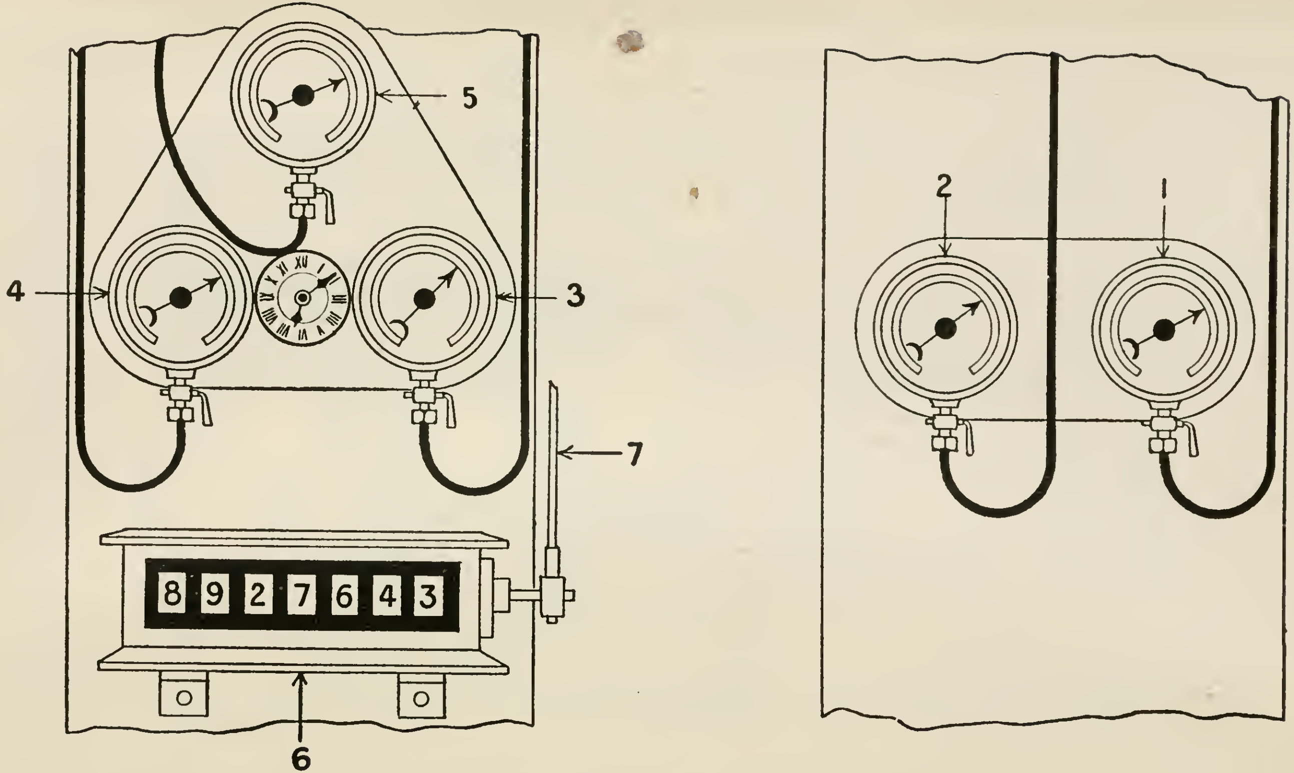
For a bolt $3\frac{1}{2}$ inches diameter the dimensions are as follows:—

Diam. of Bolt.	No. of Threads per inch.	Diam. at Bottom of Thread.	Diam. of Bolt at Reduced Part.	Area of Bolt at Reduced Part.	D.	E.	F.	G.	H.	L.	Width over Flats.	Diam. of Locking Pin.	Diam. of Locking Ring.	Bolt Head Diam.	Bolt Head Thick-ness.
In.		In.	Sq. In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.
$3\frac{1}{2}$	$3\frac{1}{4}$	3·106	$3\frac{1}{16}$	7·366	$4\frac{3}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{3}{8}$	$1\frac{1}{4}$	$2\frac{1}{4}$	5	$\frac{5}{8}$ -	$6\frac{1}{4}$	$4\frac{3}{4}$	$2\frac{5}{16}$



No. 21.—Edwards Type Air Pump (with Displacement Bucket).

Observe the air inlet ports near the bottom, also that head valves only are fitted. The pump as shown is independent and is driven by a separate engine (Naval practice).

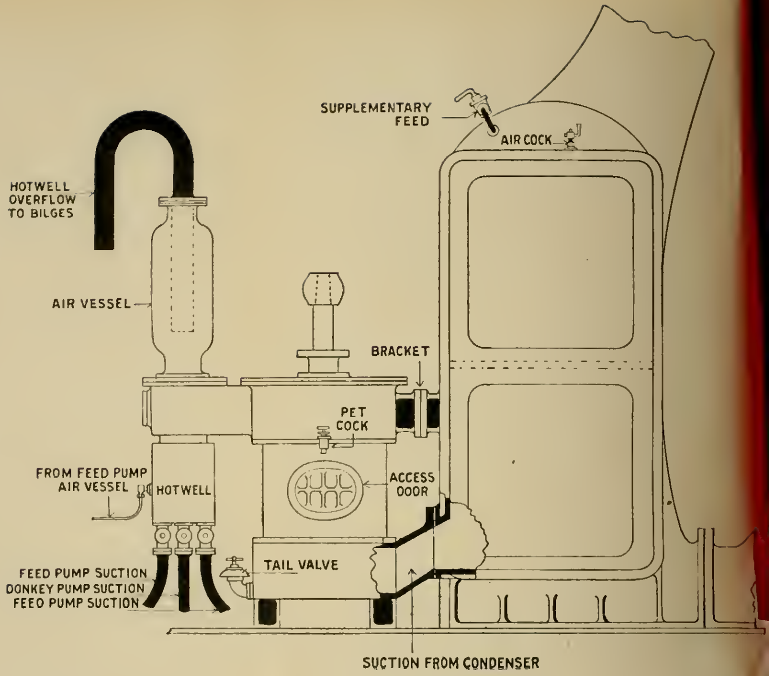


No. 22.—Engine-Room Pressure Gauges.

1, Main steam (say 190 lbs.).
2, H.P. receiver (say 185 lbs.).
3, I P. receiver (say 65 lbs.).
4, L.P. receiver (say 12 lbs.).
5, Vacuum gauge (say 25 inches).
6, Counter.
7, Counter connection gear to engine.

The above are the usual pressure gauges fitted in the engine, either on the columns as shown, or

NOTE.—The astern guide surface = 80 per cent. of ahead guide surface.



No. 23.—Condenser and Air Pump Connections.

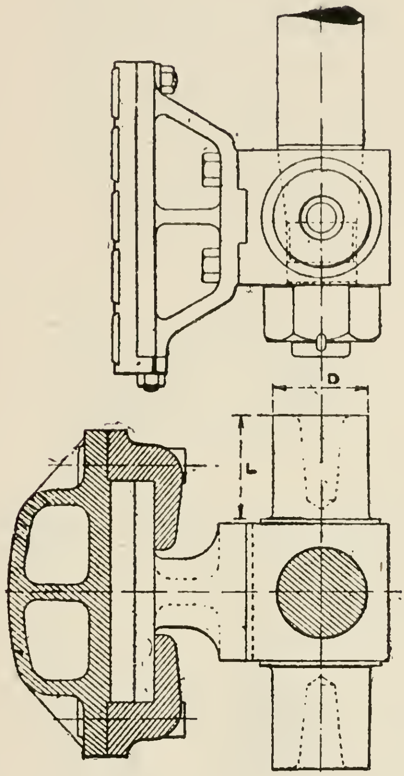
After the exhaust steam from the L.P. cylinder is condensed, the water of condensation falls to the bottom of the condenser, and together with the air and vapour present is drawn out by the air pump, passing successively through the foot valves, bucket valves, and head valves into the hot-well, some of the air and vapour escaping through the hot-well overflow pipe; the water is drawn off by the feed pumps, and passing through the suction valve and delivery valve, is forced into the feed heater. In case of accident to the main feed pumps the general service donkey is usually arranged to draw from the hot-well, if required, as shown in the sketch.

Pet Valve—The pet valve is placed just under the head valve so as to draw in air for cushioning purposes in the *down stroke* only, thus leaving the condenser vacuum unaffected.

Tail Valve.—This valve opens outwardly and is intended to relieve the pump from over pressure of water.

NOTE.—Occasionally the general service donkey is also arranged to draw from the bottom of the condenser in case of breakdown of the air pump: this is a standard connection when Weir pumps are fitted.

[*To face page* 193.

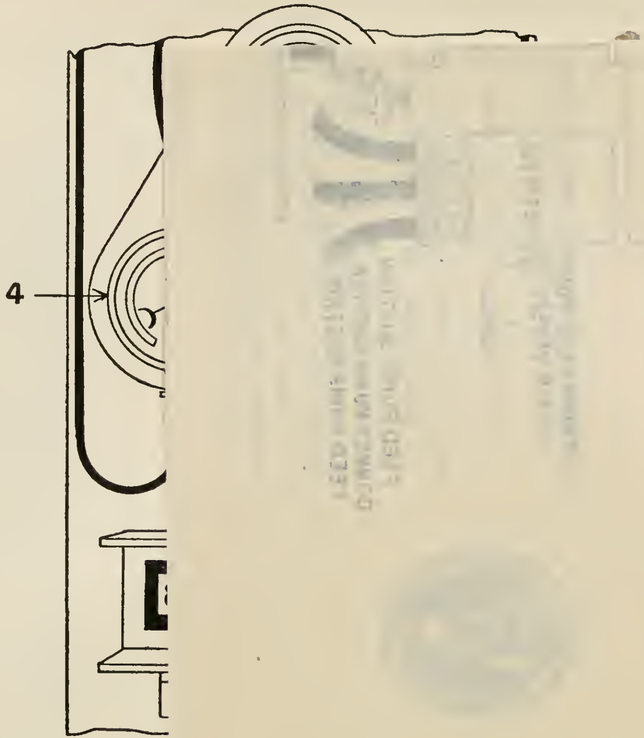


No. 24.—Piston Rod Crosshead and Shoe ("Single" Guide Type).

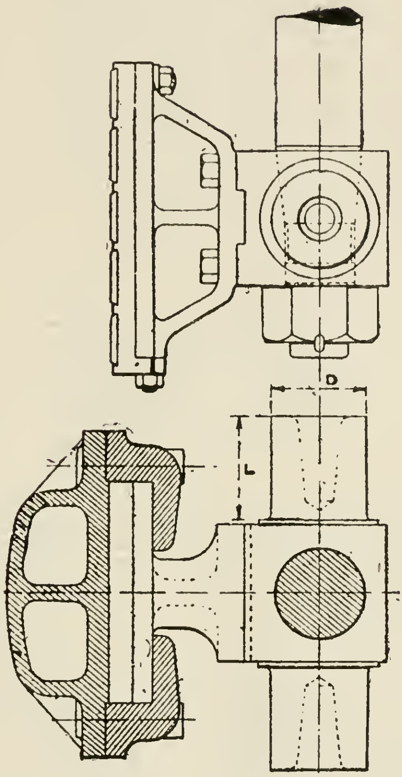
D = Crank pin diameter × ·55.

L = D × 1·2.

NOTE.—The astern guide surface = 80 per cent. of ahead guide surface.



The above

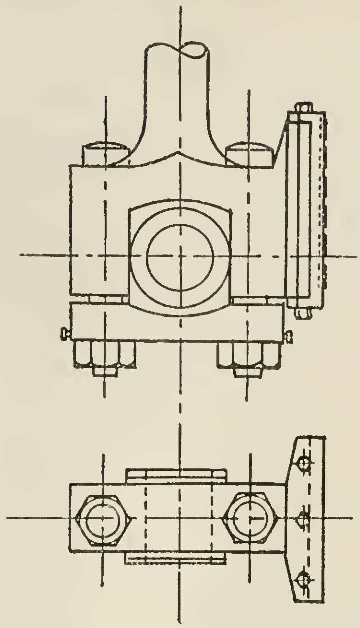


No. 24.—Piston Rod Crosshead and Shoe ("Single" Guide Type).

D = Crank pin diameter × ·55.

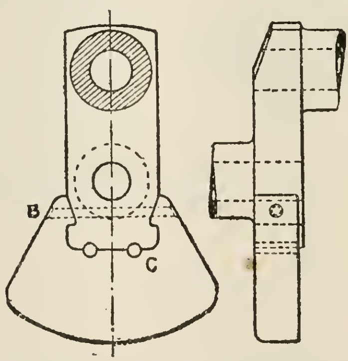
L = D × 1·2.

NOTE.—The astern guide surface = 80 per cent. of ahead guide surface.



* No. 26.—Single Guide Type Solid Crosshead.

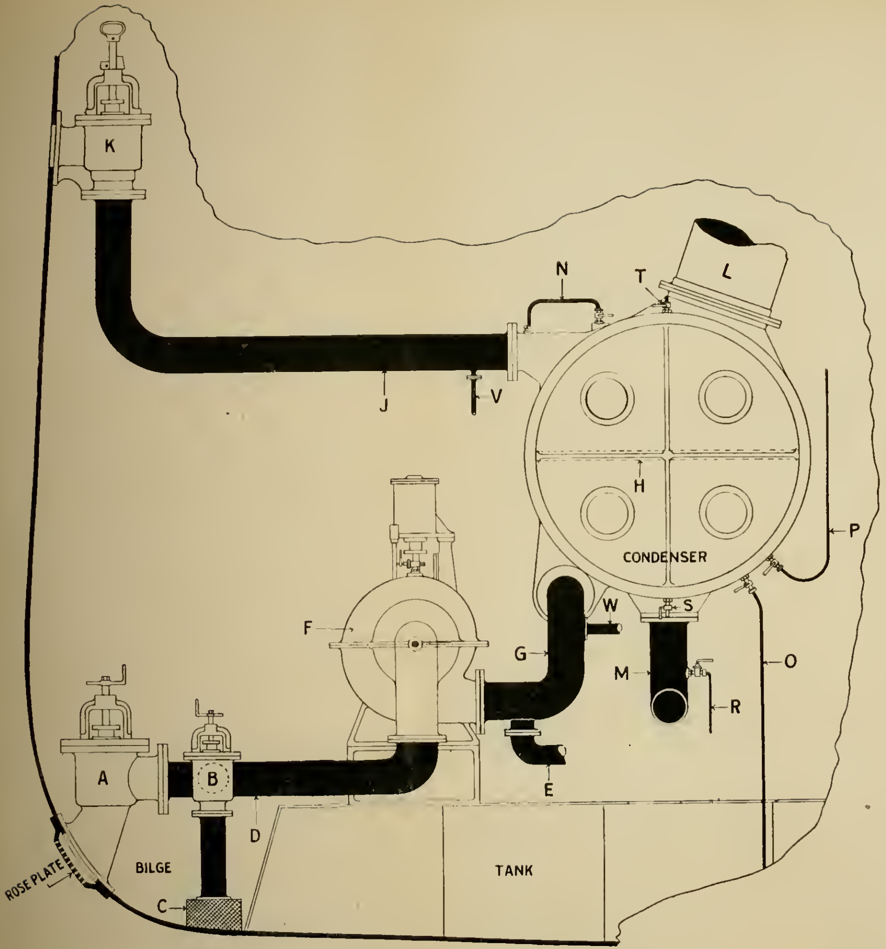
In this pattern the Crosshead Pin is shrunk into the connecting Rod Jaws.



No. 27.—Balance Weight for Crank.

B, Riveted bolt.
C, Dowel pins fitted half into web and half into weight (a driving fit).

* Reprinted by permission from "Marine Engine Design." Prof. Edward M. Bragg. D. Van Nostrand Co., New York, 1910.



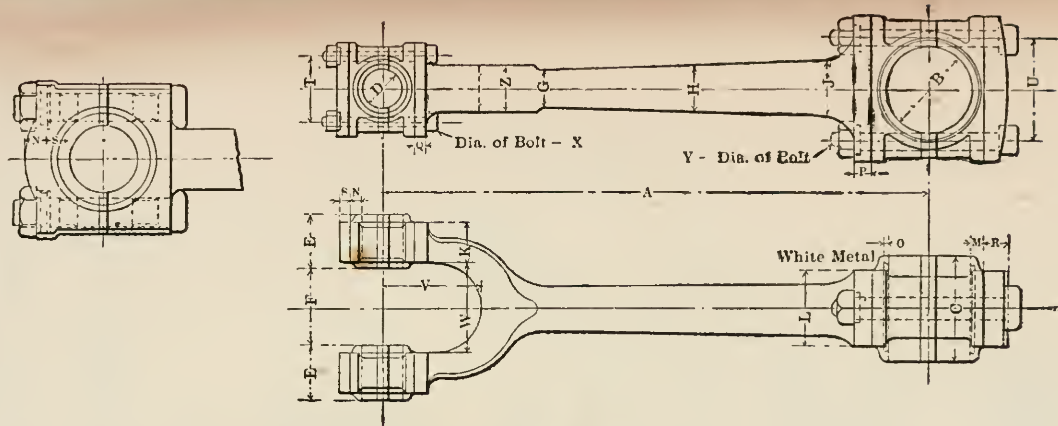
No. 25.—Condenser and Circulating Water Connections.

A, Main injection valve.
B, Bilge injection valve.
C, Bilge strum.
D, Injection pipe leading to pump suction.
E, Ballast pump circulating pipe to condenser.
F, Centrifugal circulating pump.
G, Centrifugal pump delivery to condenser.
H, Condenser division plate.
J, Circulating discharge pipe.
K, Side discharge valve.
L, Exhaust (or eduction) pipe from L.P. cylinder to condenser.
M, Air pump suction pipe from condenser bottom.
N, Auxiliary sea water feed.
O, Auxiliary fresh water feed from tanks
P, Vacuum gauge pipe.
R, Soda cock.
S, Cock to allow escape of air.
T, Cock to allow escape of air.
W, Cooling water to guides.
V, Evaporator feed.

NOTE.—In the condenser about 60 per cent. of the heat in the steam is rejected, representing an unavoidable loss: this is due to the transfer of the latent heat units of the steam to the circulating water, which transfer is necessary for condensation to take place. If the injection water enters at, say, a temperature of 65° Fahr., the discharge temperature would be somewhere about 110° Fahr., so that the discharge water is thus raised in temperature by absorbing the latent heat (approximately 1000 B.T.U. per pound of steam) of the exhaust steam.

NOTE.—The evaporator is fed with the condenser discharge water, at a temperature of, say 110°, as less heat is then required for evaporation.

[*To face page* 194.



* No. 28.—Connecting Rod.

For a shaft $12\frac{3}{4}$ inches diameter, the following are the principal average dimensions of connecting rod required:—

A = 84 inches.	F = $10\frac{1}{4}$ inches.	L = $10\frac{1}{2}$ inches.	Q = $3\frac{1}{4}$ inches.
B = $12\frac{3}{4}$,,	G = $5\frac{3}{4}$,,	M = $2\frac{3}{8}$,,	X = $3\frac{1}{4}$,,
C = 15 ,,	H = 6 ,,	N = $1\frac{1}{2}$,,	Y = 4 ,,
D = $7\frac{3}{4}$,,	J = $6\frac{1}{2}$,,	O = $\frac{3}{4}$ inch.	Z = $6\frac{3}{4}$,,
E = $8\frac{1}{2}$,,	K = $6\frac{1}{2}$,,	P = 4 inches.	

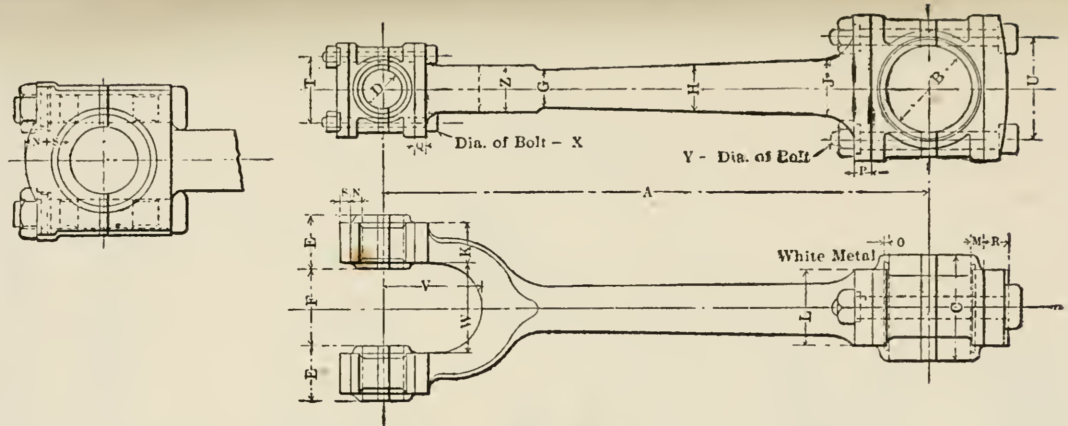
NOTE.—Distance V = $\frac{1}{2}$ depth of crosshead + depth of nut + clearance (say 1 inch).

* Reprinted by permission from "Marine Engine Design." Prof. Edward M. Bragg. D. Van Nostrand Co., New York, 1910.

B, Riveted bolt.

C, Dowel pins fitted half into web and half into weight (a driving fit).

* Reprinted by permission from "Marine Engine Design." Prof. Edward M. Bragg. D. Van Nostrand Co., New York, 1910.



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For a shaft $12\frac{3}{4}$ inches diameter, the following are the principal average dimensions of connecting rod required :—

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$B=12\frac{3}{4}$,,	$G=5\frac{3}{4}$,,	$M=2\frac{3}{8}$,,	$X=3\frac{1}{4}$,,
$C=15$,,	$H=6$,,	$N=1\frac{1}{2}$,,	$Y=4$,,
$D=7\frac{3}{4}$,,	$J=6\frac{1}{2}$,,	$O=\frac{3}{4}$ inch.	$Z=6\frac{3}{4}$,,
$E=8\frac{1}{2}$,,	$K=6\frac{1}{2}$,,	$P=4$ inches.	

NOTE.—Distance $V=\frac{1}{2}$ depth of crosshead + depth of nut + clearance (say 1 inch).

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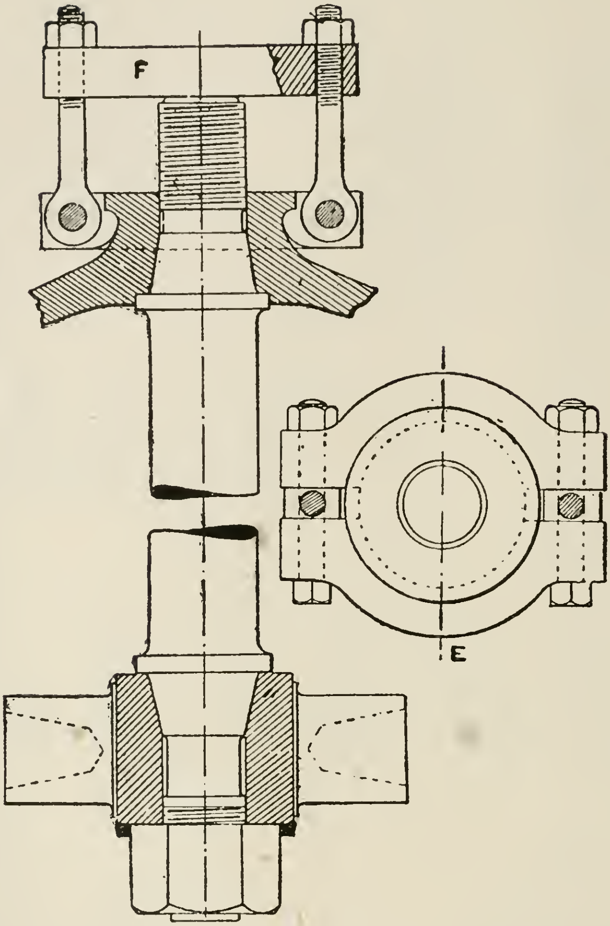
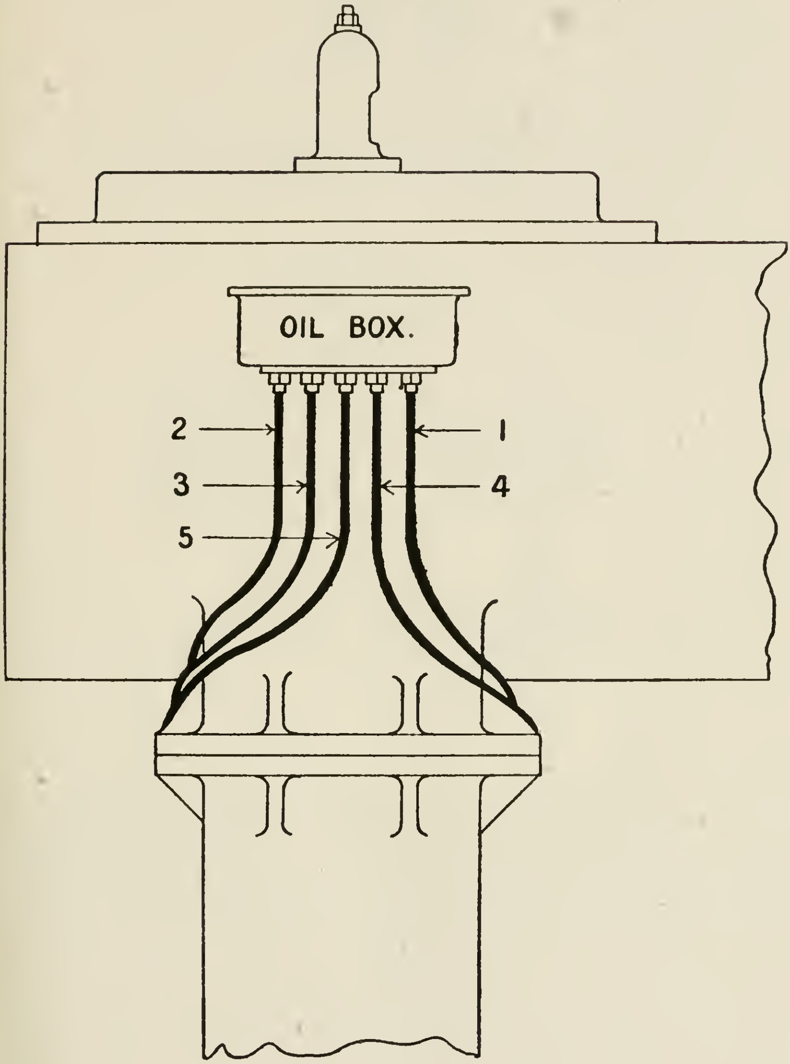


No. 29.—Piston Rod (Naval Type) showing Method of Removing Piston.

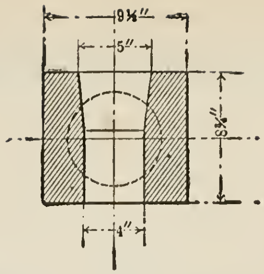
F, Dog by means of which the double clamp E draws the piston off the rod.

NOTE.—When the nuts of the hinged bolts are tightened up the clamp E acts to draw the piston off the rod.

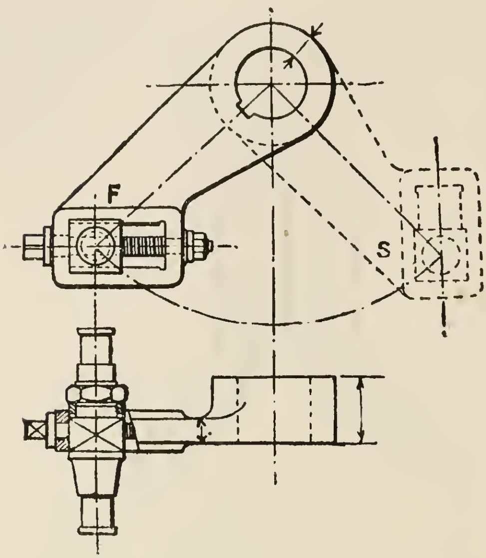


No. 30.—Syphon Feed Oil Box (on Cylinders).

1, To ahead guide.
2, To astern guide.
3, To crosshead.
4, To crank pin.
5, To crank pin.

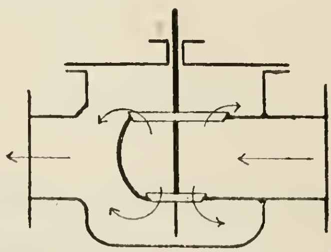


* No. 33.—Crosshead Block with Dimensions for a $5\frac{1}{4}$-inch Piston Rod.



No. 34.—Reversing Bell Crank with Expansion Slot.
F, Ahead position. S, Astern position.

Notice that linking up is only possible when in ahead gear, as when in astern gear the expansion slot is in a vertical position, so that any change of position of the block has little or no effect on the drag link.



No. 35.—Diagram Sketch of Double-Beat Valve.

Steam enters from the boilers by the right hand branch, and is admitted to the engine through both valve openings; the upper valve having the larger area allows of easy manipulation owing to the equilibrium obtained.

* Reprinted by permission from "Marine Engine Design." Prof. Edward M. Bragg. D. Van Nostrand Co., New York, 1910.

Name of Auxiliary.		Sea	Main Condenser.	Auxiliary Condenser.	Main Feed.	Auxiliary Feed	Hotwell.	Indept. Bilge.	Main Bilge	Ballast Tanks.	Overboard.	Filter.	Heater	Drain Tank.	Sanitary Tank.	Fresh Water Tank	Deck.	Distiller.	Evaporator	Refrigerator.	Circulating Dis. Overboard	Fresh Water Supply Tank	Bath Supply Tank.	See's Ash Ejector.	Engine Guides.	Tunnel and Engine Service
Circulating Pump.	suc.	●						●																		
	dis.		○								○														○	○
Ballast Pump	suc.	●						●	●	●																
	dis.		○	○							○															
Weir's Feed Pumps.	suc.	●	●				●						●													
	dis				○	○																				
General Service Donkey.	suc.	●					●		●	●																
	dis.					○					○				○		○	○						○		
Auxiliary Feed Pump.	suc.	●												●												
	dis.					○																				
Fresh Water Pump	suc.															●										
	dis.																					○				
Hot Water Pump	suc.	●																			●					
	dis.																						○			
Refrigerator Pump.	suc.	●																								
	dis.																			○						
Sanitary Pumps on M. Engines.	suc.	●																								
	dis.														○			○							○	
Evaporator Pump	suc	●																			●					
	dis.																		○							
Injector	suc.	●												●												
	dis.					○																				
Bilge Pumps on M. Engines.	suc.	● ← Ford pump only							●																	
	dis.										○						○ ← Ford. pump only.									
Feed Pumps on M Engines.	suc.						●																			
	dis.				○							○	○													
Main Condenser.	suc	●														●										
	dis																									
Air Pumps.	suc.		●																							
	dis.						○		○ ← Hotwell overflow																	

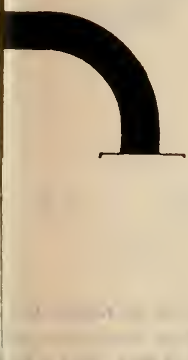
Note ● = "Suction from."
○ = "Discharge to"

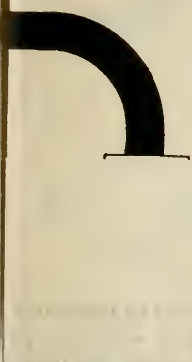
No. 31.—Pump Connection Diagram.

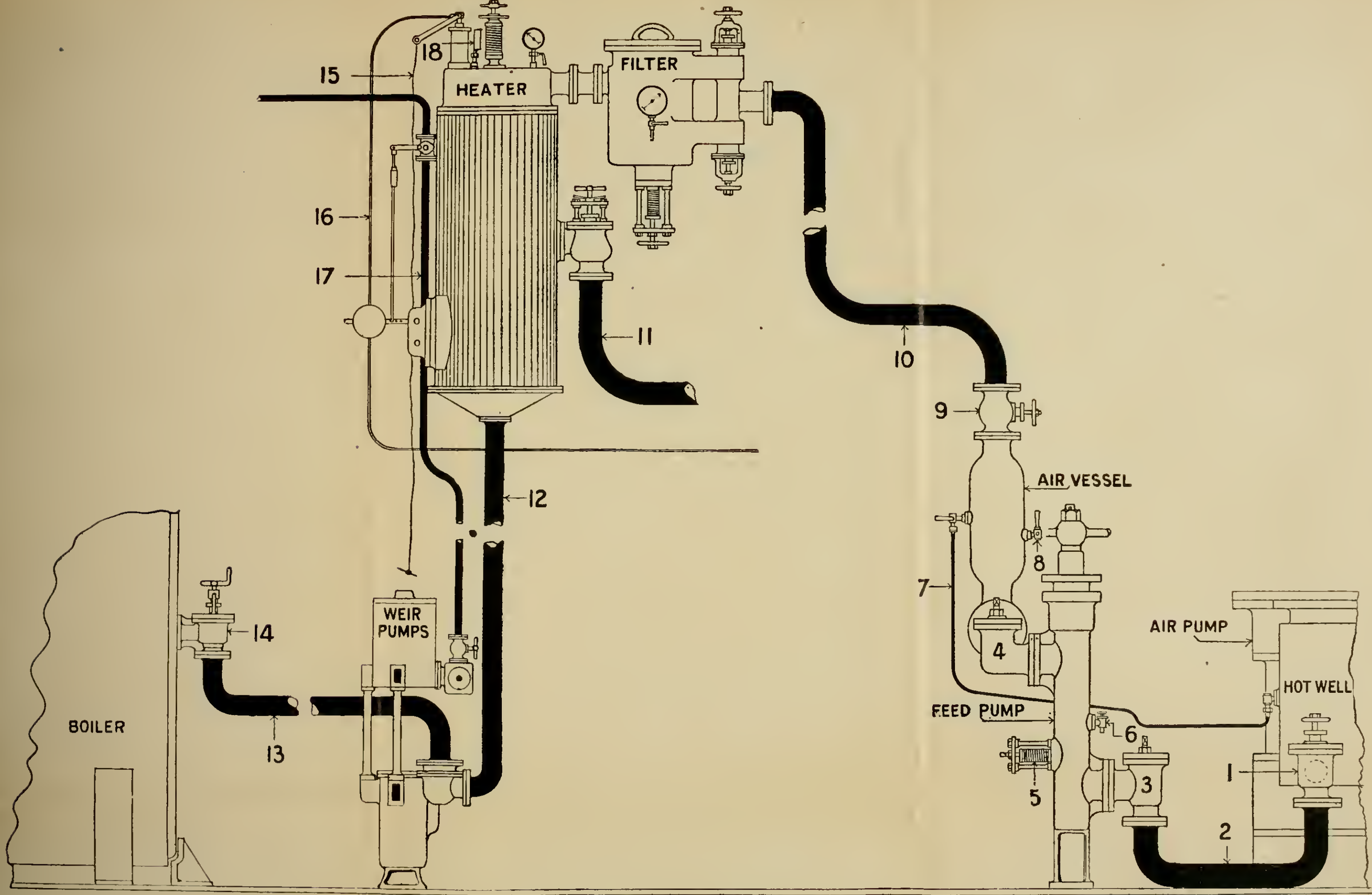
The above table shows how the various pump suction and discharge connections can be laid off in diagram form, so that any can be at once clearly located by reference. All the suction connections are shown as dark circles, and the discharge connections as open circles.

If the pumping arrangements are in any way complicated, a diagram made out similarly to the one shown above will be found of great benefit.

Referring above to the "General Service Donkey" connections it will be seen that this pump is arranged to draw from either the Sea, Hot-well, Main Bilge, or Ballast Tanks, and to discharge to Auxiliary Feed, Overboard, Sanitary Tank, Deck, or to the Distiller. The other pump connections can be traced out in the same way.







No. 32.—Feed Pump Connections.

1, Main feed pump suction from hot-well.
2, Main feed pump suction pipe.
3, Suction valve of main feed pump.
4, Delivery valve of main feed pump.
5, Relief valve of main feed pump.
6, Pet (air) valve.
7, Pressure balance connection between feed pump and hot-well.
8, Test cock (for temperature of water).
9, Regulating valve.
10, Main feed pump discharge to filter and heater.
11, Heating steam from I.P. or L.P. receiver.
12, Donkey feed pump suction from heater.
13, Donkey feed pump delivery to boilers.
14, Boiler feed check valve.
15, Cord for quick opening of heater to atmosphere when stopping engines.
16, Air pipe connection to condenser.
17, Steam to donkey feed pumps.
18, Pressure gauge.

Description.—The main feed pumps deliver the feed water at a temperature of, say, 140° into the filter; after passing through the filtering cloths it enters the feed heater at the same temperature, but is here heated to a temperature ranging from 195° to about 220°, the temperature depending on the pressure carried in the heater.

As is well known, the heating is effected by live steam placed in direct contact with the water, and this results in the condensation of the steam, the latent heat of which is thus given up to the feed water and re-enters the boilers. Had the steam gone to the condenser instead of the heater the latent heat would have been rejected in the form of heated sea water (condenser discharge). This saving of the latent heat units more than counterbalances for the loss of work by the steam having been drawn from, say, the L.P. chest, and not having expanded and done work in the L.P. cylinder in the usual way. The heater is therefore similar in action to a jet condenser as the steam is condensed direct by a spray of colder water.

[To face page 198.

SECTION IV.

SLIDE VALVES, PISTON VALVES, VALVE DATA, ETC.

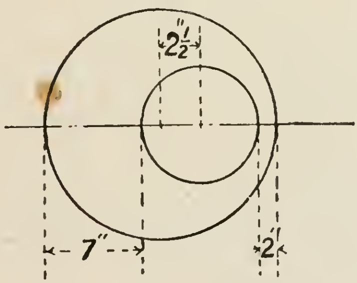
Duties of Valve.

The slide valve (or piston valve) has the following duties to perform :—

1. To admit the steam to the cylinder.
2. To cut off the supply of steam.
3. To open the port to exhaust (Release).
4. To close the port to exhaust (Compression), and so retain some of the steam for cushioning.

Valve Travel.

The travel of a valve is equal to (Steam Lap + Port Opening) $\times$ 2. Suppose lap to be 2 in. and port opening $1\frac{1}{2}$ in., then $2+1\frac{1}{2}=3\frac{1}{2}$ in., and $3\frac{1}{2}\times 2=7$ in. travel of a valve.



No. 1.

The narrow part of the eccentric subtracted from the broad part equals the travel (Sketch No. 1).

Therefore, $7-2=5$ in. Travel.

Or, take the distance from the centre of shaft to centre of eccentric and multiply by 2; this also gives the travel of valve.

Thus, $2{\cdot}5\times 2=5$ in. Travel.

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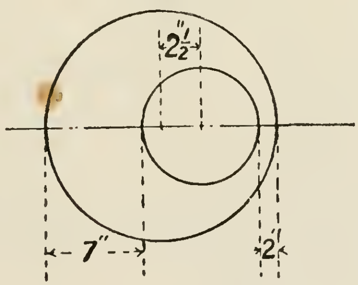


No. 1.

The narrow part of the eccentric subtracted from the broad part equals the travel (Sketch No. 1).

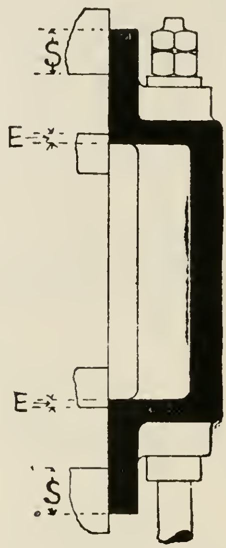
Therefore, $7-2=5$ in. Travel.

Or, take the distance from the centre of shaft to centre of eccentric and multiply by 2 ; this also gives the travel of valve.

Thus, $2{\cdot}5\times 2=5$ in. Travel.

Steam Lap is the amount the valve face covers the steam port when the valve is at half stroke, and is for cutting off the steam to cause expansion, therefore the more lap the valve has, the sooner on the stroke will the cut-off take place, and *vice versa.* The bottom steam lap is less than the top.

Exhaust Lap (usually at bottom end of valve) is the amount the exhaust edge of the valve covers the bar when the valve is at half stroke, and is for causing compression and cushioning.



No. 2.—Slide Valve.

S, Steam Lap. E, Exhaust Lap.

Minus Exhaust Lap (usually at top end of valve) is the amount the exhaust edge of the valve is *short* of the bar when the valve is at mid stroke: it causes the exhaust to open early and close late, and thus reduces the cushioning.

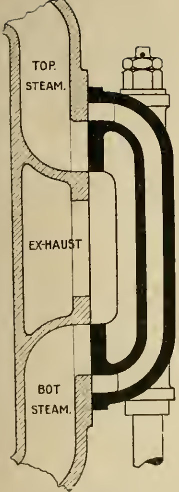
Lead is the amount the port is open for steam when the crank is on the top or bottom centre, and is for giving the engine a turning movement over the centre.

The bottom lead is always more than the top, to allow for the weight of the moving parts to be lifted up against gravity.

To cut off sooner with the main valve, steam lap must be put on and the eccentric advanced an equal amount to keep the lead the same.

No. 7. Piston Valve Ring.

Fitted with double tongue piece.

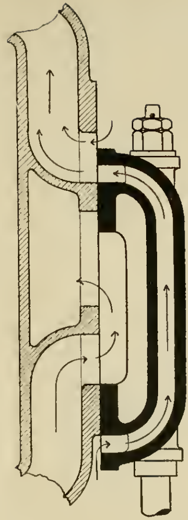


No. 3.—Trick Double Ported Valve.

At Position of Mid Travel.

For the valve shown the width of steam port = half travel of valve. The advantages of this type of valve are :—

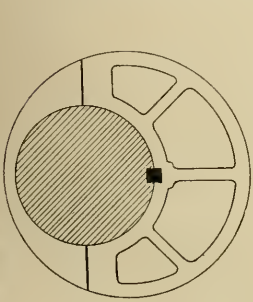
1. Reduced travel.
2. Reduced face friction.



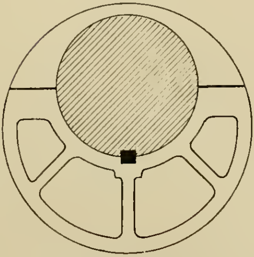
No. 4.—Trick Double Ported Valve.

At Position of Maximum Steam Opening at Top.

Notice that *half* the steam supply is admitted over the top edge of the valve, and the other half from the *bottom* by means of the internal port, as shown clearly by the arrows.

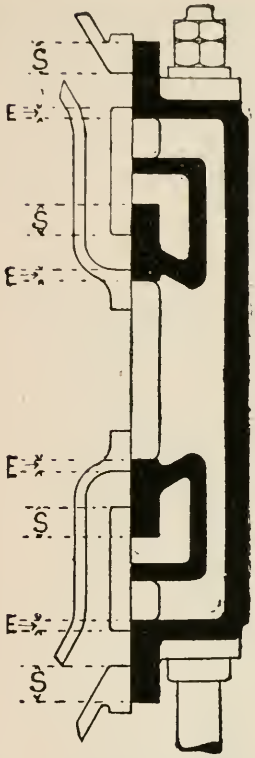


Pulley Position for above.



Pulley Position for above.

[*To face page* 200.

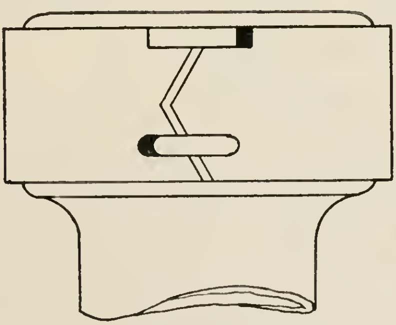


No. 5.—Common Double Ported Valve.

S, Steam Lap. E, Exhaust Lap.

No. 6.—"Trick" Double Ported Valve.

Valve Opening to Steam at Top.



No. 7.—Piston Valve Ring.

Fitted with double tongue piece.

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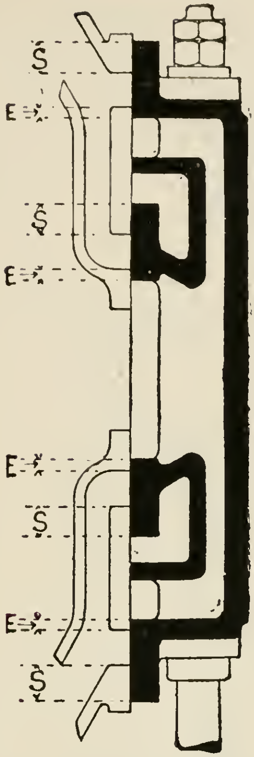

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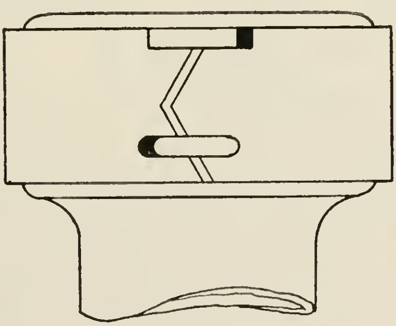
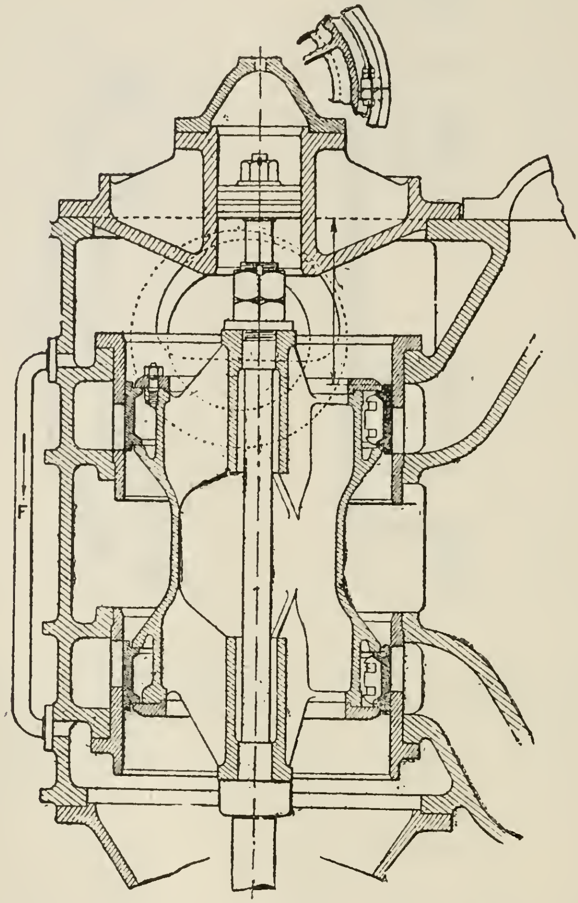
Valve Opening to Steam at Top.



No. 7.—Piston Valve Ring.

Fitted with double tongue piece.

To cut off later with the main valve, steam lap must be taken off and the eccentric put back an equal amount to keep the lead the same



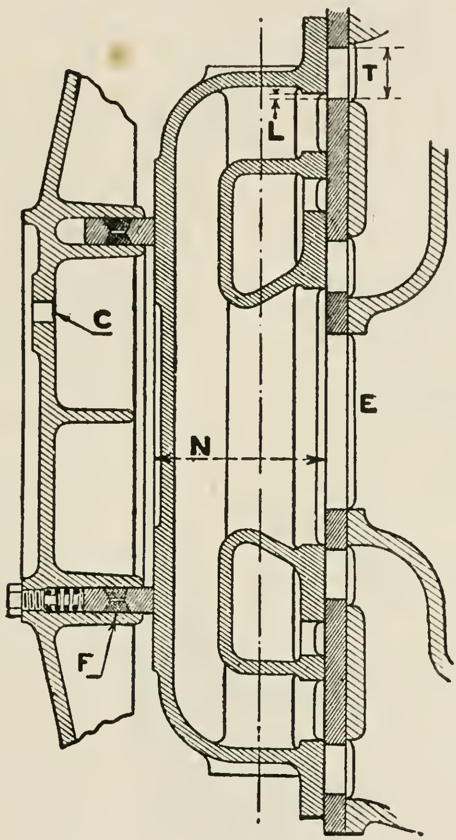
No. 8.—Piston Valve and Balance Piston (Admiralty Type).

NOTE.—The packing rings are, in this case, of the solid pattern, the cut ends being bolted together by lugs shown in the upper section view.

This valve takes steam from the ends and exhausts to the centre; the eccentric keyseat position is therefore similar to that of a slide valve: at 90° + mean steam lap and lead in advance of the crank. The balance piston has the chest steam pressure on the under side, and the condenser pressure on the upper side, a pipe connection from the top of the balance cylinder leading to the condenser.

Double Ported Valves.

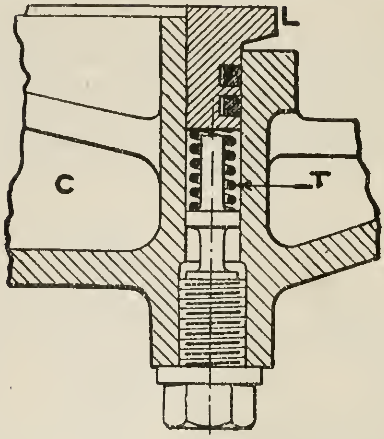
A double ported slide valve has only half the travel of a single ported valve, and the face friction is less, owing to the steam pressure in the inner ports tending to ease the valve off the cylinder face. In the common valve the inner ports receive steam from the sides, but in the Trick valve the steam passes either from bottom to top or from top to bottom.



No. 9.—Double Ported Slide Valve and Relief Ring.

T, Steam port width (about ·7 of which = steam opening).
L, Minus exhaust lap (sometimes called "internal lead").
E, Exhaust port of cylinder.
N, Depth of valve inside.
F, Packing of ring.
C, Connection to condenser (to relieve friction).

NOTE.—In the case of L.P. valves it is sometimes found of benefit to bore, say, ten or twelve ⅞-inch holes through the back metal of the valve, thus opening up additional connections to the condenser. This alteration has resulted in an improved vacuum on the back of the valve, the usual vacuum carried ranging from 18 in. to 22 in.

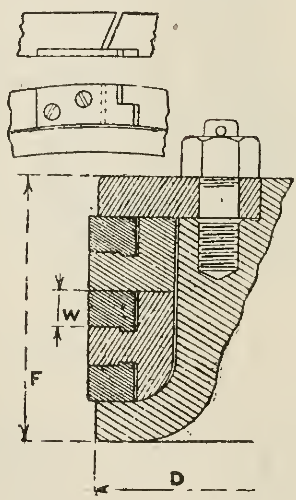


No. 10.—Slide Valve Relief Frame.

L, Faced brass ring.
T, Springs to keep ring up to back of valve.
C, Space in connection with condenser.

NOTE.—The studs (six or eight in number) can be screwed up to give additional compression to the springs.

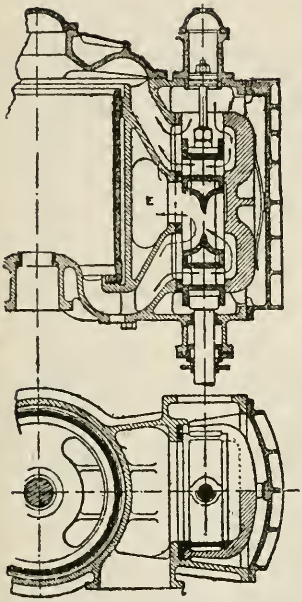
The small black sections shown in the brass ring represent soft packing.



No. 11.—"Restricted" Type Packing Rings.

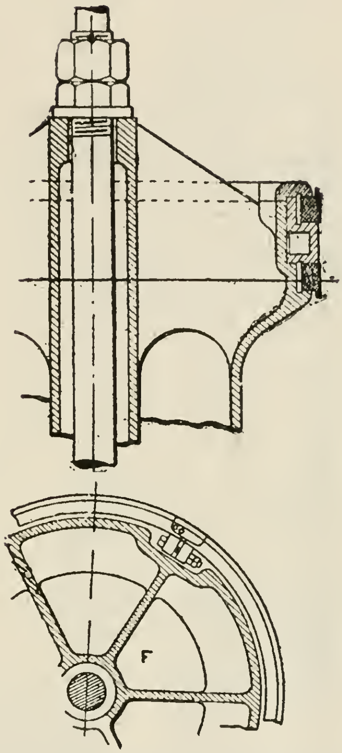
The ring expansion is limited by the small projections formed on them which fit into corresponding recesses in the carriers. The tongue piece is shown in the small views on the top.

Depth of Rings W = Lineal piston clearance × 2.



No. 12.—Andrews and Martin Balanced Slide Valve.

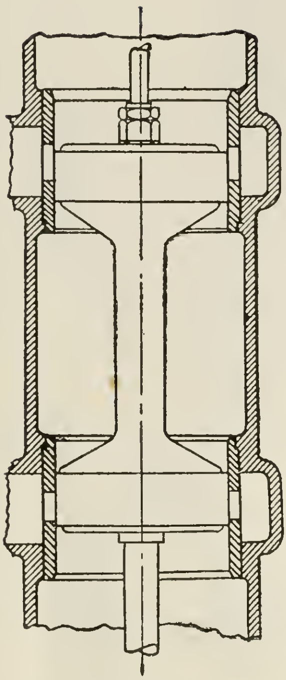
This valve is of the balanced or equilibrium type, steam being admitted from back and front through the ports shown ; the face friction is thus considerably reduced.



No. 13.—Piston Valve fitted with Two Solid Packing Rings.

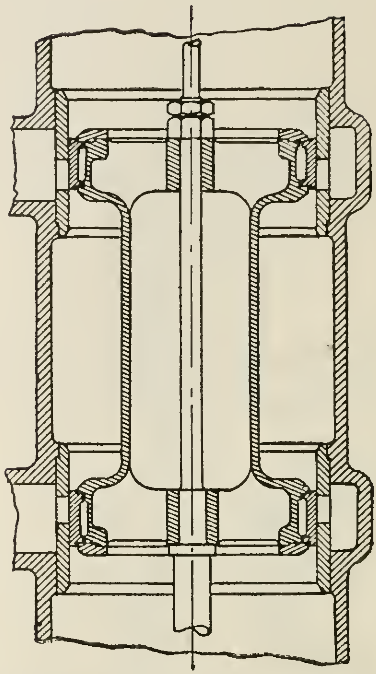
The rings are turned larger than the bore of valve chest liners, cut, bolted together as shown in the plan, and finished to fit liner diameter. A tongue piece of usual construction is also fitted.

The Andrews-Martin valve is shown in the position of steam "admission" at top, and exhaust at bottom. This valve is known among engineers as the "Matchbox" valve. The wear of the valve faces can be taken up by means of liners fitted in behind the spring on the back casing, which is adjustable.



No. 14.—Solid Type Piston Valve (Inside Steam).

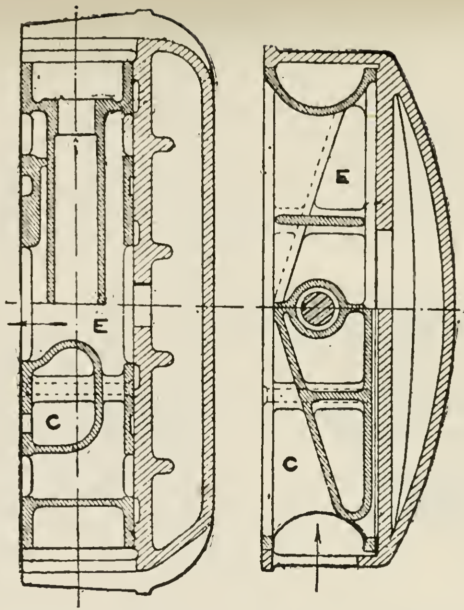
In this type of piston valve (steam inside) the eccentric keyseat position $= 90^\circ -$ mean steam lap and lead, following the crank.



No. 15.—Hollow Type Piston Valve with Rings (Outside Steam).

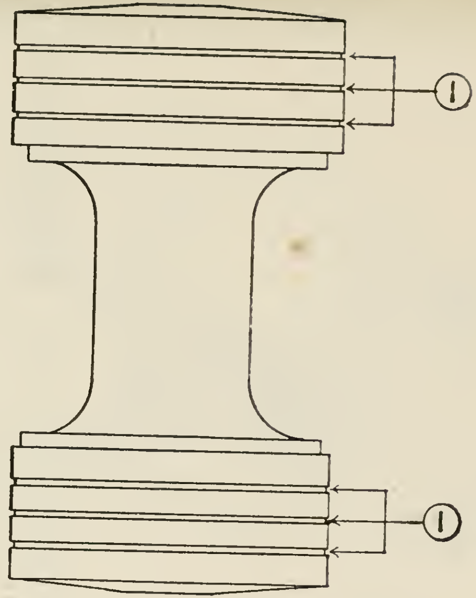
In this type of piston valve (steam outside) the keyseats are cut at an angle of $90^\circ +$ mean steam lap and lead, in advance of the crank.

NOTE.—Piston valves are usually arranged with steam inside and exhaust over ends for H.P. cylinders, and with steam over ends and exhaust in centre for I.P. or L.P. cylinders.



No. 16.—"Andrews-Martin" Patent Balanced Valve.

The above shows a double ported slide valve balanced on the Andrews-Martin system; the steam enters from the recess ports arranged on the *back* of the valve as well as from the front, and thus reduces the face friction pressure; this valve might almost be described as a modification of the piston valve.

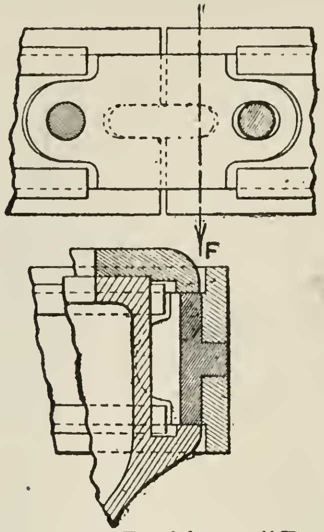


No. 17.—H.P. Piston Valve.

1, Grooves in rings for water packing.

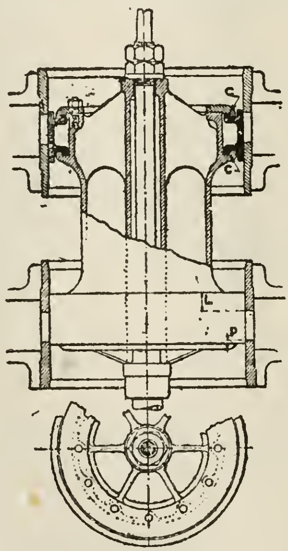
The grooves are arranged to allow of water packing (similar to Edward's Patent Air Pump) and are turned out $\frac{3}{16}$ by $\frac{1}{16}$ inch deep.

C = Steam. E = Exhaust.



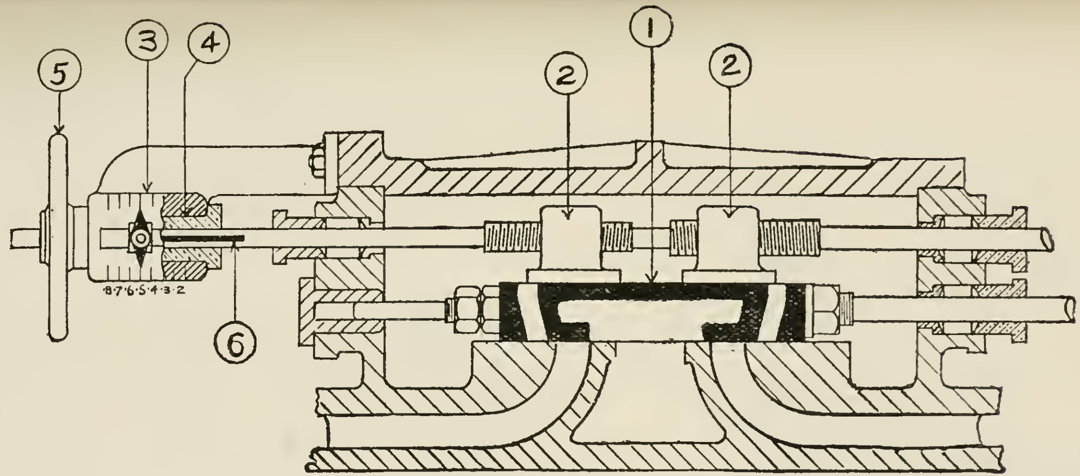
No. 18.—Piston Valve Packing: "Restrained" Type (Admiralty Type).

The split ring is allowed only a limited expansion and contraction as shown by the pin in oval hole, and by the small clearance at F. A tongue piece of ordinary pattern is fitted as shown for steam tightness.



No. 19.—Inside Steam Piston Valve with "Restrained" Type Packing Rings.

NOTE.—The clearance allowed at C C regulates the amount the ring may expand or contract; if the clearance is increased by scraping up, the ring will then expand more in proportion.



No. 19a.—Expansion Valve.

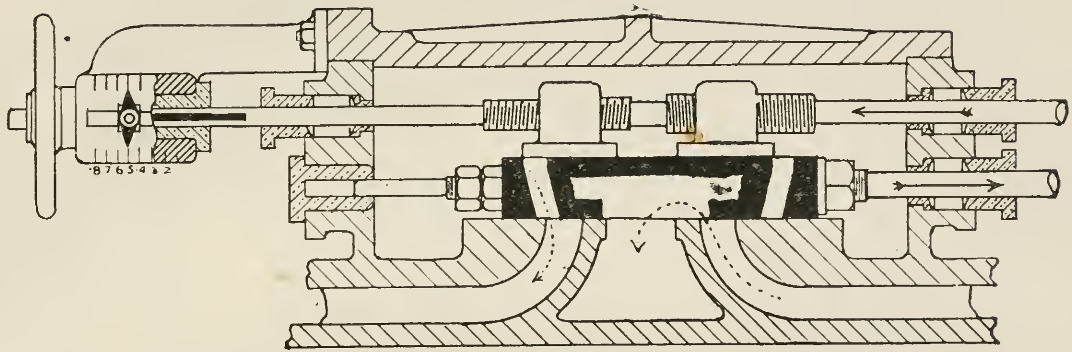
Valves in Mid-Travel Position.

(1) Main slide valve.
(2) Cut-off plates of expansion valve.
(3) Index plate for cut-off regulation.
(4) Bush which can be revolved by wheel (5).
(5) Wheel for cut-off regulation.
(6) Feather which causes valve spindle to turn when bush (4) is revolved, and which alters relative position of cut-off plates by means of R and L band screws.

Plates further apart cut off earlier on stroke.

Plates closer together cut off later on stroke.

This type of valve is fitted to the H.P. steam cylinders and air expander cylinders of cold air machines.

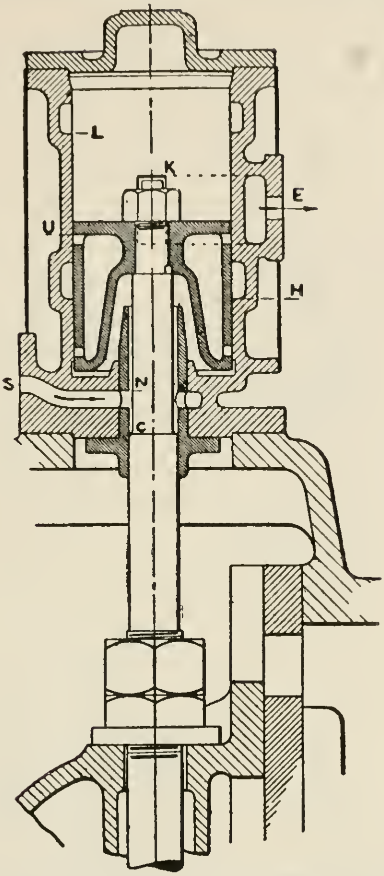


No. 19b.—Expansion Valve.

Valves in Cut-off Position.

At certain periods of the stroke both valves travel in the same direction, and at other positions the valves travel in opposite directions.

Two pulleys are fitted to the slide valve, and one eccentric drives the expansion valve, the keyseat position for the latter being, as a rule, directly opposite the crank, that is, in line with it. When fitted to main engines, in reversing it is advisable to run the expansion plates out of gear to allow of full steam for easy handling of the engines.

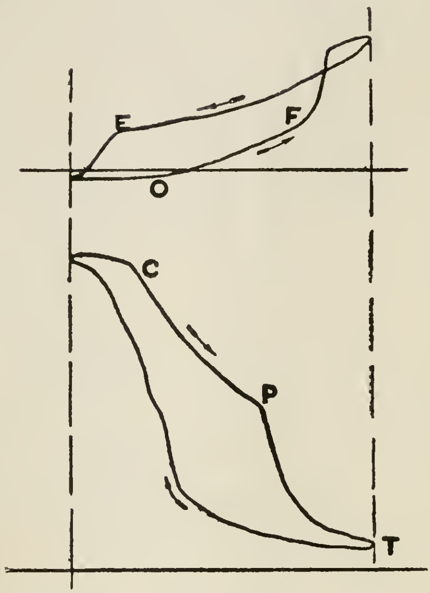


No. 20.—Joy's Patent Assistant Cylinder.

S, Steam admission port. N, Reduced diameter of rod.
C, Full diameter of rod which acts to cut off the steam supply.
U, Outlet port of piston to top of cylinder.
Note.—The inlet ports are shown at the bottom of the piston.
H, Recess for steam flow from under side of piston to inside of piston.
L, Recess for steam flow from inside of piston to upper side of piston.
E, Exhaust from cylinder.

Action.—Steam enters by port S, is cut off by rod shoulder C on up stroke, and expands in lifting up the piston; when expansion is completed. the steam exhausts into the recess H, and from there flows into the cavity of the piston, where it is retained until the small piston port U comes in line with the port L, the steam then enters the top of the cylinder and cushions the piston on top centre, afterwards assisting the piston on down stroke, and finally exhausting away (at greatly reduced pressure) by means of port E.

Notice that the same supply of steam is used both for bottom and top, also that the piston cavity acts as a receiver. The piston is double acting, but the driving pressure for the top is obtained from the exhaust steam of the bottom.

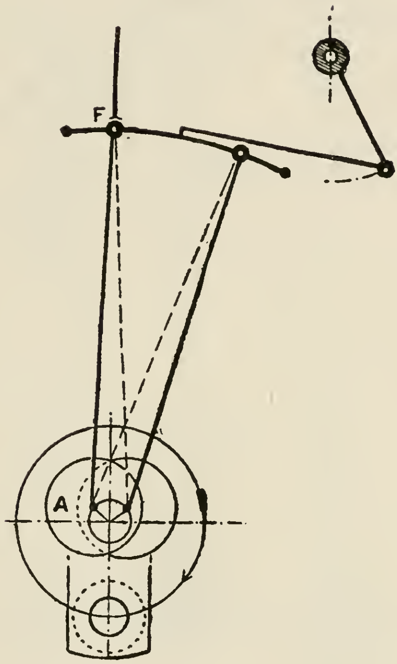


No. 21.—Diagrams from Joy Assistant Cylinder.

The lower diagram is from the bottom and the upper one from the top of the cylinder.

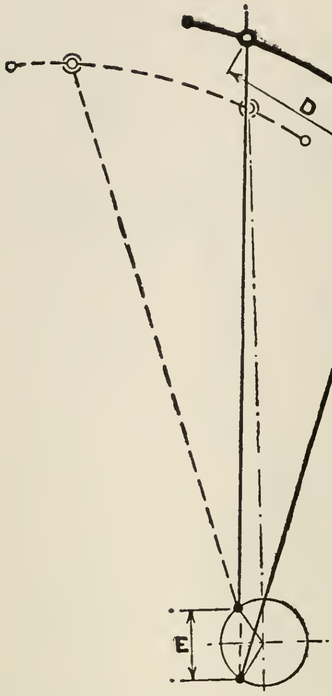
T, Admission at bottom.
C, Cut-off at bottom.
P, Exhaust at bottom to top.

O, Admission at top.
F, Compression at top.
E, Exhaust at top.



No. 22.—"Open" and "Crossed" Eccentric Rods (in "Ahead" Gear).

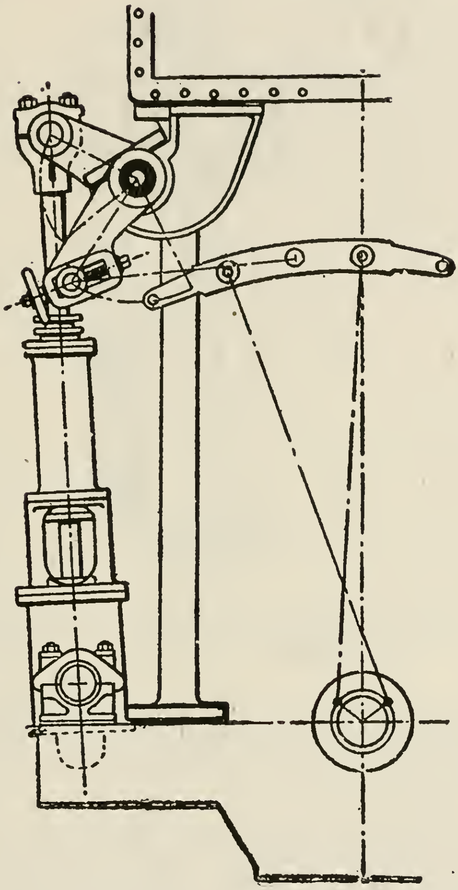
With crank on *bottom* centre, the full lines show the eccentric rods as "open" and the dotted lines as "crossed," the ahead pulley being A. "Open rods" is the usual arrangement as it allows of better link expansion when the gear is shut in (see page 257) and gives full lead in any position of link.



No. 23.—Eccentrics in "Ahead" and "Astern" Positions.

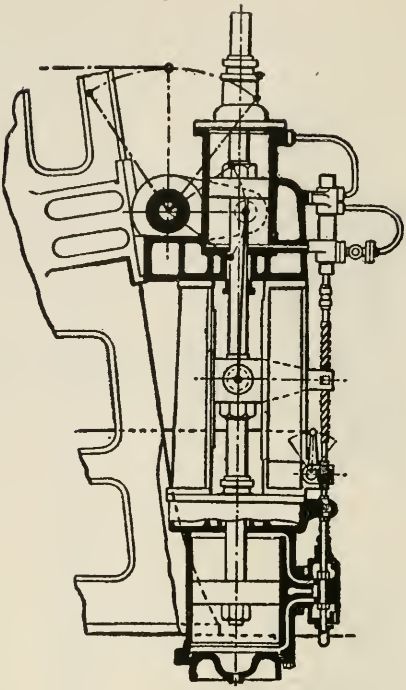
D = Valve travel × 3.
E = Distance between ahead and astern pulley.
Full lines show gear "ahead." Dotted lines show gear astern.

The difference in the link position in the two cases shows how the valve operates in reversing the engine, as the top port may be open for steam in one position, and the bottom port open for steam in the other.



No. 24.—Reversing Gear and Link Expansion Slot.

In this arrangement the reversing engine is fulcrumed on the bed-plate, the piston rod of the gear then acting direct on a forked arm keyed to the reversing (wyper) shaft. As usually arranged, the hydraulic cylinder is above and the steam cylinder below.

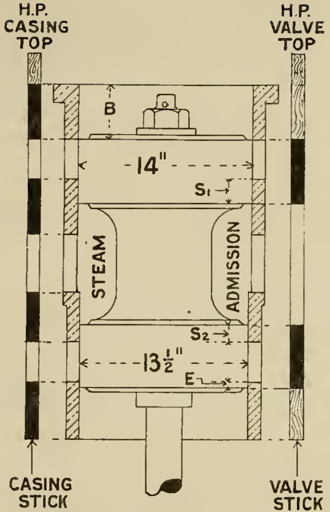


No. 25.—Combined Steam and Hydraulic Reversing Gear (Brown's Patent).

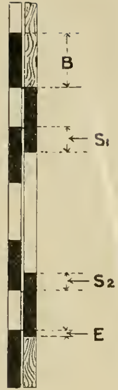
This type of reversing gear consists of a steam cylinder below and a controlling oil cylinder above, the piston rod being common to both. The piston rod connects by a crosshead and pair of links to the reversing shaft bell crank.

Action.—The steam cylinder valve has no lap, and a bye-pass valve on the oil cylinder is worked by a continuation of the steam cylinder valve spindle. The two valves mentioned are actuated by a lever (shown in sketch), also by a secondary gear connected with the reverse motion, so that a "hunting" arrangement is obtained which causes the gear when moving to bring the valves back to mid position (shut). For hand reversing a stop-cock is fitted in the bye-pass pipe of the oil cylinder and a small pump connected up to it. The oil cylinder piston is packed by means of two cup leathers (Sketch No. 26). For an engine with cylinders, $35\frac{1}{2}$ inches, 53 inches, and 63 inches (two), stroke 48 inches, the reversing engine dimensions are:—

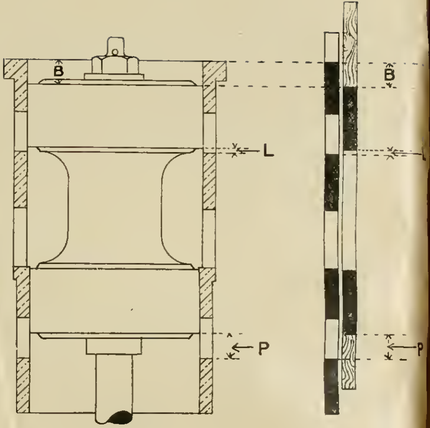
Steam cylinder diameter - - - -	16 inches
Oil cylinder diameter - - - -	$8\frac{1}{2}$ "
Stroke - - - - - - -	$20\frac{1}{2}$ "



Marking-off of Sticks.



Sticks in position for "Steam Lap" and "Exhaust Lap."



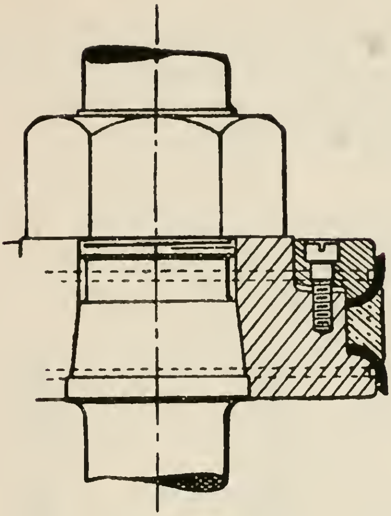
Sticks in position for "Top Lead."

No. 27.

S_1, Top steam lap.
S_2, Bottom steam lap.
E, Bottom exhaust lap.

B, Distance top piston is from top of casing, and sticks have to be placed to same distance.
P, Bottom exhaust opening at top lead position.

[To face page 215.

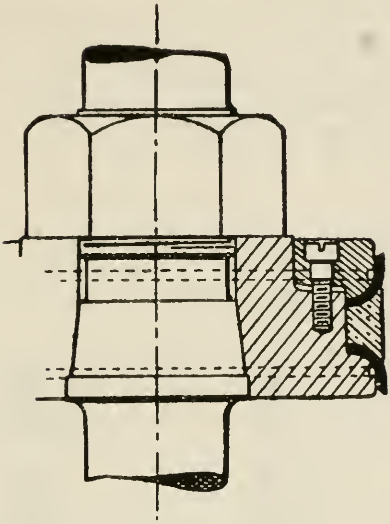


No. 26.—Leather Packing for Hydraulic Piston of Reversing Gear.

Notice the grooves cut in the piston to allow of the fluid pressure finding its way to the back of the cup leathers and thus forcing them out against the cylinder walls.

To Measure Lead of Piston Valve (Sketch No. 27).

To measure the lead or the lap of a piston valve with steam inside and exhaust at the ends, two long sticks must be cut. Having drawn the piston valve out from the chest, place one of the sticks alongside of it and mark the depth of the pistons, &c., on the stick in the exact positions they are on the valve, as shown in the sketch marked "valve stick." On the other stick, which must be equal in length to the depth of the valve casing, mark the various spaces corresponding to the bars and ports in the cylinder, shown on "casing stick." If the valve is then placed in mid travel, and the sticks put together in the same relative position, the amount of steam lap and exhaust lap will be shown, and can be measured. Tc measure the lead, top or bottom, have the valve chest cover off, and, turning the crank to the top or bottom centre as the case may be, and with the valve gear in the required position, measure how far the *top* piston is from the top of the casing; then, placing the two sticks together in a similar position, the amount of the lead will be shown, and can be measured on the sticks. The sketch shows a valve with steam lap top and bottom (S_1 and S_2), with exhaust lap on the bottom E, but having no exhaust lap on the top. Notice that the top piston is larger than the bottom one; this allows better for



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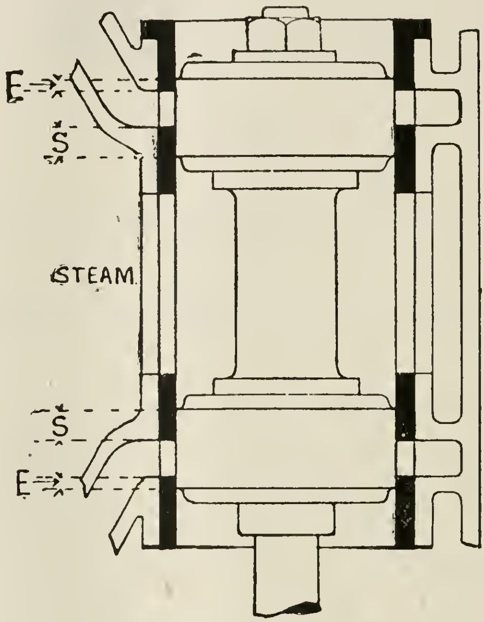
NOTE. — Sticks for ordinary slide valves (single or double ported) are made and used in the same way as above.

Lead.

Slide Valves and Piston Valves.—Advancing the eccentric increases the lead top and bottom equally.

Putting back the eccentric decreases the lead top and bottom equally.



No. 28.—Piston Valve.

S, Steam Lap. E, Exhaust Lap.

Slide Valves or Outside Steam Piston Valves.—Taking out a liner increases the top lead and decreases the bottom lead.

Putting in a liner increases the bottom lead and decreases the top lead.

To give lead to the top only, advance the eccentric for half the amount and take out a liner for half the amount.

To give lead to the bottom only, advance the eccentric for half the amount and put in a liner for half the amount.

To reduce the top lead, put back the eccentric for half the amount and line up for half the amount.

To reduce the bottom lead, put back the eccentric for half the amount and take out a liner for half the amount.

NOTE.—With double ported valves advancing the pulley for, say, $\frac{1}{8}$ in. gives $\frac{1}{4}$ in. lead in all, as the lead is duplicated by the double ports top and bottom.

Piston Valves (Inside Steam).—Taking out a liner decreases the top lead and increases the bottom lead.

Putting in a liner decreases the bottom lead and increases the top lead.

To give lead to the top only, advance the eccentric for half the amount and put in a liner for half the amount.

To give lead to the bottom only, advance the eccentric for half the amount and take out a liner for half the amount.

To reduce the top lead, put back the eccentric for half the amount and take out a liner for half the amount.

To reduce the bottom lead, put back the eccentric for half the amount and put in a liner for half the amount.

NOTE.—The upper piston is usually slightly larger in diameter than the lower one, say 14 in. diameter at top and 13 in. diameter at bottom. This is more convenient for entering or drawing the valve, it also allows of balance, the top piston "floating" and relieving the pulleys of the weight.

A piston valve (getting steam in the inside) has the following advantages over a common slide valve:—

1. Less friction, and is better balanced.
2. Only exhaust steam pressure on the valve spindle gland packing, instead of high pressure steam.
3. Reduced travel, the ports being circular and therefore longer.

Examples of Lead Adjustments.

Slide Valves.

Present Lead.			Required Lead.	
No.	Top.	Bottom.	Top.	Bottom.
1	$\frac{1}{8}$ in.	$\frac{1}{4}$ in.	$\frac{3}{16}$	$\frac{5}{16}$ in.
2	$\frac{1}{8}$ in.	$\frac{1}{4}$ in.	$\frac{1}{16}$	$\frac{3}{16}$ in.
3	$\frac{1}{8}$ in.	$\frac{1}{4}$ in.	$\frac{3}{16}$	$\frac{3}{16}$ in.
4	$\frac{1}{8}$ in.	$\frac{1}{4}$ in.	$\frac{1}{16}$	$\frac{5}{16}$ in.
5	$\frac{1}{8}$ in.	$\frac{1}{4}$ in.	$\frac{3}{16}$	$\frac{3}{8}$ in.
6	$\frac{1}{8}$ in.	$\frac{1}{4}$ in.	$\frac{1}{16}$	$\frac{1}{8}$ in.

Answers.

1. Advance pulley $\frac{1}{16}$ in.
2. Put back pulley $\frac{1}{16}$ in.
3. Take out $\frac{1}{16}$ in. liner.
4. Put in $\frac{1}{16}$ in. liner.

* { 5. Advance pulley $\frac{3}{32}$ in. and put in $\frac{1}{32}$ in. liner.
{ 6. Put back pulley for $\frac{3}{32}$ in. and take out $\frac{1}{32}$ in. liner.

A. The sum of the steam lap + lead is the same for top and bottom.

B. What is gained in lead at the bottom is lost in lap, or vice-versa.

C. Advancing the pulley increases the sum of the lap + lead at both ends.

D. Lining up increases the top steam lap and decreases the bottom steam lap.

E. Lining out decreases the top steam lap and increases the bottom steam lap.

F. Advancing or putting back the pulley does not alter the valve travel.

G. Increasing or decreasing the steam lap does not alter the valve travel.

Inside Steam Piston Valves.

Present Lead.			Required Lead.	
No.	Top.	Bottom.	Top.	Bottom.
1	$\frac{1}{8}$ in.	$\frac{1}{4}$ in.	$\frac{3}{16}$ in.	$\frac{5}{16}$ in.
2	$\frac{1}{8}$ in.	$\frac{1}{4}$ in.	$\frac{1}{16}$ in.	$\frac{3}{16}$ in.
3	$\frac{1}{8}$ in.	$\frac{1}{4}$ in.	$\frac{3}{16}$ in.	$\frac{3}{16}$ in.
4	$\frac{1}{8}$ in.	$\frac{1}{4}$ in.	$\frac{1}{16}$ in.	$\frac{5}{16}$ in.
5	$\frac{1}{8}$ in.	$\frac{1}{4}$ in.	$\frac{3}{16}$ in.	$\frac{3}{8}$ in.
6	$\frac{1}{8}$ in.	$\frac{1}{4}$ in.	$\frac{1}{16}$ in.	$\frac{1}{8}$ in.

Answers.

1. Advance pulley $\frac{1}{16}$ in.
2. Put back pulley $\frac{1}{16}$ in.
3. Put in $\frac{1}{16}$ in. liner.
4. Take out $\frac{1}{16}$ in. liner.

* { 5. Advance pulley $\frac{3}{32}$ in. and take out $\frac{1}{32}$ in. liner.
{ 6. Put back pulley for $\frac{3}{32}$ in. and put in $\frac{1}{32}$ in. liner.

Steam Lap and Lead.

RULE 1.—

Top steam lap + Lead = Bottom steam lap + Lead. Therefore, Top steam lap + Lead − Bottom lead = Bottom steam lap.

RULE 2.—

For unequal increase or decrease of lead top and bottom.

A. Alter pulley for half *sum* of lead increase or decrease.
B. Alter liners for half *difference* of lead increase or decrease.

NOTE.—If bottom lead is to be the greater, line up, but if top is to be the greater, line out.

RULE 3.—

For unequal lead increase *and* decrease, top and bottom.

A. Alter liners for half *sum* of lead increase and decrease.
B. Alter pulley for half *difference* of lead increase and decrease.

NOTE.—The nature of the question will decide whether lines have to be inserted or taken out, also whether the pulley has to be advanced or put back.

EXAMPLE 1.—

Top steam lap 2 inches and lead $\frac{1}{8}$ inch; find bottom steam lap if the lead at that end is to be $\frac{1}{4}$ inch.

Then, $2+\frac{1}{8}=$ Bottom steam lap $+\frac{1}{4}$ inch.
Therefore, $2\frac{1}{8}-\frac{1}{4}=1\frac{7}{8}$ inches steam lap at bottom. Answer.
So that, (Top) 2 inches $+\frac{1}{8}$ inch $=$ (Bot.) $1\frac{7}{8}$ inches $+\frac{1}{4}$ inch $=2\frac{1}{8}$ inches (in both cases).

EXAMPLE 2.—

Present Lead, Top $\frac{1}{8}$ inch, Bottom $\frac{1}{4}$ inch.
Required ,, ,, $\frac{1}{4}$,, ,, $\frac{1}{2}$,,
The sum of the lead increase $=\frac{1}{8}+\frac{1}{4}=\frac{3}{8}$.

Then, Pulley advance $=\frac{3}{8}\div 2=\frac{3}{16}$ inch.
And, Liner to go in $=(\frac{1}{4}$ inch $-\frac{1}{8}$ inch$)\div 2=\frac{1}{16}$ inch thick.

NOTE.—Advancing pulley $\frac{3}{16}$ inch increases lead at both ends by $\frac{3}{16}$ inch, but by lining up for the odd $\frac{1}{16}$ inch, the top is now reduced by $\frac{1}{16}$ and the bottom still further increased by $\frac{1}{16}$, giving finally $\frac{1}{4}$ inch at top and $\frac{1}{2}$ inch at bottom, as required by the question.

EXAMPLE 3.—

Present Lead, Top $\frac{1}{4}$ inch, Bottom $\frac{1}{8}$ inch.
Required ,, ,, $\frac{1}{8}$,, ,, $\frac{3}{8}$,,
Sum of Lead difference $=\frac{1}{8}+\frac{1}{4}=\frac{3}{8}$ inch.
Then, Liner to go in, $\frac{3}{8}$ inch $\div 2=\frac{3}{16}$ inch.
And, Pulley advance $=\dfrac{\frac{1}{4}\text{ inch}-\frac{1}{8}\text{ inch}}{2}=\frac{1}{16}$ inch.

NOTE.—The $\frac{3}{16}$-inch liner put in increases bottom lead to $\frac{5}{16}$ inch, and reduces top lead to $\frac{1}{16}$ inch, but the pulley advanced $\frac{1}{16}$ inch again corrects this by giving $\frac{1}{16}$ inch more at both ends, thus obtaining $\frac{1}{8}$ inch at top and $\frac{3}{8}$ inch at bottom, as required by the question.

NOTE.—If minus lead is given, treat this as so much additional or plus lead required, then proceed as explained above.

EXAMPLE 4.—

The top lead is $-\frac{1}{4}$ inch and the bottom lead is $\frac{1}{2}$ inch. The lead required is $\frac{1}{8}$ inch on top and $\frac{3}{8}$ inch on bottom.

Then, $+\frac{1}{4}+\frac{1}{8}=\frac{3}{8}$ **inch more lead required on top.**

And, $\frac{1}{2}-\frac{3}{8}=\frac{1}{8}$ **,, less ,, ,, ,, bottom.**

Therefore by Rule 3, $\dfrac{\frac{3}{8}+\frac{1}{8}}{2}=\frac{1}{4}$ **inch liner out.**

And, $\dfrac{\frac{3}{8}-\frac{1}{8}}{2}=\frac{1}{8}$ **inch pulley forward.**

Answer.— $\left\{\begin{array}{l}\frac{1}{4}\text{-inch liner to be taken out.}\\ \frac{1}{8}\text{-inch pulley put forward.}\end{array}\right.$

EXAMPLE 5.—

The original lead was, top $\frac{1}{4}$ in., bottom $\frac{1}{2}$ in., the present lead, on testing, is found to be, top $-\frac{3}{4}$ in., bottom $-1\frac{1}{8}$ in. Find how much the pulley has worked back on shaft, and the thickness of liner which has dropped out from under foot of rod.

Then, **Total lead decrease** $=(\frac{3}{4}''+\frac{1}{4}'')+(1\frac{1}{8}''+\frac{1}{2}'')=1''+1\frac{5}{8}''=2\frac{5}{8}''$.

Pulley has gone back $=\dfrac{2\frac{5}{8}''}{2}=1\frac{5}{16}''$. **Answer.**

Liner thickness (out) $=\dfrac{1\frac{5}{8}''-1''}{2}=\frac{5}{16}''$. **Answer.**

This question will be much easier understood if the student takes it backwards, that is, assume that the present lead is top $-\frac{3}{4}$ in., and bottom $-1\frac{1}{8}$ in., and that the lead required is, top $\frac{1}{4}$ in., and bottom $\frac{1}{2}$ in.

Then by rule previously enunciated—

Total lead increase $=\frac{3}{4}''+\frac{1}{4}''+1\frac{1}{8}''+\frac{1}{2}''=2\frac{5}{8}''$.

Advance pulley half sum $=\dfrac{2\frac{5}{8}''}{2}=1\frac{5}{16}''$. **Answer.**

Line up rod half difference $=\dfrac{1\frac{5}{8}''-1''}{2}=\frac{5}{16}''$. **Answer.**

These answers reversed give the solution to the question as originally stated.

Lead of Double Ported Slide Valves.—For this type of valve the examples given for the single ported valve also hold good, but it must be remembered that the pulley or liner alterations only refer to one of the two top leads or one of the two bottom leads, as advancing the pulley, say, $\frac{1}{16}$ in., will give $\frac{1}{8}$ in. extra lead in all top and bottom. In the same way, lining up for say $\frac{1}{16}$ in. gives $\frac{1}{8}$ in. more lead at bottom and $\frac{1}{8}$ in. less lead at top, and taking out a $\frac{1}{16}$ in. liner gives $\frac{1}{8}$ in. more lead at top and $\frac{1}{8}$ in. less lead at bottom. This is owing to the duplicating of the leads at both top and bottom due to the double ports.

*** NOTE.—In slide valve example No. 5 observe that the pulley requires to be advanced half the sum of the two lead increases top and bottom, which is equal to**

$\frac{1}{16}$ in. + $\frac{1}{8}$ in. or $\frac{3}{16}$ in. in all; the pulley is therefore advanced half of this, or $\frac{3}{32}$ in. and a $\frac{1}{32}$ in. liner put in to make up the difference top and bottom.

In example No. 6, as the leads have to be decreased top and bottom, the pulley is put back $\frac{3}{32}$ in. and a $\frac{1}{32}$ in. liner taken out.

* For the inside steam piston valve notice that liners are taken out instead of being put in, and and put in instead of being taken out, to give similar lead results top and bottom.

Valve Setting.—The following example of valve setting from an engine of 2500 I.H.P., will give a fair idea as to the varying proportions of lap, lead, and port opening usually arranged for.

Valve Setting.

Cylinders, 27 in., 43 in., and 72 in.; stroke, 51 in.; pressure, 180 lbs.

	H.P. Valve Travel 7 in. (Piston Valve).		M.P. Valve Travel 7 in. (Slide Valve).		L.P. Valve Travel 7 in. (D.P. Slide Valve).	
	Top.	Bottom.	Top.	Bottom.	Top.	Bottom.
Steam lap	$2\frac{1}{16}$ in.	$1\frac{15}{16}$ in.	$1\frac{7}{8}$ in.	$1\frac{3}{4}$ in.	$1\frac{13}{16}$ in.	$1\frac{11}{16}$ in.
Port opening	$1\frac{7}{16}$ in.	$1\frac{9}{16}$ in.	$1\frac{5}{8}$ in.	$1\frac{3}{4}$ in.	$1\frac{11}{16}$ in.	$1\frac{13}{16}$ in.
Lead	$\frac{1}{4}$ in.	$\frac{3}{8}$ in.	$\frac{3}{8}$ in.	$\frac{1}{2}$ in.	$\frac{1}{2}$ in.	$\frac{5}{8}$ in.
Cut-off	$33\frac{7}{8}$ in.	$29\frac{3}{8}$ in.	$35\frac{1}{4}$ in.	$31\frac{1}{2}$ in.	$35\frac{3}{4}$ in.	$31\frac{1}{2}$ in.
Per cent. of stroke	·62	(mean)	·66	(mean)	·66	(mean)
Exhaust lap	$\frac{1}{16}$ in.	$\frac{25}{32}$ in.	$\frac{3}{8}$ in.	$1\frac{1}{8}$ in.	$\frac{7}{16}$ in.	$1\frac{3}{16}$ in.
Release	$4\frac{3}{4}$ in.	$3\frac{3}{4}$ in.	$3\frac{1}{4}$ in.	$2\frac{1}{8}$ in.	$3\frac{3}{8}$ in.	$2\frac{1}{2}$ in.
Compression	8 in.	$8\frac{1}{2}$ in.	$9\frac{1}{2}$ in.	10 in.	$10\frac{1}{2}$ in.	11 in.

Referring to the above table of valve setting it is important to note the following:—

1. The top steam lap is more than the bottom steam lap.
2. The bottom port opening is more than the top port opening in proportion to the difference of lap and lead.
3. The sum of any lap and port opening top or bottom is just equal to half the valve travel.

Referring to the H.P. valve:—

Top steam lap	$= 2\frac{1}{16}$ in.	Bottom steam lap	$= 1\frac{15}{16}$ in.
Top port opening	$= 1\frac{7}{16}$ in.	Bottom port opening	$= 1\frac{9}{16}$ in.
Half travel	$= 3\frac{1}{2}$ in.	Half travel	$= 3\frac{1}{2}$ in.

The same holds good for each valve.

4. The cut-off is earlier on the up stroke in all three engines; this is due to the angularity of the crank and connecting rod when link motion valve gear is fitted.

The reverse should be the case, were it possible, as on the up stroke the weight of the working parts have to be raised against the force of gravity.

With patent valve gear such as Brock's, Morton's, Joy's, &c., the cut-off can generally be arranged to be later on the up stroke, or equal on both strokes.

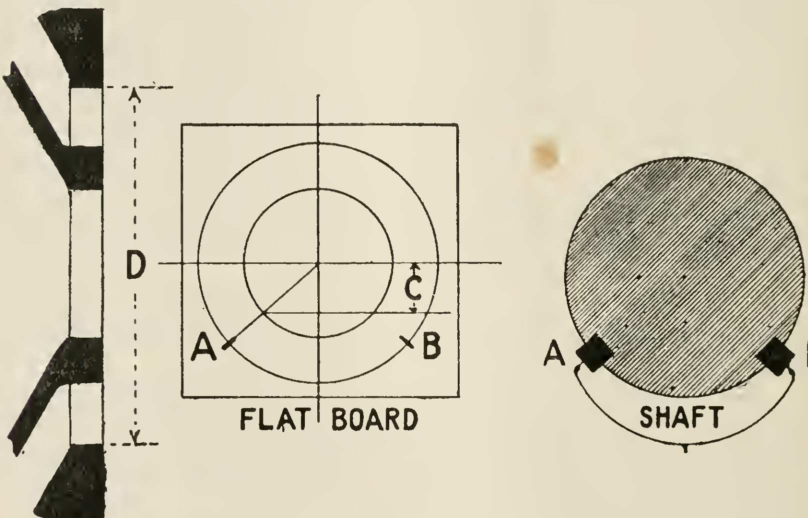
5. As less compression is required at top than bottom, it will be seen that the top exhaust lap is always less than the bottom, and is generally "minus" exhaust lap.

To Find (approximately) the Depth of a Slide Valve.

If the valve is broken up or not to be got, proceed as follows:— Obtain a flat board, and draw out on it, full size, the shaft diameter, the valve travel, and two centre lines.

NOTE.—The valve travel can easily be found by taking the difference of the narrow side and the broad side of the eccentric pulley.

Next, calliper on the shaft the exact distance between the eccentric keyseat centres, as shown at A, B of the shaft sketch, and transfer this distance to the shaft circle on the flat board, also marked



No. 29.

A, B. Now, from any one of these two points run in a line to the shaft centre, and where this line cuts the travel circle draw a horizontal line giving the distance C.

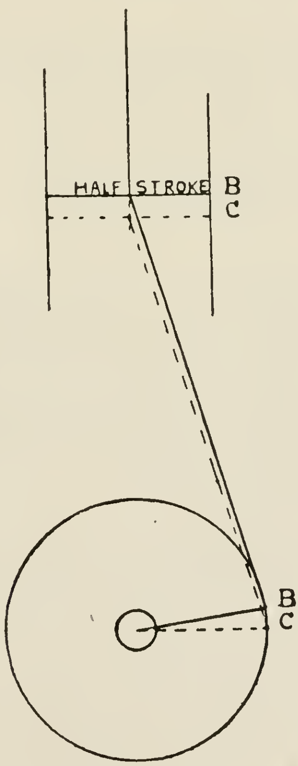
The distance C is equal to the steam lap and lead added together so that if a certain lead is determined on, say $\frac{1}{8}$ in., and subtracted from distance C, the amount left will then be the steam lap. Finally, measure the distance from the top of the top steam port to the bottom of the bottom steam port, as shown at D, and add to it *twice* the steam lap, to allow for the top and bottom; the result will then be approximately the required depth of valve.

$$\text{Or, } D + (C - \text{lead}) \times 2 = \text{valve depth.}$$

NOTE.—The most accurate method is by that of the valve diagram (see page 249), but as the cut-off is not given in the question as stated above, this method cannot be applied.

Connecting Rod Angle, &c.

When the piston is at half stroke, as at B on the sketch, the crank is lying at the angle B above the horizontal. Again, if the crank is placed exactly horizontal, as at C, the piston will be a little *lower* than half stroke, as C on the sketch. The cause, in both cases, is the angle of the connecting rod, and the shorter the rod is made the greater



No. 30

will be the difference between the piston and crank positions at half stroke.

If the slide valve has the same amount of steam lap and lead top and bottom, and the valve gear is of the ordinary link motion type, the effect of the connecting rod angle is to cut off the steam sooner on the up stroke than on the down stroke.

In practice this difference of cut-off is partly corrected by lining up the slide valve, so that the top lap is more and the bottom lap less, and the bottom lead more and top lead less.

Patent Valve Gears.

Patent valve gears are fitted with the object of correcting the defects peculiar to the ordinary Stephenson link motion, and which are as follows :—

1. Wire drawing of steam owing to slow motion of gear at moment of cut-off.

2. Variation in lead and compression when linked up (usually increased).

3. Difference in cut-off on up and down stroke, with equal steam lap top and bottom, due to effect of connecting rod angle with crank, the cut-off being much earlier on the up stroke.

4. Space saved in fore and aft direction, as the valves can (in certain cases) be placed on the sides of the cylinders.

5. Eccentrics are done away altogether in certain gears (Joy, Morton), while in others (Hackworth, Brock, Bryce-Douglas, Bremme-Marshall) one eccentric only is required.

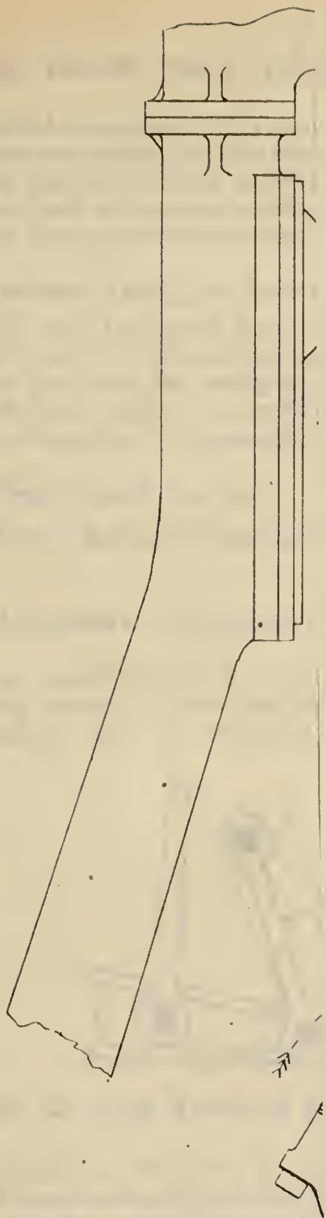
Advantages of Patent Gears.—1. Most of the patent gears arrange for a quick travel of valve at the instant of cut-off, and thus reduce the wire drawing losses by giving a much *sharper* cut-off.

2. In the majority of patent gears the lead remains constant for all positions of the link, whether "full out" or "shut in."

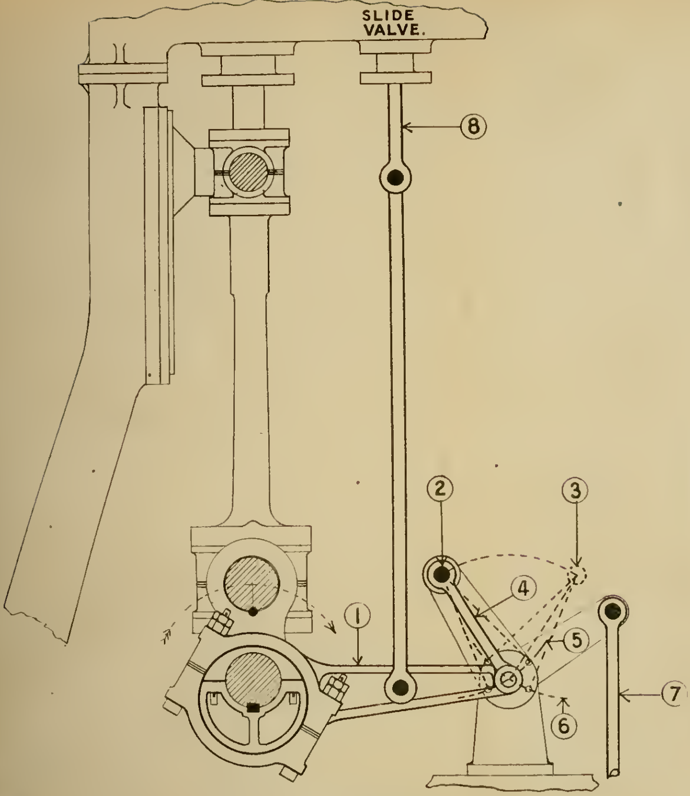
3. Certain gears are arranged with compensating rods or links which give equal cut-off on both strokes, or allow for a later cut-off on the up stroke, which is better still.

Disadvantage of Patent Gears.—The chief disadvantage of patent gears lies in the number of joints, required, the slight wear of which (in the majority of gears) upsets the valve adjustment to a more or less serious degree, as the wear of, say, $\frac{1}{64}$ in. in a brass may become magnified to three or four times that amount at the valve by means of the lever or link connected to it.

In some gears as many as sixteen small pins and brasses are fitted, all of which require to be kept in practically perfect adjustment, if the correct setting of the slide valves or piston valves is to be maintained.



1, Eccentric rod.
2, Swinging link centre on
3, Astern position of swing
4. Swinging link (Radius



No. 31.—Marshall Valve Gear.

1, Eccentric rod.
2, Swinging link centre on reversing bell crank.
3, Astern position of swinging link centre.
4, Swinging link (Radius Rod).
5, Link travel for "ahead."
6, Link travel for "astern."
7, Reversing engine.
8, Slide valve spindle.

[To face page 222.

The general experience of engineers is that the disadvantages of patent gear more than balance the advantages, with the result that the ordinary Stephenson link motion will be found fitted in even the most modern and up-to-date marine engines, as being simpler and more reliable than patent valve gears of any type.

Marshall-Bremme Gear (one Eccentric) (Sketch No. 31).

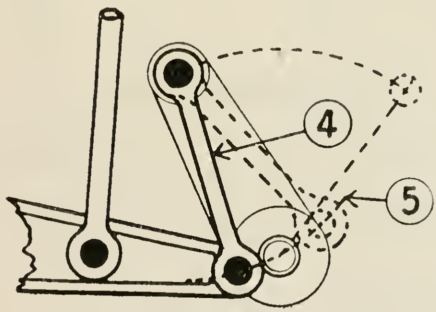
Marshall's and Bremme's gear are similar, the chief difference being that the valve link is connected to the *end* of the eccentric rod in Bremme's gear, and the swinging link to the middle, whereas in Marshall's the valve link is connected to the eccentric rod about the middle of its length, and the swinging link at the end.

Marshall Gear (Sketch No. 31).

Type of Valve fitted.—Slide valve or piston valve with steam over ends

Position of Eccentric.—Opposite crank (180°).

Action.—The eccentric rod 1 is connected to the swinging link 4, which is hung on a pin 2 from the bell crank. The gear is shown in "ahead" position, and the travel of the link is shown at 5. The



No. 32.—Link Travel of Marshall Valve Gear.

"astern" position of the gear is shown at 3 and 6, as then the bell crank is moved over to the right by the rod 7 from the reversing engine. When the swing link is at position 3, the free end travels the arc 6, and thus changes the direction of the valve travel.

The small Sketch, No. 32, shows the travel of the swinging link produced by the eccentric when in ahead position.

Bremme Gear (Sketch No. 33).

As before described, in this gear the valve link is placed at the end of the eccentric rod, which thus reverses the motion of the valve,

The general experience of engineers is that the disadvantages of patent gear more than balance the advantages, with the result that the ordinary Stephenson link motion will be found fitted in even the most modern and up-to-date marine engines, as being simpler and more reliable than patent valve gears of any type.

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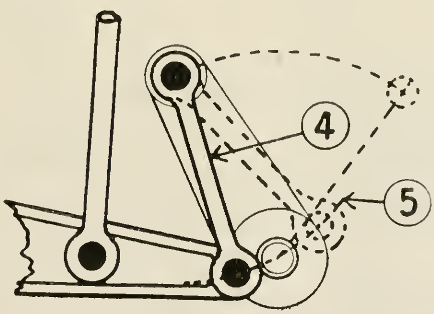
Type of Valve fitted.—Slide valve or piston valve with steam over ends

Position of Eccentric.—Opposite crank (180°).

Action.—The eccentric rod 1 is connected to the swinging link 4, which is hung on a pin 2 from the bell crank. The gear is shown in "ahead" position, and the travel of the link is shown at 5. The



No. 32.—Link Travel of Marshall Valve Gear.

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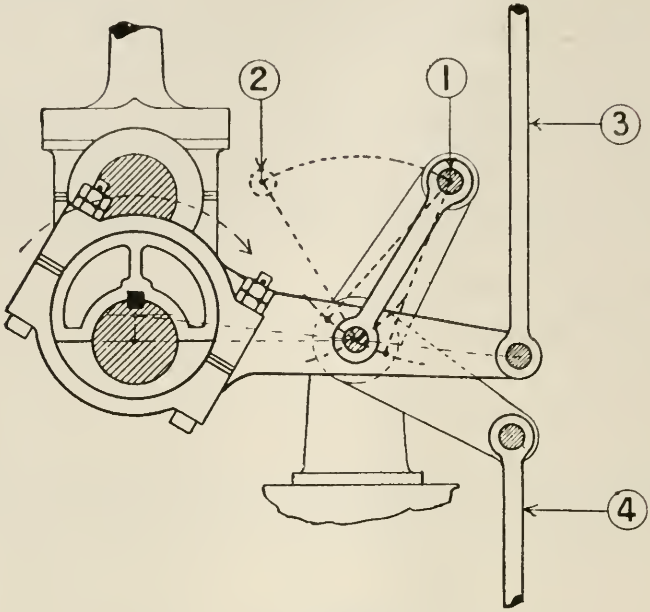
The small Sketch, No. 32, shows the travel of the swinging link produced by the eccentric when in ahead position.

Bremme Gear (Sketch No. 33).

As before described, in this gear the valve link is placed at the end of the eccentric rod, which thus reverses the motion of the valve, so that the pulley position is now *with* the crank in place of being opposite to it as in Marshall's gear, otherwise the gear is similar.



No. 33.—Bremme Valve Gear.

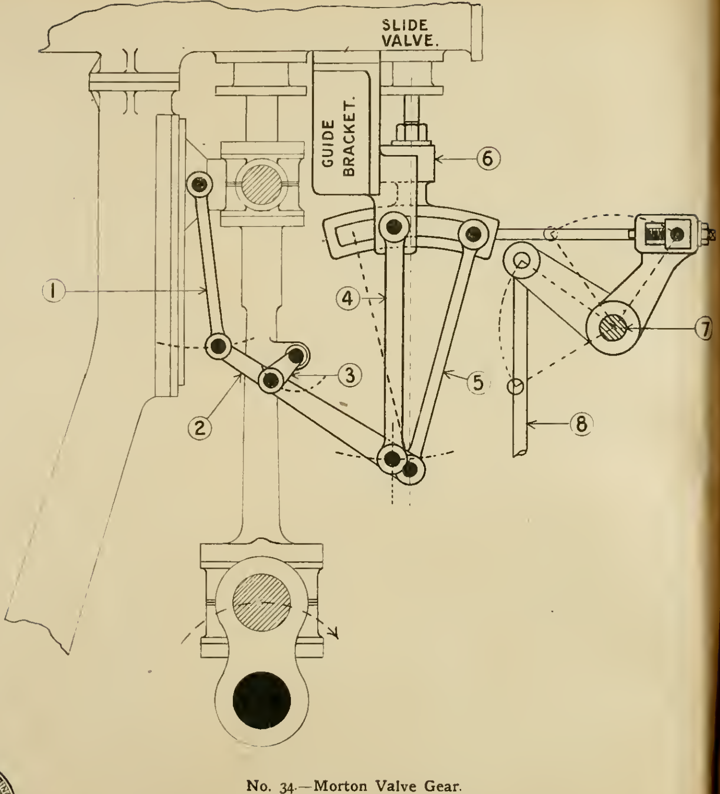
1, "Ahead" position of link.
2, "Astern" position of link.
3, Valve rod link.
4, Reversing engine rod.

Type of Valve fitted.—Slide valve or piston valve with steam over ends.

Position of Eccentric.—With the crank.

Action.—The action of the swing link 2 is as before described for Marshall's gear, but it should be noticed that the angle of the link is reversed in this case, position 1 being for "ahead" and 2 for "astern." The distance between the swinging link pin on the eccentric rod and the valve link allows for the required lead, and it should be noted that the end of the eccentric rod and valve link describe an irregular ellipse when in motion, the long sides of which incline to the vertical and produce the quick travel of valve at the cut-off positions.

It may also be stated that with equal steam lap top and bottom the cut-off and release can be arranged to take place earlier on the down stroke than on the up stroke, this result being obtained by the



No. 34.—Morton Valve Gear.

1, Link suspeoded from crosshead.
2, Lever connecting to quadrant rod 5 through small lever 3.
3, Compensating lever.
4, Suspension links from guide bracket.
5, Quadrant rod.
6, Crosshead of valve spindle, solid with quadrant.
7, Wyper shaft.
8, Reversing engine rod

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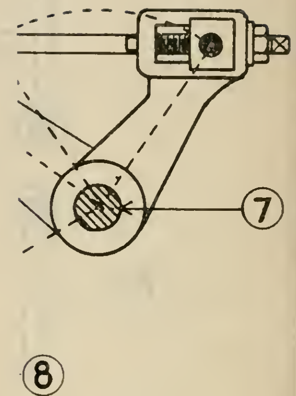

cket.

with quadrant.

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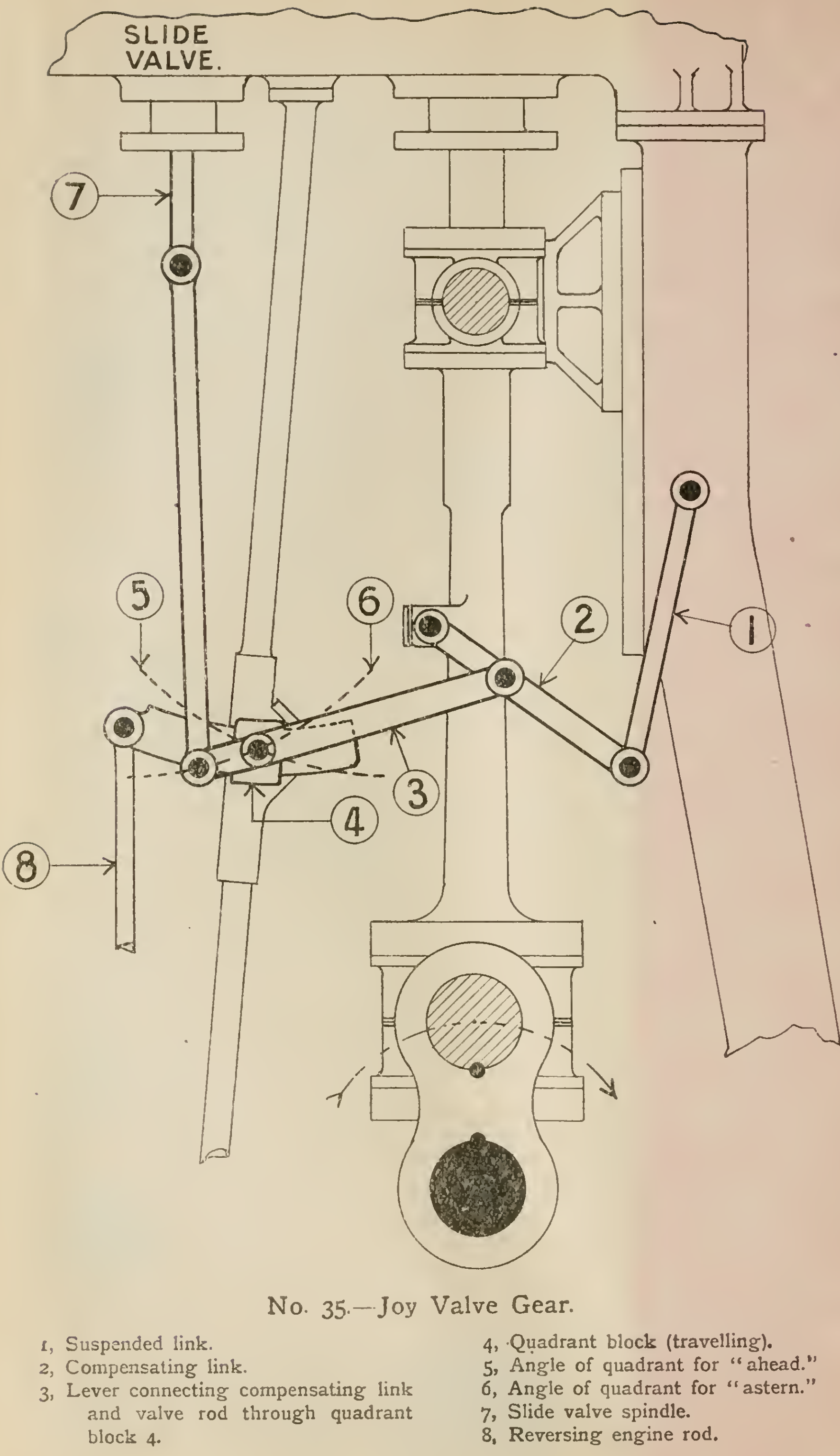


No. 35.—Joy Valve Gear.

1, Suspended link.
2, Compensating link.
3, Lever connecting compensating link and valve rod through quadrant block 4.
4, Quadrant block (travelling).
5, Angle of quadrant for "ahead."
6, Angle of quadrant for "astern."
7, Slide valve spindle.
8, Reversing engine rod.

[To face page 225.

length given to the swinging link, the oscillations of which produce the difference in cut-off mentioned. It should also be noted that the lead remains constant for all grades of expansion.

Morton Valve Gear (Sketch No. 34).

This valve gear consists of a series of levers and links, connected to a vertically moving quadrant which is *in one* with the valve spindle, no eccentric being required, as the valve motion is obtained from the connecting rod.

Type of Valve fitted.—Slide valve or piston valve with steam over ends.

Action.—The valve lever 2 is suspended from the crosshead by the link 1, from the connecting rod by the pivoted arm 3, and from the guide bracket by the heavy links 4, at a point *near* the outer end, the end of the lever being connected direct to the valve link 5, which gives vertical motion to the quadrant and thus to the valve: the small compensating arm 3 corrects the unequal effect of the connecting rod. Notice that the centres of the suspension link 4 are *not in line* with the valve spindle, and when the crank is centred the links 4 are parallel to the centre line of the engine, and in this position the valve link 5 may be moved over the quadrant from one side to the other without moving the valve. When linked up, the lead remains the same as in full gear, and equal steam lap gives equal cut-off top and bottom.

Joy Valve Gear (Sketch No. 35).

This gear is similar to Morton's in the fact that links and a quadrant take the place of eccentrics, the connecting rod supplying the necessary motion to the valve.

Type of Valve fitted.—Slide valve or piston valve with steam over ends.

Action.—The suspended link 1 on the column connects to a vibrating link 2 on the connecting rod, and the valve lever 3 is fixed to a pin on this link at an intermediate position. Near the other end of the valve lever is a fulcrum point which slides back and forward on the quadrant bar by means of a block 4, the actual end of the lever being connected direct to the valve link and spindle 7. The angle given to the quadrant by the reversing engine 8 determines the direction of rotation, 5 being for "ahead" and 6 for "astern," the block 4 sliding back and forward in the dotted arc shown.

When the quadrant is in a horizontal position (as shown in Sketch No. 35) the gear is in the neutral position. It should be noted that the quadrant bar is hinged at the centre to a supporting bracket bearing on the left column.

The leverage given by the distance from the fulcrum point on the lever to the end allows for the "steam lap + lead" travel of the valve, while the to-and-fro travel of the block 4 on the quadrant bar allows for the additional port opening travel required.

Hackworth Gear (Single Eccentric) (Sketch No. 36).

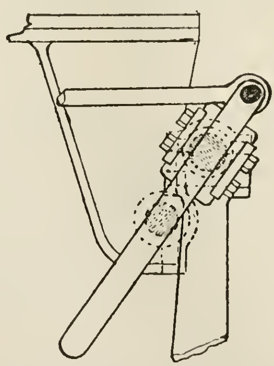
This gear works on the same principle as the Bremme-Marshall Gears, but instead of the swinging link an inclined bar is employed, on which a bearing block connected to the end of the eccentric rod slides up and down with the motion of the pulley.

Type of Valve fitted.—Slide valve or piston valve with steam over ends.

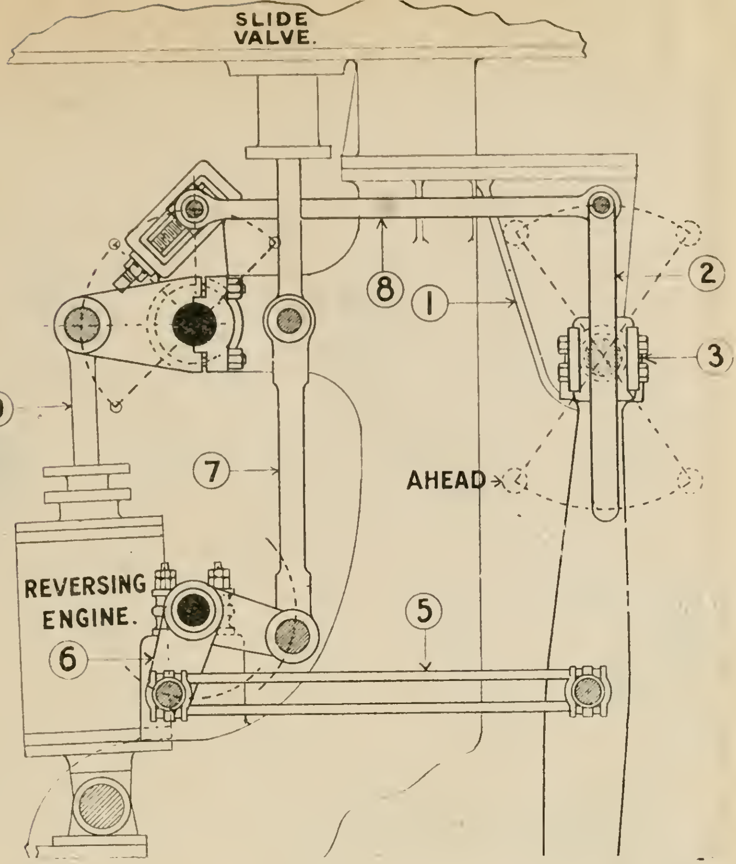
Position of Eccentric.—At 90° leading the crank.

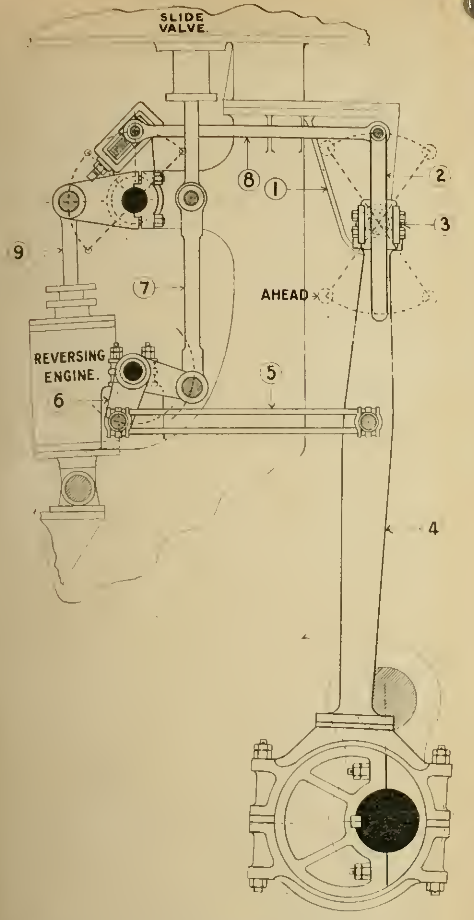
Action.—The bracket 1 supports the pair of slide bars 2 by a pin and brass as shown, and the angle given to the slide bar by the reversing engine rod 9 determines the position of the gear whether "ahead" or "astern." The motion of the eccentric rod is carried to the valve link 7 through the bell crank 6 by means of the double link 5 connected to the eccentric rod by a large pin joint; adjustable slippers 3 are fitted to the end of the eccentric, and the dotted lines show the angle of the bars for ahead or astern running.

In the Sketch the gear is shown in mid position, the valve travel being then equal to the steam lap + lead for either end, the side way slide motion of the eccentric rod in the bars allowing for the additional port opening required. The slide bars angle is changed by means of the usual drag link 8 from the reversing engine, and an expansion slot of the usual type is fitted for working "linked-up."



No. 37.—Eccentric Rod at Limit of upper travel on Slide Bar.

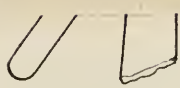

SLIDE VALVE.
8
1
2
3
9
7
AHEAD
REVERSING ENGINE.
6
5



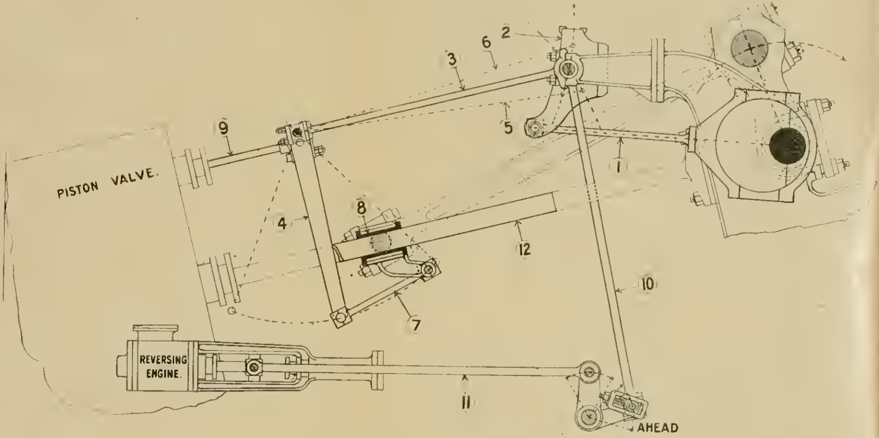
No. 36.—Hackworth Valve Gear.

1, Supporting bracket for slide bars.
2, Slide bars for slippers 3.
3. Slippers on end of eccentric rod.
4, Eccentric rod.
5, Links connecting eccentric rod and valve bell crank
6, Valve bell crank working on fixed bearing.
7, Valve link.
8, Drag link.
9, Reversing engine rod.

[*To face page* 226.



No. 37.—Eccentric Rod at Limit of upper travel on Slide Bar.



No. 38.—Brock Valve Gear.

1, Eccentric rod.
2, Rocking quadrant.
3, Valve link.
4, Lever connected at one end to valve link, and at the other end to crosshead link 7, also connected to the valve spindle near the end.
5, "Ahead" position of valve link.
6, "Astern" position of valve link.
7, Crosshead link
8, Crosshead slipper.
9, Valve spindle.
10, Drag link.
11, Reversing engine rod.
12, Guide on column.

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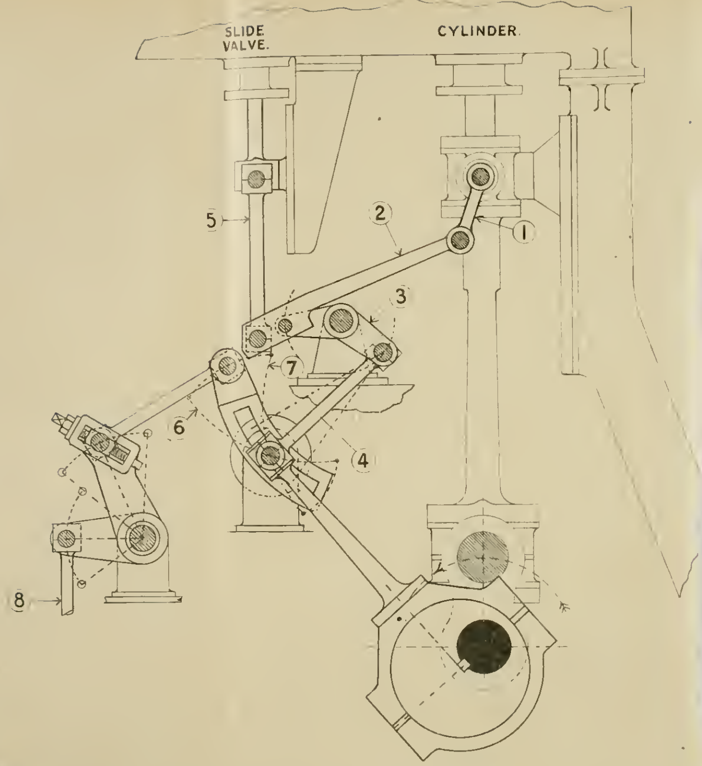
No. 37.—Eccentric Rod at Limit of upper travel o



No. 39.—Bryce-Douglas Valve Gear.

1, Liok suspended from crosshead.
2, Lever connecting crosshead with valve rod 5 through a fulcrum on bell crank 3.
3, Bell crank working oo fixed bearing.
4, Quadrant rod.
5, Valve rod.
6, "Ahead" position of quadrant.
7, "Astern" pnsition of quadrant.
8, Reversing engine rod.

[To face page 227.

The small Sketch, No. 37, shows the gear in "ahead" position, with the slipper at the upper limit of its travel on the inclined bars. In this gear the bearing of the bell crank (6) shaft is subject to the most wear, and requires the most frequent overhaul, as the setting of the valve is affected by wear down of this bearing.

Brock Gear (Single Eccentric) (Sketch No. 38).

In this gear a rocking quadrant actuated by the eccentric is employed to convey the motion to the valve.

Type of Valve fitted.—Piston valve with steam inside.

Position of Eccentric.—Following the crank at an angle of 90° less steam lap and lead.

NOTE.—As in the case of other single eccentric gears (such as Hackworth's and Bremme's) the travel of the pulley exceeds the actual travel of the valve, as the motion is reduced down from the extended end of the quadrant in the present case.

Action.—The eccentric rod 1 gives motion to the rocking quadrant 2, which is hinged on a bracket cast on the engine framing, and this is transmitted to the valve by means of the link 3 and travelling lever 4. This lever is held at the other end to a bracket cast on the crosshead, and travels to and fro with the piston rod stroke. The valve spindle is connected to lever 4 at a point *near* the end, thus giving a small leverage which allows for the "lap + lead" travel of the valve, the remainder of the travel required to give port opening being obtained by the rocking motion of the quadrant produced by the eccentric 5 being the "ahead" position, and 6 the "astern" position of the link, as shown by the dotted lines. The drag link 10, moved by the reversing engine links 11, changes over the link block to the "ahead" or "astern" position as required, and the gear can be linked up by means of the expansion slot shown in the reversing bell crank. In all positions of gear the lead remains constant, and is unaffected by linking up.

NOTE.—The Sketch shows the gear as applied to a diagonal type paddle engine, but if the reader turns the page round so that the gear assumes a vertical position with the shaft below, the position of the gear as applied to an ordinary triple expansion marine engine will be obtained, and can be studied.

Bryce-Douglas Gear (Single Eccentric) (Sketch No. 39).

In this gear a fixed quadrant and travelling block, actuated by a single pulley, is employed, together with a link, lever, and bell crank.

Type of Valve fitted.—Slide valve or piston valve with steam over ends.

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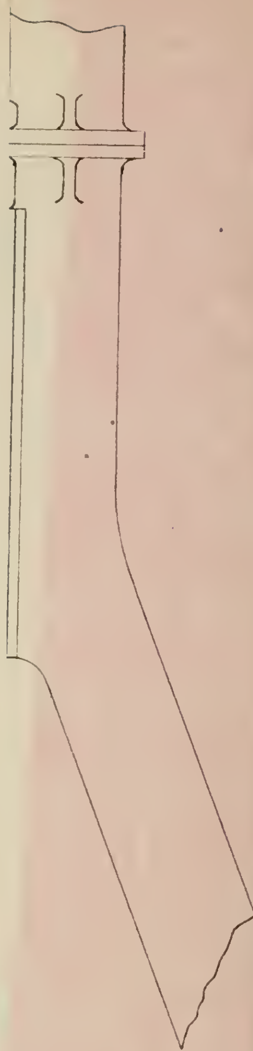


[To face page 227.

The small Sketch, No. 37, shows the gear in "ahead" position, with the slipper at the upper limit of its travel on the inclined bars. In this gear the bearing of the bell crank (6) shaft is subject to the most wear, and requires the most frequent overhaul, as the setting of the valve is affected by wear down of this bearing.

Brock Gear (Single Eccentric) (Sketch No. 38).

In this gear a rocking quadrant actuated by the eccentric is employed to convey the motion to the valve.

Type of Valve fitted.—Piston valve with steam inside.

Position of Eccentric.—Following the crank at an angle of 90° less steam lap and lead.

NOTE.—As in the case of other single eccentric gears (such as Hackworth's and Bremme's) the travel of the pulley exceeds the actual travel of the valve, as the motion is reduced down from the extended end of the quadrant in the present case.

Action.—The eccentric rod 1 gives motion to the rocking quadrant 2, which is hinged on a bracket cast on the engine framing, and this is transmitted to the valve by means of the link 3 and travelling lever 4. This lever is held at the other end to a bracket cast on the crosshead, and travels to and fro with the piston rod stroke. The valve spindle is connected to lever 4 at a point *near* the end, thus giving a small leverage which allows for the "lap + lead" travel of the valve, the remainder of the travel required to give port opening being obtained by the rocking motion of the quadrant produced by the eccentric 5 being the "ahead" position, and 6 the "astern" position of the link, as shown by the dotted lines. The drag link 10, moved by the reversing engine links 11, changes over the link block to the "ahead" or "astern" position as required, and the gear can be linked up by means of the expansion slot shown in the reversing bell crank. In all positions of gear the lead remains constant, and is unaffected by linking up.

NOTE.—The Sketch shows the gear as applied to a diagonal type paddle engine, but if the reader turns the page round so that the gear assumes a vertical position with the shaft below, the position of the gear as applied to an ordinary triple expansion marine engine will be obtained, and can be studied.

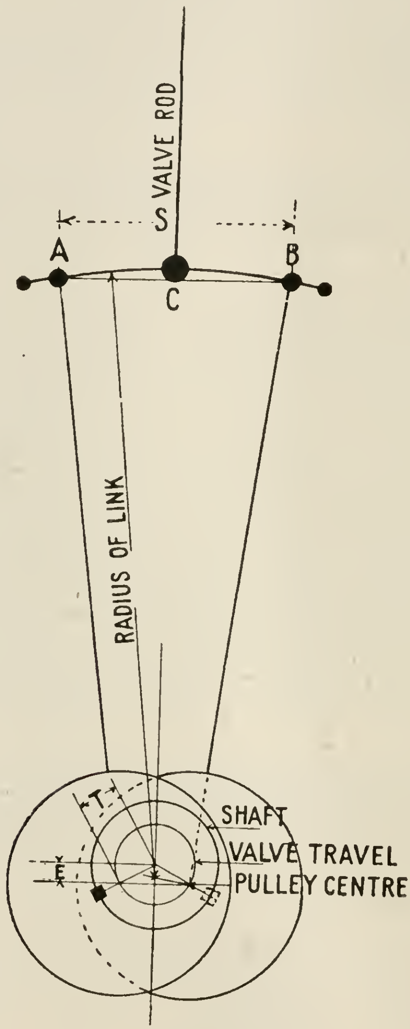
Bryce-Douglas Gear (Single Eccentric) (Sketch No. 39).

In this gear a fixed quadrant and travelling block, actuated by a single pulley, is employed, together with a link, lever, and bell crank.

Type of Valve fitted.—Slide valve or piston valve with steam over ends.

Position of Eccentric.—At an angle of 90° plus lap and lead in advance of the crank.

Action.—The eccentric rod block travels back and forward in the slot of the quadrant, which is hinged on a bracket bearing as shown, the angle given to the quadrant by the reversing engine 8 determining the direction of rotation whether "ahead" or "astern," 6 being ahead and 7 astern: from the quadrant the motion is carried to a fixed



No. 40.

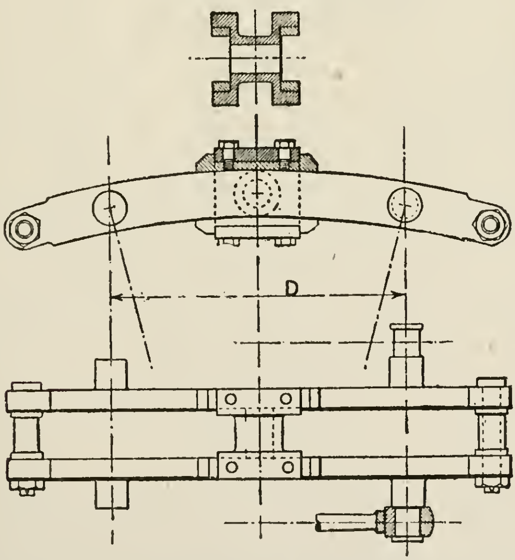
bell crank 3 by a link 4, the other arm of the bell crank finally giving the motion to the valve link 5, through the lever 2, suspended from the engine crosshead by the small link 1. Observe that the lever 2 is fulcrummed near the end by a pin to the bell crank, and the leverage to the valve spindle obtained in this way allows for the "steam lap+lead" travel of the valve, the additional travel necessary to give the required port opening being obtained by the travel of the block of link 4 in the quadrant; the lead is therefore constant for all positions of the gear, whether "full out" or "shut in."

As before stated, the quadrant is *hinged* by the centre, and is canted over to the required ahead or astern angle by the drag link of the reversing engine, and remains in that position, the block travelling back and forward by the action of the eccentric.

Link Motion.—In the most modern types of reciprocating engines, the "Stephenson" link motion gear is generally fitted, patent valve gears having been not altogether satisfactory in many respects, experience proving the superiority of the old type of gear.

Observe that the link radius is equal to the distance from the pin centre B to the pulley centre, and the centre of curvature is found by describing an arc from the pulley centre to the shaft centre line, as shown by the small cross.

S = valve travel × 3.
T = throw or eccentricity.
E = steam lap plus lead.



No. 41.—Reversing Quadrant and Block.

NOTE.—Distance D should be equal to three times the valve travel.

Linking Up.—Assuming pin A to be in line for "full gear," in linking up, the pin B is moved over towards the valve spindle block C, so that the effect is to *reduce* the valve travel.

The general results of linking up are as follows:—

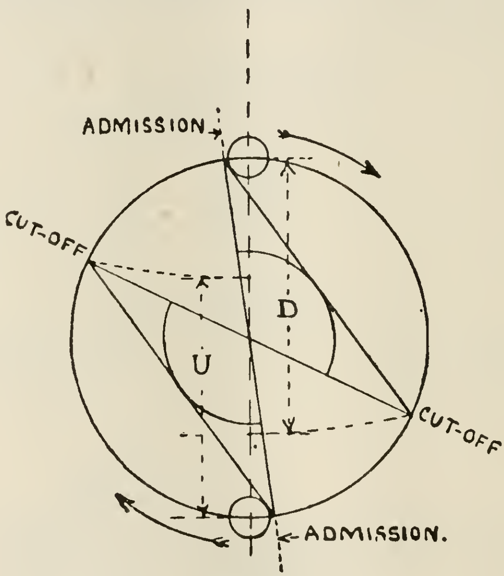
(1) Travel reduced.
(2) Port opening reduced (producing wire-drawing of the steam).
(3) Cut-off sooner.
(4) Lead increased (with rods "open," crank on bottom).
(5) Compression increased.

Generally speaking, all points occur earlier.

NOTE.—With the gear in mid position as shown, if the engine is turned one revolution with the turning gear, the valve will travel a distance equal to twice the steam lap and lead.

EXAMPLE.—To prove by a valve diagram that with equal laps on the valve, top and bottom and ordinary link motion valve gear, the steam is cut off sooner on the up stroke.

NOTE.—This difference in cut-off is caused by the connecting rod and crank angle.



No. 42.

Referring to No. 42, set off top and bottom of the valve travel circle, small circles equal in radius to the lead; also from the centre of the travel circle set off with the compasses the amount of steam lap on the valve; now draw tangents to both the lead circles and the lap arcs, and the crank angles at "Admission" and "Cut-off" will be obtained.

Next, take with the compasses the length of the connecting rod as radius, and putting the pencil on the crank position at "Cut-off," and the needle on the centre line, draw arcs *inwards* to the centre line to obtain the distance the steam is carried on the down and up strokes respectively. It will then be found, on measuring, that the distance U is less than the distance D, that is, the cut-off occurs earlier on the up stroke than on the down stroke, owing to the angle formed by the connecting rod and the crank.

NOTE.—For complete explanation of Valve Diagram, see page 248.

Eccentric Keyseat Templates.

Without Steam Lap and Lead.—If a valve has no steam lap lead, as, for example, a stearing gear engine valve, the keyseat position is at right angles, or 90° to the crank leading it (Sketch No. 43). In this case the steam is carried the full length of stroke, and, with the crank on the centre, the valve is exactly at mid position, ready to open for steam. The valve travel will then be equal to twice the port opening.

With Steam Lap and Lead.—When a slide valve has steam lap and lead the sum of the *mean* steam lap and lead must be measured down from the centre, and a horizontal line drawn through the travel circle, then lines drawn out to the shaft circle through the points of intersection from the centre, will give the correct keyway positions ahead and astern (Sketch No. 44). The valve travel circle diameter is equal to *twice* the steam lap and steam port opening. The keyseat position is therefore 90°, plus lap and lead, in advance of the crank, as when the crank is centred the valve is lower than mid position by a distance equal to the steam lap and lead.

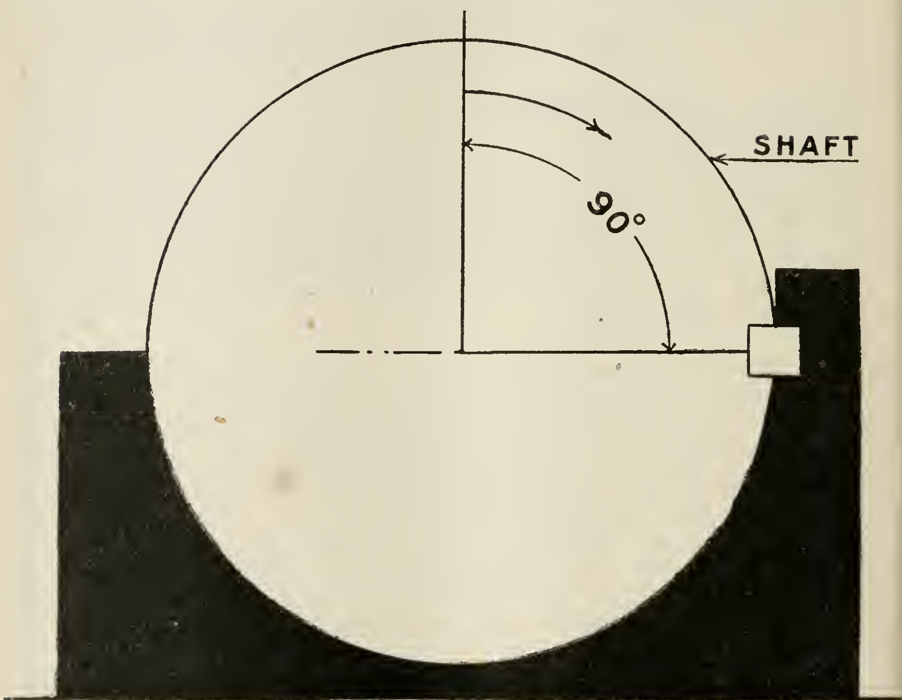
Piston Valves.—For a piston valve of the inside steam type as commonly constructed, the valve travel motion is reversed from that of a slide valve, as, instead of moving down to give lead and steam to the top port the valve requires to move up. This necessitates the position of the keyways being changed to scarcely the opposite side of the shaft (Sketch No. 46), the position being therefore 90° *behind* the crank *less* mean steam lap and lead. In setting off the keyseat template the mean steam lap and lead have to be measured up from the shaft centre.

To sum up, for a common slide valve or a double ported slide valve the keyway is cut at an angle greater than 90° leading the crank, but for a piston valve the keyway is cut at an angle less than 90° following the crank.

NOTE.—After the keyways are cut and the pulley secured to the shaft, a liner may require to be fitted under the rod if a slide valve, or a liner taken out if a piston valve, to give more lead at bottom than top.

Eccentric Keyseats.

Crank on Top Centre.

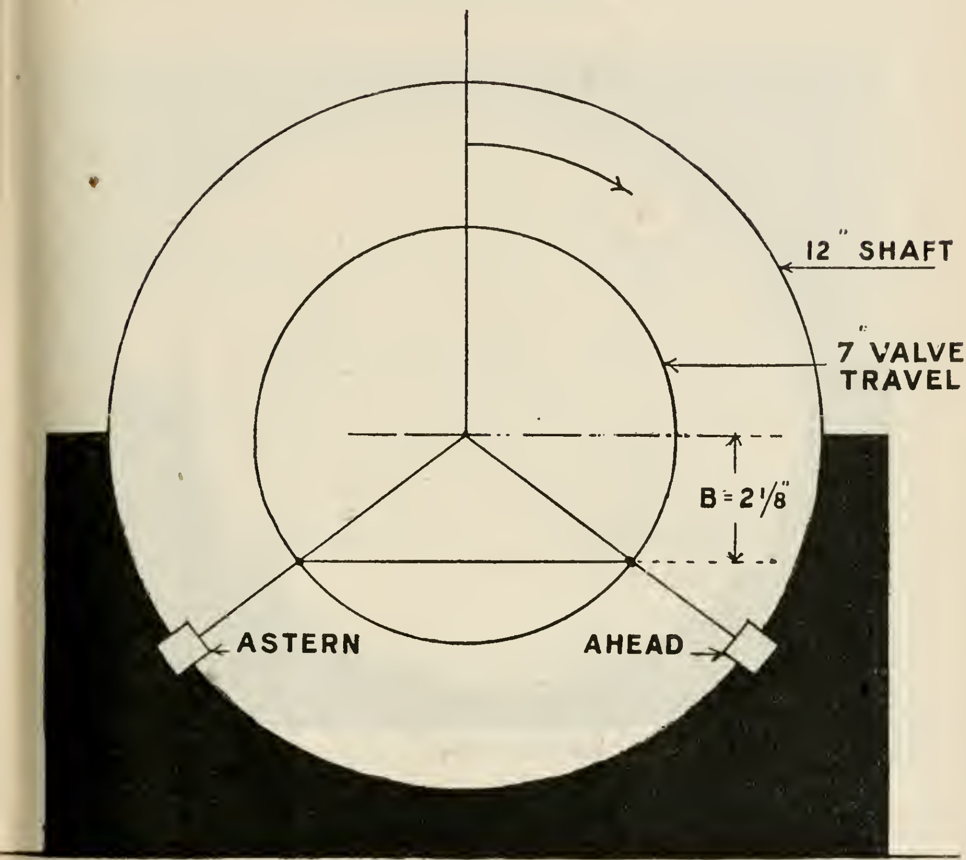


No. 43.—Valve without Steam Lap and Lead

(Steering Gear Engine Valve).

Keyseat at right angles to crank, if a slide valve leading the crank, if a piston valve following the crank.

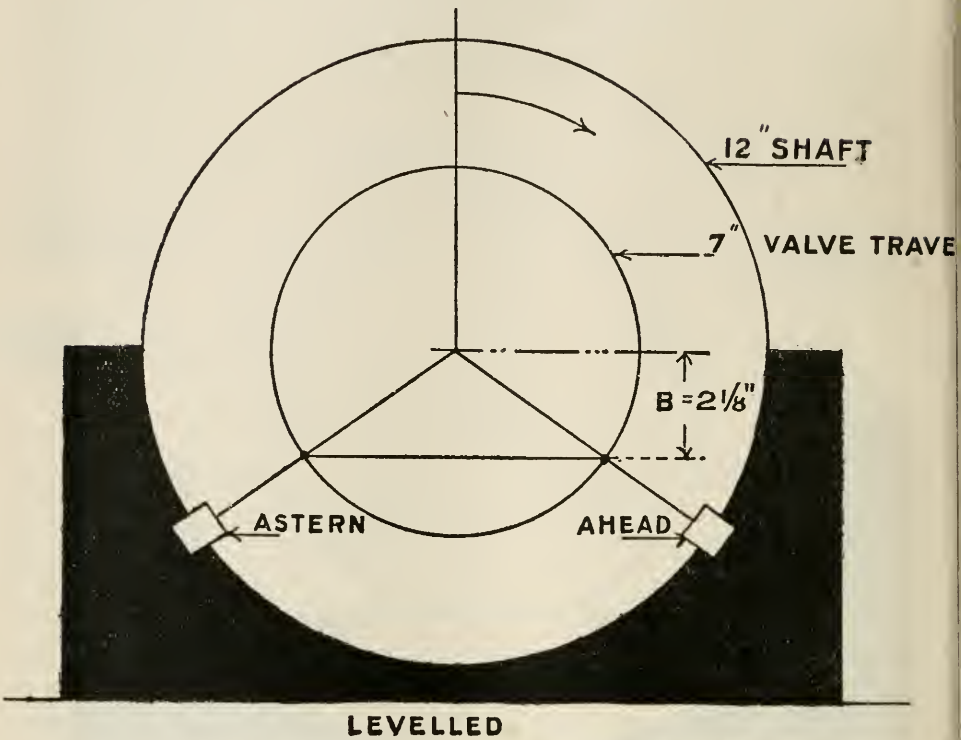
Crank on Centre.



No. 44.—Slide Valve with Steam Lap and Lead.

Shaft, 12 in. diameter.
Mean steam lap, 2 in.
Mean port opening, $1\frac{1}{2}$ in.
Mean lead, $\frac{1}{8}$ in.
Then, $(2+1\cdot5)\times 2=7$ in. valve travel.
And, B = steam lap + lead $= 2+\frac{1}{8}=2\frac{1}{8}$ in.

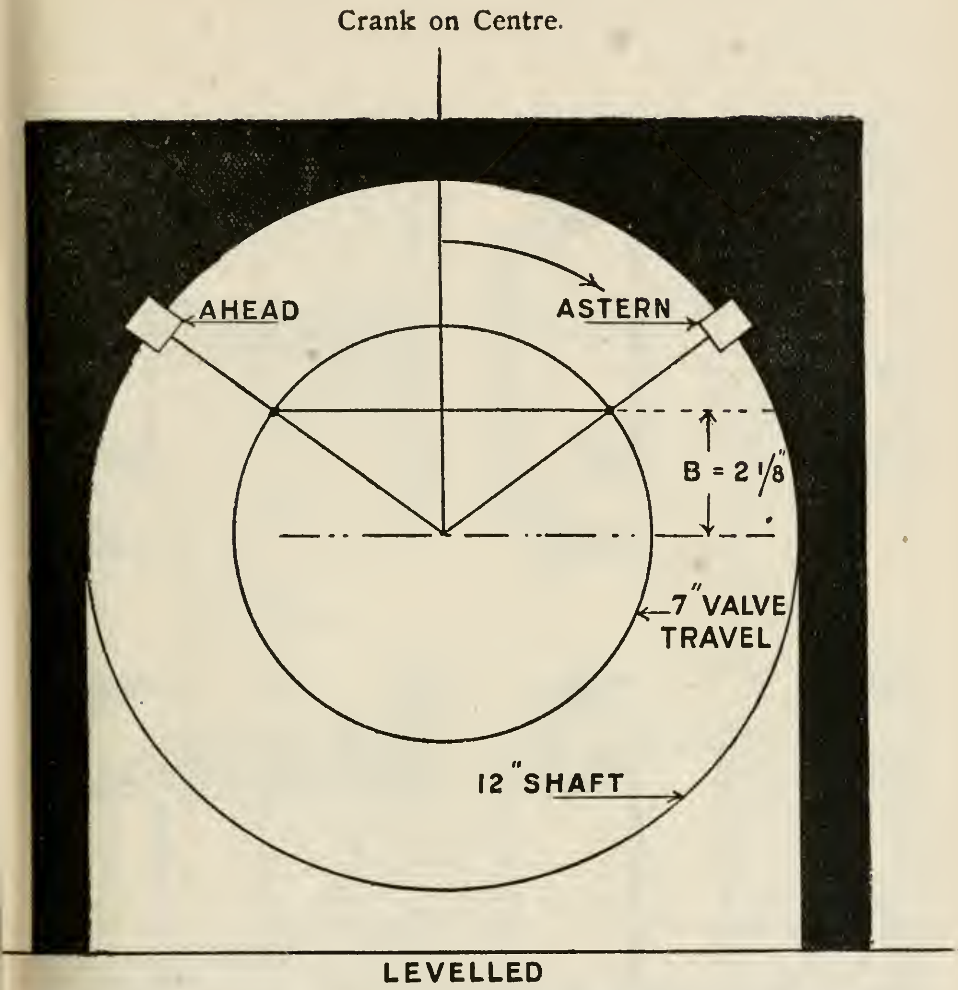
Crank on Centre.



No. 45.—Double Ported Slide Valve.

Shaft, 12 in. diameter.
Mean steam lap (each of two, top or bottom), 2 in.
Mean port opening „ „ „ $1\frac{1}{2}$ in.
Mean lead, $\frac{1}{8}$ in.
Then, $(2+1{\cdot}5)\times 2=7$ in. valve travel.
And, B = steam lap + lead $= 2+\frac{1}{8}=2\frac{1}{8}$ in.

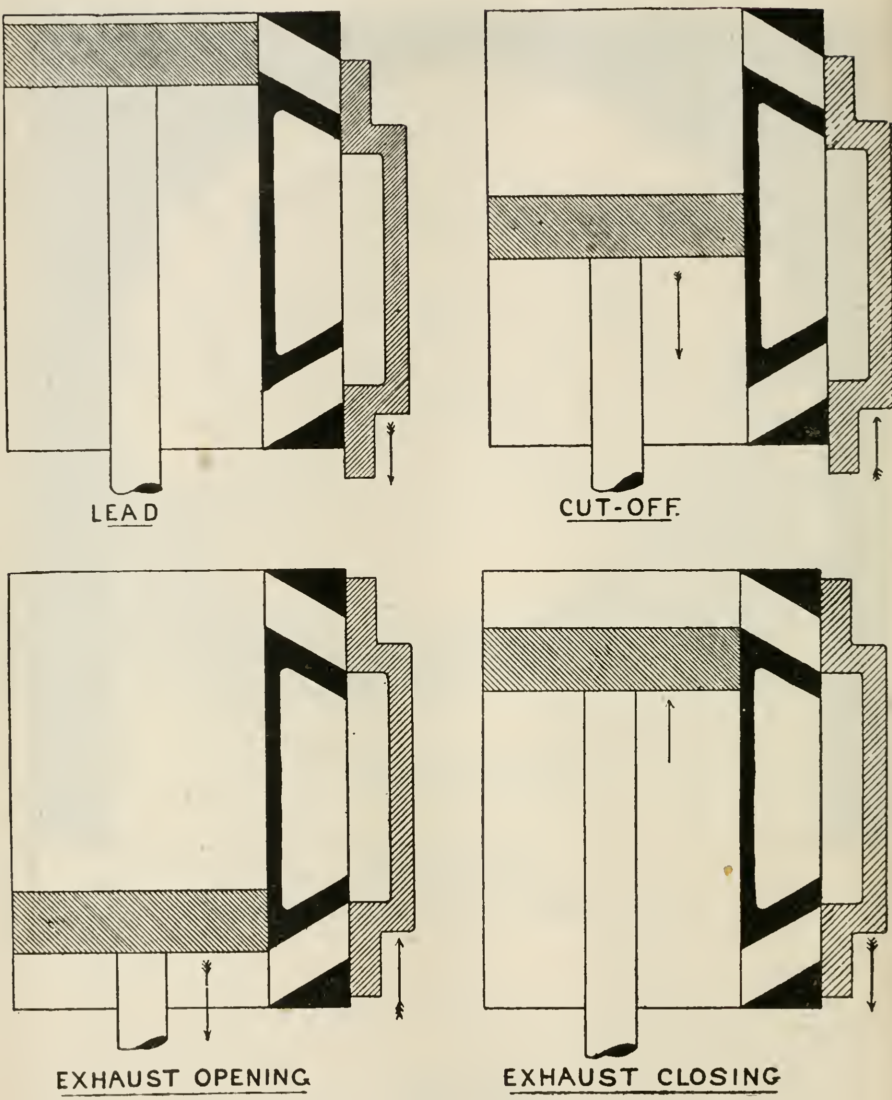
NOTE.—Only one of the two top or bottom laps and port openings are taken, and not the combined or total lap and port opening at either end.



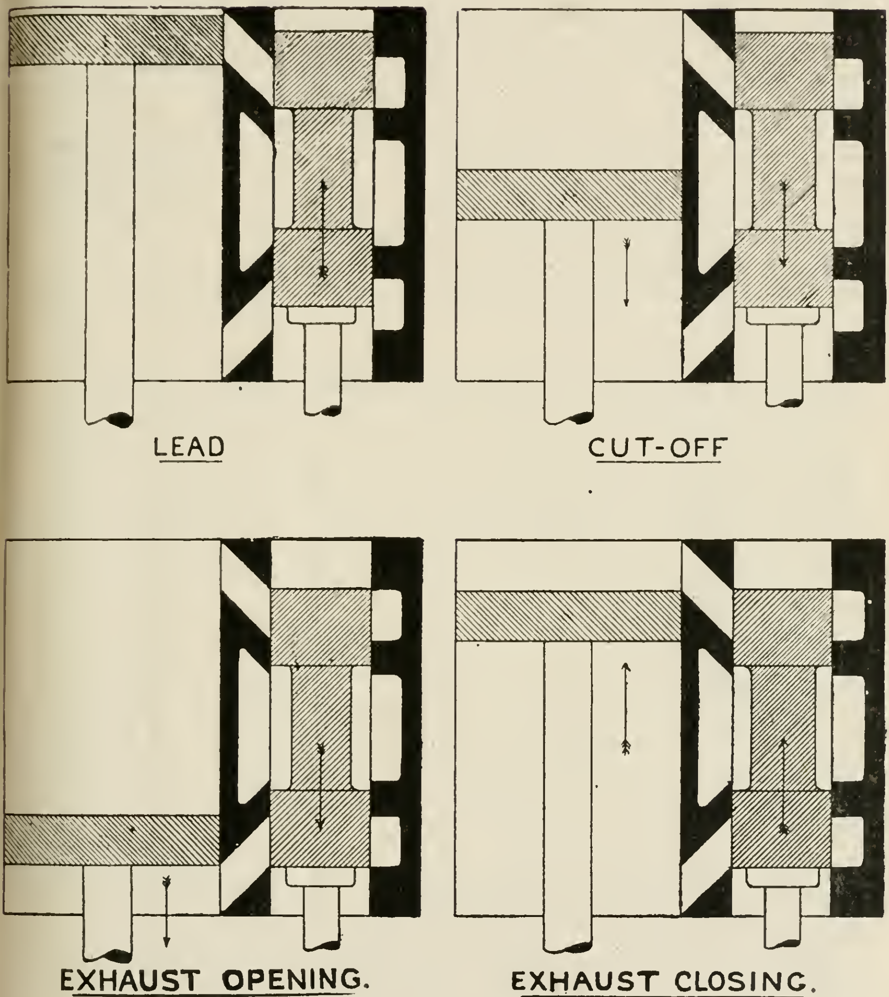
No. 46.—Inside Steam Piston Valve.

Shaft, 12 in. diameter.
Mean steam lap, 2 in.
Mean port opening, $1\frac{1}{2}$ in.
Mean lead, $\frac{1}{8}$ in.
Then, $(2+1{\cdot}5)\times 2=7$ in. valve travel.
And, $B = \text{steam lap} + \text{lead} = 2+\frac{1}{8}=2\frac{1}{8}$ in.

NOTE.—As the valve motion is reversed from that of a slide valve, the mean steam lap and lead are measured up from the centre with crank on top.



No. 47.—Slide Valve and Piston Positions.



No. 48.—Piston Valve and Piston Positions.

Action of Steam in a Cylinder.

During one revolution the action of the steam on one side of the piston is as follows:—With the crank on the top centre the top steam port is open for the amount of lead; the valve then moves down and opens the port further, and the piston moves down to, say, half stroke, when the valve moving up again cuts off the supply of steam. The steam in the cylinder expands and forces the piston down towards the end of the stroke, and when *near* the bottom centre the port opens to the exhaust; the piston then completes the stroke and travels up again, and when *near* the top centre the port is closed to exhaust. The steam thus retained in the cylinder is compressed by the returning piston to an increased pressure, and cushioning is effected; the piston next reaches the top centre, and the port again opening for lead, the same cycle of operations is repeated.

Notice that "exhaust opening" occurs when the piston is near the end of one stroke, and "exhaust closing" when the piston is near the end of the other stroke.

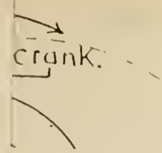
NOTE.—A piston valve travels in the reverse direction to that of a slide valve in the above cycle of operations, as will be seen by comparing the sketches of each.

Observe that in "lead" and in "cut-off" the valve is in the same position, but going down for "lead" and going up for "cut-off." Also that for "exhaust opening" and "exhaust closing" the valve is also in the same position, but going up for "exhaust opening" and going down for "exhaust closing." Again notice that the piston is *near* the bottom for "exhaust opening," and near (not at) the top for "exhaust closing."

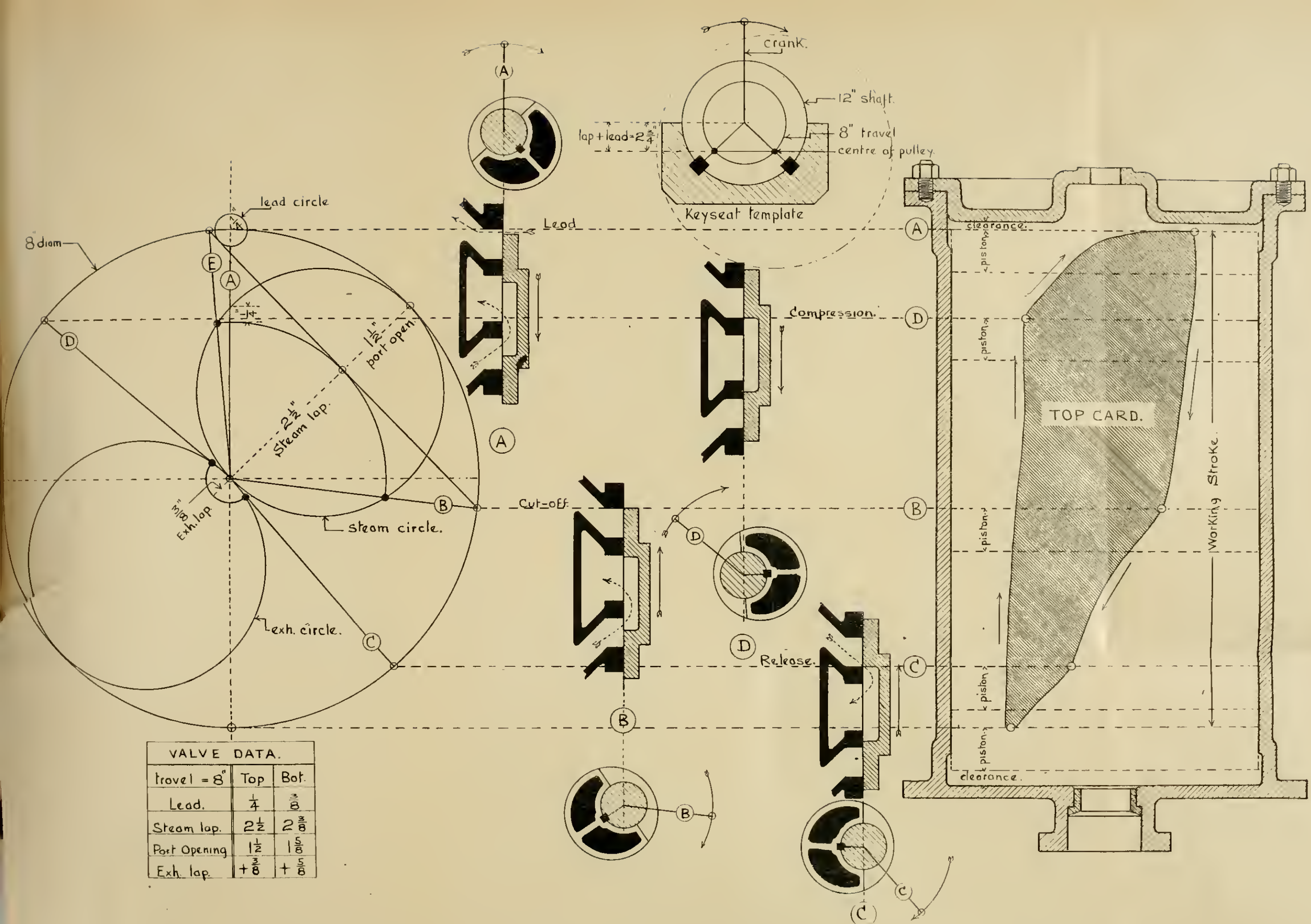
Observe that as steam is entering from between the pistons, the valve requires to travel in the reverse direction to that of a slide valve to give similar results.

In "lead" and in "cut-off" the valve is in the same position, but is going *up* for "lead" and going *down* for "cut-off." In "exhaust opening" and in "exhaust closing" the valve is in the same position, but is going *down* for "exhaust opening" and *up* for "exhaust closing."

NOTE. — Between the positions of "Cut-off" and "Exhaust Opening" ("Release") the steam in the cylinder expands in approximate accordance with Boyle's Law of Expansion; that is, the volume is increasing and the pressure decreasing proportionally: between the positions "Exhaust Closing" ("Compression") and "Lead" the steam is also following out this law but reversed in action, as in this case the volume is decreasing and the pressure increasing proportionally.



(shut in)	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
ern - -	$8\frac{3}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	2	$1\frac{15}{16}$	$1\frac{7}{8}$	$2\frac{3}{8}$	$-\frac{3}{16}$	$+\frac{7}{16}$	$24\frac{7}{8}$	$23\frac{3}{8}$	$2\frac{3}{4}$	$2\frac{1}{2}$	$3\frac{9}{16}$	$3\frac{5}{16}$



VALVE DATA.		
travel = 8"	Top	Bot.
Lead.	$\frac{1}{4}$	$\frac{3}{8}$
Steam lap.	$2\frac{1}{2}$	$2\frac{3}{8}$
Port Opening	$1\frac{1}{2}$	$1\frac{5}{8}$
Exh. lap.	$+\frac{3}{8}$	$+\frac{5}{8}$

No. 48a.—Valve and Piston Positions.

For one revolution, top side of piston.

(Angle of connectiog rod and crank neglected.)

A. Lead B. Cut off. C. Release (exhaust opening). D. Compression (exhaust closing). E. Admission.

The above diagram illustrates the comparative positions of piston and valve referred to the valve diagram, cylinder stroke, and indicator card: the relative positions of crank and eccentric are also shown throughout. Observe that the valve data table given corresponds with the keyseat template, but it should he carefully noted, that after cutting the keyseats and bolting on the pulleys a liner is required to go in under the foot of the rod to give the required difference in steam lap and lead top and bottom, in this case a liner $\frac{1}{16}$ in. thick, or equal in thickness to half the lead difference, as $\frac{\frac{3}{8}-\frac{1}{4}\text{ in.}}{2} = \frac{1}{16}$ in. liner required. By "admission" is meant the position of the crank when steam *begins* to enter the cylinder, "lead" being the amount the port is actually open when the crank is on the dead centre, in the present case $\frac{1}{4}$ in. Notice where this shows on the valve diagram.

The following points should be carefully noted.

1. **Top Steam Lap + Lead = Bottom Steam Lap + Lead.**
2. **,, ,, + Port opening = ,, ,, Port opening = half valve travel.**

Therefore Top $2\frac{1}{2} + \frac{1}{4} = 2\frac{3}{4}$. Also, Top $2\frac{1}{2} + 1\frac{1}{2} = 4$ in. (half travel).

,, Bot. $2\frac{3}{8} + \frac{3}{8} = 2\frac{3}{4}$. ,, Bot. $2\frac{3}{8} + 1\frac{5}{8} = 4$,, ,,

NOTE.—The reader is advised to study carefully the various positions of valve and piston as shown above.

NOTE.—For detailed description of valve diagram construction see page 248.

To face page 258.

Valve Setting Tables.

The following tables of valve settings, showing lead, steam lap, exhaust lap, cut-off, &c., for both up and down strokes, are from a set of engines of a modern fast passenger steamer, and should be carefully studied. The differences in lead, steam lap, port opening, cut-off, &c., occurring on the up and down strokes, and chiefly due to the effect of the angle of the connecting rod and crank, when link motion is fitted, are of great importance to the student, and the writer strongly advises special attention to this subject, as being one of particular interest and benefit to the marine engineer.

No. 1—Type ;—Fast Passenger Steamer.—I.H.P., 4500 ; Speed, 21 knots ; Cylinders, 27, 44, 70 inches ; Stroke, 2 feet 9 inches ; Boiler pressure, 185 lbs. ; Revolutions, 180 ; H.P. cylinder M.E.P. = 65 lbs. ; I.P. cylinder M.E.P. = 32·2 lbs. ; L.P. cylinder M.E.P. = 16 lbs. ; Link motion valve gear.

H.P. Piston Valve.

Expansion Grade.	Valve Travel.	Lead.		Steam Lap.		Port Opening.		Exhaust Lap.		Cut-off.		Exhaust Opening (from end of stroke).		Exhaust Closing (from end of stroke).	
		Top	Bot.	Top.	Bot.	Top.	Bot.	Top.	Bot.	Top.	Bot.	Top.	Bot.	Top.	Bot.
	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.
5 (full out)	8	$\frac{3}{8}$	$\frac{7}{16}$	$1\frac{13}{16}$	$1\frac{3}{4}$	$2\frac{3}{16}$	$2\frac{1}{4}$	$-\frac{3}{16}$	$+\frac{5}{16}$	$26\frac{1}{4}$	$23\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{1}{16}$	$2\frac{3}{16}$
S (shut in)	$6\frac{3}{4}$	$\frac{11}{16}$	$\frac{21}{32}$	$1\frac{13}{16}$	$1\frac{3}{4}$	$1\frac{9}{16}$	$1\frac{5}{8}$	$-\frac{3}{16}$	$+\frac{5}{16}$	$21\frac{3}{8}$	$16\frac{7}{16}$	$5\frac{1}{16}$	$5\frac{9}{16}$	6	$5\frac{1}{2}$
stern - -	$8\frac{1}{16}$	$\frac{13}{32}$	$\frac{11}{32}$	$1\frac{13}{16}$	$1\frac{3}{4}$	$2\frac{1}{2}$	$1\frac{15}{16}$	$-\frac{3}{16}$	$+\frac{5}{16}$	$27\frac{1}{8}$	$22\frac{3}{4}$	$2\frac{1}{4}$	$2\frac{3}{4}$	3	$2\frac{7}{16}$

I.P. Piston Valve.

Expansion Grade.	Valve Travel.	Lead.		Steam Lap.		Port Opening.		Exhaust Lap.		Cut-off.		Exhaust Opening (from end of stroke).		Exhaust Closing (from end of stroke).	
(full) -	8	$\frac{3}{8}$	$\frac{7}{16}$	2	$1\frac{15}{16}$	2	$2\frac{1}{16}$	$-\frac{3}{16}$	$+\frac{7}{16}$	$24\frac{3}{4}$	$21\frac{3}{4}$	$2\frac{7}{8}$	$2\frac{3}{4}$	$3\frac{3}{8}$	$3\frac{1}{2}$
(shut in)	$6\frac{7}{8}$	$\frac{11}{16}$	$\frac{25}{32}$	2	$1\frac{15}{16}$	$1\frac{7}{16}$	$1\frac{17}{32}$	$-\frac{3}{16}$	$+\frac{7}{16}$	$19\frac{1}{4}$	$15\frac{5}{8}$	$5\frac{3}{4}$	$5\frac{3}{4}$	$6\frac{3}{4}$	$6\frac{5}{8}$
tern - -	$8\frac{3}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	2	$1\frac{15}{16}$	$1\frac{7}{8}$	$2\frac{3}{8}$	$-\frac{3}{16}$	$+\frac{7}{16}$	$24\frac{7}{8}$	$23\frac{3}{8}$	$2\frac{3}{4}$	$2\frac{1}{2}$	$3\frac{9}{16}$	$3\frac{5}{16}$

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Valve Setting Tables.

The following tables of valve settings, showing lead, steam lap, exhaust lap, cut-off, &c., for both up and down strokes, are from a set of engines of a modern fast passenger steamer, and should be carefully studied. The differences in lead, steam lap, port opening, cut-off, &c., occurring on the up and down strokes, and chiefly due to the effect of the angle of the connecting rod and crank, when link motion is fitted, are of great importance to the student, and the writer strongly advises special attention to this subject, as being one of particular interest and benefit to the marine engineer.

No. 1—Type ;—Fast Passenger Steamer.—I.H.P., 4500 ; Speed, 21 knots ; Cylinders, 27, 44, 70 inches ; Stroke, 2 feet 9 inches ; Boiler pressure, 185 lbs. ; Revolutions, 180 ; H.P. cylinder M.E.P. = 65 lbs. ; I.P. cylinder M.E.P. = 32·2 lbs. ; L.P. cylinder M.E.P. = 16 lbs. ; Link motion valve gear.

H.P. Piston Valve.

Expansion Grade.	Valve Travel.	Lead.		Steam Lap.		Port Opening.		Exhaust Lap.		Cut-off.		Exhaust Opening (from end of stroke).		Exhaust Closing (from end of stroke).	
		Top	Bot.	Top.	Bot.	Top.	Bot.	Top.	Bot.	Top.	Bot.	Top.	Bot.	Top.	Bot.
	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.
5 (full out)	8	$\frac{3}{8}$	$\frac{7}{16}$	$1\frac{13}{16}$	$1\frac{3}{4}$	$2\frac{3}{16}$	$2\frac{1}{4}$	$-\frac{3}{16}$	$+\frac{5}{16}$	$26\frac{1}{4}$	$23\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{1}{16}$	$2\frac{3}{16}$
3 (shut in)	$6\frac{3}{4}$	$\frac{11}{16}$	$\frac{21}{32}$	$1\frac{13}{16}$	$1\frac{3}{4}$	$1\frac{9}{16}$	$1\frac{5}{8}$	$-\frac{3}{16}$	$+\frac{5}{16}$	$21\frac{3}{8}$	$16\frac{7}{16}$	$5\frac{1}{16}$	$5\frac{9}{16}$	6	$5\frac{1}{2}$
stern - -	$8\frac{1}{16}$	$\frac{13}{32}$	$\frac{11}{32}$	$1\frac{13}{16}$	$1\frac{3}{4}$	$2\frac{1}{2}$	$1\frac{15}{16}$	$-\frac{3}{16}$	$+\frac{5}{16}$	$27\frac{1}{8}$	$22\frac{3}{4}$	$2\frac{1}{4}$	$2\frac{3}{4}$	3	$2\frac{7}{16}$

I.P. Piston Valve.

(full) -	8	$\frac{3}{8}$	$\frac{7}{16}$	2	$1\frac{15}{16}$	2	$2\frac{1}{16}$	$-\frac{3}{16}$	$+\frac{7}{16}$	$24\frac{3}{4}$	$21\frac{3}{4}$	$2\frac{7}{8}$	$2\frac{3}{4}$	$3\frac{3}{8}$	$3\frac{1}{2}$
; (shut in)	$6\frac{7}{8}$	$\frac{11}{16}$	$\frac{25}{32}$	2	$1\frac{15}{16}$	$1\frac{7}{16}$	$1\frac{17}{32}$	$-\frac{3}{16}$	$+\frac{7}{16}$	$19\frac{1}{4}$	$15\frac{5}{8}$	$5\frac{3}{4}$	$5\frac{3}{4}$	$6\frac{3}{4}$	$6\frac{5}{8}$
tern - -	$8\frac{3}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	2	$1\frac{15}{16}$	$1\frac{7}{8}$	$2\frac{3}{8}$	$-\frac{3}{16}$	$+\frac{7}{16}$	$24\frac{7}{8}$	$23\frac{3}{8}$	$2\frac{3}{4}$	$2\frac{1}{2}$	$3\frac{9}{16}$	$3\frac{5}{16}$

L.P. Double Ported Slide Valve.

Expansion Grade.	Valve Travel.	Lead.		Steam Lap.		Port Opening.		Exhaust Lap.		Cut-off.		Exhaust Opening (from end of stroke).		Exhau Closin (from e of strok	
		Top	Bot.	Top.	Bot.	Top.	Bot.	Top.	Bot.	Top.	Bot.	Top.	Bot.	Top.	B
	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	I
·60 (full) -	8	$\frac{7}{16}$	$\frac{1}{2}$	$2\frac{5}{16}$	$2\frac{1}{4}$	$1\frac{11}{16}$	$1\frac{3}{4}$	$-\frac{1}{2}$	$+\frac{1}{2}$	$21\frac{3}{4}$	$18\frac{1}{8}$	$4\frac{3}{4}$	$5\frac{1}{8}$	$3\frac{7}{8}$	3
·44 (shut in)	$7\frac{1}{4}$	$\frac{11}{16}$	$\frac{13}{16}$	$2\frac{5}{16}$	$2\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$-\frac{1}{2}$	$+\frac{1}{2}$	$15\frac{7}{16}$	$12\frac{7}{8}$	$8\frac{1}{8}$	$8\frac{5}{8}$	$7\frac{5}{16}$	6
Astern - -	$7\frac{15}{16}$	$\frac{7}{16}$	$\frac{1}{2}$	$2\frac{5}{16}$	$2\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{7}{8}$	$-\frac{1}{2}$	$+\frac{1}{2}$	$21\frac{5}{16}$	$19\frac{5}{16}$	$4\frac{7}{8}$	$4\frac{3}{4}$	$3\frac{13}{16}$	4

Piston Clearances.

	Top.	Bot.
	In.	In.
H.P. - - - -	$\frac{3}{8}$	$\frac{1}{2}$
I.P. - - - -	$\frac{1}{4}$	$\frac{3}{4}$
L.P. - - - -	$\frac{13}{32}$	$\frac{3}{4}$

Referring to the foregoing tables, the following points should be carefully noted :—

1. Half valve travel = Steam Lap + Port Opening.

Example 1.—H.P. valve at ·75 cut-off grade, valve travel = 8 inches.

Then, **Top**
$$\left\{\begin{array}{ll} \text{Steam lap} & = 1\frac{13}{16} \text{ inches} \\ \text{Port opening} & = 2\frac{3}{16} \quad ,, \\ \hline & \quad 4 \text{ inches} = \text{half valve travel,} \end{array}\right.$$

and, **Bottom**
$$\left\{\begin{array}{ll} \text{Steam lap} & = 1\frac{3}{4} \text{ inches} \\ \text{Port opening} & = 2\frac{1}{4} \quad ,, \\ \hline & \quad 4 \text{ inches} = \text{half valve travel.} \end{array}\right.$$

EXAMPLE 2.—H.P. valve at ·58 cut-off grade, valve travel = $6\frac{3}{4}$ inches.

Then, Top $\left\{\begin{array}{ll} \text{Steam lap} & = 1\frac{13}{16} \text{ inches} \\ \text{Port opening} & = 1\frac{9}{16} \quad " \\ \hline & 3\frac{3}{8} \text{ inches} = \text{half valve travel,} \end{array}\right.$

and, Bottom $\left\{\begin{array}{ll} \text{Steam lap} & = 1\frac{3}{4} \text{ inches} \\ \text{Port opening} & = 1\frac{5}{8} \quad " \\ \hline & 3\frac{3}{8} \text{ inches} = \text{half valve travel.} \end{array}\right.$

This holds goods in nearly all cases, the exceptions being those due to eccentric rod angles at linked up positions, which result in a very small difference in the sum of the steam lap and port opening as compared with half the valve travel.

2. In "gear full out" positions, the difference of the lead top and bottom is just equal to the difference in the steam lap top and bottom, or in other words, what is lost in steam lap is gained in lead, or the reverse.

EXAMPLE 1.—H.P. valve at ·75 cut-off grade.

Then, Top $\left\{\begin{array}{ll} \text{Steam lap} & = 1\frac{13}{16} \text{ inches} \\ \text{Lead} & = \frac{3}{8} \quad " \\ \hline & 2\frac{3}{16} \text{ inches,} \end{array}\right.$

and, Bottom $\left\{\begin{array}{ll} \text{Steam lap} & = 1\frac{3}{4} \text{ inches} \\ \text{Lead} & = \frac{7}{16} \quad " \\ \hline & 2\frac{3}{16} \text{ inches.} \end{array}\right.$

Therefore the sum of the top steam lap and lead is equal to the sum of the bottom steam lap and lead.

NOTE.—This necessary difference in lap and lead is obtained by means of liner adjustment under the valve rod, liners having to go in if a slide valve, but to come out if for a piston valve with inside steam.

3. With the link shut in, the following effects are produced :—

A. Reduced valve travel (with open rods, crank on bottom).

B. Reduced port opening (exactly equal to difference in valve travel).

C. Increased lead.

D. Earlier cut-off, exhaust opening ("Release"), and exhaust closing ("Compression").

NOTE.—The steam lap and exhaust lap remain constant throughout as shown in the tables, as these are part of the valve dimensions, and are therefore unaffected

by link alteration. The amount of steam or exhaust lap can only be varied by either pinning on a brass strip to give an increase, or by chipping off a piece of the valve to give a decrease. By linking up all points occur sooner, and the port opening is decreased in proportion to the decrease in valve travel.

E. Owing to the angle of the connecting rod and crank, when link motion is fitted, the cut-off on the up stroke (bottom) is invariably sooner than on the down stroke, and this cannot be avoided, although the reverse order of things would be much more suitable if it could be arranged for. Although the valve is lined up as far as possible to give less steam lap, and therefore more port opening on the bottom than the top, even this fails to equalise the cut-off, as reference to the tables of valve settings reproduced will show.

F. In astern gear the link is often designed to give an increased valve travel, and consequently port opening, to allow of rapid reversing of the engines, and it should be observed that the gear cannot be "shut in" in this position as the expansion slot in the reversing bell crank is generally arranged so as to lie in a *horizontal* position when in "ahead" gear and in a *vertical* position for "astern" gear (see page 43), thus making the position of the block in the slot non-effective as regards linking up when vertical.

G. The exhaust lap on the top of each valve is negative, and that on the bottom positive, much more compression being required on the bottom than on the top, but as this difference of exhaust lap is neutralised by the angle of the connecting rod, the actual position of exhaust opening and closing is not much different for either the up or the down stroke, and is often at the wrong end (see table).

"Extra" Gear.—Occasionally the link radius bar is extended, so that a small additional travel can, if desired, be given to the valve, known as extra gear, which gives a still later cut-off than full gear by the port opening being thus increased. In the "extra gear" position of the link the lead is slightly *decreased*, the travel of valve increased, and the port opening increased, with a correspondingly later cut-off, say from ·60 at "full gear" to ·67 at "extra gear." The extra gear is generally brought into action for a special spurt on trial trip runs, but of course can be used at any time if it is required to increase the power and speed.

In engines of well-balanced power the cut-off is generally latest in the H.P., earlier in the I.P., and earliest of all in the L.P. cylinder.

No. 2—**Type ;—Cargo Steamer.**—Speed, 11·2 knots ; I.H.P., 2360 ; Cylinders, 27, 46, 76 inches ; Stroke, 48 inches ; Receiver pressures, H.P., 180 lbs., I.P., 55 lbs., L.P., 16 lbs. ; Vacuum, 27 inches ; Revolutions, 63.

Valve Settings.

H.P. Piston Valve.

Expansion Grade (mean of top and bottom).	Valve Travel.	Lead.		Steam Lap.		Port Opening.		Cut-off.	
		Top.	Bot.	Top.	Bot.	Top.	Bot.	Top.	Bot.
	In.	In.	In.	In.	In.	In.	In.	In.	In.
·72 (full gear) -	$6\frac{1}{2}$	$\frac{7}{32}$ F	$\frac{3}{8}$	$1\frac{9}{16}$	$1\frac{1}{2}$	$1\frac{15}{16}$	2	$35\frac{3}{4}$	$33\frac{1}{2}$
·50 (shut in) -	$5\frac{3}{8}$	$\frac{3}{8}$ B	$\frac{9}{16}$	$1\frac{9}{16}$	$1\frac{1}{2}$	$1\frac{1}{8}$ B	$1\frac{1}{4}$ B	$25\frac{3}{8}$	$22\frac{1}{16}$
I.P. Double Ported Slide Valve.									
·65 (full gear) -	6	$\frac{3}{8}$	$\frac{3}{8}$ F	$1\frac{5}{8}$	$1\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$33\frac{1}{4}$	$29\frac{1}{8}$
·44 (shut in) -	$4\frac{7}{8}$	$\frac{1}{2}$	$\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{1}{2}$	$\frac{13}{16}$	$\frac{15}{16}$	$23\frac{1}{2}$	$19\frac{3}{8}$
L.P. Double Ported Slide Valve.									
·53 (full gear) -	8	$\frac{1}{2}$	$\frac{7}{16}$	$2\frac{1}{2}$	$2\frac{7}{16}$	$1\frac{1}{2}$	$1\frac{9}{16}$	$28\frac{1}{2}$	$22\frac{1}{2}$
·31 (shut in) -	$6\frac{7}{8}$	$\frac{5}{8}$	$\frac{5}{8}$	$2\frac{1}{2}$	$2\frac{7}{16}$ F	$\frac{15}{16}$	1 B	$17\frac{1}{4}$	$12\frac{3}{4}$

NOTE.—The small letter B signifies "bare" and the letter F "full."

Observe that *half* the valve travel is equal to steam lap and port opening, and that if the travel is decreased by shutting in the link the port opening (not the steam lap) is decreased in exact proportion. Also notice that at all grades of expansion the up stroke cut-off is sooner than the down stroke, the cause as previously explained being the angle of the connecting rod and crank when link motion valve gear is fitted.

"Linked-up" Effects.

1. Valve travel reduced.
2. Lead *increased.*
3. Steam lap unaltered.
4. Port opening reduced.
5. Cut-off, exhaust opening, and exhaust closing all occur earlier on the stroke.

If H.P. link is the one shut in then less steam is admitted to the engine as a whole, which means less revolutions, I.H.P., and speed. If the I.P. or L.P. links are shut in, no appreciable difference in total power results, but the pressures in the I.P. or L.P. receivers are varied, which produces a change in the distribution of the power developed in each cylinder of the engine, the adjustment of which depends greatly on the setting of the links of each valve.

No. 3—Type;—Cargo Steamer.—Cylinders, 27, 43, 72 inches; Stroke 51 inches; Boiler pressure, 180 lbs.; I.H.P., 2550; Link motion gear.

Valve Settings.

H.P. Piston Valve.

Expansion Grade (mean of top and bottom).	Valve Travel.	Lead.		Steam Lap.		Port Opening.		Cut-off.		Exhaust Lap.		Compression (from end).	
		Top.	Bot.	Top.	Bot.	Top.	Bot.	Top.	Bot.	Top.	Bot.	Top.	Bot.
	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.
·62	7	$\frac{1}{4}$	$\frac{3}{8}$	$2\frac{1}{16}$	$1\frac{15}{16}$	$1\frac{7}{16}$	$1\frac{9}{16}$	$33\frac{7}{8}$	$29\frac{3}{8}$	$\frac{1}{16}$	$\frac{25}{32}$	8	$8\frac{1}{2}$
I.P. Double Ported Valve.													
·66	7	$\frac{3}{8}$	$\frac{1}{2}$	$1\frac{7}{8}$	$1\frac{3}{4}$	$1\frac{5}{8}$	$1\frac{3}{4}$	$35\frac{1}{4}$	$31\frac{1}{2}$	$\frac{3}{8}$	$1\frac{1}{8}$	10	$10\frac{1}{2}$
L.P. Double Ported Valve.													
·66	7	$\frac{1}{2}$	$\frac{5}{8}$	$1\frac{13}{16}$	$1\frac{11}{16}$	$1\frac{11}{16}$	$1\frac{13}{16}$	$35\frac{3}{4}$	$31\frac{1}{2}$	$\frac{7}{16}$	$1\frac{3}{16}$	$10\frac{1}{2}$	11

Example 1.—Referring to the foregoing:—

Half travel $= 7 \div 2 = 3{\cdot}5 =$ Steam lap + port opening.

H.P. valve, top steam lap $= 2\frac{1}{16}$ inches.

,, ,, port opening $= 1\frac{7}{16}$,,

Then, $2\frac{1}{16} + 1\frac{7}{16} = 3\frac{1}{2}$ inches half travel.

L.P. valve, bottom steam lap $= 1\frac{11}{16}$ inches.

,, ,, port opening $= 1\frac{13}{16}$,,

Then, $1\frac{11}{16} + 1\frac{13}{16} = 3\frac{1}{2}$ inches half travel.

Example 2.—

Rule—

Top steam lap + lead = Bottom steam lap + lead.

H.P. valve, top steam lap $= 2\frac{1}{16}$ inches.

,, ,, lead $= \frac{1}{4}$ inch.

Then, $2\frac{1}{16} + \frac{1}{4} = 2\frac{5}{16}$ inches.

Again, H.P. valve, bottom steam lap $= 1\frac{15}{16}$ inches.

,, ,, lead $= \frac{3}{8}$ inch.

Then, $1\frac{15}{16} + \frac{3}{8} = 2\frac{5}{16}$ inches.

EXAMPLE 3.—

L.P. valve, top steam lap $=1\frac{13}{16}$ inches.
" " port opening $=1\frac{11}{16}$ "

Then, $1\frac{13}{16}+1\frac{11}{16}=3\frac{1}{2}$ inches half travel.

L.P. valve, top steam lap $=1\frac{13}{16}$ inches.
" " lead $=\frac{1}{2}$ inch.

Then, $1\frac{13}{16}+\frac{1}{2}=2\frac{5}{16}$ inches.

Again, L.P. valve, bottom steam lap $=1\frac{11}{16}$ inches.
" " lead $=\frac{5}{8}$ inch.

Then, $1\frac{11}{16}+\frac{5}{8}=2\frac{5}{16}$ inches.

Again observe that in each cylinder the cut-off takes place *earlier* on the up stroke.

No. 4—Type;—Fast Cargo Steamer.—I.H.P., 2200; Cylinders, 26, 44, 70 inches; Stroke, 48 inches.

Valve Settings.

H.P. Piston Valve.

Expansion Grade (mean of top and bottom).	Valve Travel.	Lead.		Steam Lap.		Port Opening.		Exhaust Lap.		Exhaust Opening.		Exhaust Closing.	
		Top	Bot.	Top	Bot.	Top.	Bot.	Top.	Bot.	Top.	Bot.	Top.	Bot.
	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.
·66	$6\frac{3}{4}$	$\frac{3}{16}$	$\frac{5}{16}$	2	$1\frac{13}{16}$ F	$1\frac{3}{8}$	$1\frac{9}{16}$	$+\frac{1}{8}$	$+\frac{9}{16}$	$3\frac{3}{8}$	$3\frac{1}{2}$	$6\frac{3}{4}$	6
I.P. Double Ported Slide Valve.													
·66	7	$\frac{3}{16}$	$\frac{5}{16}$	2	$1\frac{13}{32}$	$1\frac{9}{16}$ F	$1\frac{11}{16}$	$-\frac{3}{32}$	$+\frac{11}{32}$	$4\frac{7}{8}$	4	$5\frac{3}{4}$	$4\frac{3}{4}$
L.P. Double Ported Slide Valve.													
·66	7	$\frac{3}{16}$	$\frac{1}{4}$ F	2	$1\frac{13}{16}$	$1\frac{9}{16}$ F	$1\frac{3}{4}$ B	0	$+\frac{5}{8}$	$3\frac{3}{4}$	$3\frac{5}{8}$	6	$5\frac{7}{8}$

NOTE.—In the above example the sum of the steam lap and port opening shows a slight difference from half the valve travel in the case of the I.P. and L.P. valves, this being accounted for by the angle and crossing over action of the eccentric rods.

The mean cut-off in each cylinder = 32 inches.

Therefore, 32 inches ÷ 48 = ·66 Expansion Grade.

No. 5—Type ;—Large Cargo Passenger Steamer.—Speed, 14 knots ; I.H.P., 6000 ; Quadruple expansion cylinders, $31\frac{1}{2}$, 45, 64, 92 inches ; Stroke, 60 inches ; Boiler pressure, 200 lbs.

VALVE SETTINGS.

H.P. Piston Valve (20 inches diameter).

Expansion Grade (mean of top and bottom).	Valve Travel.	Lead.		Steam Lap.		Port Opening.		Cut-off.		Exhaust Lap.		Release.		Compression.		Piston Clearance.	
		Top	Bot.	Top.	Bot.	Top.	Bot.	Top.	Bot.	Top	Bot.	Top.	Bot.	Top	Bot.	Top	Bot.
	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.
Full gear	$9\frac{1}{2}$	$\frac{1}{8}$	$\frac{1}{4}$	$2\frac{9}{16}$	$2\frac{7}{16}$	$2\frac{7}{32}$	$2\frac{1}{4}$	$44\frac{1}{4}$	$39\frac{1}{2}$	$\frac{1}{8}$	$\frac{15}{16}$	4	$2\frac{7}{8}$	7	$7\frac{5}{8}$	$\frac{3}{8}$	1
Shut in -	$7\frac{19}{32}$	$\frac{11}{32}$	$\frac{7}{16}$	$2\frac{9}{16}$	$2\frac{7}{16}$	$1\frac{5}{16}$	$1\frac{9}{32}$	$32\frac{1}{2}$	$25\frac{3}{8}$	$\frac{1}{8}$	$\frac{15}{16}$	8	7	14	14	$\frac{3}{8}$	1
1st I.P. Martin and Andrews' Patent Valve (see page 207).																	
Full gear	$10\frac{1}{2}$	$\frac{3}{16}$	$\frac{3}{8}$	$2\frac{13}{16}$	$2\frac{9}{16}$	$2\frac{3}{8}$	$2\frac{9}{16}$	43	$38\frac{3}{8}$	$\frac{29}{64}$	$\frac{9}{32}$	$5\frac{3}{4}$	$5\frac{1}{8}$	6	$6\frac{3}{8}$	$\frac{5}{8}$	1
Shut in -	$8\frac{9}{16}$	$\frac{7}{16}$	$\frac{5}{8}$	$2\frac{13}{16}$	$2\frac{9}{16}$	$1\frac{9}{32}$	$1\frac{5}{8}$	$24\frac{1}{4}$	$26\frac{1}{4}$	$\frac{29}{64}$	$\frac{9}{32}$	$11\frac{1}{8}$	$10\frac{1}{8}$	$11\frac{3}{4}$	$12\frac{3}{4}$	$\frac{5}{8}$	1
2nd I.P. Martin and Andrews' Valve.																	
Full gear	$10\frac{1}{2}$	$\frac{1}{4}$	$\frac{1}{2}$	$2\frac{13}{16}$	$2\frac{1}{2}$	$2\frac{13}{32}$	$2\frac{3}{4}$	$44\frac{5}{8}$	40	$\frac{29}{64}$	$\frac{9}{32}$	$5\frac{3}{4}$	$5\frac{3}{4}$	5	$5\frac{1}{4}$	$\frac{5}{16}$	$\frac{29}{32}$
Shut in -	$8\frac{1}{2}$	$\frac{17}{32}$	$\frac{25}{32}$	$2\frac{13}{16}$	$2\frac{1}{2}$	$1\frac{13}{32}$	$1\frac{3}{4}$	$31\frac{3}{8}$	$28\frac{3}{8}$	$\frac{29}{64}$	$\frac{9}{32}$	$11\frac{5}{8}$	$11\frac{1}{8}$	10	$10\frac{7}{8}$	$\frac{5}{16}$	$\frac{29}{32}$
L.P. Double Ported Slide Valve.																	
Full gear	$9\frac{1}{2}$	$\frac{5}{16}$	$\frac{5}{8}$	$2\frac{13}{16}$	$2\frac{1}{2}$	$1\frac{7}{8}$	$2\frac{17}{64}$	$39\frac{1}{2}$	$35\frac{1}{4}$	$\frac{5}{16}$	$\frac{29}{32}$	$4\frac{7}{8}$	$5\frac{1}{4}$	$10\frac{7}{8}$	$9\frac{1}{2}$	$\frac{5}{16}$	$\frac{13}{16}$
Shut in -	$7\frac{11}{16}$	$\frac{17}{32}$	$\frac{13}{16}$	$2\frac{13}{16}$	$2\frac{1}{2}$	$1\frac{5}{32}$	$1\frac{9}{16}$	27	$24\frac{1}{2}$	$\frac{5}{16}$	$\frac{29}{32}$	9	$9\frac{1}{2}$	18	$16\frac{5}{8}$	$\frac{5}{16}$	$\frac{13}{16}$

NOTE.—By "release" is meant the position of the piston from end of stroke at moment of exhaust opening.

By "compression" is meant the position of the piston from end of stroke at moment of exhaust closing.

Mean Cut-off and Expansion Grade.—These are determined as follows :—

H.P. Cylinder, Full Gear—

Top cut-off $=44\frac{1}{4}$ inches,
Bottom cut-off $=39\frac{1}{2}$,,

Then, $\frac{44 \cdot 25+39 \cdot 5}{2}=41 \cdot 875$ inches mean cut-off.

Expansion Grade $=41 \cdot 875$ inches $\div 60$ inches (stroke) $= \cdot 69$ of stroke.

1st I.P. Cylinder, Full Gear—

Top cut-off $=43$ inches,
Bottom cut-off $=39\frac{3}{8}$,,

Then, $\frac{43+39 \cdot 375}{2}=41 \cdot 18$ inches mean cut-off.

Expansion Grade $=41 \cdot 18$ inches $\div 60$ inches $= \cdot 68$ of stroke.

2nd I.P. Cylinder, Full Gear—

Top cut-off $=44\frac{5}{8}$ inches,
Bottom cut-off $=40$,,

Then, $\frac{44 \cdot 625+40}{2}=42 \cdot 31$ inches mean cut-off.

Expansion Grade $=42 \cdot 31$ inches $\div 60$ inches $= \cdot 70$ of stroke.

L.P. Cylinder, Full Gear—

Top cut-off $=39\frac{1}{2}$ inches.
Bottom cut-off $=35\frac{1}{4}$,,

Then, $\frac{39 \cdot 5+35 \cdot 25}{2}=37 \cdot 37$ inches mean cut-off.

Expansion Grade $=37 \cdot 37$ inches $\div 60$ inches $= \cdot 62$ of stroke.

No. 6—Type;—Cargo Steamer.—I.H.P., 1630; Speed, 11·6 knots; Cylinders, 26, 42, 70 inches; Stroke, 48 inches; Boiler pressure, 180 lbs.; H.P. receiver, 175 lbs.; I.P., 56 lbs.; L.P., 9 lbs.; Vacuum, 24 inches; Revolutions, 60·5.

VALVE SETTINGS.

H.P. Piston Valve.

Expansion Grade (mean).	Valve Travel.	Lead.		Steam Lap.		Port Opening.		Cut-off.		Exhaust Lap.		Release.		Compression.	
		Top	Bot.	Top.	Bot.	Top.	Bot.	Top.	Bot.	Top	Bot.	Top	Bot.	Top.	Bot.
	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.
·52	7	1/4	3/8	2 5/16	2 3/16	1 3/16	1 5/16	27 1/2	22 3/4	1/16	13/16	5 3/4	5 1/2	9 3/4	9 3/4
I.P. D.P. Valve.															
·62	7	3/8	1/2	2	1 7/8	1 1/2	1 5/8	32 3/4	28 1/4	7/16	1 1/4	3 3/4	2 1/2	10 3/4	10 3/4
L.P. D.P. Valve.															
·56	7	1/2	11/16	2 1/16	1 15/16	1 3/8 F	1 5/8 B	29 1/2	25 1/4	5/16	1 1/4	4 3/4	3 1/2	12 1/2	12 3/4

NOTE.—The above is a very fair example of valve setting for a set of engines of the power given, and the results as tabulated represent good practice. It will be noticed that the "compression" or exhaust closing position occurs earlier in the I.P. than the H.P., and earlier still in the L.P. This is to allow for the heavier weight of the moving parts in these cylinders, the inertia of which has to be overcome at the end of each stroke.

Observe, then, that—

1. The amount of lead opening increases with each cylinder.
2. " " exhaust lap " " "
3. The moment of compression (exhaust closing) is earlier in each cylinder from H.P. to L.P.

Valve Diagrams.

By means of the well-known "valve diagrams" devised by Zeuner the eminent Swiss engineer and scientist, the required steam lap, port opening, exhaust lap, &c., may be calculated for a valve, and the following example from actual practice should be carefully studied.

EXAMPLE.—Engine stroke, 42 inches; connecting rod, 7 feet 6 inches in length; valve travel to be $7\frac{1}{2}$ inches; top lead, $\frac{1}{8}$ inch; bottom lead, $\frac{1}{4}$ inch; bottom exhaust lap $+\frac{1}{4}$ inch; top exhaust lap $-\frac{1}{4}$ inch.

(*A.*) Find the required steam lap and port opening for a maximum down stroke cut-off of ·7. (*B.*) Also find the cut-off on the up stroke due to the necessary difference in steam lap and lead at the bottom as compared with the top.

NOTE.—The sum of the steam lap and lead is the same top and bottom, therefore what is gained in lead at the bottom end is lost in steam lap; this also proportionally alters the maximum port opening.

Application.—Stroke, 42 inches $\times \cdot 7 = 29{\cdot}4$ inches cut-off on down stroke.

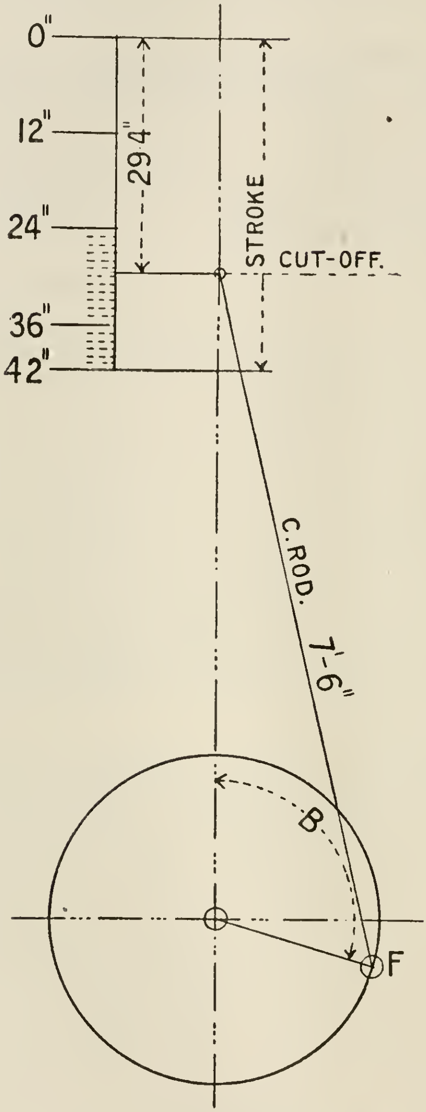
(*A.*) First set off to a small scale (say 1 inch to 1 foot) the small diagram of crank-pin circle and crosshead or piston travel as shown in No. 49, proceeding as follows:—

1. Set off vertically 42 inches by scale, then measure down from the centre of this, or half stroke the connecting rod length of 7 feet 6 inches which gives the centre of shaft; next with a radius of 21 inches (half stroke) set off the crank-pin travel circle. Now measure down from the top of stroke 29·4 inches (shown by the inch divisions from the 24-inch distance) and with the connecting rod length of 7 feet 6 inches in the compasses set the needle point on the crosshead centre and make a mark, F, on the crank-pin circle: this mark F is the position of crank-pin centre at cut-off. Connect the shaft centre and F, the crank-pin centre, which gives the crank angle at the "cut-off" position.

2. To a scale of full size, or at least half size, set off the valve travel circle of $7\frac{1}{2}$ inches diameter, and with the lead radius of $\frac{1}{8}$ inch also set off an arc from the top diameter of the circle marked as L; now transfer the angle of the crank at cut-off from the small diagram to the larger one, by describing a radius as at B, which is again repeated on the large diagram at B, and the length between taken in the compasses from the small diagram and measured off on the large one as shown: this gives the exact angle of the crank at cut-off, which is carried out as shown to point 2 on the valve travel circle.

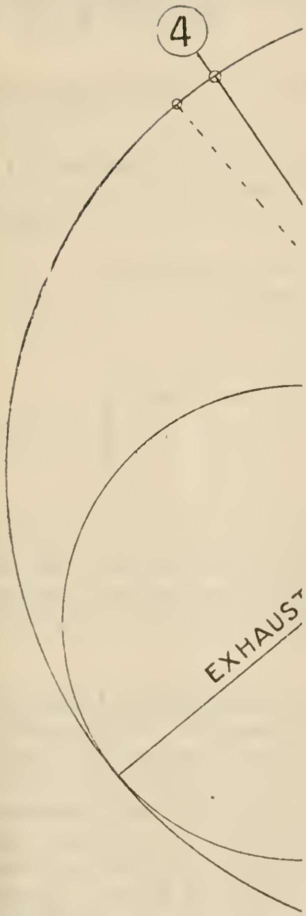
3. Draw a line from point 2 tangentially to the lead arc L, and where it cuts the valve circle put in by hand a small locating circle, also one at point 2; now bisect the line extending between points 1 and 2, by either describing arcs as shown, or by trial with the dividers, and draw out a line from the centre. We then find the required steam lap and maximum port opening as measured, the steam lap being $2\frac{3}{16}$ inches, and the port opening $1\frac{9}{16}$ inches, the sum of the two ($2\frac{3}{16}+1\frac{9}{16}$ inches) being, of course, equal to half the valve

travel, or $3\frac{3}{4}$ inches; finally describe a circle on the line as shown, called the primary valve circle.



No. 49.

4. Set off a small semicircle, E, $\frac{1}{4}$ inch radius on the line last mentioned, which semicircle represents the minus exhaust lap of the



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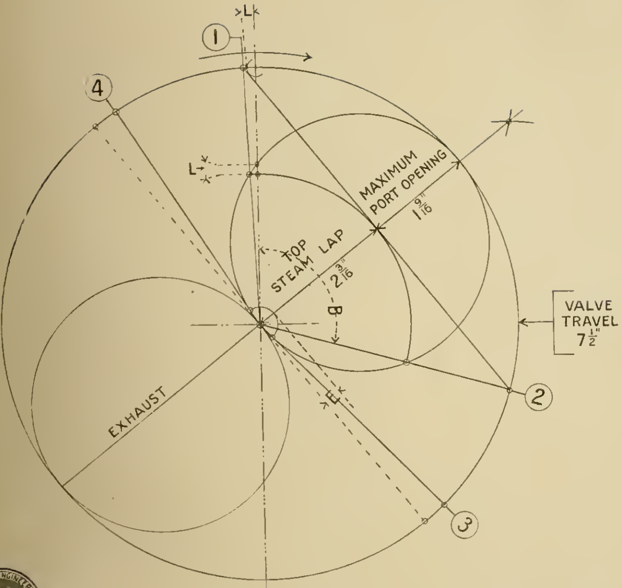
L=Le

1, Crank Angle at
2, ,, ,,



No. 50.—Valve Diagram for Top.

L = Lead $\frac{1}{8}$ inch. E = $-\frac{1}{4}$ inch Exhaust Lap.

1, Crank Angle at Lead.
2, ,, ,, Cut-off.
3, Crank Angle at Release.
4, ,, ,, Compression.

[*To face page* 250.

top, and run out lines from the centre through the points intersected on the primary valve circle; the lines thus drawn give the crank angle at exhaust opening, or "Release" (3), and exhaust closing, or "Compression" (4).

NOTE.—If the valve had neither plus nor minus exhaust lap, the dotted line shown (drawn parallel to the line 1, 2) would then give the crank angles at exhaust opening and closing.

Points of Importance.—1. Notice that, "Port opening" + "Steam lap" = Half travel.

2. The exhaust opening $= 3\frac{3}{4} + \frac{1}{4} = 4$ inches, but as the actual port width is less than this the opening is necessarily limited to the port width.

3. No. 1 crank angle at lead opening ("Admission").
 " 2 " " Cut-off.
 " 3 " " exhaust opening ("Release").
 " 4 " " " closing ("Compression").

Therefore, { From 1 to 2 steam is being admitted,
 " 2 " 3 " " expanded,
 " 3 " 4 " " exhausted,
 " 4 " 1 " " compressed.

Observe that in the vertical centre line the lead L is shown measured between the steam lap curve and the primary valve circle, and this (if the drawing is accurately done) will be exactly equal to the small arc L at the top, in this case $= \frac{1}{8}$ inch.

Release and Compression.—To find how far the piston is from the end of the stroke at "release" and "compression," transfer the angles of the crank at positions 3 and 4 *back to the small diagram* from the large one as shown in Sketch No. 51 (as described for *B*), then set off the connecting rod length 7 feet 6 inches on the small diagram upwards from point 3 and from point 4 to the centre line, at which positions draw horizontal lines for the crosshead centres; now set off a few inches by scale from either end, and measure how many inches are included from the crosshead centre line and end of stroke, which will be the positions of release and compression required. Notice that release occurs 5 inches from the bottom end of stroke, and compression $4\frac{3}{8}$ inches from the top end of stroke.

(*B.*) Referring to the data already given we find that the top steam lap and lead $= 2\frac{3}{16} + \frac{1}{8} = 2\frac{5}{16}$ inches, so that if the bottom lead is to be $\frac{1}{4}$ inch, then $2\frac{5}{16} - \frac{1}{4} = 2\frac{1}{16}$ inches steam lap at bottom, as the sum of the two must be the same.

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top, and run out lines from the centre through the points intersected on the primary valve circle; the lines thus drawn give the crank angle at exhaust opening, or "Release" (3), and exhaust closing, or "Compression" (4).

NOTE.—If the valve had neither plus nor minus exhaust lap, the dotted line shown (drawn parallel to the line 1, 2) would then give the crank angles at exhaust opening and closing.

Points of Importance.—1. Notice that, "Port opening" + "Steam lap" = Half travel.

2. The exhaust opening $= 3\frac{3}{4} + \frac{1}{4} = 4$ inches, but as the actual port width is less than this the opening is necessarily limited to the port width.

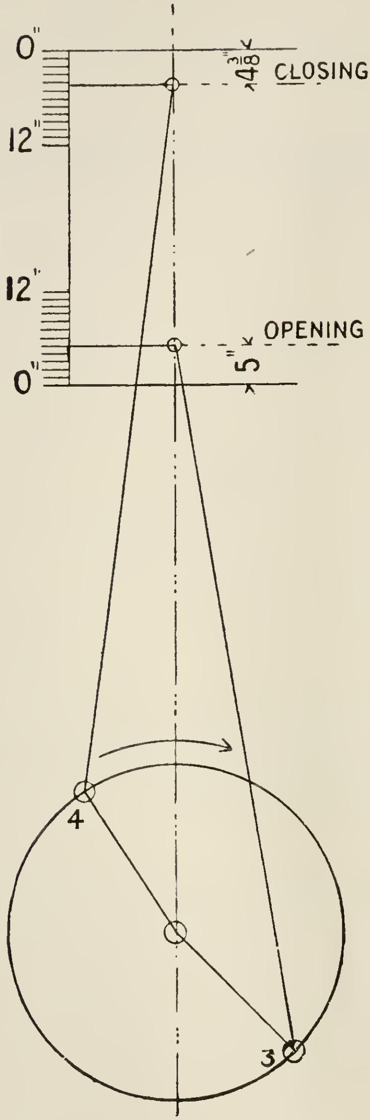
3. No. 1 crank angle at lead opening ("Admission").
" 2 " " Cut-off.
" 3 " " exhaust opening ("Release").
" 4 " " " closing ("Compression").

Therefore,
From 1 to 2 steam is being admitted,
" 2 " 3 " " expanded,
" 3 " 4 " " exhausted,
" 4 " 1 " " compressed.

Observe that in the vertical centre line the lead L is shown measured between the steam lap curve and the primary valve circle, and this (if the drawing is accurately done) will be exactly equal to the small arc L at the top, in this case $= \frac{1}{8}$ inch.

Release and Compression.—To find how far the piston is from the end of the stroke at "release" and "compression," transfer the angles of the crank at positions 3 and 4 *back to the small diagram* from the large one as shown in Sketch No. 51 (as described for *B*), then set off the connecting rod length 7 feet 6 inches on the small diagram upwards from point 3 and from point 4 to the centre line, at which positions draw horizontal lines for the crosshead centres; now set off a few inches by scale from either end, and measure how many inches are included from the crosshead centre line and end of stroke, which will be the positions of release and compression required. Notice that release occurs 5 inches from the bottom end of stroke, and compression $4\frac{3}{8}$ inches from the top end of stroke.

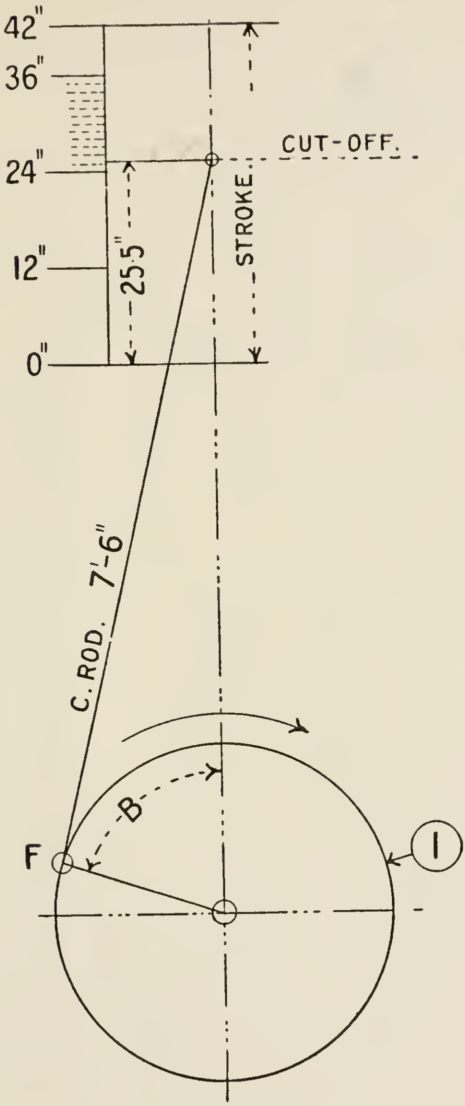
(*B.*) Referring to the data already given we find that the top steam lap and lead $= 2\frac{3}{16} + \frac{1}{8} = 2\frac{5}{16}$ inches, so that if the bottom lead is to be $\frac{1}{4}$ inch, then $2\frac{5}{16} - \frac{1}{4} = 2\frac{1}{16}$ inches steam lap at bottom, as the sum of the two must be the same.



No. 51.

Application (Sketch No. 54).

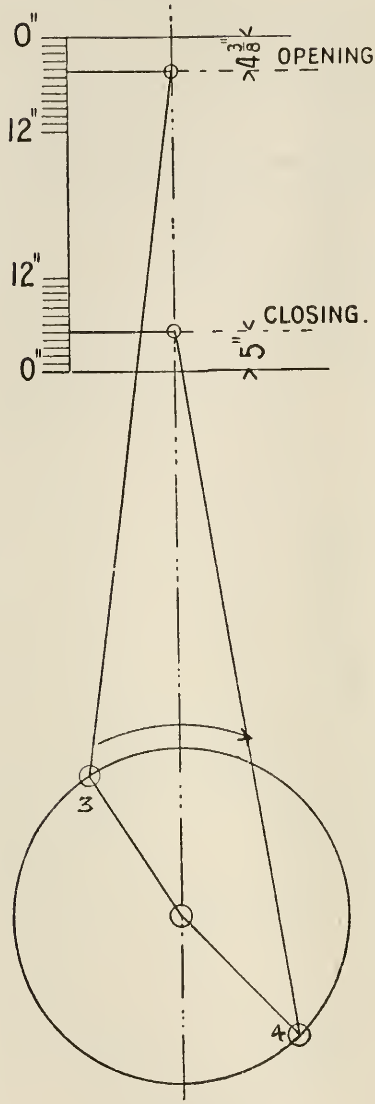
1. As before, set off the valve travel circle $7\frac{1}{2}$ inches diameter either full size or half size, and set off the lead arc at the bottom centre with $\frac{1}{4}$ inch radius in the compasses. Now take in a radius of $2\frac{1}{16}$ inches representing the steam lap at bottom, and describe an



No. 52.

arc from the centre of the circle as shown ; next draw a line tangential to both lead and lap arcs, and where this cuts the valve travel circle in points 1 and 2 it gives the crank angles at "lead" and "cut-off."

2. Transfer the angle B of crank at "cut-off" (position 2) from the valve travel diagram back to the small diagram at F (Sketch 52), and



No. 53.

with the connecting rod length 7 feet 6 inches in the compasses, and the needle point on position F, mark the centre line for the crosshead centre as shown ; finally measure by the scale of the small diagram how many inches are included between the bottom end of stroke and the crosshead, which will give the up stroke cut-off, in this case 25.5 inches. The positions of "Release" and "Compression" can be

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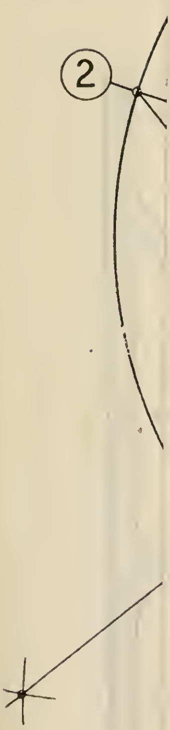

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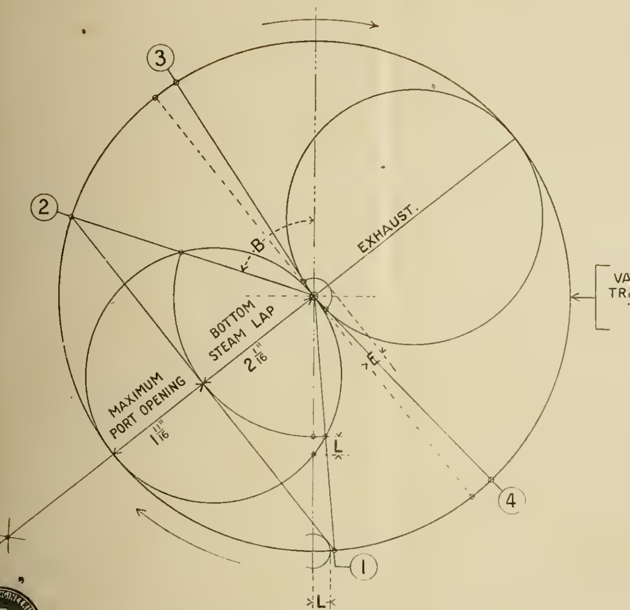


No. 54.—Valve Diagram for Bottom.

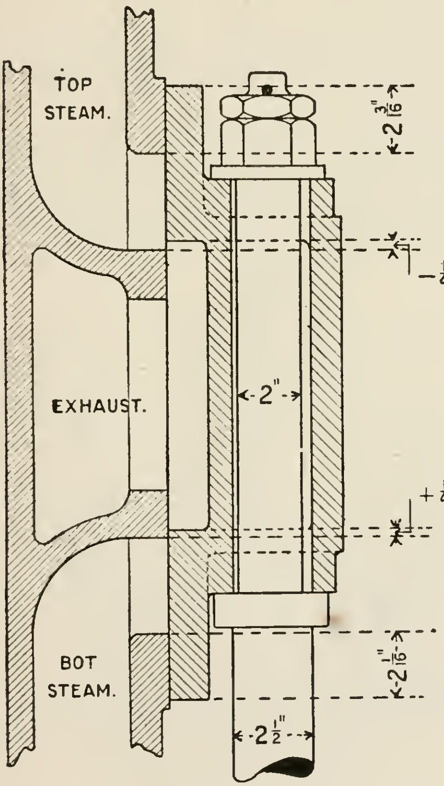
L = Lead ⅛ inch. E = + ⅛ inch Exhaust Lap.

1, Crank Angle at Lead.
2, " " Cut-off.
3, Crank Angle at Release.
4, " " Compressioo

[*To face page* 254.

located by the method described for the down stroke, and when applied give a release of $4\frac{3}{8}$ inches from top end of stroke and compression of 5 inches from bottom end of stroke (Sketch 53).

3. Set off the bottom exhaust lap $\frac{1}{4}$ inch on circle described on the *right half* of the valve travel diameter and shown as E. Now draw out crank lines at the intersection of this arc on the circle mentioned, and the positions of crank angle at release 3, and compression 4 will be found. As before described for the down stroke, set off these angles back on the small diagram, measure up the connecting rod lengths, and the positions of "Release" and "Compression" from the end of stroke can be measured as shown (Sketch 53).



No. 55.—Slide Valve.

For Valve Diagram. Showing the required steam lap and exhaust lap top and bottom as determined by the valve diagram.

NOTE.—Lining up a slide valve decreases the bottom steam lap and increases the top steam lap; but, if a piston valve, the reverse effects are obtained, the bottom steam lap being increased and the top decreased. The total steam lap remains constant in both cases.

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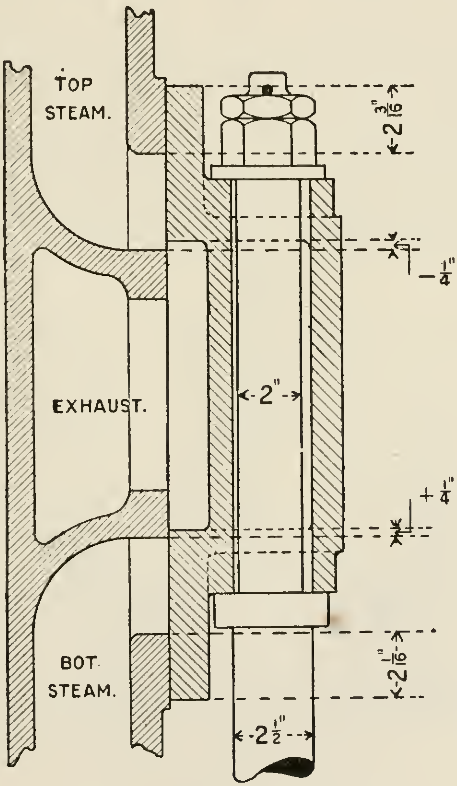
located by the method described for the down stroke, and when applied give a release of $4\frac{3}{8}$ inches from top end of stroke and compression of 5 inches from bottom end of stroke (Sketch 53).

3. Set off the bottom exhaust lap $\frac{1}{4}$ inch on circle described on the *right half* of the valve travel diameter and shown as E. Now draw out crank lines at the intersection of this arc on the circle mentioned, and the positions of crank angle at release 3, and compression 4 will be found. As before described for the down stroke, set off these angles back on the small diagram, measure up the connecting rod lengths, and the positions of "Release" and "Compression" from the end of stroke can be measured as shown (Sketch 53).



No. 55.—Slide Valve.

For Valve Diagram. Showing the required steam lap and exhaust lap top and bottom as determined by the valve diagram.

NOTE.—Lining up a slide valve decreases the bottom steam lap and increases the top steam lap; but, if a piston valve, the reverse effects are obtained, the bottom steam lap being increased and the top decreased. The total steam lap remains constant in both cases.

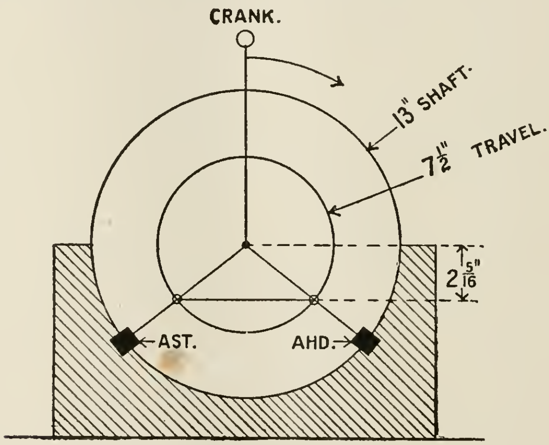
As before stated, release occurs $4\frac{3}{8}$ inches from end of stroke, and compression 5 inches from end of stroke.

General.—The general valve data now work out as tabulated below:—

Valve Travel, $7\frac{1}{2}$ In.	Lead.	Steam Lap.	Port Opening.	Cut-off.	Release.	Compression.
	Inch.	Inches.	Inches.	Inches.	Inches.	Inches.
Top -	$\frac{1}{8}$	$2\frac{3}{16}$	$1\frac{9}{16}$	29·4	5	$4\frac{3}{8}$
Bottom -	$\frac{1}{4}$	$2\frac{1}{16}$	$1\frac{11}{16}$	25·5	$4\frac{3}{8}$	5

NOTE.—Half travel $3\frac{3}{4}$ inches – Bottom Steam lap $2\frac{1}{16}$ inches = $1\frac{11}{16}$ inches Bottom Port opening.

Observe that even with *less* steam lap on the bottom end of valve than on the top the bottom or up stroke cut-off is earlier than the down stroke. This is due to the angle formed by the connecting rod and crank, and occurs in most cases in practice.



No. 56.—Keyseat Template.
For Valve Diagram.

"Open" and "Crossed" Eccentric Rods.

By this is understood the position of the eccentric rods *when the crank is on the bottom centre*, as in running the rods open and cross each other alternately all the time. For slide valves or *outside steam* piston valves the rods are usually arranged as "open," but with inside steam piston valves the rods are fitted "crossed" when the crank is on the bottom centre. This is to obtain the full benefit of link expansion, as if the rods were arranged the reverse way the lead would be diminished when linked up, and the range of expansion more limited, as shown on diagram No. 58. To obtain similar effects with piston valve gear of the inside steam type, as the motion of valve is reversed, the fitting of the rods must be also reversed, so that crossed rods take the place of open rods for the latter type of valve. The general effects may be summarised as follows (see also page 211).

Effects of Linking-up (Slide Valves).

Arrangement of Eccentric Rods.	Valve Travel.	Lead.	Cut-off.	Release.	Compression.
"Open" Rods (crank on bottom).	Reduced.	Increased.	Earlier.	Earlier.	Earlier.
"Crossed" Rods (crank on bottom).	Reduced.	Decreased.	Earlier.	Earlier.	Earlier.

NOTE.—The amount of steam lap and exhaust lap of the valve remains unchanged, but the port opening is less, in exact proportion to the reduced valve travel.

The special disadvantages of linking up are:—

1. Excess wire drawing of steam, due to reduced port opening.

2. Rapid increase of compression, which reduces effective area of indicator diagram. As before stated, it should be noted that to obtain the above effects with inside steam valves the rods require to be *crossed* instead of open, with crank centred on bottom: this is due to the fact that with slide valves and open rods, when the link is shut in the valve is slightly lowered by the angle described by the rod in moving over if crank is on the top centre, and slightly raised if crank is on the bottom centre, thus increasing the lead; but with inside steam valves the lead would be diminished in like proportion; therefore, to obtain equal effects, the rods must be arranged as crossed with crank on bottom, when the valve is of the inside steam type.

Taking, then, a three-cylinder triple-expansion engine of the usual type, the eccentric rod and crank positions are therefore arranged as follows:—

Cranks and Eccentric Rods.

H.P. inside steam piston valve,	- Eccentric Rods crossed (crank on bottom).
I.P. common slide valve, -	- Eccentric Rods open (crank on bottom).
L.P. double ported slide valve,	- Eccentric Rods open (crank on bottom).

This arrangement allows of equal linking-up effects in all three engines.

Diagram for Gear Linked up.

To construct an approximate valve diagram showing the various effects produced by shutting in the gear, proceed as follows :—

Method.

1. Measure the length of the eccentric rods from centre of pin on quadrant to *centre of pulley*. Measure the distance between the eccentric pulley centres and between the eccentric rod pins on link bar (usually equal to three times valve travel).

2. Find a radius for the equivalent valve travel circle arc, by the method devised by the late Mr Macfarlane Gray, and which reads as follows :—

$$\text{Arc radius} = \frac{\text{Eccentric rod length} \times \text{distance between pulleys}}{2 \times \text{distance between quadrant pins}}.$$

Assuming, then, that for the case already described, the eccentric rod length centre to centre is 96 inches, the distance between the quadrant bar pins $22\frac{1}{2}$ inches, the distance between the pulley centres can be measured on the previous "full gear" diagram, or can be measured direct on the new "linked up" diagram.

In sketch No. 58 it will be seen that the previous top diagram is shown complete, but in dotted lines. On this the linked up diagram can be filled in, and the points of difference, in "Lead," "Cut-off," "Release," and "Compression" can then be easily compared.

3. First draw a horizontal line across from E to C, and this measured will be the required distance between the pulleys, which in this case is $5\frac{7}{8}$ inches.

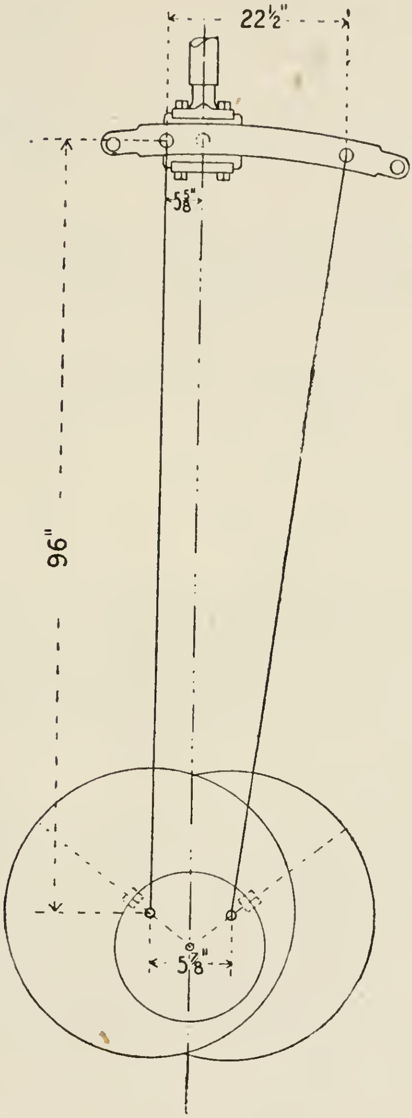
Now apply the rule given to find the radius of the arc shown connecting C and E.

Thus, $$\text{Radius} = \frac{96 \text{ in.} \times 5{\cdot}875 \text{ in.}}{2 \times 22{\cdot}5 \text{ in.}} = 12{\cdot}5 \text{ inches.}$$

With a radius of $12\frac{1}{2}$ inches in the compasses describe the arc as shown.

4. Now suppose that the link expansion gear is shut in so that the cut-off is ·40 (40 per cent.) of the stroke, and that in this grade the quadrant block occupies the position shown, that is, $5\frac{5}{8}$ inches from the pin on quadrant; or exactly one-fourth of the total distance, as $22{\cdot}5 \div 5{\cdot}625 = 4$.

Now mark on point D on the diagram arc in the same relative position, that is, DE = one-fourth of CE, and from the point D draw in a centre line on which construct the "steam" circle shown in full lines, then continue this line to the other side, and describe the "exhaust" circle. The sum of the two circle diameters is equal to the linked-up valve travel which in the present case measures 6 inches. To complete the other points notice where the steam lap arc, and small minus



No. 57.—Quadrant and Eccentrics.

exhaust lap arc, intersect the circles described, and project radial lines out to the valve travel circle for the crank angles to correspond.

If this is done carefully it will be found that all points now occur sooner, and that the lead is increased from $\frac{1}{8}$ in. to $\frac{7}{16}$ in.

1. Crank at lead	- - -	Full gear.
1B. " "	- - -	Linked up.
2. Crank at cut-off	- - -	Full gear.
2B. " "	- - -	Linked np.
3. Crank at release	- - -	Full gear.
3B. " "	- - -	Linked up.
4. Crank at compression	- -	Full gear.
4B. " "	- - -	Linked up.

If the various crank angles are now transferred back to the small scale engine stroke diagram, as described previously, the new positions of release, compression, &c., can be determined.

In the example shown the linked-up data work out as follows :—

Valve Data (Linked up).

Valve Travel, 6 in.	Lead.	Steam Lap.	Port Opening.	Cut-off.	Exhaust Lap.	Release.	Compression.
	In.	In.	In.	In.	In.	In.	In.
Top - -	$\frac{7}{16}$	$2\frac{3}{16}$	$\frac{13}{16}$	$16\frac{7}{8}$	$-\frac{1}{4}$	$10\frac{1}{2}$	11

NOTE.—It should be carefully noted that the gear is of the open rods type, that is, with the crank on bottom centre the rods are open as shown in the Sketch No. 22. If with a slide valve the rods are arranged "crossed" the lead would be less, and the expansion range much more limited, as the arc C,E, would then require to be taken from a centre above the shaft in place of below, and this would result in the arc being convex to the shaft centre line instead of concave as at present.

To find Valve Travel. (No. 59.)

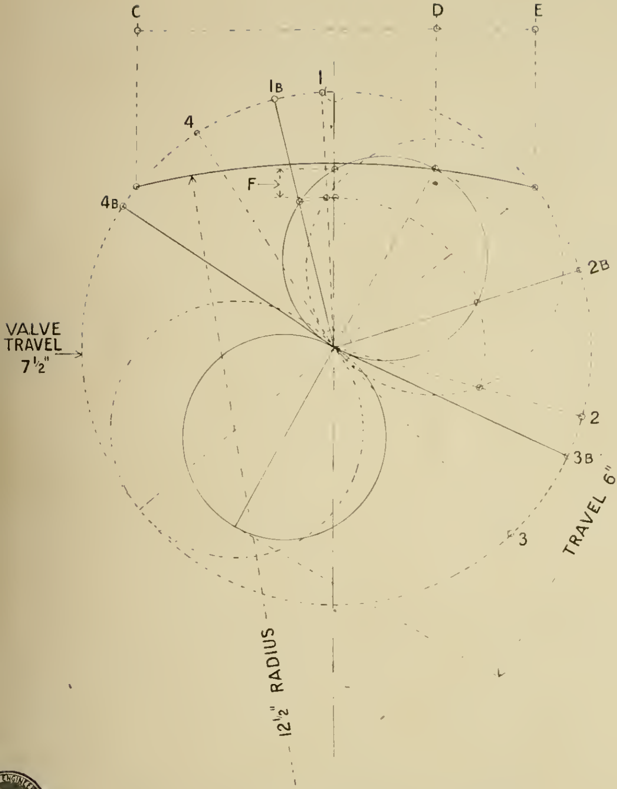
In a case where the steam port opening is decided upon first of all, the required valve travel (and therefore steam laps) may be determined by the following construction. First set off, to a suitable scale, ACB, the crank pin travel circle, OB being the angle of crank at cut-off. Join AB, and bisect it at point C, now join AC, and set off at D, DE, the port opening determined on, less half the lead ; next draw a line at E parallel to AD, and where the line so drawn cuts AC at F, draw a short line parallel to OC into the centre line at G ; then FG is equal to *half* the valve travel, so that twice FG will give full travel of valve required.

Bellis and Morcom Engine.

This type of engine is compound with the valve chest common to both cylinders placed between them, the valves of the piston type being arranged tandem fashion, and on the same valve spindle.

Action.—The engine cranks are opposite each other, or are at an angle of 180°. The H.P. piston valve receives steam in the centre

No. 60.—Bellis and Morcom High Speed Compound Engine.



No. 58.—Valve Diagram showing Effects of Linking Up.

1, Crank angle at "lead," full gear.
1B. " " " linked up.
2, Crank angle at "cut-off," full gear.
2B, " " " linked up.
3, Crank angle at "release," full gear.
3B, " " " linked up.
4, Crank angle at "compression," full gear.
4B, " " " linked up.

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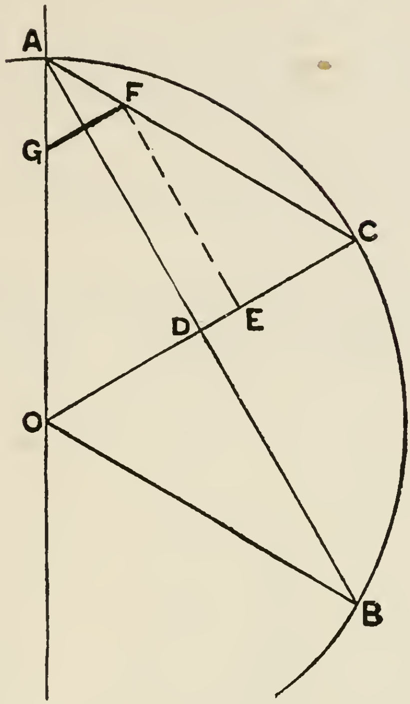
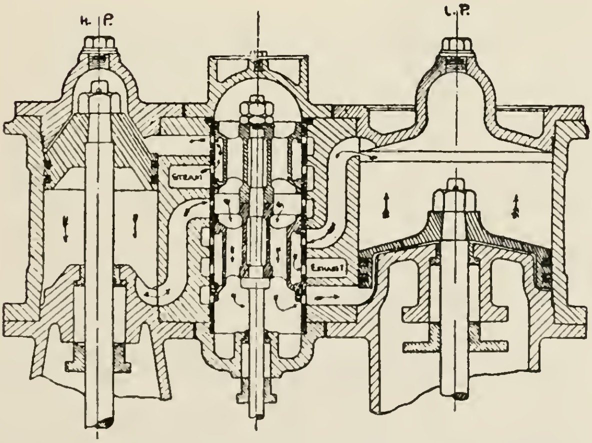


No. 59.—Construction to find Valve Travel.



No. 60.—Bellis and Morcom High Speed Compound Engine.

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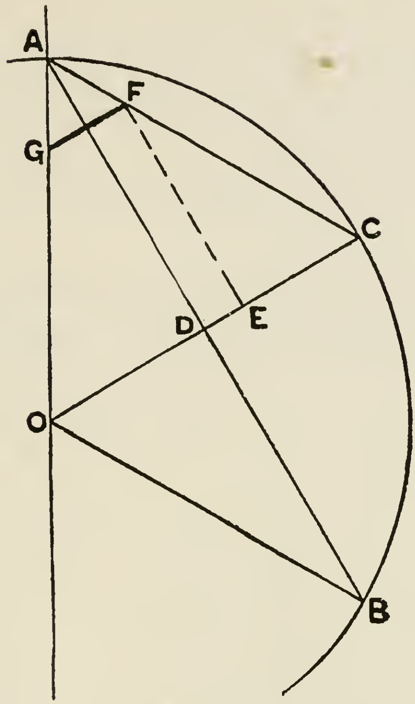
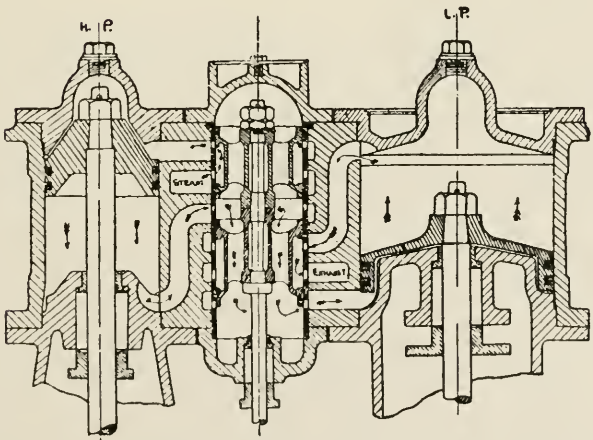
Action.—The engine cranks are opposite each other, or are at an angle of 180°. The H.P. piston valve receives steam in the centre



No. 59.—Construction to find Valve Travel.



No. 60.—Bellis and Morcom High Speed Compound Engine.

or inside edges, and the L.P. valve receives steam at the ends or outside edges, so that the exhaust of the H.P. valve being over the ends serves as the admission steam for the L.P. valve, the exhaust of the L.P. valve opens up to the exhaust casing and exhaust pipe to the condenser. Notice that the valves are hollow cast to allow of steam flow from end to end.

The sketch and following data of valve setting for this type of engine are taken from the *Mechanical World* of September 1910.

Data for Bellis and Morcom Engine.

Cylinders, 10 inches diameter and 17 inches diameter; stroke, 9 inches; piston valves, $6\frac{1}{2}$ inches diameter; revolutions, 400.

Valve Travel, $2\frac{3}{4}$ Inches.	H.P. Cylinder.		L.P. Cylinder.	
	Top.	Bottom.	Top.	Bottom.
	Inch.	Inch.	Inch.	Inch.
Steam Lap - - -	$\frac{27}{32}$	$\frac{13}{16}$	$\frac{3}{4}$	$\frac{11}{16}$
Lead - - - -	$\frac{1}{32}$	$\frac{1}{16}$	$\frac{1}{8}$	$\frac{3}{16}$
Cut-off (Mean) - - -	·64		·66	

NOTE.—Observe that the sum of the top or bottom steam lap and lead is the same, or, in other words, what is gained in lead at the bottom end is lost in steam lap, the sum of the two remaining the same.

Referring to table—

H.P. { Top, $\frac{27}{32} + \frac{1}{32} = \frac{28}{32} = \frac{14}{16} = \frac{7}{8}$ inch.
Bottom, $\frac{13}{16} + \frac{1}{16} = \frac{14}{16} = \frac{7}{8}$ "

L.P. { Top, $\frac{3}{4} + \frac{1}{8} = \frac{7}{8}$ "
Bottom, $\frac{11}{16} + \frac{3}{16} = \frac{14}{16} = \frac{7}{8}$ "

As the same pulley drives *both valves*, the sum of the steam lap and lead requires to be the same for each cylinder, as shown above.

Port Opening.

Half travel = Steam Lap + Port Opening.

Therefore, $2\frac{3}{4}$ inches $\div 2 = 1\frac{3}{8}$ inches = Half travel.

H.P. { And, $1\frac{3}{8}$ inches $- \frac{27}{32}$ inch $= \frac{17}{32}$ inch Port opening at top.
Also, $1\frac{3}{8}$ " $- \frac{13}{16}$ " $= \frac{9}{16}$ " " bottom.

L.P. { Again, $1\frac{3}{8}$ " $- \frac{3}{4}$ " $= \frac{5}{8}$ " " top.
And, $1\frac{3}{8}$ " $- \frac{11}{16}$ " $= \frac{11}{16}$ " " bottom.

Effects of Link Adjustments on I.H.P.

No.	Link Alteration.	Effect on I.H.P.
1	H.P. shut in - -	H.P. power practically unaltered. I.P. and L.P. power decreased. Total power reduced.
2	H.P. opened out -	H.P power practically unaltered. I.P. and L.P. power increased. Total power increased.
3	I.P. shut in - -	H.P. power decreased. I.P. power increased. L.P. power unaltered. Total power unaltered.
4	I.P. opened out -	H.P. power increased. I.P. power decreased. L.P. power unaltered. Total power unaltered.
5	L.P. shut in - -	H.P. power unaltered. I.P. power decreased. L.P. power increased. Total power unaltered.
6	L.P. opened out -	H.P. power unaltered. I.P. power increased. L.P. power decreased. Total power unaltered.

From the foregoing it will be evident that the back pressure of one engine varies with the cut-off in the next engine, so that if, say, the I.P. link is shut in the H.P. back pressure will rise, but if the I.P. link is opened out the H.P. back pressure will fall; the same holding good for the L.P. cut-off in relation to the I.P. back pressure.

The back pressure on a piston depends then on the following:— 1. On the point of cut-off, which gives a proportional terminal pressure and back pressure for a receiver of given capacity. 2. On the cut-off of the next engine, which, if early, increases the back pressure of the preceding engine piston, and which, if late, lowers that back pressure as before described.

As quite a number of readers do not appear to understand how in link alterations Nos. 1 and 2 the H.P. power remains practically unaltered, the following explanation is given by the writer (who is often referred to on this point), with the hope of clearing up the matter.

When the H.P. is shut in the cut off is earlier on the *same initial pressure* line, giving a lower expansion curve pressure and a lower back pressure line, the one practically making up for the other, so that the actual area of the diagram is almost unaltered.

In the same way, when the H.P. link is opened out the cut-off is later on the same initial pressure line, giving a higher expansion curve pressure, and correspondingly a higher back pressure line, the one practically balancing the other, so that, as before, the card area is only very slightly affected. Any difference in H.P. power is principally due to *difference in revs.*, as with H.P. shut in the revs. are less, and with H.P. opened out the revs. are more.

To further assist in making this point clear a set of cards are given on page 263*b* (taken from the author's "Marine Engine Indicator Cards") showing the actual effects produced when the H.P. link is shut in.

To any readers who may have the chance of experimenting, the writer would advise an actual test with link adjustments, when the results will be found to be as here stated.

Effects of Various Link Adjustments.

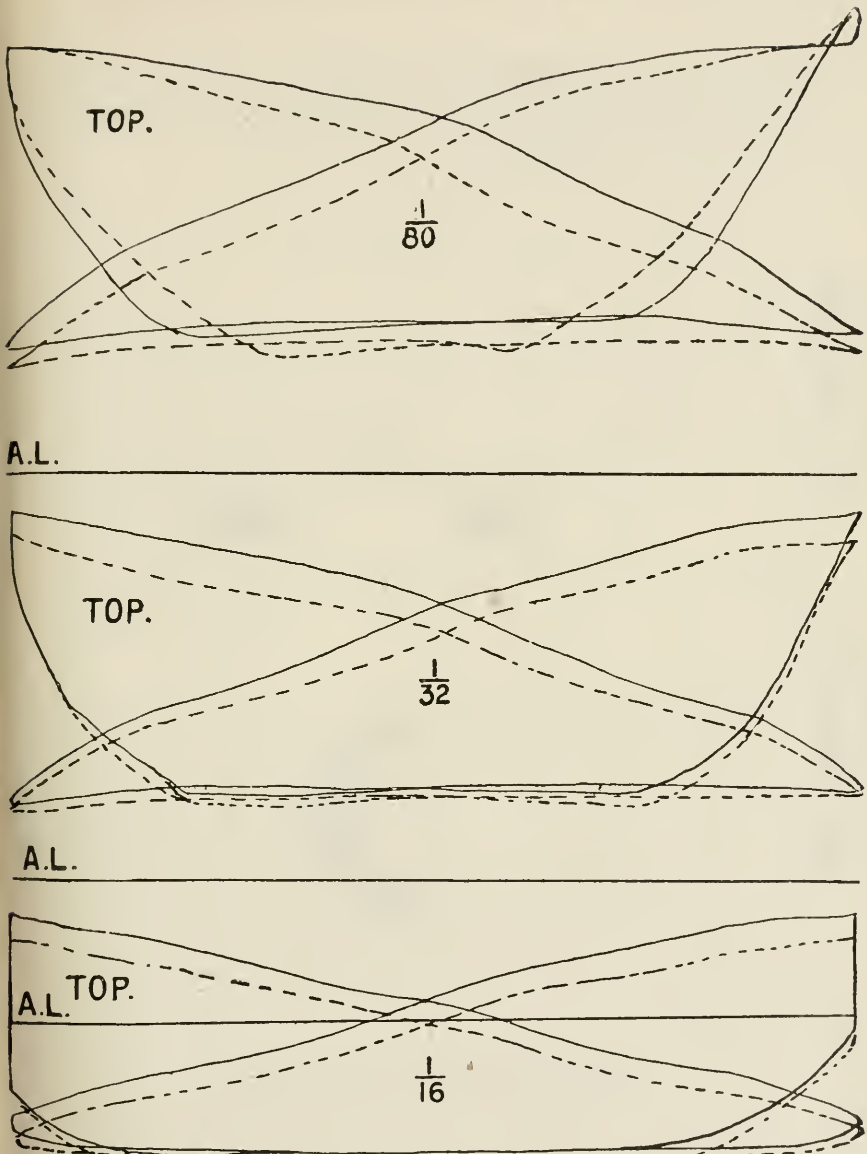
	Link Adjustment.	Effect on I.H.P. Distribution in Each Cylinder.			Effect on Total I.H.P.
		H.P.	I.P.	L.P.	
1.	I.P. shut in and L.P. opened out.	Power reduced	Power increased	Power reduced	Unaltered
2.	I.P. opened out and L.P. shut in	Power increased	Power reduced	Power increased	Unaltered
3.	I.P. and L.P. both shut in	Power reduced	Power about the same as before	Power increased	Unaltered
4.	I.P. and L.P. both opened out	Power increased	Power about the same as before	Power reduced	Unaltered
5.	H.P. link and L.P. link shut in	Power only slightly reduced	Power much reduced	Power slightly reduced	Reduced
6.	All engines linked up by means of reversing wheel	Power reduced	Power reduced	Power reduced	Reduced

Note.—When the reversing wheel is employed to link up, the I.H.P. of *each cylinder* is reduced by about the same amount.

Note.—If the links cannot be altered, similar effects can be obtained by valve and eccentric adjustments as follows:—

In place of shutting in the links, piece valve (increase steam lap) top and bottom, and advance pulley.

In place of opening out the links, chip valve (reduce steam lap) top and bottom, and put pulley back an equal amount.



No. 60a.—Set of Cards.

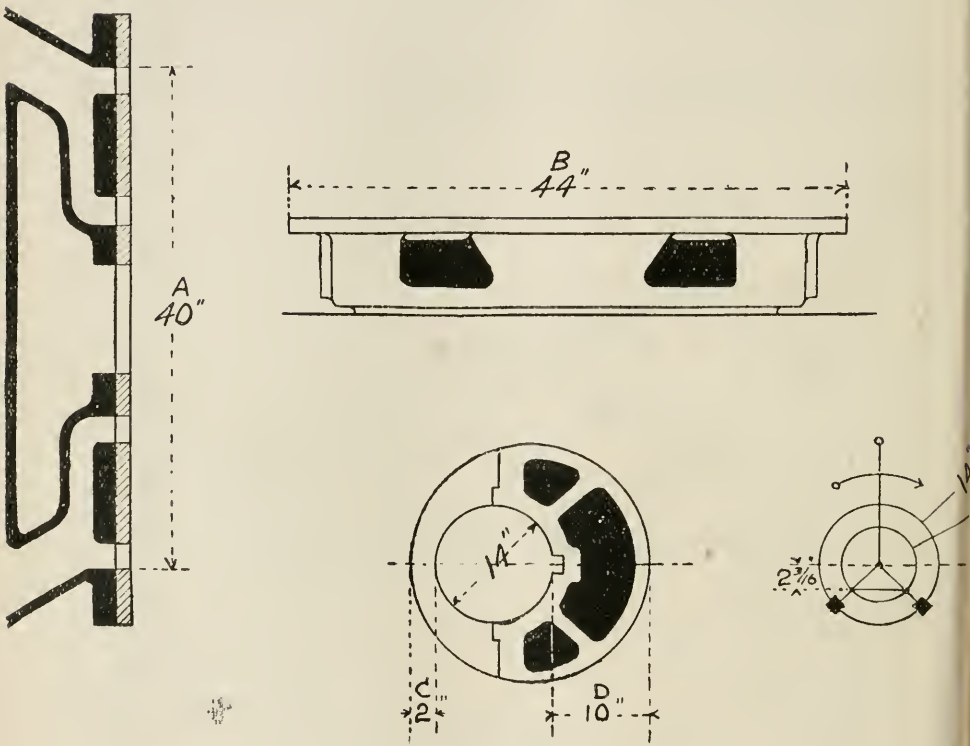
Dotted lines show effects produced by—

A, Shutting in H.P. link; or

B, Increasing steam lap of H.P. Valve and advancing the eccentric.

Effects on Horse Power:
- H.P. power only slightly less.
- I.P. power reduced.
- L.P. power reduced.
- Total power reduced.

To Mark off Eccentric Keyseat Positions on Shaft, the Valve having been Removed from the Casing and no Valve Data Available for Reference.



No. 60b.—Valve and Eccentric.

The sketches show the cylinder face, say, 40″ in depth from upper edge of top steam port to lower edge of bottom steam port; the valve face, say, 44″ in depth; and the eccentric 10″ in width on one side, and 2″ on the other side.

Proceed as follows :—

Mean steam lap $= \dfrac{44'' - 40''}{2} = 2''$ mean steam lap for both top and bottom.

Assume lead, say, $\frac{1}{8}''$ top, $\frac{1}{4}''$ bottom.

Then, Mean lead $= \dfrac{\frac{1}{8}'' + \frac{1}{4}''}{2} = \frac{3}{16}''$

So that mean steam lap and lead $= 2'' + \frac{3}{16}'' = 2\frac{3}{16}''$.

Valve Travel $= 10'' - 2'' = 8''$.

Next proceed to set out keyseat diagram (shown on right) by drawing the outer circle of shaft $14''$ diameter, also inner circle of valve travel $8''$ diameter, then measure down from centre the mean steam lap and lead $= 2\frac{3}{16}''$, project out the lines, and mark off keyseat positions in the usual manner. After bolting on the pulleys the valve will have the *same* amount of lap and lead at both ends, and to correct this, line up the rod for *half the lead difference*, that is, $\dfrac{\frac{1}{4}'' - \frac{1}{8}''}{2} = \frac{1}{16}$ liner under rod. The setting will now become top lead $\frac{1}{8}''$, bottom lead $\frac{1}{4}''$, top steam lap $2\frac{1}{16}''$, bottom steam lap $1\frac{15}{16}''$. The valve will then be correctly set.

To Determine the Required Difference in a Slide Valve Steam Lap to obtain a Given Alteration in Cut-Off.

In place of the valve diagram method as described on page 249, the following practical method may be applied :—

1. With mark on guide and shoe at top centre position, measure down on guide the distance the steam has to be carried on the stroke before cut-off, then set the shoe to this mark by turning gear.

2. Remove the valve casing cover and measure the amount the top edge of valve shows either below the port top edge or above it.

3. If valve top edge is short of port top edge, pin on a brass strip (increase the steam lap top and bottom) equal to *half* of this amount, and put the pulley forward the same amount as pinned on.

4. If, however, the valve top edge is over the port top edge, then chip off (reduce the steam lap) for *half* of this amount top and bottom, and put the pulley back the same amount as chipped off.

Observe that when the steam lap is increased the pulley requires to be *advanced* to keep the lead constant ; also, when the steam lap is reduced the pulley requires to be *put back* to keep the lead constant.

Note.—For an inside steam piston valve the edges to be altered would be reversed inside for outside, &c.

SECTION V.

GENERAL NOTES AND DESCRIPTIONS.

THE Author is indebted to the Editor of the *Scottish Bankers' Magazine* for permission to reproduce the following article on the Manufacture of Metals from the pages of that journal.

Manufacture of Iron and Steel.

Sources.—Native iron, as it is called, possessing similar properties to that extracted from ores, has been found in Greenland and elsewhere in small quantities, but for practical purposes it is from the ores we derive our iron supply. These are widely distributed throughout the earth, and vary considerably in their characteristics and purity. The chief kinds in use are (1) the *Magnetic* (loadstone or black oxide), which is the richest of all, containing as much as up to over 70 per cent. of iron. A high class iron is made from this ore in conjunction with charcoal in Sweden. (2) *Red Hæmatite*, which contains up to 60 per cent. of iron. This ore is plentiful in the district of West Cumberland and North Lancashire, and also in the north of Spain. (3) *Brown Hæmatite*, which is similar to the red, and from which the bulk of the French and German iron is made. And (4) the *carbonates* of iron, called spathic when comparatively pure, and also blackband and clayband ironstone. This ore contains from 37 to 48 per cent. of iron. It has the advantage of being found usually along with coal measures; and it has been the staple ore of Scotland. Great deposits also exist in the Cleveland district, but not of equal quality.

The Blast Furnace.—The reducing or extraction of the ore is effected by smelting in a blast furnace. In the case of the poorer ores where more impurities are present, calcining or roasting may be a preliminary operation. The effect of this is to get rid of carbonic acid, water, and such other undesirable ingredients which are volatile, and so render the material more suitable for treatment in the blast furnace.

Blast furnaces, fairly familiar objects, are large, circular, tower-like erections. The interior, which is not straight in form but contracts towards top and bottom, is lined with refractory fire-brick and ganister (a very refractory siliceous rock); around this is an annular space or ring filled with loose material to allow of expansion, and the outer wall of masonry is enclosed in iron sheathing strongly bound together. The furnaces range from 40 feet or so to 100 feet, and even more, in height, with internal capacity of 500 to 25,000 cubic feet or over. The modern furnaces are the highest, but it has been found that practical difficulties in working counterbalance the advantages of greater height when carried beyond a certain point. One advantage of the higher furnace is to render previous calcining of the ore less necessary, the same effect being accomplished in the upper part of the furnace. At the top of the furnace is a gallery or platform from whence the charge is admitted. The mouth of the furnace is closed by means of a large cone, which can be lowered by a chain when charge is being admitted, and then closed again. The closed top is a modern advance. Formerly the mouth was open and the great, lurid flames belching out made the blast furnace a picturesque feature of the district where it was erected. Many will remember the time when "Dixon's blazes," as they were familiarly called, formed a landmark in Glasgow; and when shipmasters on the Ayrshire coast could shape their course by the glare of the Ardeer furnaces. But the old order changes, and the picturesque has to give way to the practical.

The closed tops came into being when the gases generated in the blast furnace were utilised with resulting efficiency and economy. By-products have also become a feature of present-day practice. With the utilising of the gases and slag, recovery of ammonia, and so on, a combination of iron and chemical work is now common.

The Charge.—The charge consists of fuel, ore, and flux. The first is commonly coke, but may also be coal or a combination of both; and charcoal is mostly used in Sweden, where coal does not abound. The ores, as has been said, vary considerably. The flux is commonly limestone, although other agents are also used. It is introduced in consequence of the impurities remaining in the ore. It combines with the silica and other prejudicial matter, and forms a slag or cinder separated from the iron. A strong blast of air is introduced through piping surrounded by water tuyeres (outer casings). Powerful blowing engines force the blast into furnace and through the charge therein. The water circulating through the tuyeres serves to cool the inlet where the heat becomes intense, and might cause trouble by fusing the parts. At one time a cold blast was used, and another of the most important modern improvements was the invention and use of the hot blast by Neilson of Dundyvan. By heating the air before its admission into the furnace great efficiency and economy of working were attained. Various forms of heating stoves have been devised, and these are now usually fired by the escaping blast furnace gases.

Pig-Iron.

The proportions of fuel and flux are so far determined by the nature of the ore used, and there are modifications of appliances and methods according to varying circumstances and requirements. The close tops and greater height of the furnaces have led to thinner walls and iron casings. As the charge becomes affected by the intense heat, chemical and other changes take place, impurities being taken up by the flux, though some others partially remain, as sulphur, phosphorus, and carbon. After these changes fusion speedily ensues, and the molten iron falls to the bottom, the slag floating on the top, while other waste elements escape in the form of gases. Once started, a blast furnace may be kept in operation for years. At the bottom of the furnace is an aperture called a tapping-hole, kept closed until the melting of the iron is completed. A large bed of sand is formed in front of the furnaces in which channels are made with smaller furrows branching off from them. These are called sows and pigs respectively, whence the term *pig-iron*. The tap-hole being opened, the melted iron runs out like a stream of liquid fire, flows down the large furrows into the smaller ones, where, on cooling, it assumes the familiar form of the oblong bars called pig-iron. The cinder or slag is drawn off over a dam at a higher level, independent of the iron.

The pig-iron thus produced is of different grades, the fracture disclosing divergencies of character and quality. The colour varies, ranging from grey to white, and the degree of hardness varies correspondingly. Grey is the softest, running through the mottled to the whitest, which is also the hardest. The quality is distinguished by numbers, beginning with No. 1, and these numbers also indicate the suitability of the iron for further uses, all not being alike adapted for the same purposes. The grey iron is most suitable for foundry purposes, the white for forging.

Castings.—We now arrive at the parting of the ways, so to speak The pig (or cast) iron may be applied, broadly speaking, in two ways. It may be employed for the producing of castings, or it may be converted into malleable iron or steel. Although large and rough castings might be made direct from the blast furnace, and malleable iron direct from the ore, these methods have been discarded, as it is more advantageous to make both indirectly from the pig-iron.

The grey or softer pig-iron, as has been said, is most suitable for castings. But as there is great variety in the size, shape, intricacy, and purpose of castings, ranging from, say, a boot protector to a large steam-engine cylinder, mixture and manipulation of the different brands of iron are often required. For example, the bottom of a pan mill subject to constant grinding and heavy pressure, naturally requires to be of very hard metal; whereas, an ornamental article showing design will require fluid and easily running metal. Again, castings which have to be machined—that is, turned, planed, drilled,

&c., cannot be economically treated if too hard, and yet generally require toughness and strength.

The Foundry.—The castings are produced in the foundry, to which the pig-iron is conveyed; and there it first undergoes a process somewhat analogous to that already described. The iron, along with the fuel and flux, is remelted in a furnace called a cupola, and drawn off at a tap-hole, the slag being afterwards thrown down through an aperture at bottom. Frequently scrap iron (old castings broken up) is used along with the pig-iron. Obviously the additional refining gives purer and better metal, but all is regulated very much by the purpose for which the castings are required. The metal runs from the tap-hole into a large iron ladle lined internally with fire-clay, and for pouring the metal into the lesser moulds small hand ladles are used.

The castings may be made in sand or loam moulds. Patterns or models of the articles required have, mostly, to be made in the first instance, and may be of stucco, wood, or iron, but generally the last when many articles are required, as it stands tear and wear better, and lasts longer. The sand is enclosed in an iron frame or box, made in halves and hinged; and the patterns of the article required, which may be simple or complicated and in one or more pieces, are embedded in the sand so as to form an exact mould. The patterns are then withdrawn and the box closed, an aperture being left for pouring in the metal, and small holes pierced to allow the escape of air and gases. When sufficiently cool the castings are removed from the boxes and dressed—that is, cleaned and filed.

Loam is a mixture, such as sand, clay, and horse manure, faced with blacking or coal dust, and built in a pit or brick-work. Appliances called loam boards are used in forming the moulds, and the skill of the moulder is called into requisition in building these together. For water and gas pipes, iron moulds coated with plumbago are now sometimes used. More expensive to begin with, they save afterwards, seeing they are not destroyed at each casting like sand or loam moulds.

What are called chilled (hardened) castings are made in metal or mixed metal and sand moulds, in which the melted iron cools rapidly and becomes extremely hard. This does for shot, &c. Cast iron is distinguished by its granular formation, which can be seen on fracture, its brittleness, and its hardness. It cannot be welded or riveted, and is not pliable. So-called malleable castings are of the opposite kind from the chilled, and are soft and to a certain extent pliable. For this process the articles are placed in powdered hæmatite ore or similar preparation, packed in chests, heated in a furnace for several days, and allowed slowly to cool until annealed.

Malleable Iron.

For the manufacture of malleable or wrought iron a different process is employed. Malleable iron, as the name implies, is ductile and fibrous. It is softer than cast iron and can be bent, twisted, welded, and riveted. The difference between it and cast iron is principally due to the larger proportion of carbon in the latter. It has been described as free from carbon, but this is hardly correct, as usually it contains a very small percentage of that element.

Puddling Furnace.—The first step is to treat the pig-iron in a puddling furnace. This is of the reverberatory type—that is, one where the fuel and the iron do not come into contact as in the open and closed hearth furnaces. As usual, the inner lining of the furnace is of a refractory material encased in strong iron outer sheathing. The bed or hearth is divided into two parts by a low wall or bridge, the fuel being placed in one and the charge of pig-iron in the other. The flame passes over the bridge against the roof, which is so shaped as to reverberate or throw it down in a fierce heat upon the iron and then pass on to the flue. The furnace may hold a charge of about 4 cwt. of metal, and is worked by two men, a fore and an under hand. When partially heated the furnace is fettled—that is, plastered with composition embodying oxide of iron in the form of hæmatite made into paste with water, or else of slag from previous meltings, which comes in cheaper. Lumps of metal are thrown in, the fire is then raised, and by means of long iron bars (changed as required) the puddlers, in turn, keep working or stirring and distributing the metal. During melting the iron is decarbonised (divested of the carbon remaining in the pig-iron) and various other impurities removed. Before complete fusion the mass becomes of a pasty consistency, and is well "rabbled" or distributed by the puddlers. When melted it seethes and bubbles, and shortly afterwards begins to thicken, the iron separating from the impurities unites in solid pieces which gradually become welded together, while the waste or slag is run off in liquid form. It will be obvious that the process of puddling is a very exhausting one to the men engaged in it, and means of accomplishing the required work by mechanical action have been devised. These need not be described, as the object is the same whether manual labour or machinery be the method employed. Besides the mechanical contrivances for puddling, experiments have been made in the opposite direction by constructing rotary furnaces actuated by machinery. These have not, however, displaced the stationary furnaces.

The mass of iron having combined, is now removed from the furnace, and for convenience of handling is divided into parts or balls, and at once taken to a steam hammer to be beaten. Thereafter the balls can again be united and hammered into shape. During these operations the waste or slag is being further pressed out. The next stage is to convey the iron to the roughing or cogging mills with rolls of large diameter through which it is passed, the openings between the rolls being gradually reduced. The iron is thus converted into slabs, while it is also being additionally cleansed. By reheating and rerolling the quality can be improved up to a certain point, but beyond that harm results from burning, &c. The final step is the passing of the wrought iron through the rolling-mills with rollers having sections of the different forms required. These mills are massive in construction, and driven by powerful steam-engines. The billets are passed backwards and forwards through the rolls—the mills being reversible—and the rollers brought closer until the desired sizes and shapes are obtained. These are principally sheets, plates, round, square, and flat bars, angles, and other sections.

The quality of the iron is denoted (after the puddled bars) as common, best, best best, and treble,—the superior quality being the result of the reheating and rolling of the previous grade. The qualities of different works, however, vary. There are some special brands—such as Lowmoor and the products of other Yorkshire works—of a high class, where the results are dependent on special methods and local advantages as regards fuel and ore.

Steel.

Steel is a material of valuable and unique properties. It may be extremely hard, or soft enough to be bent, twisted, hammered, or drawn out to the thinnest sheet or finest wire. It can be so hard as to cut any other metal or material and to scratch glass, or it may be elastic to a degree.

It has been described as a metal intermediate between cast and wrought iron as to the carbon it contains. But this, correct enough as far as it goes, is somewhat misleading, as it does not take all considerations into account. According to the amount of carbon it contains, steel is harder or softer. According, also, to the proportion of carbon it contains, the tensile breaking strain and the limit of elongation vary. The tensile breaking strain will range from 20 to as much as 80 tons per square inch, or more if the steel be wire-drawn.

Cementation Process.—The highest quality of steel is made by what is termed the cementation process. The best puddled wrought-iron bars, preferably Swedish charcoal iron, are used. The cementing furnace is of fire-brick, circular or rectangular in form, having a wide, conical top and dome-like chimney. The fireplace is in the centre at the bottom, with door at each end for firing. On each side, supported above it, are pots or chests, so arranged that the flames pass underneath and all around, rising against the arched top, which, as it becomes heated, radiates down on the pots. These are perhaps 15 feet long or more, and 3 feet deep or thereby, A manhole at each end permits of charging the pots or converters. but these are closed during the working. A layer of charcoal is first deposited on the bottom of the pots, then a layer of bars with spaces between also filled with charcoal, and so on in alternate layers.

The charcoal is partly fresh and partly previously used. Then a plaster of cement (used charcoal and ground waste) is applied to the top, and the whole is sealed with clay to exclude air. The charge will range from 12 to 18 tons, according to size of furnace, and may even reach 30 tons. One or two of the bars are longer than the others and project outside for testing purposes. The furnace is heated for eight or ten days, when a testing-bar is withdrawn and examined to see if the material has been sufficiently carbonised. Afterwards the furnace is left for a day or two to cool down gradually. It will be found that the carbon is not uniformly distributed through the bars, but is greatest at the surface, decreasing as the centre is reached. The skin is raised in blisters (blister steel). If not intended to go through the further crucible process, the bars are then sheared into short lengths, bound together in bundles, powdered with mixture of clay or sand, heated to welding-point, and then subjected, while in a plastic state, to rapid action of a tilting-hammer or rolling-mill till welded. This gives what is called *single shear steel.* By repeating the process double shear steel is obtained, and so on. This shear steel is of cheaper description than the crucible steel, and serves for shear blades, certain kinds of knives, &c.

Crucibles.—For a finer steel under same process, to attain uniformity, the bars after being withdrawn from the furnace are remelted and mixed. This is done in crucibles. What is called the melting-house varies somewhat in arrangement, as light or heavy ingots are to be produced. The melting furnaces form a series of rectangular chambers, separated by thin brick partitions, the tops being level with floor of house, while the fireplace and ashpit are at bottom, accessible from underground passages running in front of them. Each hole or chamber is lined with ground ganister (a refractory siliceous rock), leaving an opening sufficient to hold two pots or crucibles resting on stands placed over the fire-bars. Crucibles differ in dimensions, but a common size is 18 inches high by 9 inches diameter. They are made of a mixture principally of fire-clay, with added coke dust, old ground pots, and for some purposes plumbago. They have to be very carefully made, and are dried, annealed, and fitted with lids before use. The first melting will take four or five hours for complete fusion, but subsequent meltings about half that time in consequence of the previous heat being maintained. The molten steel is poured from the crucibles into ladles with tap-holes for discharging into the ingot moulds, or for small moulds with hand ladles. The crucibles in good condition are at once replaced in the furnace for a second heat. Great care has to be exercised throughout in order to secure sound ingots. It will be obvious that the specially high class of iron used and the slow method makes this description of steel costly, but on the other hand there is nothing that can take its place.

Tempered Steel.—From the high class steel are made the various tools for hand and machine use to operate on metal, wood, and stone, also cutlery, surgical instruments, swords, springs of various kinds, saws, &c. To render it suitable for these varied uses it has to be tempered—a distinctive feature of steel. When heated and then suddenly cooled, and afterwards reheated to the given temperature required, it becomes available for the particular service to which it has to be applied. The cooling is usually effected by plunging in water. But if hardened and tempered in oil a somewhat greater toughness, more elastic limit, and tensile strength are the result. Workmen have a simple means of ascertaining the temperature by the colours which the steel assumes at the different stages of heating. These may be described as follows: pale straw, straw, yellow, brown, purple, bright blue, deep blue, according as the temperature increases.

Bessemer Process.—There were many purposes where a class of steel, not necessarily possessing the properties of the crucible product, would be of immense advantage, and this led to research and experiment. On the assumption that steel was a material containing a proportion of carbon intermediate between cast and wrought iron, the inference followed that a mixture of these would give steel, and this method was tried but without success. The fact overlooked was that cast iron contained impurities (in particular sulphur and phosphorus) in a degree sufficient to render it useless for steel making. There was also a practical difficulty in regulating the exact proportion of carbon necessary. The Bessemer process, however, accomplished the purpose, supplanting in great measure the use of malleable iron for many purposes, particularly rails and other railway material, forgings, plates, and various sections for structural work. The first step in success was by using selected pig-iron, containing very little of the objectionable elements, and the addition of spiegeleisen also exercised a purifying effect. It was found that by first eliminating all the carbon and then adding the spiegeleisen (a particular pig-iron containing a considerable and known amount of carbon) or ferro-manganese in proper proportion, the difficulties were overcome. Finally, Thomas and Gilchrist devised their method of lining the converter with a refractory basic material containing lime and magnesia. These agents having an affinity for phosphorus absorbed this impurity, and thus the bulk of the impurities being removed, ordinary pig-iron could be utilised.

The essential feature of the process is the forcing of a current of air through melted pig-iron in a special vessel named the converter. The original Bessemer method is called the acid process, because of the converter being lined with siliceous (ganister) material.

The Bessemer converter is an iron vessel, and may contain from 5 to 15 tons, lined as usual with refractory material, the bottom being perforated with holes through which a blast of air is conveyed. It is mounted on axles which allow of its being swivelled round; one of these being hollow is utilised to admit the blast. The converter after being heated is brought to a horizontal position, giving access to the mouth. Melted pig-iron is poured in, the blast turned on, and the converter raised to its vertical position. The blast is sufficiently strong to counteract the weight of the charge, so that the metal does not run down through the perforations. The air is forced through the molten metal, burning up the carbon, and a kind of fairy fountain effect is set up by the flames, spray, and colour which ensue, while, to paraphrase Tennyson, "The blast is roaring and blowing." The near exhaustion of the carbon is indicated by the waning of the flames. The converter is once more brought to the horizontal position, while the melted spiegeleisen is added in the proportion required to give the necessary amount of carbon, and then reversed. The mixing of the metals causes at first a violent agitation. Altogether about twenty minutes is occupied in converting, say, 10 tons of hæmatite pig-iron into steel. The temperature at the close of the blast is said to reach 3250° Fahr.

It will be readily understood that the intense heat generated in all these smelting operations, together with the corrosive action of the slags, necessitates the use of materials for lining the different furnaces capable of withstanding as far as possible the destructive effect of these agents—hence the frequent allusions to refractory linings. For the most part special deposits of fire-clay and of siliceous or flinty rock ground to form ganister have been found best adapted for the purpose.

Basic Process.—The distinctive feature of what is called the basic process of producing the steel—as distinguished from the original acid Bessemer process—is the substitution of a basic lining of magnesian lime for the coating of the converter instead of the acid ganister. As already noted, the basic material extracts the phosphorus from the iron, which cannot be done by the ordinary process, and so renders it possible to use Cleveland and other lower class iron in the making of steel. With some modifications the processes and appliances are, in general, similar to those already described.

The amount of slag produced in the basic process is much larger than in the other, but is of a different character, and as it contains a considerable proportion of calcium phosphates useful for fertilising land, it is employed after suitable treatment for this purpose. Apart from the capability of using the inferior iron by the basic method, the large quantity of material which can be handled at a time, the rapidity of the conversion, and the comparative absence of manual labour combine to render the Bessemer process of steel making one of the most important of the present day, and great quantities of material are turned out for the purposes already named.

Siemens-Martin Process.—Another successful method largely in use is the Siemens-Martin, by which a mild steel with a low percentage of carbon, approximating to wrought iron, is obtained. A bath of melted pig-iron of high quality is first made, and to this there is gradually added wrought iron and steel scrap in small quantities at a time. Impurities are removed during the melting, and the presence of the scrap already refined by the previous puddling gives very good results. Spiegeleisen is added to supply the desired amount of carbon, as in the Bessemer process. The furnace employed is generally a Siemens Regenerative Gas Furnace.

The roof and sides are of refractory silica or brickwork, and the bed (of considerable depth) of sand of a like nature. There are doors for introducing the charge and also for stirring and mixing. Externally it is encased in iron plates. The regenerative chambers are built underneath the furnace, with spaces between. The gas is produced in separate furnaces, of which there are several patterns. The gas and air necessary for combustion, ascending through one set of regenerators, are admitted by separate valves through portholes into the furnace hearth. The furnace having been already heated, an intense heat and volume of flames soon ensue. The hot air and gas enter at one end of furnace, and the flames and waste gas pass out through ports at the other end down to the other set of regenerators, and thence to chimney. There are means for reversing this cycle, so that a constant heat is maintained. Different kinds of gases are used, and in America natural gas is sometimes employed. As much as 50 tons of metal can be treated at one operation, or it may be only a few tons, according to the size of furnace. The proportions of pig-iron and scrap vary according to conditions and requirements; and the melting of a 10-ton charge will occupy three and a half to four hours. A small sample is ladled out to see if the desired degree of decarbonisation (removal of the carbon) is reached, then the spiegeleisen or ferro-manganese is added, and these quickly melting, the tap-hole can be opened, and the steel flows out into a large ladle with a stopper in bottom, and thence is discharged into the moulds. In order to ensure soundness in the ingots several devices are used, principal among these being the addition of some alloy at the end of the melting and the compression of the steel while in the fluid state.

Siemens Process.—In the Siemens process, besides the pig-iron and scrap used in the Siemens-Martin method, rich hæmatite ores are introduced, and somewhat different chemical reactions take place, but the working arrangements do not differ much, though longer time is required for fusion. If pig-iron of a lower class with many impurities be used, then the sand-bed of the furnace must be replaced by a basic material, as in the Bessemer converter.

There are also some important alloys used in the manufacture of steel. For instance, nickel alloyed with steel intensifies its hardness, while it can be rolled or hammered to advantage. Harveyised nickel steel is much used for armour-plates. Chromium is also used as an alloy. It imparts greater strength, toughness, and ductility. It is employed in the production of armour-plates, and also of projectiles, conferring on the latter the property of keeping intact when striking steel plates at the highest velocity. In the making of large and heavy forgings, whether in wrought iron or steel, besides the steam hammers already alluded to, very powerful hydraulic presses have been devised and come into use for shaping and compressing.

It may be said that the great development of the machinery and appliances for dealing with iron and steel in recent years, and the developments in the treatment of these materials themselves, together with the reciprocal interaction of these two factors, have made possible advances unthought of even at the beginning of the present generation. It is difficult to forecast what further progress may be in store in the future.

Strength and Composition.—The following table gives the average tensile and crushing strengths of Cast Iron, Wrought Iron, Mild Steel, Nickel Steel, and Hard Steel, also the per cent. proportion of Carbon, &c., in each.

Metal.	Tensile Strength.	Crushing Strength.	Carbon.	Manganese, Silicon, Phosphorus, Sulphur, &c.	Nickel.
	Tons per Sq. In.	Tons per Sq. In.	Per cent.	Per cent.	Per cent.
Cast Iron - -	7	45	3·5	2·8	...
Wrought Iron -	20	16	...	·3	...
Mild Steel - -	28 to 30	22	·16	·6	...
Nickel Steel -	40	...	·3	·6	3·5
Hard Tool Steel	50	100	1·1	...	...

Alloys.

Alloy.	Tensile Strength.	Copper.	Zinc.	Tin.	Antimony.
	Tons per Sq. In.	Per cent.	Per cent.	Per cent.	Per cent.
Brass - - -	12	66	33	...	...
Naval Brass - -	27	62	36	1	...
Muntz Metal - -	22	60	40	...	...
Gun-metal - -	16	91	...	9	...
					Phosphor.
Phosphor Bronze -	16	92	...	7	·5
					Antimony.
Babbit's White Metal	...	8·3	...	83·3	8·3
					Lead.
Parson's White Metal	...	1	30·5	68	·5
Spelter for Brazing -	...	50	50	...	...

Properties of Metals and Alloys.

Cast Iron - -	Can be cast.
Wrought Iron - -	,, forged, welded.
Mild Steel - -	,, forged, welded.
Nickel Steel - -	,, forged, tempered, cast.
Hard Steel - -	,, forged, tempered, cast.
Brass - - -	,, cast.
Naval Brass - -	,, cast, rolled, forged (hot).
Muntz Metal - -	,, cast, rolled, forged (hot).
Gun-metal - -	,, cast, rolled.
Phosphor Bronze -	,, cast, rolled.
Babbit's White Metal -	,, cast.
Parson's White Metal -	,, cast.

Composition of Steel.

The following extract from Greenwood's work on steel and iron gives the average composition of various steels :—

Analyses of Steel.

In 100 parts of	Soft Siemens-Martin Steel.	Siemens Steel Plates.	Crucible Steel for Forgings.	Hard Tool Steel.
Carbon - - - -	·167	·21	·36	1·144
Manganese - - -	·044	·36	·30	·104
Silicon - - - -	·023	·047	·02	·166
Sulphur - - -	·013	·052	·02	...
Phosphorus - - -	·062	·035	·03	...
Copper - - - -	·076	...	trace	...

Mild steel can be forged and welded, but cannot be tempered. Hard steel only can be tempered.

Composition of Manganese Bronze.

	Per cent.
Copper - - - - -	52
Zinc - - - - - -	46
Tin - - - - - -	1
Iron - - - - - -	about 0·5
Aluminium - - - - -	,, 0·5
Manganese - - - - -	trace

Test for Steel.

The Board of Trade test for a piece of steel plate to be used for a furnace or combustion chamber is as follows :—Strips of the metal 8 by 2 inches are heated to a cherry red, and afterwards plunged into water of 80° temperature : the strips must then be able to stand being bent over without fracture until the sides are parallel at a distance equal to three thicknesses of the plate.

Tempering Steel.

In tempering a piece of steel, as for example, a chisel, the tool is first heated to a cherry red and the point of it dipped into water; if the metal be then rubbed with a piece of stone, the various colours will appear as the heat travels along to the point. When the required tint shows, the tool must be plunged into cold water and kept there until cold, and the temper will be fixed.

The following are the colours and corresponding temperatures required in tempering different articles:—

Article.	Colour.	Temperature.
Turning tools -	Straw - -	450°
Chipping chisels	Purple - -	530°
Springs - -	Blue - - -	570°

Annealing.

In working or flanging a steel plate the smaller number of heats the plate has to undergo the better, as the effect of repeated heating of part of a plate is to strain the metal; the necessary flanging should therefore be done in one heat if at all possible. After the flanging is done the plate should be annealed, that is, heated uniformly throughout and allowed to cool down slowly, as this has the effect of taking out the strain and bringing back most of the strength lost previously by local heating.

Case-Hardening.

Case-hardening consists of putting a thin skin of steel on parts made of iron by the addition of carbon, so that the surface of the metal is hardened.

This is usually done by enclosing the iron parts to be case-hardened in an air-tight box together with substances rich in carbon, such as bones and horns of animals, or prussiate of potash, and the box is then left in a furnace for a number of hours.

The heat causes carbon to deposit on the surface of the iron, and thus changes it into steel.

After being withdrawn from the furnace the articles require to be plunged into water.

Gear which only requires to be lightly case-hardened is usually packed in finely ground bone, and heated for a few hours; if a deeper case-hardening effect is required, then the articles are packed up in the boxes with coarser bone powder, and heated for ten hours or more, but if a greater depth still of hardening is required, it is then necessary to repack and again heat up, finally dipping in the cooling water bath. For certain classes of work, wood charcoal is employed instead of bone.

Brazing.

Brazing is hard soldering, and consists of the joining together of parts made of copper or brass, such as, for example, a brass flange to a copper pipe.

The pieces to be joined are first carefully cleaned, then fitted in place and clamped together in the required position, and, after they have been covered over with spelter (composed of one part copper and one part zinc), heat is applied by means of a charcoal fire, and the spelter runs into the spaces of the joint. Borax is sprinkled over the parts as a flux to make the spelter run easily. After cooling, the spelter sets hard and the parts are then firmly soldered together.

NOTE.—Soft solder is made up of equal parts of tin and lead, resin or spirits of salts being employed as a flux.

Welding.

In welding, two pieces of metal are joined by being first heated to about 1600° Fahr. (white heat) and then hammered together.

The ends to be joined require to be scarfed or tapered away at an angle, and before putting the two surfaces together sand (if iron) or borax (if steel) is sprinkled over them as a flux, and the hammering proceeded with. It is important that the two pieces be heated to as nearly the same temperature as possible before joining.

NOTE.—The flux (sand or borax) acts to clean the surfaces of the magnetic oxide which forms on the heated surfaces, and which would otherwise prevent perfect adhesion.

Strength of Materials.

Tensile strength of	nickel steel,	34	tons	per square inch.
,, ,,	boiler steel,	28	,,	,,
,, ,,	wrought iron,	20	,,	,,
,, ,,	Muntz metal,	20	,,	,,
,, ,,	brass,	12	,,	,,
,, ,,	copper,	12	,,	,,
,, ,,	cast iron,	7	,,	,,

NOTE.—Nickel steel is mild steel with about 3·2 per cent. of nickel added.

Crushing Strengths.

Crushing strength of	hard steel,	100	tons	per square inch.
,, ,,	cast iron,	40	,,	,,
,, ,,	wrought iron,	16	,,	,,

Alloys.

An alloy is a combination of two or more metals.

Brass consists of about 2 parts of copper and 1 part of zinc.

Muntz Metal consists of about 3 parts of copper and 2 parts of zinc.

White Metal consists of about 84 per cent. of tin, and the remainder of copper and antimony.

NOTE.—White metal melts at about 600° Fahr.

Stresses on Various Parts.

Boilers.

On the stays the stress is *tensile.* On the shell plates the stress is *tensile.* On the furnace the stress is *compression.* On the tubes the stress is *compression.* On the stay tubes the stress is *compression* and *tensile.* On the back tube plates the stress is *compression.* On the combustion chamber dogs the stress is *compression* on top edge and *tensile* on bottom edge. On the rivets the stress is *shearing.*

Engines.

On the shafting the stress is torsion, but the tail end shaft has also a bending stress due to the propeller weight outside of the stern tube.

On the shafting from the thrust shaft aft there is an end compressive stress going ahead, and a tensile stress going astern.

On the crank-shaft there are also bending and crushing stresses combined with torsion.

For these reasons the tail shaft and shaft crank are usually made a little larger in diameter than the tunnel shafting (about 1 inch).

Coupling bolts have a shearing stress. Crank webs and pump levers have a bending stress.

Built Shafting.—Instead of shrinking the webs on to the shaft, some engineering firms force them on by hydraulic pressure, the ram exerting a load of anything from 100 to 125 tons. The holes in the webs are bored out a few thousandths less in diameter than the pin or shaft, and for a shaft of, say, 14 inches diameter, the difference would amount to $\frac{14}{1000}$ inch, which is just under $\frac{1}{64}$ inch.

Strength of Shafting.

The strength of a **solid shaft** varies as the cube of its diameter, therefore, the comparative strength of two shafts of, say, 8 inches diameter and 10 inches diameter will be—

$$\frac{\text{Diameter}^3}{\text{Diameter}^3}=\frac{10^3}{8^3}=\text{Ratio of strengths}=\text{as } 1 : 1{\cdot}95.$$

A **hollow shaft** is stronger than a solid one of the **same sectional area,** as, the diameter being greater, the leverage of the power acting to twist it is less in proportion.

In addition to this, the removal of the central core of metal reduces the risks of flaws, which often develop at the centre and then extend outwards. Internal inspection for flaws is also to some degree possible.

NOTE.—The torsional stress is *o* at the centre of a shaft, and increases from that point out to the circumference; the mean stress may therefore be taken as acting at a leverage of half of the shaft radius, or one-fourth of the shaft diameter.

The strength of a hollow shaft varies as

$$\frac{D^4-d^4}{D} \qquad D=\text{outer diameter.} \quad d=\text{inner diameter.}$$

A shaft will stand twice as much torsion stress as bending stress, the Constant for torsion being 5·1, and for bending 10·2.

RULE (TORSION).—

$$5{\cdot}1 \times \text{Load} \times \text{Crank Length} = \text{Torsional Stress per square inch} \times \text{Shaft}^3.$$

Therefore,

$$\sqrt[3]{\frac{5{\cdot}1 \times \text{Load} \times \text{Crank Length}}{\text{Torsional Stress}}} = \text{Diameter of Shaft},$$

or,

$$\frac{5{\cdot}1 \times \text{Load} \times \text{Crank Length}}{\text{Shaft Diameter}^3} = \text{Torsional Stress}.$$

NOTE.—For Torsional Stress the maximum Load may be taken as approximately equal to that on the piston, or,

$$\text{Piston area} \times \text{Pressure} = \text{Load}.$$

RULE (BENDING).—

$$10{\cdot}2 \times \text{Load} \times \text{Length} = \text{Bending Stress} \times \text{Shaft}^3.$$

Twisting Moment (T.M.) = Crank Length × pounds load on pin.

Bending Moment (B.M.) = Length × pounds Load.

EXAMPLE.—Calculate the bending stress per square inch on a tail end shaft 12 inches diameter, distance from stern post to centre of propeller boss 30 inches, weight of propeller 10 tons. Also express the Bending Moment in inch-pounds.

Then, $10{\cdot}2 \times \text{B.M.} = d^3 \times \text{stress},$

$$\therefore \quad \frac{10{\cdot}2 \times \text{B.M.}}{d^3} = \text{stress},$$

$$\text{B.M.} = 30 \text{ inches} \times 10 \times 2240 = 672000 \text{ inch-pounds}.$$

$$\text{Stress} = \frac{10{\cdot}2 \times 672000}{12^3} = 3966 \text{ lbs. per square inch}.$$

Power and Revolutions.

For a given power the higher the revolution speed the less the diameter of shaft required, as the stress decreases with the speed.

To find the diameter of shaft necessary for a certain length of crank and piston load (allowing 7000 lbs. as the safe torsional stress).

RULE.—

$$\sqrt[3]{\frac{5{\cdot}1 \times \text{Load} \times \text{Crank Length}}{7000}} = \text{Diameter of Shaft}.$$

NOTE.—The above rule requires the extraction of the *cube* root, as shown by the sign.

Torsional Stress on Shafts and Constant, 5·1.—To prove that the strength of solid shafting depends on the cube of the diameter, and to determine the origin of the Constant 5·1.

EXPLANATION.—The shaft area is the shearing area resisting torsion, and the mean leverage is equal to one-half of the shaft radius, or one-fourth of the shaft diameter, as the stress is 0 at shaft centre and maximum at the radius, therefore, $\frac{0 + \text{Radius}}{2} = $ mean leverage, or, which is the same thing, $\frac{\text{Diameter}}{4} = $ mean leverage. If, then, we find the shaft area, and multiply by the stress (8000 lbs. per square inch for steel) and by the mean leverage, we obtain the *resistance* to torsion offered by the shaft metal.

Thus, $\text{Diameter}^2 \times \cdot 7854 \times \text{Stress} \times \frac{\text{Diameter}}{4} = \text{Shearing Moment},$

or, $\text{Diameter}^2 \times \frac{3 \cdot 1416}{4} \times \text{Stress} \times \frac{\text{Diameter}}{4} =$,, ,,

or, $\frac{\text{Diameter}^2 \times 3 \cdot 1416 \times \text{Stress} \times \text{Diameter}}{4 \times 4} =$,, ,,

or, $\frac{\text{Diameter}^2 \times 3 \cdot 1416 \times \text{Stress} \times \text{Diameter}}{16} =$,, ,,

or, $\frac{\text{Diameter}^3 \times 3 \cdot 1416 \times \text{Stress}}{16} =$,, ,,

Notice that $\text{Diameter}^2 \times \text{Diameter} = \text{Diameter}^3$, also that instead of $\cdot 7854$ we may say $\frac{3 \cdot 1416}{4}$, which is equal to $\cdot 7854$; it will thus be seen that the strength varies as the "Diameter cubed."

Again, $\text{Load} \times \text{crank leverage} = \frac{D^3 \times 3 \cdot 1416 \times \text{Stress}}{16};$

or, $\frac{\text{Load} \times \text{crank leverage}}{\frac{3 \cdot 1416}{16}} = \text{Diameter}^3 \times \text{Stress};$

or, $\frac{\text{Load} \times \text{crank leverage} \times 16}{3 \cdot 1416} = \text{Diameter}^3 \times \text{Stress};$

or, $\text{Load} \times \text{crank leverage} \times 5 \cdot 1 = \text{Diameter}^3 \times \text{Stress};$

so that in dividing by $\frac{3 \cdot 1416}{16}$, we invert it, and obtain $\frac{16}{3 \cdot 1416}$, which gives $5 \cdot 1$ Constant for torsion, and as the resistance to bending stress is only half of this, then $5 \cdot 1 \times 2 = 10 \cdot 2 =$ Constant for bending stress.

Twisting Stresses and Tunnel Shaft Diameter.—The ratio of piston travel to crank-pin travel per revolution is in the ratio of $2 : 3 \cdot 1416$, as during one revolution the piston travels through *two* strokes and the crank-pin through a circle equal in diameter to the stroke. The crank-pin has therefore more travel than the piston, and, by the principle of work, what is gained in travel is lost in pressure, so that the average pressure on the crank-pin is *less* than the average pressure on the piston in the ratio of $3 \cdot 1416 : 2$.

EXAMPLE.—The stroke is 4 feet and the total mean pressure on the piston 36000 lbs. Find the mean pressure on the crank-pin.

Then, $4 \times 2 \times 36000 = 4 \times 3{\cdot}1416 \times p$,

and, $p = \dfrac{4 \times 2 \times 36000}{4 \times 3{\cdot}1416} = 22918$ lbs.

Mean Twisting Moment.

The mean twisting moment (T.M.) = Crank Length × Pounds (L × P).

Therefore, T.M. = 24 inches × 22918 = 550072 inch-pounds.

For one minute the equation will read as follows:—

$$\text{I.H.P.} \times 33000 \times 12 \text{ inches} = \text{Crank Length} \times 2 \times 3{\cdot}1416 \times \text{Pounds} \times \text{Revs.}$$

Therefore, $\dfrac{\text{I.H.P.} \times 33000 \times 12}{2 \times 3{\cdot}1416 \times \text{Revs.}} = \text{Crank Length} \times \text{Pounds (T.M.)}$.

EXAMPLE.—I.H.P. 1,400, stroke 4 feet, and revolutions 62. Find the mean T.M.

Then, $1400 \times 33000 \times 12 = 24 \text{ inches} \times 2 \times 3{\cdot}1416 \times 62 \times \text{Pounds}$.

Therefore, $\text{T.M.} = \dfrac{1400 \times 33000 \times 12}{2 \times 3{\cdot}1416 \times 62} = 1423180$ inch-pounds,

and, $\text{Pounds} = 1423180 \div 24 \text{ inches} = 59298$ lbs.

Notice that the Twisting Moment in inch-pounds divided by the crank length in inches brings out the pounds applied at end of crank.

Maximum T.M.

The foregoing only takes into account the mean or average T.M., and to allow for the usual cut-off and the varying effects of the crank angle a constant of about 1·2 is usually employed, so that

$$\text{Maximum T.M.} = \text{Mean T.M.} \times 1{\cdot}2.$$

Shaft Diameter.

From the foregoing principles the required diameter of tunnel shaft for a three-crank engine can be calculated as follows:—

$$\text{I.H.P.} \times 33000 \times 12 \times \overset{\text{C.}}{5{\cdot}1} \times 1{\cdot}2 = d^3 \times 2 \times 3{\cdot}1416 \times \text{Revs.} \times \text{Stress.}$$

Therefore, $d = \sqrt[3]{\dfrac{\text{I.H.P.} \times 33000 \times 12 \times \overset{\text{C.}}{5{\cdot}1} \times 1{\cdot}2}{2 \times 3{\cdot}1416 \times \text{Revs.} \times \text{Stress}}}$.

Allowing a stress of 7,000 lbs. per square inch the shaft diameter will be equal to

$$d = \sqrt{\frac{1400 \times 33000 \times 12 \times 5{\cdot}1 \times 1{\cdot}2}{2 \times 3{\cdot}1416 \times 62 \times 7000}} = 10{\cdot}7 \text{ inches Diameter.}$$

BOARD OF TRADE RULE FOR TUNNEL SHAFTS.—

$$\text{Shaft Diam.} = \sqrt[3]{\frac{\text{Crank Length} \times \text{Absolute boiler pressure} \times \text{L.P.}^2}{\frac{1295}{\text{C.}} \times \left(2 + \frac{\text{L.P.}^2}{\text{H.P.}^2}\right)}}.$$

The boiler pressure = 160 lbs. (gauge) and the cylinders 22 inches, 38 inches, 64 inches diameter.

Therefore,

$$\text{Shaft Diam.} = \sqrt[3]{\frac{24 \times 175 \times 64^2}{1295 \times \left(2 + \frac{64^2}{22^2}\right)}} = 11 \text{ inches (nearly).}$$

It will be noticed that the Board of Trade Rule brings out a slightly larger shaft than that previously calculated, which, of course, allows for a larger margin of safety.

Combined Twisting and Bending Stresses.—As the crank-shaft is subjected to a combined twisting and bending stress, both of which require to be allowed for in estimating the size, the Board of Trade Constant is 1110 in place of 1295: for tunnel shafts this gives a larger diameter of shaft, which also holds good for the thrust length; but the propeller shaft requires to be still larger, as the Constant for this length is 943.

Board of Trade Shaft Constants.

(For Engines with Three Cranks at 120°.)

	Tunnel Shaft.	Crank and Thrust Shafts.	Propeller Shafts.
Constant - -	1295	1110	943

Lloyd's Rules for Shafting.

Triple Expansion Engines with Cranks at 120°.

(1.) Diameter of tunnel shafting $= (\cdot038 \times A + \cdot009 \times B + \cdot002 \times C + \cdot0165 \times S) \times \sqrt[3]{P}.$

Where
A = H.P. cylinder diameter.
B = M.P. cylinder diameter.
C = L.P. cylinder diameter.
S = Stroke in inches.
P = Boiler pressure (gauge).

(2.) Diameter of crank-shaft and of thrust shaft = diameter of tunnel shaft $\times \frac{21}{20}$.

(3.) Diameter of propeller shaft =

$$\text{diameter of tunnel shaft} \times \left(\cdot63 + \frac{\cdot03 \times \text{diameter of propeller in inches}}{\text{diameter of tunnel shaft}}\right)$$

(with liner from end to end).

(4.) Diameter of propeller shaft =

diameter of tunnel shaft $\times \frac{21}{20}$ (with two separate liners).

EXAMPLE 1.—Determine the required diameter of tunnel shaft, thrust shaft, crank-shaft, and tail shaft (with continuous liner) according to Lloyd's Rules for an engine with cylinders, 24, 40, and 66 inches diameter. The stroke is 42 inches, and the boiler pressure 190 lbs. The propeller diameter is 14 feet.

$$\text{Tunnel shaft diameter} = (\cdot038 \times 24 + \cdot009 \times 40 + \cdot002 \times 66 + \cdot0165 \times 42) \times \sqrt[3]{190} =$$
$$(\cdot912 + \cdot360 + \cdot132 + \cdot6930) \times 5\cdot74 = 2\cdot0970 \times 5\cdot74 = 12\cdot03 \text{ inches.}$$

Say, $12\frac{1}{8}$ inches diameter.

$$\text{Crank-shaft and thrust shaft diameter} = 12\cdot125 \times \frac{21}{20} = \begin{cases} 12\cdot73 \text{ inches. Say,} \\ 12\frac{3}{4} \text{ inches diameter.} \end{cases}$$

$$\text{Propeller shaft diameter} = 12\cdot125 \times \left(\cdot63 + \frac{\cdot03 \times 168}{12\cdot125}\right) =$$
$$12\cdot125 \times \left(\cdot63 + \frac{5\cdot04}{12\cdot125}\right) = 12\cdot125 \times (\cdot63 + \cdot4156) = 12\cdot125 \times 1\cdot0456 = 12\cdot68 \text{ inches.}$$

Say, $12\frac{3}{4}$ inches diameter.

NOTE.—$\sqrt[3]{P}$ means extract cube root of boiler pressure.

14 feet diameter = 168 inches.

EXAMPLE 2.—Calculate the required diameter of tunnel shaft, thrust shaft, crank-shaft, and propeller shaft for an engine with cylinders of 27, 42, and 73 inches diameter, the stroke being 48 inches, and the boiler pressure 180 lbs. per square inch. The diameter of the propeller is 15 feet 6 inches, and the tail shaft is brass-lined from end to end.

$$\text{Then tunnel shaft diameter} = (\cdot038 \times 27 + \cdot009 \times 42 + \cdot002 \times 73 + \cdot0165 \times 48) \times \sqrt[3]{180} =$$
$$(1\cdot026 + \cdot378 + \cdot146 + \cdot792) \times 5\cdot64 = 2\cdot342 \times 5\cdot64 = 13\cdot2 \text{ inches.}$$

Say, $13\frac{1}{4}$ inches diameter.

$$\text{Crank-shaft and thrust shaft diameter} = 13\cdot25 \times \frac{21}{20} = \begin{cases} 13\cdot912 \text{ inches. Say,} \\ 14 \text{ inches diameter.} \end{cases}$$

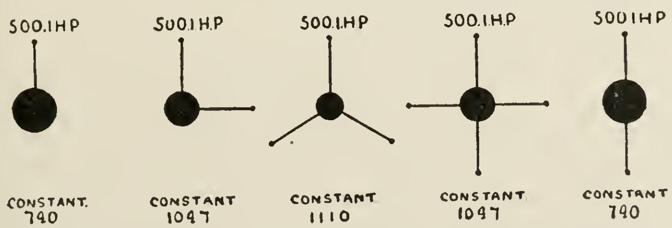
Propeller shaft diameter =

$$13\cdot25 \times \left(\cdot63 + \frac{\cdot03 \times 186}{13\cdot25}\right) = \begin{cases} 13\cdot25 \times (\cdot63 + \cdot421) = 13\cdot25 \times 1\cdot051 = 13\cdot92 \text{ inches.} \\ \text{Say, 14 inches diameter.} \end{cases}$$

NOTE.—15 feet 6 inches = 186 inches diameter of propeller.

Crank Angles.

Suppose we take five different engines, each to develop the same I.H.P. No. 1 has one crank, No. 2 has two cranks at 90°, No. 3 has three cranks at 120°, No. 4 has four cranks at 90°, and No. 5 has two



No. 1.—Board of Trade Constants for Shafting.

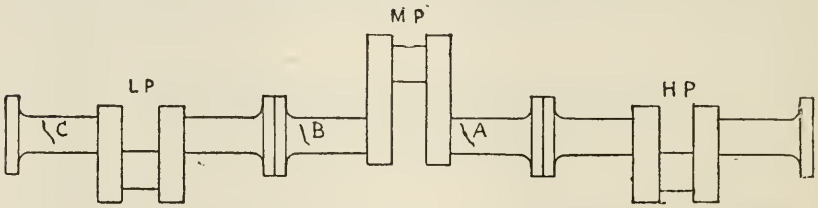
cranks opposite each other; then the accompanying sketch shows the comparative diameter of shafts required in each case, and the corresponding Board of Trade constants.

Observe that one crank and two cranks at 180° require the same diameter of shaft, the constant being the same. Also the two cranks at right angles and the four cranks at right angles require the same diameter of shaft.

The three-crank engine requires the smallest diameter of shaft, as in this shaft arrangement of cranks the twisting stress on the shaft is more evenly divided, and gives less variation than in any of the other arrangements.

Flaws on Shafting.

Suppose that in a triple-expansion engine each cylinder develops about 300 I.H.P.; then between the H.P. crank-pin and the M.P. crank-pin we have 300 horse-power on the shaft; between the M.P. crank-pin and the L.P. crank-pin we have 600 horse-power on the shaft; and between the L.P. crank-pin and the propeller we have



No. 2.—Flaws on Crank Shafting.

900 horse-power on the shaft, as all the power effectively developed travels along the shafting aft to the propeller, each engine successively adding its share.

If the M.P. shaft shows a flaw at A, link up the M.P. gear, and this will reduce the stress on the weak part, as less horse-power will now be developed in the H.P. engine, and more in proportion in the M.P.; therefore less will be transmitted through the weak part of the shaft.

If a flaw shows at B on the M.P. shaft, if possible turn that shaft end for end, as this will give only half the stress on the weak part. Shutting in the M.P. gear would still further reduce the stress, as more power will now be developed in the M.P. and less in the H.P.

For a flaw at C on the L.P. shaft, as before, turn the shaft end for end, and link up the L.P. gear, on the same principle as before.

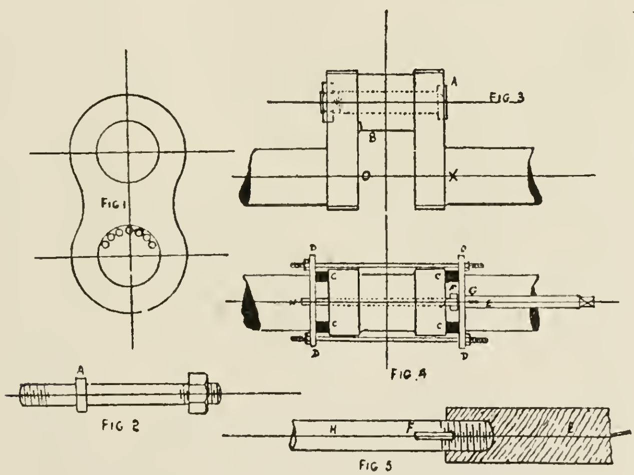
If the H.P. shaft is a duplicate of the other two, in the event of flaws appearing in either the M.P. or L.P. shafts, change the H.P. shaft for either one which has the flaw, instead of doing as above described, as the further forward the weak shaft is placed, the less stress will there be on it.

Crank-Shaft Repairs.

The following description of crank-shaft repairs, supplied by John M'Callum, Esq., are reprinted from the pages of *International Marine Engineering.*

(1.) **Loose Web.**—The crank-shaft had been kept under observation, as a certain amount of slackness had been noticed between the shaft and the web. Later on, when at sea, knocking was heard, and it was located in the loose web. The repair was effected by drilling a number of holes round the end of the shaft, as shown in Fig. 1, and fitting taper screw pins, $1\frac{3}{8}$ inches diameter, in each hole, and screwing them up as tightly as a key with a good leverage would accomplish. $1\frac{3}{8}$ inches was the size of the largest screw tap on board.

Holes were drilled in the end of the shaft for about one-third of its circumference, as shown in the figure; the line of the holes being arranged to leave $\frac{1}{2}$ inch of metal between them and the edge of the shaft. The holes were drilled 6 inches deep, and screwed with taper tap only. Each hole and pin was finished, and the pin screwed up as



No. 3.—Repair for Loose Crank Web, &c.

far as it would go before commencing the next. The line of screwed pins swelled the shaft in the neighbourhood of the holes, and tightened the shaft in its web sufficiently to carry the ship to her home port without further trouble. When arrived in port a new shaft end was fitted to the existing web. The possibility of this being done, and so avoiding the expense of new shaft and web, had been foreseen when the method described above was adopted. If plugs had been fitted interlocking the shaft and web, apart from the fact of their liability to slack back when at work, a new shaft and a new web would have been required on arrival home.

(2.) **Flaw in Fillet.**—When at sea, a flaw was noticed in the fillet of the low-pressure crank-pin, which though slight at first developed very rapidly, necessitating a very careful repair, or stoppage of the ship would have been necessary. The flaw extended over about one-third of the circumference of the after fillet and underneath the pin, "crank on top," as shown at B in Fig. 3. In the writer's experience, defects in crank-shafts, whether solid or built up, usually occur at this point, and are mainly attributable to the faulty position of the thrust block. Instead of the thrust being taken up by the shaft bearing, the thrust block has to be adjusted, and a nip is thrown on the crank-pin at the point B in Fig. 3, which usually results in a loose pin in a built-up crank-shaft, or a flaw, as in this case, in a solid one. As the runs between the ports were not long the engines were kept running, but were well watched, while the engineers thought out the difficult problem of repair and got the necessary appliances ready.

The repair was effected by fitting a pin through the crank, as shown at A in Fig. 3, a spare main-bearing bolt with collar and nut, shown in Fig. 2, being employed for the purpose. Drills and ratchets were got all ready before the engine was stopped, and drilling was commenced from each end, a $1\frac{3}{8}$-inch hole being drilled through both webs and crank-pin, the holes meeting in the centre. Coloured labour being plentiful on the coast, the holes were got out in good time.

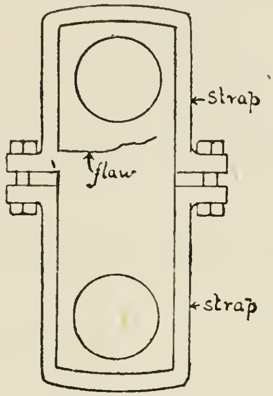
After the holes were drilled the complete hole was bored by means of the arrangement shown in Fig. 4. A boring bar E was fitted with a pilot end H screwed into the boring bar, to save forging down and to keep the bar true. The slot for the cutting tool was arranged, as shown in F in Fig. 5, so that the body of the bar E took up the weight of the cutting. Another slot was cut at G in the boring bar, and a larger cutting tool was placed in it, which came into operation after the cutter F.

Fig. 4 shows the arrangement of the boring bar. The plates D D were used to support it, with the distance pieces C C, long bolts, as shown, holding the whole thing together, and keeping the bar and the boring central with the pin. The hole was recessed at each end to receive the collar and nut, as shown in Fig. 3, and the nut was cut down for the recess.

When the boring was finished the crank was well warmed by means of a wood fire, the bolt tapped in with an anvil slung in ropes for the purpose, the bolt end outside of the collar being left on until the bolt was right in its place. The nut was then put on, hardened up, and the bolt end riveted over it. The ship completed her charter, and ran home with the repaired shaft. A new shaft had been sent out, but it was not necessary to use it.

Repairs for Flawed or Broken Shafts.

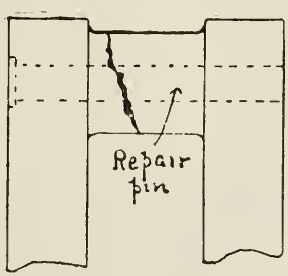
Flaw on Crank Web.—Two iron or steel straps, say about 2 inches thick, heated and shrunk on, then bolted as shown.



No. 4.—Repair for Flaw in Crank Web.

The bolts should be as large as possible, for ordinary size of shafting about $2\frac{1}{2}$ inches diameter, but larger than this if they can be obtained. Coupling bolts would do very well in most cases.

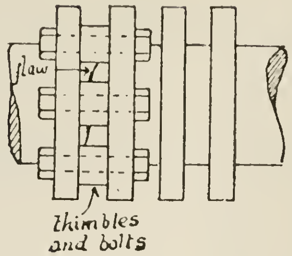
Broken Crank-Pin.—Bore out the crank-pin to about one-third of its diameter, and put in a repair-pin, a driving fit. A small locking-



No. 5.—Repair for Broken Crank Pin.

pin screwed half into crank-pin and half into repair-pin will keep it in place.

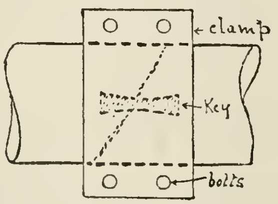
Repair for Thrust Shaft—The collars to be bored or slotted out, and bolts with thimbles fitted between the collars where the flaw or



No. 6.—Repair for Broken Thrust Shaft.

break is situated. The thrust block rings next the broken part will have to be removed.

Repair for Tunnel Shaft.—The shaft to be cut for keys of the shape shown, the number of keys depending on the extent of the



No. 7.—Repair for Broken Tunnel Shaft.

flaw or crack, and the shaft clamped round over the keys and securely bolted.

NOTE.—In the foregoing cases the revolutions will require to be reduced.

Notable Shaft Repair.—The following is a brief description of the method adopted in repairing the broken tail end shaft of the steamer "Fazilka," of the British India Steam Navigation Company, in the Indian Ocean.

The binding of the shaft together by means of a set of bottom end brasses applied as a clamp, and the further locking of the clamp by steel pins driven in through the brass into the shaft, constitute the most noteworthy and original points of this repair.

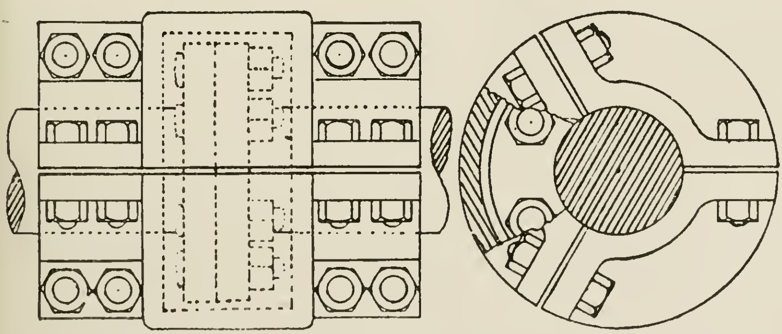
The tail end shaft gave way in the stern tube at two places, so that a piece fully 3 feet 6 inches in length was detached, and in breaking also broke through the stern tube. The engines were promptly stopped and the stern of the ship afterwards tipped to prevent the entrance of water to the tunnel while the work of repair was being carried on.

A number of holes were first bored into the tube and the metal broken away, so that a hole was made large enough to allow of access to the shaft inside. The broken piece was removed and the propeller shaft disconnected from the last tunnel length, and pushed out aft until the broken ends touched. Two sets of bottom end brasses were then taken and used as clamps to bind together the two parts of the shaft, and to obtain one of these sets the H.P. engine had to be disconnected. Plates of ½ inch thickness were fitted across the two brasses top and bottom, and the bolts passed through to more effectually support the broken parts, and, to allow of going astern, holes, two of 2 inches diameter and two of 2¼ inches diameter, were bored into the shaft through the brasses, and steel pins driven in. It will be noticed that a gap was left between the last tunnel length of shafting and the tail end, owing to the bringing together of the broken ends of the latter. To join these the after length of the tunnel shaft (13½ inches diameter) was *cut through*, by first boring round it twenty-two holes of about 1 inch diameter and cutting between them. The flange was then brought aft and coupled to the propeller shaft, and the cut parts of the tunnel shaft connected by means of a "Thomson" patent coupling. To complete the job and make the whole, as far as possible, one solid mass, molten metal was run in to fill up the various spaces left by the ragged ends of the broken shaft inside of the clamps, and when the engines were turned round the parts were found to hold together satisfactorily enough to allow of a reduced speed being easily maintained.

The work spent on repairing the shaft occupied about three weeks' time, but this was mainly due to temporary failure of one or two of the methods tried, which are not given here in detail. It is sufficient to add that the "Fazilka" steamed safely, and without assistance, into Colombo Harbour by the use of her M.P. and L.P. engines, working at a reduced boiler pressure.

"Thomson" Patent Coupling.

This coupling is specially designed for clamping up a broken shaft, and forms the best means of repair. It consists of three



No. 8.—"Thomson" Patent Coupling.

pieces bolted together, and is so arranged that it may extend between two lengths of shafting if required, an enlarged part of the coupling allowing for the flanges.

Stopping of Engines.

Sudden stopping of the engines may be caused by the following:—

1. Slide valve loose on spindle, or spindle broken.
2. Stop valve seat lifted with valve.
3. Go-ahead eccentric broken, or loose on shaft.
4. Throttle valve turned round on its spindle.

If any steam connection on the H.P. is opened and steam blows out, this proves that the stop valve and throttle valve are clear.

If the slide valve is loose on the spindle, the piston of that engine will most likely stop on the top centre, as the valve will stick on the top, and the bottom port remain full open to steam. If the steam be shut off quickly, the noise of the valve dropping down would locate the trouble.

In the case of a valve slack on the spindle the following generally holds good:—

1. If H.P. inside steam piston valve is slack on spindle, H.P. engine will stop with piston on *bottom* centre.
2. If I.P. or L.P. slide valve (or *outside* steam piston valve) is slack on spindle, the corresponding engine will stop with piston on *top* centre.

Failing the foregoing, testing, by means of the indicator cock connections top and bottom, will usually locate the valve which has slackened back.

GENERAL ENGINE BREAKDOWNS.

If the H.P. engine breaks down, take out the valve and disconnect the engine, and work with the I.P. and L.P. The pressure should be reduced to about 100 lbs., as the I.P. cylinder is of larger diameter the H.P., and therefore weaker.

If the I.P. engine breaks down, take out the valve and disconnect the engine (if the pumps are not worked off it). The steam will then pass direct from the H.P. exhaust to the L.P. chest. The boiler pressure may require to be reduced in this case, owing to the fact that the steam in expanding down to the L.P. pressure does not drop in temperature, as no work is done during this expansion, and to obtain the same condenser vacuum the boiler pressure may, as stated, require to be reduced. To develop the same I.H.P. the consumption will be more, owing to the great difference of temperature existing between admission and exhaust in the H.P. cylinder, and the consequent excessive condensation of steam causing loss of heat.

NOTE.—In the foregoing cases it is advisable to leave in the valve spindle to close up the gland.

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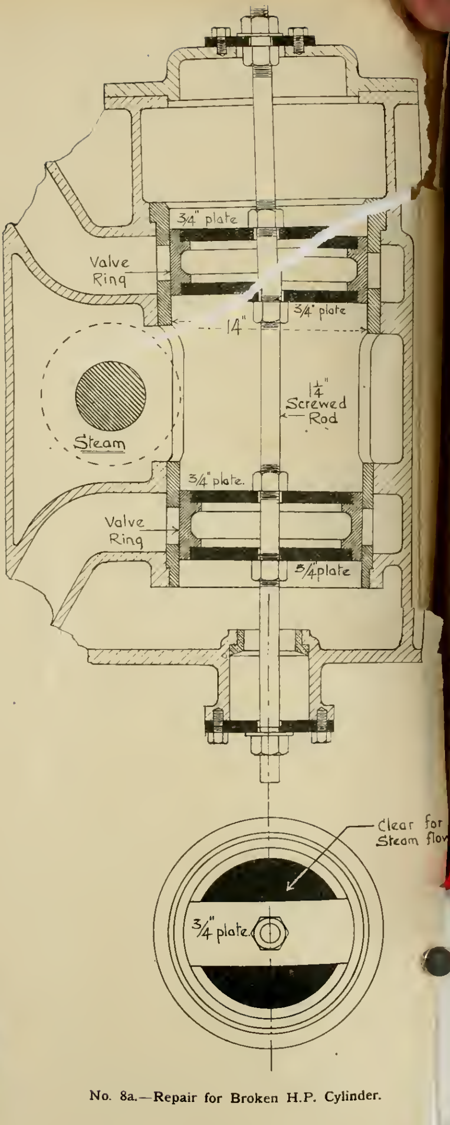
gear, eccentric straps, &c., leaving in the valve rod to close up the gland, by lashing it to guide bracket. Then reduce the pressure until the I.P. and L.P. gauges show, say, 4 or 5 lbs. higher than formerly, and go on again. The H.P. piston, having steam in both sides, will be balanced by the pressure.

It will be found that the pressure will drop considerably by the free expansion of the steam in the H.P. cylinder.



No. 8a.—Repair for Broken H.P. Cylinder.

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The following methods of repair in cases of breakdown require the least time to carry out under conditions of urgency, and where economy is of secondary importance.

1. H.P. Cylinder Broken Beyond Repair.

The sketch (No. 8*a*) illustrates the best method of repair which requires the least time in getting under way again.

Remove broken parts of cylinder, also piston, and gear as required, then draw H.P. piston valve and remove rings of same. Next expand out the rings by inserting liners at the cut portion, and drive them into the casing opposite the ports, taking care that the cut in the rings is in line with the bridge metal of the ports. To prevent the rings from being blown out of position, fit plates on a long screwed bar as shown, with nuts top and bottom of the rings, securing the bar by washers, plates, and nuts at top and bottom of the valve casing.

The pressure should be reduced from, say, 180 lbs. to 110 or 120 lbs.

This method of repair was carried out very successfully by Chief Engineer D. M'Culloch and the engine room staff of the S.S. "Numidian" of the Allan Line in August 1911, the time occupied on the work being 26½ hours.

The consumption, speed, and pressures before and after the breakdown in the case of the "Numidian" were as follows:—

	Before Breakdown.	After Breakdown.
Boiler steam - -	160 lbs.	100 lbs.
Consumption - -	56 tons.	48 tons.
Speed - - - -	13·5 knots.	10·5 to 11 knots

After breakdown the I.P. steam was 45 lbs., L.P. steam 7 lbs., and the revs. 52. The vessel afterwards steamed over 3300 miles as repaired without trouble of any kind developing.

2. H.P. Piston Valve Broken Beyond Repair.

In this case merely draw the valve, and disconnect the valve gear, eccentric straps, &c., leaving in the valve rod to close up the gland, by lashing it to guide bracket. Then reduce the pressure until the I.P. and L.P. gauges show, say, 4 or 5 lbs. higher than formerly, and go on again. The H.P. piston, having steam in both sides, will be balanced by the pressure.

It will be found that the pressure will drop considerably by the free expansion of the steam in the H.P. cylinder.

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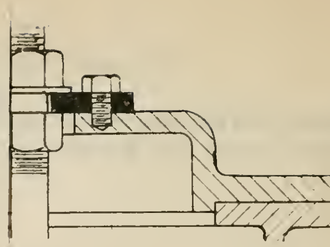

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require to be reduced. To develop the same I.H.P. the consumption will be more, owing to the great difference of temperature existing between admission and exhaust in the H.P. cylinder, and the consequent excessive condensation of steam causing loss of heat.

NOTE.—In the foregoing cases it is advisable to leave in the valve spindle to close up the gland.

The following methods of repair in cases of breakdown require the least time to carry out under conditions of urgency, and where economy is of secondary importance.

1. H.P. Cylinder Broken Beyond Repair.

The sketch (No. 8*a*) illustrates the best method of repair which requires the least time in getting under way again.

Remove broken parts of cylinder, also piston, and gear as required, then draw H.P. piston valve and remove rings of same. Next expand out the rings by inserting liners at the cut portion, and drive them into the casing opposite the ports, taking care that the cut in the rings is in line with the bridge metal of the ports. To prevent the rings from being blown out of position, fit plates on a long screwed bar as shown, with nuts top and bottom of the rings, securing the bar by washers, plates, and nuts at top and bottom of the valve casing.

The pressure should be reduced from, say, 180 lbs. to 110 or 120 lbs.

This method of repair was carried out very successfully by Chief Engineer D. M'Culloch and the engine room staff of the S.S. "Numidian" of the Allan Line in August 1911, the time occupied on the work being 26½ hours.

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2. H.P. Piston Valve Broken Beyond Repair.

In this case merely draw the valve, and disconnect the valve gear, eccentric straps, &c., leaving in the valve rod to close up the gland, by lashing it to guide bracket. Then reduce the pressure until the I.P. and L.P. gauges show, say, 4 or 5 lbs. higher than formerly, and go on again. The H.P. piston, having steam in both sides, will be balanced by the pressure.

It will be found that the pressure will drop considerably by the free expansion of the steam in the H.P. cylinder.

3. I.P. Cylinder Cover Broken Beyond Repair.

Remove broken parts of cover, draw valve, and disconnect valve gear, then close up ports by means of wooden plugs and cement. Also remove piston rings to eliminate pull on piston due to vacuum formed on under side during up stroke.

Reduce boiler pressure if found necessary and go on again.

4. H.P., I.P., or L.P. Piston Broken Beyond Repair.

Take off cylinder cover and remove piston, then screw down cover again and go on, leaving the valve working as before. Reduce the boiler pressure as the case requires.

NOTE.—The steam will enter the cylinder and exhaust out of it again as previously.

Remarks.—In most of the foregoing examples it would be found that after repair the speed, power, and consumption would be reduced, but that the consumption per I.H.P. per hour would be increased, although in many cases the total consumption also rises. In the case of the H.P. cylinder being cut out it would be advisable to open out the I.P. link to gain power by carrying steam later on the stroke of that engine which is now virtually the H.P. It may also be found necessary to line up the slide valves to improve handling of the engines by increasing the bottom lead.

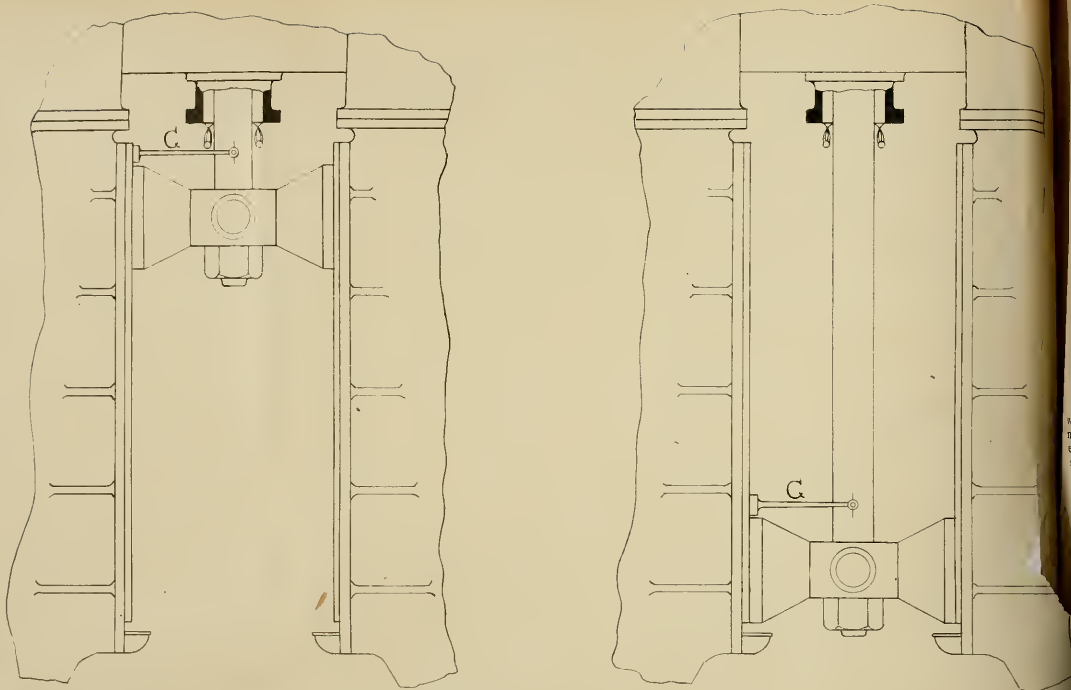
In all cases, when steam is allowed to enter a cylinder freely as described in No. 4, a certain amount of loss by condensation must be expected. This loss is, of course, much reduced if the ports are blanked off and the cylinder cut out altogether, which should be done if the circumstances allow.

ENGINE TESTING.

1. To Test if Guides are Parallel to Piston Rod Line.

(*With all Gear in Position.*)

First draw out gland and remove packing, then place engine on top centre, and see by caliper or gauge measurements that rod is exactly fair in stuffing box, also test rod for fairness by means of a steel straight-edge. Next, by a surface gauge or a stiff wire shaped into a gauge take distance from guide face surface at top to a mark made on piston rod (see sketch). Place engine on bottom and again test rod for fairness in stuffing box by caliper measurements, and with gauge measure distance from guide face surface at bottom to *same mark* on piston rod: if the two measurements agree the rod is proved to be working fair in the cylinder, and the guides proved to be parallel to the rod.



No. 8b. Method of Testing if Guides are Parallel to Piston Rod.

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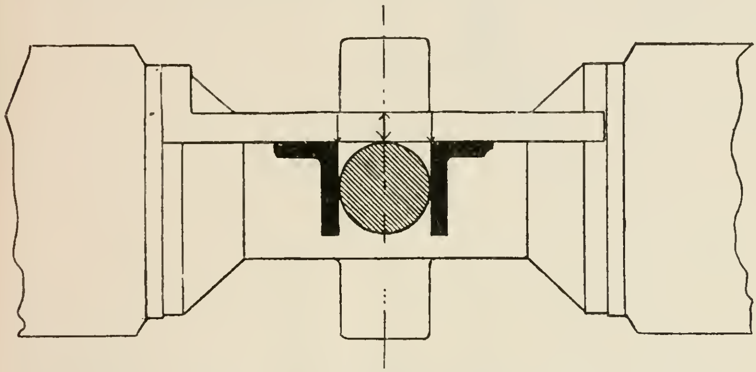
NOTE.—As the guides are closer together at top than at bottom when cold (about $\frac{20}{1000}''$ or so) it is advisable to make the test before the engines are cooled down otherwise due allowance requires to be made for the difference mentioned. If on testing the rod in stuffing box it is found not to be in dead centre, then the rod and piston will not be working truly in the cylinder bore, and of course the other test will then be unnecessary.



No. 8c.—Method of Locating Centre of Piston Rod between Guides.

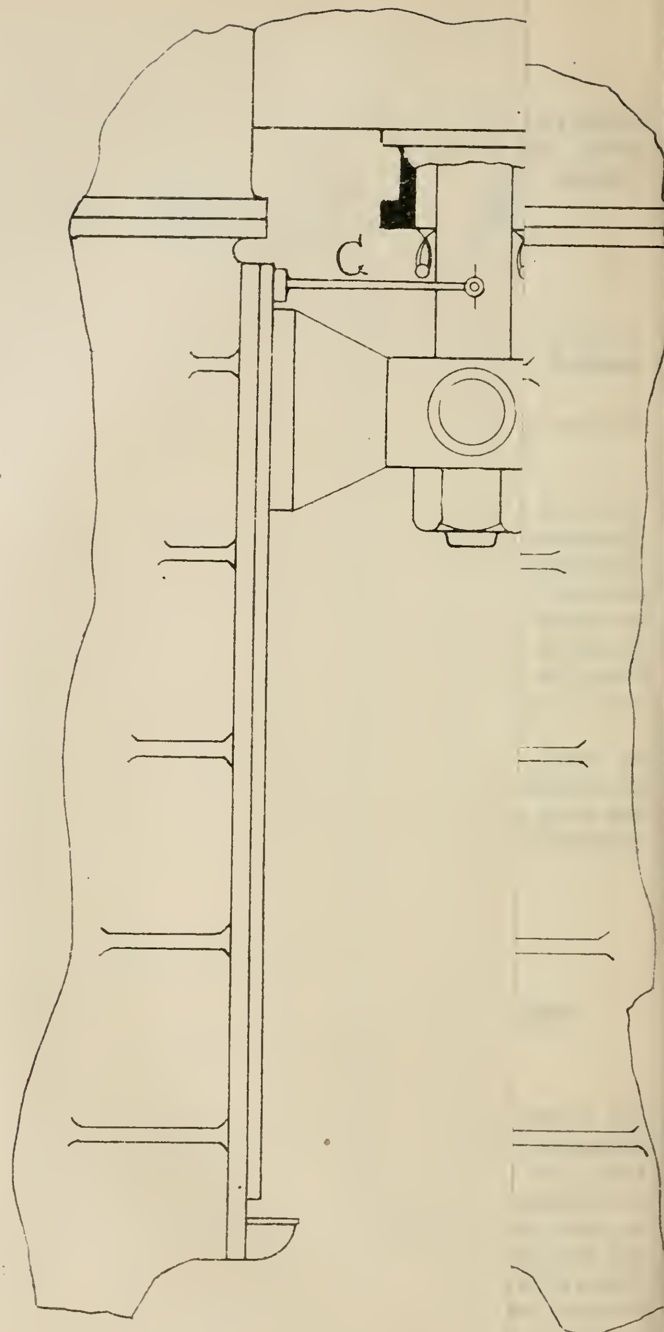
The sketch (No. 8*c*) shows another method of finding a centre from which to test the fairness of the guides to the piston rod line. By means of the squares shown the centre of the piston rod can be exactly located as indicated by the mark. If now a wire is fixed to, say, a small plate bolted on the gland flange, and is then led down to the shaft centre, this line can be used as a test for the guide face setting, or piston rod trueness of line.

2. To Test if Guides are Parallel to Shaft Centre Line.

Turn the crank up to near the top centre and place a long steel straight-edge diagonally across the guide face (sketch No. 8*d*), with the lower end lying forward, and with a gauge measure the distance from the straignt-edge to the crank pin centre forward: next slope the straight-edge at the opposite angle on the guide face with the lower end lying aft, and with the gauge measure the distance between straight-edge and the crank pin centre aft: if the two measurements agree the shaft line (assuming, of course, that the crank pin centre is exactly parallel to the shaft centre) and guides are fair to each other; if not, the guides will require to be lined up to correspond.

3. To Test for Wear of Crosshead Shoes.

This can generally be done by means of feelers placed between the crosshead shoe and the guide face at top and bottom positions of



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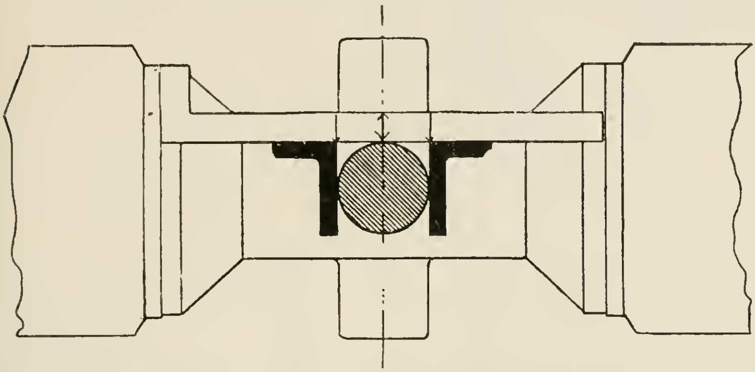
NOTE.—As the guides are closer together at top than at bottom when cold (about $\frac{20}{1000}''$ or so) it is advisable to make the test before the engines are cooled down otherwise due allowance requires to be made for the difference mentioned. If on testing the rod in stuffing box it is found not to be in dead centre, then the rod and piston will not be working truly in the cylinder bore, and of course the other test will then be unnecessary.



No. 8c.—Method of Locating Centre of Piston Rod between Guides.

The sketch (No. 8*c*) shows another method of finding a centre from which to test the fairness of the guides to the piston rod line. By means of the squares shown the centre of the piston rod can be exactly located as indicated by the mark. If now a wire is fixed to, say, a small plate bolted on the gland flange, and is then led down to the shaft centre, this line can be used as a test for the guide face setting, or piston rod trueness of line.

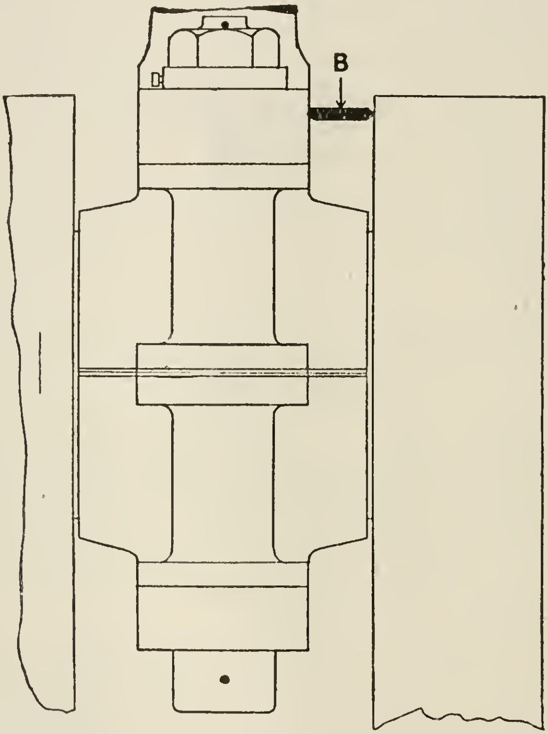
2. To Test if Guides are Parallel to Shaft Centre Line.

Turn the crank up to near the top centre and place a long steel straight-edge diagonally across the guide face (sketch No. 8*d*), with the lower end lying forward, and with a gauge measure the distance from the straignt-edge to the crank pin centre forward: next slope the straight-edge at the opposite angle on the guide face with the lower end lying aft, and with the gauge measure the distance between straight-edge and the crank pin centre aft: if the two measurements agree the shaft line (assuming, of course, that the crank pin centre is exactly parallel to the shaft centre) and guides are fair to each other; if not, the guides will require to be lined up to correspond.

3. To Test for Wear of Crosshead Shoes.

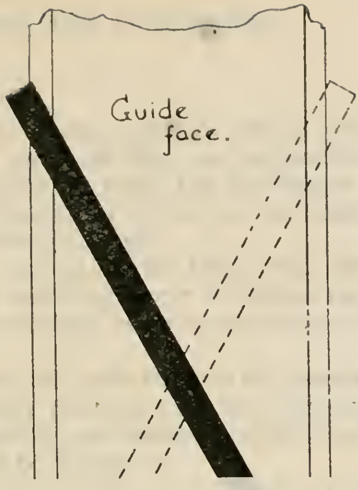
This can generally be done by means of feelers placed between the crosshead shoe and the guide face at top and bottom positions of crosshead travel. Due allowance must be made for the difference in distance between the guide faces top and bottom if the engines are cooled down when testing, and it is also necessary to be sure that to begin with the guides are parallel to the rod as described in test No. 1. The shoes can then be lined up as required.

4. To Test if Cylinder Line is at Right Angles to Shaft Centre Line

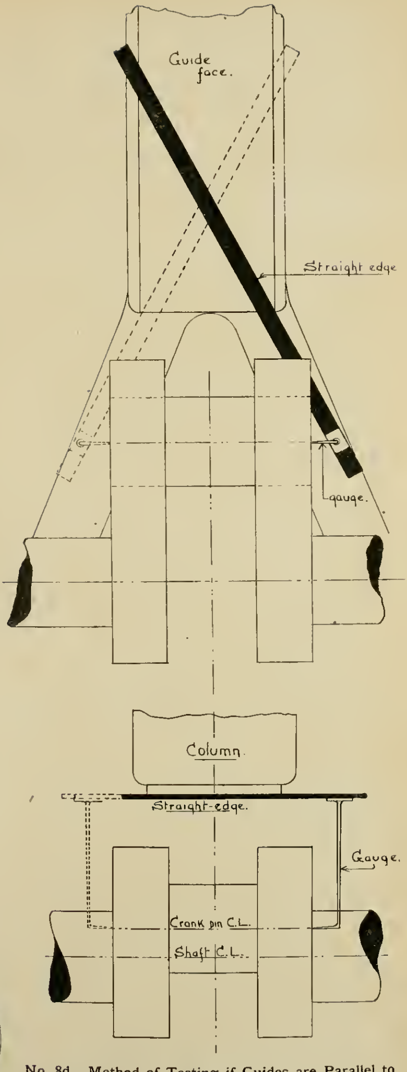


No. 8e.—Method of Testing Setting of Shaft Line and Cylinder Line.

To test if the centre line of the cylinders is exactly at right angles to the centre line of the shaft, slacken back bottom ends, place crank on top, and measure the distance between the butt of the connecting rod and the crank cheek, as shown at B; now place crank on bottom centre and again measure between butt of rod and crank; and the two distances will be the same, if the cylinders and shaft centre lines are set fair (at 90° to each other).



No. 9.—Bucket Air Pump.



No. 8d.—Method of Testing if Guides are Parallel to Shaft Centre Line.

[To face page 290d.

Pumps.

The suction valve of a feed pump should be placed low down on the barrel, and the delivery valve as high up as possible. This arrangement allows of better working when the feed-water temperature increases, as the air or vapour will rise clear of the pump suction, and allow of a better vacuum being formed.

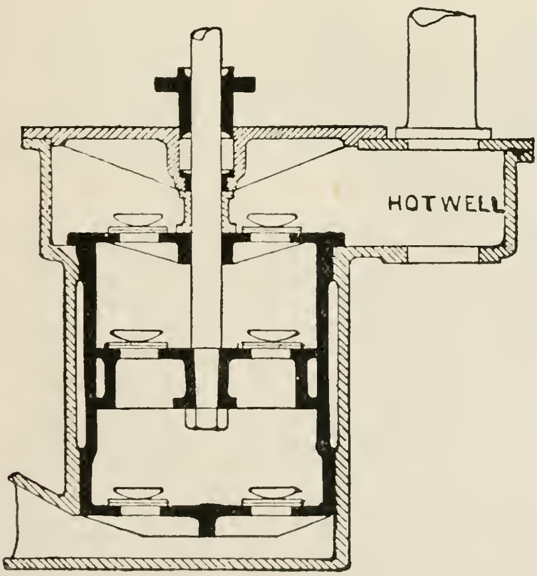
Most patent feed donkey pumps are not fitted with relief valves, the reason being that should anything occur to choke the delivery valves, the pump would stop working by over-pressure on the water side.

With feed pumps worked off the main engines the case is different for if, say, the delivery valve seat rises up with the valve, the pumps would of course still go on working, and unless a relief valve is fitted, the chest or connections would be damaged.

The feed relief valve should be placed on the pump chest between the suction and delivery valves, so that, should the delivery valve seat lift up with the valve, the relief will act and prevent damage to the chest. If the check valve on the boiler stuck or was left shut, the relief valve would also act and save the feed pipe from bursting.

In an ordinary feed pump, it should be noted that, with the chest cover off, the *top* of the suction valve is always open to the pump plunger, and the *bottom* of the delivery valve is open to the plunger.

A **Plunger** pump has suction and delivery valves, and is a single-acting force pump.



No. 9.—Bucket Air Pump.

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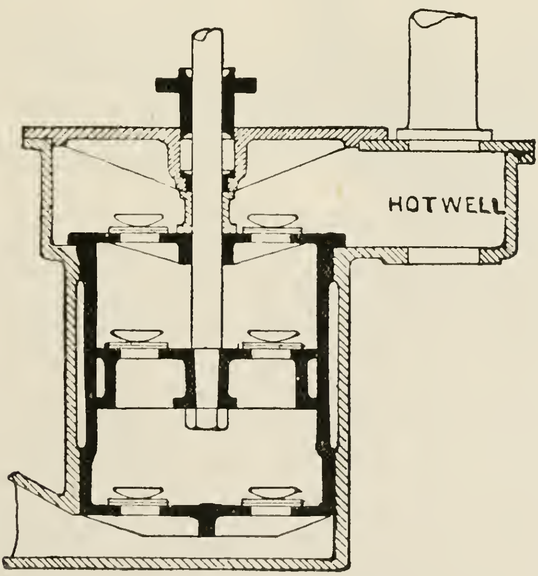


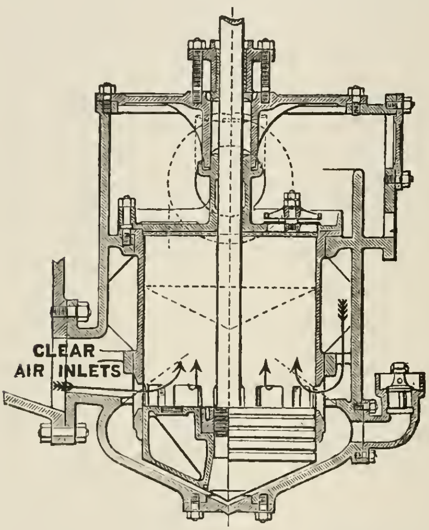
No. 9.—Bucket Air Pump.

A **Bucket** pump has foot, bucket and head valves, and is a single-acting lift pump.

Broken or leaky foot valves do not, in most cases, affect the vacuum, as the pump is placed much lower down in position than the bottom of the condenser, but the foot valves act to control the action of the pump and induce more regular flow. Broken or leaky bucket valves affect the vacuum most, and broken or leaky head valves next.

The Edwards Patent Air Pump.

In the Edwards air pump, as will be seen in the sectional illustration, foot and bucket valves are dispensed with; the water flows continuously *by gravity* into the base of the pump, and being dealt with mechanically by the conical bucket working in connection with a base of similar shape, is projected silently and without shock at a high velocity through the ports into the barrel.



No. 10.—Edwards Air Pump.

Packing for Gland of Edwards Air Pump.

1. At bottom of gland place two rings of soft packing of, say, hemp or cotton, which should be occasionally soaked in hot tallow with graphite.

2. Above this fit in a solid white metal ring with grooves turned in the bore for water packing.

3. Above this ring finally place a flat ring of rather stiff soft packing.

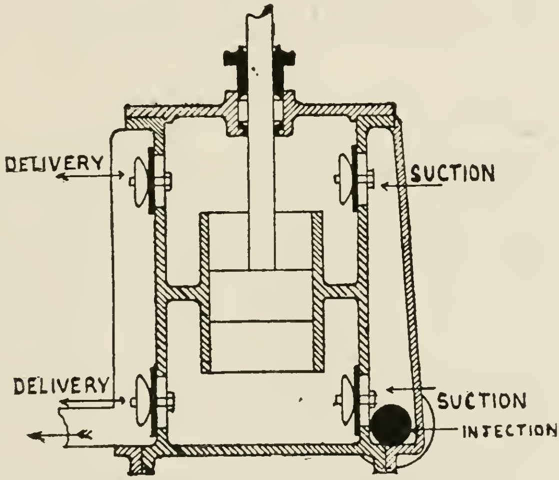
When running see that a supply of fresh water is kept on top of gland to fill up the grooves of the middle packing ring.

As soon as the ports open, there are clear inlets for the admission of the air, and the water is immediately afterwards injected, thereby tending to compress the air in the barrel and carry in more air with it.

Another important feature of the Edwards pump is that the top clearance is reduced to a minimum. Before any air pump can discharge, the pressure in the pump must exceed that of the atmosphere, and thus all air remaining in the pump is compressed. As soon, however, as the bucket descends, and the pressure is reduced, the air in the clearance water is given off, and expanding, occupies space in the pump which should be available for a fresh supply from the condenser, consequently the smaller the top clearance the more efficient is the pump.

The practical advantages of the Edwards pump are numerous; there being no foot and bucket valves, the risk of breakdown and stoppage through the failure of valves, which cannot be examined while the pump is at work, is eliminated, and there being only one set of valves instead of three sets, the cost of maintenance and the time necessary for overhauling are reduced to a minimum.

By means of the door at the top of the pump the only valves used can be readily examined, and if necessary can, in some cases, be renewed while the engines are running, without loss of water or vacuum.



No. 11.—Double-acting Circulating Pump.

A **Piston** pump has suction and delivery valves at either end, and is a double-acting pump. On the up stroke the bottom suction and top delivery valves are open, and on the down stroke the top suction and bottom delivery valves are open. Air-valves are usually fitted to either end of the pump to admit air to allow of cushioning of the water.

A **Centrifugal** pump, as the name implies, works from the centre, and consists of a cast-iron chamber containing hollow vanes keyed to a spindle; a small engine coupled direct to the spindle rotates the vanes, and the water entering, by suitable passages, the pump casing at the centre, where the vacuum is created by the rapid vane rotation, travels through the hollow vanes and is delivered tangentially at the circumference of the vane circle: thus peripheral force is converted into pressure head. This type of pump has no valves.

The usual driving speed of the pump is from 180 to 220 revolutions per minute. If by mistake the engine is started to run in the wrong direction, the water will merely be churned by the impeller, and no effective discharge will take place. Sometimes a steam pipe and cock is fitted on the pump chamber to assist in starting by blowing through a jet of steam, which, on condensing, produces a partial vacuum. In place of this, the pump may require to be first "primed," that is, filled up with water; the air escaping meantime by a small air-cock on the top. In ordinary practice, however, neither of the above aids to starting are required, as the position of the pump being lower than the sea level, the water flows by the force of gravity into the pump chamber, the air being allowed to escape by the small cock fitted on top of the pump casing. This type of pump is of low discharge pressure and would not be suitable for, say, boiler feeding, unless two or more pumps of the same type were coupled up "in series" (as in coal mine practice), which arrangement would result in increase of water pressure.

Breakdown of Pumps.

If the **air pump breaks down,** feed the boilers by "Weir's" pump, which usually has a suction direct from the condenser.

If no such connection is fitted, remove the air pump bucket and valves, leave on the cover and rod, close the hot-well overflow pipe, and allow the water in the condenser to drain into the hot-well. Draw the feed from there by the main feed pumps or the donkey pump. The vacuum will of course go back in this case, and most likely disappear altogether.

If the **circulating pump breaks down,** put on the ballast donkey to the condenser for circulating; if it is not suitable for this, then the engines must be worked jet condensing.

To effect this draw a number of the condenser tubes, and open up the air pump discharge valve; also when under weigh again take the boiler density oftener, as the feed will be chiefly salt water.

To find number of tubes to draw:—

$$\frac{\text{Injection pipe diameter}^2}{\text{Condenser tube diameter}^2} = \text{Number of tubes.}$$

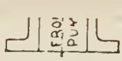
Loss of Vacuum.

Vacuum may be lost through the following causes:—

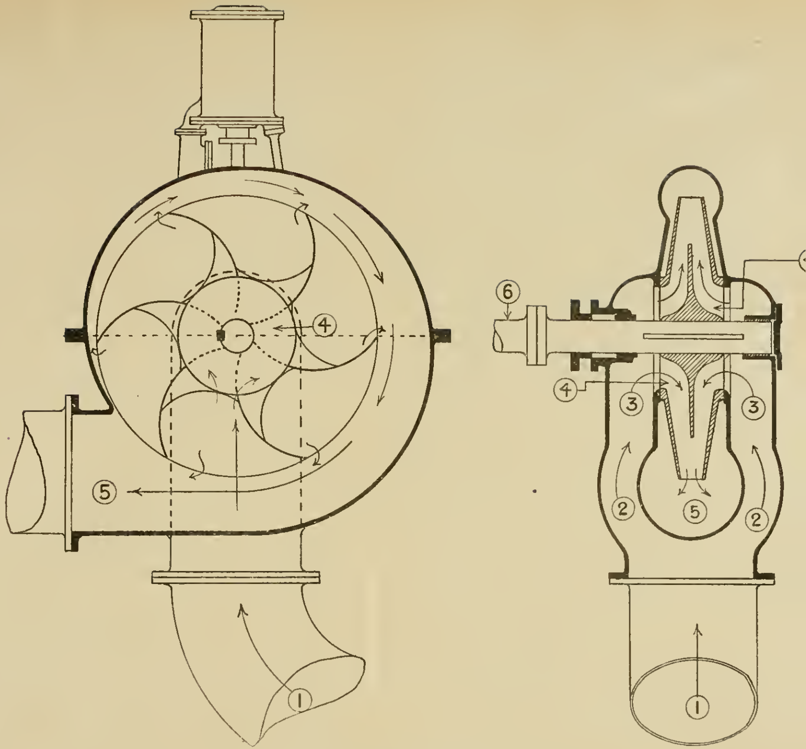
1. Head or bucket valves of air pump broken.
2. Valves of circulating pump broken, or injection choked.
3. Division plate in condenser door carried away.
4. Leaky L.P. gland.

To find the probable cause of the vacuum going back, feel the temperature of both ends of the condenser. If both ends are cold, broken air pump valves are the cause, or leaky L.P. gland. If both

No. 13.—Closed Type Air Vessel.



No. 14.—Open Type Air Vessel.



No. 12.—Centrifugal Type Circulating Pump.

1, Inlet from main injection.
2, Water flow to each side of pump.
3, Water inlet to pump at centre.
4, Vacuum position.
5, Delivery at periphery of pump
6, Driving shaft of engine.

DATA.

Diameter in inches of suction or delivery pipe $= \sqrt{\dfrac{\text{I.H.P.} \times \cdot 045}{\cdot 7854}}$.

NOTE.—Allow ·045 square inch of pipe area per I.H.P.
Diameter of impeller = Diameter of pipe × 2·5.
Diameter of inlet opening at centre = Diameter of pipe × 1·1.

EXAMPLE.—Determine the diameter of circulating pump suction and delivery pipe, the diameter of the impeller, and the diameter of inlet opening at pump centre for an engine of 1800 I.H.P.

Then, Diameter of circulating pipe $= \sqrt{\dfrac{1800 \times \cdot 045}{\cdot 7854}} = 10$ inches,

and, Diameter of impeller $= 10 \times 2{\cdot}5 = 25$ inches,

also, Diameter of inlet at centre $= 10 \times 1{\cdot}1 = 11$ inches.

[To face page 294.

ends are warm, either broken circulating pump valves or choked injection valve is the cause. If one end is cold and the other end warm, the division plate in the condenser is most likely carried away.

NOTE.—24 inches of vacuum means that the air pump has drawn 12 lbs. of air pressure out of the condenser, leaving 3 lbs. absolute back pressure on the L.P. piston.

Pet Valves.

On feed or bilge pumps the pet valve is usually placed between the suction and delivery valves.

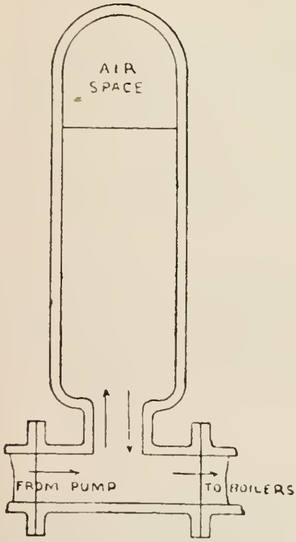
On double-acting circulating pumps a pet valve is placed on the suction side of the pump at both ends.

On air pumps the pet valve is placed high up on the pump chamber, just under the head valve, so that the air drawn in for cushioning the water will not affect the vacuum under the bucket. Many air pumps have no pet valves fitted.

On horizontal double-acting piston air pumps no pet valves are fitted, as, no matter where they might be placed, the air drawn in would affect the vacuum more or less seriously, since suction valves are fitted at both ends of the pump.

Air Vessels.

Air vessels are usually fitted on single-acting pumps, to give a continuous flow of water similar to that of a double-acting pump.



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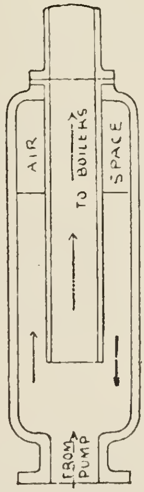

No. 14.—Open Type Air Vessel.

To find the probable cause of the vacuum going back, feel the temperature of both ends of the condenser. If both ends are cold, broken air pump valves are the cause, or leaky L.P. gland. If both ends are warm, either broken circulating pump valves or choked injection valve is the cause. If one end is cold and the other end warm, the division plate in the condenser is most likely carried away.

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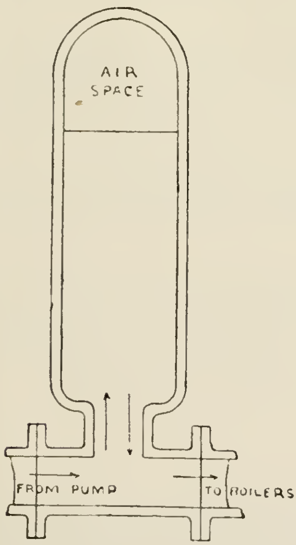

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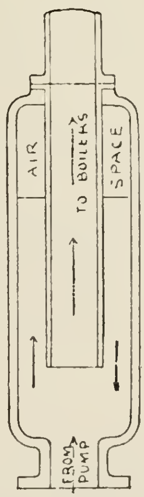


No. 14.—Open Type Air Vessel.

They are made in two ways—(1) A plain dome with the air for compression in the top, the water entering and leaving by the same branch. (2) A dome open at the top, with a pipe extending down about three-quarters of the length of the vessel; the pipe is to allow of an air space round it, and the water in this case passes up through the pipe and top of vessel into the main feed pipe.

On the down stroke of the pump plunger the water pressure compresses the air in the top of the vessel, and on the up stroke, the pressure being released, the air reacts on the water, and forces it out of the chamber and along the feed pipe, thus giving a continuous flow, similar to that obtained by a double-acting pump.

NOTE.—The Board of Trade appear to object to cocks or connections of any kind being fitted to air vessels.

The Condenser and Air Pump.

The function of the air pump is to draw out the air and water from the condenser, and, by the vacuum formed, to reduce the back pressure on the L.P. piston.

As all water contains a certain proportion of air, the feed pumps deliver air into the boilers along with the feed water, and most of the air so brought in is carried back with the steam to the condenser, where it remains in the form of back pressure. Air cannot be condensed in the same manner as steam, and for this reason the air pump is required to take out the air and reduce the back pressure by the formation of a vacuum in the condenser.

The valves most necessary in the air pump to maintain the vacuum are the bucket and head valves; foot valves are not absolutely necessary when the pump is placed lower down than the bottom of the condenser, as the water will then flow by gravity into the pump chamber.

The air pump only delivers about $\frac{1}{2}$ inch depth of water over the valves each stroke, the remainder of the pump being filled with air and vapour.

On the up stroke of the air pump the foot valves and head valves are open, and on the down stroke the bucket valves only are open.

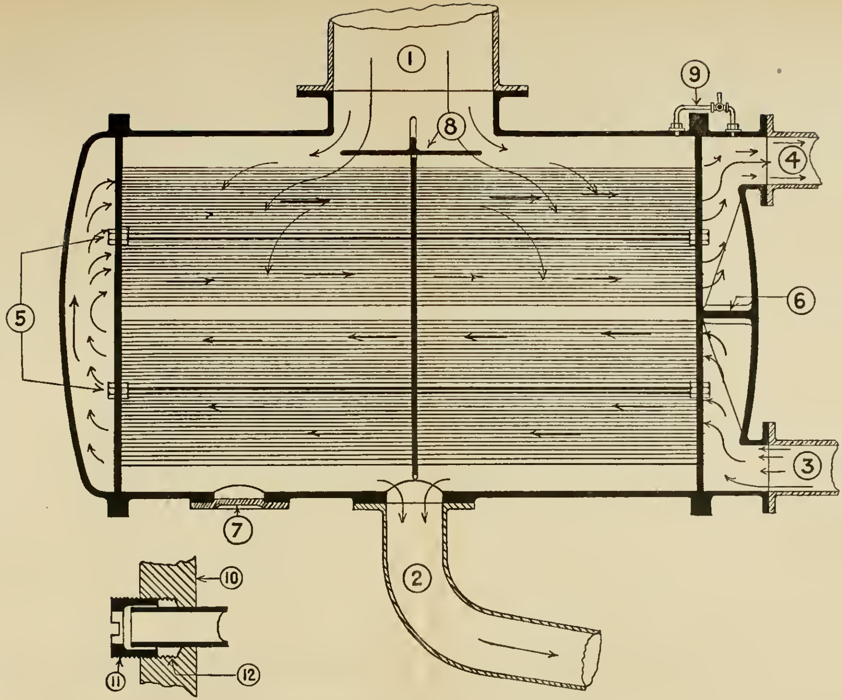
If the vacuum gauge indicates 24 inches, it means that the air pump has taken 12 lbs. of air pressure out of the condenser, which leaves a back pressure of 3 lbs. gross, as $15 - 12 = 3$ lbs.

Oscillating Engines.

The steam enters the valve chest through one of the trunnions (usually the outer), and after doing its work in the cylinder exhausts through the other trunnion.

The trunnions allow the steam to pass to and from the valves, and also allow for the oscillation of the cylinder.

balanced by a slight increase in the quantity of circulating water passing through the tubes, which ensures a more rapid flow. The steam flow through the condenser is also more direct and more convergent from top to bottom than in the usual pattern of condenser as fitted.



No. 15.—Surface Condenser (Two Flow Independent Type).

1, Exhaust or eduction pipe from L.P. cylinder to condenser.
2, Air pump suction.
3, Delivery from circulating pumps to condenser.
4, Circulating discharge from condenser to ship's side.
5, Stays.
6, Division plate.
7, Hand hole.
8, Baffle plate for steam.
9, Supplementary feed.
10, Tube plate.
11, Screwed ferrule.
12, Packing space.

DATA.

Cooling Surface.

I.H.P. = 1800.
Allow 1.4 square feet cooling surface per I.H.P.
Tubes, 10 feet by $\frac{3}{4}$ inch external diameter.

Then, number of tubes $= \dfrac{1800 \times 1{\cdot}4}{10 \times \frac{{\cdot}75}{12} \times 3{\cdot}1416} = 1283$ tubes.

NOTE.—The cooling surface of a tube in square feet = length in feet × circumference in feet.

Circulating Water.

Assume exhaust steam pressure = 5 lbs. absolute and temperature 162°, hot-well temperature 130°, sea 60°, and discharge 102°,

then,	$1115 + {\cdot}3 \times 162 = 1163{\cdot}6$ total heat of steam,
and,	$1163{\cdot}6 - 130 = 1033{\cdot}6$ units to be extracted.
Again,	$102 - 60 = 42$ units absorbed by each pound of cooling water,
then,	$1033{\cdot}6 \div 42 = 24{\cdot}6$ lbs. cooling water required per lb. steam.

Assuming 15 lbs. steam per I.H.P. per hour,

then, $1800 \times 15 \times 24{\cdot}6 = 664200$ lbs. of circulating water required per hour.

Air Pump Water.

Exclusive altogether of expanded vapour and neglecting losses, the air pump will lift per hour 27000 lbs. of feed water and discharge it into the hot-well, as $1800 \times 15 = 27000$ lbs.

[*To face page* 296

Condenser Data.

Position.	Temperature Degs. Fahr.
Sea (outside) - - - -	65°
Circulating water - - -	67°
Condenser (1st stage circulation) -	92°
Discharge (2nd stage circulation) -	122°
Exhaust - - - - -	152°
Hotwell - - - - -	Top. 143° | Bottom. 140°
Filter - - - - - -	154°
Heater - - - - -	220° (pressure, 10 lbs.)

NOTE.—The heater coil drain to hotwell accounts for the rise in temperature shown at filter. Heater of the closed type with steam heating coils. After first passage through the tubes the circulating water temperature rose from 67° to 92°, and after second passage through tubes from 92° to 122° (discharge).

The Weir "Uniflux" Condenser.

A characteristic of the "Uniflux" condenser is its special contour, whereby the entering steam is caused to traverse the cooling surface at practically uniform velocity throughout its passage.

Provision is also made in the bottom of the condenser for ensuring equality of distribution, so that no short circuiting of any portion of the surface can take place. There are no baffle plates or partitions to interfere with the steam flow in the body of the condenser. The flow is therefore direct towards the bottom of the condenser from the exhaust inlet to the draft plate. As a result of the maintenance of the steam velocity the heat transmission value of the condensing surface becomes greatly increased, while by the directness of the flow the whole of the surface is uniformly operated, and no ineffective zones can exist.

As compared to the ordinary type condenser, the Weir "Uniflux" condenser has less cooling surface per H.P., but this is counterbalanced by a slight increase in the quantity of circulating water passing through the tubes, which ensures a more rapid flow. The steam flow through the condenser is also more direct and more convergent from top to bottom than in the usual pattern of condenser as fitted.

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The trunnions allow the steam to pass to and from the valves, and also allow for the oscillation of the cylinder.

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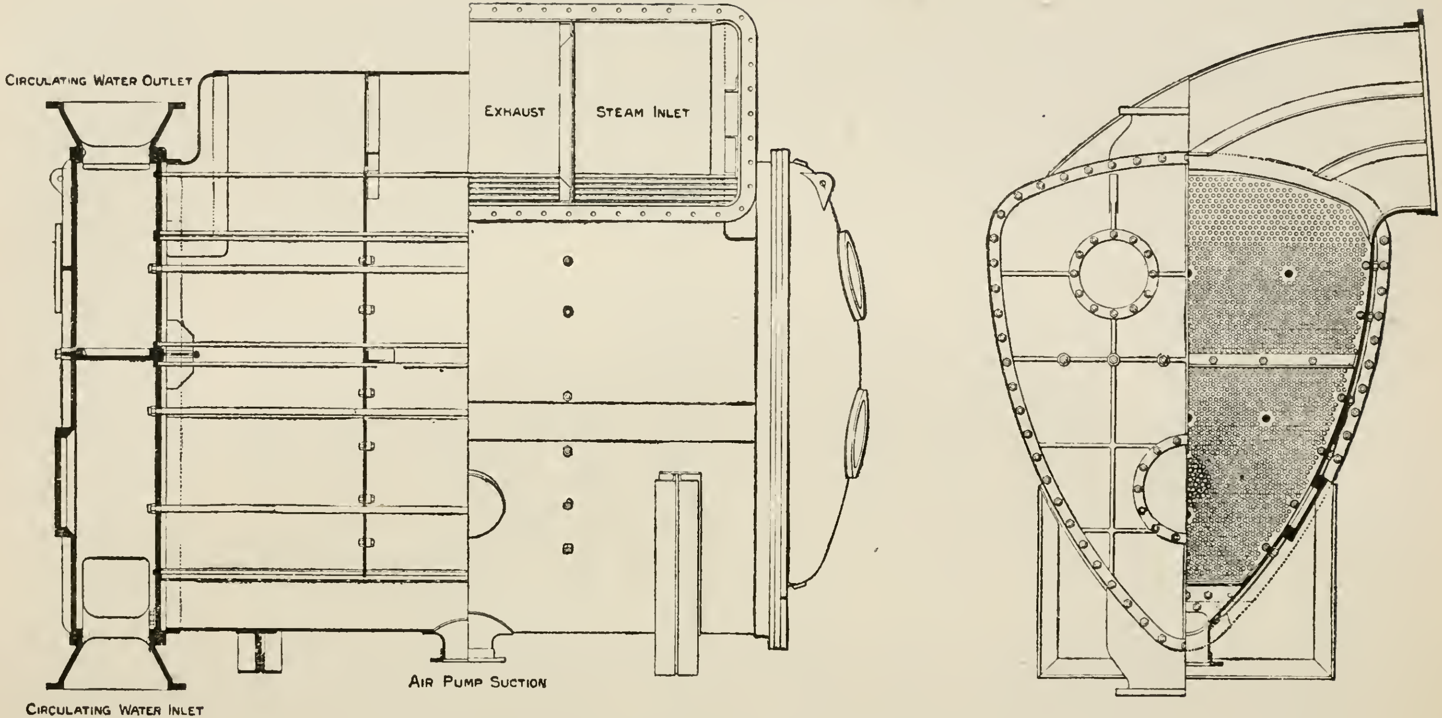
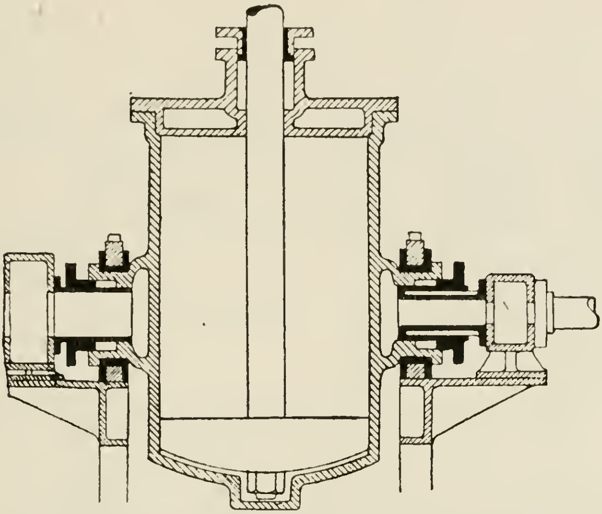


No. 15a.—Weir "Uniflux" Condenser.

An expansion joint is formed between that part of the steam pipe which enters the trunnion and the trunnion itself.

The flanges of the steam pipe are bolted to a bracket on the engine framing, as will be seen by referring to the sketch, and this does away with the necessity of having a collar on the pipe and long tuds, as are usually fitted to expansion joints of steam pipes.

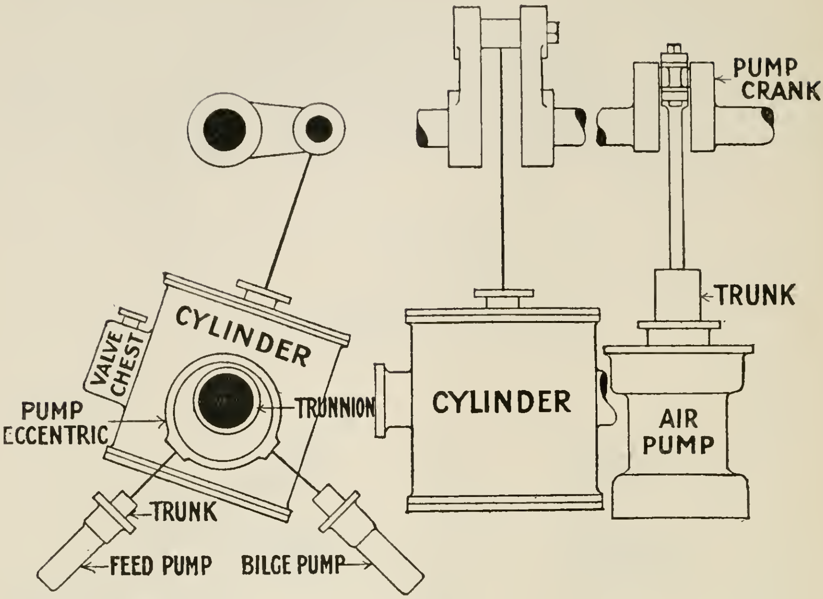


No. 16.—Oscillating Cylinder and Trunnions.

To test if one or other of the trunnion bearing brasses have worn down, slacken back the piston rod head bolts and measure the clearance between the crank web and piston rod brass at the top centre and the bottom centre; if the sizes do not agree one of the trunnions has worn down.

In oscillating engines the feed and bilge pumps are worked by large eccentrics, fixed either on the shaft or on the trunnions, and the air pump by an extra crank on the main shaft: to allow of this the pumps are usually of the trunk pattern (see sketches).

The valve gear of the oscillating engine (see sketch) consists of two vertical pillars, connecting the top and bottom framing, with a slotted quadrant working between them on brass shoes. The reversing link from the eccentrics connects to a block on the quadrant. The radius of the slot in the main quadrant is taken from the centre of the trunnion with the gear at half position, the radius of the reversing link slot is taken from near the shaft centre. Usually two valves are fitted, one on either side, each supplying steam simultaneously to the cylinder or exhausting simultaneously from the cylinder. The block on which the reversing link slides is a fixture, it should be

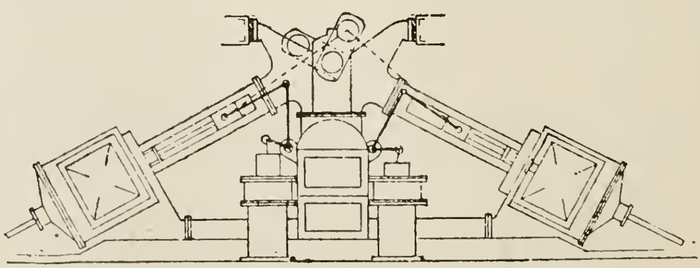


No. 18.—Method of working Pumps in Oscillating Engines.

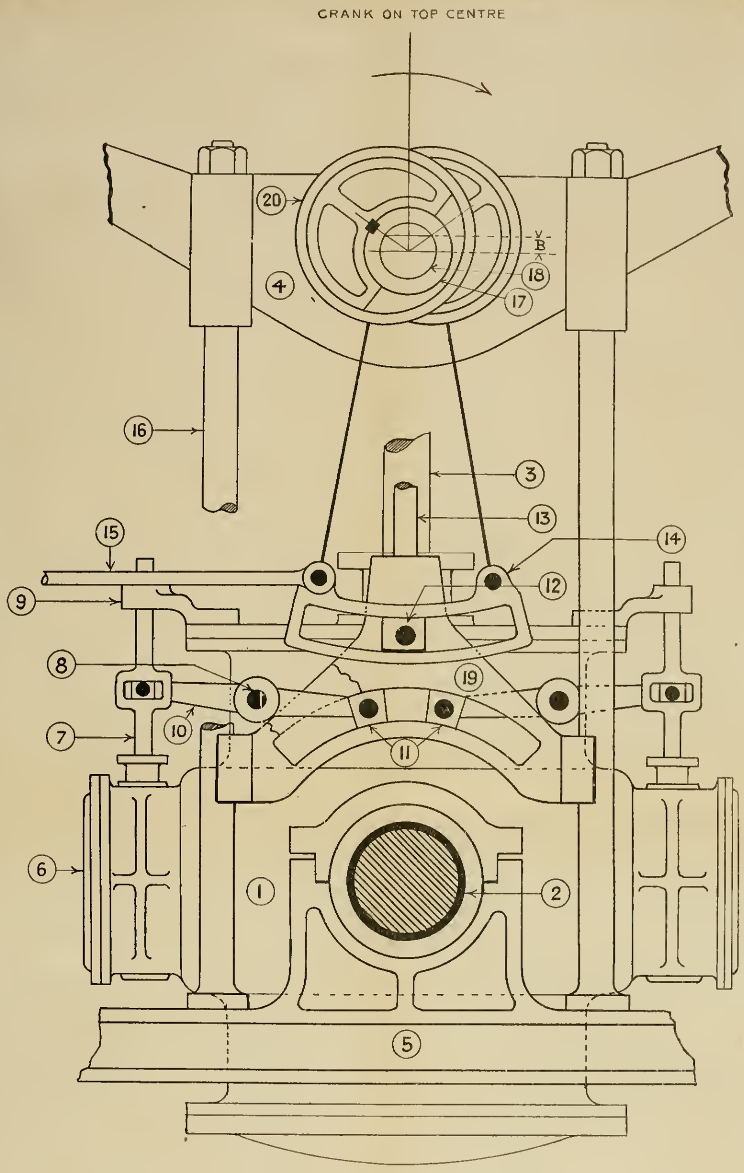
noted, to the main quadrant, and the drag link changes the position of the link and rods from ahead to astern as required. A rocking lever works on a pin fixed on the *side of the cylinder*, and one end of the lever moves in the slot of the quadrant as the cylinder oscillates; the other end is connected to the valve spindle, and so gives the necessary travel to the valve, but in the opposite direction.

NOTE.—For a slide valve, the eccentric key centre is placed behind the crank at an angle of 90°, minus lap and lead, because the rocking lever reverses the motion, but if a piston valve (inside steam), the key is cut at an angle of 90° plus lap and lead before the crank, as the one position corrects the other.

Diagonal Engines.



No. 19.—Diagonal Engine and Air Pumps.



No. 17.—Oscillating Engine and Valve Gear Complete.

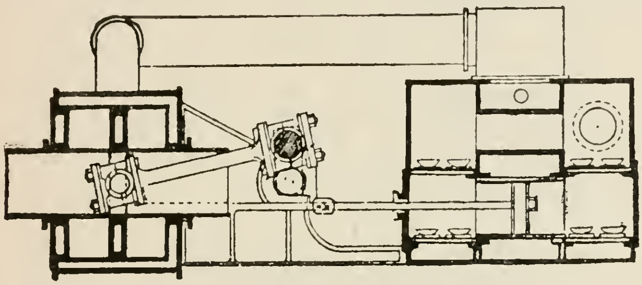
1. Cylinder belt through which the steam passes from the admission trunnions to the valve chest, or through which the steam exhausts from the valve chest to the exhaust trunnions.
2. Trunnions on which the cylinder oscillates and which also serve for admission or exhaust of the steam.
3. Piston rod connected direct to crank pin, no connecting rod being fitted.
4. Engine top framing containing the shaft bearings.
5. Engine bottom framing.
6. Valve chest.
7. Valve rod.
8. Pin fixed to cylinder side and on which the rocking lever moves.
9. Guide bracket for valve rod.
10. Rocking lever, one end of which slides to and fro in the main quadrant with the cylinder oscillations and the other end of which is connected to the valve spindle by a movable joint, thus conveying the eccentric travel to the valve, but reversed in direction.
11. Moving blocks of rocking levers.
12. Fixed block on main quadrant and on which the reversing link slides by means of the drag link.
13. Tail guide rod of main quadraot.
14. Reversing link.
15. Drag link.
16. Vertical pillars or columns on which work the shoes of the main quadrant
17. Shaft.
18. Valve travel circle.
19. Main quadrant.
20. Ahead pulley. B = steam lap and lead.

[To face page 298.

In diagonal engines the pumps are often worked by bell crank levers connected to the piston rod crosshead (see sketch), and the pumps are usually made of the trunk type.

Trunk Engines.

A trunk engine has no piston rod, but simply a connecting rod extending between the crank-pin and the piston. The trunk passes



No. 20.—Trunk Engine and Double-Acting Air Pump.

through both ends of the cylinder, and is bolted by a flange to the piston. For a right-handed propeller this type of engine is placed on the starboard side of the engine-room, so that when going ahead the stress will be thrown on the top side of the trunk and piston; for a left-handed propeller the engine would be placed on the port side to obtain the same result.

The air pump shown on the sketch is of the double-acting type, and has foot and head valves at either end, and a solid piston; the condenser suction is below, and the hot-well above. When the engines are of the inverted type, the pump is worked from the main shaft by an eccentric, or by a pin on the crank web.

Paddle Engine Cranks.

In nearly all paddle engine crank-shafts, the crank-pin is only fixed to one web, and is an easy fit in a brass bush in the other web (see sketch). This is to allow for the extra wear which takes place at the outside bearing due to the weight of the paddle wheels. The crank-pin is fitted into one web by a taper and nut, and in the other web it is simply a loose fit in a brass bush as previously stated.

This arrangement allows for the shaft wearing down outside, and prevents undue stresses being thrown on the web and on the pin.

In ordinary paddle engines with double cranks the crank-pin is fixed to the inner web and is loose in the outer web, but in disconnecting paddle engines the reverse is the case, that is, the pin is fixed to the *outer* web, and is loose in the inner one. This is to allow

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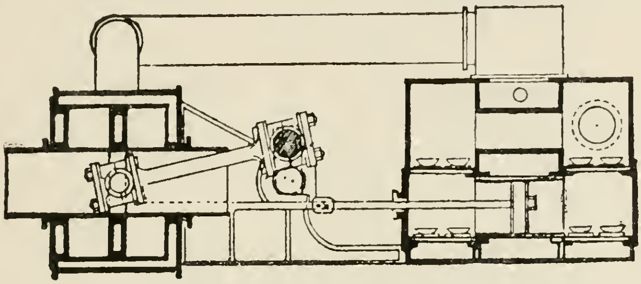

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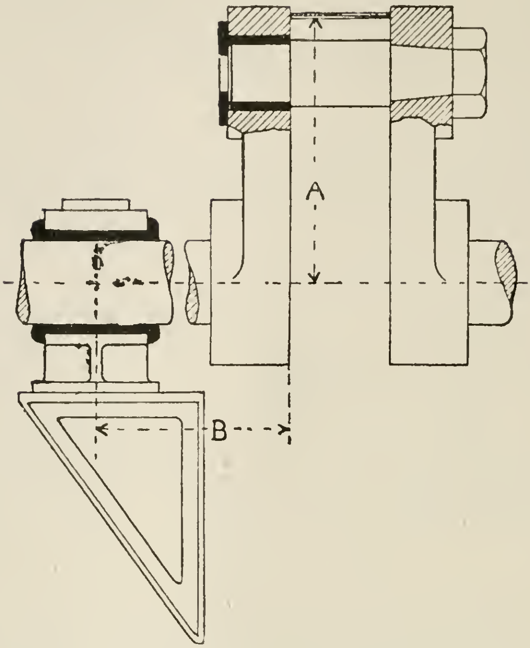


No. 21.—Method of Testing Fairness of Paddle Shaft.

of the inside crank being drawn away from the pin when the engines have to be disconnected.

From the above it will easily be understood that to test if the outer bearing is wearing down, the cranks must be placed on the top centre, and the distance between the webs at the top taken by a length stick; then the cranks put on the bottom centre, and the same distance again measured; if it is found that the top width is greater than the bottom, the outer bearing will be down.

To find Thickness of Liner for Outer Bearing.

Referring to the sketch, suppose that the distance between the webs is 1 inch more at the top than at the bottom, and the distance from the top of the web to the centre of the shaft is A, and from the centre of the outer bearing to the inside of the web is B, then by proportion as follows:—

As $A : B :: \frac{1}{2}$ inch = thickness of liner required.

NOTE.—Observe that it is only half the amount that the webs are open at top more than at bottom, which is taken for the third term.

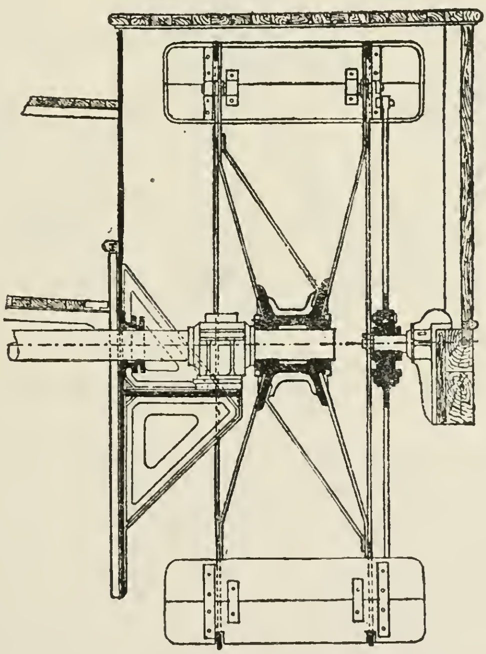
Paddle Shaft Flexible Coupling.

If the crank-shaft of a paddle engine is made similarly to that of a screw steamer, that is, with the webs and pins forged solid, or built up, allowance for wear down is arranged for by a flexible coupling fitted between the paddle shaft and the crank-shaft (see page 325).

The coupling consists of a hard rubber ring fitted between the two flanges with the coupling bolts an easy fit in one side, the holes being made larger for that purpose. This allows for the outside bearing wearing down, and prevents the pin and webs from being subjected to heavy stresses in consequence.

Feathering Paddle Wheel.

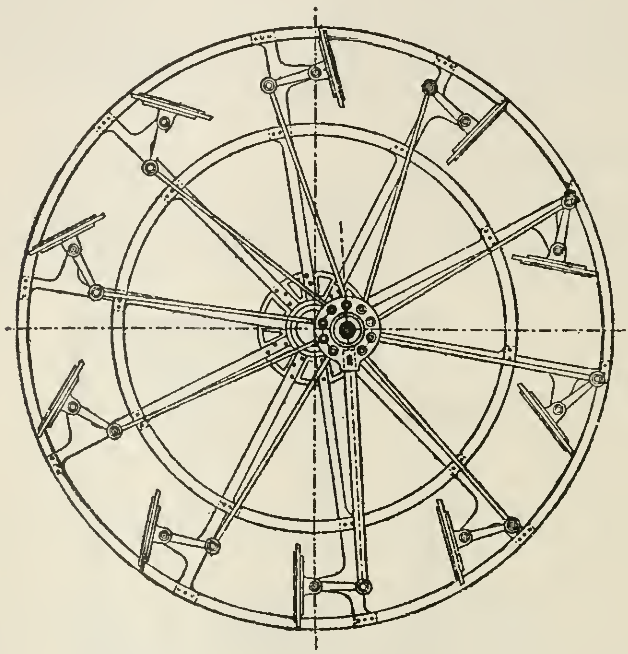
In a feathering paddle wheel the floats are hung by pins to brackets on the circumference of the wheel, the brackets being bolted to the rim and forming continuations of the paddle arms.



No. 22.—Feathering Paddle Wheel.

The floats have small feathering levers (see sketch) fitted on the back, to which the eccentric rods are connected by pins in brass bushes, the other end of the rods being fitted in a similar manner to the feathering strap.

The feathering pin is fixed to a bracket bolted to the outer paddle box, and its centre is placed forward of the shaft centre and higher up so that the pin position is "eccentric" to that of the shaft; the feathering strap revolves round the pin which is stationary. One rod, called the "driver," is made of greater strength than the others, and,

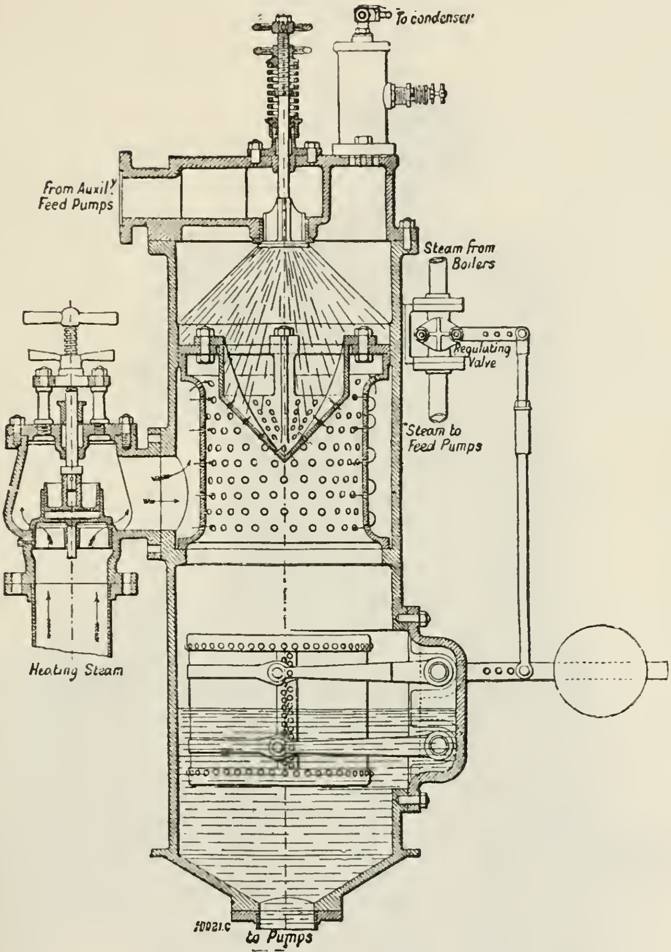


No. 23.—Feathering Paddle Wheel.

instead of being secured to the strap by an eye and pin, is fitted in to a specially arranged recess and firmly bolted there; the other end of this rod is connected to the float in the same manner as the other rods: the driving rod gives the motion to the feathering strap, and therefore to the other rods. The feathering pin being eccentric to the shaft, the floats are feathered as the wheel turns round, that is, they alter their angle so as to enter the water in a vertical position, and thus obtain a good thrust, and leave it at an inclined position, which prevents loss of power by water being lifted up.

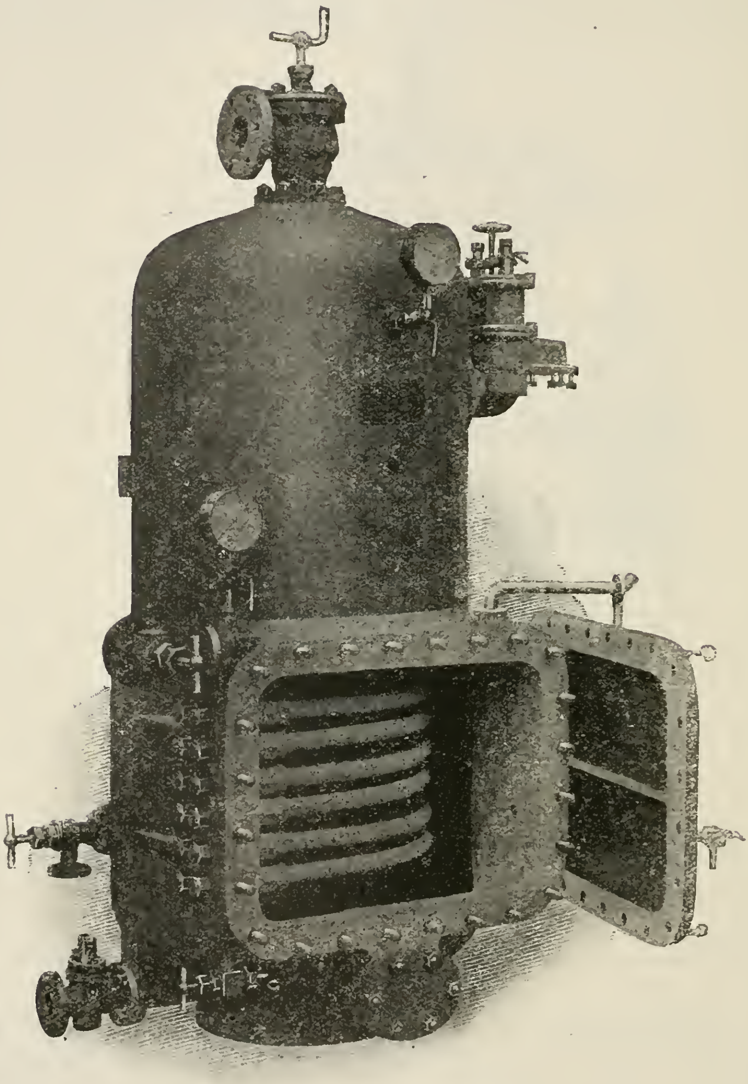
The various pins mentioned are brass bushed, and lubricated with water.

Engine-Room Appliances.

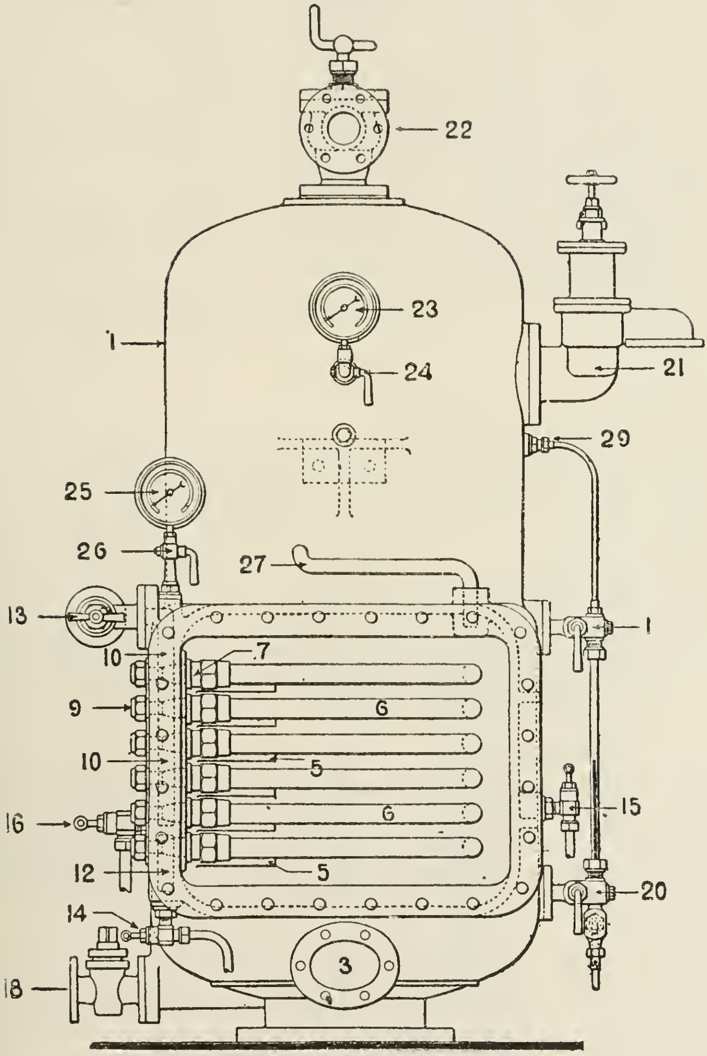


No. 24.—"Weir" Feed Heater

In this well-known type of feed-water heater, the heating steam is taken from the low-pressure receiver of the main engines and exhaust of the auxiliary engines, and enters the heater through a non-return valve on the side; the feed water is forced up into the top of the heater by the main feed pumps on the engine, and, the pressure of the water overcoming the tension of the spring, forces open the internal valve and allows the water to enter the body of the chamber, through a perforated ring in the form of a fine spray, which,



No. 25.—Weir Vertical Type Evaporator.



No. 26.—Weir Vertical Type Evaporator.

meeting the heating steam entering by another set of perforations becomes raised in temperature to that of the steam: the air present in the water is set free by the heat, and rising, escapes from the heater by a cock on the top to either the condenser or the atmosphere. The float shown near the bottom of the heater is connected by a system of levers to the steam stop valve of "Weir's" pumps, which take away the hot feed water and deliver it into the boilers, and as the water level in the chamber falls, the float sinking shuts off the steam supply to the pumps, so that they work slower, and, on the other hand, when the water level is high, the float rising correspondingly opens the steam valve, and the pumps work faster in proportion.

Two pressure gauges are fitted on the top of the heater, one to indicate the water pressure entering, and the other to indicate the pressure in the body of the chamber, upon which latter depends the temperature of the hot feed water.

NOTE.—With a pressure of 5 lbs. in the L.P. receiver, the feed temperature in the heater will be about 220°.

Advantages of Feed Heating.

The practical advantages of feed heating are as follows:—

1. Less straining of the plates in high-pressure boilers.
2. Less air enters the boilers, and consequently the corrosion due to oxygen is reduced.
3. The boilers steam better with hot feed water, as circulation is checked when cold water enters a boiler.
4. The heat of condensation of the steam is given up to the feed water instead of being given to the circulating water and lost over the side, as would be the case if the steam condensed in the condenser.

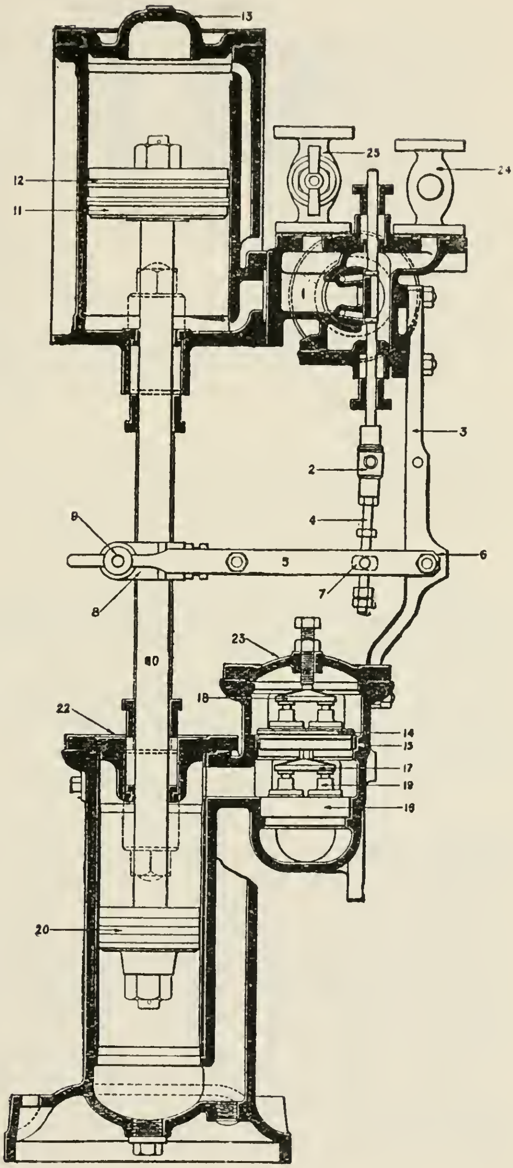
"Weir" Evaporator.

The Weir Vertical Type Evaporator is a single casting of close-grained cast iron. The heating surface is formed of heavy solid drawn copper tubes attached by hollow conical couplings, at one end to the steam inlet chamber cast on the side of the evaporator, and at the other end to the corresponding outlet chamber cast alongside it.

The tube space is separated from the steam space by a deflector. The arrangement of this deflector allows the steam to rise, but throws down the water so that it is returned again to the water space. With this arrangement it is almost impossible to make the evaporator prime under anything like reasonable limits.

The usual mountings consist of steam inlet valve, steam outlet valve, feed check valve, brine valve, drain valve and coupling to hot-well, blow-off cock, safety valve, gauge glass fittings, pressure gauge, compound gauge, salinometer valve, also gun-metal feed pump to work off main engines.

1, Steam Slide Valve Chest.
2, Double Joint.
3, Front Stay.
4, Bottom Spindle.
5, Valve Gear Levers.
6, Front Stay Bush.
7, Ball Crosshead.
8, Main Crosshead.
9, Crosshead Pin.
10, Piston Rod.
11, Piston Body.
12, Piston Rings.
13, Cylinder Cover.
14, Discharge Valve Seat.
15, Discharge Valve Seat Ring.
16, Suction Valve Seat.
17, Suction Valve Guard.
18, Discharge Valve Guard.
19, Water Valves.
20, Bucket.
22, Pump Cover.
23, Valve Chest Covers.
24, Steam Stop Valve.
25, Exhaust Stop Valve.



No. 27.—Sectional View, Weir Standard Feed Pump.

List of Parts of Evaporator.

1. Shell of Evaporator.
2. Main door of Evaporator (1) for withdrawing tube coils (6).
3. Hand cleaning door.
4. Baffle plate, or deflector.
5. Shelves for supporting tube coils (6).
6. Evaporating tube coils.
7. Inlet steam couplings for coils (6).
8. Drain outlet couplings for coils (6).
9. Coupling nuts for 7 and 8.
10. Inlet steam header.
11. Drain header.
12. Drain collecting pocket.
13. Inlet valve for steam coils (6).
14. Valve for drain from coils (6) to hot-well.
15. Feed check valve.
16. Brine valve.
17. Salinometer cock.
18. Cock for blowing off to sea.
19. Top cock for water gauge.
20. Bottom cock for water gauge.
21. Safety valve.
22. Outlet valve for generated steam.
23. Compound gauge for generated steam in shell (1).
24. Cock for compound gauge (23).
25. Pressure gauge for inlet steam to coils (6).
26. Cock for pressure gauge (25).
27. Swing crane bar for door (2).
28. Eye bolt for supporting door (2) on crane bar (27).
29. Connection from top of water gauge (19) to steam space in Evaporator shell (1).

Weir's Patent Direct-Acting Feed Pumps.

(To work in connection with Main Feed Pumps and Heater.)

The Weir pump is of the direct-acting type, and has suction and delivery valves for top and bottom independently.

The pump is vertical, single cylinder, and is usually supplied in pairs.

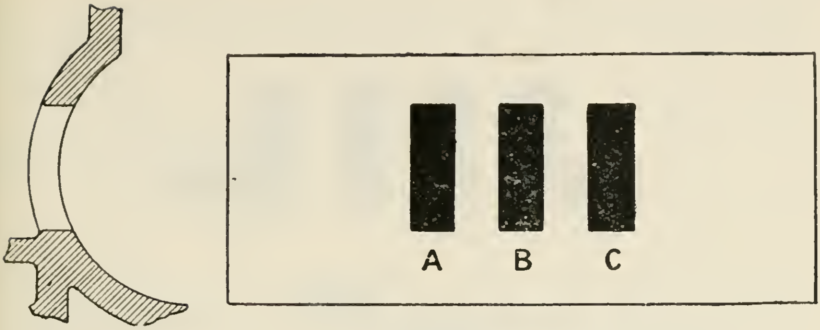
The steam piston is fitted to the top end of the rod, and the water piston to the bottom end, the latter being smaller in area than the former.

Weir's feed pump is a slow-speed, high-pressure, full-stroke pump.

Group Valves.—These are milled out of the solid metal, and are arranged to give a large area with a small lift. Each seat contains a number of small valves, and in all cases these are duplicate, with a lift of $\frac{1}{4}$ inch. The delivery valves have light springs fitted. The suction seat contains a *larger* number of small valves than the delivery seat; it will therefore be noted that the delivery valves have *less* area than the suction valves, and in addition have small springs fitted to keep them down.

The screwed pin in the centre of the valve box covers is for keeping the valve guards in position, and is not for regulating the lift of the valves.

The Weir Steam Valves.—The steam valves of Weir's pumps are simple in action, and the main valve face (known as the "shuttle" valve) is half round instead of flat, and instead of travelling up and down is moved by steam horizontally from side to side. The ports



End View. **Cylinder Face (Half Round).**

A, Steam Port to Cylinder Bottom. C, Steam Port to Cylinder Top.
B, Exhaust Port.

No. 28.—Weir Pump Cylinder Ports.

are therefore arranged to allow of this, and are cast side by side in place of one above the other. The result is, of course, the same, as the left-hand port leads to the bottom of the cylinder, and the right-hand port to the top (see sketch).

Main Valve.—As before stated, this valve is moved by steam horizontally in the chest, and in this way opens up the cylinder ports from steam to exhaust at the end of each stroke. It must be remembered, however, that previous to this the steam passing through the main valve into the cylinder ports has already been cut off by the expansion valve at three-quarters stroke. The ends of the main valve are round, and work in extended cylindrical casings at each side of the chest, the valve being moved across by steam alternately admitted and exhausted from the ends which act as pistons.

The main valve has two faces. That on the front contains four ports, two steam and two exhaust. The face on the back contains five ports (see sketch).

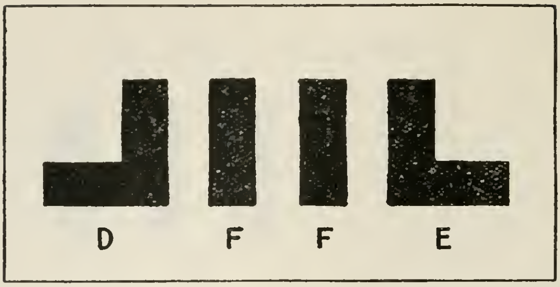
The port E leads from back to front and admits steam to the cylinder bottom by port A.

The port D leads from back to front and admits steam to the cylinder top by port C.

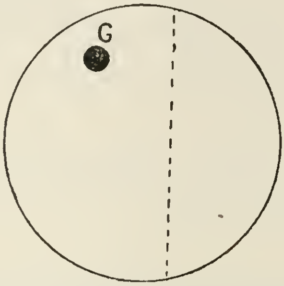
The port G admits steam to, or allows of exhaust from, the *left* side of the chest in which the round end of the main valve works steamtight.

The port H admits steam to, or allows of exhaust from, the *right*-hand side of the chest in which the round end of the main valve works steamtight.

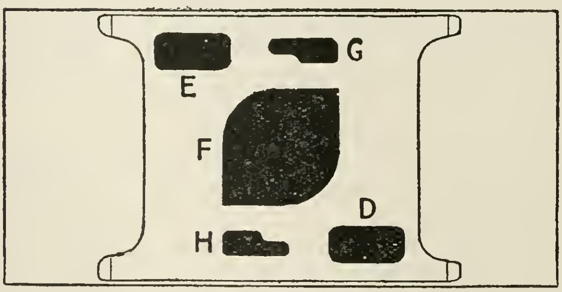
The centre exhaust port F is common to all the ports. Observe that port G leads to the left-hand end, where a small hole allows the steam to pass out and act on the piston end of the main valve to



No. 29.—Main (Shuttle) Valve Front Face (Half Round).



No. 30.—End View (Left).



No. 31.—Shuttle Valve Back Face (Flat).

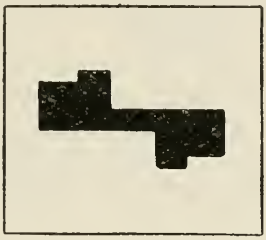
force it across; it also allows of exhaust to take place from that end to the exhaust port F by means of the small auxiliary or expansion valve.

Auxiliary Valve.—This small valve works vertically on the back of the main valve, and admits steam to the ports E D and G H alternately, or allows of exhaust from H and G to port F.

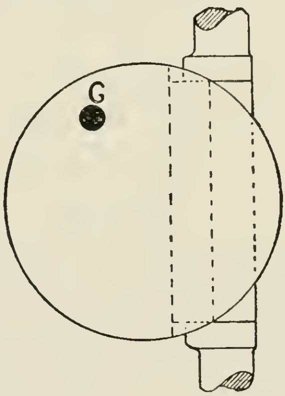
The auxiliary valve is moved up and down by the levers from the pump rod striking a pair of adjustable nuts fitted in the valve spindle, the distance between the nuts allowing of a certain amount of lost motion.

The auxiliary valve has two separate functions :—

1. To work the main slide valve by admitting steam and exhausting it from the valve ends ; and
2. To cut off steam at a definite part of stroke.

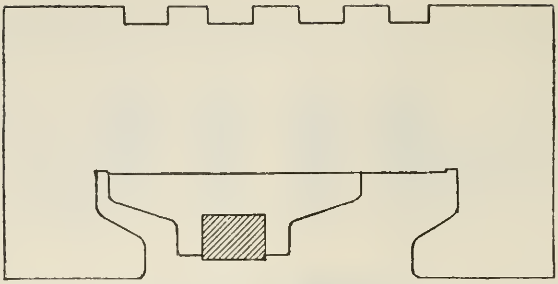


No. 32.—Expansion (Auxiliary) Valve Face.



No. 33.—End View of Main and Auxiliary Valves in Position.

Action of Valves (Up Stroke).—When the cylinder piston is on the bottom centre, the main valve is at the right-hand side of the chest, and the auxiliary valve at its lowest position. Steam is then entering through the main valve port E into the bottom port A of the cylinder, and to the left-hand end of the main valve by port G : this continues as the piston moves upward until about half stroke, when the lever strikes the nut on the auxiliary valve spindle, and the valve coming up cuts off the steam entering port E at about three-quarters stroke ; the steam in the main valve and cylinder then expands and completes the stroke. When the piston reaches the end of the stroke the auxiliary valve opens up the exhaust port G from the left-hand end of the main valve, and the steam acting on the other end, forces the valve across, thus opening the bottom cylinder port A to the exhaust



No. 34.—Plan of Main and Auxiliary Valves in Position for Steam to Bottom of Cylinder.

and at the same time opening the top cylinder port C to steam for the down stroke.

For the down stroke the same process is gone through with the other ports, the main valve moving in the opposite direction, that is, from left to right. It is important to note that the main valve does not move until one end is opened to exhaust: it is then forced across by the steam pressure on the opposite end.

The auxiliary valve after opening the end to exhaust by port G or H, is arranged to close again, so that steam is retained to act as a cushion to the main valve when it flies over, and thus prevent hammering on the chest end covers. It will be seen from the foregoing that the valve gear is positive in action, the main valve being always open to *steam* for one end of the cylinder, and to *exhaust* for the other end, with the piston on the corresponding centre. It is therefore impossible for the pump to stick on the centre points.

Bye-Passes.—Small bye-passes are fitted at each end of the cylinder to admit steam full stroke when required. This may be necessary when starting the pumps, as the cylinder may then contain a quantity of water. "Knocking" can also be reduced by suitable adjustment of the bye-passes.

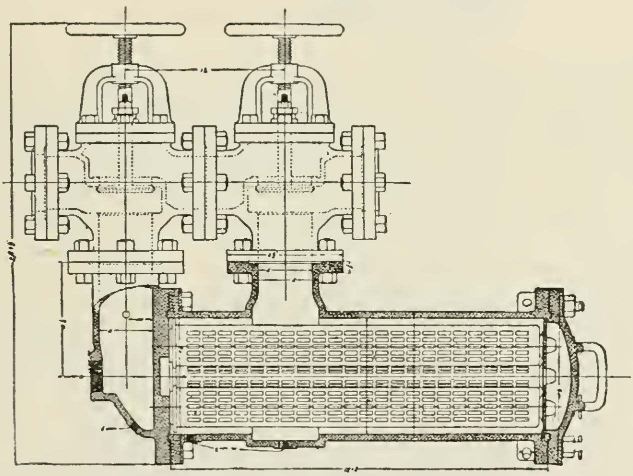
The bye-passes are formed either by notches cut in the edges of the cylindrical caps, which may be opened or shut by turning round the caps, or by parallel plug cocks, one at each side, which can be adjusted to admit as much steam as required for the occasion.

To Adjust the Length of Stroke of a Weir Feed Pump.

This is done by screwing the valve spindle up into the joint until the piston comes to rest against the cylinder cover, without having raised the auxiliary valve sufficient to throw the main valve. By noting the distance of the crosshead from the gland when the piston is against the cylinder cover, the spindle can be brought down until the valve throws and the piston clearance is $\frac{1}{2}$ inch. The lock nut should now be carefully fixed to prevent the spindle from shaking loose. The bottom stroke adjustment is done by slacking the nuts on the bottom end of the spindle. These should be screwed down until the piston rests against the bottom of the cylinder, and by again noting the distance of the crosshead from the pump gland you can again adjust it for the valve to throw when the pump has ½ inch of clearance. Always run the pump *slowly* when adjusting the stroke. It is most important that any one having the working of a "Weir" Pump should become familiar with the method of adjusting the stroke.

Feed-Water Filters.

A feed-water filter is employed to prevent oil and grease from entering the boilers with the feed water. It is usually placed between the feed pumps and the feed check valve, and consists of a cast-iron



No. 35.—Feed Water Filter.

(American Steam Gauge and Valve Manufacturing Co.)

chamber containing a series of perforated brass plates with filter cloths fitted between them, or brass grids arranged in a similar manner.

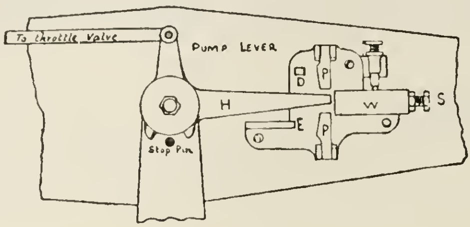
The feed water is forced through the perforations and cloths, and leaves behind the greasy matter, which is blown out at intervals.

The fittings on the filter are :—Bye-pass valve (used when cleaning out or changing the cloths), safety valve, pressure gauge, soda cock, and drain cock.

Sometimes, instead of cloths, ashes or charcoal are used as the filtering medium.

Aspinall's Patent Governor.

Aspinall's patent governor is usually bolted to the side of the pump levers, and consists of a frame containing a hinged weight W, which operates on two pawls P, P; the pawls when in action striking a lever connected to the throttle valve and so regulating the steam supply to the engines.



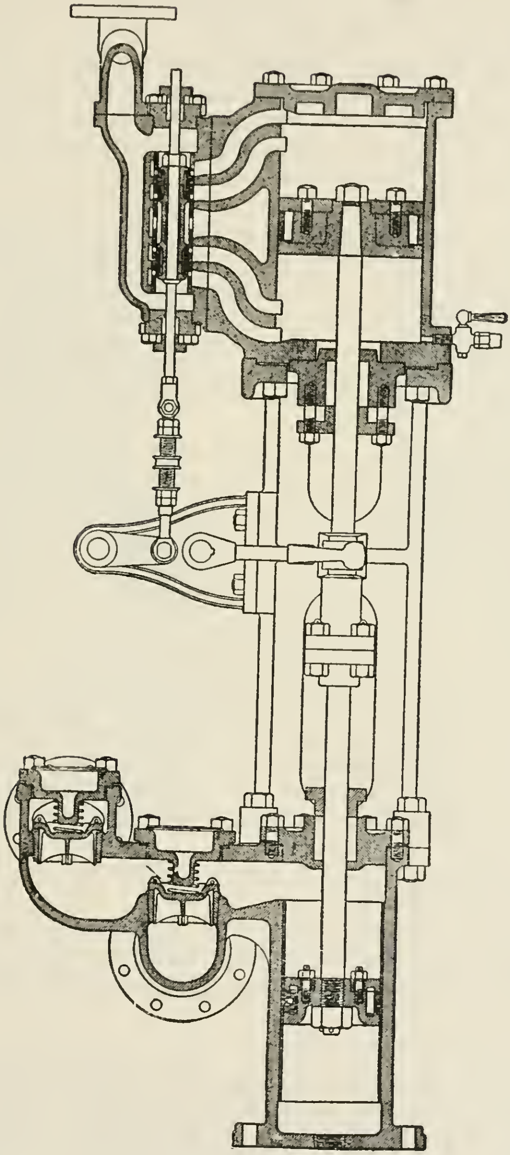
No. 36.—Aspinall Governor.

When the revolutions increase by about 5 per cent. above the normal speed, the weight W is left behind by the increase of momentum, and this reverses the position of the pawls, causing the bottom one to fall out and strike the lever H, and lift it on the upward stroke; the throttle valve is by this means closed: on the downward stroke the detent D is lifted, and this again sets free the weight W, and, if the racing is over, the top pawl P strikes the lever H and brings it back again to its original position, which has the effect of reopening the throttle valve. An emergency gear is also fitted, which locks the weight W in the shut-off position, in the event of a serious accident happening, such as, for example, the shaft breaking or the propeller coming off. This type of governor is fitted to the turbine steamer "Carmania."

Vorthington Vertical Type Feed Pumps (Sketch No. 37).

These pumps are supplied in pairs, the crosshead of one pump actuating the steam valve of the *other* pump. The steam valve is generally of the piston type and is allowed about $\frac{1}{4}$ inch or $\frac{3}{8}$ inch "lost motion" to allow of full steam admission each stroke; double ports are arranged for at top and bottom, the outer one being for steam admission only and the inner one for *exhaust* only, compression and cushioning being provided for by the *piston travelling over and closing the inner or exhaust ports*, and so retaining the required amount of compression steam.

The water valves are spring loaded, and are, of course, four in number, one suction and one delivery for either end of pump. By the momentary pause which occurs at the end of each stroke, the suction valve of one end and the delivery valve of the other end get time to seat themselves quietly and without shock.



No. 37.—Worthington Patent Feed Pump.

To Set the Valves of a Worthington Type Pump.

The steam valve of a Worthington pump has no outside lap, consequently, when in its *central position*, it just covers the steam ports leading to opposite ends of cylinder.

By *lost motion* is meant the distance a valve rod travels before moving the valve; or, if the steam chest cover is off, the amount of *lost motion* is shown by the distance the valve can be moved back and forth before coming in contact with the valve rod nut.

To set the piston in the middle of its stroke, open the drip cocks and move piston by prying on the crosshead (not on lever), until it comes in contact with the cylinder head; make a mark on piston rod at face of steam end stuffing box follower; move piston back to contact stroke at opposite end. Make second mark on piston rod half-way between first mark and the aforesaid follower. Then if piston is again moved until second mark coincides with the face of same follower, it will be exactly at the middle of its stroke.

Bear in mind that the piston on one side moves the valve on opposite side.

(*a*) When the steam valve is moved by a single valve rod nut.

Place one piston in the middle of its stroke; disconnect link from head of valve rod on opposite side. Then set the valve in its *central position;* place valve nut evenly between jaws on back of valve, screw valve rod in or out until eye on valve rod head comes in line with eye of valve rod link; then reconnect. Repeat the operation on opposite side and the valves will be properly set.

(*b*) When the valve rod has more than one lock nut.

Place one piston in the middle of its stroke and the opposite slide valve in *central position;* adjust lock nuts so as to allow about $\frac{3}{16}$ inch *lost motion* on each side of jaw, and valve is set. Do not disconnect the valve motion. Repeat operation on other side.

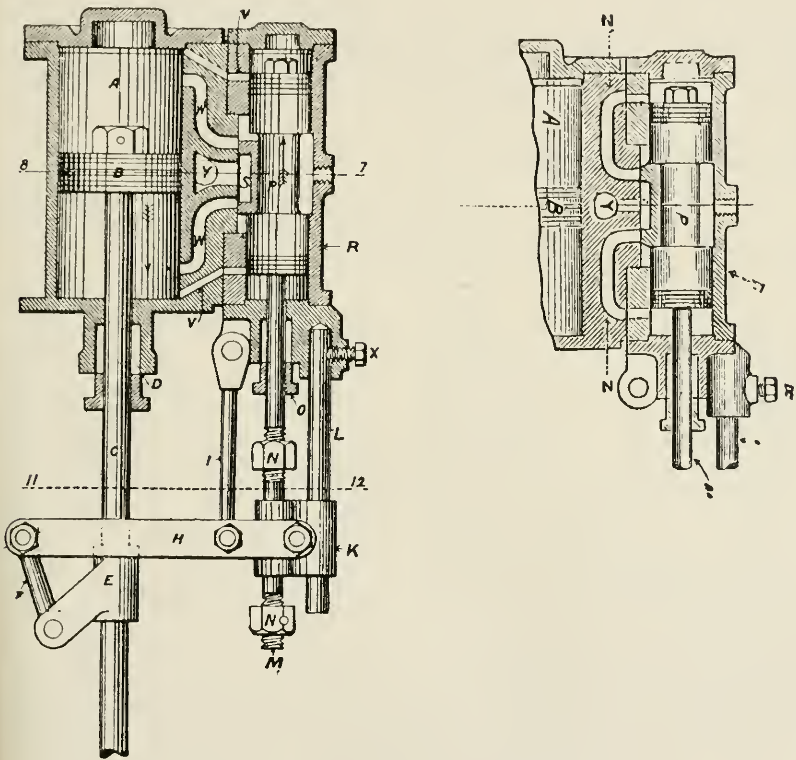
The best way to divide the *lost motion* equally is to move the valve each way until it strikes the nut or nuts, and see if the port openings are equal.

It is advisable to place both pistons at the middle of their strokes before touching either slide valve.

Too much *lost motion* will tend to lengthen the stroke, and may cause the piston to strike the cylinder heads; *vice versâ*, when there is not enough *lost motion* the stroke will be perceptibly shortened.

Lamont Pump (Sketch No. 38).

Action.—P is the auxiliary piston, which, with both slide valves S and T, is moved by the main piston B before coming to rest, by means of the links and levers to a little over the central position, and this puts one end of the auxiliary cylinder R in communication with pressure and the other end with the exhaust, whereby the auxiliary piston and valves S and T are moved by the steam pressure on P to the extent necessary to give full port openings to the main cylinder, and at the same time the auxiliary piston P covers the supplementary port V to the end of the main cylinder, which is open to the exhaust and uncovers the one at the end open to steam pressure. The opening and closing of the supplementary ports V¹, V by the piston P takes place simultaneously with the opening of steam or exhaust to the ends of the auxiliary cylinder R, so that as soon as either supplementary port V¹ or V is uncovered by P, steam is admitted to that end of the auxiliary cylinder by the slide



No. 38.—Lamont's Patent Pump.

valve T, and conversely as soon as the slide valve T opens either end of the auxiliary cylinder to exhaust the supplementary port in that end V¹ or V is covered by P. The drawing shows the piston P and the auxiliary valve T in the central position, but the main valve S is still open, and will cause the main piston B to move in the direction of the arrow until it passes the port W, which is open to the exhaust, when it will gradually be brought to rest by the imprisoned steam forming a cushion between the piston and the end of the cylinder, as this space is now completely closed. The stop nuts N N are fixed on the valve spindle M in such a position as just to move P a little over the central position before the main piston B comes to rest. The auxiliary valve T will then admit steam by the auxiliary port Z to the end of the auxiliary cylinder R, and at the same time the other end of the auxiliary cylinder will be opened to the exhaust through the port Z^1. The pressure of the steam will thus cause the auxiliary piston, with both slide valves S and T, to move to the other end, and so reverse the steam and exhaust passages to the main cylinder, but as the main piston B still covers the port W in the cylinder, the steam will only be admitted through the supplementary port V until the piston B has moved so as to uncover the port W and allow the full flow of steam. The compression and slow reversal of motion is the same for both ends of the stroke.

Engine and Boiler Data.

A good idea of the general proportions and dimensions for a steamer of given speed and power will be obtained from the following data taken from a modern fast ocean-going steamer:—

Length over all	434 ft.	Boilers, Number	2 single-end; 2 double-end
,, between perps.	420 ,,	Pressure	185 lbs.
Breadth, moulded	50 ,,	Diameter	15 ft. 6 in.
Depth, ,,	30 ft. 6 in.	Length	10 ft. 6 in.; 18 ft.
Height between decks,	7 ,, 11 ,,	Number of furnaces	18
Gross tonnage	5600 tons	H.S. total	14840 sq. ft.
Engines	single, triple	G.S. ,,	465 ,,
Diameter cylinders	31, 51, and 85 in.	I.H.P. on trial	6500
Stroke	54 ,,	Speed ,,	17 knots
Valves, piston on H.P., slide on I.P. and L.P.		H.S. ÷ I.H.P.	2·28
		I.H.P. ÷ G.S.	13·98
		H.S. ÷ G.S.	31·91

Pressure Gauge Indications.

The boiler pressure gauge indicates the pressure in the boiler above the atmosphere, and therefore the initial pressure in the H.P. cylinder (less a few pounds drop of the pressure).

The M.P. gauge indicates roughly the back pressure on the H.P. piston, and the initial pressure in the M.P. cylinder.

The L.P. gauge indicates roughly the back pressure on the M.P. piston, and the initial pressure in the L.P. cylinder.

The vacuum gauge indicates how many pounds of pressure the air pump has taken out of the condenser, and if this be subtracted from the atmospheric pressure of 15 lbs., it will give the pressure still left in the condenser, and therefore acting on the L.P. piston as back pressure (approximately).

Example.—The boiler pressure gauge indicates 160 lbs., the M.P. gauge 53 lbs., the L.P. gauge 8 lbs., and the vacuum gauge

24 inches. Allow a drop of pressure between boilers and engines of, say, 5 lbs.

Then, Boiler pressure - - - -	160 lbs. gauge.
Initial pressure in H.P. cylinder -	155 lbs. ,,
Back pressure in H.P. - - -	55 lbs. gauge.
Initial pressure in M.P. - - -	53 lbs. ,,
Back pressure in M.P. - - -	10 lbs. gauge.
Initial pressure in L.P. - - -	8 lbs. ,,
Back pressure in L.P. - - -	4 lbs. gross.
Pressure in condenser - - -	3 lbs. ,,

Notice that the gross back pressure on the L.P. piston is obtained by taking the vacuum in pounds from the atmospheric pressure. Thus: 24 inches $\div 2 = 12$ lbs., and $15 - 12 = 3$ lbs. pressure in the condenser, which is, within a pound or so, the back pressure on the L.P. piston.

NOTE.—Allow a difference of about 2 lbs. between the back pressure of the one engine and the initial pressure in the next engine, as naturally a fall of pressure must take place between the two.

The Barometer.

The barometer is an instrument used in measuring the pressure or weight of the atmosphere.

The mercurial type of barometer consists of a glass tube fully 31 inches in length, closed at the top, and open at the bottom to a cup containing mercury. Between the mercury and the top of the tube there is a vacuum practically perfect, and the pressure of the atmosphere acting on the surface of the mercury in the cup forces it up the tube against the vacuum in the top, and so indicates the air pressure.

When the weight of the atmosphere is 15 lbs. per square inch, the difference in level of the mercury in the cup and tube will be 30 inches, and this is termed the height of the barometer.

It will thus be seen that every pound of atmospheric pressure raises the mercury 2 inches up the tube, and the atmosphere being 15 lbs., then 15×2 inches $= 30$ inches of mercury. If the atmospheric pressure fell to 14 lbs. the barometer would show 28 inches, and if the atmosphere increased to $15\frac{1}{2}$ lbs. the barometer would indicate 31 inches, and so on, every pound causing a difference of 2 inches, and every $\frac{1}{2}$ lb. 1 inch in the mercury level.

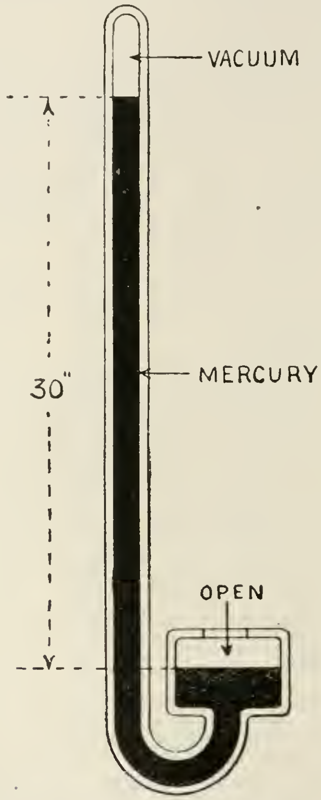
High up, as, for example, on a mountain top, the barometer will show less than 30 inches, because the air pressure will be less than 15 lbs.; and low down, as at the bottom of a mine, the barometer height will be more than 30 inches, as the atmospheric pressure will exceed 15 lbs.

If water were used instead of mercury, then the height of the barometer would be $34\frac{1}{2}$ feet, because 15 lbs. $\times 2{\cdot}305 = 34{\cdot}5$ feet: this is, at the same time, the theoretical limit that a pump can draw up water. In practice the limit is 26 feet, as it is impossible to obtain a perfect vacuum, even with the best pump fittings, &c.

NOTE.—One pound of pressure per square inch is equal to a column of water 2·305 feet in height.

If the pump has to draw water at a high temperature, then the limit of lift becomes less, as the temperature rises, and at, say, 200° Fahr. the pump would hardly draw up the water any height, owing to the vapour formed destroying the vacuum. This is the reason that "Weir's" feed-heater is placed high up in the engine-room, so that the water may fall by gravity to the pump suction.

The barometer can be made to act as a vacuum gauge by opening up the closed end, and connecting it by a pipe to the condenser, the



No. 39.—Mercurial Barometer.

other end being still left open to the atmosphere. Every pound of pressure taken out of the condenser by the air pump will cause the mercury to rise 2 inches in the tube; therefore if, say, $12\frac{1}{2}$ lbs. are drawn out by the air pump, the difference of level in the cup and the tube will be 25 inches. If there be no vacuum in the condenser, the level of the mercury will be the same in the cup and in the tube, that is, there will be no difference of level.

Aneroid Barometer.—The barometer in general use at the present day is of the "aneroid" type, consisting of a vacuum box at the back, and a set of very sensitive levers connecting to the indicating pointer on the dial face.

The variations in the atmospheric pressure act on the back of the vacuum box, and either slightly force it in, or allow it to ease back, and this setting in motion the set of finely adjusted levers and gear, causes the pointer to move round the dial and indicate correspondingly the pressure of the atmosphere.

The Thermometer.

The thermometer is an instrument used in measuring the temperature of bodies. It consists of a glass tube of fine bore, partly filled with mercury or spirits, and having a bulb at the bottom end, and the top end sealed.

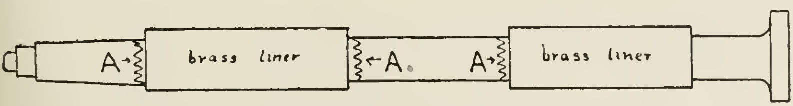
In graduating the instrument, the tube is placed in a closed vessel with the steam from boiling water surrounding it, and the heat causes the mercury to expand and rise in the tube ; when the mercury stops rising a mark is made at the level and fixed as 212°, representing the boiling point of fresh water in the atmosphere. The tube is next placed in a dish containing pieces of broken ice, and the cold causes the mercury to contract and fall in the tube, the point when the liquid falls being marked as 32°, which is the melting point of ice, or the freezing point of water.

Between the 212° and the 32° marks, the scale is divided into 180 equal parts, each part representing 1° of temperature.

NOTE.—Mercury solidifies at $-38{\cdot}5°$ and boils at 725° Fahr.

Tail Shaft Corrosion.

On propeller shafts fitted with two separate liners corrosion occurs at the places marked A, that is, at the end of the brass liners. This is caused by galvanic action between the iron or steel of the shaft and the brass of the liners in sea water, and the effect is intensified by the heavy stresses thrown on the shaft when the stern rises and



No. 40.—Corrosion on Tail End Shaft.

falls in a heavy seaway, and the propeller and shaft strike the water. When a tail shaft breaks it usually gives way at one of the places marked A.

The bending stresses are concentrated at the end of the liners, owing to the difference in diameter of the liners and the shaft, and it will be noticed that the wasting away of the latter is due to a combined mechanical and chemical action, one leading up to and assisting the other.

If the sea water is prevented from coming in contact with the shaft, galvanic action cannot take place, and the shaft is preserved.

Therefore all remedies for the preservation of the tail shaft consist of arrangements for keeping the sea water from the shaft. The three most important methods are:—

1. The brass liners carried the whole length of the shaft.

2. A rubber sleeve extending between the two brass liners, and made watertight at the ends.

3. A packed gland at the stern post (outside) between the boss and the stern tube, and oil let into the stern tube for the shaft to run in instead of sea water, the shaft being without brass liners, and running on white metal instead of lignum vitæ strips (see Cedarvall's patent stern tube).

Iron Shafts.—Many makers prefer good iron for tail end shafts instead of steel, for the reason that under conditions of galvanic action iron does not corrode so fast as steel.

Propeller Pitch.

The pitch of a propeller means the distance the propeller would travel in one revolution if working in a solid nut.

As the propeller works in water, the actual advance is less, owing to slip. The usual amount of slip is from 5 to 15 per cent.

The hollowed after surface of the blades is called the "thrust surface," and the rounded forward surface the "drag surface." In running ahead the after surface of the blades thrusts back the water and the resultant reaction thrusts forward the steamer.

In running astern, the drag surface thrusts the water forward, and the reaction resulting sends the steamer aft.

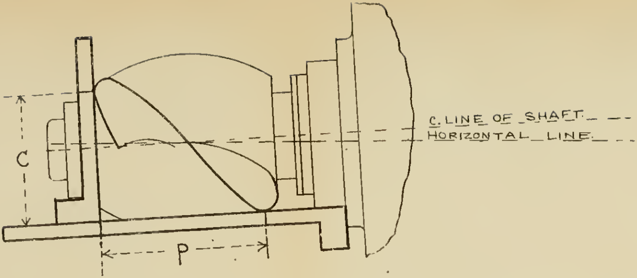
To Measure the Pitch.—With the ship in dry dock, and shaft horizontal, turn the engine so that one of the blades is horizontal, take a string with a weight tied to each end of it, and hang the string over the blade, about two-thirds out from the boss. Now take a straight-edge and fix it parallel to the shaft, and just touching the bottom end of the blade where the string hangs.

The distance between the string is piece of pitch $=p$, and the length of string from the top of the blade to the straight-edge is piece of circumference $=c$. Next measure from the string to the centre of the boss, and multiply by 2 and by 3·1416. This will give the full circumference $=$ C, at the position of the string and straight-edge; then by proportion as follows:—

As piece of circumference : Full Circumference : : piece of pitch : Full Pitch.

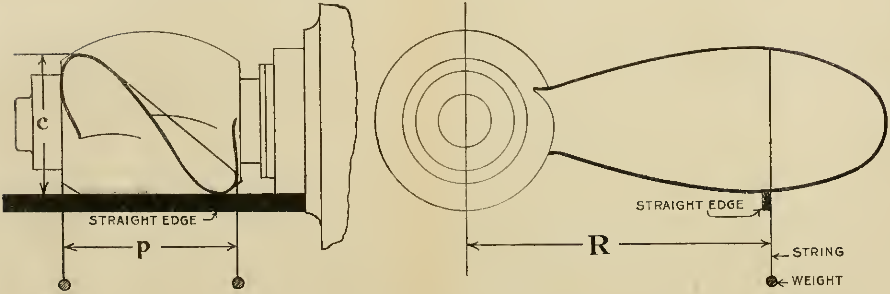
Or, as c : C : : p : Full Pitch.

NOTE.—If the pitch varies radially, take the pitch as described at three different radii, and the mean of the three may then be taken as the average pitch.



No. 41a. To Measure Pitch when Shaft is not Horizontal.

When shaft line is not dead horizontal, but is slightly inclined as in sketch, the plumb line method of pitch measurement does not apply, and steel or wood squares or straight edges should be arranged as shown to obtain part pitch and part circumference, after which the full pitch can be determined in exactly the same manner as shown in No. 41.



No. 41.—To Measure Pitch of Propeller.

(With Shaft Horizontal.)

p = Piece of pitch.
c = Piece of circumference.
C = Full circumference at position of weights.
R = Radius, and Radius × 2 × 3·1416 = C.
Then, as c : C : : p : Full pitch

NOTE. If a right-hand propeller is taken off and put on again reversed (assuming the shaft parallel at boss), then the propeller will still remain a right hand screw, but the efficiency will be reduced for ahead running, and increased for astern running.

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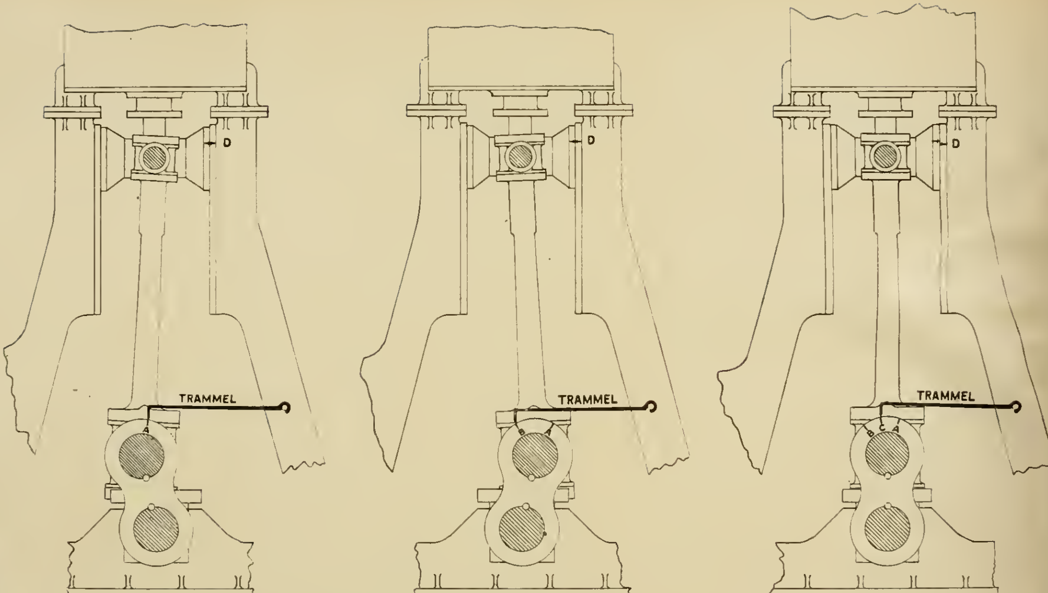


41a.—Crank on Centre.

A First Mark B Second Mark C. Centre Mark. D. Position of Guide and Shoe at each Crank Angle.

[*To face page* 323.

To Find the Cut-off.

To measure the distance the steam is carried on the down stroke, put the crank on the top centre and mark the crosshead and guide, take off the valve casing cover, and turn the engine round with the turning gear until the valve comes up and closes the port; when it does so, or cuts off, stop the turning gear, and the distance measured down from the mark on top centre to where the crosshead stops will be the cut-off or distance the steam is carried.

NOTE.—The exact time of closing the port is best determined by inserting a slip of paper into the steam port, so that the edge of the valve "nips" the paper on the instant of closing the port.

Crank on Centre.

To put the crank on the top centre, first turn the engine up to near the top, and mark the guide and shoe at D; then with a trammel fixed on the column, and long enough to reach the crank, mark the top of the crank at A. Now turn the engine over the centre until the marks on the guide and shoe come together again at D, and again mark the crank top with the trammel at B. Find the centre between the two marks on the top of the crank, and make a mark C, and turn the engine until the mark C comes in line with the trammel point. The engine will then be on the top centre.

The crank is put on the bottom centre by the same method, the trammel being applied to the bottom end of the crank instead of the top.

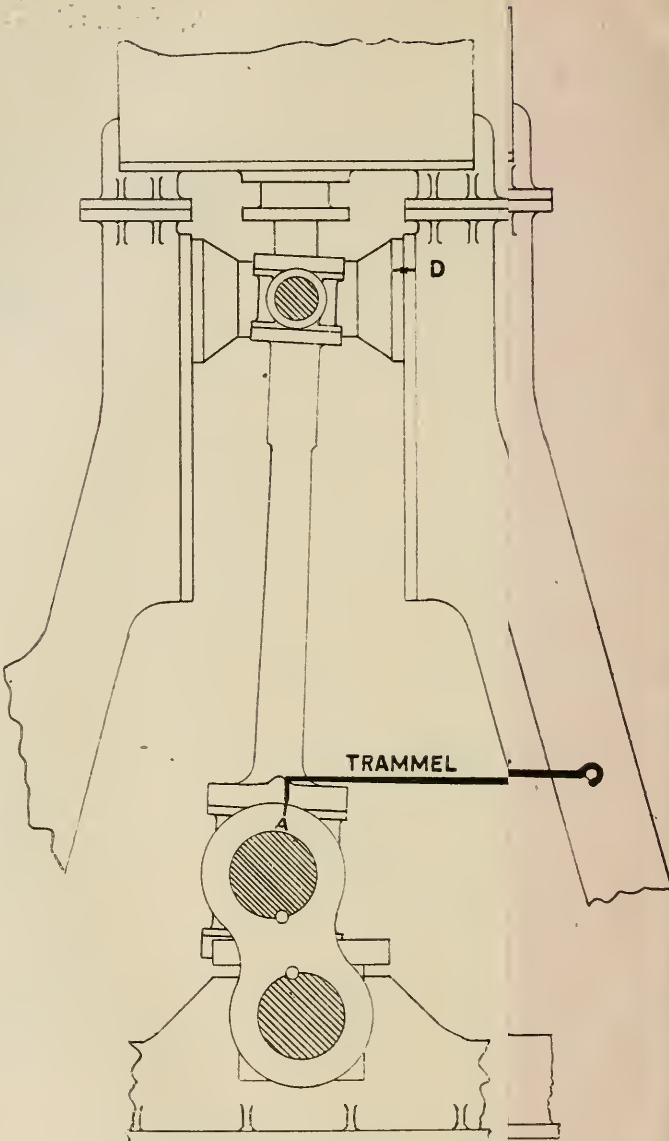
Cutting of Key Seats (see also page 232).

To mark the position of the eccentric key seats on the shaft, first put the crank on the top centre, and with the eccentric gear connected up, turn round the pulley until the valve is open for the required top lead, and mark the shaft. Having fixed the pulley by a set pin, turn the engine to the bottom centre, and easing back the set pin, shift the pulley further round the shaft until the valve is open for the required lead at the bottom, and again mark the shaft. Find the centre between the marks on the shaft, and this will be the position of the key seat. Cut the key seat, bolt on the pulley, and put a liner under the rod to make up the difference of leads top and bottom.

NOTE.—If the lead is to be $\frac{1}{8}$ inch at the top, and $\frac{1}{4}$ inch at the bottom, the liner will require to be $\frac{1}{16}$ inch thick.

To Set Valve in Mid Travel.

Turn round the engine until the valve comes to its top position, and mark the valve rod at the gland; then turn the engine until



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NOTE.—The exact time of closing the port is best determined by inserting a slip of paper into the steam port, so that the edge of the valve "nips" the paper on the instant of closing the port.

Crank on Centre.

To put the crank on the top centre, first turn the engine up to near the top, and mark the guide and shoe at D; then with a trammel fixed on the column, and long enough to reach the crank, mark the top of the crank at A. Now turn the engine over the centre until the marks on the guide and shoe come together again at D, and again mark the crank top with the trammel at B. Find the centre between the two marks on the top of the crank, and make a mark C, and turn the engine until the mark C comes in line with the trammel point. The engine will then be on the top centre.

The crank is put on the bottom centre by the same method, the trammel being applied to the bottom end of the crank instead of the top.

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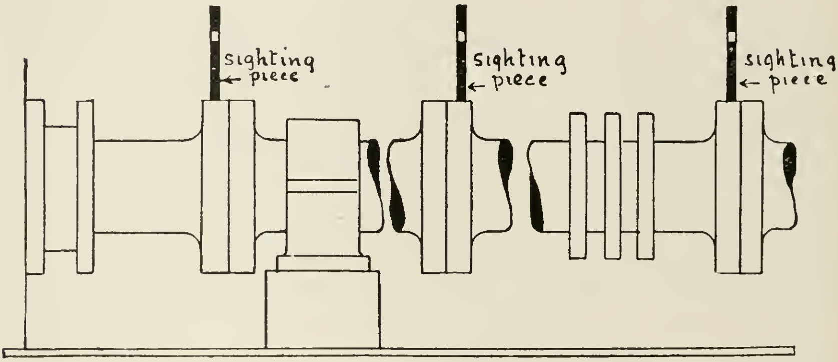
NOTE.—If the lead is to be $\frac{1}{8}$ inch at the top, and $\frac{1}{4}$ inch at the bottom, the liner will require to be $\frac{1}{16}$ inch thick.

To Set Valve in Mid Travel.

Turn round the engine until the valve comes to its top position, and mark the valve rod at the gland; then turn the engine until the valve comes to its bottom position, and again mark the rod at the gland; next divide the two marks, and set the valve to the centre mark so found, which will be the exact mid position of the valve. By marking the cylinder face, top or bottom, at the position of the steam edge of the valve, the steam lap can be found by simply taking the distance between the marks and the edge of the steam ports of the cylinder.

Shaft Sighting.

To test the fairness of a line of shafting by "sighting," obtain three strips of iron all the same length, and with a small hole in two of them, say $\frac{1}{8}$ inch in diameter, cut in each at the same distance up,



No. 42.—Method of "Sighting" a Shaft.

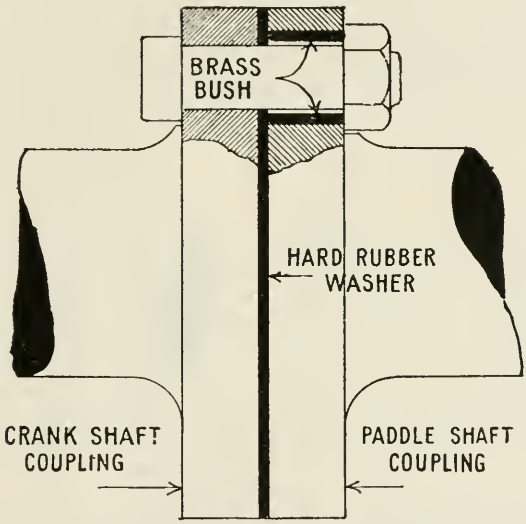
and in the third one have cut a vertical slot; now fix one strip on the thrust shaft coupling, another on the tail end shaft coupling, and place the other one with the vertical slot on the intermediate shaft couplings turn about, and with a light behind the hole of the strip on the propeller shaft coupling sight the shaft from the strip on the thrust shaft.

If the light is visible at the same level through all the strips the shaft is fair, but if the light is sighted higher up on the portable strip the corresponding length of shaft has worn down (see sketch).

Lining Up Shafting.

A $\frac{1}{16}$-inch hole is drilled through forward engine-room bulkhead; hole in stern post bridged and a $\frac{1}{16}$-inch hole drilled through bridge, both drilled holes to correspond with shaft centre. A light is then placed forward of engine-room bulkhead hole, and it can easily be seen through these two $\frac{1}{16}$-inch holes. After-tank bulkhead hole is then bridged, a $\frac{1}{16}$-inch hole drilled through bridge, and bridge adjusted so that the light can be seen through these three $\frac{1}{16}$-inch holes. Circles can now be described around these holes in the stern post and after-tank bulkhead with the $\frac{1}{16}$-inch holes for centres, bridges removed, and boring bar set by circles. Of course any number of intermediates may be erected before bridges are taken out, to get height of seatings, &c. All errors of a sagging line are thus avoided.

To line from tail-shaft coupling, the tunnel-shaft bearings, of which there is often only one for each section of shafting, are not



No. 43.—Flexible Coupling.

used. Each length of shafting is blocked on two blocks in such a way that the overhanging ends balance the portion between blocks, otherwise the shaft will sag, throwing couplings out. Faces of couplings are left a little apart for lining up, so that a very keen taper wedge can be used between them. If now the after coupling of after line shaft section is central with forward coupling of tail shaft, and the taper wedge enters the same distance all around between their faces, tail shaft and after section of line shaft are surely in line, and the next length forward may be proceeded with. When all the shafting is in, engine bed with crank-shaft in place is set in the same way, and forward crank-shaft centre will correspond with the $\frac{1}{16}$-inch hole drilled in forward engine-room bulkhead. Before blocks are removed from under line shaft, the tunnel shaft bearings are put in place.

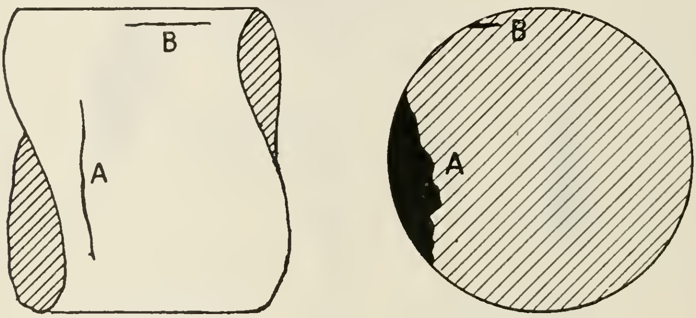
Paddle Shaft Flexible Coupling (Sketch No. 43).

Sometimes in paddle steamers allowance for the outside bearing wearing down is arranged for as shown, but in this case the crankshaft, crank-pin, and webs are solid.

A hard rubber washer is fitted in between the coupling faces, and the bolts fit into brass bushes in the outer or paddle shaft coupling. These bushes are made large enough in diameter to give a slight clearance to the bolts as shown, which thus gives flexibility to the shaft.

Flaws in Shafts.

In the sketches below two flaws, A and B, are shown on the shaft, one (B) running longitudinally, and the other (A) extending circumferentially. Notice that the circumferential flaw A seriously affects the strength of the shaft by decreasing the sectional area available to



No. 44.—Shaft Flaws.

resist torsion, whereas the flaw B, which extends along the shaft, produces very little difference in the area or strength. It should be remembered that the strength of a solid shaft to resist torsion varies as the diameter *cubed.*

NOTE.—The depth of a flaw can be found by either boring a hole into it or by chipping out.

To Test the Fairness of Shafting.

To test if the main shafting is down, slacken back the coupling bolts, and with a feeler, test the distance between the two flanges all round.

To Test the Fairness of the Rocking Shaft.

To test if the pump lever rocking shaft is down at one end or the other, place the levers at half stroke and disconnect the crosshead links, then measure the distance between the engine crosshead pin and the pin on the end of the lever on the after side and on the forward side; if the two measurements do not agree, one of the two brasses has worn down.

Link Brasses and Pump Clearance.

If the engine crosshead link brasses are tightened up, the pump clearance will be increased on the top and decreased on the bottom.

If the pump crosshead link brasses are tightened up, the pump clearance will be decreased on the top and increased on the bottom.

NOTE.—The foregoing assumes that the links are suspended from the engine crosshead: if overhung the clearance will be reversed.

To Test the Fairness of the Piston Rod.

To test if the piston rod is working fair between the guides, ease back the gland and take the distance between the guide and piston rod at the top and bottom centres.

Air Pump Clearance.

To measure the air pump clearance top and bottom, put the crank on the top centre; this will place the pump on the bottom centre; then mark the rod, say at the gland. Next put the crank on the bottom centre; this will place the pump on the top centre, and again mark the rod. If the pump links are then disconnected, and the pump lifted against the cover and the rod marked, we will have the top clearance; and if the pump is lowered to the bottom and the rod marked, it will give us the bottom clearance.

To find Length of Valve Spindle.

If the valve rod is broken and the length for a new rod required, this can be found as follows:—

Turn the engine up to the top centre (see page 323) and shore up the valve to the lead determined upon; also suspend the links in line with the valve spindle for the required ahead position. Now pass up through the gland and valve a long length stick or rod, and mark on it the distance from the under side of the valve to the centre of the link block, which will therefore be equal to the required length of valve spindle between these points. The position of the washer or collar under the valve, and the position of the nuts above the valve, can also be marked on the stick, and the necessary extra length of rod allowed for accordingly.

To find the Connecting Rod Length.

To find the length of the connecting rod, put the engine at half-stroke, and measure from the centre of the crosshead pin to the centre of the shaft.

To find the Eccentric Rod Length.

To find the length of the eccentric rod (the pulleys being on the shaft), put the crank on the top centre, and set the valve to the required lead, then measure from the centre of the link block to the top of the eccentric strap; this gives the exact length of the eccentric rod.

Another method is as follows:—Place the valve at mid-stroke, and measure the distance between the centre of the link block to the centre of the shaft; then subtract from this half of the pulley diameter and the thickness of the eccentric strap at the place where the rod joins it: this will leave the length of the rod.

Piston Clearance.

The piston and cylinder cover clearance at top is usually about $\frac{5}{8}$ inch, and at bottom about $\frac{3}{4}$ or $\frac{7}{8}$ inch.

The bottom clearance requires to be more than the top, to allow for the wear down of the top end and bottom end "brasses" or "white metals."

To Measure the Piston Clearance.

It is measured by turning the crank to the top centre and marking the shoe and guide, then disconnecting the crosshead brasses and lifting or wedging up the piston until it touches the cover; then again mark the guide, and the distance between the marks will be the top clearance.

The bottom clearance is measured in a similar manner.

Excessive Clearance.

Excessive clearance means a distinct loss of heat, because the steam must first fill up the clearance spaces before it can do work on the piston, and, when the exhaust opens, this steam is exhausted out of the cylinder.

Prevention of Ridge in Cylinder.

Cylinders are usually made bell-mouthed at the bottom to prevent the piston wearing a ridge on the cylinder or liner.

Defective Check Valve.

With two boilers, if the check valve of one gets so damaged that it cannot be shut and the boiler is getting too much feed, regulate the feeding by the stop valves, partly shutting down the one on the boiler which is getting too much water and opening up the one on the boiler that is not getting enough of feed water. The difference of evaporation will then keep the water from entering the one boiler and allow it to enter the other. The same result can be obtained by firing one boiler more than the other.

U Tube in Pressure and Vacuum Gauges.

The gauge tube is of flattened section, as shown in the sketch, and when acted on by increase of pressure, the tube section tends to become more circular, but when acted on by decrease of pressure the tube section tends to become more flattened. This difference in section produces a reaction, the effect of which is to straighten out the tube when the tube becomes more circular in section, and to curl up the tube when the section becomes flatter. For pressure, the tube loses diameter in one direction and gains diameter in the other, being forced more into a circular shape by the action of the steam pressure, whereas in a vacuum gauge, for example, the reverse effect takes place, that is, the tube becomes less circular in section, and takes its length on a more curved line in consequence.

It will easily be understood that with the tube quite flat in section the curvature of its length would be at a maximum, and with the tube fully circular in section its line would naturally be straight.

Therefore (1) under pressure the long diameter of the tube decreases, and the short diameter increases, and this results in the tube length straightening out and indicating (by means of the small quadrant and toothed gearing) increase of pressure.

(2) Under decrease of pressure the long diameter increases and the short diameter decreases, and this results in the tube length forming a smaller curve, and indicating decrease of pressure, or vacuum.

SECTION

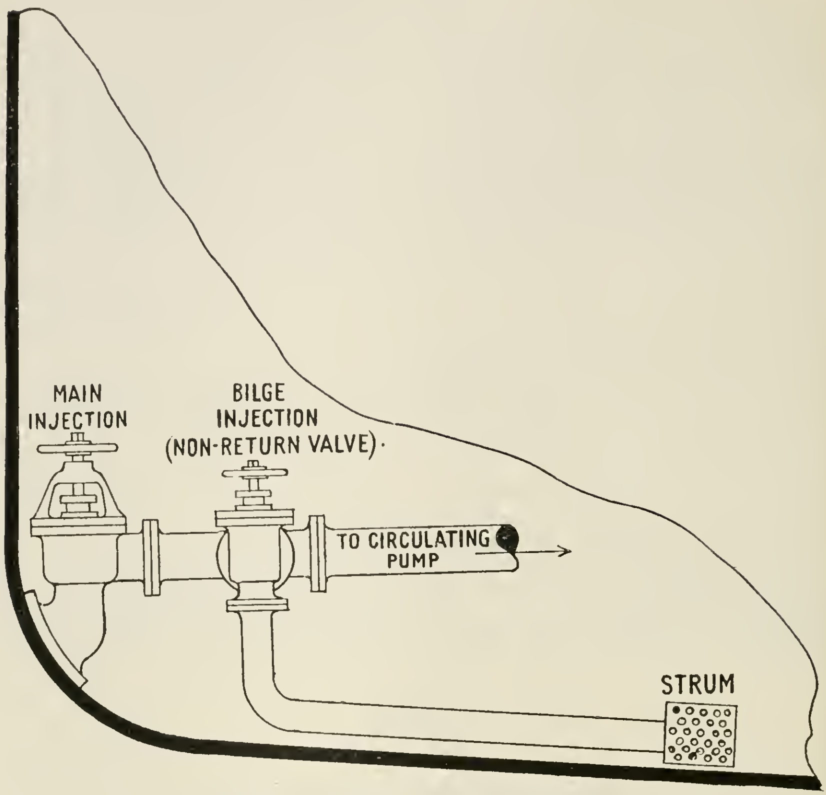
No. 45.—Pressure Gauge Tube.

NOTE.—If the pressures are equal inside and outside of the U tube, the gauge will register 0: therefore when the atmospheric pressure increases, the gauge will register less pressure in like proportion, although the actual pressure in the tube is exactly the same as before: similarly, if the atmospheric pressure decreases, the gauge will register more, although the actual pressure in the tube will be the same.

Main and Bilge Injection (Sketch No. 46).

The illustration shows the usual arrangement of the main and bilge injection connections.

Observe that the bilge injection pipe leading from the bilge strum is connected to the main injection pipe by means of a non-return valve, which is peculiar in the fact that the valve and spindle are separate. The valve can be shut down but not lifted up by the screwed spindle, as only the pressure of the water below acts to lift the valve.



No. 46.—Injection Connections.

This type of valve is advisable, as it prevents the return of water back to the bilges.

NOTE.—If, when using the bilge injection, the strum becomes choked up, cut the pipe above the strum, and fix a basket over the end of the pipe, or close up the pipe end and pierce a number of holes through the pipe.

Thrust.

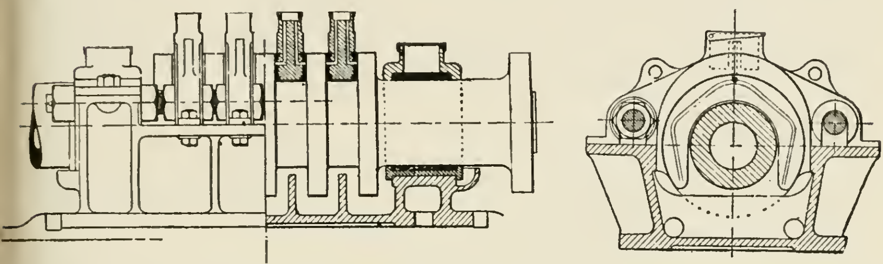
With a right or left hand propeller and the engines going ahead, the thrust is on the *after* side of the thrust block rings, and on the *forward* side of the shaft collars.

With a right or left hand propeller and the engines going astern the thrust is on the forward side of the thrust block rings, and on the after side of the shaft collars.

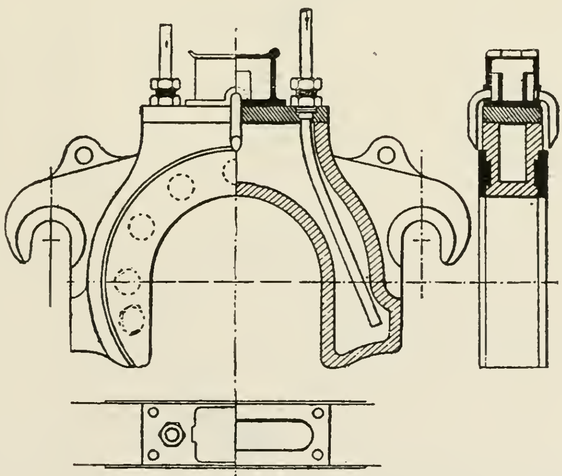
The total or combined effective horse-power of the engines come on the thrust block and is transmitted to the ship's hull through the holding down bolts.

The following are the points in connection with the thrust block demanding the most careful attention :—

1. Proper lubrication.
2. Proper adjustment of the rings.
3. Secure bolting down to the ship.



No. 47.—Thrust Block (Horse-Shoe Type), with bearing at each end



No. 48.—End View of Thrust, showing Oil and Water Service, &c.

Crank-Pin and Piston Travel.

The travel of the crank-pin is about one-half more than that of the piston, for while the piston travels two strokes (up and down), the crank-pin travels round a circle, the diameter of which is equal to the stroke.

The speed of the crank-pin is about one-half more than that of the piston, or in exact proportion to the extra distance it has to travel.

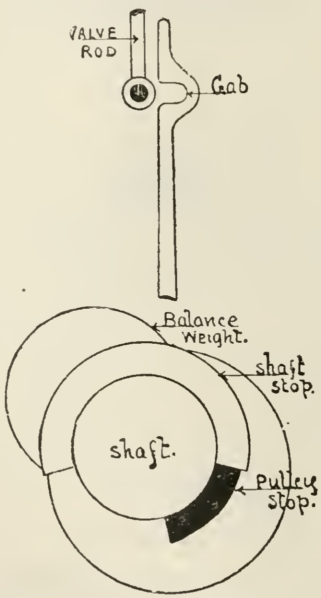
Flaws in L.P. Crank-Pin.

The after end of the L.P. crank-pin often develops flaws, usually due to the wearing down of the after lengths of shafting throwing heavy stresses on the L.P. shaft.

A built shaft is not so liable to flaws as a solid shaft, owing to the webs and pins being separate pieces; also with a built shaft, if flaws develop on the pin a new one can be fitted without entailing the condemnation of the whole shaft.

Loose Eccentric.

This type of single eccentric is not keyed on the shaft, but is loose circumferentially, and is driven round by a stop on the shaft, striking a corresponding stop on the pulley.

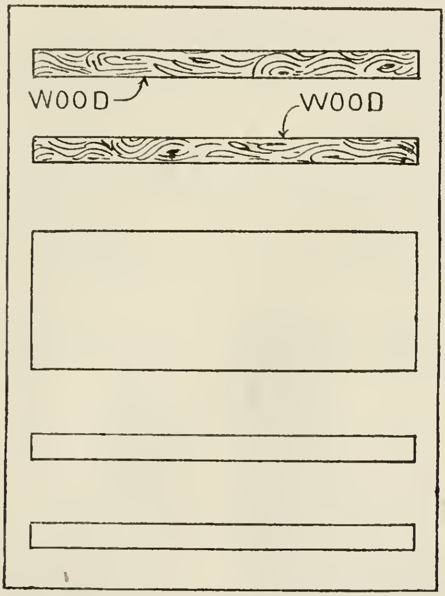


No. 49.—Single Eccentric.

The position of the centre of the pulley stop is found by the same method as is used in finding the key seat for ordinary eccentrics (see page 232).

The balance weight is to counterbalance the weight of the broad side of the pulley, and is fitted on the narrow side.

The top of the eccentric rod has a gab or clutch, which, when in gear, fits on a pin in the end of the valve spindle, and gives the motion to the valve. In reversing, this gab is thrown off the pin by a hand lever, and the valve is put in the reverse position by another hand lever, and when the shaft travels round, and the shaft stop strikes the pulley stop on the other side, the gab is again put in gear with the valve spindle, and the engine continues to run in that direction. A spring is often fitted on the rod to force the gab on to the pin.



No. 50.—L.P. Cylinder with Top Ports Closed Up.

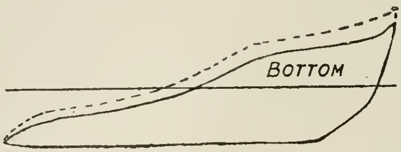
Broken L.P. Cylinder Cover.

If the L.P. cover breaks and cannot be repaired, the L.P. engine can be run on the atmospheric principle, that is, with atmospheric pressure on the top side, and steam on the under side.

To arrange for this, take off the L.P. valve chest cover, draw the valve, and drive wooden plugs into the two top steam ports, taking care that the plugs are clear of the face: then replace valve and cover and go ahead.

The atmospheric pressure will now assist the down stroke of the piston, and the steam pressure, as before, will act on the up stroke.

As the pressure carried in the L.P. chest is often only a few pounds above the atmosphere, the difference in pressure on each side of the piston will not, in most cases, be excessive.



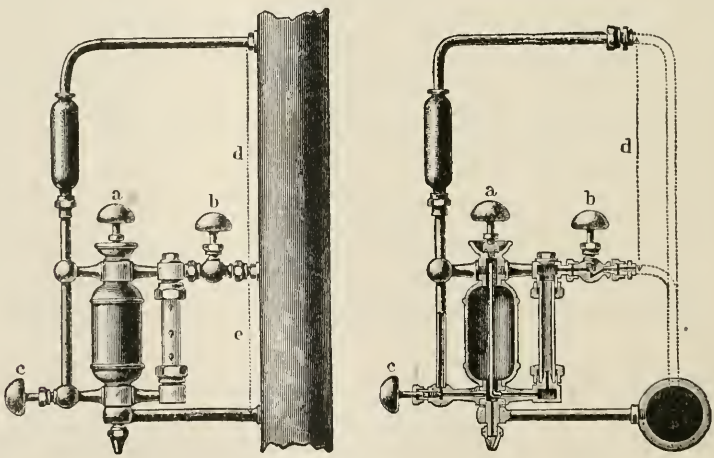
No. 51.—L.P. Card with Top Ports Closed Up.

With the top steam ports closed up the bottom ports will receive steam of a higher pressure than before, as shown by the dotted line, and the back pressure on the M.P. piston will be more in proportion. Observe that the atmospheric line now represents the top diagram.

Sight Feed Lubricator.

The principle of construction is that of displacement. Oil being lighter than water rises to the surface of the water, and is forced down the internal tube to the sight glass and steam chest.

NOTE.—The specific gravity of oil is ·9, and of water 1.



No. 52.—Improved Sight Feed Lubricator.
(By Messrs Schäffer & Budenberg Limited.)

Action.—Steam from the steam pipe condenses in the small condenser shown on the left of the sketches, and the water of condensation entering the bottom of oil chamber displaces the oil, which rises and flows down the small internal tube.

The oil then enters the sight tube by the nozzle shown, and passes up the sight feed tube in drops, from the top of which it is forced into the steam chest. Sometimes the condenser is formed of a copper coil placed a good height above the lubricator.

Instructions for Using.—The oil chamber is charged by unscrewing the plug *a*. The sight feed glass should then be filled up with water or a solution of salt and water (concentrated). To start the lubricator first open the valve *b*, then gradually open the valve *c*, and the drops of oil will be seen ascending through the water in the glass. Valve *c* regulates the amount of lubrication desired. When the engine is stopped, the two valves *b* and *c* must be closed.

Hydraulic Accumulator.

As will be seen from the sectional drawing, the accumulator piston is steam loaded by a reduced pressure of 80 lbs. per square inch. The pumping engine first of all pumps up the ram against the steam pressure, and when the piston reaches the top the pumps are automatically stopped by a rod and lever connected to the ram. The water is then stored up at a pressure of 800 lbs. per square inch and ready for use in the cranes, hoists, &c. The relative area of ram and piston being as 1 is to 10, a pressure of 80 lbs. per square inch on the one gives a pressure of approximately 800 lbs. per square inch on the other. As the water pressure is used in the cranes the ram and piston descend, until at a certain position the automatic gear in connection acts and starts the pumping engine: the ram is then raised back again to its former position.

NOTE.—The ram is packed by a leather ring which constitutes the best hydraulic packing yet discovered. The water pressure inside the ring forces it out against the ram and against the chamber.

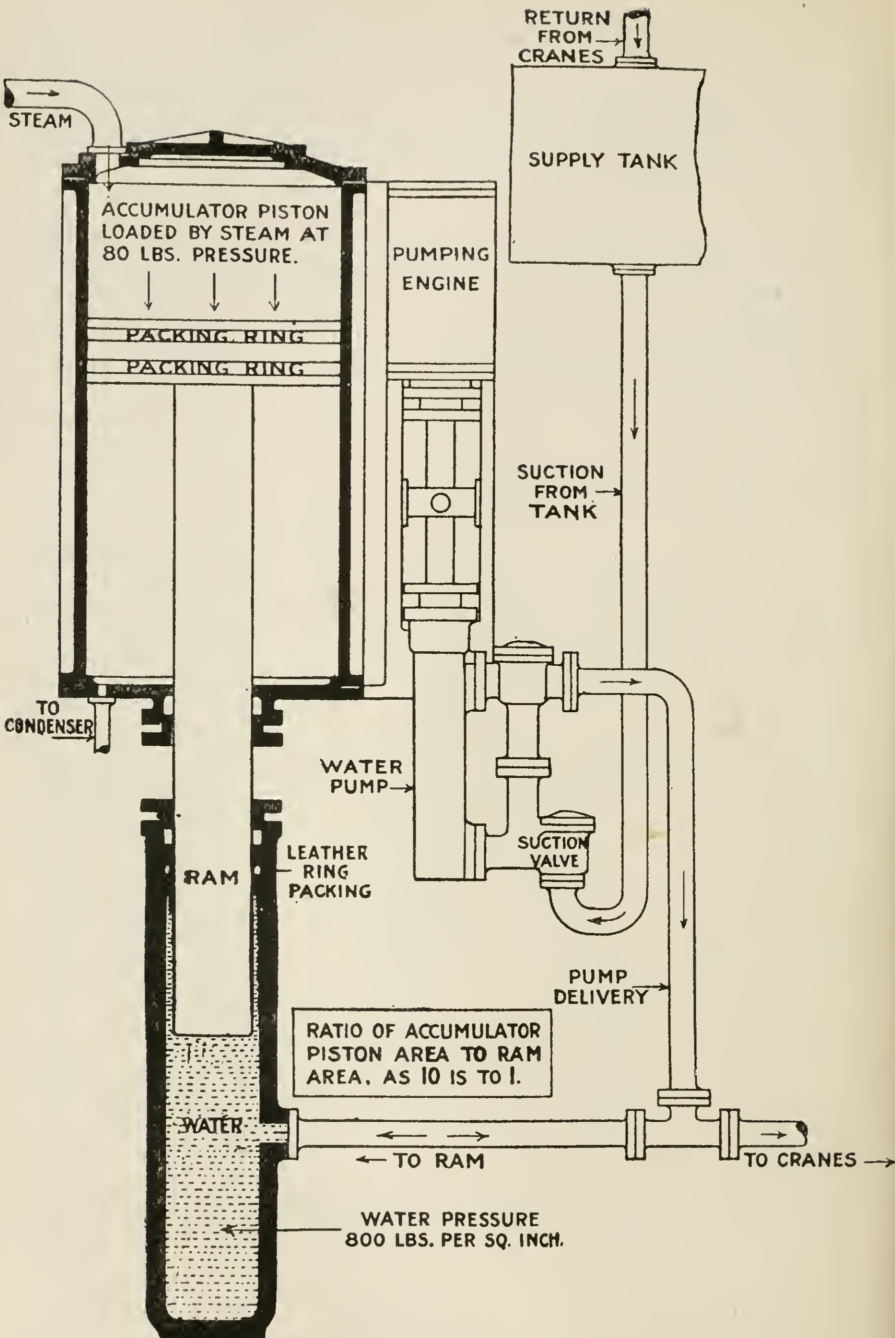
The crane consists of three rams, one large central lifting ram and two smaller side rams for slewing round the crane post.

The water is admitted by a hand valve to each ram as required, and after doing the work of lifting or slewing exhausts by return pipes back to the supply tank of the pumping engine (see sketch of accumulator).

When the water is admitted by the hand valve the ram is raised, and when exhausted the ram is lowered, but if the valve is put in mid position the ram is locked and therefore maintains the position it may be in at the time.

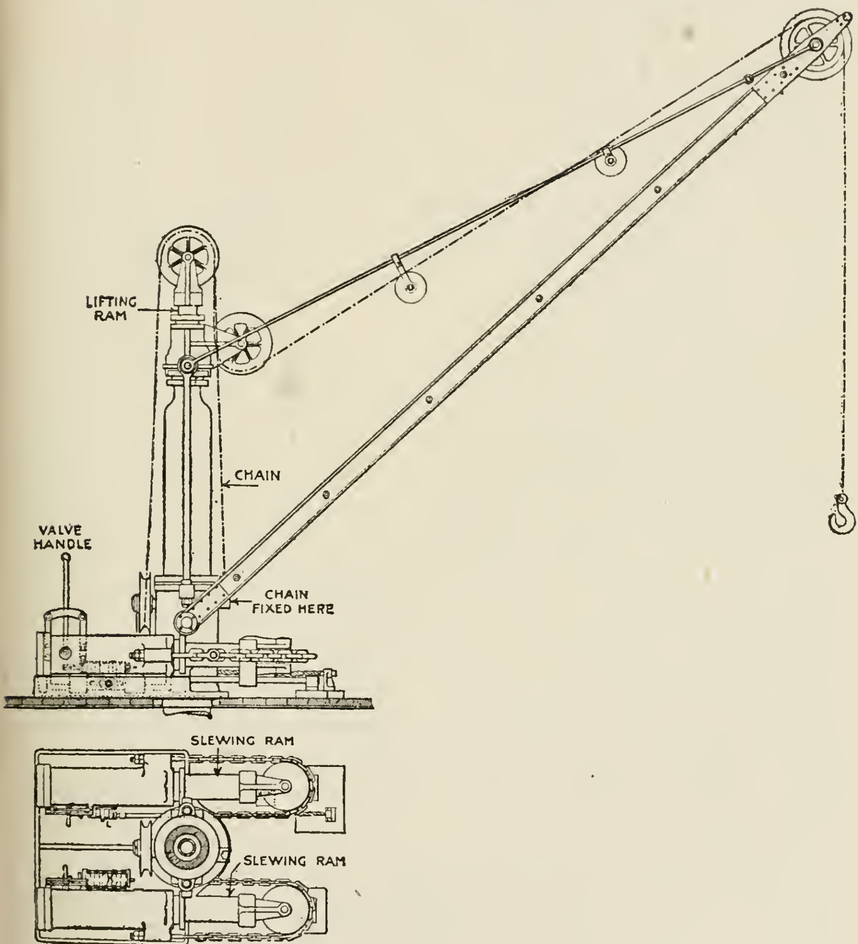
Advantages.

Among other advantages possessed by the hydraulic system, the following may be specially mentioned :—



No. 53.—Hydraulic Accumulator.
(Brown Bros., Edinburgh.)

(1.) Great smoothness of working.
(2.) Quickness of handling.
(3.) Absence of noise in working cargo; a consideration in passenger steamers.



No. 54.—Hydraulic Crane.
(Brown Bros., Edinburgh.)

Brown's Patent Combined Steam and Hydraulic Reversing Engine.

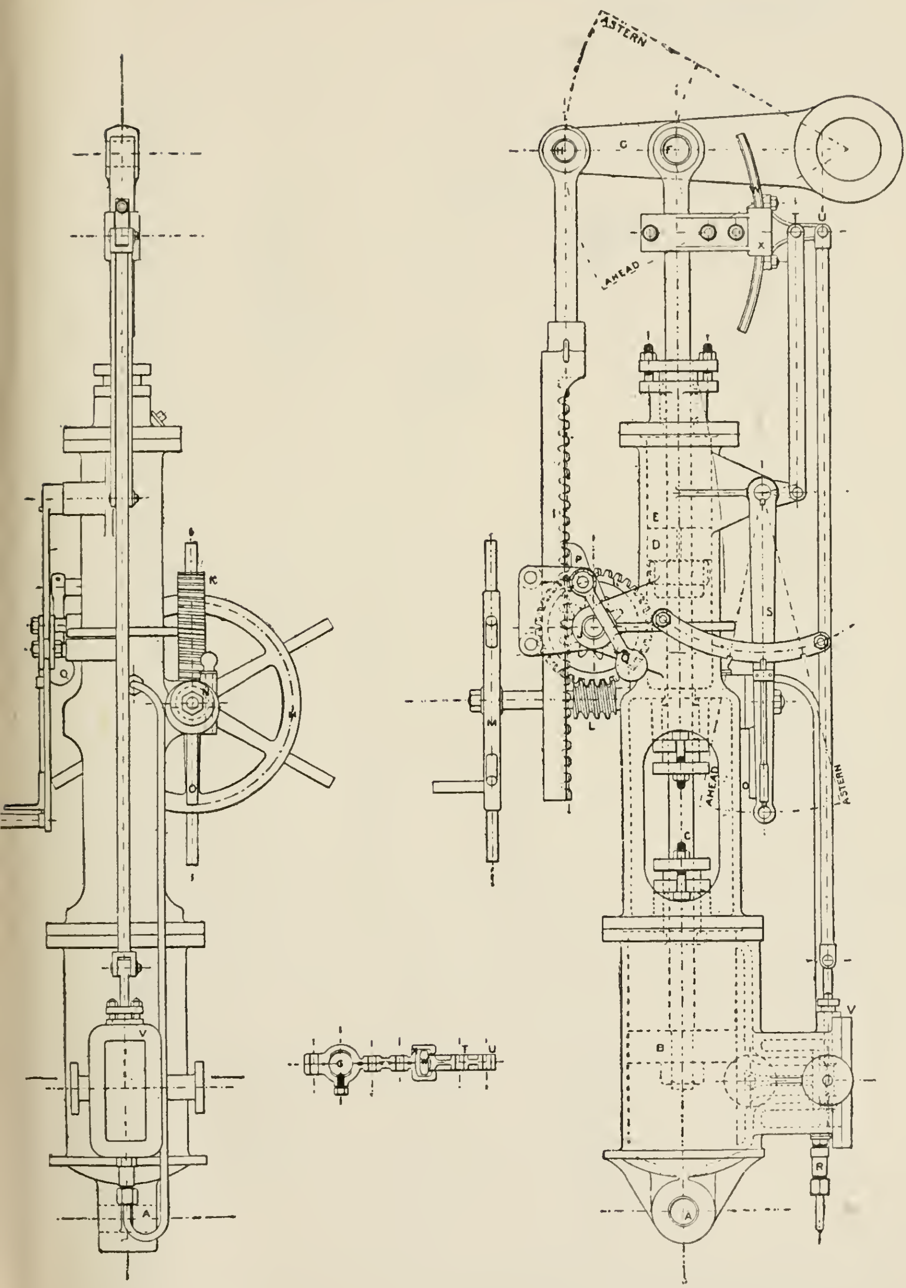
The engine as shown is attached to the bedplate or column of the main engine by the oscillating joint A formed on the end of the steam cylinder. In this cylinder is fitted a piston B with rod C, upon which is cottered a block piston D, working in the hydraulic cylinder E, the fluid being allowed to pass from one end to the other of the cylinder E by means of a small hole bored in the piston D. The rod C passes through stuffing boxes on the steam and hydraulic cylinders, terminating in a joint F, which lays hold of the weight shaft lever G. The lever is carried out to the joint H, upon which works the rod and rack I geared into the pinion J, both being shrouded to the pitch line. Upon the pinion shaft is keyed a worm-wheel K, which is actuated by the bronze worm L, this being revolved by the hand-wheel M. The worm and hand-wheel shaft are thrown out of gear with the worm-wheel K by the eccentric N, which is turned by the handle O, and held in position by the checkpin shown in dotted lines. When the hand-wheel and worm are disengaged, the rack and worm-wheel are free to revolve on the engine, making a stroke either way. This hand gear, therefore, forms no integral part of the starting engine, and is unaffected by any derangement of either hydraulic or steam cylinder or the steam valve.

A locking arrangement on the hand gear is provided, so that the main valve gear can be linked up in any position of the ahead stroke. This consists of a pawl P, which is made to engage the teeth of the rack, and the engine is held up against the pawl by means of the slide valve being left slightly open to steam. The pawl is provided with a balance weight Q, so that, on the engine being reversed for the astern position, immediately pulls the pawl out of gear.

The hydraulic cylinder is kept charged by means of the condensed steam in the bottom of the valve casing being driven by the steam through the non-return valve R, and led by a small copper pipe, as shown, into the lower end of the cylinder.

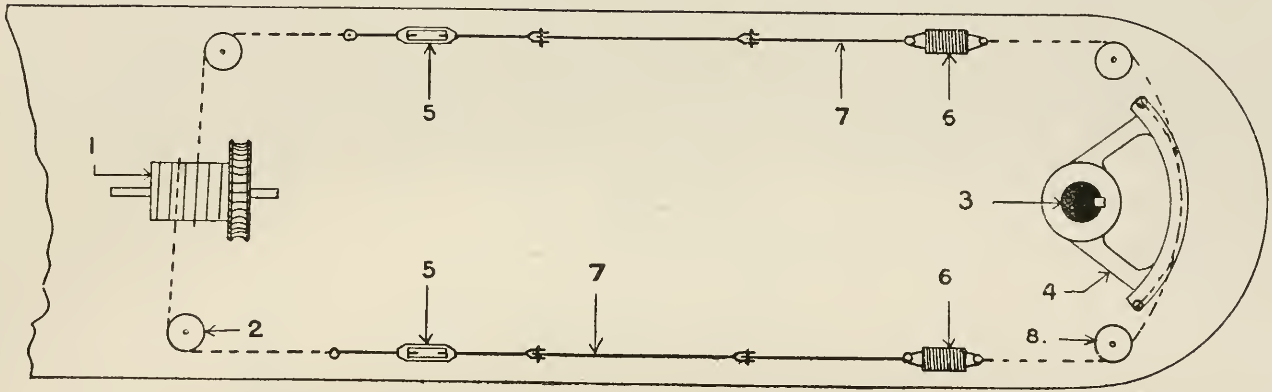
The engine is handled by a simple reversing lever S, which is connected by two links to the fulcrum of a lever at T. This lever has its end extended to U, which is connected direct by a link to the valve spindle V. A curved link W is securely attached to the lever T U, and on this there slides a block X, which is carried from the piston rod.

The action of this valve gear is as follows:—When the lever S is pulled into the ahead position, the lever T U, which is attached to the curved link W sliding in the block X, is depressed. The valve spindle is also moved down, opening the top end of the cylinder to steam. The piston rod now begins to move down, and carries with it the guide block X, which forces the end of curved link to move into the centre line of the block, thus moving the point U of the lever T U in an upward direction, the fulcrum at T by means of the reversing handle S being held stationary. In this way the valve spindle V is brought back into its original position, thus bringing the engine to rest. The same operation is performed for the astern or any intermediate position.



No. 55.—Brown's Patent Combined Steam and Hydraulic Reversing Engine.

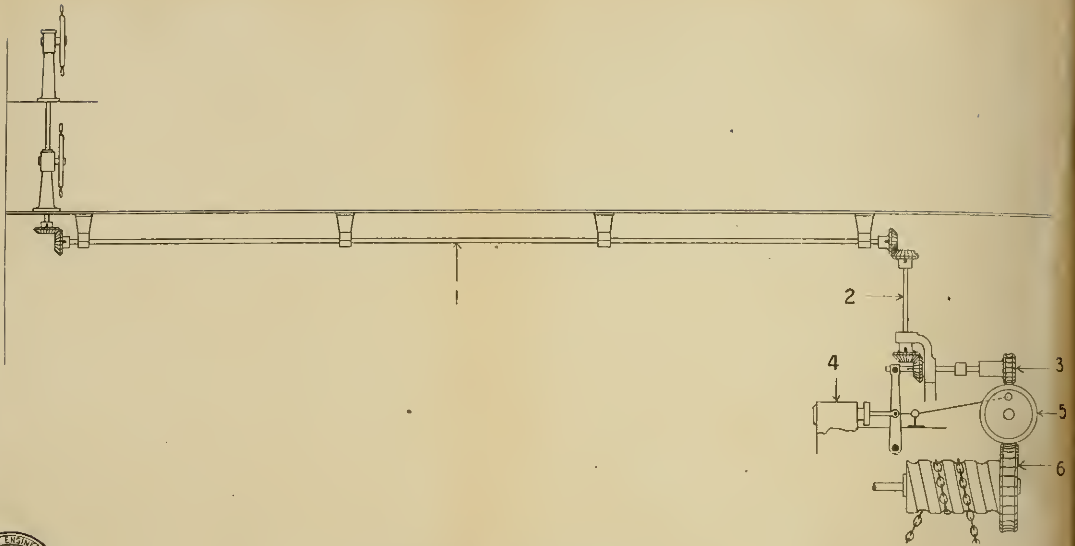
Steering Gears.



No. 56.—Plan of Steering Gear.

1, Chain drum.
2, Guide pulley.
3, Rudder stock.
4, Rudder quadrant.
5, Stretching screws.
6, Springs.
7, Steering rods.
8, Guide pulley.

NOTE. — The stretching screws are fitted so that the "slack" of the chains and rods may be taken up when required. The springs are for reducing shock and for easy transmission of the engine power to the rudder. Observe that chains are fitted at both the engine end and rudder end of the gear, the ends of the chains aft being hooked to the rudder quadrant.



No. 58.—Steering Engine Transmission Gear.

1, Horizontal shaft connecting steering wheel and engine.
2, Vertical shaft connecting (by bevel wheels) to control valve lever.
3, Hunting or return gear from engine worm to control valve.
4, Control valve.
5, Engine worm.
6, Worm wheel of chain drum.

NOTE.—The small spindle connected to the control valve lever gears into wheel 3 by a screw.

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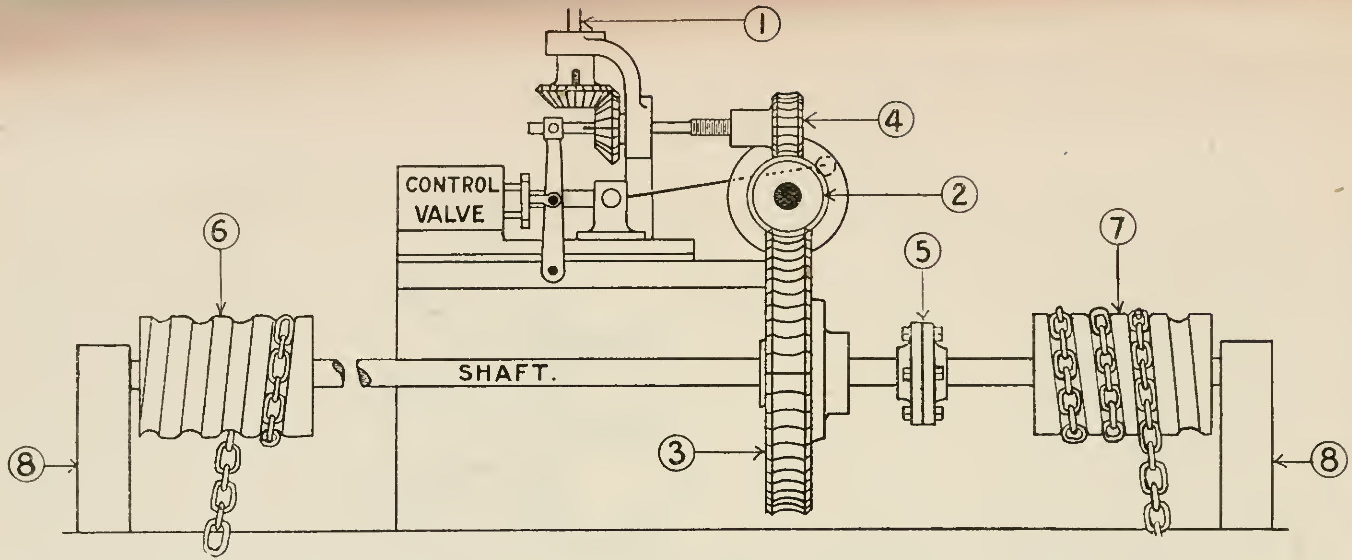
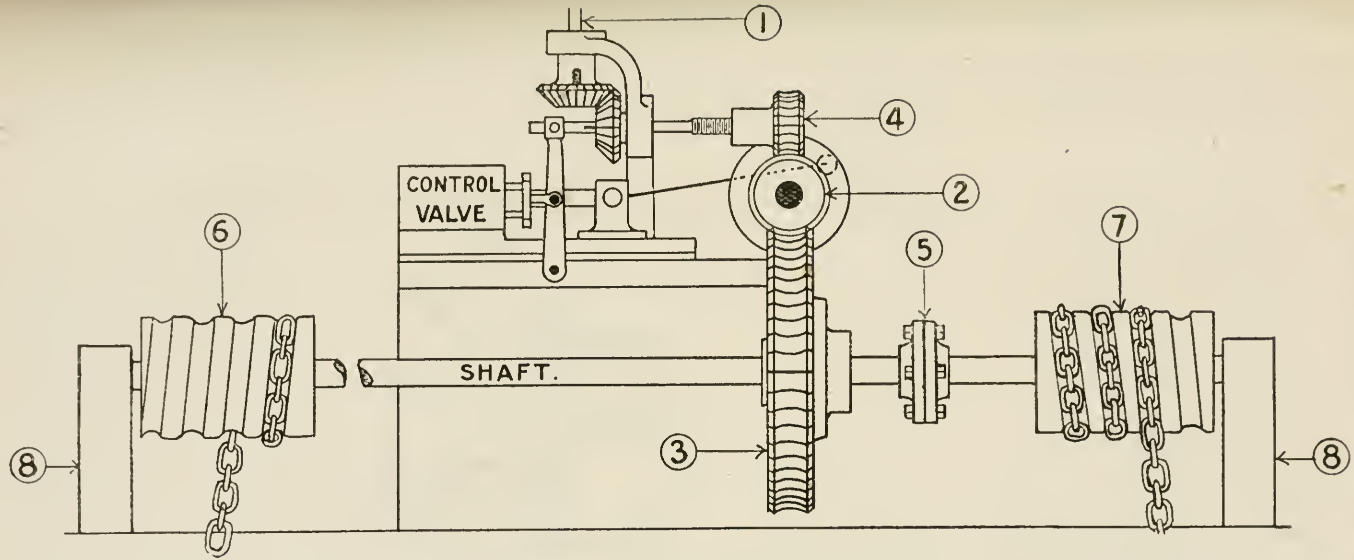


No. 57.—Modern Type Steering Gear, with Double Drum and Direct Leads.

1, Vertical shaft gearing by bevel wheels with control valve lever.
2, Engine worm.
3, Worm wheel on drum shaft (keyed and bolted).
4, Hunting or return wheel from worm wheel to control valve.
5, Coupling (one on either side).
6, Port chain drum.
7, Starboard chain drum.
8, End bearing.

NOTE.—As the chain coils round one drum, it uncoils on the other drum.



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3, Worm wheel on drum shaft (keyed and bolted).
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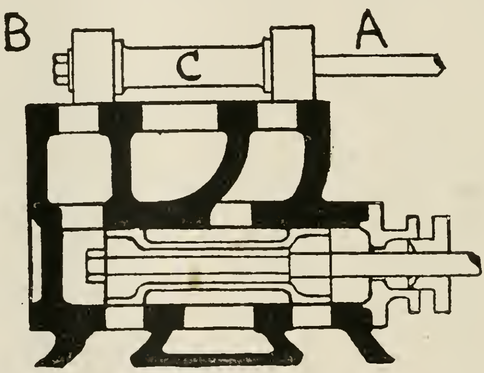
5, Coupling (one on either side).
6, Port chain drum.
7, Starboard chain drum.
8, End bearing.

NOTE.—As the chain coils round one drum, it uncoils on the other drum.

Steering Gear Engines.

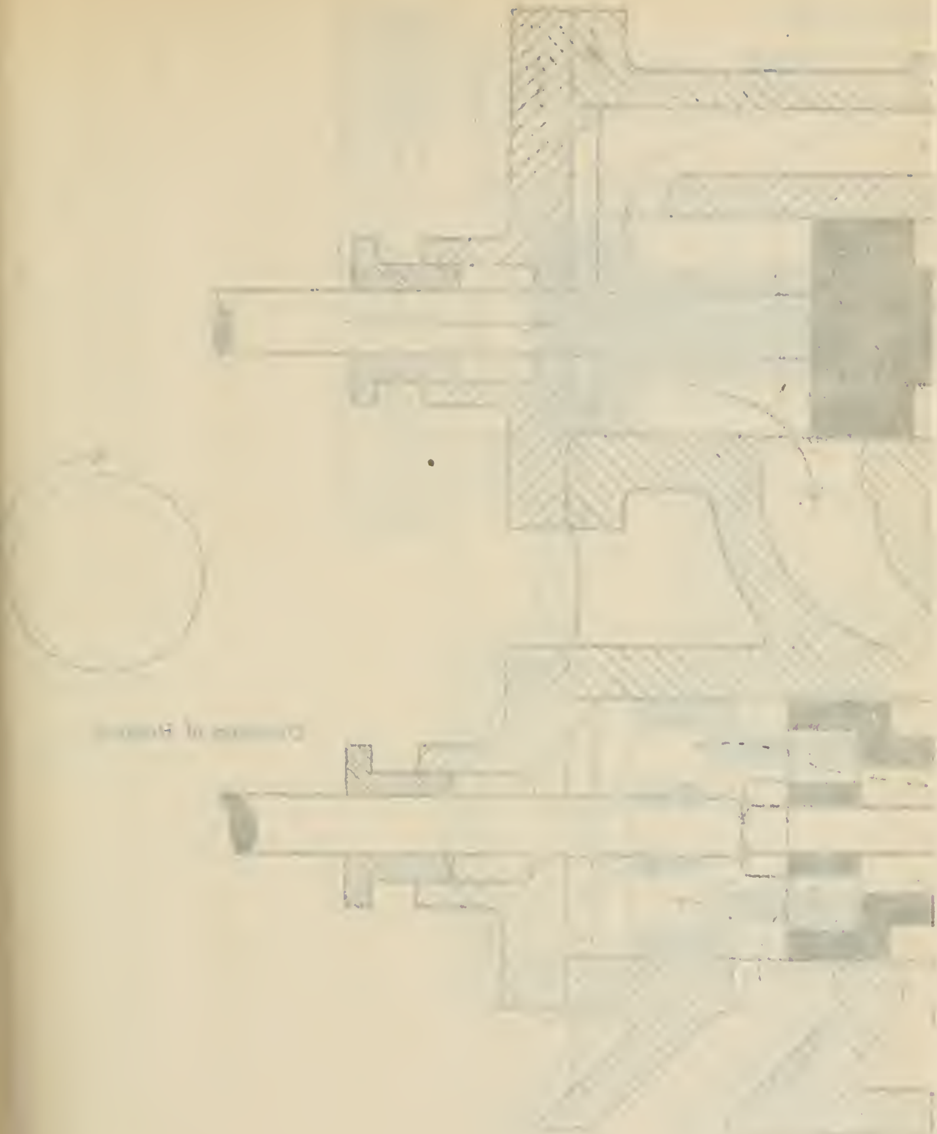
The majority of patent steering gear engines are fitted with three valves—a central control valve and two piston valves or slide valves, one for each cylinder of the engine. The control valve distributes the steam to the engine valves so that the gear may run either to port or starboard as required, and this being the case it will be observed that each piston valve or slide valve requires only one eccentric, the control valve acting as the reversing gear. The piston valves or slide valves have little or no steam lap, so that the steam is carried for the full length of the stroke, and to allow of this the eccentric keyseats are cut at right angles to the cranks.

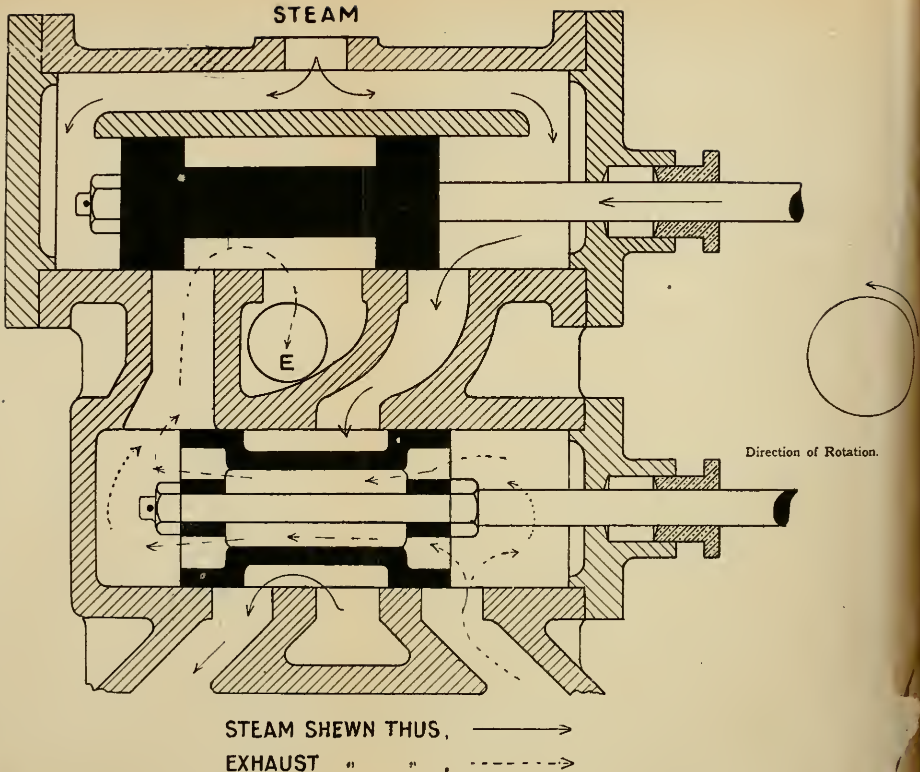
Control Valve.—This valve is sometimes a flat valve, but more generally a round or piston valve. It is operated directly (1) by hand from the steering wheel on the bridge, and (2) automatically by a counteracting return gear from the chain drum or crank-shaft of the steering engine.



No. 59.—Steering Gear Engine Valves.

When the engine is stationary, if the hand-wheel is turned, the control valve opens and starts the engine; this again has the effect of bringing into play the return gear in connection with the crank-shaft, so that as the hand gear (if still moving) tends to open the control valve the return gear tends to close it: if both gears, hand and automatic, run at the same speed the control valve remains open, but if the hand gear is stopped the automatic gear shortly afterwards brings the engine to a stop by closing the control valve. From this it will be seen that to keep the steering gear running the hand-wheel must be kept in motion, otherwise the automatic gear will bring the control





E. EXHAUST PORT.

No 59a.—Steering Gear Valves.

Steam is being admitted to the cylinder from the centre of piston valves, and is exhausting to ends of same, back to exhaust port E, and the direction of engine rotation is as shown.

NOTE.—In this type of gear the steam is admitted from the ends of the control valve for both directions of running, as clearly shown in the sketches.

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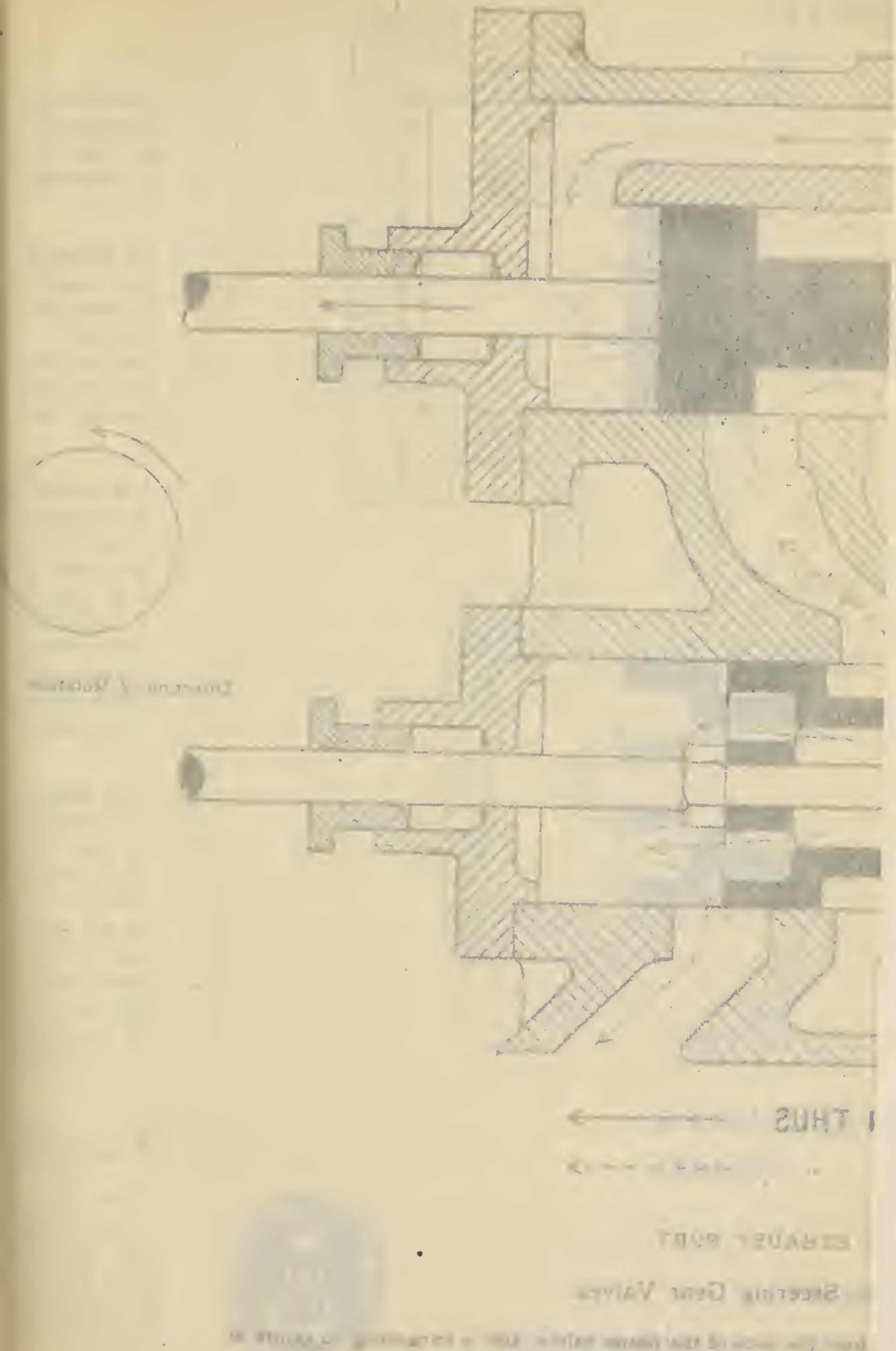

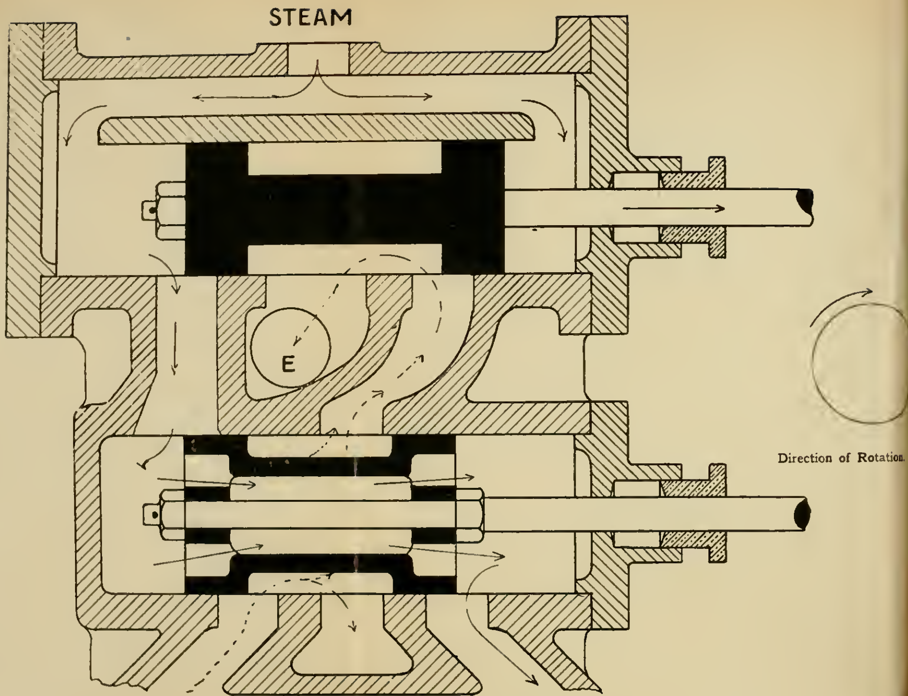

STEAM SHEWN THUS, ——————→

EXHAUST " " , - - - - - - - →

E. EXHAUST PORT.

No. 59b.—Steering Gear Valves.

Steam is being admitted to the cylinder from the ends of the piston valves, and is exhausting to centre of same back to exhaust port E; the directioo of engine rotation is as shown, and is now reversed.

[*To face page* 343.

valve back to mid-position and thus stop the gear. As before stated, the control valve does away with the necessity of having two eccentrics for each valve, one only being required, as the reversal of rotation is obtained by the action of the control valve.

Engine Valves.—The valves of the engine are generally of the hollow piston type, although in some cases special flat valves are used, as in the gear of Messrs Alley & M'Lellan.

The hollow piston valves are arranged so as to receive steam at the ends and exhaust in the centre, or, to receive steam at the centre and exhaust at the ends, the ports being suitably cast to admit of this (see sketch of control valve).

Action of Valves.—If the steam is admitted to the ends of the piston valves by the control valve C being moved in the direction of A, the cylinders obtain the steam from the ends and the exhaust takes place in the centre of the piston valves, the engine running so that the rudder is brought over to, say, the port side; but if the steam is admitted to the *centre* of the piston valves, by the control valve C being moved in the direction B, then the cylinders receive steam from the centres of the valves and exhaust at the ends: the direction of motion being thus reversed, the rudder is brought over to the centre again and so to the starboard side.

Types of Gear.—The following five types of steam steering gear engines in general use will give a very good idea of the principle of working and of the mechanism employed by some of the best makers of this important piece of auxiliary machinery.

It will be noticed that, with the exception of Messrs Hastie's gear, the control valve is moved laterally to open and close the ports, but in the gear produced by Messrs Hastie & Co. the control valve *moves round* somewhat after the manner of a Corliss valve; the spindle therefore of this gear does not move laterally, but simply revolves for part of a ircle, as will be seen by examining the sketch.

Steering Gear by Messrs Caldwell & Co.

In this gear the control valve is opened or closed by a cam in connection with the "sun and planet" motion contained in the brass casing. The toothed wheel A is in connection with the hand-wheel on the bridge, and the toothed planet wheel B is in one with the cam, while the toothed casing C is in connection by bevel wheels with the chain drum.

of Rotation.

page 343.

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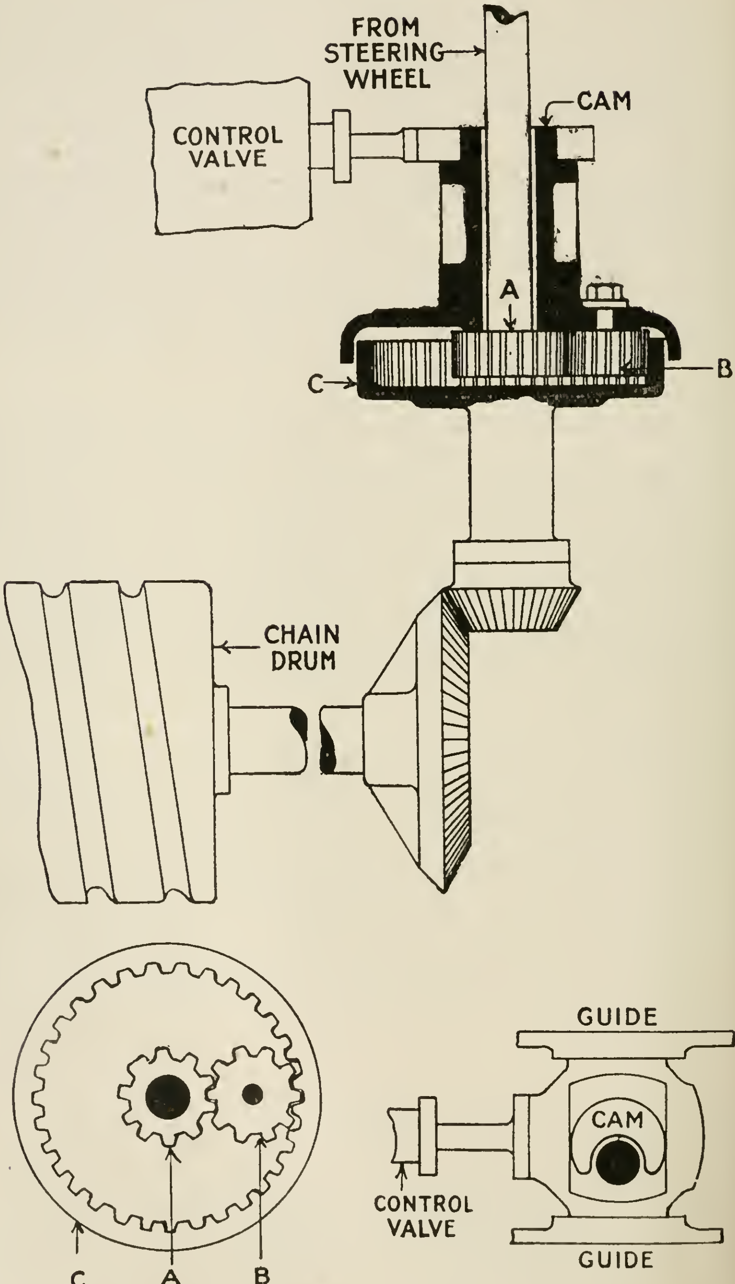
It will be noticed that, with the exception of Messrs Hastie's gear, the control valve is moved laterally to open and close the ports, but in the gear produced by Messrs Hastie & Co. the control valve *moves round* somewhat after the manner of a Corliss valve; the spindle therefore of this gear does not move laterally, but simply revolves for part of a ircle, as will be seen by examining the sketch.

Steering Gear by Messrs Caldwell & Co.

In this gear the control valve is opened or closed by a cam in connection with the "sun and planet" motion contained in the brass casing. The toothed wheel A is in connection with the hand-wheel on the bridge, and the toothed planet wheel B is in one with the cam, while the toothed casing C is in connection by bevel wheels with the chain drum.



Sun and Planet Motion.

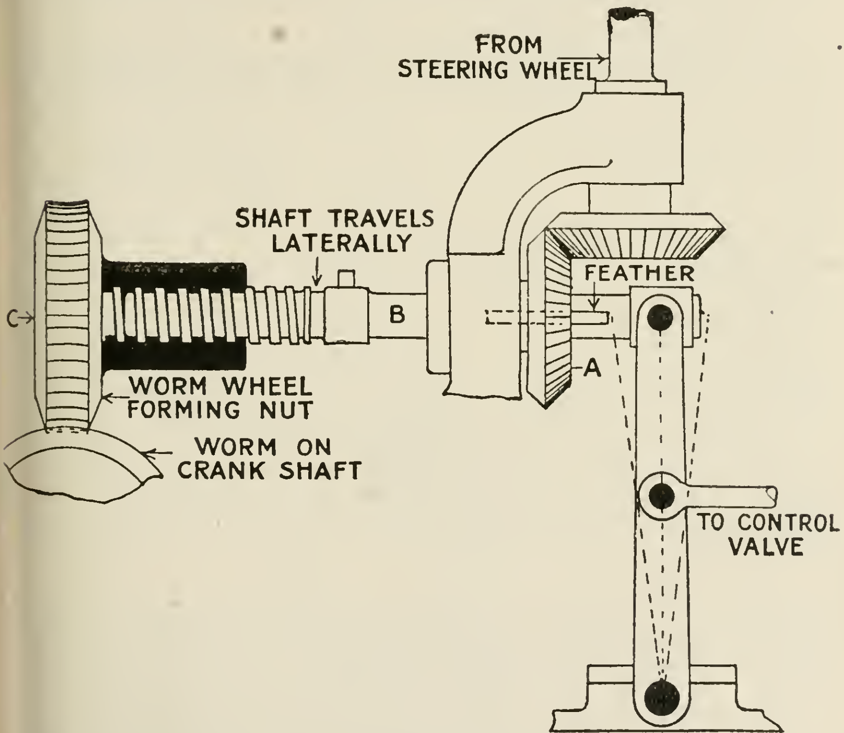
No. 60.—Steering Gear by Messrs Caldwell & Co.

Action.—(1.) If wheel A is moved round by the steering wheel with C stationary, B moves round the casing and opens the control valve, thus starting the engine and setting C in motion.

(2.) If A is stopped, C moves round in the reverse direction, carrying B and the cam back again until the control valve is brought to mid-position and the engine stopped.

(3.) If A and C move at the same relative speed the control valve will remain open and the engine keep running, but if the speed of C exceeds that of A the control valve will close and the engine stop.

In the same way, to open the control valve further the speed of A must exceed that of C.



No. 61.—Steering Gear by Messrs Bow, M'Lachlan, & Co.

Steering Gear by Messrs Bow, M'Lachlan, & Co.

In this gear the control valve is moved horizontally by the spindle B turning in the nut wheel C and so acting on the lever in connection with the control valve spindle to move it from mid-position to the right or to the left as required. The bevel wheel actuated from the steering wheel turns the spindle B, which being screwed into the nut C causes a lateral movement. The feather shown on the spindle allows the spindle to move horizontally without turning, if actuated by the nut wheel C travelling round.

Action.—(1.) If wheel A is moved round by the steering wheel with C stationary, the spindle B turns and moves either in or out of the nut wheel C and opens the control valve, thus starting the engine and setting wheel C in motion.

(2.) If A is stopped, C moves round and causes the spindle to travel (without turning) back again to mid-position of the control valve, thus stopping the engine. The feather and slot referred to allow of this taking place.

(3.) If A and C move round at the same relative speed the control valve will remain open and the engine keep running, but if the speed of C exceeds that of A the control valve will close and the engine stop.

In the same way, to open the control valve further the speed of A must exceed that of C.

Steering Gear by Messrs Davis & Co.

In this gear an expansion or regulating steam valve is fitted in addition to the control valve. The expansion valve is operated by a nut, which travels up or down the vertical shaft connecting the steering wheel and control valve spindle and opens the expansion valve a certain amount on either side, admitting the steam to the control valve in proportion to the amount of work to be done.

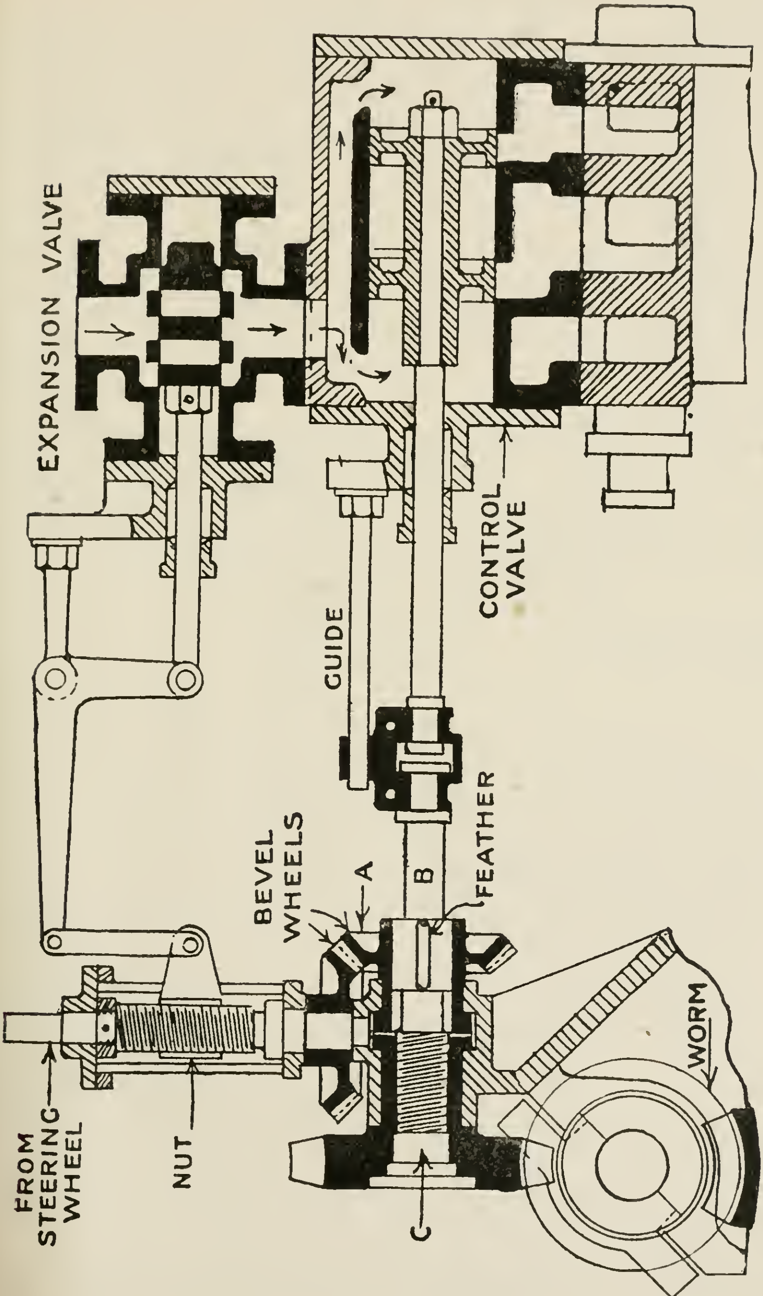
The spindle B of the control valve works in or out of the nut wheel C as required, and admits the steam to the main valves of the engine. The wheel A is connected to the spindle B by a slot and feather which allows of a lateral movement of the spindle B when it is not turning.

Action.—(1.) If wheel A is moved round by the steering wheel the spindle B moves either in or out of the nut wheel C and opens the control valve, thus starting the engine and setting wheel C in motion.

(2.) If A is stopped C moves round and causes the spindle B to travel (without turning) back again to mid-position of the control valve, thus stopping the engine. The feather and slot referred to allow of this taking place.

(3.) If A and C move round at the same relative speed the control valve will remain open and the engine keep running, but if the speed of C exceeds that of A the control valve will close and the engine stop.

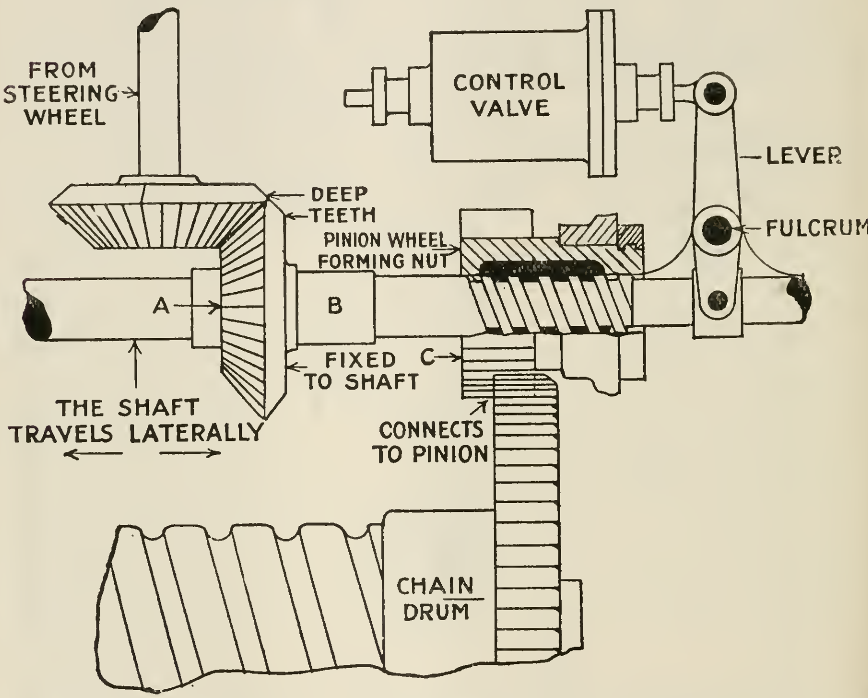
In the same way, to open the control valve further the speed of A must exceed that of C. The expansion valve is arranged to open when full steam is required in the engine.



No. 62.—Steering Gear by Messrs Davis & Co.

Steering Gear by Messrs Alley & M'Lellan.

In this gear the spindle B moves horizontally for a small distance in or out of the teeth of the bevel wheels, which are cut specially deep to allow of this, and, acting on the lever in connection with the control valve spindle, moves the valve to right or left as required. The small travel of the spindle B is increased at the control valve



No. 63.—Steering Gear by Messrs Alley & M'Lellan.

by the leverage obtained. The pinion wheel C, which gears with the chain drum wheel, forms the nut in which the spindle travels.

Action.—(1.) If wheel A is moved round by the steering wheel with C stationary, the spindle B moves horizontally to right or left and opens the control valve, thus starting the engine and setting C in motion.

(2.) If A is stopped C moves round, causing B to move back again until the control valve is brought to mid-position and the engine is stopped.

(3) If A and C move round at the same relative speed the control valve will remain open and the engine keep running, but if the speed

of C exceeds that of A the control valve will close and the engine stop.

In the same way, to open the control valve further the speed of A must exceed that of C.

Steering Gear by Messrs Hastie & Co.

In this gear the control valve moves round on its axis instead of travelling laterally as in the others, and so opens the ports to the centre or to the ends of the engine valves. The control valve mechanism consists of three bevel wheels, one of which travels round the other two and operates the rolling quadrant connected to the control valve. The left-hand bevel wheel of the three is keyed to the rotating spindle actuated by the steering wheel, and the right-hand bevel wheel is keyed to the worm-wheel, which, observe, is clear of the spindle. The top bevel wheel B is free, and is one with the rolling or "tumbling" quadrant which connects by teeth to the control valve spindle to move it round to right or left as required.

It will thus be seen that the top wheel B travels *round* between the teeth of the other two bevel wheels. With B at top the control valve is in mid-position and therefore closed, but if it travels down on either side the control valve is opened.

Action.—(1.) If A is moved round by the steering wheel with C stationary, wheel B travels round and down between the other two bevel wheels and opens the control valve, thus starting the engine and setting C in motion.

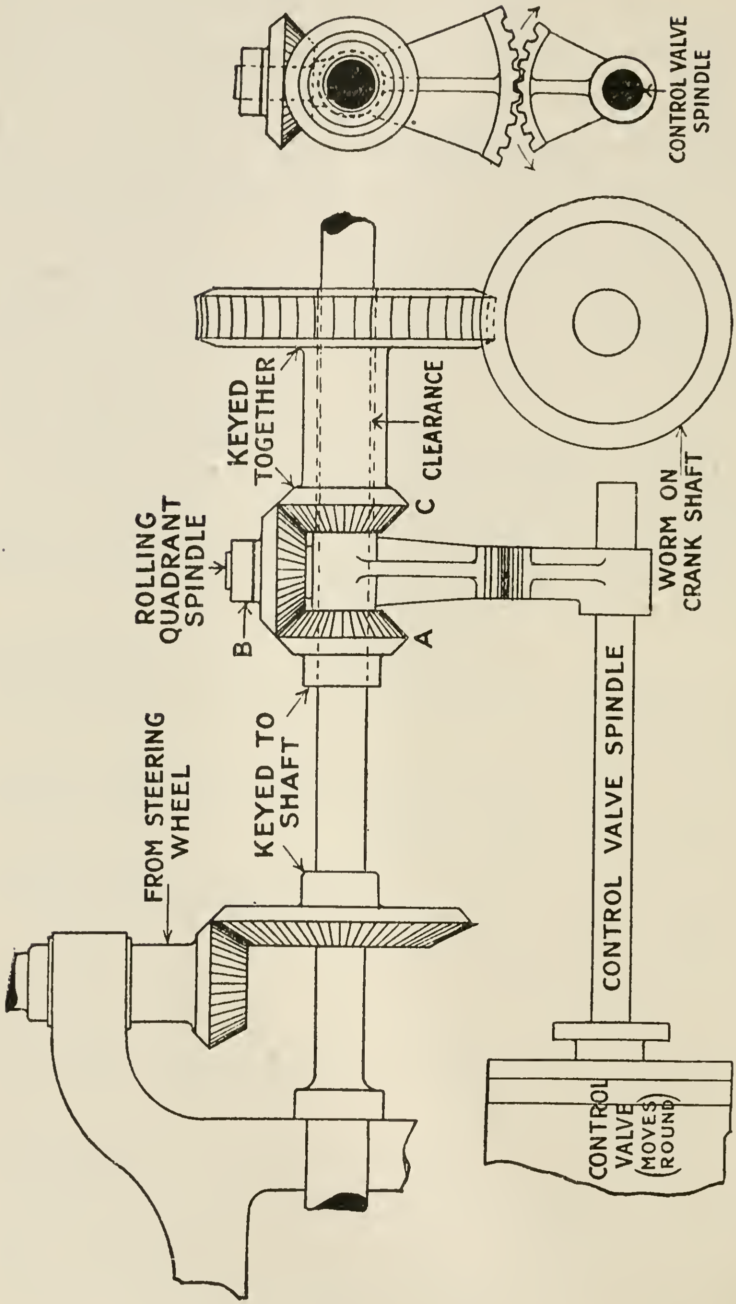
(2.) If A is stopped C moves round in the reverse direction, causing B to travel back and up again to the top position, so that the control valve is closed and the engine stopped.

(3.) If A and C move round at the same speed one counterbalances the other, and the control valve will remain open and the engine keep running, but if the speed of C exceeds that of A the control valve will close and the engine stop.

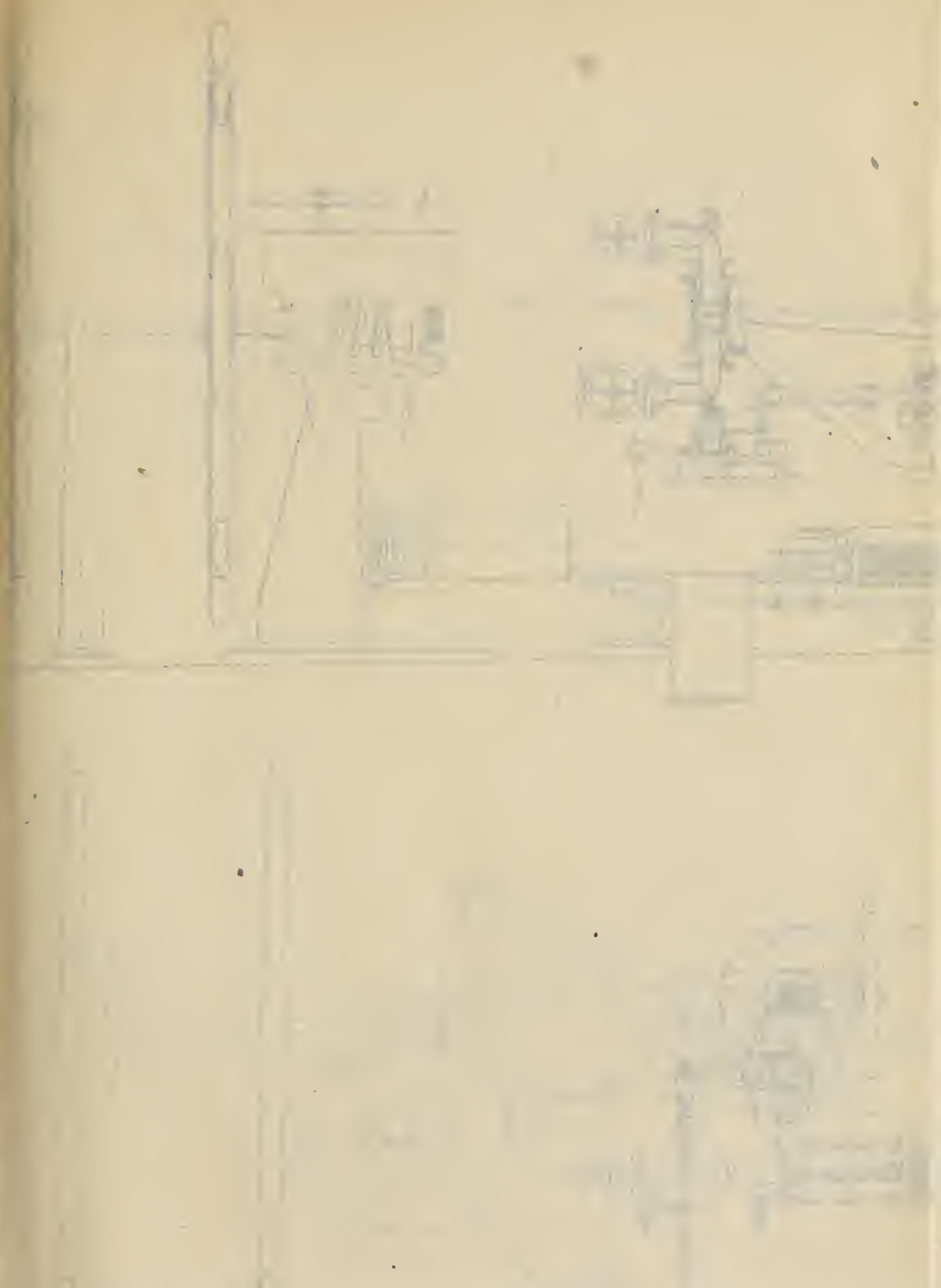
In the same way, to open the control valve further the speed of A must exceed that of C.

NOTE.—In all of the foregoing types of steering gear engines a limit is fixed to the running of the engine by means of "stop" arrangements, which prevent damage to the steering gear in general. The "stops" check the travel of the rudder chains beyond a fixed point.

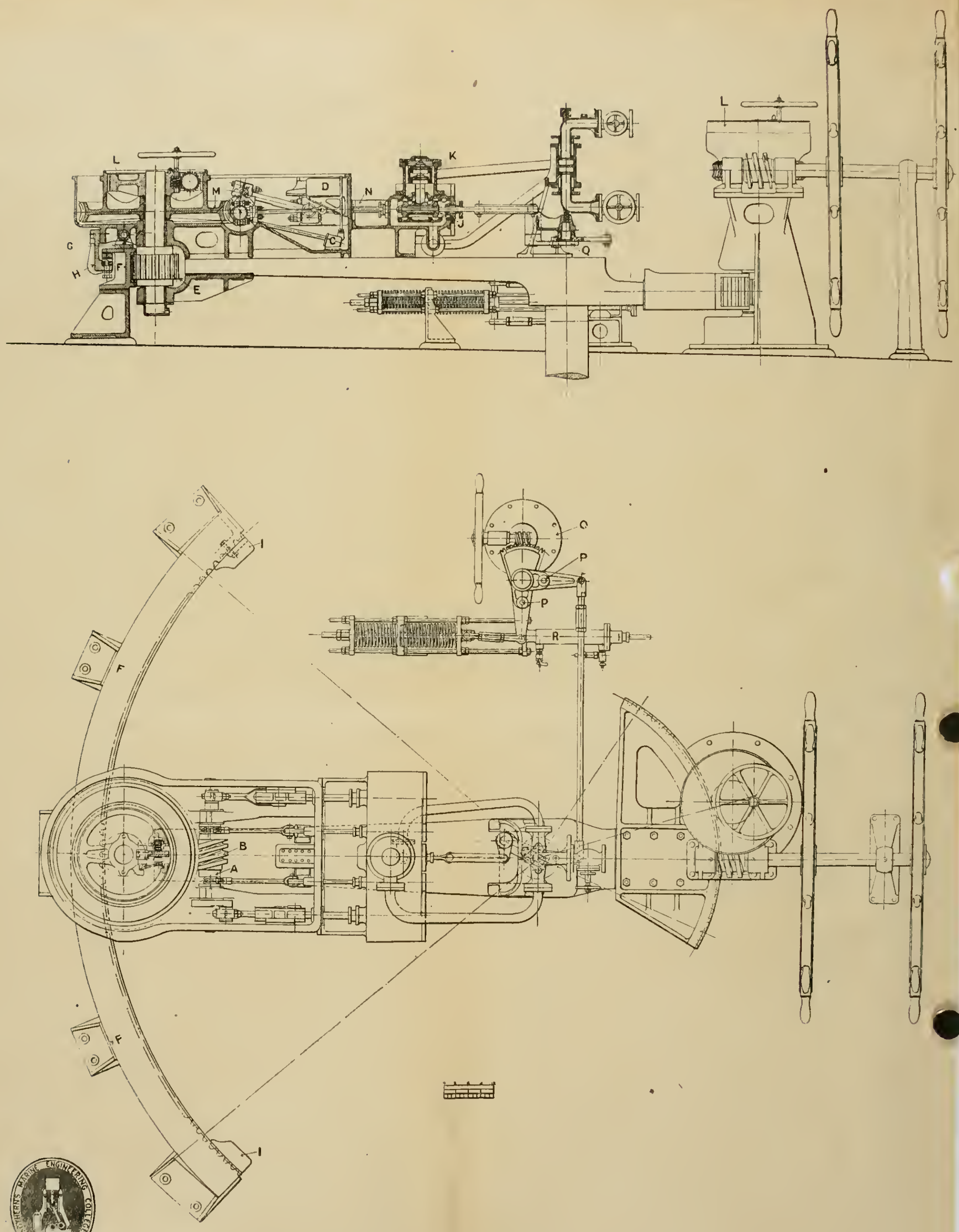
It should be noted that when the steamer is going ahead, say at full speed, with the rudder hard over to either port or starboard, the friction of the steering gear prevents the chain barrel from revolving, and so allowing the rudder to slip back again to mid-position. This will perhaps be understood when it is remembered that though the worm may cause the worm-wheel to revolve, it is almost a mechanical



No. 64.—Steering Gear by Messrs Hastie & Co.



a greater angle of rudder can be arranged for, if necessary. This does away with any need for disconnecting the control gear when using the hand or other gear for moving the rudder.



No. 65.—Brown's Patent Steam Tiller (Direct Geared Type).

impossibility for the worm-wheel to cause rotation of the worm, in fact, the teeth would give way first.

Supposing, however, that the worm-wheel and barrel *did* revolve and cause the worm to go round, the effect would be to set in operation the control valve, which, opening, would then start the steering engine to run in the opposite direction, and thus prevent further movement of the chain barrel.

NOTE.—By having two cylinders and the valves for each without steam lap, certainty of action is ensured, as one or other of the cylinder ports will always be open to receive steam, and so effect instant starting of the gear. Also, the key seats being at right angles to the crank, this allows the engine to run either way as required.

Description of Brown's Patent Steam Tiller.

(Direct-Geared Type.)

This type of gear, which contains many improvements over the older designs of direct gears, has been introduced specially for ships that require a smaller power of gear than the 1905 design with countershaft. The general arrangement is similar to the old direct gear, except that the worm-wheel has been considerably increased in diameter, and the worm is now double-threaded. The oil pumps are worked direct from the eccentrics, and placed in a well at the bottom of the engine pan. The friction clutch is increased in size, so as to make it more powerful. Brown's patent economic valve, which absolutely shuts off steam every time the engine comes to rest, is also fitted. The great advantage possessed by this valve is that, should dirt or any foreign matter get in so as to obstruct its working, it can only remain in the open position, so that the gear does not become disabled, and would work under the same conditions as an ordinary steering gear without this attachment. The cut-off gear Q embodies the 1907 patents, with which no stops on the valves are ever reached, thus preventing any strain or damage to the gear by forcing over the telemotor when no steam is on the engine. This gear gives a very quick opening at the commencement of the movement of the control gear, and the motion of the valve is gradually reduced to practically nil, though the control gear moves uniformly. The same thing occurs in closing—that is, it commences to close very slowly, and just at closing the motion is very quick and decided. This gives a very fine and delicate control of the gear without any danger of reaching the stops on the control valve or its gear. With the standard design and a rudder angle of 40°, it is possible, when there is no steam on the gear, for the tiller to be hard over in one direction while the telemotor is pulled hard over to the other; and a greater angle of rudder can be arranged for, if necessary. This does away with any need for disconnecting the control gear when using the hand or other gear for moving the rudder.

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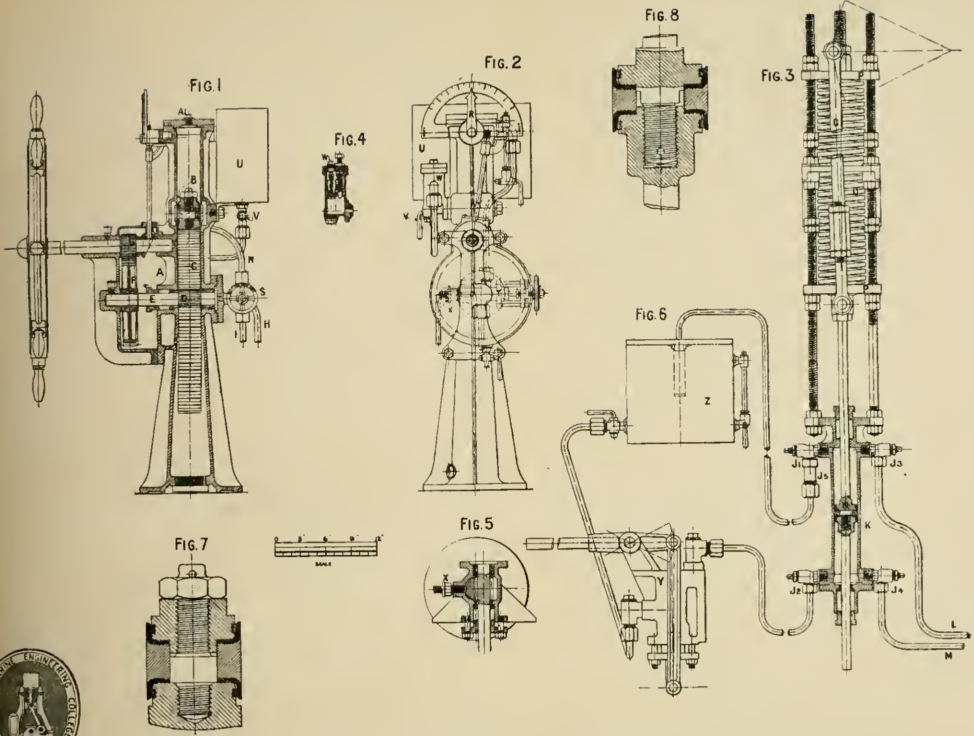
The crank-shaft A is forged from high-tensile steel ; and the worm B is cast on to it from a special hard and tough bronze. The valveless oil pumps C are placed in a well, and discharge up into the tank D, from whence the oil is carried by small brass pipes to the various bearings, guides, &c. The main pinion E is machine cut from a forging of high-tensile steel, and is solid with its shaft. The rack F, having machine-moulded knuckle teeth of special design, is made in halves, which are interchangeable, bolted together, so that, should the teeth in the middle get worn through long usage, the two ends can be turned in to the centre, the worn portions going of course to the outside. The slipper G slides on the top of the rack, and helps to carry the weight of the tiller, engine, &c. The slipper or friction block H is made of cast steel. The slides on inclined planes for about 10° each side of midships, being tightest at the centre ; and it has been found to hold the tiller quite steady, and act perfectly. The stops I attached to the ends of the rack fit into the teeth, and are secured by through bolts, so that they can be moved when the rack is changed. The form of control valve J, fitted with this gear, is of the well-known piston type, having one of Brown's patent economic valves K fitted on top. The worm-wheel M forms part of the friction clutch casing, and has its teeth cut in a special machine, so that great accuracy is obtained. The distance piece between the cylinders N has been increased in length, so that no part of the piston rod that goes into the engine pan passes into the steam cylinders or the packing in the stuffing boxes. This prevents oil being taken over from the engine pan into the cylinders, and thus into the feed water.

The hand gear has been redesigned and brought up to date. Both the steam and hand gears are connected and disconnected by the large friction brakes L, which give every satisfaction, and quite take the place of the separate brakes frequently fitted. As the "cut-off" is of the floating type, the steam and hand gear can be put in and taken out in any position of the rudder, it not being necessary to connect them in the position in which they were disconnected, or bring the gear to the rudder.

The mechanical standard O has also been considerably modified in design and brought up to date. This is, of course, for use in lieu of the telemotor, or when it is desired to control the steering gear aft. When changing from the telemotor to this standard, or *vice versâ*, the connection can be made in a second or two, as all that is necessary is to take out a pin from one hole (marked P) and put it in the other, according to which change is being made. These pins are of brass, and have a large eye at the top, with a tapered point for easy entrance, so that they can be easily unshipped and put in.

R is the receiving cylinder of the telemotor installation, and is of the latest design, with patent single spring.

This gear can, of course, be arranged to work with control shafting in place of the telemotor, if desired ; but as the type of telemotor is



No. 66.—Brown's Patent Hydraulic Steering Telemotor.

[To face page 352.

thoroughly reliable, its adoption can be recommended in every case where the distance from the steering gear to the steering position is of any considerable length.

Description of Brown's Patent Hydraulic Steering Telemotor (Sketch No. 66).

When the distance between the steering engine and the position of the steering wheel is considerable, as in most modern ships, and it is desired to have as frictionless as possible a connection between the steering wheel and the steering engine, the telemotor shows to the greatest advantage over shafting and its equivalents. One of the advantages is that the small copper pipes can be led almost anywhere, provided they are protected from heat and damage; in fact, through places where it would be very undesirable to have shafting, such as saloons, berths, &c., as there is no noise, or oiling required, and no motion, except, of course, the fluid through the pipes, so that there is no danger of anything getting foul of bevel wheels, &c., and disabling the gear. The telemotor described and illustrated herein is the outcome of the original inventor's and makers' experience up to this date.

Fig. 1 shows the vertical section of the transmitting cylinder A, fitted with the piston B attached to the rack C, into which gears a pinion D, the shaft E of which is made to revolve by the hand-wheel through pinion and spur wheels F and G, by which a suitable number of turns of the hand-wheel are obtained.

Pipes H and I from the top and bottom of the cylinder respectively are led to the gear aft, and are joined up to either end of the cylinder K by means of the pipes L and M, the connection being made according to which way the after cylinder is required to move in relation to the forward one.

(Fig. 3.) This cylinder is fitted with a piston N with the usual piston rod, and connecting links O which are attached by a lever to the controlling valve of the steering engine. The piston rod is fitted with two crossheads P P, between which lies a spiral spring Q, under initial compression, and which is compressed further by any motion of the piston, the object being to always cause the piston to return to mid-position—the steering engine control valve, of course, moving with it—when the pressure on both sides is equal, or tending to become so.

When the apparatus is fully charged with fluid, any movement of the steering wheel will bring about a corresponding movement of the piston in the receiving cylinder K, and consequently the valve gear of the steering engine.

In pulling the wheel round, it will be found to become sensibly stiffer until it is hard over, so that the steersman feels the amount of helm he is giving the ship, much in the same way as in steering by hand with the antiquated winding drum and chains; and, on "letting

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The increase of resistance of one large spring is very much less than with the old design with two springs of small diameter, and as the minimum power required here is fixed by the amount required to move the control valve of the steering engine and bring the steering wheel back to its central position, it follows that with the single spring considerably less power is required to put the wheel hard over. Further advantages are that a much better design of spring is possible, with a larger factor of safety, and being co-axial with the cylinder it is more efficient, as, with the two springs, one on each side, there is generally a cross-winding action, or tendency of the cross-heads to bear hard on the guide rods, due to it being practically impossible to get two springs to exert equal resistance, or give out equal power, through equal ranges of motion.

The telemotor on the bridge is fitted with an indicator R, as shown in Fig. 2, which, when everything is in order, shows the actual position of the helm. It is possible, however, that the piston packing leathers may in time become worn, so as to admit of considerable leakage, and it may happen that the piston B may be working altogether in the top or bottom of the cylinder A, with the piston aft in the mid-position, and the ship steered on a straight course with the indicator pointing at, say, 20°, and this may go on until the indicator pointer is almost past the degrees marked on the quadrant, still without disabling the gear, as the capacity of the cylinder A is considerably more than that of the cylinder K aft.

To readjust the indicator and position of the helm, it is only necessary to turn the steering wheel until the indicator is brought to zero, and the piston enters the bye-pass or central position, allowing a free communication of liquid between both sides of the system, when the compressed spring aft will immediately bring everything into correspondence. As, however, the piston B in steering a ship is always more or less passing the centre position in "porting" or "starboarding" even to the smallest extent, the tendency is for the piston N in the cylinder K always to return to the mid-position every time the forward piston B is in that position.

The bye-pass is now greatly improved, being formed by drilling two rows of very small holes, which are the correct distance apart longitudinally, so that the two leather packings of the piston B are between them, thus allowing a free passage for the fluid through the holes and so round the piston. This, of course, allows the pressure between the two sides of the system to come into equilibrium should there be any difference, and the spring on the cylinder K then brings the piston N and the control valve of the steering engine to their central position. The small holes are not liable to catch the leathers and turn the edges over, as frequently happened with the old style of bye-pass where the opening extends right round. This is quite obvious when it is explained that the holes are only just over $\frac{1}{16}$ inch diameter and nearly $\frac{1}{4}$ inch apart, the leathers being thus supported by the metal between the holes. Another advantage is that the cylinder is all in one piece and can be bored right through at one operation, which prevents any possibility of getting out of line at this part, such as was always liable to take place with the old arrangement, through the joints not being properly nipped up, or any defect in machining, &c.

It is sometimes necessary to set the gear so that the central position does not actually represent the rudder as true fore and aft, but a certain amount of permanent helm is required to counteract the action of the propeller, &c., when the ship is under weigh. This is done by making the connecting links longer or shorter, as the case may require, by means of the adjusting nuts provided, thus altering the central or shut position of the steam control valve.

In some exceptional cases, where it might be inconvenient to adjust the gear by running the indicator into its midship position, should the two pistons have got out of correspondence, there is provided the hand-wheel S which opens the stop valve T, giving a free communication between the top and bottom parts of the cylinder, taking the place of the automatic adjustment by bye-pass at the central position, and so allowing the indicator to be brought to zero without moving the rudder aft. **This valve must be shut and kept so when working.** It should only be used in cases where there is very little room to manœuvre the ship, as in narrow waters. The reason for this precaution is that it may be opened when unnecessary and left open or slightly open. In port it may be left open with advantage, as should any one move the wheel, nothing is moved aft, and so no damage can result.

A small tank U is provided with gauge glass, as shown. This is usually charged with a mixture of glycerine and water, one part of the former to two or three of the latter. **The cock** V is provided for shutting the tank off from the system when charging up, &c., **but it must always be kept open when the telemotor is being used for steering.**

As it is very important that the whole system of pipes and cylinders should be fully charged, and no air should be present, it is necessary to provide for the expansion and contraction of the fluid, due to changes of temperature, &c., and for this purpose a valve box is fitted on the bye-pass, a section of which is shown on Fig. 4. It contains a small inlet and outlet valve, the latter being simply an ordinary safety valve loaded above the working pressure, which is about 200 lbs. per square inch. As the temperature rises, a portion of the fluid passes into the tank U, and as the pressure falls the fluid returns through the inlet valve.

The entire telemotor on the bridge is constructed of gun-metal, so as not to affect the compass. The motor cylinder aft is of the same material, and the pipes are of solid drawn copper of diameters varying from $\frac{3}{8}$ to $\frac{5}{8}$ inch internal diameter, according to the length fitted. These are easily run, and may be bent into any number of corners without adding materially to the friction of the gear.

A hand pump Y with tank Z is provided for charging up the system, and suitable pipes for connecting are supplied according to the arrangement of the gear on the ship. The cock on the tank is for shutting off the fluid from the pump when not in use. Screw-down valves J^1 and J^2 are provided for shutting off the pump and its connections when the system is charged, the discharge pipe from the pump being connected to J^2 and the return pipe to J^1. Two similar valves, J^3 and J^4, are provided so that they can be closed when it is desired to open out the cylinder K, and so prevent loss of fluid from the system. **When working, these latter valves must be kept open, as also when charging up.** A spring-loaded valve J^5 is provided, as shown, so that when charging up a system where the forward cylinder is a great height above the cylinder K, the fluid is retained in the pipes instead of coming down and leaving a vacuum or empty space at the highest point.

Instructions for Charging, Adjusting, and Working.

It is of the utmost importance that all joints be watertight, as any leakage will empty the small tank. After all the pipes are coupled and the connections made to cylinders and to tank in wheel-house, close the cock underneath the tank and fill to about one-third full with fresh water. For cold climates, add 30 per cent. glycerine, which keeps the parts lubricated, and will resist frost to about zero Fahrenheit (see table of freezing temperatures of various mixtures of water and glycerine on page 359). Put the hand-wheel in mid gear, which will be seen by the pointer coming between the two zero marks on indicator. This opens the bye-pass between the top and bottom ends of the cylinder, and allows the whole system to be charged by one operation from the after part of the ship.

Open the cocks on the side of the cylinder K or motor cylinder, and see the cocks J^3 and J^4 are open. When pumping, great care should be taken that the liquid in tank Z never gets so low as to allow the pump to draw air, as the good working of the gear depends upon the air being expelled. The liquid will shortly be seen to run from the small pipe back into the tank Z, but the pumping must be continued for some time, say three times as long as it took to come back. By this time the air should nearly all have been driven out, and each stroke of the pump should show a corresponding rush, and not a continuous flow back through the return pipe to the tank.

Being satisfied as to this, the air cock J^1 on the top of the cylinder should be closed, and a slight but continuous strain kept on the pump. Now, go forward to the wheel-house, and on the valve casing cover on the transmitting cylinder A will be seen a brass plug

W^1; remove it, and press down the spindle of the inlet valve, which is immediately underneath, when the liquid will rush up owing to the pressure being kept on by the pump from aft. When the casing is quite full, and no more air bubbles up, screw in the plug W^1; also the plug A^1 on the top of the transmitting cylinder should be slacked back to allow any air imprisoned in the cylinder to escape, afterwards tighten up the plug, close the cock J^2 on the under side of the motor cylinder K, when the installation will be fully charged; open the cock V underneath the tank U, and all is ready for use. The tank U in the wheel-house should be kept half full.

The gear may now be tried by putting the wheel over to port and starboard, and noticing aft if a corresponding movement takes place in the piston of the motor cylinder. Should it not respond on one side or the other, then an internal leakage may be suspected; in which case, examine the leathers in the telemotor and motor cylinder.

To take out for examination or renewal the leathers on the piston B (a section of this piston with its leathers, springs, nuts, &c., is shown to a larger scale in Fig. 7), it is only necessary to remove the cylinder cover and turn the wheel so as to bring the piston up. The rack is sufficiently long to enable the piston to be run up right out of the cylinder and so be easily got at. If the bye-pass valve T is opened, and the cover left on until the piston comes against it, this can be done with little, if any, loss of fluid. To get at the leathers in the after cylinder K (Fig. 8 shows the piston with its leathers, springs, nuts, &c., to a larger scale), it is necessary to shut valve J^4, remove the cylinder cover, slack off and remove the two large nuts that bear on the top yoke P, when the piston rod, &c., can be drawn sufficiently far out to examine or renew the leathers. As soon as the piston comes out of the cylinder, valve J^3 should be shut, so that no more fluid may be lost. It should not be closed before the piston is out, or difficulty may be experienced in getting it so. All the leathers used for the two pistons (four in all) are exactly alike, which is a great advantage, as only one size of leather has to be carried as spare instead of two sizes as in the older designs. Care should be taken that any new leathers obtained are the proper depth, as the action of the automatic bye-pass on the cylinder A may be rendered inoperative, if they are too deep and cover the holes. The leathers in the pistons themselves will not cause any trouble until actually worn out, and even when in a leaky condition will work quite well and keep in correspondence with the gear aft, in virtue of the spring always putting the rudder in a fore and aft central position when the piston enters the bye-pass portion of the cylinder.

The inlet and relief valves in the valve box W are not working but automatic valves; they merely open and shut as occasion requires, to allow for expansion and contraction of the fluid in the pipes due to change of temperature.

After having made any repairs that may have been necessary, and before recharging, it is advisable to clean out the pocket underneath these valves, the purpose of which is to collect any dirt or sediment that may have been in the liquid. This is done by removing the brass plug in the bottom, when the small quantity of liquid that flows out of the pocket will carry anything with it.

When first charging up after erection, or after any repairs or alterations to pipes, &c., it is advisable to disconnect the pipes from both cylinders and force clean water through them, so as to wash out any dirt or other foreign matter that may be in the pipes, and so prevent it getting into the cylinders and valve boxes.

In addition to the stuffing box of the valve T, there are only three more—one on the cylinder on the bridge and two aft—and as the water pressure need never exceed 250 lbs. per square inch, there is no reason for any serious loss of the fluid in the tank. Keep the stuffing boxes *full* of greasy cotton packing, and screw up as lightly as is necessary to secure *tightness*, but not *stiffness*. It is advisable to occasionally examine the leathers in the telemotor and motor cylinder aft when the ship is in port. The necessity for this can be ascertained by pulling the steering wheel hard over to port and securing it there. The motor cylinder will be found to have responded to same extent. If the gear is now left, say for half an hour, the spring in the motor cylinder will have moved the piston towards midship position if there is any leakage in the port leather. A similar trial may be made to starboard, which will test these leathers.

It need not be expected that these leathers should be quite tight, but the motor piston should remain over for say ten minutes without any serious movement towards midship position, that being about the maximum time that, in practice, a helm would be held hard over; and so any little deviation due to leaky leathers would be at once adjusted when the steering wheel is let go, the motor springs running it back to zero, and the bye-pass allowing the free circulation of the fluid.

Fig. 5 is a section of the telemotor through the centre of the shaft, and shows a screwed plug X. When it is desired to take out the shaft E (the indicator being at zero), this plug is withdrawn and the other end screwed into the cylinder until its point enters a recess in the rack C. The rack is thus kept in its central position until the shaft and the pinion are replaced.

Care should be taken to lubricate with good oil the various working parts of the gear.

A glycerometer and thermometer are supplied with each installation, so that it is possible to test the actual proportion of glycerine in the fluid at any time when the gear is not in use, by drawing some of the fluid out of the circuit and testing in a similar manner to that adopted for ascertaining the density of the water in boilers, the glycerometer reading right off the percentage of glycerine.

Non-Freezing Fluid for Telemotors.

Water containing Refined Glycerine.	Safe to Work to Fahrenheit.
25 per cent.	$+18°$
33 ,,	$+10°$
50 ,,	$-20°$
60 ,,	$-30°$, getting thick
70 ,,	Too thick to work at -25.

Metallic Packing (Sketches Nos. 67 and 68).

The United States type of packing is entirely metallic, and is thus specially suitable for high pressure steam or gas. Consisting as it does of various members or sections carefully fitted into each other, any side play of the rod is compensated for, the accommodating nature of the springs, cones, rings, and blocks forming the packing, and which constitutes perhaps the most valuable point of this well-known system of packing; the regulation of the packing block pressure is automatic, constant, and reliable, and is regulated to suit equally well the out and in stroke of the rod.

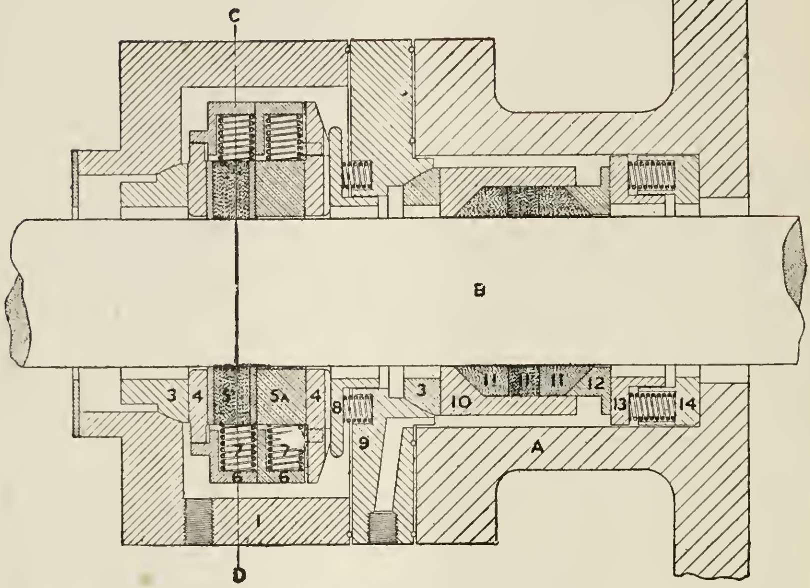
To exert a minimum packing pressure against a rod, a packing must be of the floating type, and this important result is attained in the U.S. packing, which is automatic and floating, exerting a minimum but effective pressure against the rod, and thus preventing the escape of steam in the case of high pressures and intermediate cylinder rod glands, and the admission of air in the case of the L.P. cylinder rod gland. The packing is free to "follow the rod," and this being so, the pressure of the packing against the rod is reduced to a minimum.

Description.

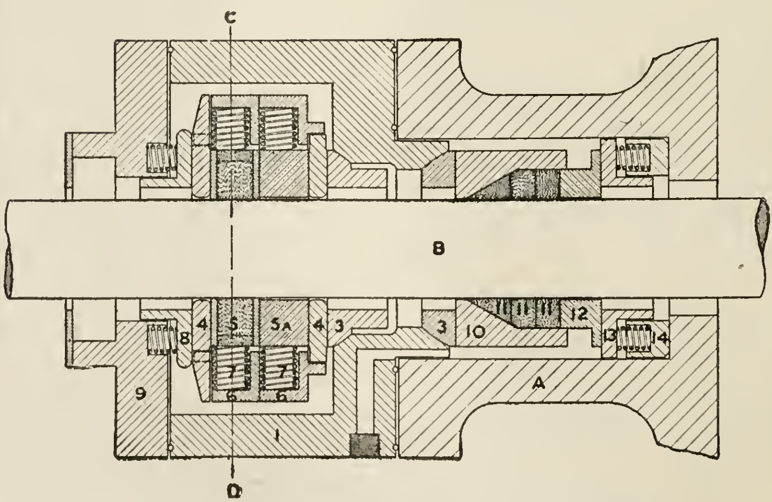
Duplex Packing is designed for use with high pressures. It consists of a block packing as described above used in conjunction with a cone packing which includes a set of white metal rings (11) placed in a vibrating cup (10), the interior of which is partly conical. The duplex follower ring (12) holds the cone rings in position, and transmits to the latter the pressure from the duplex follower springs, which are held in the ring (14) and protected by the spring cover (13). In this arrangement the inner cone packing checks the steam pressure, and the outer block packing is thus assisted, and the escape of steam absolutely prevented.

Atmospheric Duplex Packing.—For use on low-pressure condensing cylinders. It consists, like the Duplex Packing, of two parts, but with this difference: in the Duplex Packing both parts are steam setting, and operate in the same direction to prevent the escape of steam: in the Atmospheric Packing the parts are placed face to face and act in

United States Marine Type Packings.



No. 67.—Duplex Packing.



No. 68.—Atmospheric Duplex Packing.

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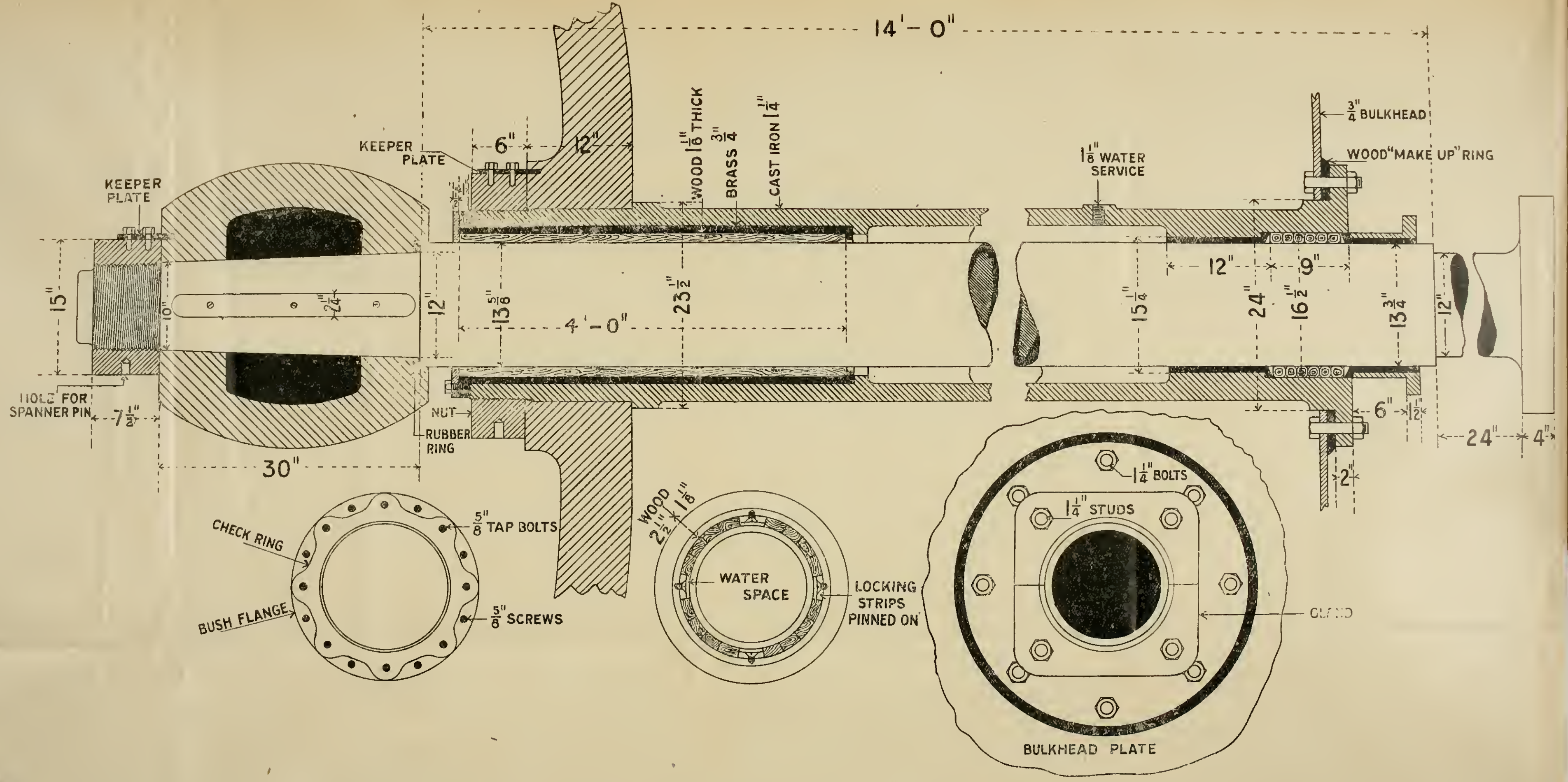
are pinned on to nut at back of boss to prevent turning, and to the large nut screwed on to the tube aft of the stern post for the same object. The propeller boss is fitted to the shaft on a taper ($\frac{3}{4}$ inch per foot), a feather or key being also sunk on the shaft, and secured by pins as shown. The boss should have a bearing fit on the feather at the sides, but should be clear at the top of feather.

For a right-hand propeller the boss nut should have a left-hand



Check Ring and After-Bush. Section through Tube. Gland and Flange of Tube.

No. 69.—Stern Tube and Propeller Shaft.
(With dimensions for a 12-inch shaft.)

[To face page 301.

opposite directions. The inner packing only is steam setting and prevents the escape of steam. The outer part is open to and set by the atmosphere. When there is a vacuum in the cylinder, the atmospheric pressure is actually used to tighten the outer packing and automatically prevent the passage of air, which would impair the vacuum.

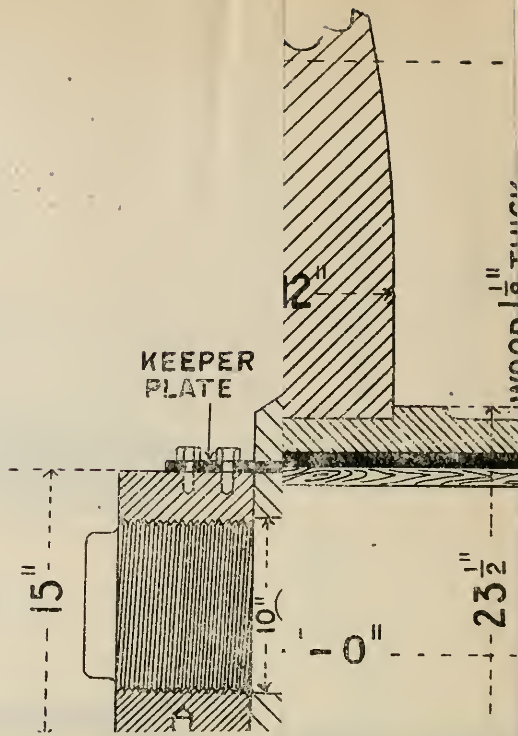
DESCRIPTION OF COMPONENT PARTS.

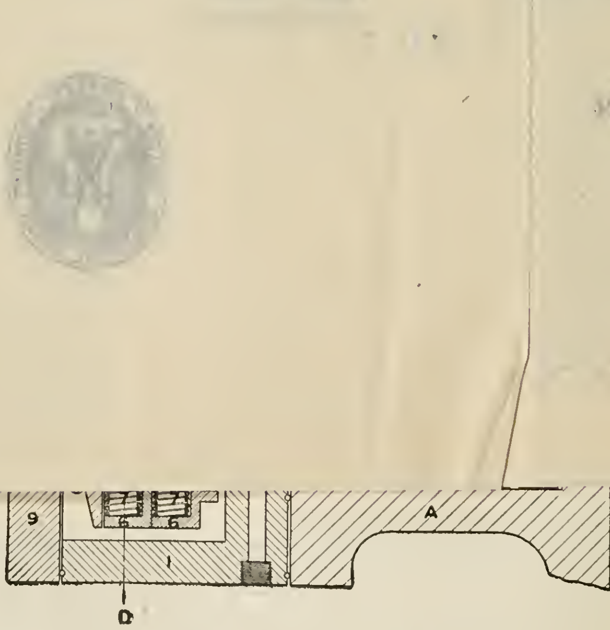
A, Stuffing Box.
B, Piston Rod.
1, Packing Case.
2, Stud Bolts.
3, Ball Joints.
4, Sliding Plates.
5, Packing Blocks.
5A, Guide Blocks.
6, Horn Rings.
7, Block Springs.
8, Spring Cover Plate.
9, Follower Bush and Springs.
10, Duplex Vibrating Cup.
11, Duplex Cone Rings.
12, Duplex Follower.
13, Duplex Spring Cover.
14, Duplex Spring Holder and Springs.

Stern Tube and Shaft (Sketch No. 69).

Description.—The stern tube is generally constructed of cast iron, the thickness varying from about $1\frac{1}{2}$ to $2\frac{1}{2}$ inches. The tube is larger in diameter at the forward end (24 inches) and slightly less ($23\frac{1}{2}$ inches) at the stern-post end for convenience in fitting in or in taking out. The forward end is flanged and bolted to the after bulkhead, a "make-up" liner of lead or wood being inserted between the two as shown. The forward end is supplied with a stuffing box packing and gland to keep water out of the tunnel, and the after end runs on a bearing composed of lignum vitæ (hard wood), the wood strips being fitted dovetail fashion into the after brass bush. Waterways are left at four or more positions to allow access of water to the shaft after bearing. The wood strips are kept in place forward by a collar or lip on the bush, or on the stern tube, and aft by a "check ring," which is secured by means of tap bolts to the flange of the brass bush. The bush itself is pinned in turn to the stern tube by countersunk screws, as shown in the drawing. A rubber "stop" ring is fitted hard up between the end of shaft liner and the propeller boss, which is recessed out to allow the rubber ring to be fitted, the object of the rubber ring being to prevent the access of water to the metal of the shaft, and thus prevent galvanic action taking place between the shaft metal and the brass of the liner, which would result in corrosion of the shaft. "Keeper" plates are pinned on to nut at back of boss to prevent turning, and to the large nut screwed on to the tube aft of the stern post for the same object. The propeller boss is fitted to the shaft on a taper ($\frac{3}{4}$ inch per foot), a feather or key being also sunk on the shaft, and secured by pins as shown. The boss should have a bearing fit on the feather at the sides, but should be clear at the top of feather.

For a right-hand propeller the boss nut should have a left-hand





No. 68.—Atmospheric Duplex Packing.

opposite directions. The inner packing only is steam setting and prevents the escape of steam. The outer part is open to and set by the atmosphere. When there is a vacuum in the cylinder, the atmospheric pressure is actually used to tighten the outer packing and automatically prevent the passage of air, which would impair the vacuum.

DESCRIPTION OF COMPONENT PARTS.

A, Stuffing Box.	5A, Guide Blocks.	10, Duplex Vibrating Cup.
B, Piston Rod.	6, Horn Rings.	11, Duplex Cone Rings.
1, Packing Case.	7, Block Springs.	12, Duplex Follower.
2, Stud Bolts.	8, Spring Cover Plate.	13, Duplex Spring Cover.
3, Ball Joints.	9, Follower Bush and Springs.	14, Duplex Spring Holder and Springs.
4, Sliding Plates.		
5, Packing Blocks.		

Stern Tube and Shaft (Sketch No. 69).

Description.—The stern tube is generally constructed of cast iron, the thickness varying from about $1\frac{1}{2}$ to $2\frac{1}{2}$ inches. The tube is larger in diameter at the forward end (24 inches) and slightly less ($23\frac{1}{2}$ inches) at the stern-post end for convenience in fitting in or in taking out. The forward end is flanged and bolted to the after bulkhead, a "make-up" liner of lead or wood being inserted between the two as shown. The forward end is supplied with a stuffing box packing and gland to keep water out of the tunnel, and the after end runs on a bearing composed of lignum vitæ (hard wood), the wood strips being fitted dovetail fashion into the after brass bush. Waterways are left at four or more positions to allow access of water to the shaft after bearing. The wood strips are kept in place forward by a collar or lip on the bush, or on the stern tube, and aft by a "check ring," which is secured by means of tap bolts to the flange of the brass bush. The bush itself is pinned in turn to the stern tube by countersunk screws, as shown in the drawing. A rubber "stop" ring is fitted hard up between the end of shaft liner and the propeller boss, which is recessed out to allow the rubber ring to be fitted, the object of the rubber ring being to prevent the access of water to the metal of the shaft, and thus prevent galvanic action taking place between the shaft metal and the brass of the liner, which would result in corrosion of the shaft. "Keeper" plates are pinned on to nut at back of boss to prevent turning, and to the large nut screwed on to the tube aft of the stern post for the same object. The propeller boss is fitted to the shaft on a taper ($\frac{3}{4}$ inch per foot), a feather or key being also sunk on the shaft, and secured by pins as shown. The boss should have a bearing fit on the feather at the sides, but should be clear at the top of feather.

For a right-hand propeller the boss nut should have a left-hand

screw, so as to be self-locking when revolving. Needless to say the stern tube is put in place from the inside of the ship.

Observe that a small check collar is cast on the tube forward of the stern post for tightening up the nut. In the drawing shown the diameter of the shaft liner is $13\frac{3}{4}$ inches forward and $13\frac{5}{8}$ inches aft: this is for convenience in fitting in or drawing out the shaft.

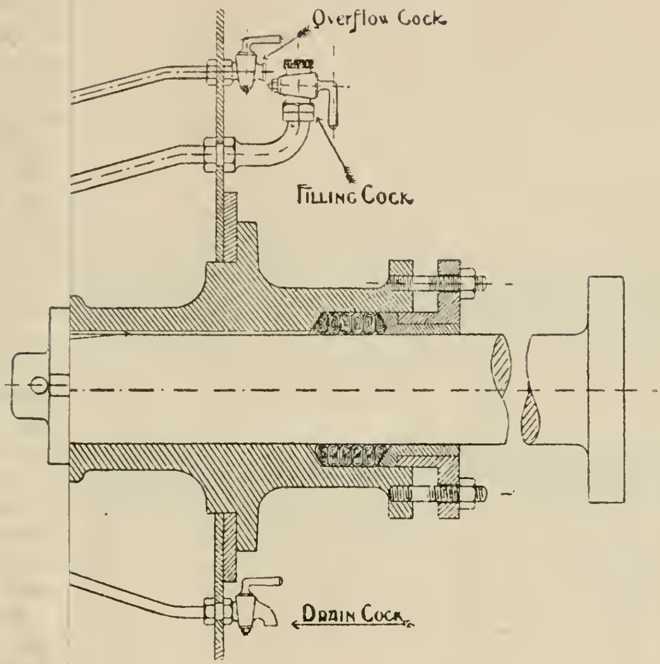
In running, it is better that a small drip of water should show at the gland, as otherwise the gland brass bushes and packing may heat up, and possibly tear up the shaft liner.

A water service is fitted, leading from the tube to the bulkhead just over the gland, the temperature of which will indicate if the shaft inside the tube is running cool or otherwise. To allow for partial repacking at sea, the gland is often fitted in halves, with long studs, so that the gland need not be taken off altogether when inserting the turn of packing required.

Cedervall's Patent Stern Tube (Sketch No. 70).—The lubrication of propeller shaft bearings within the stern tube has hitherto been chiefly effected by the eakage of a certain amount of water into the bearings, and its combination there with the oily surfaces of lignum-vitæ strips; or by forcing a mixture of oil and tallow between the plain unprotected bearing surfaces. In the one case the shaft, although partially protected by a sleeve of brass, has certain parts constantly exposed to the corrosive action of sea water; while the other method, under usual conditions, does not afford a satisfactory means of lubrication, as the water washes away the lubricant from the parts where it is most required, and the resultant corrosion and wear of shaft and bearings are most excessive.

To overcome imperfections, and to reduce first cost, "Cedervall's Patent Protective Lubricating Box" has been invented. The principal objects of this invention are to absolutely prevent the access of any external water to the stern tube, and to provide a reservoir of oil capable of supplying a steady and continuous lubrication to the whole bearing surface. The invention consists, essentially, of an annular box of brass or gun-metal, containing an inner packing ring, which is pressed outwards by a series of small spiral springs. The box fits over the shaft, and is fixed to the forward face of the propeller boss by means of screws, thus turning with the propeller, and the inner movable ring presses against the prepared face of the stern tube bush. The springs, while of ample strength, are of such elasticity that, irrespective of any play which the shaft may have in revolving or reversing, the ring maintains a watertight joint with the end of the tube. As the ring is faced with antifriction metal and well lubricated by oil from the inside, it revolves with the minimum amount of friction.

The method of applying the protective lubricating box, and the arrangement for supplying the lubricant, are shown in the drawing. Three or more grooves are cut in the stern tube bush.



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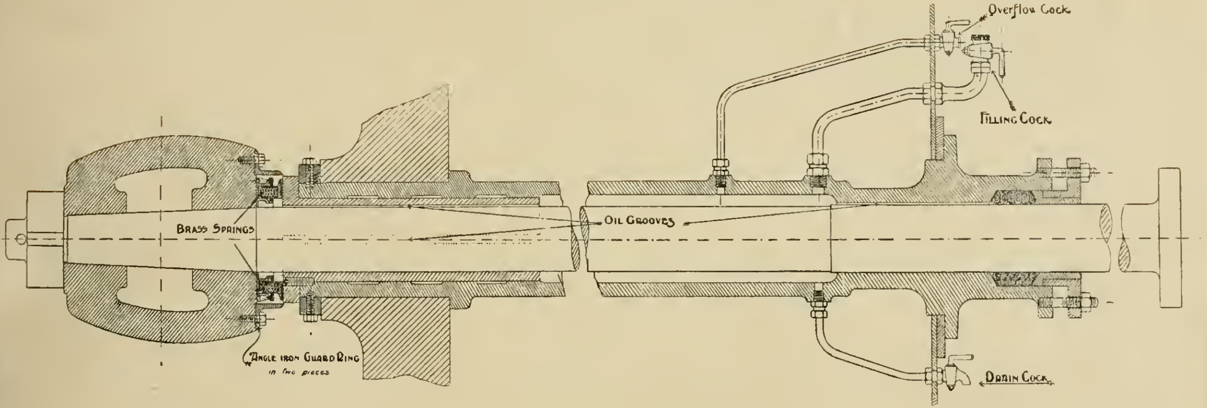
4. Shore off coupling of tail shaft solidly from stern tube end by means of the wooden blocks mentioned before, and remove gland and packing.

5. Remove nut at back of boss, by means of blows from a hammer on the large spanner, having previously secured the propeller from turning.



No. 70.– Cedervell's Patent Stern Tube.

[To face page 362.

The top one is for the escape of air, and the others are for leading the oil to the inside of the movable ring of the protective box. From the stern tube three pipes are carried through the aft bulkhead, or to any other convenient place, and fitted with cocks. The oil is forced in by means of a small hand pump, and when all the spaces are filled, the oil shows at the overflow cock.

From the foregoing description it will be obvious that the protective lubricating box has several highly important advantages. Its adoption does away with the necessity for expensive liners and metal bearings, the plain cast-iron stern tube bush being all that is required, and as a consequence of the efficient and uniform lubrication, coupled with the exclusion of all dirt and gritty substances from the bearings, the wear and tear on the propeller shaft is reduced to a minimum, and the usual vibration at the after part of most steamers is practically eliminated. Experience with vessels already fitted with the patent lubricating box amply proves its efficiency, as after several years' constant working the shafts on examination exhibit bearing surfaces quite as good as any smoothly working bearing connected with the engine proper.

The safeguard which this immunity from corrosion and absolute wear affords against breakdowns of the shafting must be obvious to, and appreciated by, all having experience with the present expensive and not very efficient mode of fitting and lubricating propeller shaft bearings. Although heating of the stern tube bearings is most unlikely to happen with the arrangements shown, should it occur, the oil may be discharged at the drain cock, and water forced through the bearing by means of a hose attached to the filling cock. To obviate the possibility of the box being fouled by ropes, ice, or other floating bodies, a strong guard ring, made in halves, is fitted over the box, as shown.

Drawing the Propeller Shaft.—The usual method of drawing out the tail end shaft for examination or repair is as follows :—

With steamer in dry dock, have all necessary working gear at hand, such as: chain and wire rope tackle, strong wooden blocks, screw or hydraulic jacks, light ram for coupling bolts, &c.

1. Fit up suitable staging round propeller.

2. Disconnect tunnel shafting and remove to one side two lengths of same.

3. Place tackle in position for drawing out tail end shaft (usually consisting of rope and chain blocks).

4. Shore off coupling of tail shaft solidly from stern tube end by means of the wooden blocks mentioned before, and remove gland and packing.

5. Remove nut at back of boss, by means of blows from a hammer on the large spanner, having previously secured the propeller from turning.

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with the end of the tube. As the ring is faced with antifriction metal and well lubricated by oil from the inside, it revolves with the minimum amount of friction.

The method of applying the protective lubricating box, and the arrangement for supplying the lubricant, are shown in the drawing. Three or more grooves are cut in the stern tube bush.

The top one is for the escape of air, and the others are for leading the oil to the inside of the movable ring of the protective box. From the stern tube three pipes are carried through the aft bulkhead, or to any other convenient place, and fitted with cocks. The oil is forced in by means of a small hand pump, and when all the spaces are filled, the oil shows at the overflow cock.

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4. Shore off coupling of tail shaft solidly from stern tube end by means of the wooden blocks mentioned before, and remove gland and packing.
5. Remove nut at back of boss, by means of blows from a hammer on the large spanner, having previously secured the propeller from turning.

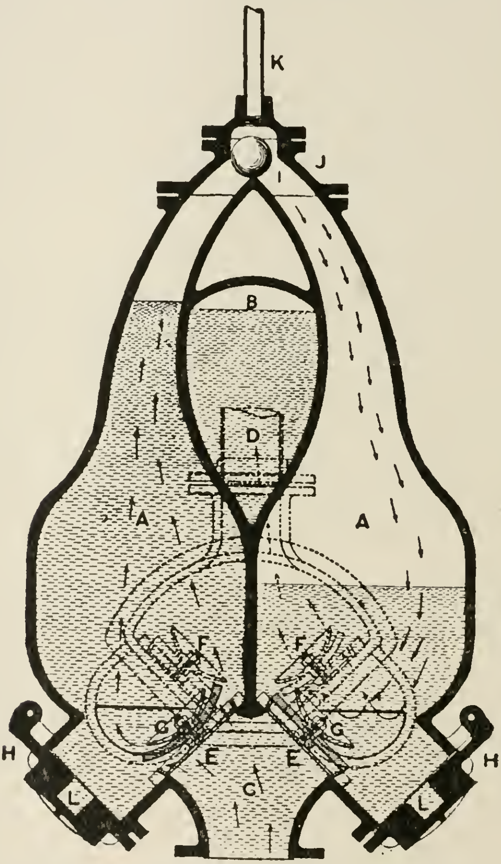
6. Drive in steel wedges hard up between boss and stern post, and by means of a ram (hydraulic) force boss off the taper.

7. Connect up propeller to tackle, and draw tail shaft gradually into tunnel, supporting it by blocks as it emerges.

NOTE.—Very often the boss is found difficult to start, and when this is found to be the case one of the following methods may be tried :—

1. Build a fire below the boss, and when heated up apply blows from a large hammer on the end of the shaft, or the pressure of a ram on the steel wedges.

2. Bore a number of holes into the metal of the boss, then try the heating up, &c., as before. The holes are to allow of easier expansion of the boss when heating up.



No. 71.—Pulsometer Pump.

Description of the Pulsometer Type Pump.

The body casting comprises the two working chambers A A, the air vessel B, and the discharge box, which is shown by the dotted lines in the illustration. Valves G G are fitted between the suction branch

C and the working chambers, and a second set of similar valves F F is arranged in the passages connecting the discharge box with the working chambers. Hand-holes L L are provided to give access to the suction valves, whilst the discharge valves are reached by removing the cover of the discharge box. Surmounting the body there is the "neck" casting J, which contains the gun-metal ball I, and is fitted with the steam pipe K. The air vessel B communicates with the suction branch C, by means of a prolongation running down in front of and between the two chambers A A.

The action of the pump consists of the alternate filling and emptying of each working chamber, as the condensation and pressure of the steam respectively exert an upward and downward force on the surface of the water. The alternation of these operations is effected by means of the ball valve I in the following manner:—Assuming one of the chambers A be open to steam and full of water, the steam entering by the steam pipe K, and past the ball I, passes into the chamber and presses upon the small surface of water exposed. This depresses it and drives it through the discharge valves F F into the rising main D. The moment, however, the water in the chamber falls to the level of the opening in the branch leading to the discharge box, the steam blows through, and the consequent disturbance of the water surface causes the instantaneous condensation of the steam. The vacuum thus formed in the emptied chamber immediately pulls the control ball I over on to the corresponding seat and cuts off further admission of steam, allowing the vacuum to be completed. Water immediately enters through the suction pipe C, and lifting the inlet valve G, rapidly fills the chamber again. A similar operation has been taking place in the opposite chamber, the period occupied by filling one chamber corresponding with that of emptying the other, and these operations continue alternately in the two chambers so long as the pump is supplied with steam and water. The alternations follow so rapidly and with such regularity that the stream of water is practically continuous. A small "snifting" valve is fixed in the upper part of each of the working chambers. Their function is to introduce a small quantity of air at each pulsation for the purpose of cushioning the ball as it changes its position and to separate the steam from the water by a non-conducting film, thus preventing loss of steam by condensation during the forcing part of the stroke.

Air Pump Valves and Vacuum.

The condenser vacuum is affected most of all by broken or leaky bucket valves, next by broken or leaky head valves, and least of all by defective foot valves, particularly so in the case of the newer type of independent condenser which is placed much higher than the bottom of the air pump and allows of complete drainage of the water. Foot valves, although not much required for the maintenance of the vacuum, allow the pump to work steadier and more regularly than would be the case if these valves were omitted or were broken.

Hot-well Temperature and Condenser Back Pressure.

Neglecting air leakage, the pressure in the condenser can be determined from the hot-well temperature as follows:—Note the hot-well temperature and look up the "Table of Saturated Steam," page 622, for the corresponding pressure.

EXAMPLE.—The temperature of the water in the hot-well is 141°. Find the corresponding vapour pressure.

ANSWER.—On looking up the Table, page 622, we find that the vapour pressure for this temperature is 3 lbs. absolute, which is, of course, the back pressure in the condenser.

In actual practice air leakage reduces the hot-well temperature for a given degree of vacuum.

NOTE.—The actual back pressure on the L.P. piston is usually from 1 to 2 lbs. in excess of this, as a slight difference of pressure must of necessity exist between the two positions of steam flow.

From the foregoing it will be evident that it is impossible to have both a high vacuum and high hot-well temperature as the two vary in inverse ratio. With a high temperature of hot-well water the vapour corresponding to the temperature is also high, with, of course, a proportionally reduced vacuum in the condenser.

General Definitions.

Heat.—Heat is a form of energy. When the molecules of a body are set in rapid motion or vibration, heat results, and the more rapid the vibration the more intense is the heat generated. The amount of heat given to a body produces a difference in its temperature. Heat may, then, be expressed as molecular energy, and the value of one unit—generally known as one British Thermal Unit, or simply one B.T.U.—is equal to 778 foot-pounds of work. It should always be borne in mind that Heat and Work are mutually interchangeable, Heat giving out Work, and Work done producing Heat.

Foot-Pound.—1 foot-pound of work is equal to a weight of 1 lb. raised 1 foot.

Power.—Power is the amount of work done in a given time (as, for example, per minute).

Horse-Power.—1 Horse-Power (Indicated) is equal to a weight of 33000 lbs. raised 1 foot in one minute or to a weight of 1 lb. raised 33000 feet in one minute.

Unit of Heat.—To raise the temperature of 1 lb. of water one degree requires the expenditure of 778 foot-pounds of work. This is known as the Mechanical Value of one Heat Unit.

Sensible Heat.—Sensible Heat raises the temperature of a body, and is measured by the thermometer.

Latent Heat.—Latent Heat changes the condition of a body (as, for example, ice to water, or water to steam) without adding to its temperature. To change or evaporate into steam 1 lb. of water at 212° temperature requires 966 units of Latent Heat.

Total Heat.—Total Heat is the sum of the Sensible and Latent Heats.

Energy.—Energy is the capacity to do work. All energy really originates from the heat of the sun.

Potential Energy is *stored up* energy, as, for example, a raised weight, a coiled spring, gunpowder, and steam in a boiler.

Kinetic Energy is the energy of motion, as, for example, a moving piston rod, or pump plunger, a revolving shaft, and machines in general.

NOTE.—**Potential Energy when set free changes to Kinetic Energy.**

Force.—Force is that which moves or tends to move a body, as, for example, the force of steam, the force of water, the force of gravity, &c.

Inertia.—Inertia is the natural property possessed by bodies at rest to remain at rest unless acted on by some force, or, if set in motion to continue in motion unless acted on by other forces, such as friction, &c.

Centrifugal Force.—Centrifugal Force means a force acting outwards from the centre. An example of this is the centrifugal circulating pump where the water enters at the centre and is forced outwards to the circumference or periphery of the vanes.

Friction.—Friction depends on the pressure exerted and nature of the surfaces in contact, and is independent of surface area. For example, if a small guide shoe is changed for a larger one the *total friction* is still the same, but the pressure per *square inch* on the shoe is less The coefficient of friction for lubricated metals is ·08, which means that ·08 of the pressure exerted is absorbed in overcoming friction.

Steam.—Steam is an invisible gas obtained by the evaporation of water. It may be expanded to a lower pressure, or compressed to a higher pressure: it can also be condensed back again to water.

Stress.—Stress means the forces set up in a material to resist strain or fracture, as, for example, a pressure of, say, 100 lbs. acting on a surface of 10 square inches will produce a tensile stress of 1000 lbs. on a stay of 1 square inch area.

Strain.—Strain means change of form in a structure due to stress, as for example, when a rod is lengthened by tensile stress, or shortened by compressive stress.

Specific Gravity.—Specific Gravity means the weight of a body compared with water and of the same volume. The specific gravity of Wrought Iron is 7·7, of Mercury 13·5, and of Oil ·9.

NOTE.—Water is taken as representing the figure 1.

Efficiency.—The efficiency of an engine is lowered by (1) Boiler losses, (2) Engine losses, (3) Mechanical losses, and (4) Propeller losses. The average combined efficiency of a marine boiler, engine, and propeller is only about 6 per cent. of the total, or is represented by the fraction $\frac{1}{16}$.

Specific Heat (Capacity for Heat).—Is the heat required to raise 1 lb. of anything 1° in temperature compared with the heat required to raise 1 lb. of water 1°.

Specific Heats.

Water (at 39°)	1·00
Steam (at 212°)	·48
Ice	·5
Wrought Iron	·113
Mercury	·033

From the above it will be seen that the amount of heat required to raise 1 lb. of water 1° in temperature would be sufficient to raise 1 lb. of wrought iron nearly 9° in temperature, as $1 \div \cdot 113 = 8{\cdot}8°$.

Hyperbolic Expansion Curve.—This is known as the "Isothermal" or even temperature curve of a gas, and is obtained from the law of Boyle which states that: Pressure × Volume = Constant.

From this it follows that if the volume of a gas be doubled the pressure falls to half, or if, as shown in the diagram, the original volume of the gas is increased three times, the pressure falls to one-third. Observe that the steam is cut off at one-third stroke, and at the end of the stroke the final pressure is only one-third of the initial pressure.

Adiabatic Expansion Curve.—If heat is neither given to nor taken away from the gas the curve follows out that shown and is then called the "adiabatic" or varying temperature curve. Notice that during expansion the adiabatic line falls below the "isothermal" and during compression rises above it.

Entropy.—An "Entropy" diagram represents heat and work, the area representing heat units per pound, and the depth of the diagram the absolute temperature of the gas. The length of the diagram, or parts of the length, represent the "entropy." This diagram is of great value in estimating the expenditure of energy in steam or gas engines.

Gravity.—The attraction of the earth, known as gravity, causes an accelerating effect in falling bodies of 32 feet per second. This number is commonly expressed as $g = 32$.

Momentum.—Momentum means the force or energy acquired by a moving body, and is equal to the quantity of Matter multiplied by its Velocity; or, Mass × Velocity = Momentum.

Atmospheric Pressure.—At the sea level the Atmospheric Pressure varies between $14\frac{1}{2}$ lbs. (average, 14·7 lbs.) and 15 lbs. per square inch. This pressure is measured by the barometer.

Gauge Pressure.—"Gauge" Pressure is pressure above that of the atmosphere. Ordinary steam gauges indicate pressures *above* the atmosphere only.

Gross, or Absolute, Pressure.—The gauge pressure added to the atmospheric pressure is equal to the "Gross" or "Absolute" Pressure.

Initial Pressure.—The pressure at the commencement of the stroke is called the "Initial" Pressure.

Final, or Terminal, Pressure.—"Terminal" Pressure is the pressure at the end of the stroke.

Effective Pressure.—The "Effective" Pressure is the difference between the steam pressure on one side of the piston and the exhaust pressure on the other side. If the steam pressure is, say, 80 lbs., and the exhaust pressure 10 lbs., then, $80 - 10 = 70$ lbs. Effective Pressure (not *mean* effective).

Mean Effective Pressure is the average effective pressure exerted on the piston throughout the stroke, or during one revolution. This is the pressure required in calculating the Indicated Horse-Power of an engine.

Combustion is a chemical process, and consists of the combining (chemically) of Carbon of coal with Oxygen of the air, producing CO_2 and heat.

Complete combustion produces CO_2 and water.
Incomplete ,, ,, CO and smoke.

NOTE.—The small percentage of water formed in combustion is due to the combination of the Hydrogen of the Coal and the Oxygen of the air, giving H_2O.

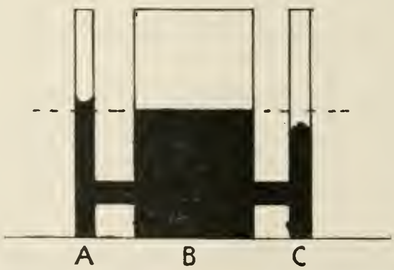
Conservation of Energy.

By this is meant that energy, like matter, is indestructible, and can only be transformed from one state to another. Energy is said to be wasted or lost in overcoming friction, for example, and this reduces the useful energy of a machine, but the total energy remains the same as originally supplied. A dynamo engine of a certain horse-power transforms mechanical energy into electrical energy, but the amount of electrical energy given out by the dynamo is less than the amount of mechanical energy supplied by the engine, as part of the energy is wasted in overcoming friction, weight, &c. Nevertheless the sum of the energy wasted and the useful energy given out by the dynamo is equal to the energy originally supplied by the engine, and can be all accounted for

Capillary Attraction.

The force which causes the oil in an oil cup to creep up the worsted, and so flow down the pipe, is known as "capillary attraction," and is due to the attraction of the molecules of the oil to those of the cotton strands. The absorption of water in a sponge is due to the same force, and the difference in level of a liquid outside and inside of a tube of very fine bore, as shown by the sketches, is another example of the same.

If in tube A the liquid moistens the tube, the level rises as shown above the normal and is concave. If in tube C the liquid does not moisten the tube, then the level is below the normal and is convex.



No. 72.—Examples of Capillary Attraction.

Vessel B shows the normal level of the liquid when free from the influence of capillary attraction.

Siphon.

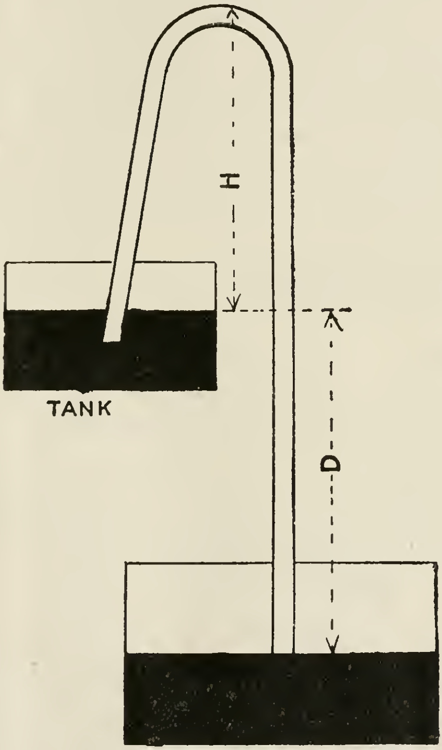
By means of a bent pipe with a long and a short leg known as a siphon, water may be caused to flow from one tank to another one lower down in position.

For the efficient working of a siphon the following requirements are necessary :—

(1.) The height H (sketch) must not exceed 26 feet, which is the practical lift of a pump by the atmospheric pressure effect.

(2.) The bent pipe must first be filled with water to start the flow, and this is usually done by drawing out the air in the pipe and so forming a vacuum.

NOTE.—The siphon will work equally well with cold or hot water, in which, it will be noted, it differs from a pump. The weight of the water in the length D of the pipe is the cause of the flow of water from the one tank to the other, and the longer this is made the faster will the upper tank be emptied of its contents.



No. 73.—Action of Siphon.

Density of Steam.

By density of steam is meant the *weight* per cubic foot volume. The density increases with the pressure, as will be seen on referring to the Steam Table, page 622.

If the specific volume of the steam be given, the density can be determined as follows :—

RULE.—Density $= 1 \div$ specific volume in cubic feet.

EXAMPLE.—At 180 lbs. pressure absolute, the specific volume of the steam is 2·49 cubic feet per lb.; express the density.

Then density $= 1 \div 2{\cdot}49 = {\cdot}401$ lb. nearly; therefore the density or weight of each cubic foot of steam at the pressure given is ·401 of a pound.

Shaft or Brake Horse-Power.

It is now well known that so far no method has been devised, or, in fact, is likely to be devised, for the indicating of the horse-power as done in the case of reciprocating engines, but the actual power transmitted along the shafting to the propeller may be determined by means of the "torsion meter," an instrument which measures the twist or torque put on the shaft by a given power. For accuracy of results it is advisable to have the shafting calibrated beforehand, as different builds of shafts and materials give slightly varying results.

It should be noted that the shaft horse-power or brake horse-power, as measured by the torsion meter, is the useful horse-power, and that the I.H.P. by comparison is a matter of indifference, the effective horse-power being actually transmitted along the shafting to the propeller being of chief importance.

Equivalent I.H.P.

Repeated trials have proved that the ratio of shaft horse-power by torsion meter as compared to indicated horse-power is usually in the ratio of 90 to 100, or ·9 to 1.

Therefore, $\text{Equivalent I.H.P.} = \text{Shaft Horse-Power} \div {\cdot}9.$

EXAMPLE.—The collective shaft horse-power by torsion meter is found to be 8100; calculate the equivalent I.H.P.

Then, $\text{Equivalent I.H.P.} = 8100 \div {\cdot}9 = 9000.$

Dryness Fraction (or Factor).—In considering the actual work done by steam, it is important that the dryness fraction be taken into account, as the result greatly depends on this quantity. After work is done by adiabatic expansion, the steam contains a certain amount of water, which proportionally reduces the internal heat still left in the steam. The dryness fraction is the ratio between the weight of dry steam per pound and the weight of the dry steam and water added together;

Or, $$\frac{\text{Weight of dry steam}}{\text{Weight of dry steam} + \text{weight of water}} = \text{Dryness Fraction.}$$

Suppose the water to be 25 per cent. of each pound weight of mixture,

Then, $$\frac{100-25}{100} = \frac{75}{100} = \frac{15}{20} = \frac{3}{4} = \text{Dryness Fraction.}$$

So that after expansion and work done by the steam the actual units or foot-pounds of energy left are, in this case, equal to the internal heat units multiplied by the fraction $\frac{3}{4}$.

Total Heat of Steam.—By the total heat of saturated, or boiler steam, is meant the number of heat units required to produce 1 lb. of steam from a temperature of 32° Fahr. to any given temperature and pressure. The total heat includes the latent heat of steam formation and the sensible or thermometer heat.

Rule.— $1083 + \cdot3 \times T° =$ Total Heat (above 32° Fahr.).
Where, $T° =$ Temperature of the steam (Fahr.).

Internal Heat of Steam.—By this is meant the heat or energy required to change 1 lb. of water into steam at any given pressure.

External Heat of Steam.—By this is meant the heat required to produce increase of volume (water to steam) against an external resistance or pressure.

Latent Heat of Steam.—The sum of the Internal heat and External heat is equal to the latent heat.

The Latent Heat can be calculated as follows :—

Rule.— $1114 - \cdot7 \times T° =$ Latent Heat.

Where, $T° =$ Temperature of the steam (Fahr.).

EXAMPLE.—Calculate the Total Heat, Latent Heat, and Sensible Heat of 1 lb. of steam at 160 lbs. pressure by gauge.

$160 + 15 = 175$ lbs. Absolute pressure and 371° Temperature (from Table, page 622).

Then,	$1083 + \cdot3 \times 371 = 1194\cdot3$	Total Heat,
and	$1114 - \cdot7 \times 371 = 854\cdot3$	Latent Heat.
Therefore,	$371° - 32° = 333\cdot9$	Sensible Heat.

NOTE.—The above are all calculated from a temperature of 32° Fahr.

Potential Energy is the energy contained or stored up in steam of a given pressure and temperature, the amount of energy contained increasing with the pressure and the temperature.

Kinetic Energy is the result of setting free the potential or stored-up energy of the steam, which then shows as active energy in the performance of work. In a steam-engine the steam acts on the pistons, and by causing motion to take place work is done, and, as a result, the steam falls in pressure and in temperature. In a turbine, the steam at a given pressure and velocity leaves the first row of guide blades, and striking the first row of moving blades gives up part of its kinetic energy, which results in a decrease in pressure and in heat.

Adiabatic Expansion.—If steam expands in a cylinder or turbine casing, and neither receives heat from any external source nor gives out any heat externally, then the expansion is said to be "adiabatic," and all work done in the cylinder or turbine is obtained at the expense of the internal heat of the steam, which in falling in pressure and temperature conforms to this condition, and part of which condenses. In the cylinders of a marine engine of the reciprocating type, the expansion is approximately hyperbolic or isothermal, and in a turbine the expansion is approximately "adiabatic."

Hyperbolic or Isothermal Expansion.—This is founded on the well-known law of Boyle and Marriot that the pressure of a gas varies inversely as the volume; or, as it is expressed—

Rule.— $P_1 \times V_1 = P_2 \times V_2 = \text{Constant.}$

Where, P_1 = Initial pressure. Where, P_2 = Final pressure.
,, V_1 = Initial volume. ,, V_2 = Final volume.

Therefore, $\frac{P_1 \times V_1}{P_2} = V_2;$

and, $\frac{P_1 \times V_1}{V_2} = P_2;$

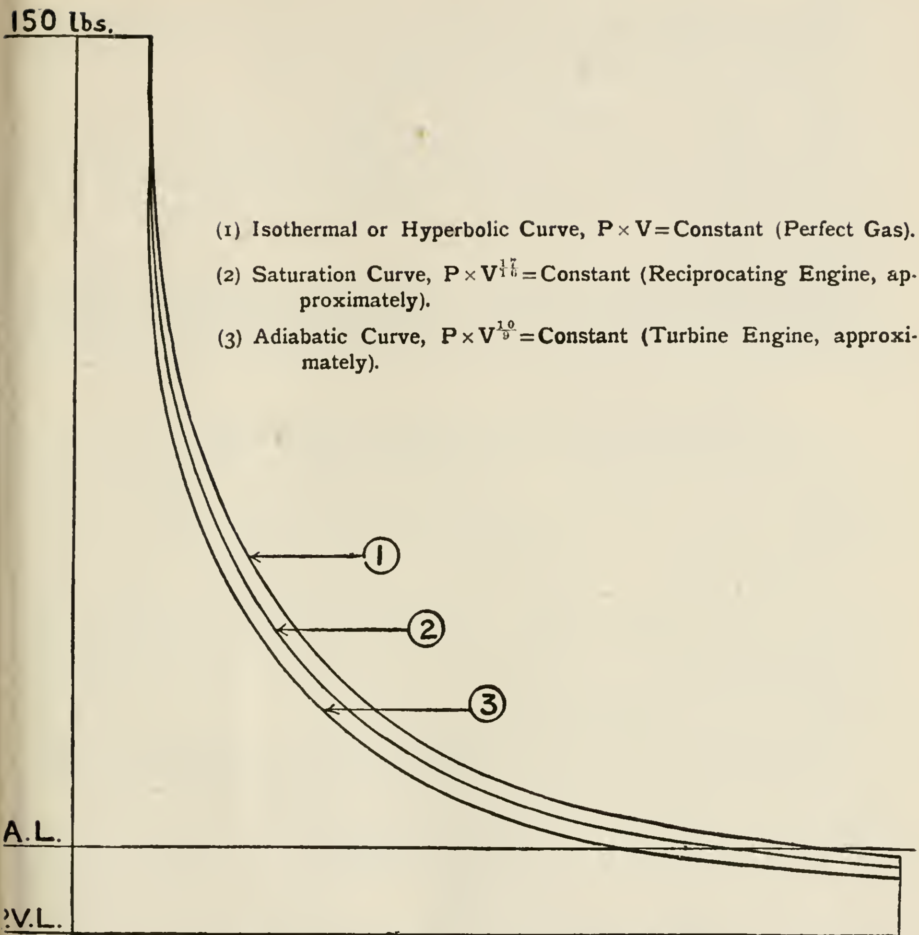
or, $\frac{P_2 \times V_2}{P_1} = V_1;$

and, $\frac{P_2 \times V_2}{V_1} = P_1.$

Foot-Pound.—A foot-pound is the work done in raising a weight of 1 lb. up through a distance of 1 foot.

Torque.—Torque is the turning movement to which a shaft is subjected when a force is exerted to rotate the shaft against a resistance such as that of the screw propeller in water. In ordinary engines the turning effort or torque is applied by means of the crank, and in turbines by the direct energy of the steam acting on the periphery of the blade circle of the rotor.

Heat.—Heat is merely a form of energy, and as such exists in two states—(1) in that of Potential or stored-up energy, and (2) in that of Kinetic or active energy. When the molecules of a body or gas are set in rapid motion or vibration, heat is developed and work done. Consequently in the case of a steam-engine, either of the reciprocating type or turbine type, the energy which produces rotation of the shaft is obtained by means of the transformation of heat energy into mechanical work.



No. 74.—Expansion Curves of Steam.

British Thermal Unit (B.Th.U.).—This is taken as being equal to 778 foot-pounds of work or energy, and signifies that one heat unit, when transformed into mechanical energy, gives out 778 foot-pounds of work.

Saturated Steam.—Steam taken direct from the boilers is known as "saturated steam," as the density, or weight of water per cubic foot, is constant for any given pressure, as also is the temperature and volume. The steam supplied to all marine engines (without superheaters) is therefore of this quality, and calculations as to expansion, work done, and fall of pressure, are usually made on this assumption. The steam supplied to the H.P. turbine of a turbine engine is therefore saturated steam. Sometimes the term "dry saturated steam" is used to distinguish this quality of steam from wet steam, or steam containing water from priming.

"Wet" Steam.—If water is carried off with the steam due to priming taking place in the boilers, the steam contains more water per cubic foot than is natural to the "saturation" pressure, volume, and temperature, and it is then known as "wet steam," or "wet saturated steam."

Superheated Steam.—If saturated steam from the boilers is passed through the tubes of a superheater, the water contained in the steam is evaporated out of it, with the following results:—

1. Rise of temperature.
2. Increase of volume if pressure is kept constant; or,
3. Increase of pressure if volume is kept constant.

The chief advantage of superheated steam lies in the fact that cylinder condensation is practically eliminated, as the steam does not then readily condense when exposed to cooled surfaces: leakage is also reduced.

Another point of importance is that the specific heat of this steam being only ·48 (some authorities give ·5), one B.T.U. of heat supplied to the steam has the effect of raising its temperature fully two degrees, as $1 \div \cdot 48 = 2 \cdot 08$.

Boyle's Law of Expansion.

Boyle's Law of expansion states that "The pressure of a gas varies inversely as the volume if kept at constant temperature."

Or, $P \times V = C$; therefore, $C \div V = P$, or, $C \div P = V$,
where P = Absolute pressure,
,, V = Volume in cubic feet,
,, C = Constant.

This means that the pressure multiplied by the volume is always equal to a constant number, or in other words what is lost in pressure is made up in volume or *vice versa*, so that the result of the multiplication is always the same.

Example 1.—The H.P. initial pressure is 185 lbs. gauge pressure, and the cut-off ·6; find the pressure at the end of the stroke.

Rule—

$$P \times V = C; \text{ therefore, } (185 + 15) \times \cdot 6 = 120 = C.$$

Again, $\overset{C}{120} \div \overset{V_2}{1} = 120$ lbs. absolute $= P_2$,

and, $120 - 15 = 105$ lbs. gauge pressure.

It should be noted that the initial pressure is $185 + 15$, or 200 lbs. absolute, and the volume ·6, also that at the end of the stroke (release) the volume will be equal to 1.

EXAMPLE 2.—Initial pressure, 160 lbs. gauge, and volume, 2·56 cubic feet; find the volume when the pressure drops down to 80 lbs. gauge.

$$\text{Then,}\quad 160+15=175=P_1,\ 80+15=95=P_2.$$

$$\text{Therefore,}\quad \overset{P_1}{175}\times\overset{V_1}{2\cdot56}=448=C,$$

$$\text{and,}\quad \overset{C}{448}\div\overset{P_2}{95}=4\cdot71\text{ cubic feet}=V_2.$$

NOTE.—On referring to the Steam Table, page 622, it will be seen that the actual volume of saturated steam at 95 lbs. pressure absolute is 4·54 cubic feet in place of 4·71 cubic feet as brought out by Boyle's Law, which difference is principally due to the fact that, under practical conditions, fall of pressure is accompanied by fall of temperature.

EXAMPLE 3.—The L.P. initial pressure is 11 lbs. by gauge, and the cut-off ·5 stroke; find the terminal pressure at end of stroke.

$$\text{Then,}\quad 11+15=26\text{ lbs.}=P_1,\text{ and }\cdot5=V_1,$$

$$\text{so that,}\quad 26\times\cdot5=13\text{ lbs. absolute}=C,$$

$$\text{then,}\quad 13\div1=13=P_2\text{ absolute.}$$

Notice that $V_2=1$, that is, the whole volume of the cylinder.

Cylinder Clearance Allowance.—For even approximate results it is necessary that the clearance volume of the cylinder should be allowed for, so that the rule corrected for this reads thus:—

(Cut-off + clearance) × Initial pressure = (Release + clearance) × Terminal pressure.

EXAMPLE 4.—H.P. initial pressure, 165 lbs. gauge; cut-off, ·6; clearance volume, 10 per cent. that of cylinder; find gauge pressure at end of stroke (release).

$$\text{Then,}\quad \text{Clearance}=\frac{1\times10}{100}=\cdot1\text{ (assuming cylinder as unit 1).}$$

$$\text{Therefore,}\quad (\cdot6+\cdot1)\times180=(1+\cdot1)\times P,$$

$$\text{so that,}\quad P=\frac{\cdot7\times180}{1\cdot1}=114\cdot5\text{ lbs. absolute,}$$

$$\text{and,}\quad 114\cdot5-15=99\cdot5\text{ lbs. gauge terminal pressure.}$$

NOTE.—If the I.P. receiver is, say, 1·4 times the capacity of the H.P. cylinder, then, 114·5 ÷ 1·4 = 81·7 lbs. absolute, and 81·7 − 15 = 66·7 lbs. on I.P. receiver gauge.

EXAMPLE 5.—Apply Boyle's Law and find the H.P., I.P., and L.P. terminal gauge pressures, also the I.P. and L.P. receiver pressures; given H.P. initial, 155 lbs. gauge; H.P. cut-off, ·6; I.P. cut-off, ·5; L.P. cut-off, ·4; clearance volume, 10 per cent. in each case. I.P. receiver = 1·4 times H.P. cylinder volume, and L.P. receiver = 1·5 times I.P. cylinder volume.

H.P. Cylinder.

$$155 + 15 = 170 \text{ absolute,}$$

$$\frac{1 \times 10}{100} = \cdot 1 \text{ clearance.}$$

Then, $(\cdot 6 + \cdot 1) \times 170 = (1 + \cdot 1) \times P.$

Therefore,

$$P = \frac{\cdot 7 \times 170}{1 \cdot 1} = 108 \text{ lbs. absolute, and } 108 - 15 = 93 \text{ lbs. gauge terminal pressure.}$$

I.P. Receiver.

H.P. terminal pressure = 108 lbs. absolute,

Therefore, Receiver = $108 \div 1 \cdot 4 = 77$ lbs. absolute,

and, $77 - 15 = 62$ lbs. gauge.

I.P. Cylinder.

$$(\cdot 5 + \cdot 1) \times 77 = (1 + \cdot 1) \times P.$$

Therefore, $P = \frac{\cdot 6 \times 77}{1 \cdot 1} = 42$ lbs. absolute,

and, $42 - 15 = 27$ lbs. gauge terminal pressure.

L.P. Receiver.

I.P. terminal pressure = 42 lbs. absolute.

Therefore, Receiver = $42 \div 1 \cdot 5 = 28$ lbs. absolute.

and, $28 - 15 = 13$ lbs. gauge.

NOTE.—The foregoing is only approximate, as the cut-off in following cylinder affects the pressure in previous receiver.

L.P. Cylinder.

$$(\cdot 4 + \cdot 1) \times 28 = (1 + \cdot 1) \times P.$$

Therefore, $P = \frac{\cdot 5 \times 28}{1 \cdot 1} = 12 \cdot 7$ lbs. absolute.

Observe that the last pressure found is equal to about 2¼ lbs. *below* that of the atmosphere.

Boyle's law of expansion may, as before stated, be expressed as follows:—

$$P_1 \times V_1 = P_2 \times V_2, \text{ or, } P_1 \times V_1 = \text{Constant.}$$

Therefore, $\frac{P_1 \times V_1}{V_2} = P_2$, or, $\frac{P_1 \times V_1}{P_2} = V_2$.

Again, $\frac{P_2 \times V_2}{P_1} = V_1$, or, $\frac{P_2 \times V_2}{V_1} = P_1$.

Where P_1 = Initial absolute pressure,
,, V_1 = Initial volume,
,, P_2 = Terminal absolute pressure,
,, V_2 = Terminal volume.

At *constant temperature* the initial pressure absolute multiplied by the volume is equal to the terminal pressure absolute multiplied by the volume, which simply means that as the pressure decreases the volume increases proportionally, or *vice versa*. What is lost in pressure is gained in volume, or what is gained in pressure is lost in volume. After the cut-off takes place we have an example of decrease in pressure and increase in volume, and when the exhaust closes we have an example of decrease of volume and increase of pressure (compression).

EXAMPLE 1.—H.P. initial pressure, 165 lbs. gauge; cut-off, ·6; find the pressure at end of stroke.

(1.) Then, $P_1 \times V_1 = P_2 \times V_2 = 180 \times \cdot 6 = P_2 \times 1.$

Therefore, $\frac{180 \times \cdot 6}{1} = P_2 = 108$ lbs. absolute,

and, $108 - 15 = 93$ lbs. by gauge. Answer.

Observe that the volume at cut-off is ·6 of stroke, and at the end of stroke the volume is 1 or the full stroke, also $165 + 15 = 180$ lbs. absolute pressure. At the end of the stroke the pressure is therefore 93 lbs. gauge, but when the steam flows into the M.P. chest the pressure drops still further owing to the receiver capacity being greater than that of the preceding cylinder, assuming that the M.P. receiver is 1·4 times the capacity of the H.P. cylinder.

Then, $108 \div 1 \cdot 4 = 77$ lbs. absolute,

and, $77 - 15 = 62$ lbs. gauge in M.P. chest.

NOTE.—It must be remembered that in all steam expansion problems the pressures must be expressed as absolute or gross.

EXAMPLE 2.—H.P. initial pressure, 170 lbs. gauge; cut-off, ·6; M.P. receiver capacity, 1·4 times that of H.P. cylinder; cut-off in M.P cylinder, ·5; L.P. receiver capacity, 1·5 times that of M.P. cylinder; cut-off in L.P. cylinder, ·5. Determine (1) the H.P. terminal gauge pressure, (2) the M.P. receiver gauge pressure, (3) the M.P. terminal gauge pressure, (4) the L.P. receiver gauge pressure, and (5) the L.P. terminal absolute pressure.

Then, $170 + 15 = 185$ lbs. absolute H.P. initial pressure,

and, $P_1 \times V_1 = P_2 \times V_2 = 185 \times \cdot 6 = P_2 \times 1.$

Therefore,

$\frac{185 \times \cdot 6}{1} = 111$ lbs. absolute, and $111 - 15 = 96$ lbs. gauge terminal pressure H.P.

Again,

$111 \div 1 \cdot 4 = 79 \cdot 2$ lbs. absolute, and $79 \cdot 2 - 15 = 64 \cdot 2$ lbs. gauge M.P. receiver pressure.

(2.) $P_1 \times V_1 = P_2 \times V_2 = 79 \cdot 2 \times \cdot 5 = P_2 \times 1.$

Therefore,

$\frac{79 \cdot 2 \times \cdot 5}{1} = 39 \cdot 6$ lbs. absolute, and $39 \cdot 6 - 15 = 24 \cdot 6$ lbs. gauge terminal pressure M.P.

Again, $39 \cdot 6 \div 1 \cdot 5 = 26 \cdot 4$ lbs. absolute in L.P. receiver,

or, $26 \cdot 4 - 15 = 11 \cdot 4$ lbs. gauge in L.P. receiver.

(3.) $$P_1 \times V_1 = P_2 \times V_2 = 26{\cdot}4 \times {\cdot}5 = P_2 \times 1.$$

Therefore, $\frac{26{\cdot}4 \times {\cdot}5}{1} = 13{\cdot}2$ lbs. absolute terminal L.P. pressure.

Observe that the L.P. terminal pressure is *below* that of the atmosphere.

It should be noted that the foregoing rule assumes the steam to act as a perfect gas, whereas in actual practice the conditions are somewhat different as shown below.

(1.) Difference due to initial cylinder condensation and re-evaporation.

(2.) Difference due to the steam being of the condition known as "saturated" (see page 622).

(3.) Difference due to work done by the steam on the piston.

These differences produce a drop in the pressure for any given expansion below that as determined by the rule given. The saturated steam expansion curve drawn out on page 375 illustrates clearly the difference referred to and should be compared with the isothermal curve.

Steam Expansions by Pressures and by Volumes.—The differences in the action of the steam under practical conditions, as compared with Boyle's Law, naturally results in a difference in the number of expansions obtained throughout the cylinders.

EXAMPLE 1.—Find the total number of expansions by pressures if the H.P. initial pressure is 165 lbs. gauge, and the L.P. terminal pressure 12 lbs. absolute; also by volumes if the cylinder ratio is as 1 : 2·7 : 7·2 and the H.P. cut-off ·6.

RULE.—H.P. initial pressure (absolute) ÷ L.P. terminal pressure (absolute) = No. of Expansions by Pressures.

$165 + 15 = 180$ lbs. absolute.

Therefore, $180 \div 12 = 15$ Expansions by pressures.

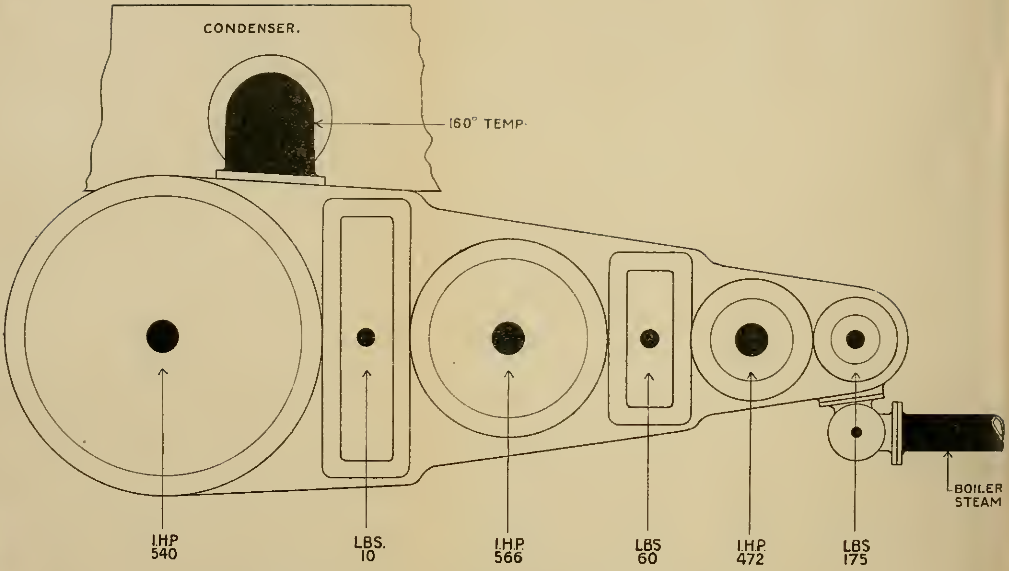
RULE.—L.P. ratio ÷ H.P. cut-off = No. of Expansions by Volumes.

Therefore, $7{\cdot}2 \div {\cdot}6 = 12$ Expansions by volumes.

In practice the pressure at the end of L.P. stroke, being less than that found by Boyle's Law, gives a correspondingly increased number of expansions as compared with the number of expansions obtained by the volumes.

Charles' Law.

A. The pressure of a gas at constant volume varies with its absolute temperature.



No. 74a.—Pressure and I.H.P. for Heat Efficiency Calculation.

[To face page 381.

B. The volume of a gas at constant pressure varies with its absolute temperature.

EXAMPLE 1.—A superheater contains 200 cubic feet of steam at a constant pressure of 165 lbs. gauge; find the volume when the temperature is raised to 450° Fahr.

NOTE.—Absolute temperature = Fahr. Temp. + 461°.

Therefore, $165 + 15 = 180$ lbs. Absolute and Temperature of 373° Fahr.

and, $373 + 461 = 834°$ Absolute Temp.,
again, $450 + 461 = 911°$,, ,,
then, As $834 : 911 :: 200 : 218{\cdot}4$ cubic feet. Answer.

EXAMPLE 2.—If the pressure of a gas is 150 lbs. absolute, and the temperature 366° Fahr., find the pressure if the gas is heated up to 400° Fahr.

Then, $366 + 461 = 827$, and $400 + 461 = 861$.
Therefore, As $827 : 861 :: 150 : 157{\cdot}4$ lbs. Absolute. Answer

Heat or Thermal Efficiency.

DATA—

Cylinders—24, 40, 65 inches.
Stroke—3 feet 6 inches.
Revolutions—72.
H.P. steam—175 lbs. gauge.
I.P. steam—60 ,,
L.P. steam—10 ,,
Vacuum—24 inches.

I.H.P. of H.P. cylinder - -	472
I.H.P. of I.P. cylinder - -	566
I.H.P. of L.P. cylinder - -	540
I.H.P. collective - - -	1,578

Coal per twenty-four hours—28 tons.
Coal per I.H.P. hour—1·65 lbs.

RULE.—

Work done ÷ Heat supplied = Efficiency,

and, Heat supplied − Work done = Heat rejected.

Again, Heat supplied (per pound water or steam) $= 1115 + {\cdot}3 \times T° - t°$.

Where, $T°$ = H.P. Initial steam temperature.
$t°$ = L.P. Exhaust steam temperature.

Work done in heat units per I.H.P. $= \frac{33000 \times 60}{778} = 2545$ Heat Units per hour.

NOTE.—778 foot-pounds of work = 1 Heat unit value.

Heat supplied per I.H.P. hour = pounds feed water per I.H.P. × Heat per pound.

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381.

B. The volume of a gas at constant pressure varies with its absolute temperature.

EXAMPLE 1.—A superheater contains 200 cubic feet of steam at a constant pressure of 165 lbs. gauge; find the volume when the temperature is raised to 450° Fahr.

NOTE.—Absolute temperature = Fahr. Temp. + 461°.

Therefore, 165 + 15 = 180 lbs. Absolute and Temperature of 373° Fahr.

and, 373 + 461 = 834° Absolute Temp.,
again, 450 + 461 = 911° „ „
then, As 834 : 911 : : 200 : 218·4 cubic feet. Answer.

EXAMPLE 2.—If the pressure of a gas is 150 lbs. absolute, and the temperature 366° Fahr., find the pressure if the gas is heated up to 400° Fahr.

Then, 366 + 461 = 827, and 400 + 461 = 861.
Therefore, As 827 : 861 : : 150 : 157·4 lbs. Absolute. Answer

Heat or Thermal Efficiency.

DATA—

Cylinders—24, 40, 65 inches.
Stroke—3 feet 6 inches.
Revolutions—72.
H.P. steam—175 lbs. gauge.
I.P. steam—60 „
L.P. steam—10 „
Vacuum—24 inches.

I.H.P. of H.P. cylinder - -	472
I.H.P. of I.P. cylinder - -	566
I.H.P. of L.P. cylinder - -	540
I.H.P. collective - - -	1,578

Coal per twenty-four hours—28 tons.
Coal per I.H.P. hour—1·65 lbs.

RULE.—

Work done ÷ Heat supplied = Efficiency,

and, Heat supplied − Work done = Heat rejected.

Again, Heat supplied (per pound water or steam) = $1115 + \cdot 3 \times T^\circ - t^\circ$.

Where, T° = H.P. Initial steam temperature.
t° = L.P. Exhaust steam temperature.

Work done in heat units per I.H.P. $= \frac{33000 \times 60}{778} = 2545$ Heat Units per hour.

NOTE.—778 foot-pounds of work = 1 Heat unit value.

Heat supplied per I.H.P. hour = pounds feed water per I.H.P. × Heat per pound.

Application.—To apply the above rules to the case shown in the sketch facing page 381 :—

Heat supplied $= 1{,}115 + \cdot 3 \times 377 \cdot 5 - 160 = 1068 \cdot 25$ Heat Units per pound steam.

NOTE.—$175 + 15 = 190$ lbs. absolute, and temperature (from Table, page 622) $377 \cdot 5$.

Heat given up as work per I.H.P. hour $= 2545$ units.

$$\left.\begin{array}{r}\text{Pounds water (or steam) per I.H.P. hour} \\ \text{(Evaporation assumed as 8·8 lbs.)}\end{array}\right\} = \frac{28 \times 2240 \times 8 \cdot 8}{1578 \times 24} = 14 \cdot 6 \text{ lbs. nearly.}$$

NOTE.—The evaporation of water per pound coal is the most troublesome item to obtain with any degree of accuracy; it can, of course, be determined by actual evaporative tests, but the most satisfactory method is that adopted in Admiralty trials where measuring tanks are employed which record the actual amount of feed water entering the boilers during a given period. It should be noted that the steam (or water) used per I.H.P. is the true test of economy as the quality of coal varies greatly, and therefore does not constitute a reliable standard of comparison.

Then, $14 \cdot 6 \times 1068 \cdot 25 = 15489 \cdot 625$ Heat Units supplied per hour

Therefore, Efficiency $= 2545 \div 15489 \cdot 625 = \cdot 164$,

and $\cdot 164 \times 100 = 16 \cdot 4$ per cent. Thermal Efficiency.

It will thus be seen that of 15489·625 Heat Units supplied only 2545 Heat Units appear as actual work done in the engine.

Pressures, Volumes (Sketch facing page 381).

The following data of pressures, volumes, and temperatures throughout the range of cylinders and receivers should be carefully studied by the student.

H.P. valve chest { Pressure = 175 lbs. gauge.
Specific Volume = 2·43 cubic feet per pound.
Temperature = 377·5° Fahr.

I.P. valve chest { Pressure = 60 lbs. gauge.
Specific Volume = 5·68 cubic feet per pound.
Temperature = 307·5° Fahr.

L.P. valve chest { Pressure = 10 lbs. gauge.
Specific Volume = 15·97 cubic feet per pound.
Temperature = 240° Fahr.

Condenser - - { Pressure = 2·8 lbs. absolute.
Specific Volume = 117 cubic feet per pound.
Temperature = 140° Fahr.

NOTE.—The above figures assume saturated steam at all stages of expansion, but this is not strictly the case in practice, as the steam expands to a certain amount adiabatically, which results in reduced steam volume per pound at the latter stages of expansion, part of the steam condensing in the performance of work. The "dryness fractions" of the steam produced in this way may therefore show somewhat like the following :—

	H.P. Valve Chest.	H.P. Cylinder.	I.P. Cylinder.	L.P. Cylinder.
Dryness Fraction of Steam -	1 (dry)	·9	·8	·78

As a set-back against the above it should, however, be noted that the steam in the receivers is more or less superheated, owing to simple expansion into the receivers from the preceding cylinders, without actual work being done during such expansion.

Taking the I.P. chest the pressure is 75 lbs. absolute, and the specific volume 5·68, then $5{\cdot}68 \times {\cdot}8 = 4{\cdot}524$ cubic feet, as the actual volume when the dryness is ·8, the remainder of the steam having condensed in the performance of work.

Steam Consumption per Revolution.—For methods of calculating the steam (or feed water) consumption of an engine see author's "Marine Indicator Cards."

Water formed by Initial Condensation.—The weight of water produced by cylinder initial condensation may be closely approximated by the following rule, assuming that the H.P. initial steam is of dry saturated quality and free from priming water.

Steam condensed = Pounds steam in H.P. per rev. − (pounds steam in L.P. per rev, + steam condensed by work done in engine).

Example.—The weight of steam used per revolution in the H.P. cylinder from the H.P. cards is 6·2 lbs., and the steam used in the L.P. cylinder from the L.P. cards is 4·25 lbs. If the weight of steam condensed by work done is 1·3 lbs., find the amount of water formed by initial condensation in the cylinders.

Then, $\text{weight} = 6{\cdot}2 - (4{\cdot}25 + 1{\cdot}3) = {\cdot}65$ of a lb. per revolution.

Work done during Adiabatic Expansion.—To calculate the work done, or, which is the same thing, the units of heat given up or converted into work during the adiabatic expansion of steam in a turbine, the following data are required :—

The absolute temperature of the steam before and after expansion.
The latent heat of the steam before and after expansion.
The dryness factor of the steam before and after expansion.

Let, T_1° = Absolute temperature before expansion.
,, T_2° = Absolute temperature after expansion.
,, H_1 = Latent heat before expansion.
,, H_2 = Latent heat after expansion.
,, f_1 = Dryness factor before expansion.
,, f_2 = Dryness factor after expansion.

The heat energy given out in British Thermal Units =

$$f_1 \times H_1 - f_2 \times H_2 + T_1^\circ - T_2^\circ = \text{B.T.U.}$$

EXAMPLE.—Find the work done per pound of steam in expanding adiabatically from an H.P. initial pressure of 180 lbs. gauge, to a terminal L.P. pressure of 10 lbs. absolute, the dryness fractions being ·99 and ·76 respectively.

Then, $180 + 15 = 195$ lbs. absolute $= 379{\cdot}7$ temperature from Table, page 622.
Latent heat $= 846{\cdot}5$ B.T.U. from Table.
And, 10 lbs. absolute $= 193{\cdot}3$ temperature.
Latent heat $= 1140{\cdot}3$ B.T.U.
Therefore, $379{\cdot}7 + 461 = 840{\cdot}7$ absolute temperature,
and, $193{\cdot}3 + 461 = 654{\cdot}3$ absolute temperature.

Then,
$$\overset{f_1}{{\cdot}99} \times \overset{H_1}{846{\cdot}5} - \overset{f_2}{{\cdot}76} \times \overset{H_2}{1140{\cdot}3} + \overset{T_1}{840{\cdot}7} - \overset{T_2}{654{\cdot}3} =$$

$$838{\cdot}035 - 866{\cdot}628 + 840{\cdot}7 - 654{\cdot}3 = 1678{\cdot}735 - 1520{\cdot}928 = 157{\cdot}807 \text{ B.T.U.}$$

$$\text{Foot-pounds} = 157{\cdot}807 \times 778 = 122768{\cdot}4 \text{ foot-pounds.}$$

Advantages of High Pressure Steam.

To prove the economy of high pressure steam as compared with low pressure steam.

Compound 80 lbs. pressure = 324° temperature.
Triple 180 ,, ,, = 380° ,,
$1115 + .3 \times 324 = 1212{\cdot}4$ units of heat required.
$1115 + .3 \times 380 = 1229$ units ,, ,,

Then $1229 - 1212{\cdot}4 = 16{\cdot}6$ additional units of heat required to give more than double the pressure.

High pressure steam is stronger and more expansive than low pressure steam, therefore a lesser quantity of it will do the same work.

$$\text{So that, per cent. extra fuel} = \frac{16{\cdot}6 \times 100}{1212{\cdot}4} = 1{\cdot}36 \text{ per cent.}$$

It can be proved by calculation that the higher pressure steam gives out fully 20 per cent. more power than the lower pressure steam, owing to the greater range of expansion obtained: therefore the clear gain in economy resulting from the higher pressure $= 20 - 1{\cdot}36 - 18{\cdot}64$ per cent. (see also pages 398-399).

Advantages of Using a Number of Cylinders.

Neglecting the advantage of two or more cranks in regard to the stresses on the crank shafting, the principal gain by having, say, three cylinders is that the pressure is lowered a certain amount in each cylinder, and the drop of temperature does not take place all at once, but is divided into three stages; the re-evaporation of the H.P. doing work in the M.P. cylinder, and the re-evaporation of the M.P. doing work in the L.P. cylinder, only the L.P. re-evaporation of the L.P. being lost in the condenser; therefore the condensation losses, due to the cylinder cooling down during exhaust, are much reduced.

If we were to use steam of 200 lbs. pressure in one cylinder, instead of in three cylinders, the great difference of temperature occurring between admission and exhaust would cause excessive condensation to take place owing to the cooling of the cylinder during exhaust, but, by dividing this drop of temperature into a number of cylinders the condensation losses are proportionally decreased owing to the counterbalance by re-evaporation.

In a single cylinder engine the re-evaporation creates back pressure, which reduces the power, and thus represents a loss.

Cylinder Ratios and Steam Expansions.

In compound engines the ratio of H.P. to L.P. cylinder is usually about 1 to 4, and in triple expansion engines about 1 to 7, or 1 to 7·25.

To find cylinder ratios—L.P. dia.^{2} ÷ H.P. dia.^{2} = Ratio.
or L.P. dia.^{2} ÷ I.P. dia.^{2} = Ratio.

If the steam is cut off at half stroke in the H.P. cylinder, this equals two expansions in the H.P.; then $2 \times 4 = 8$ expansions altogether in compound engines.

For triple, $2 \times 7 = 14$ expansions, or $7 \div \frac{1}{2} = 14$ expansions.
Or, L.P. Ratio ÷ H.P. cut-off = total Expansions.

EXAMPLE.—Cylinder Ratios are as 1 : 2·7 and 7·2 H.P. cut-off ·6; find No. of expansions by volumes.

Then, $7{\cdot}2 \div {\cdot}6 = 12$ Expansions (Volumes).

NOTE.—To find number of expansions by pressures, divide initial absolute pressure by final absolute pressure.

If we take the cut-off at one-third stroke, the $3 \times 4 = 12$ expansions in compound engines, and $3 \times 7 = 21$ expansions in triple engines.

In working out the number of steam expansions the size of the H.P. cylinder and L.P. cylinder only are required, as the I.P. makes no difference in the result, the reason being that the *initial* and *final* volumes of the steam limit the expansion range.

Cut-off and Pressures.

The final pressure, or pressure at the end of the stroke, is exactly proportional to the cut-off.

If the cut-off is at half stroke, the final pressure will be half of the initial pressure, and so on.

EXAMPLE.—Initial pressure, 100 lbs. gross; cut-off, 15 in.; and stroke, 30 in.; find the pressure at 20 in. of the stroke, at 25 in., and at the end of the stroke.

$$\frac{100 \text{ lbs.} \times 15 \text{ in.}}{20 \text{ in.}} = 75 \text{ lbs. at 20 in. of stroke.}$$

$$\frac{100 \text{ lbs.} \times 15 \text{ in.}}{25 \text{ in.}} = 60 \text{ lbs. at 25 in. of stroke.}$$

$$\frac{100 \text{ lbs.} \times 15 \text{ in.}}{30 \text{ in.}} = 50 \text{ lbs. at end of stroke.}$$

NOTE.—The above are all gross pressures; therefore if the answer is required to be expressed as gauge pressure, 15 lbs. must be subtracted from the result in each case.

Expansion of Steam and Heat.—Steam expanding in a cylinder, and doing work on the piston, falls in pressure and in temperature; the fall in heat ("heat drop") corresponding to 1 B.T.U. for each 778 foot-pounds of work done.

Steam expanding *without* doing work, as for example from the exhausting position of one cylinder to the receiver of the next engine, falls in pressure by expansion, but only slightly in temperature; in a word, the steam becomes superheated. Steam therefore in the I.P. or L.P. receivers, at a given pressure, is usually at a higher temperature than that corresponding to the pressure shown on the gauge.

A similar result is obtained with reduced steam from a reducing valve, as the steam having only to do a small amount of work in compressing the spring, falls in pressure, but not nearly so much, proportionally, in temperature; the reduced steam is thus, to a certain degree, superheated.

Expansion of Water by Heat.

The following table shows the relative volume of water at various pressures and temperatures, and the gradual expansion with rise of temperature should be noted.

Temperature and Relative Volume of Water.

Pressure (Gauge).	Temperature.	Volume.
0	212°	1·043
60	307·5°	1·093
160	370·8°	1·136
180	379·7°	1·142
200	381·7°	1·148
220	389·9°	1·154

From the above it will be seen that for a given weight of water (say 1 lb.) the level will be higher with a higher temperature. An example of this may be found in the case of the water gauge glass, which in many cases shows a lower level than the actual water level in the boiler, the difference being due to the colder water in the glass occupying a lower level.

Suction Lift of Pumps.

The suction lift of a pump depends on the vacuum obtained and on the atmospheric pressure, so that when the barometer indicates, say, 28 inches, the pump lift will be less than when the barometer indicates, say, 30 inches, the difference of 1 lb. accounting for nearly 2 feet of difference in lift. Strictly speaking, no such force as suction exists, only difference in pressure, but the word suction is still in use with reference to valves, &c.

At a higher level than the sea the atmospheric pressure is less, therefore a pump will have less lift in due proportion. It should also be remembered that the vacuum will *show* less when the atmospheric pressure is lessened, as the pressure on the outside of the U tube tending to bend it is reduced with the same pressure inside the tube; this will naturally give the tube less curvature.

Vertical Suction Lift of Pumps at Different Atmospheric Pressures.

Barometer Reading.	Practical Vertical Lift (approx.).
29·4 inches	25 feet.
28 ,,	24 ,,
26·6 ,,	23 ,,
22·8 ,,	19 ,,
19·7 ,,	17 ,,

Material or Shafting.—Shafting is now generally constructed of ingot steel, that is, large masses of steel known as ingots, which after casting are afterwards turned down by machine to the required diameter, the ingots being originally much in excess of the size wanted to ensure soundness of material. Good scrap iron forged is also employed for shafting, propeller lengths particularly, many builders preferring this to steel, owing to its less corrosive action in sea water.

Stresses on Shafting.—All lengths of shafting are subjected to a torsional stress due to the twisting moment set up by the cranks; in addition to this the crank shafting has to withstand a bending stress, produced by the bending moment of half the piston load multiplied by the distance from centre of crank-pin to centre of main bearing.

The propeller shaft is also subjected to a bending stress, set up by the bending moment of the propeller weight multiplied by the distance from centre of boss to stern post. To allow for these excess stresses it is necessary to increase the diameter of these shafts over that of the tunnel lengths.

The general stresses on shafting may therefore be classed as follows:—

Crank Shafting	Torsional. Bending.
Tunnel Shafting	Torsional. End compression (ahead). End tensile (astern).
Propeller Shafting	Torsional. Bending. End compression (ahead). End tensile (astern).

It will be obvious that if the propeller rises out of the water during racing the bending moment and stress will be much increased.

To Line up Crank Shaft.

1. Disconnect bottom ends and hang up connecting rods and eccentric rods.
2. Take out keeps and top half bearings.
3. Jack up the shaft until coupling faces all fair, or test with bridge gauge for level.
4. Take out bottom half bearings, refill with W.M. and bore out true.
5. Replace bearings, lower shaft into place, and test again before finally connecting up the engine.

NOTE.—If the wear-down is slight and the engine not of large power, liners may be fitted in below the bottom half bearings instead of rebushing the same.

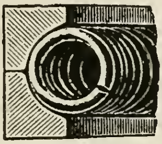
Pistons.—For highest efficiency the piston rings require to join steam-tight joints at three places.

1. Between rings and cylinder walls.
2. Between rings (top edge) and junk ring.
3. Between rings (bottom edge) and piston flange.

At the same time the rings require to be a floating fit, otherwise the springs or compression of the rings would be ineffective in preventing steam leakage from one side of the piston to the other.

All patent types of piston rings aim at forming the threefold joint referred to, which, however, is not easily attained in practice. The Buckley, Lancaster, and other rings fitted with coiled springs are designed so that the spring pressure exerts a force outwardly and at the same time presses the two half rings away from each other, thus forming a steam-tight joint between the upper ring and the junk ring, and between the lower ring and the piston flange, the idea being to prevent steam leakage to the back of the rings, which, if taking place, results in excessive friction and wear of the rings and cylinder barrel. This will perhaps be understood when it is stated that a pressure of about 4 lbs. per square inch on the rings is sufficient to prevent piston leakage in ordinary cases.

On the down stroke the friction between the rings and cylinder barrel will naturally produce contact between the upper edge of the top ring and the junk ring and so form a steam-tight joint, but for the same reason the lower ring is apt to come away from the piston flange unless held firmly in position by the piston springs, if not, the exhaust pressure (about 60 lbs. in the case of H.P. cylinders) will be admitted to the back of the rings and results in the serious frictional effects referred to. On the up stroke the positions of the rings are reversed, the lower edge of the bottom ring bearing hard against the piston flange and so preventing the admission of steam to the back of the rings, but the upper ring will now be loose, unless kept in place by the springs, and will allow the admission of exhaust pressure steam to the back of the rings.



No. 75.—Lancaster Piston Ring and Spring.

The rings are made from a special mixture of cast iron, containing principally cold blast iron and hematite, giving a tensile strength of 20 tons per square inch, making them exceedingly strong, close-grained, hard, and capable of taking a high polish.

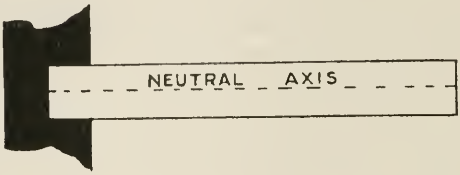
To alter the length and tension of the spring (if necessary) the ends are screwed into each other, and by varying the distance which one end screws into the other, the diameter of the spring can be altered so as to put more or less pressure on the rings, thus furnishing compensation for increased steam pressure or wear. The spring has ample bearing inside the rings so that it cannot wear away.

In the case of patent feed pumps the water pistons require even more careful adjustment, as if leakage to the back of the rings occurs the water pressure is much in excess of the steam piston leakage pressure. The side of the water piston where the rings would be loose may have a pressure of 220 lbs. or more in the case of boiler feeding, and if leakage occurred would result in enormous friction between the rings and pump barrel.

Beams.

Many examples of beam construction occur in marine engineering practice, such as, for example, in main-bearing keeps, escape-valve bridges, pump levers, crank webs, tail-end shafts, &c., and a few simple examples of the general principles involved should be found of service.

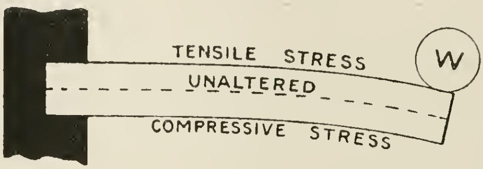
Neutral Axis.—A beam or lever fixed at one end (cantilever) and loaded at the other end (Sketch No. 1) is subjected to a tensile stress



No. 76.—Neutral Axis of a Beam.

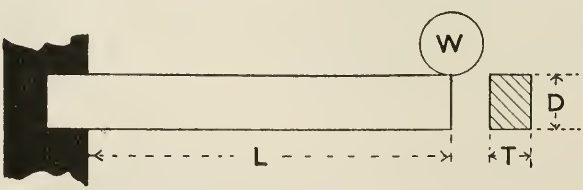
at the upper edge and a compressive stress at the lower edge (Sketch No. 2); but at the neutral axis, situated midway between the two, there is neither tension nor compression stress.

This will perhaps be understood when it is noticed that the upper edge is *lengthened* (Sketch No. 2) by the effect of the load producing



No. 77.—Stresses on Loaded Beam.

a bending tendency, and the lower edge proportionally *shortened*, but at the neutral axis the length remains unchanged, and the beam is therefore subject to neither tension nor compression at this position. From the neutral axis upwards the tensile stress increases from 0 to a maximum, and from the neutral axis downwards the compressive stress increases from 0 to a maximum, and allowing for a mean position of stress and for the beam cross-sectional area, the constant number 6 is obtained, which is employed in the equation connecting the bending moment, load, and stress per square inch.



No. 78.

No. 1 (Sketch No. 78).

RULE.—

$$6 \times W \times L = D^2 \times T \times \text{Stress}.$$

$$\text{Bending Moment} = L \times W.$$

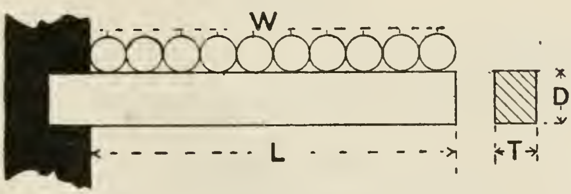
Therefore,
$$\frac{6 \times W \times L}{D^2 \times T} = \text{Stress, or } \sqrt{\frac{6 \times W \times L}{T \times \text{Stress}}} = D,$$

$$W = \frac{D^2 \times T \times \text{Stress}}{6 \times L},$$

$$L = \frac{D^2 \times T \times \text{Stress}}{6 \times W},$$

$$T = \frac{6 \times W \times L}{D^2 \times \text{Stress}}.$$

NOTE.—The strength of a beam varies directly as the Depth² and Thickness and inversely as the Length, or as $\frac{D^2 \times T}{L}$.



No. 79.

No. 2 (Sketch No. 79).

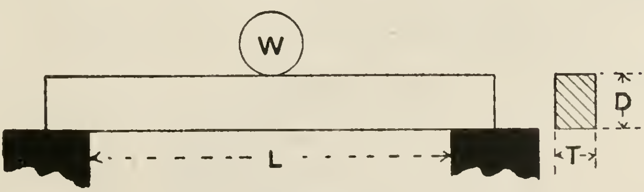
RULE.—

$$6 \times W \times L = D^2 \times T \times \text{Stress} \times 2.$$

Therefore,
$$S = \frac{6 \times W \times L}{D^2 \times T \times 2},$$

$$\text{Bending Moment} = \frac{L \times W}{2}.$$

NOTE.—If Length is in feet and Weight in pounds, then B.M. (Bending Moment) is expressed in Foot-Pounds.



No. 80.

No. 3 (Sketch No. 80).

RULE.—

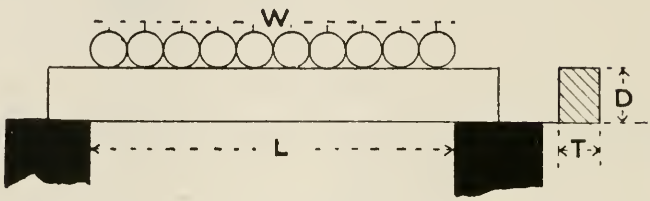
$$6 \times W \times L = D^2 \times T \times \text{Stress} \times 4.$$

Therefore,

$$\text{Stress} = \frac{6 \times W \times L}{D^2 \times T \times 4},$$

$$\text{Bending Moment} = \frac{L \times W}{4}.$$

NOTE.—If Length is in feet and Weight in tons, then B.M. is expressed in Foot-Tons.



No. 81.

No. 4 (Sketch No. 81).

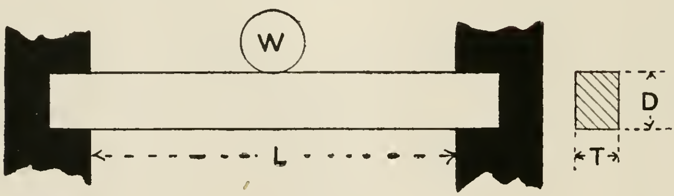
RULE.—

$$6 \times W \times L = D^2 \times T \times \text{Stress} \times 8.$$

Therefore,

$$\text{Stress} = \frac{6 \times W \times L}{D^2 \times T \times 8},$$

$$\text{Bending Moment (B.M.)} = \frac{L \times W}{8}.$$



No. 82.

No. 5 (Sketch No. 82).

RULE.—

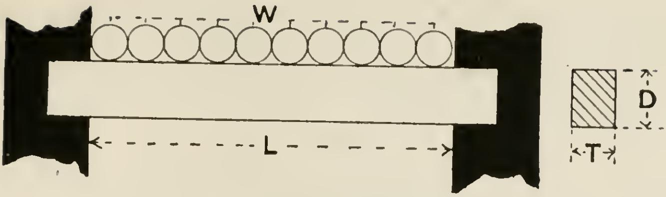
$$6 \times W \times L = D^2 \times T \times \text{Stress} \times 8. \quad \text{(Same as last example.)}$$

Therefore,

$$\text{Stress} = \frac{6 \times W \times L}{D^2 \times T \times 8},$$

$$D = \sqrt{\frac{6 \times W \times L}{T \times \text{Stress} \times 8}},$$

$$\text{B.M.} = \frac{L \times W}{8}.$$



No. 83.

No. 6 (Sketch No. 83).

RULE.—

$$6 \times W \times L = D^2 \times T \times \text{Stress} \times 12.$$

Therefore,

$$\text{Stress} = \frac{6 \times W \times L}{D^2 \times T \times 12},$$

$$T = \frac{6 \times W \times L}{T \times \text{Stress} \times 12},$$

$$D = \sqrt{\frac{6 \times W \times L}{T \times \text{Stress} \times 12}},$$

$$\text{B.M.} = \frac{L \times W}{12}.$$

EXAMPLE NO. 1.—Find the bending stress per square inch on a beam fixed at one end and loaded at the other end by a weight of 5 tons. The beam is 6 feet long, 10 inches deep, and 3 inches thick.

C.

Then, $6 \times 72 \text{ inches} \times 5 \times 2240 = 10^2 \times 3 \times \text{Stress}.$

Therefore, $\text{Stress} = \dfrac{6 \times 72 \times 5 \times 2240}{10^2 \times 3} = 1612{\cdot}6$ lbs. per square inch.

EXAMPLE NO. 2.—Calculate the required depth of lever for a lever safety valve. Length from valve to weight 25 inches, weight 20 lbs., thickness of lever $\frac{1}{2}$ inch, and the stress on lever metal 3000 lbs. per square inch.

Then, $6 \times 25 \text{ inches} \times 20 = D^2 \times {\cdot}5 \text{ inch} \times 3000.$

Therefore, $\text{Depth} = \sqrt{\dfrac{6 \times 25 \times 20}{{\cdot}5 \times 3000}} = 1{\cdot}4$ inches (say $1\frac{1}{2}$ inches).

EXAMPLE NO. 3.—Piston 24 inches diameter, pressure 120 lbs. per square inch, distance between centres of connecting rod bottom end bolts 18 inches. Find the required thickness of the cap if the width is 9 inches and the stress not to exceed 6000 lbs. per square inch.

NOTE.—Assume the construction to be as that of a beam supported at both ends and loaded throughout. (Sketch No. 81.)

$$\text{Load} = 24^2 \times {\cdot}7854 \times 120 = 54360 \text{ lbs.}$$

Then, $6 \times 54360 \times 18 \text{ inches} = D^2 \times 9 \text{ inches} \times 8 \times 6000.$

Therefore, $D = \sqrt{\dfrac{6 \times 54360 \times 18}{9 \times 8 \times 6000}} = 3{\cdot}7$ inches (say $3\frac{3}{4}$ inches).

EXAMPLE NO. 4.—A condenser door weighs 1000 lbs., and when taken off is hung on a bar $1\frac{1}{2}$ inches thick and 18 inches in length. Find the required depth of the bar if the stress on the metal is not to exceed 4000 lbs. per square inch.

NOTE.—This case is similar to Sketch No. 78.

Then, $6 \times 18 \text{ inches} \times 1000 = D^2 \times 1\cdot5 \text{ inches} \times 4000.$

Therefore, $D = \sqrt{\dfrac{6 \times 18 \times 1000}{1 \times 4000}} = 4\cdot2$ inches (say $4\frac{1}{4}$ inches).

EXAMPLE NO. 5.—Find the required depth of the bridge bar of a feed-pump spring-loaded relief valve if the distance between the pillar studs is 5 inches and the width of bridge 3 inches, valve $2\frac{1}{2}$ inches diameter, and loaded to 50 lbs. per square inch; stress 3000 lbs.

NOTE.—This case may be assumed as being similar to Sketch No. 82.

$$\text{Load} = 2\cdot5^2 \times \cdot7854 \times 170 = 833 \text{ lbs.}$$

Then, $6 \times 5 \text{ inches} \times 833 = D^2 \times 3 \text{ inches} \times 8 \times 2250.$

Therefore, $D = \sqrt{\dfrac{6 \times 5 \times 833}{3 \times 8 \times 2250}} = \cdot68$ inch (say $\frac{11}{16}$ inch).

Consumption and Speed.

At ordinary speeds the consumption or I.H.P. varies as the cube of the speed.

EXAMPLE.—The consumption per day is 14 tons and the speed 11 knots; find the consumption if the speed is increased to 12 knots.

$$\text{As } 11^3 : 12^3 :: 14 = 18\cdot17 \text{ tons per day.}$$

From this it will be seen that to increase the speed by 1 knot per hour the consumption increases 4·17 tons per day.

EXAMPLE.—A twin screw steamer develops 2000 I.H.P. in each set of engines, or 4000 I.H.P. in all, and runs at a speed of 14 knots; find the speed when running with one set of engines only, and developing 2000 I.H.P.

$$\text{As } \sqrt[3]{4000 : 2000 :: 14^3} :: 11 \text{ knots,}$$

so that with 4000 I.H.P. the speed will be 14 knots, and with 2,000 I.H.P. 11 knots.

NOTE.—The cube root requires to be extracted.

Speed and Slip.

The engine speed in knots per hour is worked out as follows:—

$$\frac{P \times R \times 60}{6080} = \text{engine speed per hour.}$$

P = propeller pitch. 60 = minutes per hour.
R = revolutions per minute. 6080 = feet per knot.

If the ship's speed per hour be subtracted from the engine speed, the difference is the apparent slip. To express the slip as a percentage proceed as follows :—

As Engine knots : Slip knots : : 100 per cent. = per cent. of slip (apparent).

EXAMPLE.—The pitch is 20 feet, the revolutions per minute 70, and the speed of the ship 12 knots ; find the per cent. of slip.

$$\frac{20 \times 70 \times 60}{6080} = 13 \cdot 81.$$

And, $13 \cdot 81 - 12 = 1 \cdot 81$ knots slip.

Then, as $13 \cdot 81 : 1 \cdot 81 :: 100 :: 13 \cdot 1$ per cent. of slip.

To find the percentage of slip if the distance run by the ship per day is known, and the revolutions for the same time as indicated by the counter.

EXAMPLE.—The distance gone by the ship in twenty-four hours is 360 knots, and the counter indicates 135112 revolutions for the same period ; find the per cent. of slip if the propeller pitch is 18 feet.

$$\frac{135112 \times 18}{6080} = 400 \text{ knots by engine.}$$

$400 - 360 = 40$ knots slip.

Then, as $400 : 40 :: 100 :: 10$ per cent. slip.

Indicated Horse-Power and Consumption.

After finding the mean pressure from a pair of diagrams, the I.H.P. is obtained as follows :—

$$\frac{D^2 \times \cdot 7854 \times S \times 2 \times R \times M.P.}{33000} = \text{I.H.P.}$$

D = diameter of piston in inches.
S = stroke in feet.
M.P. = mean pressure.
R = revolutions per minute.
2 = two strokes per revolution.
33000 = foot-lbs. per I.H.P. per minute.

The I.H.P. of each cylinder must be worked out separately, and the results added together to obtain the total I.H.P. of the engine.

To find the consumption of coal per I.H.P. per hour divide the coal used per day in lbs. by the I.H.P. and twenty-four hours.

EXAMPLE.—The total I.H.P. of the three cylinders amounts to 1,100, and the consumption per day is 18 tons ; find the pounds of coal burnt per I.H.P. per hour.

$$\frac{18 \times 2240}{1100 \times 24} = 1 \cdot 52 \text{ lbs. of coal per I.H.P. per hour.}$$

EXAMPLE.—The H.P. cylinder of a compound engine develops 420 I.H.P., and the L.P. 460 I.H.P.; if the consumption is 20 tons per day, find the coal used per I.H.P. per hour.

$$420+460=880 \text{ total I.H.P.}$$

Then, $$\frac{20\times 2240}{880\times 24}=2\cdot 12 \text{ lbs. of coal per I.H.P. per hour.}$$

The cylinders of a triple expansion engine are 24, 40, and 65 inches in diameter, stroke 3 feet 6 inches, and revolutions 70 per minute. The mean pressures obtained from the indicator cards are—H.P. 55·6 lbs., M.P. 24·8 lbs., and L.P. 8·9 lbs. The consumption per day is 22·5 tons.

Find (1) the I.H.P. of each engine; (2) the total I.H.P. developed; and (3) the consumption per I.H.P. per hour.

$$\frac{24^2\times \cdot 7854\times 3\cdot 5\times 2\times 70\times 55\cdot 6}{33000}=384\cdot 1 \text{ I.H.P. in H.P. cylinder.}$$

$$\frac{40^2\times \cdot 7854\times 3\cdot 5\times 2\times 70\times 24\cdot 8}{33000}=475\cdot 9 \text{ I.H.P. in M.P. cylinder.}$$

$$\frac{65^2\times \cdot 7854\times 3\cdot 5\times 2\times 70\times 8\cdot 9}{33000}=451 \text{ I.H.P. in L.P. cylinder.}$$

Then, $$384\cdot 1+475\cdot 9+451=1{,}311 \text{ total I.H.P. developed.}$$

and, $$\frac{\overset{\text{Tons.}}{22\cdot 5}\times 2240}{1311\times 24}=1\cdot 6 \text{ lbs. of coal per I.H.P. per hour.}$$

NOTE.—For average cases the consumption should be somewhere between 1·3 and 1·6 lbs. per hour per I.H.P.

Coal Consumption, I.H.P., Speed, and Distance Run.

In comparing coal consumption, speed, and distance run, it should be remembered that within moderate speeds the coal consumption (or I.H.P.) varies as the speed^2 × distance, which means that the coal burnt per knot varies as the speed^2. This amounts to the same thing as stating that the consumption varies as the cube of the speed. Therefore we have the following laws which apply to moderate speeds for any given steamer :—

1. The consumption (coal or water) (or I.H.P.) varies as the speed^3.
2. The consumption per knot varies as the speed^2.
3. The consumption over a voyage varies as the speed^2 × distance.

NOTE.—Above a certain speed limit the I.H.P. may vary as the 4th, 5th, or 6th power of the speed. From the foregoing it will be seen that if a steamer runs short of coal, port may be reached if the speed is reduced, for although the time taken is much longer, this is more than balanced by the reduced daily consumption which, under the reduced speed conditions, may be found sufficient to last the voyage.

EXAMPLES.—

(1.) The consumption per day is 14 tons and the speed 11 knots; find the consumption if the speed is increased to 12 knots.

$$\text{As } 11^3 : 12^3 :: 14 = 18{\cdot}17 \text{ tons per day.}$$

From this it will be seen that to increase the speed by 1 knot per hour the consumption increases 4·17 tons per day.

(2.) A twin screw steamer develops 2000 I.H.P. in each set of engines, or 4000 I.H.P. in all, and runs at a speed of 14 knots; find the speed when running with one set of engines only, and developing 2000 I.H.P.

$$\text{As } \sqrt[3]{4000 : 2000 :: 14^3} :: 11 \text{ knots,}$$

so that with 4000 I.H.P. the speed will be 14 knots, and with 2000 I.H.P. 11 knots.

(3.) The speed is 14 knots and the coal consumption 40 tons per day; find the reduced consumption if the H.P. is linked in to give a speed of only 12 knots.

$$\text{As } 14^3 : 12^3 :: \overset{\text{Tons}}{40} = 25{\cdot}2 \text{ tons (nearly).}$$

Therefore the consumption falls from 40 tons to 25·2 tons, a difference of 14·8 tons per day.

(4.) A steamer after having run 1000 nautical miles at a speed of 10 knots and consumed 72 tons of coal, has yet to run 1200 miles, but the coal has run short, as the bunkers only contain 65 tons; find the economical speed required so that port may be made.

Then, $$65 \times 1000 \times 10^2 = 72 \times 1200 \times K^2.$$

Therefore, $$\frac{65 \times 1000 \times 10^2}{72 \times 1200} = K^2 = 75{\cdot}23,$$

and, $$\sqrt{75{\cdot}23} = 8{\cdot}7 \text{ knots (nearly).}$$

Therefore by reducing the speed from 10 knots to 8·7 knots (say $8\frac{1}{2}$ knots), the coal will last the voyage, although a longer time is taken to complete the distance.

(5.) Distance run, 2880 miles; speed, 12 knots; coal consumed, 300 tons; time taken, ten days; find the required speed to make port, and the time taken, distance yet to go 1500 miles, and coal supply only 100 tons.

Then $$100 \times 2880 \times 12^2 = 300 \times 1500 \times K^2.$$

Therefore, $$\frac{100 \times 2880 \times 12^2}{300 \times 1500} = K^2 = 92{\cdot}16,$$

and, $$\sqrt{92{\cdot}16} = 9{\cdot}6 \text{ Knots.}$$

$$\text{Time taken} = 1500 \div 9{\cdot}6 \times 24 = 6{\cdot}5 \text{ Days.}$$

Therefore the coal will last 6·5 days at a reduced speed of 9·6 knots, the steamer covering a distance of 1500 miles, as $1500 \div 9{\cdot}6 \times 24 = 6{\cdot}5$ days, but would only last 3·33 days at 12 knots, and cover a distance of only 959 miles, as $3{\cdot}33 \times 12 \times 24 = 959$.

The proof can be shown as follows:—

300 Tons ÷ 10 Days = 30 Tons per day at 12 Knots.

Then, As $12^3 : 9{\cdot}6^3 :: 30 : 15{\cdot}36$ Tons at 9·6 Knots,

so that $30 \times 3{\cdot}33 = 99{\cdot}9$ Tons,

and, $15{\cdot}36 \times 6{\cdot}5 = 99{\cdot}8$ Tons.

H.P. Cut-off and Consumption.

The consumption (either coal or steam), and therefore the Horse-Power developed, vary as the cube of the speed (at moderate speeds). As the H.P. valve cut-off is the approximate measure or rate of the steam consumption, and therefore the coal consumption, then the variation in cut-off required for a given speed may be approximated as follows:—

Example.—Speed, 12 knots, with H.P. cut-off at ·6; find the H.P. cut-off required to reduce the speed to 11 knots.

Then, As $12^3 : 11^3 :: {\cdot}6 : {\cdot}46$ cut-off. **Answer.**

Therefore the H.P. link must be run in to cut-off at ·46 (or 46 per cent.) to reduce the speed from 12 knots to 11 knots.

NOTE.—In actual practice the consumption is found to vary more often as the 4th power of the speed in place of the cube of the speed.

Efficiency.

The general average efficiency of the boilers, engines, working parts, and propellers are as follows:—

No. 1.—Boiler Efficiency $= \frac{10}{15}$ or ·66, or 66 per cent.

NOTE.—1 lb. good coal contains about 14500 heat units, and to change 1 lb. water into steam, if the steam temperature is 212°, and the feed water temperature 212° (see p. 622), requires 966 heat units.

Therefore, Theoretical evaporation $= 14500 \div 966 = 15$ lbs. water.

In practice, however, 1 lb. evaporates only 9 or 10 lbs. water.

Therefore, Boiler efficiency $= \frac{10}{15}$.

No. 2.—Steam Efficiency.—The heat efficiency of the steam acting on the piston to develop horse-power varies from 10 to 15 per cent. in actual practice.

NOTE.—Maximum Theoretical Efficiency of an engine $= \dfrac{T^\circ - t^\circ}{T^\circ + 461^\circ}$.

Where, T° = Steam temperature.
t° = Exhaust ,,
461° = Absolute temperature Constant.

Example.—The initial pressure is 180 lbs., and temperature 380° Fahr., the exhaust temperature is 200° Fahr.; express the maximum heat efficiency of the engine.

Then, Efficiency $= \dfrac{380 - 200}{380 + 461} = {\cdot}21$, or 21 per cent.

In practice only about 56 per cent. of this efficiency can be obtained.

Therefore, Actual efficiency $= \dfrac{21 \times 56}{100} = 11{\cdot}7$ per cent. efficiency.

Combined Boiler and Engine Efficiency.—Suppose that 1·6 lbs. of coal are burnt per I.H.P. per hour.

Then, Combined efficiency $= \dfrac{60 \times 33000}{1{\cdot}6 \times 14500 \times 778} = {\cdot}116$, or 11·6 per cent.

NOTE 1.—Heat Value per lb. coal = 14500 B.T.U.
,, 2.—Mechanical Value of each B.T.U. = 778 foot-lbs. of work.
,, 3.—60 minutes per hour.

This result, it should be noted, corresponds with the last.

Mechanical Efficiency.—The power lost in driving the air pump, feed and bilge pumps, together with that required to overcome the friction and weight of the moving parts, amounts to about 10 or 12 per cent. of the total power.

Therefore, Mechanical efficiency $= 100 - 10 = 90$ per cent.

Combined efficiency of Boilers, Engines, Shafting, &c. $= \dfrac{11{\cdot}6 \times {\cdot}90}{100} = 10{\cdot}44$ per cent.

Propeller Efficiency.—The actual power utilised in driving the ship through the water is only about 60 or 65 per cent. of that delivered to the propeller, as blade friction, slip, useless effort of rotation, &c., waste the remainder of the power.

Therefore, Propulsive efficiency $= {\cdot}60$, or 60 per cent.

Combined Efficiency of Boilers, Engines, Shafting, and Propeller.

$$\text{Total efficiency} = \frac{10{\cdot}44 \times 60}{100} = 6{\cdot}264 \text{ per cent.}$$

Therefore, of 100 per cent. coal, or power supplied at the boiler end of the plant, only 6·26 per cent. of this is utilised by the propeller to drive the steamer, or the combined total loss is equal to $100 - 6{\cdot}26 = 93{\cdot}74$ per cent.

Efficiency of High-Pressure Steam and Low-Pressure Steam Compared.

$$\text{Heat Efficiency} = \frac{T^\circ - t^\circ}{T + 461} \text{ (see p. 398).}$$

Where, T° = temperature of steam.
,, t° = ,, ,, exhaust.
,, 461 = absolute temperature constant.

Take steam of, say, 80 lbs. gauge pressure, or 95 lbs. absolute; also steam of, say, 180 lbs. gauge pressure, or 195 lbs. absolute, with exhaust steam temperature of 160° in each case.

Then, 95 lbs. = 324° Fahr. (from table).
And, 195 ,, = 380° ,, (,, ,,).
Exhaust temperature = 160°.

Low Pressure—

$$\text{Efficiency} = \frac{324^\circ - 160^\circ}{324 + 461} = {\cdot}209 \text{ (nearly).}$$

High Pressure—

$$\text{Efficiency} = \frac{380^\circ - 160^\circ}{380^\circ + 461} = {\cdot}261.$$

$$\text{Per cent. economy} = \frac{({\cdot}261 - {\cdot}209) \times 100}{{\cdot}209} = 24{\cdot}8 \text{ per cent.}$$

So that the theoretical economy of the higher pressure is 24·8 per cent. over that of the lower pressure. In actual practice the gain is about 15 per cent. (see p. 384).

Squared Paper Diagrams or Curves.

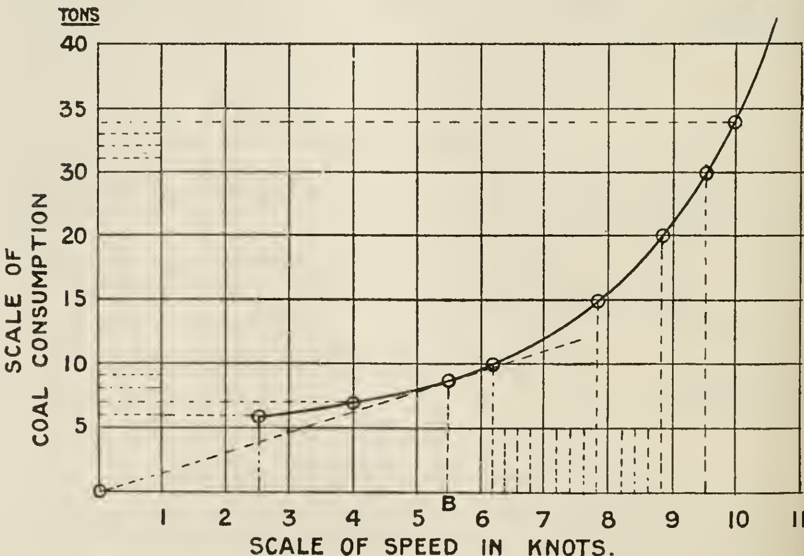
By means of paper ruled off into squares (which can be easily made by ruling off a series of vertical and horizontal lines, say $\frac{1}{2}$ inch apart), useful diagrams known as "speed power" curves can be constructed which will be found of immense service to the chief engineer of a steamer. The writer is quite aware that the majority of marine engineers do not trouble themselves much with so-called "theoretical" diagrams, but he would direct the special attention of readers to the use of the curves here referred to, which are of a strictly practical nature, and if once grasped will be found both interesting and instructive. The curves can be applied as a comparison or check on speed, consumption slip, horse-power, revolutions, &c., and a few examples are given by way of application.

Speed and Consumption Curve.

EXAMPLE I.—From observation, notes are made of the consumption of coal at various ship speeds, which are as follows:—

At 2·5	knots	the consumption	was	6	tons.
" 4	"	"	"	7	"
" 6·2	"	"	"	10	"
" 7·8	"	"	"	15	"
" 8·8	"	"	"	20	"
" 10	"	"	"	34	"

From the above plot out a curve of "speed and consumption."



No. 84.—Speed and Consumption Curve.

METHOD.—Draw out on a sheet of paper a number of, say, $\frac{1}{2}$-inch squares, and let each horizontal space represent 1 knot, and each vertical space or division 5 tons. Where required each horizontal division can be divided into five smaller divisions, each representing $\frac{2}{10}$ or ·2 of a knot, and each vertical division into five smaller divisions, each representing 1 ton. Number the spaces as shown, up to, say, 11 knots and 40 tons.

Now, at the first noted speed of 2·5 knots erect a vertical line, and draw out a horizontal line from the corresponding consumption of 6 tons to meet it; at the intersection describe a small circle or large dot. At the next observed speed of 4 knots draw out a horizontal line from the corresponding consumption of 7 tons, and again describe a small ring at the intersection. Repeat this successively for each noted speed and consumption, describing small circles at each point of intersection as shown. Notice that at the speed of 6·2 knots the vertical line is run up at the first subdivision between the spaces of 6 and 7 knots, this representing 6·2 knots; also that at 7·8 knots the vertical line is run up at the fourth small division between spaces 7 and 8, this being equal to 7·8 knots, and so on for each decimal of a knot. Last of all, draw either by hand or wooden curves a line running through all the small rings so found, and the result will be a speed consumption curve which shows at a glance the relation existing between speed and coal consumed at various speeds.

NOTE.—The value of the spaces may be changed to any convenient measure without in any way affecting the result. That is, each vertical space may be made to represent 2 tons in place of 5 tons, or each horizontal space made to represent half a knot, or if found more suitable 2 knots, in place of 1 knot.

Economical Speed.

By this is meant the speed which will give the greatest distance run for a given coal bunker supply, or in other words the greatest distance which can be run per ton of coal burnt.

This can be closely approximated on the diagram by simply drawing a tangential line from the left-hand bottom corner to the curve as shown, and the point of contact gives the speed and consumption for greatest economy of steaming. If, then, on a voyage the coal supply runs short, this speed would be the best to adopt under the circumstances (see page 400) to make the coal last out the voyage. The economical speed is at position B, which corresponds to about 5·5 knots.

I.H.P. and Speed Curve.

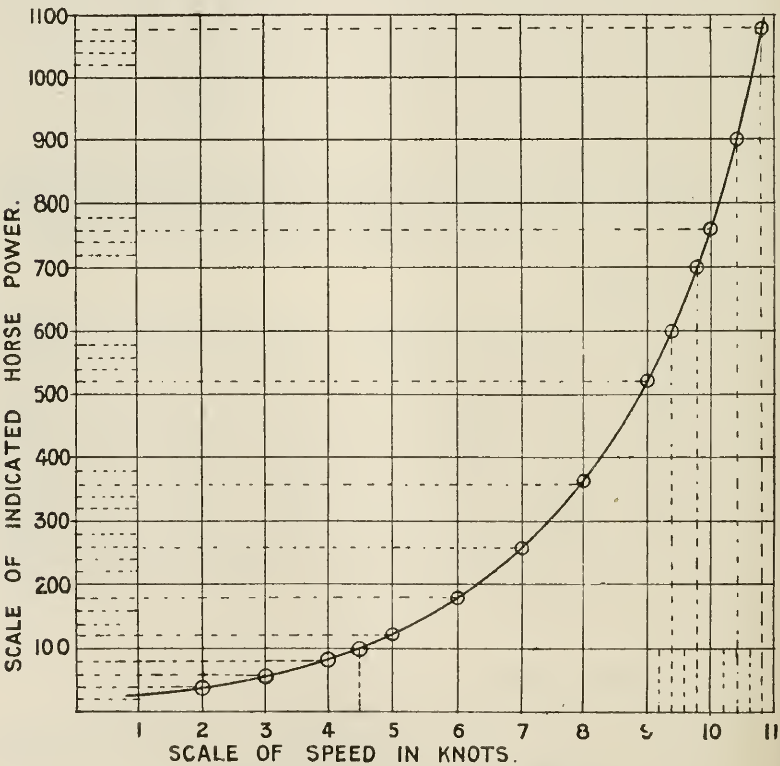
EXAMPLE 2.—This curve is constructed in a similar manner to the last, and is of very nearly the same value, as generally speaking the consumption and power are practically equivalent terms, and vary in the same relative proportion to the speed.

METHOD.—Set off horizontal spaces of ½ or ¼ inch as found most suitable, representing knots and vertical spaces of similar size, each representing, say, 100 I.H.P. Therefore if each space be divided into five smaller divisions each will be equal to 20 I.H.P.

Suppose that the following observations were made of I.H.P. at different speeds during progressive trials :—

At	2	knots the I.H.P. was		40	At	8	knots the I.H.P. was		360
,,	3	,,	,,	60	,,	9	,,	,,	520
,,	4	,,	,,	80	,,	9·4	,,	,,	600
,,	4·5	,,	,,	100	,,	9·8	,,	,,	700
,,	5	,,	,,	120	,,	10	,,	,,	760
,,	6	,,	,,	180	,,	10·4	,,	,,	900
,,	7	,,	,,	260	,,	10·8	,,	,,	1080

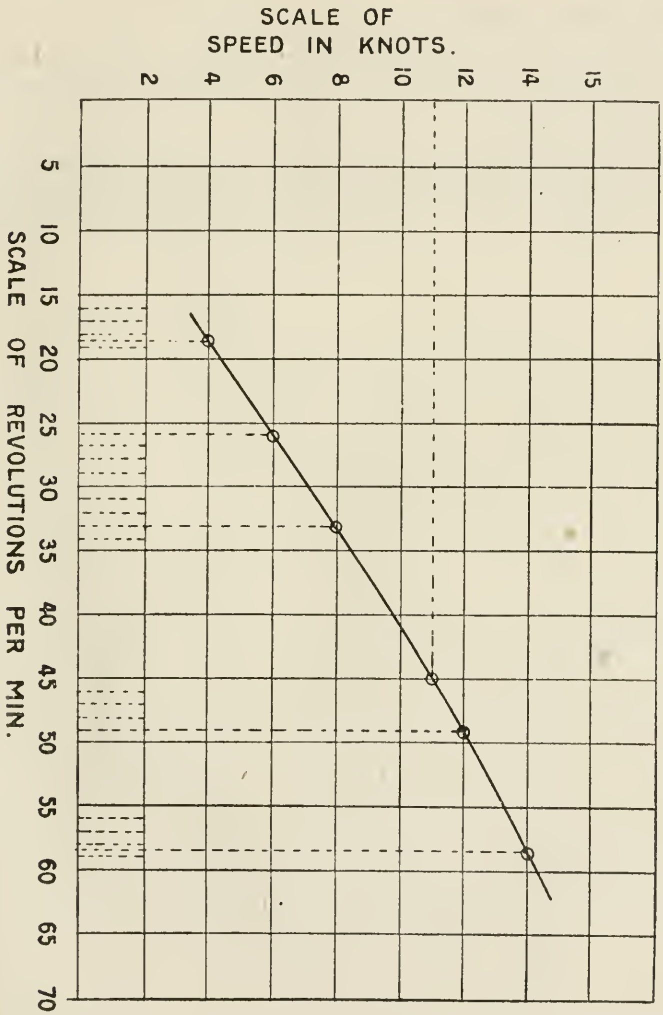
At each speed noted, where required, run up vertical lines, and connect these with horizontal lines run out from the left at the corresponding I.H.P., and at the points of intersection describe little



No. 85.—Power and Speed Curve.

circles; finally, connect these circles by a curve, which is then the "speed and I.H.P." curve required.

NOTE.—Observe that each vertical space = 100 I.H.P., therefore each fifth of each space = 20 I.H.P. Also that each horizontal space = 1 knot, therefore the fifth of each division = ·2 of a knot.



No. 86.—Speed and Revolution Curve.

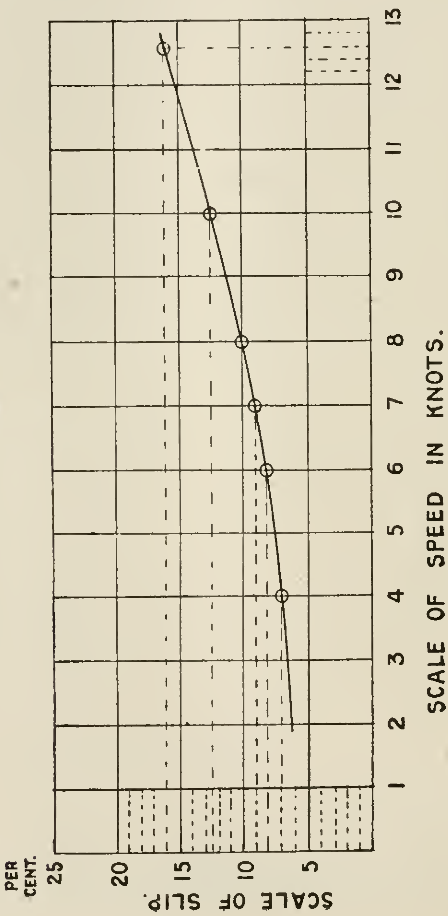
Speed and Revolution Speed.

EXAMPLE 3.—This curve is useful in showing the relation between the ship speed and engine revolutions. Set off the vertical spaces on the left as knots and the horizontal spaces as revolutions. Then at any observed or noted speed and revolutions connect the points and describe small circles as shown; repeat this for as many speeds and revolutions as may be known or noted, and draw a curve through the points so marked, which gives the speed and revolution curve.

Speed and Slip Curve.

EXAMPLE 4.—This curve shows the relation between speed and slip, which, owing to cavitation, generally shows as increase of slip with increase of speed.

METHOD.—Set off as before, horizontal spaces ($\frac{1}{4}$ inch or $\frac{1}{2}$ inch) as knots, and vertical divisions as per cent., each space representing, say, 5 per cent. of slip. Now from the log or other data connect



No. 87.

the corresponding positions of speed and per cent. slip (see page 404), and describe little circles as shown. Suppose the data to be as follows:—

At	4	knots the slip is		7	per cent.
,,	6	,,	,,	8	,,
,,	7	,,	,,	9	,,
,,	8	,,	,,	10	,,
,,	10	,,	,,	12·5	,,
,,	12·6	,,	,,	16	,,

Observe that 12·6 knots is equal to twelve spaces horizontally and three-fifths of a space or ·6, the vertical is then run up to join the horizontal line, projected over at the 16 per cent. level. As before explained, draw a line through all the points of intersection, and the result will be the "speed and slip curve."

Points to be Observed.

Example 1.—As previously explained (page 402) at intermediate speeds the consumption or power varies approximately as the cube of the speed. This can be seen in the "speed consumption curve" which shows the consumption to be 15·5 tons at 8 knots and 34 tons at 10 knots. Beyond this speed the consumption may vary as the 4th power in place of the cube, which brings out a still greater consumption ratio, but which nevertheless reduces the consumption per I.H.P. per hour, so that at low speeds the coal per I.H.P. per hour is more than at high speeds.

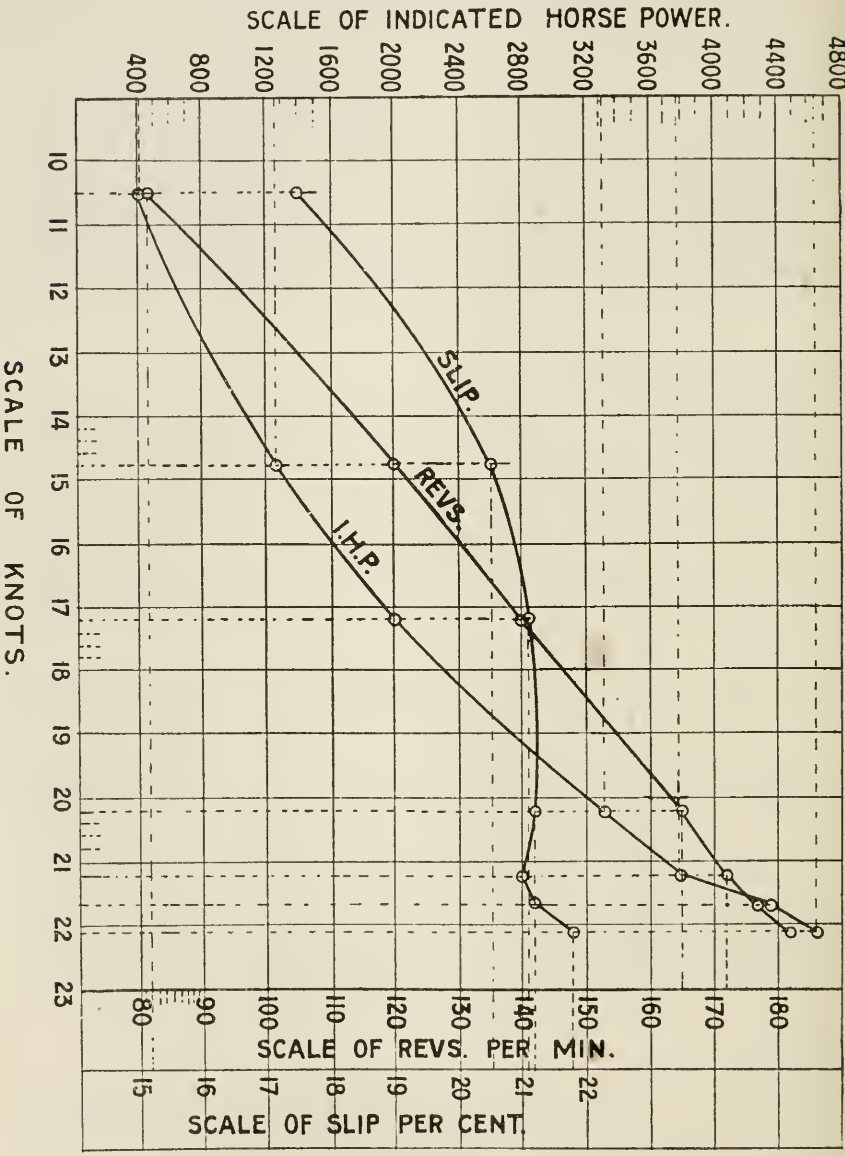
Example 2.—This curve is very similar to the last, and in some cases is practically equivalent, but generally it is found that the power varies as the cube or as the 4th power of the speed.

Example 3.—Roughly speaking the revolutions vary directly as the speed, and this is borne out by the curve shown, where it will be seen that at nearly 6 knots the revolutions are twenty-five per minute, and at 12 knots nearly fifty revolutions per minute.

Example 4.—An ordinary reciprocating engine propeller develops greatest efficiency at low revolution speed, therefore as the revolutions increase the efficiency falls off, and the slip ratio increases; this is shown by the curve, in which the slip ranges from about 6 per cent. at 6 knots to 16 per cent. at 12·5 knots.

Combined Curves.

In modern trial trip practice it is usually found more convenient to combine the various curves previously described instead of having them drawn out separately, as the various results can then be compared simultaneously, and a more comprehensive idea obtained of the general efficiency. The following is a combined curve diagram of a modern high speed passenger steamer, and the I.H.P., Speed, Slip, and Revolution curves are all shown together, thus allowing of a general comparison to be made.



No. 88.—Combined Curves of Power, Speed, Slip, and Revolutions.

In the above diagram of ship performance the spaces representing knots are divided up into tenths or fifths where required to allow of projecting the necessary points on to the I.H.P., revolutions, or slip curves. In the same way the I.H.P. divisions are divided up into four parts, each being equal to 100 I.H.P.; the slip spaces are also divided into tenths for decimals of per cent., and the revolution spaces are divided into ten divisions, each being equal to a revolution. Dotted lines are shown to illustrate as clearly as possible how the small rings which form the connecting points of the curves are located.

The three curves shown on the diagram were developed from the following trial trip data, and it should be noted that the *mean* results of a series are taken in each case.

Data for Curves.

Mean Speed in Knots.	Mean I.H.P.	Mean Slip per Cent.	Mean Revs. per Minute.
10·5	400	17·5	82
14·8	1270	20·6	120
17·2	2000	21·1	140
20·2	3325	21·2	165
21·2	3790	21	172
21·7	4393	21·2	177
22·1	4650	21·8	182

With reference to the curves, the following points should be carefully studied:—

1. Between 11 knots to 21 knots the I.H.P. varies approximately as the knots cubed, which can be proved as follows: At 11 knots the I.H.P. is 500; to find the I.H.P. at 16 knots,

Then, As $11^3 : 16^3 :: 500 = 1600$ I.H.P. (nearly),

which is exactly the figure shown on the curve at this speed. After this speed the ratio is still higher.

2. The revolutions vary almost directly as the speed, for at 10·5 knots the revolutions are 82, and at double that speed or 21 knots the revolutions are 170.

3. The propeller slip is high owing to the high engine revolution speed reducing the propeller efficiency, but at 21·2 knots the slip is less than at 19 knots, which is somewhat unusual. From 21·2 knots to the maximum speed of 22·1 knots, the slip increases rapidly, as shown by the upward tendency of the slip curve at this position.

SECTION VI.

MARINE ENGINEERING CHEMISTRY NOTES.

IT need hardly be pointed out that a slight knowledge of chemistry, or at least of chemical processes, is a necessity for the modern engineer, surrounded as he is by numerous active examples of chemical action and reaction, which have, in many cases, to be resisted or neutralised by means of other chemical actions or reactions. Unfortunately, however, many of the effects referred to, instead of being eliminated altogether, can only be minimised.

Examples.—The following are a few ordinary examples of chemical action and reaction :—

1. Combustion of coal or oil in a furnace.
2. Rusting (combustion).
3. Explosion of gases (combustion).
4. Corrosion in boilers, Feed Heaters, &c.
5. Corrosion in tank tops, condenser tubes, tail-end shafts, rudder posts, propellers, &c.
6. Scale deposits in boilers, Evaporators, &c.
7. Formation of Marsh Gas in bunkers, oil tanks.
8. Formation of CO_2 gas and Free Nitrogen in ballast tanks, &c.

From the foregoing it will be admitted that, to cope more or less successfully with the destructive effects produced by the processes referred to, a knowledge of chemistry is really necessary, as by it the engineer may know how to obtain good combustion in the furnaces, how to correct the action of acid or oxygen in boiler water, the best methods of preventing explosion of Marsh Gas in bunkers, methods of protecting the boilers against corrosion and pitting, how to keep down scale deposit in boilers, &c. &c., all of which come under the general heading of chemistry.

Composition of Coal.

Average Coal	Carbon	80	per cent.
	Hydrogen	5	,,
	Oxygen	8	,,
	Nitrogen	$1\frac{1}{2}$	,,
	Sulphur	$1\frac{1}{2}$	,,
	Ash, &c.	4	,,
		100	per cent.

Heat Values.

Carbon	= 14500	Heat Units per lb.
Hydrogen	= 62000	,, ,,
Sulphur	= 42	,, ,,

Heat in 1 lb. Coal.

1 lb. good coal contains about 14700 Heat Units, made up as follows :—

$$\frac{14500 \times 80}{100} = 11600 \text{ Heat Units from Carbon.}$$

$$\frac{62000 \times 5}{100} = \underline{3100} \text{ Heat Units from Hydrogen.}$$

$$14700 \text{ Heat Units, total.}$$

NOTE.—The heat of the other elements may be neglected.

Chemistry of Gases, &c.

In chemical formulæ the various elements are represented by the following symbols :—

C = Carbon.	**Cl** = Chlorine.
H = Hydrogen.	**Na** = Sodium.
N = Nitrogen.	**Fe** = Iron.
O = Oxygen.	**Ca** = Calcium
S = Sulphur.	

Small numbers affixed to any of the above symbols indicate the atoms or volumes of that element which go to make up the chemical compound expressed. For example, water is composed of two atoms of hydrogen and one atom of oxygen ; it is therefore expressed chemically as H_2O. Again, dry Ammonia is composed of one atom of nitrogen and three atoms of Hydrogen, and is expressed as NH_3. The prefix "Mon" means one atom, and "Di" two atoms.

The following are the most important chemical compounds to be studied and committed to memory :—

Atmospheric Air.

Composition.

	By Volume.	By Weight.
Nitrogen - - - - -	79·04	77
Oxygen - - - - -	20·96	23

Ordinary atmospheric air contains water vapour and a very small percentage of carbonic acid gas—about ·04 per cent. It also contains small proportions of Ammonia, Argon, also Aqueous Vapour and Nitric Acid.

One pound of air at ordinary atmospheric temperatures occupies about 13 cubic feet, and consists of Oxygen 23 parts, Nitrogen 77 parts by weight.

1 lb. air = $\begin{cases} \text{Oxygen } \cdot 23 \text{ of 1 lb.} \\ \text{Nitrogen } \cdot 77 \text{ of 1 lb. (nearly).} \end{cases}$

NOTE.—Atmospheric air also contains a very small proportion of CO_2 gas—about ·04 per cent.

Air Required per Pound Coal.—(1.) Assuming that each pound of coal requires in forced draught 20 lbs. of air for perfect combustion,

Then, $20 \times 13 = 260$ cubic feet of air per pound coal.

(2.) If natural draught, 24 lbs. are required,

Then, $24 \times 13 = 312$ cubic feet of air per pound coal.

Water = H_2O.

Carbon Monoxide, or **CO = Carbonic Oxide.**—This gas is obtained by incomplete combustion, due to an insufficient supply of air or oxygen. CO will change into CO_2 if sufficient oxygen be supplied to it. An example of this is flame sometimes seen at the funnel top. CO burns with a pale bluish flame. The specific gravity of CO = ·96.

Carbon Dioxide, or **CO_2 = Carbonic Acid Gas.**—This gas is obtained (one way) by perfect combustion in the furnace. It supports neither combustion nor animal life. It is used in the form of "*carbonic anhydride*" (that is, "without water") for refrigerating machines, as when reduced to a liquid under pressure and cold, it quickly evaporates again at a low temperature if the pressure is withdrawn, and expansion allowed to take place.

It can be prepared in large quantities by the action of Hydrochloric acid on Limestone.

CO_2 (together with Free Nitrogen) is also found in empty ballast tanks and boilers under the name of "foul air," and the CO_2 gas being fully one and a half times heavier than the atmosphere, accumulates at the lowest parts of the tanks, &c.

It has been recently found by careful experimental tests that "foul air" is made up of fully 85 per cent. Free Nitrogen, and only about 15 per cent. CO_2, instead of being, as was at one time supposed, entirely made up of CO_2 gas.

The presence of CO_2 and Free Nitrogen can be detected by lowering a lighted taper (or open lamp) into the suspected place, which, if extinguished, denotes these gases present in quantity. The atmosphere contains about 0·4 per cent. of CO_2, and this is understood to be one of the causes of corrosion in boilers, as the air admitted with the feed water contains Oxygen and CO_2, both of which in combination are conducive to corrosion when set free by heat in a moist atmosphere.

In testing for the presence of "foul air," a light, if lowered into the tank or boiler to be tested, will either burn smoky and black or go out altogether, according to the percentage of "foul air" present. If the light is extinguished, CO_2 is present in dangerous quantity.

Free Nitrogen.—When oxidation goes on in a ballast tank or boiler, Free Nitrogen is produced by the rapid combination of the Oxygen, which is therefore used up, and leaves behind the Nitrogen gas, Nitrogen liberated in this manner is similar in its effects on life and combustion to CO_2, in that it neither supports life nor combustion, and thus constitutes "foul air."

Light Carburetted Hydrogen, Methane, or **CH_4 = Marsh Gas.**—This gas is usually found in coal bunkers, and is generated by the gradual evaporation of the coal and its absorption of oxygen from the air especially if damp. As the gas slowly forms by chemical action, heat is generated, and the temperature rises, it may be, to the ignition point, thus causing what is known as spontaneous combustion. It should be noted that this gas will explode on the introduction of a naked light into the bunker. Small or wet coal is most liable to produce spontaneous combustion. Danger of explosion from a light exists when the proportion of CH_4 to air is in the ratio of 1 to 10; that is, 1 cubic foot of CH_4 to 10 cubic feet of air. Coal shipped direct from the mine contains more CH_4 than coal which has lain for some time previous to shipping, as in the latter case most of the Marsh Gas has passed off. Air containing 5 to 6 per cent. CH_4 will explode sharply, and when 10 per cent. is present it explodes with greatest violence, complete combustion taking place. When CH_4 is suspected in a coal bunker, the only sure way of detecting it is by means of a miner's Davy lamp, its presence being shown by a pale blue "cap" or halo on the top of the flame. When an explosion of CH_4 takes place in a bunker great caution should be exercised in entering it as the resultant gases (CO and CO_2) are highly poisonous. CH_4 itself in a pure state is also poisonous.

Marsh Gas (specific gravity = ·55) is lighter than the atmosphere, and therefore occupies the highest parts of bunkers and tanks. Marsh Gas supports neither life nor combustion.

Petroleum Vapour, or **Light Carburetted Hydrogen,** is also found in empty oil tanks, and is formed by the evaporation of the layer or skin of oil left adhering to the bottom and sides of tanks after they have been pumped out. Oil gas is of nearly the same composition as coal gas, with the difference that it contains less carbon and more hydrogen. This will be understood when it is stated that average mineral oil as used in furnaces for fuel is made up as follows:—Carbon, 84 per cent.; Hydrogen, 14·5 per cent.; and Oxygen, 1·5 per cent. Carburetted Hydrogen, if mixed with air, is highly explosive; it is also fatal to life. For this reason oil tank steamers are fitted with special fans for exhausting the foul gases from the tanks after the oil is pumped out. Petroleum vapour from heavy mineral oils is much heavier than the atmosphere (two and a half times), and occupies the lowest level of a tank.

Ammonia = NH_3.—This alkali (the opposite of an acid) is peculiar in the fact of its having what is usually the property of an acid, that is, a corrosive action on brass and copper. Ammonia being an alkaline substance, changes red litmus paper to blue; whereas an acid changes blue litmus paper to red. Litmus paper is the ordinary chemical test for acid or alkali.

NOTE.—Acids and alkalies combine to form salts, and in combining the one counteracts or destroys the effects of the other.

Ferric Oxide = Fe_2O_3.—This is obtained by the chemical combination of iron and oxygen in a damp atmosphere. When two atoms of iron combine with three of oxygen we obtain five atoms forming Fe_2O_3.

Hydrochloric Acid = HCl.—This is composed of hydrogen and chlorine, and is often found in empty boilers. The chlorine is obtained from Magnesium Chloride, a constituent of sea water.

Sodium Chloride, or **Common Salt** = $NaCl_2$.

NOTE.—Na is the chemical symbol for Sodium (Natrium).

Calcium Chloride = $CaCl_2$.—This compound is used as a brine former in refrigerating machines, where the brine pipe system is used for cooling. This salt is very much more intense in its effect, and is better in every way than common salt.

In making up the brine with calcium chloride *fresh water* only should be used, as sea water sets up corrosion in the brine pipes and brine pump. Common salt has the same injurious effect.

Acids.—An acid is a chemical compound possessing the following characteristic properties:—

(1.) Is sour to the taste.
(2.) Changes blue litmus paper to red.
(3.) Neutralises alkalies, and with them forms chemical salts.
(4.) Contains hydrogen.

Alkalies.—An alkali is a chemical compound possessing the following characteristic properties:—

(1.) Is soapy to the taste.
(2.) Changes *red* litmus paper to blue.
(3.) Absorbs CO_2 gas.
(4.) Easily combines with acids to neutralise them and form chemical salts (is then known as a "base").

Spontaneous Combustion in Coal Bunkers.

This is likely to occur in bunkers exposed to the effects of fairly high temperatures, such as may exist when the bunkers are placed very near the boilers of a steamer.

By the oxidation of coal the carbon is set free, and combines with the oxygen of the air and forms CO_2 and CO.

The gas produced by the gradual oxidation of the coal is CH_4 or Marsh Gas, and it should be noted that this gas of itself is not explosive, but only when mixed with certain proportions of atmospheric air. A lighted taper plunged into a jar of pure Marsh Gas will not produce an explosion, but if the Marsh Gas were mixed with, say, 10 cubic feet of air to 1 cubic foot of gas, a violent explosion would result.

If the air supply is reduced to one-half of the above proportion, the resulting mixture will not produce an explosion, or if the air supply is increased to *twice* the above proportion, the resulting mixture will not be explosive. The most violent explosion occurs when the proportion of air to CH_4 is as 10 is to 1, a weaker explosion, however, taking place when the proportion is as 8 is to 1.

Marsh Gas can be detected by means of a safety lamp similar to that used by miners, for if the flame of the lamp is turned down low a blue "cap" or top will form and burn above the flame, thus indicating the presence of Marsh Gas.

Marsh Gas can only be got rid of by means of ample ventilation of bunkers or tanks, or, in the case of the latter, by means of special exhausting fans, as employed in oil-carrying steamers.

NOTE.—The gases obtained after explosion of Marsh Gas (CH_4) and air are:—CO_2, H_2O, and N.

Or, Carbonic Acid Gas, Steam, and Free Nitrogen.

Marsh Gas (CH_4).

The following points regarding the explosive properties of Marsh Gas when mixed with air are of importance:—

1. Ignition point, 1200° Fahr.

2. If atmosphere contains $5\frac{1}{2}$ per cent. of Marsh Gas ignition only will take place if a light such as a match, candle, lamp, or fat electric spark is applied.

3. If atmosphere contains 10 per cent. of Marsh Gas a violent explosion will take place if a light is applied.

4. If atmosphere contains 18 per cent. of Marsh Gas ignition only will take place if a light is applied; and if the percentage of Marsh Gas is more than this the combustible nature of the mixture decreases.

Briefly, if the oxygen (or air) supply is low ($5\frac{1}{2}$ per cent.) the mixture is not favourable to combustion, and if the oxygen (or air) supply is high or excessive (20 per cent.), the same holds good; the most suitable mixture for explosion being 10 per cent. of Marsh Gas in the atmosphere, as with this proportion the chemical combination is most favourable for instantaneous combustion.

Causes of Spontaneous Combustion.—Fires in coal bunkers or in coal cargoes are due to the rapid absorption of oxygen by the coal, and the most favourable conditions for this occurrence are as follows:—

1. With freshly worked coal.
2. With small coal, slack, or dross.
3. With moist coal.
4. By heat in surrounding atmosphere.

The first signs of spontaneous combustion in a coal cargo are a peculiar smell like burning oil, and the appearance of a mist-like vapour in the air currents, also a rise in temperature of the surrounding atmosphere. Inferior coal is more likely to ignite spontaneously than good coal; and coal of small size than lump or thick coal. With small coal there are more spaces containing air, as for a given cubic mass the weight will be less than for large coal. The oxygen of the contained air is greedily absorbed by the coal under the conditions mentioned, and this results in a rise of temperature, as whenever chemical combination takes place heat is developed. The heat may in time rise to the ignition point of Marsh Gas (CH_4), which is 1200° Fahr., and fire will then break out. This firing of coal generally originates in the centre or heart of a pile of coal and well down in the mass, as on the surface the heat generated is carried off by the ventilating atmosphere, and the temperature is thus kept down below ignition point.

Prevention.—With coal cargoes, as usually stowed loose in the holds, it is practically impossible to adopt really effective preventive measures against the generation of Marsh Gas and danger of spontaneous combustion, but the following suggestions may be of use—

1. Liberal ventilation.
2. In handling coal ashore, the coal is divided up into sectional lots, with divisions between.
3. Great care as regards ventilation is necessary with small coal for the reasons given previously.
4. Coal to be kept as cool as possible.

Treatment of Fires.—When a coal cargo or bunker takes fire the following methods of dealing with the outbreak may be employed—

1. For small fires the use of sand thrown on the flame will be found sufficient for the purpose.
2. For more serious outbreaks "digging out" may in some cases be successfully resorted to; that is, the mass of coal which is burning is (if possible, and safe to do so) dug out altogether.

"**Sealing Off.**"—In other cases of fire it is advisable to close down or "seal off" the hold altogether, so that the fire may be starved of oxygen. This method is only effective, however, if the closing down is really effective, and all ingress of air actually prevented. If leaks of air, however small, take place, the fire will be intensified instead of diminished.

Water.—Water played on the flame may in some cases be effective if the supply is ample and directed on the coal actually burning, but if otherwise, the water applied may generate "water gas" and only make the state of matters worse.

CO_2.—This gas may be effectively employed to put out a fire if applied in the initial stages, but if the coal once becomes incandescent the use of CO_2 as an extinguisher may be totally ineffective.

Carbon.—In the solid state carbon exists as charcoal, coke, soot, graphite, and the diamond, the latter being carbon in the hard and purest crystalline form.

Solid carbon may be obtained by heating coal without allowing actual combustion to take place; this expels the volatile gases, and carbon in the form of coke is left as the residue. The average heat value is 14500 B.T. units per pound.

Nitrogen.—This gas does not assist combustion, as it is a quite inert gas, and is termed a "non-supporter" of combustion. It reduces the activity of the oxygen, and thus moderates oxidation and combustion. During combustion it passes off unchanged in condition, but raised in temperature.

Hydrogen.—This gas is the lightest substance known. It exists in water as H_2O. It is also combined with Carbon, Nitrogen, and Oxygen in coal, and other substances. All acids contain Hydrogen.

When Hydrogen burns in air or Oxygen, water is produced, the water consisting of two volumes of Hydrogen and one volume of Oxygen, or H_2O. The action takes place in the furnaces, and water vapour or steam is evolved by the liberation of Hydrogen from the coal, and the combination of the same with atmospheric Oxygen.

Combustion.—Allowing 24 lbs. of air per pound of coal—

Then,	$\cdot 77 \times 24 = 18\cdot 48$	lbs. Nitrogen,
And,	$\cdot 23 \times 24 = 5\cdot 52$	lbs. Oxygen.
	$24\cdot 00$	lbs. Air.

From this it will be seen that for each pound of coal 18·48 lbs. of Nitrogen require to be heated up from say 62 degrees to 650 degrees (Funnel Temperature), or 588 degrees. This illustrates the unavoidable waste of heat due to the heating of the Nitrogen of the air.

Coal Gases.—Moist or damp coal in the bunkers generates the following gases by steady absorption of Oxygen—

$$(1.)\left\{\begin{array}{l}\text{Light Carburetted Hydrogen}\\ \text{Marsh Gas}\\ \text{Methane}\end{array}\right\} = CH_4.$$

The gas CH_4 (which, it will be observed, has three different names) is colourless, tasteless, and inodorous. It burns with a yellow coloured flame, and can be detected by the use of a safety Davy type lamp, which burns with a "blue cap" at the top of the flame if this gas is present.

(2.) CO gas, or Carbonic Oxide.

This gas is also colourless, tasteless, and inodorous. It is poisonous and burns with a bluish coloured flame. The Davy test lamp has the flame surrounded by a mantle of gauze, the cooling effect of which is to reduce the temperature of the flame below the ignition point of Marsh gas. In testing for the presence of explosive gases the lamp requires to be turned down to a mere peep, and if Marsh gas is present the blue cap will then show at the top. CO can only be detected by a flame when 12 per cent. is present, whereas $\frac{1}{2}$ per cent. is dangerous to life. The only sure test is by means of a small, warm blooded animal, such as a mouse or canary, which are now used in coal mines when this gas is suspected.

General Notes.

Combustion is the chemical combination of the Carbon and Hydrogen of the coal with the Oxygen gas of the air, producing heat.

Weight for weight with Carbon, Hydrogen gives out the most heat, as 1 lb. of Hydrogen contains about 62,000 units of heat, and 1 lb. of Carbon about 14,500 units of heat, but in actual combustion the amount of Hydrogen in coal is so small (about 5 per cent.) that its heating power may be neglected altogether, and all the effective heat assumed to come from the Carbon, which constitutes four-fifths of the coal.

If twelve parts of Carbon (by weight) combine with thirty-two parts of Oxygen (by weight), Carbonic Acid gas, CO_2, is obtained, giving complete combustion.

If twelve parts of Carbon combine with less that thirty-two parts of Oxygen, Carbonic Oxide gas, CO, is obtained, giving incomplete ombustion, with a corresponding loss of heat.

The funnel gases consist chiefly of Carbonic Acid gas, Carbonic Oxide gas, Oxygen and Nitrogen, or of CO_2, CO, O, and Free N.

NOTE.—Under conditions of good combustion the funnel gases are made up as follows:—

Nitrogen = 80 per cent. CO_2 = 10 per cent.

(CO, Oxygen, and H_2O) = 10 per cent.

If CO is present in waste or funnel gases, combustion of the fuel is incomplete and a loss of heat is taking place.

Black smoke is chiefly composed of uncombined particles of Carbon.

In combustion the Sulphur of the coal combines with Oxygen and produces SO_2 (Sulphur Dioxide), or more correctly Sulphurous Anhydride. This is a colourless gas, possessing a strong, pungent smell.

Black smoke may be caused by excess of air as well as by want of air, as excess of air lowers the temperature of combustion and prevents the formation of CO_2. The surplus air absorbs heat in passing through the furnaces.

If coal is placed in a closed chamber, technically known as a retort, and subjected to destructive distillation, gases are given off, leaving behind carbon in the form of coke.

Therefore,—

1. **In complete combustion** the Carbon of the coal combines chemically with Oxygen of the air to produce CO_2 gas.

2. **In incomplete combustion** part of the Carbon of the coal combines chemically with the Oxygen of the air to produce CO gas, and part combines mechanically with the air to produce black smoke.

3. In combustion the Hydrogen of the coal combines with Oxygen of the air to produce water vapour, H_2O.

4. In combustion the Nitrogen of the air remains chemically uncombined, but passes up the funnel raised in temperature, thus carrying off heat and representing the principal loss occurring in combustion.

Burning of CO.

A light bluish coloured flame noticed burning at the back of the furnace indicates the presence of CO gas (Carbonic Oxide, or Carbon Monoxide).

During combustion the Hydrogen in the coal, set free by the heat, combines chemically with Oxygen of the air and produces water vapour, the proportions being Hydrogen two volumes, and Oxygen one volume, or H_2O (water).

Carbon particles which have not been supplied with sufficient Oxygen become mixed mechanically with the water vapour and produce black smoke, the intensity of colour depending on the proportion of solid Carbon held in suspension.

If CO_2 gas combines with an additional amount of Carbon the result is the formation of Carbonic Oxide (Carbon Monoxide),

Thus, $C + CO_2 = 2CO$.

Heat in Carbon.

With perfect combustion each pound of Carbon converted into CO_2 gas (Carbon Dioxide) gives out about 14500 Heat Units.

With imperfect combustion each pound of Carbon converted into CO gas (Carbon Monoxide) only gives out 4450 Heat Units: the serious loss of heat resulting from incomplete combustion will thus be obvious.

Scale, Density, and Corrosion.

The generation of steam being a continuous process the supply of feed water must also be continuous, and this naturally results in the concentration of the impurities introduced with the feed water; these impurities develop as scale, increase of density, corrosion, or by suspended matter (oil).

Scale Deposit is due to the more or less combined effects of heat, pressure, and concentration of impurities.

Corrosion is due chiefly to the introduction of gases and acids into the boiler with the feed, but may also be caused by galvanic action.

Scale in forming may also indirectly produce corrosion, as, by the effects of the heat when the scale matter concentrates and deposits, acids (such as Hydrochloric) are set free which cause corrosion.

Composition of Fresh Water and Sea Water.

Fresh Water.

Name.	Grains per Gallon.	Effect in Boiler.
Calcium Bicarbonate (or Carbonate of Lime)	10·8	Forms lime carbonate scale at low pressure and temperature (200° to 212°).
Calcium Sulphate (or Sulphate of Lime)	·3	Forms hard scale at 40 lbs. absolute pressure or 267° Temperature.
Magnesium Sulphate -	0·25	Remains soluble in the water.
Carbonate of Magnesium	1·25	Remains soluble in the water.
Chloride of Sodium (common salt)	1·8	Begins to deposit at 35 oz. density and water is fully saturated at 60 oz. density.
Silica, Iron Oxide, Organic Matter	3·21	Form muddy deposits.
Total Grains per gallon	17·61	

Sea Water.

Name.	Grains per Gallon.	Effect in Boiler.
Calcium Carbonate, or Carbonate of Lime	3·9	Forms scale at low pressures and at temperatures of from 200° to 212°.
Calcium Sulphate, or Sulphate of Lime	93·1	Forms hard scale at about 40 lbs. absolute pressure and 267° Temperature.
Magnesium Sulphate -	124·8	Remains soluble in the water.
Magnesium Chloride -	220·5	Decomposes at 360° Temperature, and liberates Hydrochloric Acid, the Chlorine gas of which produces severe corrosion.
Chloride of Sodium (common salt)	1850·	Begins to deposit at 35 oz. density, and water is fully saturated at 60 oz. density.
Silica, &c. - - -	8·4	Form muddy deposits.
Total Grains per gallon	2300·7	

NOTE.—At densities over 15 oz. per gallon the lime sulphate deposits at lower temperatures than 267°.

NOTE.—Per cent. of Salt (Sodium Chloride) $= \frac{1850 \times 100}{2300} = 80$ **per cent., or** $\frac{4}{5}$.

Incrustation (Scale).

A sample of boiler scale formed from Thames water was composed of the following :—

	Per cent.
Sulphate of Lime ($CaSO_4$) - - - -	20·41
Sodium Chloride ($NaCl$) - - - -	68·25
Magnesic Hydrate (MgH_2O_2) - - - -	2·81
Magnesic Chloride ($MgCl_2$) - - - -	3·14
Oxide of Iron (Fe_2O_3) - - - -	·02
Silica (SiO_2) - - - . -	·11
Organic Matter - - - - -	·10
Moisture (H_2O) - - - - -	5·16
Total - - . . -	100·00

Chemical Composition of Boiler Scale.

	Sea Water Scale.	Fresh Water Scale.
Sulphate of Lime (parts in 100) - -	85·53	3·68
Carbonate of Lime ,, ,, - -	·97	75·85
Hydrate of Magnesia ,, ,, - -	3·39	2·56
Chloride of Sodium (salt) ,, ,, - -	2·79	·45
Silica ,, ,, - -	1·1	7·66
Oxide of Iron ,, ,, - -	·32	2·96
Organic Matter ,, ,, - -	...	3·64
Moisture ,, ,, - -	5·90	3·20
Total - - - -	100·00	100·00

Scale and Oil Deposit Compared.—An oily deposit, $\frac{1}{16}$ inch thick, on the furnaces will result in more overheating of the plates than that produced by $\frac{1}{2}$ inch thickness of lime scale. Oily matter deposits more readily with a low boiler density, as with a higher density the oily mass tends to float on the surface, whereas if the water is nearly fresh the weight of the mass may cause it to sink down and deposit on the combustion chamber tops, smoke tubes, or furnaces.

Temporary Hardness.—When water is boiled at 212° temperature the CO_2 gas is driven off, and the carbonate of lime then becoming insoluble is deposited; thus the "temporary hardness" is got out of the water.

Permanent Hardness.—Lime Sulphate, Lime Chloride, and Magnesia Chloride form the permanent hardness of water, as these substances are not deposited until a temperature of 267° is reached, when the Lime Sulphate becomes insoluble and deposits in the form of Gypsum, or hard close-grained scale. The Magnesium Chloride does not deposit until a temperature of 360° is attained.

The scale formed from Lime Carbonate is more open in the pore than that from Lime Sulphate, is lighter in colour, does not adhere so closely, and is more easily removed. Lime Sulphate scale is often combined with Lime Carbonate, and thus shows as a series of layers of varying shade.

Corrosion of Boilers.

Parts Affected.—Damage to boilers is caused by internal or external corrosion, wearing away of the rusted edges by repeated caulking of same, and by cracks due to unequal expansion or contraction, which finally produce "fatigue" of the metal. The latter condition is particularly applicable to the "saddle" of the furnace where the combustion chamber is riveted to the furnace flange, and which in so many boilers gives trouble by cracking or by leakage. Leakage, if not taken up at once, will produce corrosion, as the decomposition of the water or steam passing the leak liberates Oxygen, which combining chemically with the metal produces oxide of iron. On the furnace sides above the fire-bar level the intense heat sets free oxygen bubbles, which adhere to the plate and produce corrosion by chemical action. Again, the tendency to unequal expansion in the upper and lower half of the furnace is apt to strain the metal at this position and slightly fracture the surface skin, which is thus placed in a condition most favourable to corrosion.

NOTE.—The upper half of the furnace is exposed to a temperature of about 2500°, and the lower half to a temperature of about 1000°. This is due to the fact that the upper half receives the heat of convection, whereas the lower half only receives the heat of radiation.

To sum up, the following positions are most often affected by corrosion, &c. :—

1. On water sides of furnaces above line of fire-bars (see sketch).

2. Bottom of furnace near the back end, and bottom of combustion chamber.

3. Cracks round corrugations of furnaces (expansion and contraction of metal).

4. Cracks near back ends of furnaces at combustion chamber (more often in plain furnaces).

5. Corrosion at tube plates due to leaky tubes, or at combustion chamber back and side plates due to leaky stays.

6. Corrosion at ends of furnaces due to straining set up by expansion and contraction.

7. Corrosion at bottom of boiler and combustion chambers due to deposits and to weak circulation at these parts.

8. External corrosion caused by damp (CO_2 gas) from the bilges, from wet ashes, and other similar causes. If manhole or sludge-hole doors are not tight the leakage will in time produce corrosion, as explained previously.

Furnace corrosion, cracks, combustion chamber and tube leakage are all intensified in the case of forced draught owing to the higher temperature of combustion obtained.

Causes of Corrosion.

1. Oxygen and CO_2 gas brought in with the air in the feed water and set free by heat.

2. Chlorine gas set free by heat at high pressures from the Magnesium Chloride contained in Sea water feed.

3. Galvanic action due to dissimilar metals, such as brass and

steel, &c., connected metallically in a solution of sea water or acid water, such as is found in boilers.

4. Fatty acids set free by heat from oil taken into the boilers with the feed water.

Stearic and Oleic acids are contained in vegetable and animal oils, but not in mineral oil, which is of so-called pure "Hydrocarbon" composition and free from acid of any kind. Cylinder or valve oil should always be of this class.

It should be noted, however, that *any* class of oil if deposited on the furnace crowns may bring about collapse.

Prevention of Corrosion.

Taking the case of new boilers, corrosion can be hindered to a great extent by working as follows :—

1. By using some sea water feed and frequent surfacing, or by the direct addition of lime, first obtain a light protective lime scale of $\frac{1}{32}$ inch thick on the heating surfaces.

2. Use fresh feed water at all times if possible, and avoid altogether the use of sea water as feed. For this a large evaporator is necessary.

3. Employ a feed heater which, in addition to heating up the feed water, extracts most of the air and other gases from the water gases, which (in the case of Weir's heater) pass away to the condenser.

4. Employ a feed water filter which, fitted between the hot-well and boiler check valves, traps most of the oil or grease which may be present in the feed water when it leaves the condenser.

5. Fit inside of the boiler and in clean metallic contact with the plates or stays, slabs of zinc to set up galvanic action between the zinc and steel, which action results in the wasting away of the positive element zinc and the proportionate protection of the boiler plates.

6. If necessary, use soda in moderation.

Hydrochloric Acid.—This corrosive acid is formed as a result of the decomposition at high temperature of certain chlorides present in sea water and reactions between such substances as Magnesium Sulphate and Sodium Chloride. Hydrochloric Acid giving off Chlorine produces corrosion in the steam space of boilers; as, however, it is both volatile and soluble, it may produce pitting below the water line as well as above it.

Scale Deposit and Plate Temperature.—The overheating effects produced on the furnace metal by various thicknesses of lime scale deposit are shown below :—

A scale thickness of $\frac{1}{16}$ inch requires an expenditure of 15 per cent. more fuel.
" " $\frac{1}{4}$ " " " 60 " "
" " $\frac{1}{2}$ " " " 150 " "

At a scale thickness of $\frac{1}{2}$ inch or less the plates may reach the critical temperature of over 600°, and collapse of the furnaces take place. This will easily be understood when it is stated that a temperature of 700° produces a low red heat on the plate.

A scale thickness of $\frac{1}{32}$ inch is quite sufficient to protect the plates from corrosion.

Magnesium Chloride.—This sea water chemical decomposes at a temperature of 360°, and the Chlorine set free combines with iron to produce corrosion. During decomposition of Magnesium Chloride, Magnesia is produced and deposits as a form of mud or slime.

A gallon of ordinary sea water contains about 220 grains of Magnesium Chloride, and under conditions of heat and concentration this substance splits up into Hydrochloric Acid and Magnesia: the acid mentioned combines with the iron of the boiler to form iron salts, and therefore produces corrosion.

Corrosion of Tubes in Boilers.—May be due to (1) fatty acids obtained from the decomposition of animal or vegetable oils; (2) Hydrochloric Acid produced by the decomposition, at a high temperature, of a magnesium and chlorine compound in the sea water; (3) galvanic action; (4) the presence of Carbonic Acid and air in the water.

Red and Black Iron Oxides.—Red oxide of iron (Fe_2O_3) is formed when excessive air is entering the boilers with the feed water, and Black Oxide of Iron (Fe_3O_4) when the air admission is not excessive. Air should therefore be kept out of the boilers as much as possible, as the amount of corrosion will then be reduced in proportion.

Test for Carbonic Acid.—To test for the presence of CO_2 gas in boiler water: make up equal volumes of boiler water and lime water, and if CO_2 is present, the mixture will turn cloudy or milky in appearance.

Evaporator Scale.—The scale in evaporators is chiefly Lime Carbonate, as the pressure and temperature carried is not high enough to produce deposit of much Lime Sulphate. The Lime Sulphate therefore remains in solution in the water, and is blown out by way of the blow-down cock.

Boiler Deposits.—The impurities in boiler water are deposited in the following order:—

1. Carbonate of Lime (at atmospheric pressure and 212° temperature).
2. Sulphate of Lime (at 267° temperature, 40 lbs. absolute pressure).
3. Oxide of Iron.
4. Silica, Alumina, Magnesia Hydrate.
5. Common Salt (at 35 oz. density deposit begins, and the water is fully saturated at 60 oz. density).

Leaky Tubes.—If cold air is admitted to the tubes by opening the smoke-box doors the tubes are apt to contract and leak at the ends. Leaky tubes may therefore be caused by (1) unequal expansion, or by (2) unequal contraction of the tubes and plates. Severe vibration, due to blowing down, may also produce leaky tubes. It should also be noted that leakage ultimately sets up corrosion on the plate round the tube necks and on the tubes themselves.

Density and Scale.—The following points should be carefully noted :—

1. Scale deposit is due to heat, and is quite independent of evaporation, as, even though no evaporation is going on in a boiler (under banked fires), if sea water feed is admitted scale will be deposited shortly afterwards.

2. Increase of density (over 5 oz. per gallon) is entirely due to evaporation, as, if no evaporation takes place, the boiler density cannot rise above that of sea water feed.

Therefore, the limit of density (saturation point) being 35 oz., should the boiler reach this density, and still more sea water feed be pumped in, when the water so fed in evaporates, the salt contained in it will be deposited.

NOTE.—It must be clearly understood that the salt forming the 35-oz. density does not deposit, but only the salt contained in feed water put in after the 35-oz. density has been reached.

General Notes on Scale and Salting.

Salting or saturation of the boiler water means that the density has reached $\frac{7}{32}$, or 35 oz. per gallon. If at this density more sea water feed is admitted the salt in it begins to deposit after the water evaporates.

Scale consists chiefly of sulphate of lime and carbonate of lime.

Scale is caused by the action of heat on sea water feed. The heat concentrates the sulphate and carbonate of lime, and as a result these substances deposit on the tubes, furnaces, &c. This occurs when the air and CO_2 gases are expelled by heat.

To sum up, **scale is due to heat**, and is quite independent of evaporation, as although no evaporation takes place scale may yet deposit. **Increase of density above 5 oz. (sea density) is entirely due to evaporation**, and cannot take place unless evaporation goes on.

All waters contain lime, so that even with fresh water feed scale will form (chiefly carbonate of lime), but the hardest and most troublesome scale is that deposited by sea water feed, as it is largely composed of sulphate of lime.

Sulphate of lime scale forms from a pressure of 40 lbs. absolute and a temperature of 267°.

Scale increases with the pressure, because the temperature is higher in proportion.

The scale thickness depends upon the amount of sea water put into the boiler as feed, and does not depend upon the density of the boiler.

Surfacing a boiler (having no evaporator fitted) reduces the density but increases the amount of scale, because every time the boiler is surfaced extra feed water has to be pumped in, therefore more sulphate of lime will deposit. The scale formed from fresh water feed is composed chiefly of carbonate of lime.

To *form a scale* on a new boiler, keep the density low by surfacing and feed up with sea water. This will cause rapid deposit of sulphate and carbonate of lime, and the formation of a protective scale on the heating surfaces.

As oil in boilers causes pitting, and may also bring down the furnaces by depositing on them, it is best kept out altogether, and this is done by having a feed filter fitted between the feed pump and the boiler check valve.

A feed heater is used for raising the temperature of the feed water, and partly clearing it of air before entering the boiler.

Solids in Sea Water.—The solid matter in sea water (forming $\frac{1}{32}$ of the weight) is chiefly made up (approximately) as follows :—

Sodium Chloride (common salt) - -	80 per cent.
Magnesia Chloride - - - -	10 ,,
Magnesia Sulphate - - - -	6 ,,
Calcium Sulphate (gypsum) - - -	4 ,,

In addition to the above, Calcium Carbonate exists in combination with CO_2, forming Calcium Bicarbonate in solution. The heat drives off the CO_2 and the Calcium Carbonate is left, which, depositing, forms a scale.

Calcium Sulphate.—This forms the hardest scale found in boilers, and is deposited largely from sea water feed. It is similar to plaster of Paris and marble (gypsum).

Calcium Carbonate.—This composition gives a softer scale than Calcium Sulphate unless combined with the latter, and is similar to common chalk.

It is deposited chiefly from fresh water feed when the CO_2, previously in combination with it, is set free by heat.

NOTE.—The scale found in evaporators is principally made up of Calcium Carbonate.

A piece of Calcium Carbonate scale, when broken across, is coarser in the grain than Calcium Sulphate scale.

Soda.—The addition of an alkali such as soda into the boiler feed water has the effect of tending to convert the Calcium Sulphate into

Calcium Carbonate, and thus make the scale less objectionable and less insoluble. Soda being an alkali also tends to combine with acids present in the water, and, forming chemical salts, destroys the corrosive effects of the acids.

The addition of soda into a boiler often causes the density to rise shortly afterwards, as the soda softens the hard scale, and, making it soluble, the water becomes denser in proportion.

Lime.—When lime is used in boilers, the proportion to allow is about $1\frac{1}{2}$ lbs. of lime per day per each 1000 I.H.P., dissolved in 1 gallon of water, and forming what is known as "milk of lime."

This mixture is very useful in forming a scale on a new boiler, and should be used when the boiler water shows of a red (rust) colour.

Rusting.—Iron immersed in pure water free from air will not rust, but with air present rusting takes place owing to the fact that air contains a small percentage of carbonic acid, which in contact with iron decomposes the water and sets free Hydrogen, which thus allows the Oxygen to combine chemically with the iron and form oxide of iron, Fe_2O_3 (or Ferric Oxide).

Grooving.—By this is meant corrosion produced by mechanical causes, such as unequal expansion. An example can be found in the case of vertical donkey boilers, at the bottom of the fire-box, where the expansion at the top end being much more than at the bottom (the latter being rigidly riveted to the shell), the resultant stresses set up crack the skin of the metal circumferentially, and the skin thus broken, allows of corrosion taking place to a serious extent.

Paraffin Oil.—Paraffin oil is sometimes used in boilers with the object of softening the scale, which is then found to be easily removable. The heating surfaces are treated by being rubbed over with the oil and allowed to stand for a few hours before filling up.

Carbonate of Soda.—The moderate use of soda is to be recommended, as the Soda has the effect of converting the Lime Sulphate into Lime Carbonate.

Nitrate of Silver Test for Saltness.—As boilers are now often worked at an extremely low density (1 oz. or 2 oz.), it becomes necessary to employ a more sensitive method of testing for the density than by the salinometer, and this can be carried out by a Nitrate of Silver test. A supply of this chemical having been obtained a few drops added to a glass of the boiler water will quickly discover the presence of salt by the water instantly becoming cloudy. This method is also the most suitable for testing the tightness of the condenser, as the smallest leakage will be indicated.

Nitrate of Silver Test.

This test only proves the presence of salt—not the quantity per gallon.

Solution = 4 per cent. nitrate, the remainder being water, and when made up appears as a colourless liquid (like water).

The presence of salt is indicated by the water becoming cloudy when the nitrate solution is dropped into the sample.

Caustic Soda.—Caustic soda added to boiler water has the effect of converting the Hydrochloric Acid into common salt, thus destroying the corrosive properties of the acid. The usual allowance of soda is about 1½ lbs. to 2½ lbs. per 1000 I.H.P. per 24 hours.

When pitting or corrosion is located in patches or small areas, the plates affected should be scraped clean, and washed with soda solution. A final coating of weak solution of Portland cement will reduce the danger of the corrosion spreading.

Corrosion in the form of oxide of iron formation is apt to occur in those positions of a boiler where the circulation is weak, such as the lower parts of the furnaces and combustion chambers, and if once started may extend to other parts higher up.

Heating Effect of Scale.—A scale of $\frac{1}{32}$ inch is sufficient to protect the plates from corrosion, and any increase over this may lead to serious overheating, especially in the case of forced draught boilers carrying a high water gauge air pressure.

New Boilers.—Messrs Babcock & Wilcox recommend that 10 lbs. of lime per each 1000 I.H.P. be put into the boiler when new, and from 4 to 6 lbs. per day per 1000 I.H.P. for six days afterwards: the object of this is to form a light protective scale on the heating surfaces.

In the first case the lime should be dissolved in water, and poured in through the manhole, and in the second case the mixture (made up as milk of lime) should be put into the hot-well.

Test for Acid.—Draw off some of the boiler water and put into it a few drops Methyl-Orange (obtainable at the chemist). If the water remains yellow then it indicates an alkaline condition, but if the water turns into a pink colour it indicates the presence of acid.

Action of Lime.—Lime added to the boiler water has the effect of converting the Magnesium Chloride into Magnesia and Calcium Chloride, the former being corrosive and the latter non-corrosive.

Hydrometer.

This instrument, which is simply a type of salinometer, is employed in Naval practice, and is generally graded for use at a temperature of 80° instead of 200°. The divisions or degrees represent half ounces, so that 20 indicates a density of 10 oz., 15, 7½ oz., and so on. The hydrometer thus allows of finer density readings than those obtained by the ordinary salinometer.

Oils.

Lubricating Oils.

For cylinder lubrication, mineral oil only should be used, as this class of oil is composed of Hydrogen and Carbon and is free of acids. The flash point should not be less than 400° Fahr.

The Viscosity of an oil or cohesive nature of the fluid should be such that the lubricant is not so thick as to produce friction, nor yet so thin as to be squeezed out from between the surfaces in contact. Viscosity may be said to be the **consistency** of the oil, and should be high for heavy bearings and low for lighter wearing parts. The viscosity of all oils becomes reduced with increase of temperature.

Gumminess.—If an oil evaporates easily it will naturally become gummy and lose proportionally its lubricating properties. A rough test for gumminess can be made by painting over an earthenware dish with the oil and placing it in a warm position, where it can be tested by hand at intervals for stickiness.

Classes of Oil.

Mineral Oils.
1. Extracted from Shale, and Cannel coal.
2. Found in Russia and America in oil springs. Paraffin, Petroleum, Kerosene, Benzine, Naphtha, &c.

Vegetable Oils. Colza, Linseed, Rape, Castor, Olive, Cottonseed, &c. Manufactured from the seeds of the plants named.

Animal Oils.—Sperm (whale), Seal, Neatsfoot.

At moderate temperatures vegetable oils decompose into Oleic acid, and animal oils into Stearic acid, both of which act corrosively on the cylinders, valve faces, and on the boilers if admitted with the feed water.

Oil Emulsion.—Although feed water filters collect most of the grease or oil in the water often a certain amount passes the filter cloths in the condition known as "emulsion," and the small atoms of oil in this state combine easily with the Magnesia present in the boiler water, resulting in the formation of a slimy deposit, which forms a bad non-conductor of heat, and which may bring about buckled plates or collapsed furnaces.

"Saponification."—If animal or vegetable oils enter the boilers with the feed water, and soda is present, then the fatty acids of the oil (set free by the heat) combine with the soda to form a soapy substance which is a particularly bad conductor of heat, and which if deposited on the furnace crowns is likely to bring about buckling or collapse of the same. The combination of the acids and soda is called "saponification."

Care of Boilers.—Regarding oil the following points should be attended to:—

1. Use minimum quantity of cylinder oil.
2. Work at a density which will "float" any oil which may enter boilers.
3. Clean filter cloths weekly.

To Test Acidity of Oil.

A. Make up a solution of Sodium Chloride with an equal weight of water, take a measured quantity of the solution and an equal quantity of the oil, which place together in a bottle. Shake up the bottle and allow it to stand, after which, if acid is present, it will show by settling to the bottom of the bottle. If no deposit takes place the oil is free from acid.

B. Chemically prepared papers of a pale bluish tint and known by the name of "litmus papers," can be obtained in little books at a very small cost from any chemist, and these can be applied to test the acidity of an oil. To make the test, boil some of the oil and dip one of the litmus papers into it, and if, on withdrawing, the colour has become a deeper shade only, the oil is clear of acid; but if the paper changes to a *pink* or *red* colour, it denotes with certainty the presence of acid in the oil, which should therefore be rejected for purposes of internal lubrication. The redder the paper becomes, the greater is the quantity of acid present in the oil.

Boiler water can be tested for acid in the same way.

NOTE.—A clean copper wire if immersed in oil for a few hours will show discoloration if acid is present.

Viscosity Test for Oil.—The viscosity of an oil is tested in the following manner. The apparatus consists of a small cup-shaped vessel fitted with an internal pan, in the bottom of which a small round hole is truly bored, and a thermometer dips into the oil; the oil is then heated up to a fixed temperature, say 180°, and the time taken for a measured quantity of oil to drip out through the small hole in the bottom is noted. Oil of high viscosity will take a longer period to escape than oil of low viscosity.

To Test for Animal or Vegetable Oils.—To test whether an oil is of vegetable or animal nature add a small portion of chlorine to the oil, and note the change of colour; if to brown the oil is animal, and if to white the oil is vegetable.

Alkali Test.—To test for alkalinity use red-coloured litmus paper, which will change to blue if the water is strongly alkaline.

Most of the oil used for internal lubrication of the engines finds its way to the boilers, (unless extracted by means of a filter), by being brought with the steam into the condenser, and afterwards pumped into the boilers by the feed pumps.

"Alkala" Boiler Composition.

Lambie's patent boiler composition, known as "Alkala," has proved, after severe and exhaustive tests, to be most effective in the prevention of scale deposit and corrosion; used in conjunction with the usual zinc plates, boilers working under the highest pressures are found to be in excellent condition after months of hard steaming, the scale being of a light, easily removable nature, and the pitting or corrosion checked to a remarkable degree. Alkala is a paint and is applied by brush to the parts requiring protection, such as furnaces, plates, stays, and tubes. The composition referred to is rapidly becoming known as the most efficient boiler preservative in the market.

Remedies for Pitting.

Zinc plates (see page 143) are used internally to check the wasting of the plates, and are fitted so as to form with the boiler plates a galvanic couple, of which the zinc is the positive element. The zinc plates are connected metallically to the boiler by the following methods:—(1) Studs screwed into the furnace sides; (2) metal hangers suspended from the stays. Sometimes zinc balls with a copper wire passed through them and connected to the boiler are used instead. These are called "Electrogens." Externally, feed heaters (such as Weir's) assist in keeping out the air, and feed filters assist in keeping out the grease or oil.

Galvanic Action.

If two dissimilar metals are placed in a bath of sulphuric acid, both will in time show signs of corrosion; but, if the two metals are connected by a copper wire soldered to each, then only one of them will corrode, as a galvanic couple is then formed, and the corrosive effects take place on the most electro-positive metal only. (See page 432.)

By connecting the two metals a weak electrical current is set up between them, through the liquid and wire, resulting in the wasting away of the one element which is electro-positive to the other.

Examples of the foregoing are to be found in the boilers by the zinc plates fixed inside, and on the tail end shaft at the end of the brass liners, the brass and iron or steel of the shaft being dissimilar metals and connected in a bath of sea water. Sea water, as it contains salt, acts in a like manner to sulphuric acid, but in a milder form.

NOTE.—The current flow is produced by the difference in electrical potential obtained.

Rusting.

Rusting of a metal is an example of combustion at a low temperature, and is due to the Oxygen of the air combining with iron (or steel) to form Oxide of Iron.

The chemical name for red iron rust is Ferric Oxide, or, by symbols (Fe_2O_3).

Rusting is in reality the burning of the metal, but at a low temperature (the atmospheric temperature).

Rusting can only take place when Carbonic Acid gas (CO_2) is present in quantity, as in a damp atmosphere.

Rusting cannot occur in a dry atmosphere.

A good example of rusting, or the formation of oxide of iron, is to be seen at the water line on the wet uptake of vertical boilers. The action that takes place is as follows:—The intense heat on one side of the plate sets free the Oxygen of the water adhering to the plate on the other side, and the Oxygen being present with Carbonic Acid gas combines with iron to form Oxide of Iron, or rust (Ferric Oxide).

If a rivet or stay in the combustion chamber leaks, and the leakage is not at once checked, the plate and stay will soon begin to waste away, owing to the formation of Oxide of Iron.

Oxide of Iron formation also accounts for the wasting away of boiler bottoms, just above the bilges, furnaces at the ashpits, and on the tops of tanks.

Condenser Tube Corrosion.

In most cases the corrosion of condenser tubes is due to one of the following causes:—

(1.) Galvanic action produced between the condenser metal and tubes in sea water, resulting in the loss of the zinc, of which the tubes are partly composed. This produces small holes in the tubes at various positions throughout the length, but usually near the ends, and is known as "de-zincification."

(2.) Acids from animal or vegetable oils producing corrosion on outer surface of the tubes by chemical action.

(3.) General thinning of the tube by wear, and caused by the long-continued mechanical attrition action of the water inside the tubes.

In the great majority of cases the corrosion is caused by decomposition of the zinc, as stated in No. 1 cause.

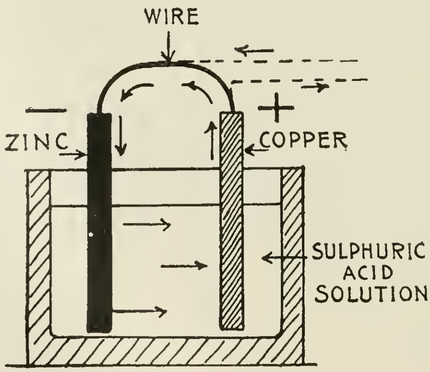
SECTION VII.

MARINE ELECTRIC LIGHTING.

General Description.

THE following elementary description of electric lighting is intended by the author for marine engineers who have had no experience with dynamos, and who may have had no opportunity to study electricity—in other words, for beginners.

Galvanic Cells or Batteries.—An electric current may be produced either by chemical or mechanical means. If chemically, by a galvanic battery or cell; and if mechanically, by a dynamo.



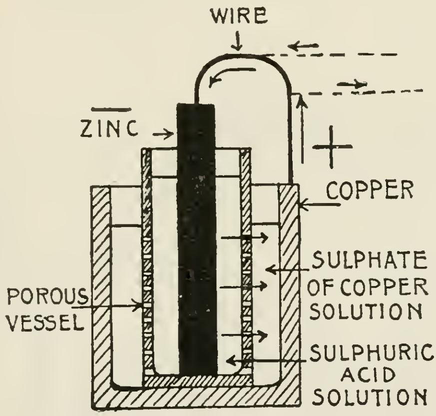
No. 1.—Simple Cell.

The sketch shows a simple cell, or galvanic couple, consisting of two plates, one of Copper and one of Zinc, placed in a bath of sulphuric acid, and connected together at the top by a wire. A current flows from the Copper plate through the wire to the Zinc plate, and back again from the Zinc through the liquid to the Copper. The Copper is called the "positive pole" but the *negative* element, and the Zinc the negative pole and the *positive* element. The dotted lines in the sketch show how the current may be led to an external circuit so as to do work, such as, for example, to operate an electric bell, instead of passing direct from one plate to the other, as shown by the full lines.

It should be noted that the current flowing in the direction described results in the dissolving of the zinc plate, and the formation of sulphate of zinc.

NOTE.—The current flow is produced by the difference of electrical potential obtained.

Daniell Cell.—One form of this well-known type of galvanic cell is shown in the sketch, in which the outer vessel is of Copper and constitutes the positive pole. The inner vessel is of porous construction to allow of the liquid to pass through gradually, and the negative pole is formed by a rod of Zinc placed inside. Two liquids are em-

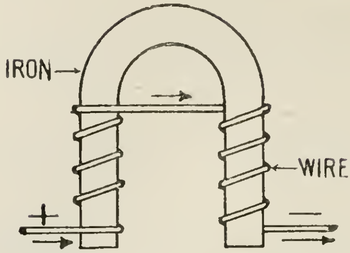


No. 2.—"Daniell" Cell.

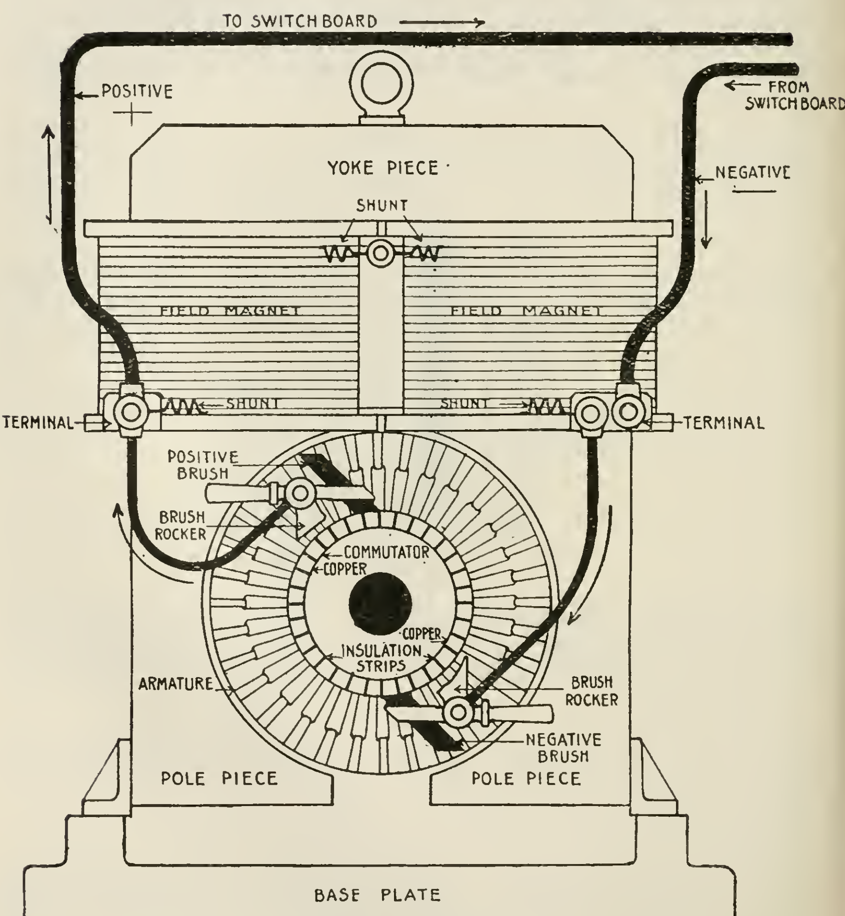
ployed—sulphate of copper solution in the outer vessel, and, as in the simple cell, sulphuric acid in the inner vessel. The current flows from the Copper to the Zinc as described previously in the case of the simple cell.

The electro-motive force of the Daniell cell is fully one volt. Dotted lines are shown in the sketch to illustrate the manner in which the current may be applied to an external circuit, instead of passing directly from one pole to the other.

Electro-Magnets.—If a bar of iron is enclosed within a coiled wire, and a current from either a galvanic battery or a dynamo passed through the wire, the bar becomes magnetised for so long as the current is passing.

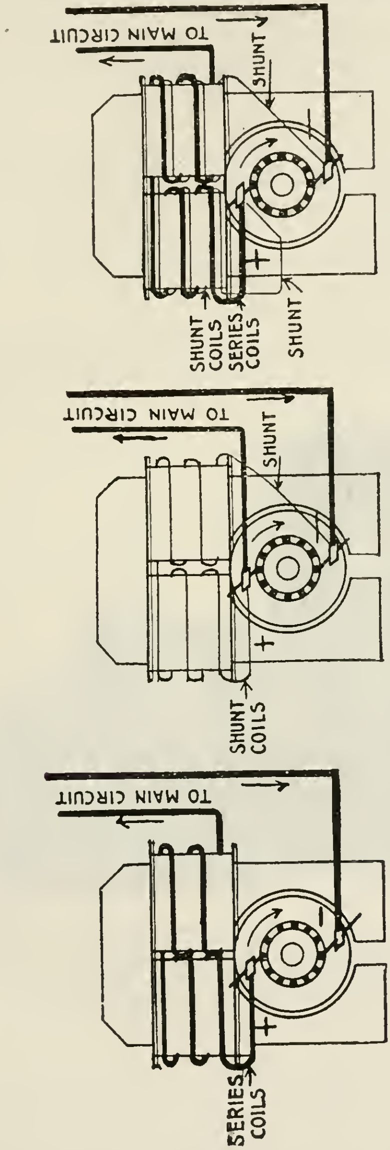


No. 3.—Electro-Magnet.



No. 4.—Dynamo.

The bar is then known as an electro-magnet. If the bar is bent into a horse-shoe shape so as to form two legs, when a current is



No. 5.—Series Wound Dynamo. Shunt Wound Dynamo.
Compound Wound Dynamo.

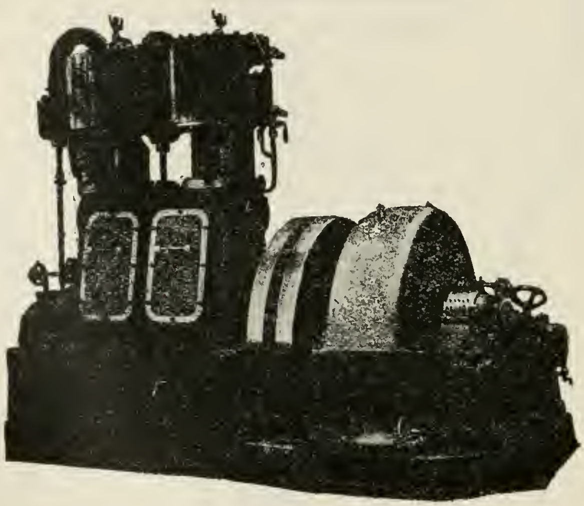
passed through the wire both legs of the bar become magnetised, and form a pair of electro-magnets. The space between the legs is called the "magnetic field," and if an armature be made to revolve in the magnetic field so as to cut the lines of force or magnetism passing across, currents are generated in the coils or conductors of the armature. The two legs constitute north and south poles.

The field magnets of a dynamo are of the type just described, the poles receiving their magnetism from the dynamo itself, as the winding of the magnets consists of wires led direct from the brushes, the wires obtaining their current direct from the armature.

A dynamo is a machine which, by the application of mechanical power, generates electro-motive force. This electrical force or power can be utilised either for lighting or for the running of motors, or for both purposes combined.

The four principal parts of a dynamo are—(1) Field magnets; (2) armature; (3) commutator; (4) brushes.

Field Magnets.—The field magnets consist (in a two-pole dynamo) of two masses of cast steel connected at the top by a "yoke piece"



No. 6.—"Belliss" Engine direct coupled to G.E.C. Witton Generator used for lighting and power on board Steamships, Shipyards, Dockyards, Factories, &c.

which gives to them the well-known horse-shoe shape. Each magnet is wound with coils of insulated copper wire which are in connection with the armature by one of three ways—(1) In series; (2) in shunt; or (3) compound, which is a combination or compound of the first two.

In *series* winding the *whole* of the current generated in the armature passes from the brushes round the field magnets, and then to the lamps and back again.

In *shunt* winding only part of the current passes round the field magnets, as the shunt wire to the magnets is smaller and finer than the series wire, and in this way offers more resistance to the current.

In *compound* winding the field magnets are wound with two sets of wire, and the whole of the current generated in the armature passes round them, but by two distinct and separate paths; first by the thick or series coils, and next by the thin or finer "shunt" coils. The object of this method of winding may be described as follows:—As lamps are switched on and more current is required, the extra current, on its way to and from the lamps, passes through the series coils of the magnets, and therefore strengthens the magnetic field in proportion. Again, if a certain number of lamps are switched off, less current passes through the series coils as less is now passing through the main wires, but more current will pass through the shunt coils, and thus tend to maintain the same strength as before in the magnetic field, and keep the voltage constant.

It will thus be seen that a compound wound dynamo is, to a great extent, self-regulating, and retains practically the same voltage, no matter how many lamps are on or off. This is of great importance in ship-lighting, and for this reason nearly all dynamos used for marine purposes are of the compound wound type.

It should be noted that the exciting current for the magnets comes from the armature itself, and though small at first, increases as more current is developed, so that the one, in a sense, supplies the other in proportion to the demand.

On examining a dynamo of the compound wound type, it will be noticed that the wires from the brushes are connected to terminals or studs fixed on an insulated plate which is placed on the side of the field magnets, and from these terminals the main leads or wires to the lamp circuits are branched off, and the series coils and shunt coils to the magnets.

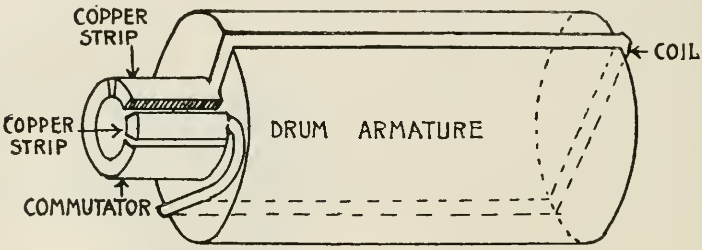
The fine shunt wire will generally be seen to connect across the two magnets and to the main terminals, as shown in the sketch.

Four-Pole Dynamos are now being supplied for ship-lighting, and it is evident that this type is rapidly coming to the front and taking the place of the older-fashioned two-pole machine. In some cases six-pole machines are supplied.

The four-pole dynamo is supplied with four sets of brushes, each alternate set being in connection, so that there are two sets of positive and two sets of negative brushes in use. The other parts of the machine are similar to those of the ordinary two-pole dynamo, but the four-pole type is of more compact form, and can be made of high magnetic strength. On examining a dynamo of this type, the thick wires of the series and the fine wire of the shunt will be noticed extending from one magnet to the other, and connecting them with the brushes and lamp cables.

Armature.—The two types of armature in general use for ship-lighting dynamos are those known as the "Ring" and "Drum" type, the latter being usually preferred and supplied as being the best suited for the work. It will therefore be sufficient to describe this type of armature in detail.

The "Drum" pattern of armature is formed of sheets of soft iron or steel discs insulated from each other and clamped together on a sleeve keyed to the spindle or driving shaft. The sleeve is cast with recesses and webs so that air may pass freely from end to end, and by the ventilation so afforded prevent excessive rise of temperature when the armature is revolving at a high speed. The iron or steel discs are slotted longitudinally for the reception of the insulated copper conductors or wires, and each slot is also carefully insulated



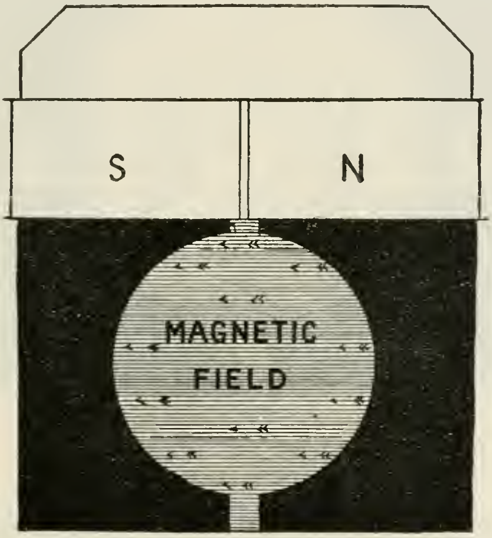
No. 7.—Drum Armature, showing one Conductor Connected.

and the copper wire embedded in it. The complete armature consists of a number of these conductors wound on the surface of the compressed iron sheets or discs, and each conductor passes from the commutator end back to the other end, and then forward again to the commutator end, where the extremities of each conductor are connected (usually soldered) to the copper strips or bars of the commutator.

Each conductor has one end connected to one copper bar of the commutator and the other end connected to the adjacent bar, so that when all the conductors are fixed and bound in place they form an enclosed ring right round the armature and commutator, and thus allow of the passage of the currents generated from the one end to the other, and so to and from the brushes and main wires.

Calling the commutator end of the armature the front, and the other end the back, it should be noted that when the armature is revolving, currents are passing from the front end to the back in *one half* of the armature circle, and from back to front in the other half of the armature circle. The currents passing from front to back are negative, and those from back to front positive, and the corresponding brushes are placed in contact with the commutator to transmit the current flow in the directions indicated.

It will be seen then, that current is constantly passing from the back end of the armature to the front and into the commutator strips, and from there to the positive brushes ; also, that current is constantly passing from the negative brushes into the commutator and along the armature from front to back. It therefore follows that if we were to connect the brushes, positive and negative, the circuit would be complete, but if we first extend the connection before joining the positive and negative we will still have the circuit complete. This is done in practice by joining cables to the brushes and carrying the wires or cables to different parts of the ship, where they are connected at



No. 8.—Diagram showing Magnetic Field and "Lines of Force."

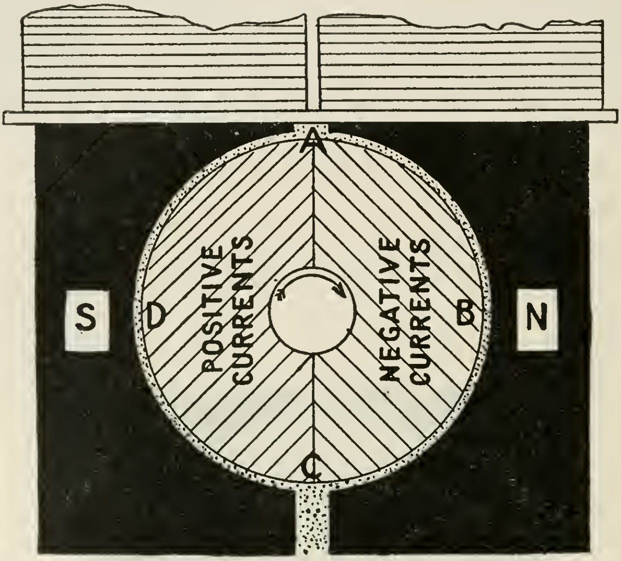
various points by the lamp wires through which the current must first pass on its way back to the negative brushes of the dynamo.

NOTE.—Field magnets are generally composed of hard steel, as when once magnetised the magnetic influence is retained for some time, and thus allows of the "building up" of current when the dynamo is first started, otherwise the machine would refuse to generate.

The armature core is made up of soft or charcoal iron discs, as this material allows of rapid magnetic saturation and of as rapid demagnetisation, which is a necessary requirement in an armature.

The above diagram illustrates, in an imaginary way, the field space of a two-pole dynamo, and shows how the lines of magnetic force are supposed to extend from the N pole to the S pole, the space between the poles being filled up with these invisible lines of force.

It can easily be imagined, then, that if the field space be filled up with an armature composed of soft iron, the lines of force will meet with less resistance in passing across from pole to pole.



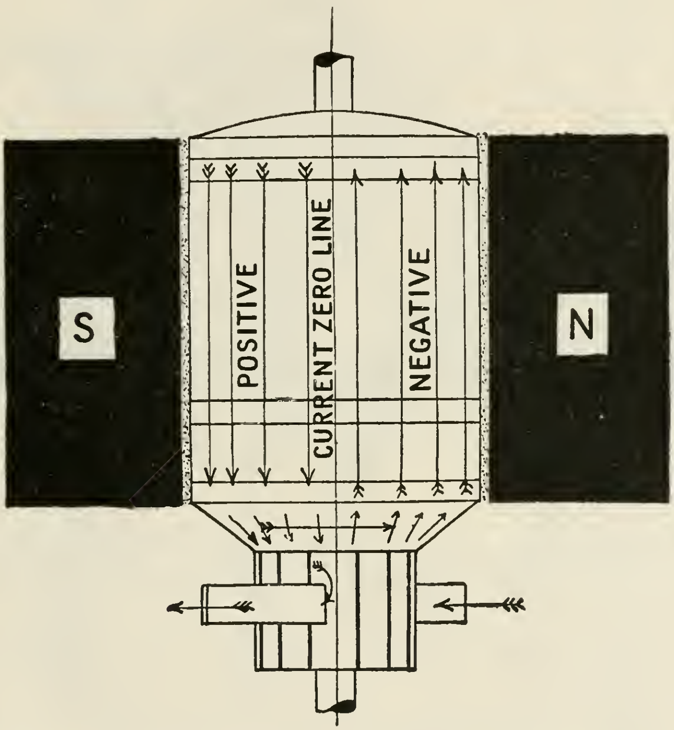
No. 9.—Diagram of Magnetic Field.

From A to B negative currents are increasing in intensity.
From B to C negative currents are decreasing in intensity.
From C to D positive currents are increasing in intensity.
From D to A positive currents are decreasing in intensity.

At positions D and B the armature conductors are cutting through the greatest possible number of "lines of force," hence the intensity of the current strength at these positions.

On the left-hand side the currents are passing from back to front, and are positive, while on the right-hand side the currents are passing from front to back, and are negative. A and C are the zero positions of current change from positive to negative, and from negative to positive.

Sketch No. 10 shows the direction taken by the current in each half of the armature circle. Notice that on the "positive" side the currents are travelling in the armature conductors from back to front (calling the commutator end the front), and so into the positive brush and on to the lamp circuits; while on the "negative" or "return" side, the currents are passing from the negative brush back into the commutator bars, and so into the armature conductors.

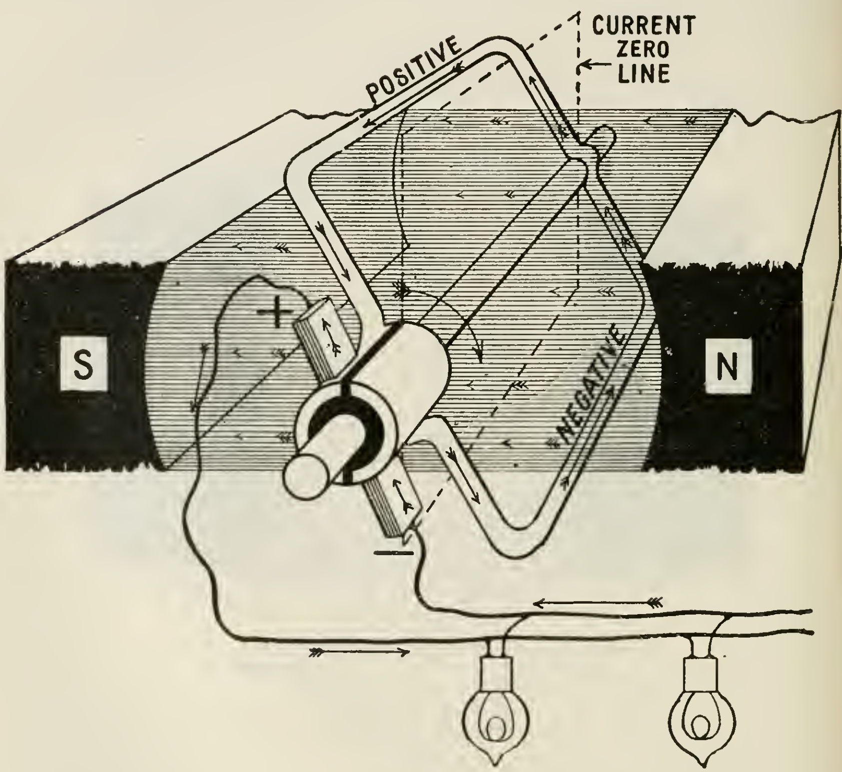


No. 10.—Plan of Armature and Field Magnets.

Sketch No. 11 is self-explanatory, and shows diagrammatically the positive and negative current flow in an armature conductor at different parts of the circle. The plus sign indicates positive connections, and the minus sign the negative connections. Observe that at the top "zero" line the current changes from positive to negative, and at the bottom "zero" line the current changes from negative to positive.

NOTE.—A volt is the E.M.F. required to produce one ampere of current when opposed by one ohm resistance.

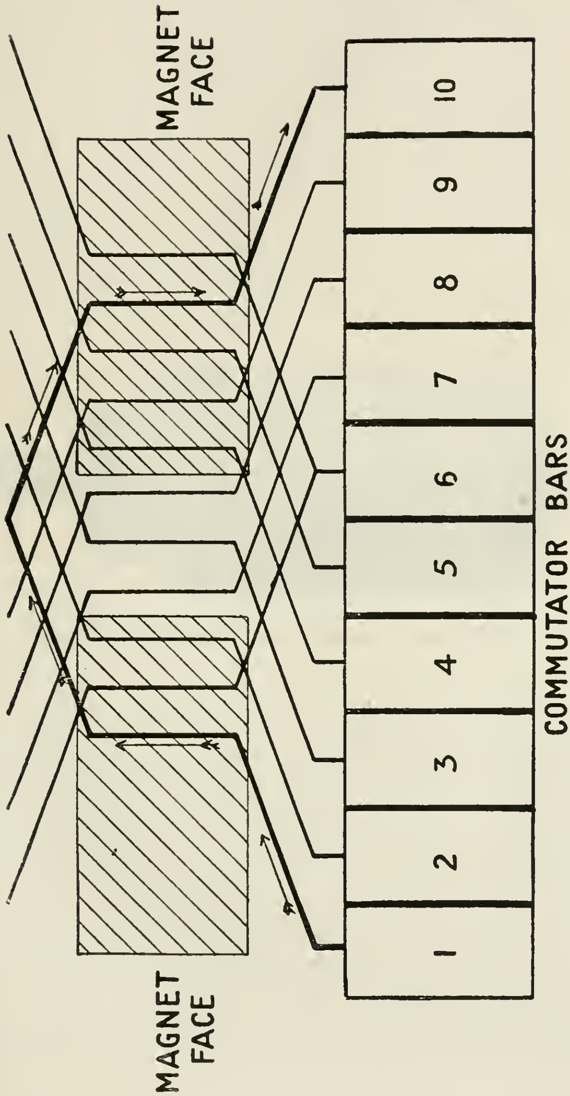
Diagram No. 12 shows one method of armature winding for a two-pole dynamo. Observe that, when arranged in a circle, the commutator bar No. 1 is adjacent to bar No. 10, and that the corresponding conductor extends between the two, being soldered to each.



No. 11.—Diagram showing Flow of Current in Single Conductor of Armature.

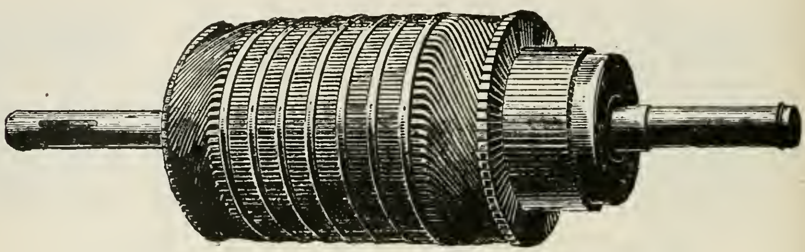
The arrow shows the run of the conductor referred to, and also shows the direction of the current flow. Each armature conductor is similarly connected.

NOTE.—In the above diagram the complete conductor connections for No. 6 bar only are shown. From this it will be seen that two ends of two different conductors are connected to each commutator bar.



No. 12.—Diagram of "Wave" Winding for Two-Pole Machine (Commutator opened out or laid out flat).

Commutator.—All dynamos are originally what is called alternating —that is, the currents passing in the armature coils are always passing alternately in opposite directions, from back to front and from front to back; but by means of a commutator these alternations of current can be made to flow always in one direction, and the dynamo is then known as of the "continuous current" type. For ordinary lighting purposes this type is always employed, but for large power stations (such as for electric tramways) the alternating type is used, as being better suited for high tensions or voltages. As mentioned before, currents are constantly passing from one end to the other in one half of the armature circle, and in the reverse directions in the other half, so that if the brushes are placed at the neutral points where the direction of the currents is reversed, the flow will then be "commuted" or continuous in the one direction. From this it will be



No. 13.—Armature and Commutator Complete.

apparent that to make a dynamo of the continuous current or direct type a commutator is necessary.

The commutator consists of hydraulic pressed solid copper bars arranged in a circle on a sleeve keyed to the driving shaft at the end of the armature, each bar being insulated from the one adjoining by mica insulation. As before described, the ends of the copper conductors of the armature are connected metallically to the commutator bars, one end of two separate wires being so connected to one copper bar. The current after passing from the armature into the commutator is absorbed or collected by the positive brushes, and after going through the lamp circuits is again delivered back to the commutator and armature by the negative brushes. Observe that a complete circuit must be described before any current can possibly pass.

Brushes.—The brushes are composed either of copper gauze or of carbon blocks, and are held up against the commutator at an angle by a holder fixed to the brush-rocker. The angle of brush adjustment can be regulated by means of a pair of handles on the rocker, and the brushes moved round the commutator circle more or less according to what may be required to prevent sparking, &c. In some dynamos the position of the brushes has to be altered according to the load on the machine in order to get sparkless collection of the current, but in modern machines fitted with carbon brushes this is not necessary.

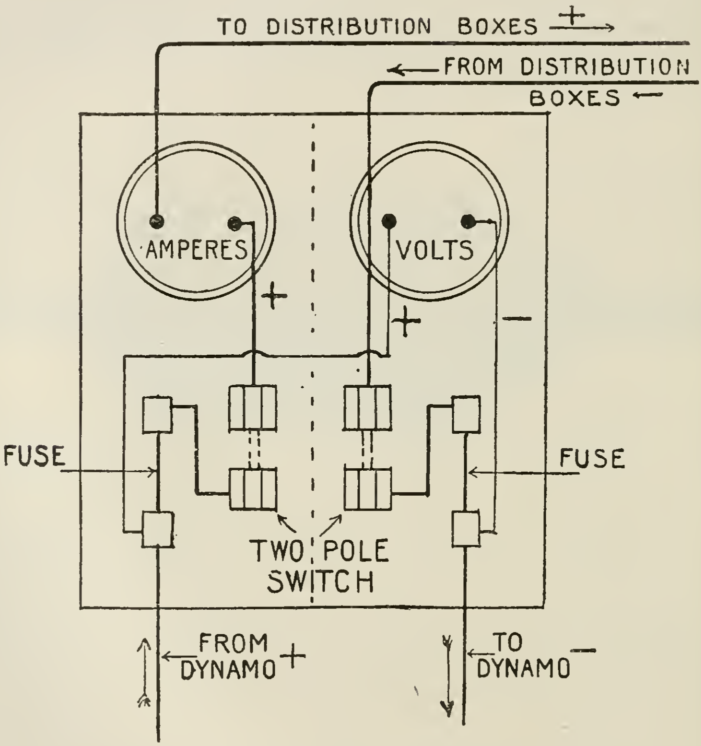
Small springs of low tension are often fitted to the brush-holders to keep the brushes pressed gently against the surface of the commutator. If the brushes press too tightly, they are apt to cause uneven wear of the commutator, and consequent sparking.

The positive brush (or set of brushes) takes the current from the dynamo and gives it to the main wires, and the negative brush (or set of brushes) returns the current back into the dynamo after it has travelled round the lamp circuits.

Armature Shaft.—One end of the armature shaft is coupled direct to the driving engine shaft, and the other, or commutator end, runs in a bearing cast on the bed-plate of the machine. This bearing is generally of the enclosed type, and supplied with revolving oil rings and oil bath which allow of constant lubrication. As the dynamo usually runs at from 250 to 500 revolutions per minute, it is of the greatest importance to have regular and reliable lubrication of the end bearing, as will easily be understood from the foregoing.

Action of Dynamo.—The principle of action of a dynamo may be described as follows:—As the armature, containing the copper conductors or coils, revolves between the two poles of the magnets, it cuts through the lines of force or magnetism which pass across from one pole to the other, and as a result, currents travelling from end to end are induced in the armature coils. The space in which the armature revolves is called the magnetic field, and this field is strengthened or excited by the magnets themselves, which are wound with insulated copper wire through which current is passing, so that, as more or less current passes round the magnets, more or less strength is obtained in the "field," and consequently in the armature coils. In a "shunt" dynamo, as lamps are switched in and more current is taken from the armature, the E.M.F. falls, hence to get the same voltage at full load as at no load it becomes necessary either to increase the speed or the "field" strength. The latter is more convenient, and is accomplished by the addition of series coils, thus "compounding" the machine. If more current is now taken from the armature more passes through the series coils, thus increasing the field strength, hence the voltage remains the same. On the other hand, when lamps are switched off, the E.M.F. of the dynamo would increase; but this is prevented by the decrease in the strength of the field due to less current passing through the series coils, the amount passing through the shunt coils remaining the same in both cases. For ship-lighting "over compound" dynamos are used, which give, say, 100 volts at no load, and 105 volts at full load, the speed being the same in both cases. From this will be seen the reason why a compound wound type of dynamo is best suited for ship-lighting where a constant voltage is required.

Switchboard.—After the current is generated in the dynamo it passes from the positive brushes and leads or cables to the switch-

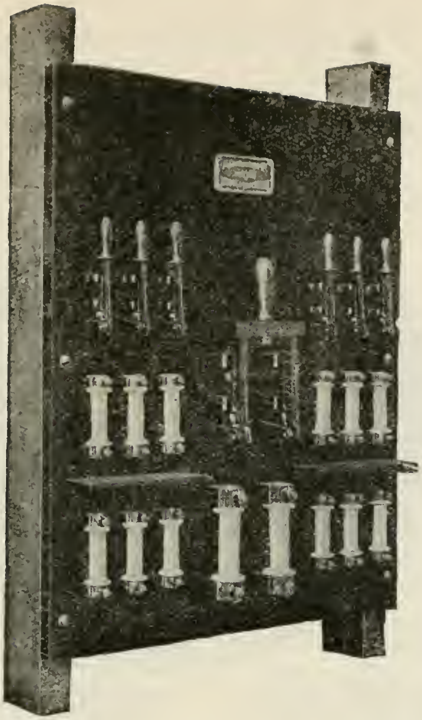


No. 14.—Diagram of Switchboard.

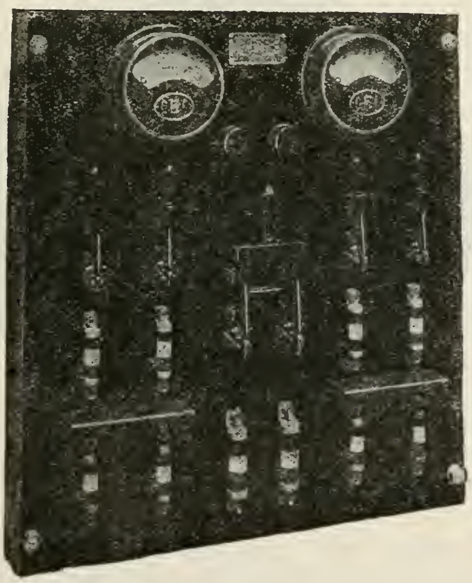
board, and from there is led away by smaller branch wires to the distribution boxes and the various lamp circuits in connection.

The switchboard contains the switches or "circuit-breakers," fuses or cut-outs, volt meter and ampere meter.

After the current has travelled from the switchboard through the lamp circuits, it returns to the switchboard by the "negative" cables or wires, and is returned again to the dynamo by the negative brushes.

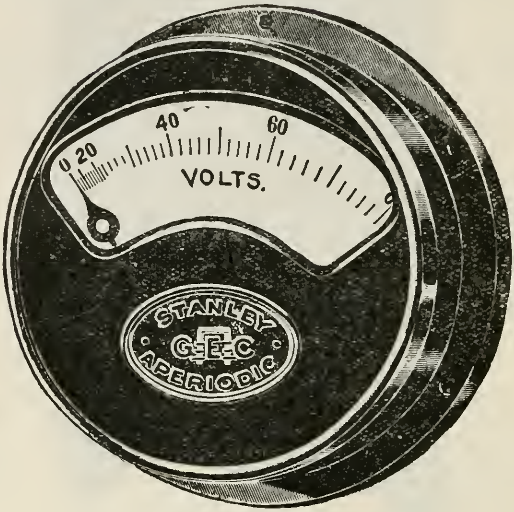


No. 15—G.E.C. Main Distribution Switchboard



No. 16.—G.E.C. Main Switchboard

Volt Meter.—The volt meter is an instrument for measuring the electrical "tension" or pressure. It is connected *between* the positive



No. 17.—G.E.C. Volt Meter.

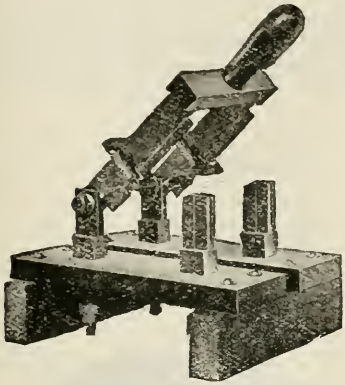
No. 18.—"Stanley" Aperiodic Ammeter.

and negative wires, and therefore indicates the difference of "potential" or pressure existing between them. A volt then, is the unit of electrical pressure, or electro-motive force, as it is called.

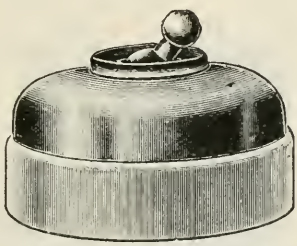
Ampere Meter.—This instrument measures the amount or quantity of current passing through the wires, and is generally connected to the positive lead only.

If each lamp requires, say, 1 ampere of current, and 65 lamps are switched on, the ampere meter would then indicate 65 amperes fully, allowing for the extra voltage required to *overcome the resistance* of the wire.

From the foregoing it will be understood that if, say, 20 lamps are switched on, and the volt meter is showing 70 volts, and the ampere meter 22 amperes, if then 60 lamps are switched on, the



No. 19.—G.E.C. "Witton" Main Switch shown in "Off" Position.



No. 20.—G.E.C. New Pattern Flat Type Switch. A very neat switch for use on board ship.

ampere meter will now show a corresponding increase, but the volt meter will remain the same as before, as, although the amount of current required is now more, the pressure of the current is still the same.

NOTE.—It should be noted that this only holds good when lamps are run on the "parallel system," as is common in ship-lighting.

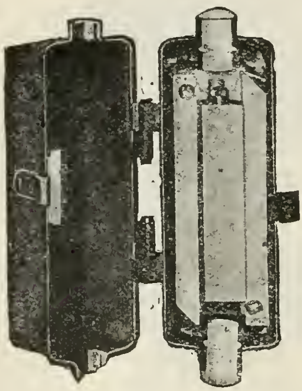
Main Switches.—Switches are employed to break the flow of the current when opened, and to allow the current to flow when closed. They are sometimes called "circuit-breakers." Switches may be of the following types :—

1. Single pole, single break.
2. Single pole, double break.
3. Double pole, single break.
4. Double pole, double break.

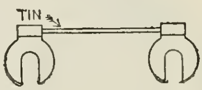
A single-pole switch breaks the current at one pole or wire (positive or negative), and if single break at one place only, if double break at two places.

A double-pole switch breaks the current at both poles or wires, positive and negative, at once, and if single break the current is broken at one place only, but if double break the current is broken at two places on each wire. The requirements of a good switch are —(1) When "on" the contact is complete with ample surface; (2) When "off" the circuit is absolutely broken, without any possibility of a short circuit; (3) A sharp or quick "break" so that sparking may be avoided.

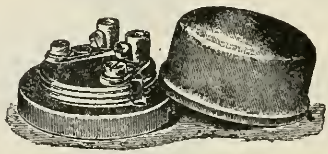
Fuses or Cut-outs.—A cut-out or fuse may perhaps be described as an automatic circuit-breaker. It consists, in most cases, of a strip



No. 21.—G.E.C. "Ironclad" Cut-out.



No. 22.—Fuse.



No. 23.—Cut-out in Case.

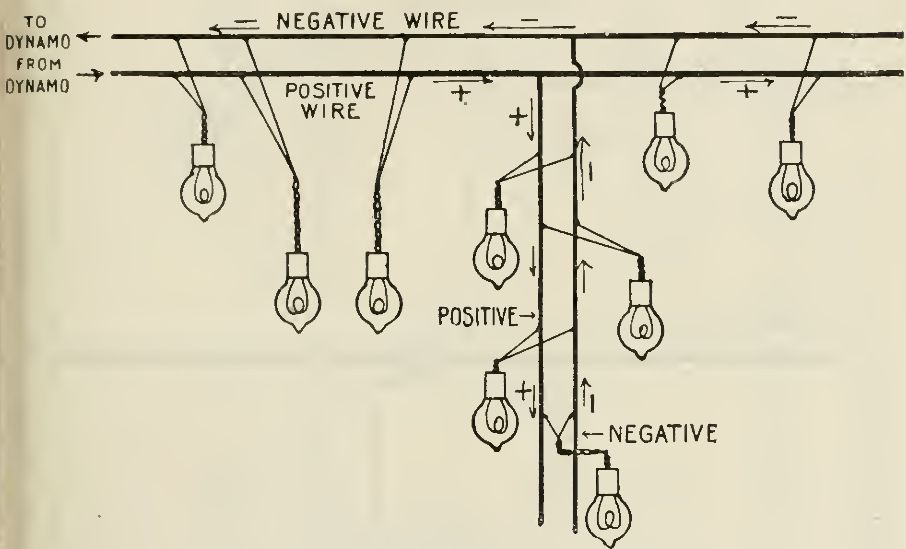
of lead or tin, placed in line with the main wire, which is cut to allow of the fuse being fitted in its place. The ends of the fuse are connected to small terminals or screws.

Cut-outs are arranged so that should an excess of current (caused by some defect or breakdown) attempt to pass, the strip of tin or lead would melt and break the circuit automatically, thus preventing further damage by burning out to the rest of the circuit beyond where the cut-out is placed. Fuses are fitted on switchboards, distribution boxes, and for general safety at various other places on the lamp circuits.

Wiring.—The main wires divide on the back of the switchboard, and branch pairs of wires, positive and negative, of smaller size, are led off to the "distribution boxes" situated at different parts of the ship.

Each pair of wires is connected to a couple of brass bars in the distribution box, and from each bar a number of wires of still smaller size are led away to supply the various lamp circuits in connection.

The incandescent lamps are placed *between* each pair of wires, positive and negative, so that every single pair of wires supply a number of lamps, the current crossing from one wire (the positive) through the lamps to the negative wire. This arrangement is what is known as the "parallel system," the requirements of which are that



No. 24.—"Parallel" System of Lighting.

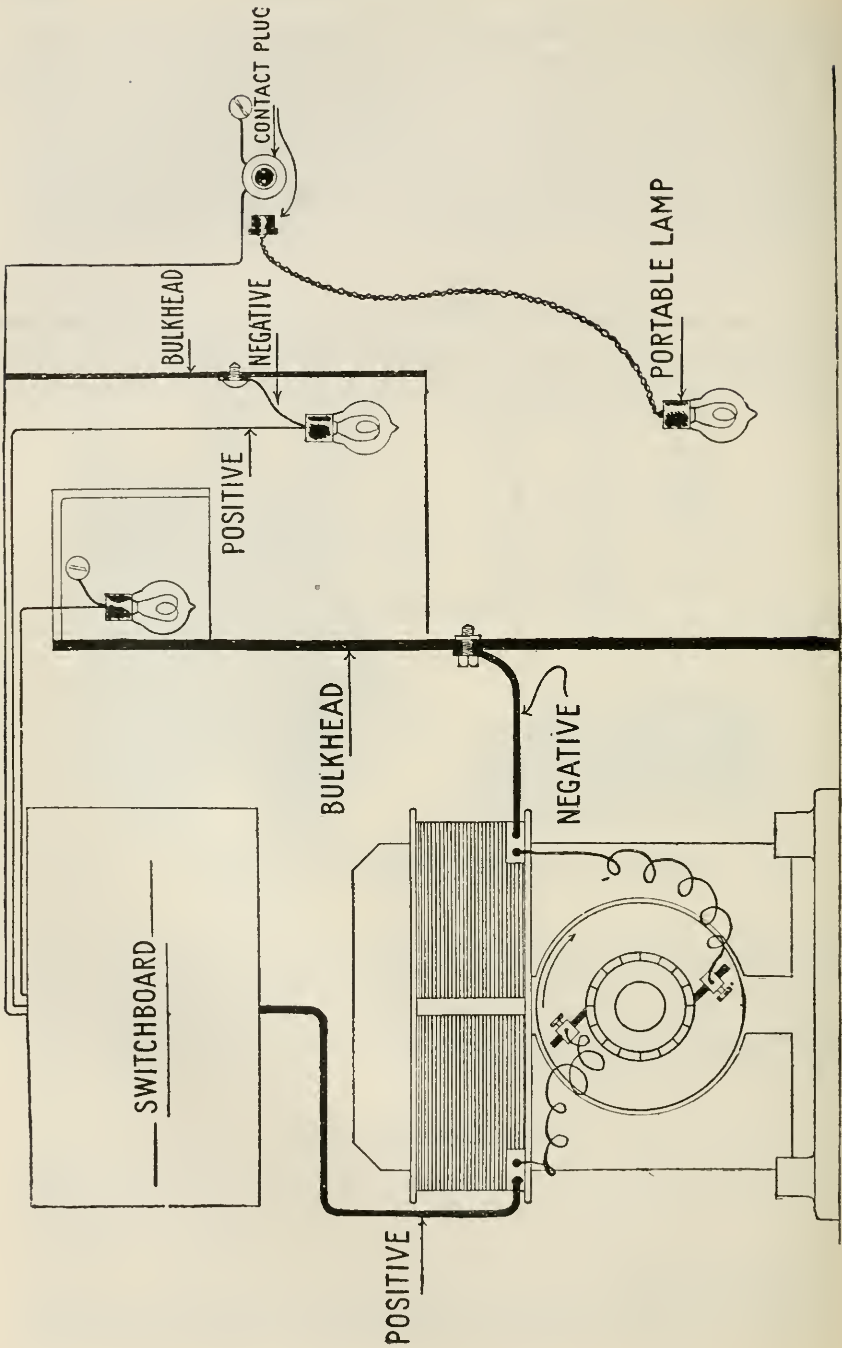
the voltage remains constant, but the number of amperes varies according to the number of lamps switched into the circuit.

NOTE.—When the quantity of current passing through a conductor is one coulcombe per second the strength of the current is said to be one ampere. This then is taken as the current strength.

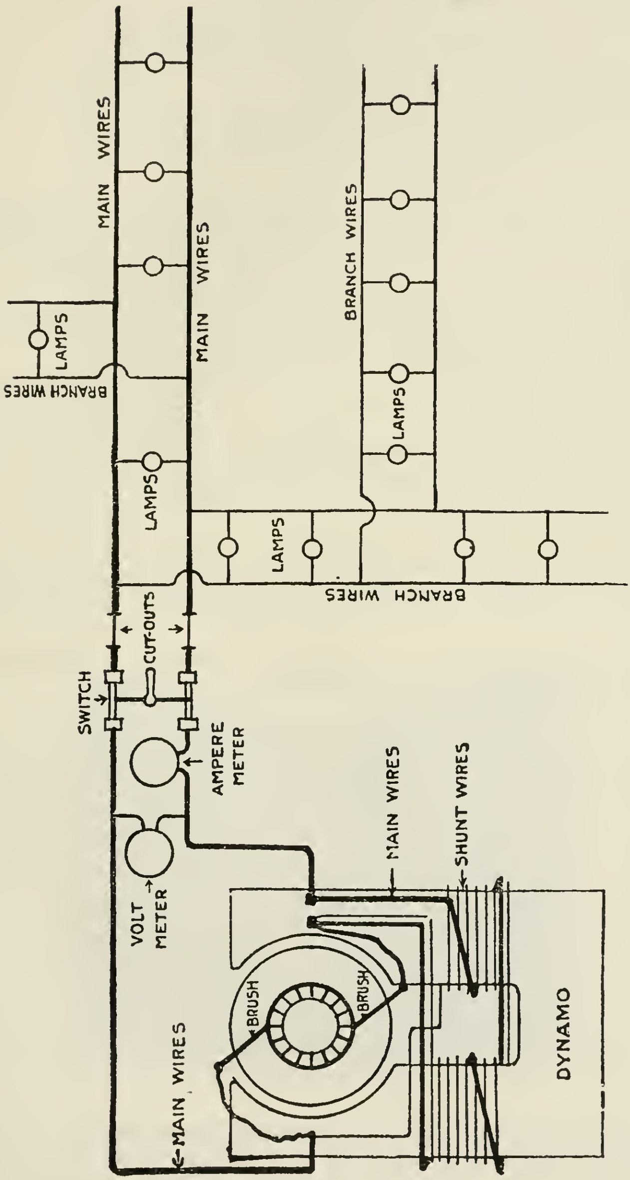
Single Wire System.

In this system of wiring the positive current is carried by a single cable or wire to the lamps, and the return from the lamps is effected by means of the metal of the ship, or, as it is called by electricians, the return is "earthed."

The sketch shows clearly how the lamp connections are made, the return wire being metallically connected to a stud screwed into



No. 25.—Single Wire System.



No. 26.—Double Wire System of Electric Lighting.

some part of the metal of the steamer. Observe that the dynamo negative cable is secured by a large stud to the bulkhead plate, the positive wire only going to the switchboard.

The advantages claimed for this arrangement are—

(1) Lower cost of installation.

(2) Less complication of wiring.

(3) Less trouble in locating faults in the circuit.

The disadvantages are—

(1) Greater danger of short circuits between lead and return.

(2) System only possesses half the insulation of the twin wire installation.

(3) Supposed cause of corrosion in condensers or other places due to galvanic action.

(4) More danger of lights going out suddenly in case of vessel grounding or in collision.

(5) Troubles experienced by rusting going on at the ship return lamp connections, owing to damp due to "sweating" of the plates, &c.

It may be stated that the majority of new installations are of the twin wire system, this being now considered the most reliable method.

NOTE.—Sea water acts very injuriously on the insulation of wires and cables, unless they are protected by metal piping or are "armoured."

Three-Wire System (see also page 649). — In some steamers two dynamos are run together on what is called the three-wire system. In this arrangement the positive wires of one dynamo are connected to the negative wires of the other dynamo, and the central or neutral wire acts as a common conductor for both.

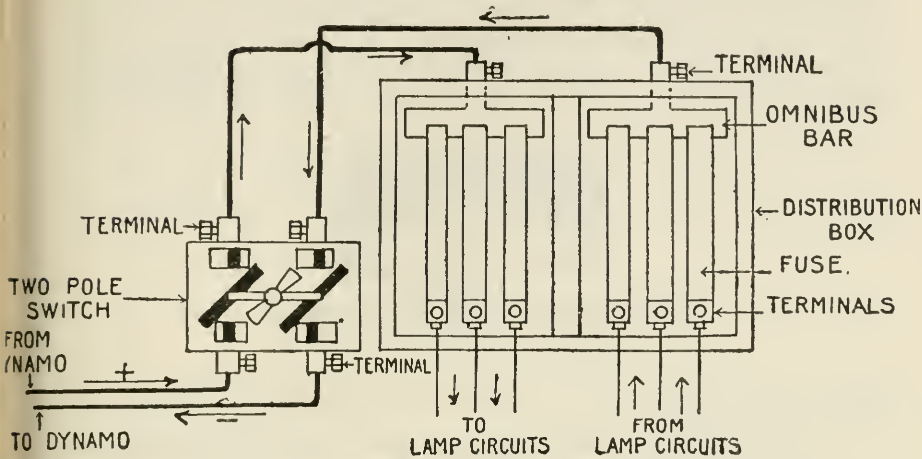
The voltage is usually 220, but as this is divided between the two, the working voltage for the circuits is only 110 volts.

The lamps are arranged so that the number of them is equally divided between the two outside wires and the central one, to balance each other and divide the current. If the same number of lamps are run on each side, the middle wire will carry no current, but should more lamps be switched in on one side than on the other, the difference of current resulting will then be carried by the central wire.

It should be noted that as the central wire has only to carry the excess or difference between the two outside wires, it can be made of less section, and is therefore of smaller size than the others.

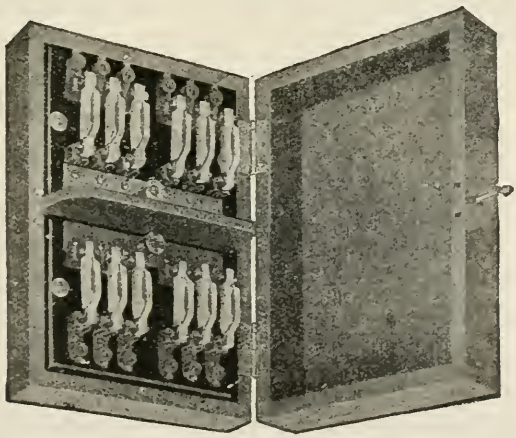
In the three-wire system the main switches and cut-outs are necessarily of the "three-pole" type.

Distribution Boxes.—The distribution boxes contain two brass bars, positive and negative (called "omnibus bars"), to which each wire,



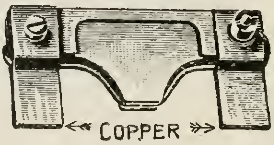
No. 27.—Distribution Box and Connections.

positive and negative, from the switchboard is connected by screws or terminals. From the two bars fuses are led to connect to a



No. 28.—Distribution Board in Cast-Iron Case for Motor Circuits.

number of smaller wires, each intended to light a certain section of the steamer. These fuses are held in earthenware holders with copper contact pieces, and as they can be withdrawn at will by hand, may therefore be made to act as "circuit-breakers" if required. Suppose the main supply wire from the switchboard to the distribution box to carry, say, 15 amperes of current, then if the box has three smaller wires led away to as many lamp circuits, each circuit wire will have about 5 amperes of current to carry. In certain cases pairs of wires from one box are led to other smaller boxes, where still



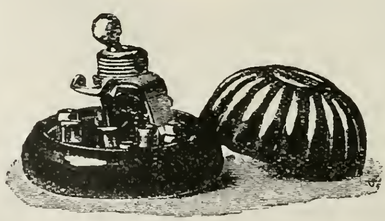
No. 29.—Distribution Box Fuse.

smaller wires, carrying less current in proportion, are connected up to the lamp circuits.

A hand switch is usually fitted to the wires at the distribution box, so that all current can be shut off from the set of wires leading from it (Sketch No. 27).

Lamp Switches.—The variety of design in lamp switches is infinite, and it will be sufficient to describe one or two of those most in use.

All switches, be it noted, whether large or small, fulfil the same object, that is, to break the current or circuit when "off" and to connect the circuit when "on."



No. 30.—Tumler or "Link" Switch.

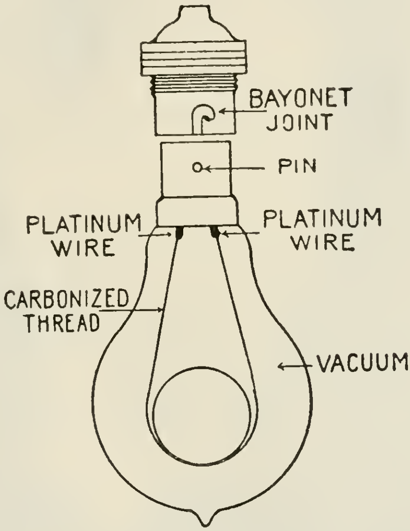
The switch most commonly met with is undoubtedly that known as the "tumler" or link type, and will be found in nearly all cabin and state-room fittings. When the small round knob of the "tumler" switch is pushed over, two small copper contact pieces are pressed against the two corresponding wire terminals, and the current flows across the bridge so formed to the lamp. A small spring acts on the lever and keeps the contact strips in position.

Sometimes a small cut-out is also fitted inside the switch box, and fits in between two screw terminals on the main wire leading to the switch. Should an excess of current come on the wire, the fuse would burn out or melt before the excess of current reached the lamp in connection with the switch.

Incandescent Lamps.

Incandescent carbon filament lamps are made of any candle-power from 8 to 500. They are usually marked as so many candle-power, as, for example, 16 c.p. or 32 c.p., &c.

The lamp consists of a brass neck piece or collar filled with cement, into which are sealed two platinum wires. The wires connect



No. 31.—Incandescent Lamp.

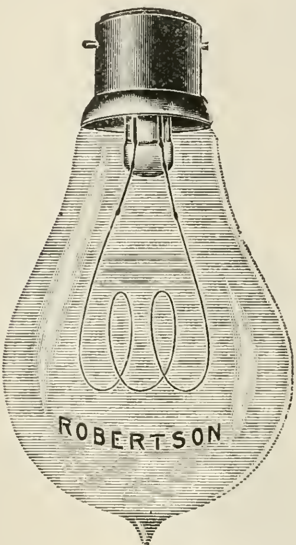
by contact pieces to the lamp wires on the outside, and to a carbonised cotton thread or filament inside the glass globe The carbonised cotton thread is made in the form of a single or double loop, so that the lighting effect may be intensified. In some cases double filaments are fitted with the same object. The glass globe is exhausted of air during manufacture, and the carbonised thread being thus in a vacuum, does not burn away when heated to a white heat. It should be noted that should air get into the globe, the thread would burn away in a few seconds, owing to the supply of oxygen allowing combustion to take place. The current in passing through the carbonised thread raises it to a white heat by the resistance offered to the current flow, and the "incandescence" resulting constitutes the light.

The "life" of an incandescent lamp varies a great deal. Some last 1000 hours, others less than this, and others again for indefinite periods.

NOTE.—The "filament" or "thread" referred to is carbonised by being heated in a vessel filled with the carbon gas of coal, the effect of which is to cause minute particles of pure carbon to deposit on the thread until it is completely covered with a fine skin of carbon.

NOTE 2.—Platinum is the only metal suitable for connecting the filament, owing to the fact that it expands at the same rate as glass when heated, and thus keeps the vacuum good.

Metallic filament lamps, however, are marked with number of watts, in Osram lamps about 1 watt = 1 c.p.



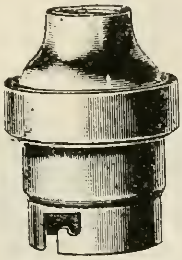
No. 32.

No. 33.

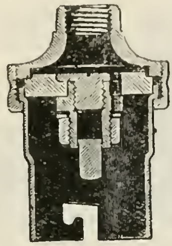
Osram metallic filament lamps are now rapidly becoming adopted by the leading steamship lines. These lamps have hitherto been very largely used for the lighting of buildings, offices, &c., but they were not used for ship lighting when they were first brought out, owing to the fact that they were rather fragile. They have now, however, been considerably strengthened, and are eminently adapted for ship work.

The great advantage of using Osram metallic filament lamps is that with an Osram lamp of 50/130 volts, it is possible to obtain 16 c.p. with a consumpt of 17 watts, compared with a 16 c.p. carbon filament lamp which consumes 64 watts. This readily shows the fact that there is a clear saving of 73 per cent. by using the

Osram lamps. This, of course, means that in many cases where more light is required in a vessel, it can be obtained by taking out the carbon filament lamps and installing Osram metallic filament lamps of a greater candle-power without increasing the size of the plant (which in all probability is fully loaded with carbon filament



No. 34.—Bayonet Joint.

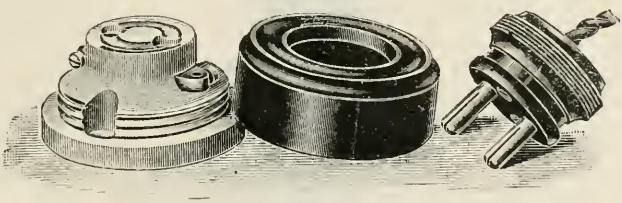


No. 35.—Section through Lamp-holder, showing Spring Contact.

lamps), and which would have to be increased if more carbon filament lamps were added.

Again, with the Osram metallic filament lamp, it is possible to obtain double the candle-power of the carbon filament lamp and still effect a saving of 50 per cent. in current.

There is no doubt whatever that in the near future metallic filament lamps will be universally adopted for ship lighting, which



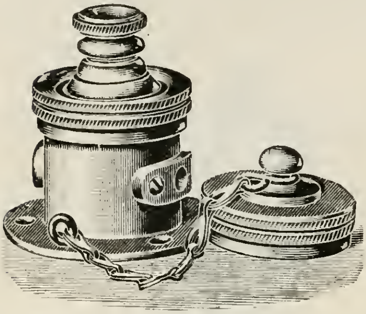
No. 36.—Wall Plug for use in Cabins, &c., for Fans, Cable Lamps, &c.

will mean that the installation can be put in at a considerably decreased cost, as the plant would be considerably smaller and the wiring throughout the vessel in proportion would be of a smaller size.

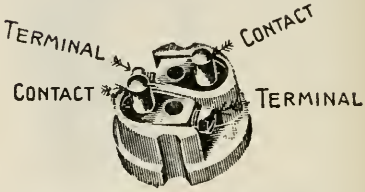
We illustrate a Robertson carbon filament and an Osram metallic filament lamp.

Lampholder.—The neck of the lamp connects to the lampholder by a bayonet joint. Two small brass contacts in the holder press down against the contact strips or plates of the lamp to which the platinum wires from the thread are attached.

The lamp wires are connected to the spring contacts of the

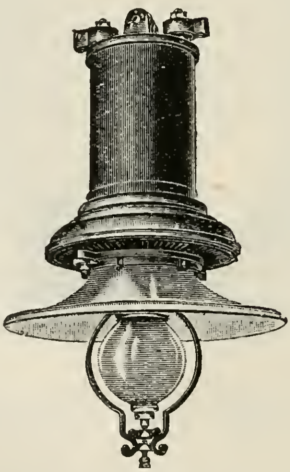


No. 37.—G.E.C. Water-tight Wall Plug, for use with Handlamps, Portables, &c.



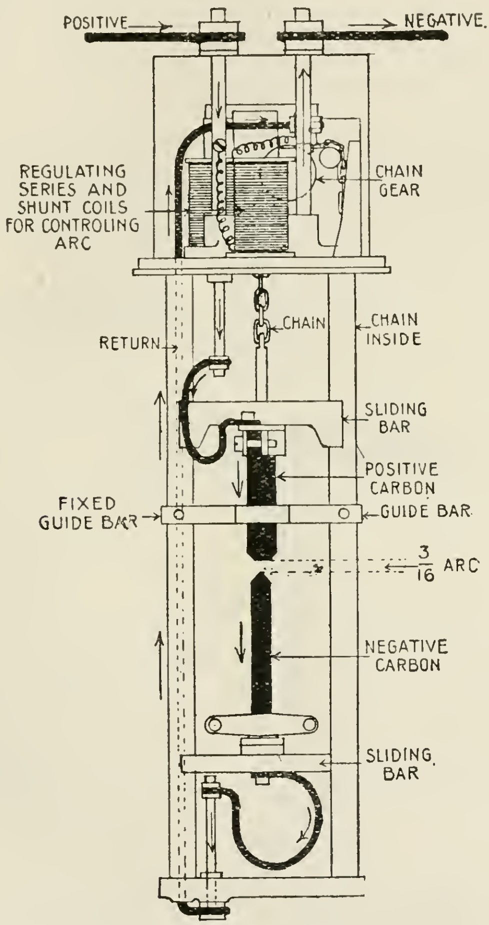
No. 38.—Lampholder Terminals and Spring Contacts.

lampholder by small screw terminals carefully insulated from each other. Sometimes a switch is supplied inside the lampholder.



No. 39.—G.E.C. "Angold" Arc Lamp, for Lighting of Decks and Holds.

Arc Lamps.—If a single wire carrying a current be cut and two carbon pencils (one to each end of the cut wire) inserted in the gap, the current will pass across from one carbon to the other, provided the space or "arc" between them is not too great. In crossing over the space, light is given out by the particles of carbon which pass from one carbon (the positive) to the other (the negative) becoming heated to a white heat. This is, roughly, the principle of the well-known arc lamp. It is important to note that before light can be obtained the two carbon pencils must first touch, and then be drawn



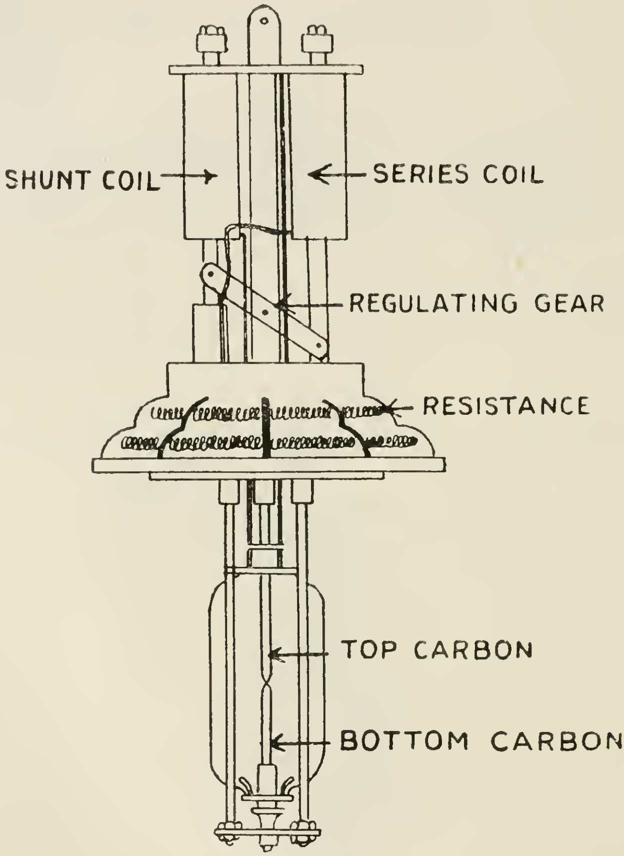
No. 40.—Arc Lamp with Case and Globe removed.

away to the required distance from each other: this is termed "striking the arc." The space between the carbons is usually from $\frac{1}{8}$ in. to $\frac{1}{4}$ in., depending on the amount of current supplying the lamp. The upper carbon is the positive one, from which the current passes to the lower or negative one. The mechanism of an arc lamp has to perform the following functions:—

1. On the current being switched on, to "strike the arc."

2. After the arc is struck, to maintain the carbons at the proper distance apart.

3. If two or more lamps are run in "series" (on the same wire), to allow the current to pass to the other lamps if anything goes wrong.



No. 41.—G.E.C. Type Arc Lamp.

The gear for regulating the arc varies a great deal in design, different makers having different methods. The general principle, however, is as follows :—

Two small bobbins wound with wire, one being of coarser (series) wire than the other (shunt), are arranged so as to form electro-magnets. When a current flows through the shunt wires, the magnetism resulting attracts a piece of metal or lever placed in connection with the small wheel and chain attached to the carbon-holders. The chain and wheel act so that as one carbon is raised the other is lowered, thus keeping the focus or arc always in the same place.

When the current is flowing from the positive carbon down to the negative one, and the space between them is properly adjusted, the coarse or series wire carries the current; but if the space becomes too great, then as the resistance to the current passing across is now increased, less passes through the series wire and more through the fine or shunt wire, and the magnetism resulting attracts the lever in connection with the chain gear, which being set in motion, draws the carbons together until the balance is restored. When the lamp is first switched on, the current momentarily passes through the shunt wires, and the effect of this is to draw quickly together the two carbons, thus striking the arc. After the connection is made in this way, most of the current then passes through the series coils and the carbons, weakening the shunt in proportion, so that the arc is correctly set. Nearly all patent arc lamps are worked on this system, called the "differential," owing to the difference in the series and shunt bobbins. Put briefly, then, when the carbon pencils are at the proper "arc," most of the current passes through the series coils; but if the arc lengthens owing to the consumption of the carbons, less current passes through the series coils and more through the shunt coils, and the shunt coils attracting a magnet, set in motion the clockwork gear which draws the carbons together again until the proper arc is established.

An automatic cut-out and substitutional resistance is usually provided in case the lamp fails to act.

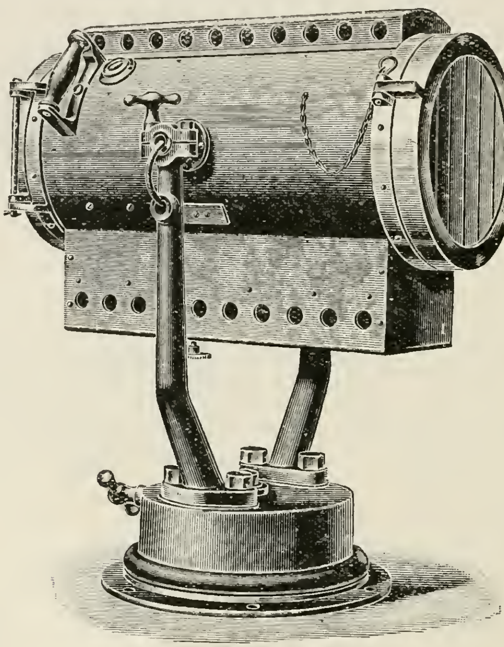
The upper or positive carbon burns away about twice as fast as the negative one, and becomes slightly hollowed at the lower end, whereas the negative carbon assumes in time a pointed or conical shape. The carbon pencils only last from six to ten hours, after which they require to be renewed, unless in the case of enclosed arc lamps, the carbons of which last for a much longer period.

Arc lamps are generally run at about 50 volts, and require from 8 to 10 amperes of current per lamp.

Projector.—The Suez Canal Regulations require each steamer passing through in the night time to be supplied with a strong projector or searchlight. The projector consists of a cylindrical casing hung on movable trunnions, and containing inside the necessarry mirror, lenses, carbons, and adjusting gear.

The regulation of the arc is in most cases obtained by the hand-feed arrangement, the carbon pencils being held in two brackets screwed on to a right and left hand threaded spindle. By means of small handwheels the arc can be focussed and adjusted as required. As in the case of other single arc lamps, a "resistance" is also employed to reduce the voltage, and obtain a steady light. The amount of current required for the projector is very high, as much as from 100 to 150 amperes being sometimes necessary, although the voltage may only be from 50 to 60.

The wire from the dynamo runs to a terminal box near the position required for the projector, and from the box the wire is led direct to the lamp. The positive terminal is marked with a + sign, and the negative with a – sign.

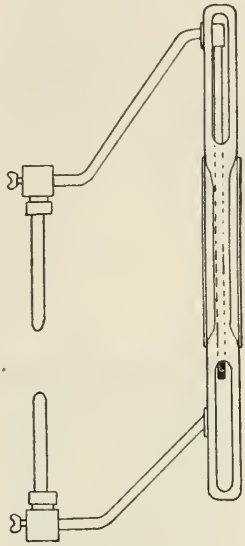


No. 42.—G.E.C. Projector.

A fuse and switch are often fitted in the terminal box for greater safety.

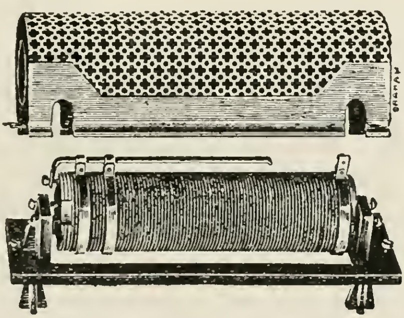
Description of G.E.C. Projector.

The case and pedestal is made of light sheet steel with copper protection guards and sight holes fitted with blue glass for examining the arc. The side standards, trunnions, lamp box, and back frame door, and frame in front working head, and all exposed parts are made of gun metal highly finished, polished, and lacquered. The vertical and horizontal movements are obtained by gearing worked by hand-wheels, but arranged so that, if desired, the gear can be thrown out and the projector left free to be moved by the handles fitted on the frame at the back. The hand feed lamp is of special construction and fitted with vertical focusing gear. The feed motion is worked by bevel wheels from the outside of the lamp box, the carbons being moved together by a left and right hand screw. The horizontal focusing gear



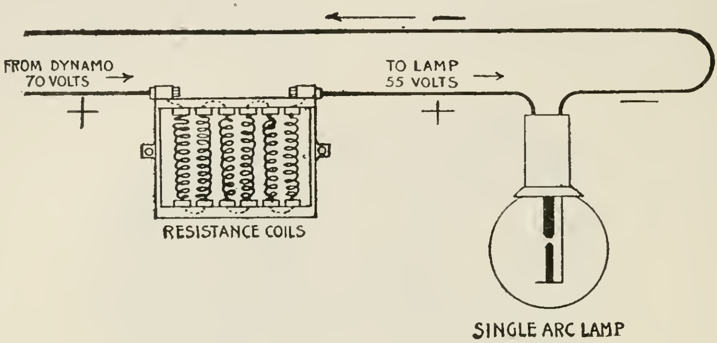
No. 43.—G.E.C. Type Projector.

is worked by a screw passing through the lamp box fitted with hand-wheel at side and end of box. The projector can also be fitted with automatic feeding gear if required. A resistance is placed in the circuit to reduce the voltage to suit arc voltage, and this also ensures steadier working.



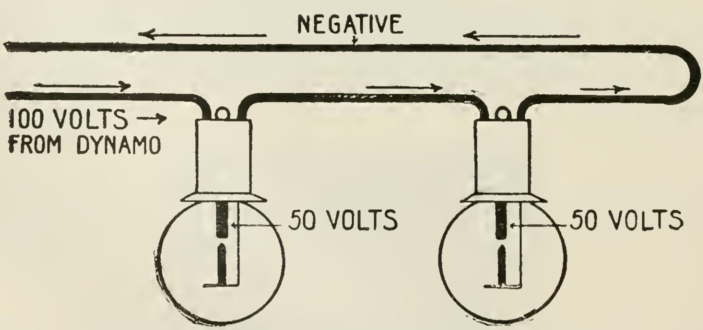
No. 44.—Resistance Coil.

Resistance Coils.—Single arc lamps are often fitted forward and aft, and for these the current is led from the switchboard by separate wires to a "resistance coil," where the voltage is reduced to suit that required by the lamp. After passing through the resistance coil the current enters the carbons of the lamp at the reduced voltage.



No. 45.—Arc Lamp and Resistance.

The resistance coil for single arc lamps consists of a metal box or case containing coils of fine platinoid wire, arranged either in vertical rows or wound on a cylinder. If two or more coils are fitted, they are connected "in series," that is, the end of one coil is joined to the end of the next, and so on. The current in passing through the coils is lowered in voltage, as the resistance of the platinoid wire is much more than that of copper wire.

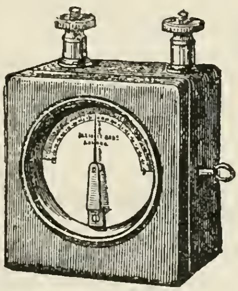


No. 46.—Two Arc Lamps "in Series."

As work is done by the wire resistance referred to, heating up of the coils and box ensues, and to allow for this the case should be made of fireproof material to prevent damage from overheating. As before stated, arc lamps are usually run at about 50 volts, and this being so, if the dynamo runs at 100 volts, as is sometimes the case, then two arc lamps can be run "in series," that is, on the same wire, each one receiving 50 volts. If the dynamo, however, only runs at, say, 70 volts, then the arc lamps must be run singly, and a resistance employed to dissipate about 15 volts, so that the lamps may each receive no more than 55 volts. This tends towards steadiness in the light.

Testing for Faults.—Testing can be done by means of a "detector" formed of a magnetic needle and galvanic battery, or by a small portable hand lamp with a length of wire connected to each of its terminals. When using the hand lamp the ends of the copper wires must be carefully stripped of insulation so that the copper is bared.

The detector can be used in most cases when the dynamo is stopped, but the lamp can only be used when the dynamo is running.



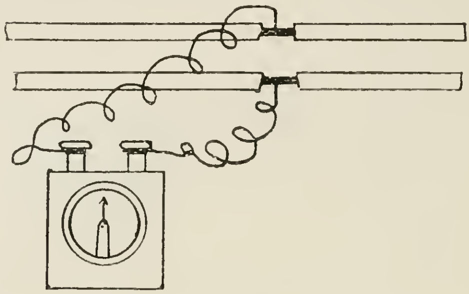
No. 47.—Detector.

The detector is supplied with a battery, so that a current will flow through it and deflect the needle whenever the positive and negative poles of the dynamo or wires to be tested are connected up to the terminals of the detector.

In the case of the portable lamp, a current must first be sent through the wires, &c., when the ends of the lamp wires are put in contact with the positive and negative connections under test, before the light will show in the lamp.

NOTE.—A "short circuit" is a connection (usually metallic) between any positive and negative part of the dynamo connections, or between any two of the wires. An "earth" is a metallic connection between one of the poles of the dynamo or wires to the metal of the ship's plates.

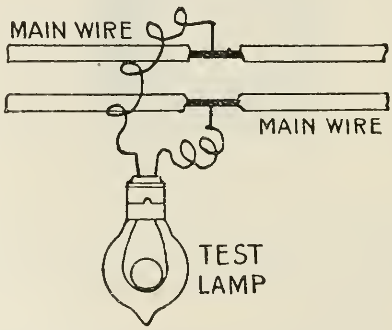
Break in Main Wires.—To discover the position of a break in a pair of wires, begin from the source of the current in question or distribution box from which the wires branch off, and baring the two wires at short distances, touch them both with the free ends of the wires connected to the detector. If the needle deflects, a current is passing at the point tested; but if after repeating this a few times the needle does not deflect at a point further on, it indicates that a break is situated somewhere between this point and the last place where the needle deflected.



No. 48.—Broken Wire Test by Detector.

The wires will therefore require to be carefully examined between the two places referred to for the location of the break.

The portable lamp will do equally well as the detector, only in

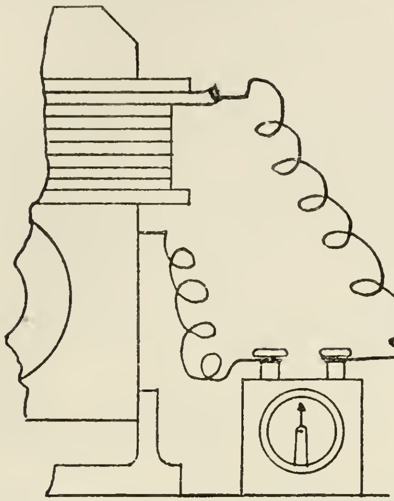


No. 49.—Broken Wire Test by Lamp.

this case the dynamo must be run to obtain a light in the lamp when the bared ends of the lamp wires are put in contact with the wires under test. If at a certain point no light shows in the lamp, it indicates a break in the current or circuit.

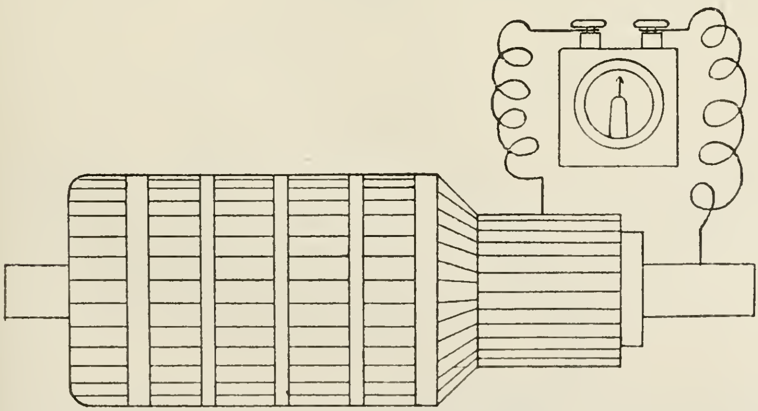
NOTE.—The switches of the circuit in question must be "on" when testing with the detector.

Leak in Magnet Coils.—To test if leakage is occurring between the magnet coils and magnet, connect one of the detector wires to the end of the coil to be tested, and after carefully cleaning and polishing up a small part of the metal work of the magnet, put the end of the other detector wire in close contact with it. If the needle deflects, it indicates a leak between that particular coil and the core of the magnet: if no deflection of the needle takes place, it proves the insulation to be intact. Each coil will require to be tested in turn.



No. 50.—Test for Leakage between Coils and Magnet.

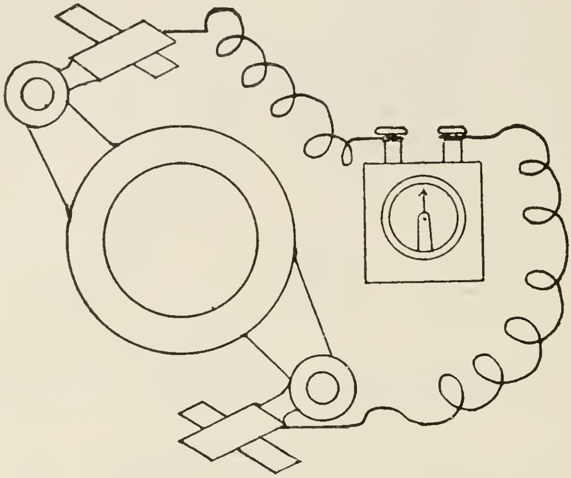
Leak between Armature Coils or Commutator and Armature Drum.—Take out the armature and support it on a pair of trestles, place one detector wire on the armature shaft or drum (either will do)



No. 51.—Test for Leakage between Armature Coils and Drum.

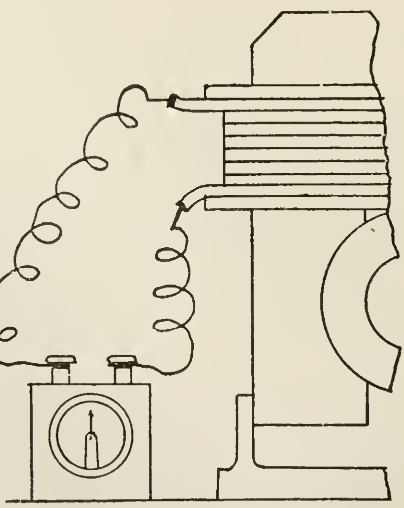
and with the other wire touch the commutator bars as the armature is slowly turned round. If the needle deflects it indicates a short circuit between the armature coils or commutator bars and the drum.

Short Circuit in Brush-holders.—Lift the brushes from the commutator and disconnect them from the cables leading to the dynamo terminals, then place one detector wire on one brush-holder, and the



No. 52.—Test for Short Circuit between Brush-holders.

other detector wire on the other brush-holder. If the needle deflects, a current is passing indicating a short circuit. Each part of the brush connections can be tested in the same manner.

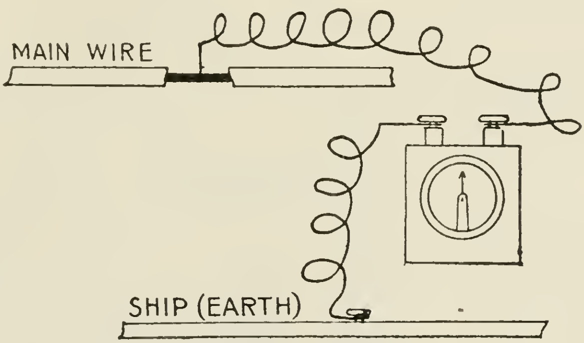


No. 53.—Test for Broken Wire (Continuity Test).

Test for Broken Wire.—Disconnect the wire to be tested so that the ends are free, and place one detector wire to each end. If the needle deflects, a current is passing, and the wire is not broken; but if the detector needle remains stationary, it indicates that the wire in question is broken, as the circuit is not complete.

Test for "Earth" Leakage.—With the main and lamp switches "on," connect one detector wire to the positive and negative wire of the dynamo in turn, and put the other detector wire in contact with the floor plates or ship's skin as the case may be. If a deflection of the needle occurs, it indicates that leakage to "earth" is taking place, that is, at some part of the circuit one of the wires is in bare contact with the metal of the ship, and the current is returning to the dynamo by that path.

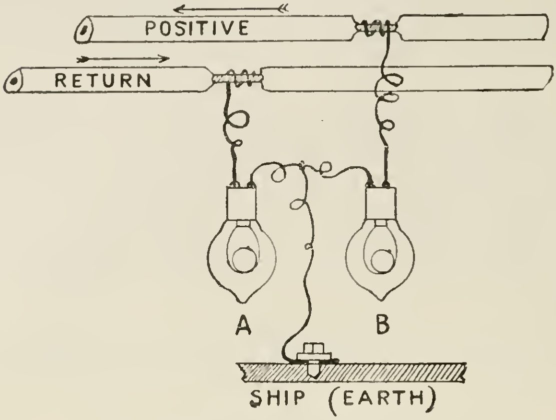
To locate the part of the circuit affected, switch off the main switches one by one till the needle comes back to its zero position,



No. 54.—Test for "Earth" Leakage.

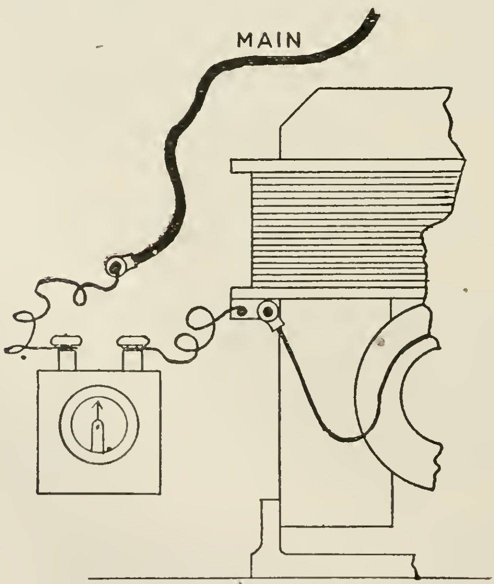
and the last switch opened will be that of the circuit affected. Now connect one of the detector wires to one of the "bus" bars or terminals of the distribution box of the circuit, and, as before, connect the other detector wire to the ship's metal. If the positive and negative fuse bridges in the box are now pulled out one by one, the needle will only move back to zero when the fuse bridge of the "earthed" wire is disconnected, and in this way the exact "earthed" wire can be located.

Another method of carrying out this test, should a galvanometer not be available, is to connect up a lamp, the lamp being of the same voltage as the dynamo. In this case it is necessary to have the dynamo running and to test both negative and positive sides of the leads, as the lamp only lights up when there is a fault on the opposite pole to that to which it is connected. For instance, should there be a fault on the positive lead to a lamp, when the test lamp is connected to the negative wire it will light up, but if connected to the positive it would remain black.



No. 55.—Earth Lamp Test.

"Earth" Lamp Test.—To test for an earth leakage arrange a pair of lamps as shown in the sketch, one connected to the positive lead,



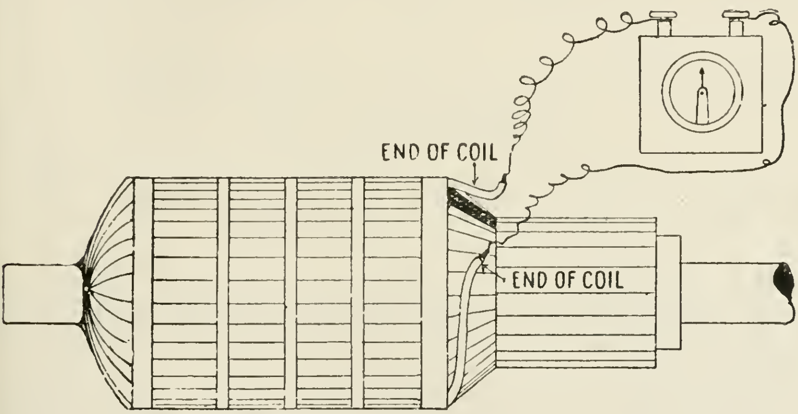
No. 56.—Short Circuit Test.

the other to the negative lead, and both connected to the ship metal by a cross wire.

With the dynamo running one of the lamps will burn brighter than the other if there is a leakage to earth, and the leak will be on the opposite wire to that of the bright lamp. For example, if lamp A burns brightest the leakage will be on the positive wire, but if lamp B burns brightest then the fault is on the negative or return wire.

To Test for Short Circuit in Main Wires.—Disconnect one of the main wires from the dynamo terminal, and insert between the wire and terminal the detector, as shown. Now switch off the lamps (not the main switches), and run the dynamo. If a deflection of the needle takes place it indicates a short circuit between the main wires, as with the lamp switched off no current should then be passing.

To Test for Broken Armature Coil.—This can only be accurately determined by the following method :—Disconnect both ends of each

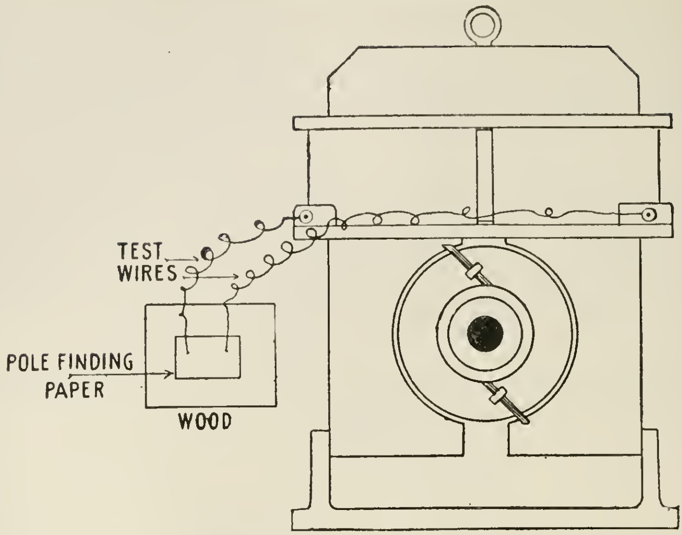


No. 57.—Test for Broken Armature Coil.

armature conductor from the commutator bars, and place one wire of the detector to each end; then if no deflection of the needle takes place it indicates a broken wire: a deflection proves the wire to be continuous or unbroken. Each armature coil must be tested separately.

To Test for Polarity of Dynamo.—It is often very convenient to know which is the positive and which the negative connections or wires of a dynamo, and these can be located as follows :—

Obtain a piece of "pole-finding" paper (procurable at any Electrical Supply Stores), and after moistening the paper place it on a piece of dry wood; now lead a suitable length of wire from each dynamo terminal, as shown in the sketch, and with the free ends of the wires touch the wetted "pole-finding" paper. A red coloured blot will then



No. 58.—Polarity Test.

appear on the paper at the wire connected to the *negative* terminal; the other will of course be the positive connection.

Hints on Running.—In the case of a new dynamo it is advisable to run the machine for a couple of hours or so with the brushes lifted from the commutator, as a test of the mechanical balance, lubrication, &c.

The armature, commutator, and field magnets should be kept absolutely free of dust, grit, oil, or moisture, as these allow of the formation of short circuits.

The commutator should be supplied with the *least amount* of lubrication possible, and that only of vaseline or mineral oil.

A commutator in good working condition presents a surface covered with an even bronze glaze or skin, and this should be maintained if at all possible.

The brushes should be placed at *exactly* opposite positions on the commutator circle (mathematically opposite).

It is safest to first get the speed up on the dynamo, and the proper voltage showing on the voltmeter, before switching in the lights.

Examine the armature conductors to see that the commutator ends of the wires are not bent and in contact with each other, as this will produce a short circuit.

Keep all small tools away from the dynamo, as the magnetic attraction may draw them into the field space and result in serious damage.

Brass or copper oil cans only should be used.

To test the armature balance, lift it out and place the shaft on two fine levelled knife edges; if the armature is then gently rolled from side to side it will come to rest with the heavy side down; this side should therefore be reduced in weight, or the other side increased in weight.

Short circuits in the armature coils show either by burning of the insulating material resulting in a strong smell, or merely by heating up of certain coils when felt by hand immediately after stopping the dynamo.

If a commutator develops an untrue surface or "flats," it should be turned up with a diamond-nosed tool, as this type of tool prevents the burring of the copper edges over the insulation.

Make sure that the binding terminals are screwed up and in metallic contact.

If the dynamo has become demagnetised it will refuse to generate current when the speed is up. To remedy this, either tap the field magnets with a light hammer, or, if this fails, reverse the brushes, that is, turn them round 180 degrees of the commutator circle (if a two-pole machine), so that they change places with each other, and run the machine for a short period with reversed current; this tends to restore the magnetic conditions: afterwards replace the brushes to their original positions.

Excessive rise of temperature in fields or armature indicates a short circuit between some of the wires.

A short circuit or earth leak may result in overloading the dynamo and produce sparking at the brushes.

In place of the ordinary galvanometer or detector a small bell and dry battery may be used for testing. When the circuit is completed by the wires from the bell terminals the bell will ring.

Whenever possible slow down and stop the dynamo before switching off the lights, as this prolongs the life of the incandescent lamps.

Before starting up the dynamo be sure that the lubrication is reliable and the oil cups filled up; also that the armature shaft is clear.

The brushes should not be lifted from the commutator while the dynamo is running, as this produces destructive sparking.

Sand-paper only should be used to polish up the commutator surface, and it should be applied by means of a board on which the sand-paper is pasted, the width of the board to be cut to the length of the commutator bars.

Hold the sand-paper board against the commutator, and have the armature shaft revolved by hand. This is best done with the armature lifted out and laid on a pair of wood trestles.

At intervals feel by hand the temperature of the magnet coils.

It is important to see that the engine is not started to run in the wrong direction, that is, against the brushes, as damage would result.

The brush position, when the machine is running without load, will not be suitable when the load is on, and the brushes must then be rocked forward to obtain a sparkless contact.

NOTE.—"Forward" means in the direction of rotation.

In polishing up the commutator in position, take care to lift up the brushes clear of the commutator surface.

The voltage of the dynamo varies in proportion to the speed of the machine.

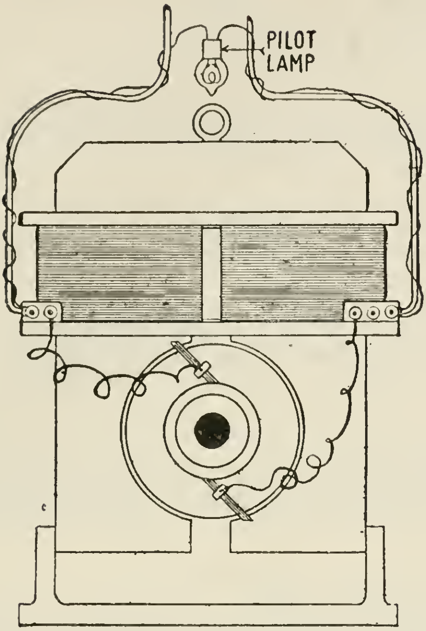
If a fuse blows or burns out it should be replaced by one of the same size, and not by a larger one as is sometimes done.

Copper gauze brushes should be kept well trimmed up and free of ragged edges.

Violent sparking at the commutator may be caused by a broken armature coil, or broken armature and commutator connection.

The "pilot" lamp serves as a guide to the voltage of the dynamo, as, being connected direct to the dynamo terminals, it indicates whether the machine is generating the required E.M.F. or not. If therefore a fault appears on a section of the lighting circuit and the pilot lamp of the dynamo is still burning brightly, it proves that the fault is not in the machine, but must be in the wiring or lamps. If the speed is too high this may show in the pilot lamp by possible burning out, and if too low, by the lamp only glowing instead of being at a white heat.

Engine-room "waste" should never be used on a dynamo, as the loose fibres are apt to detach and lodge between the commutator bars or armature coils, and ultimately bring about short circuits. A linen cloth is much to be preferred.



No. 59.—Pilot Lamp.

Become acquainted with the usual temperatures of the machine at different parts when running, so that any abnormal rise of temperature may be noticed at once, and the cause located.

When lifted out of the bearings the armature should be laid on a pair of wood trestles as mentioned elsewhere, or if laid on the floor should rest on sacking or some such soft material, as, being a delicate piece of work, it easily becomes damaged.

By careful adjustment of the brush rocker the best position of the brushes can be found, and this should give a practically sparkless contact.

See that the brushes have no side-play in the holders.

The point or toe of copper gauze brushes should be cut to an angle of about 40°.

Apply the necessary lubrication to the commutator either by the palm of the hand or by means of a piece of *linen* rag, and remember that a very small amount of mineral oil is sufficient.

If the armature is much out of balance it will probably injure both the commutator and the brushes.

The disadvantage of carbon brushes is a tendency to heat up if not accurately adjusted, as, for example, by excessive compression on the holder springs.

If the dynamo is situated in a part of the steamer where the temperature is high (say 100°), sparking will ensue at the brushes and commutator, owing to the increased resistance due to heating.

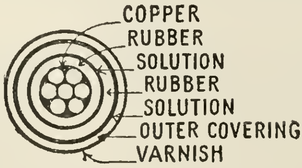
If the dynamo is placed near the condenser corrosion of the tubes may result, due, it may be supposed, to galvanic action. This has occurred in several cases which have come under the writer's observation.

When the brushes become ragged at the bearing edges, they can be quickly repaired by cutting off the rough parts with a knife run along a straight-edge, a cut also being taken off the corners at an angle.

Jointing of Wires.—In joint making the following materials are required, all of which are obtainable at Electrical Supply Stores :—

1. Solder sticks.
2. Resin.
3. Pure rubber strip or tape.
4. Rubber solution (Challerton's compound is one of the best).
5. Prepared tape.
6. Shellac varnish.
7. Emery cloth.
8. Fine copper wire (for binding).

The number of separate layers of rubber tape and solution depend on the thickness of the insulation originally on the wire, the heavier the insulation the greater the number of layers, and *vice versa.*



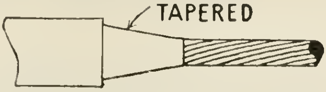
No. 60.—Section through Main Wire.

In lapping the rubber tape over the soldered part of the joint, only carry it up to the ends of the rubber on the wires, and not beyond this point.

Before applying the first lapping of rubber strip, file up all rough edges of solder or of wire so that a smooth-jointed surface is obtained.

In cutting the ends of wires previous to jointing, it is advisable to take off each layer of insulation in steps, as shown in the sketches. This allows of the covering of the jointed part fitting in better with the original layers of insulation.

Before applying the insulation layers, the rubber ends should be cut down to form a taper to the bare wire, as shown in the sketch. The lapping of the rubber strip over this ensures closeness of joint.



No. 61.—Tapering of Insulation.

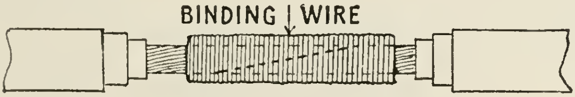
To cause adhesion the solution must be applied between each layer of rubber strip, and last of all a good coating of shellac varnish.

The following instructions as to jointing of wires are issued by Messrs Scott & Mountain, Electrical Engineers, Newcastle-on-Tyne :—

Main Cables.

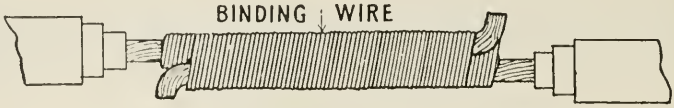
Preparing Ends.—Remove the two outside tapes for about 5 in. from each of the ends intended to be jointed. Bare the conductor of its covering of indiarubber and inside lapping of tape for about $1\frac{1}{2}$ in., and clean the wires with emery cloth.

Metal Joint.—Solder together the wires composing the strand for about 1 in., and scarf two ends with a fine file. Bring the two scarfed ends together and solder them. If this is carefully done, the conductor will be of uniform size. Over the joint bind spirally a fine copper wire, and solder the whole together. Resin, and not acid, must always be used for soldering.



No. 62.—Scarfed Joint.

Insulating Joint.—Taper each end of insulation with a sharp jointer's knife for $1\frac{1}{2}$ in. from the conductor to the outside of the indiarubber. Cover the metal joint with one lap of $\frac{1}{2}$-in. broad indiarubber, coated with cotton tape. Over the cotton tape lap spirally pure indiarubber strip (1 in. broad), stretching it at the same time, and building up the joint, by a series of coverings in alternate directions, to the same size as the indiarubber coating of the wire, or slightly larger, to allow for the thickness of binding wire. A very small portion of indiarubber solution should be applied over each coat, and sufficient time allowed for the spirit to evaporate before putting on another coat; this will cause the indiarubber strips to unite together.



No. 63.—Joints for Heavy Wires.

Outer Protection.—Two coverings of prepared tapes ($1\frac{1}{2}$ in. broad) are to be laid on in opposite directions, with strong shellac varnish between them, and then, outside, another covering of waterproof tape, and finally varnished over all.

Branch Wires.

Preparing Ends.—Remove the braiding tape and indiarubber for about 4 in. from each end intended to be jointed. Unlap the cotton serving next the conductor for about $1\frac{1}{2}$ in. (do not cut it off).

Metal Joint.—Thoroughly clean the ends of the wire with fine emery cloth, and scarf them with a fine file. Bring the two scarfed ends together and solder them. If this is carefully done the conductor will be of uniform size. Over the joint bind spirally a fine copper wire, and solder the whole together. Resin, and not acid, must always be used for soldering.

Insulating Joint.—Cover the metal joint evenly and as thinly as possible with the cotton which had been previously unwound from the ends. Over the cotton covering lap spirally pure indiarubber tape ($\frac{1}{2}$ in. broad), stretching it at the same time, and building up the joint by a series of coverings in alternate directions, to the same size as the indiarubber covering of the wire, or slightly larger, to allow for the thickness of binding wire. A very small portion of indiarubber

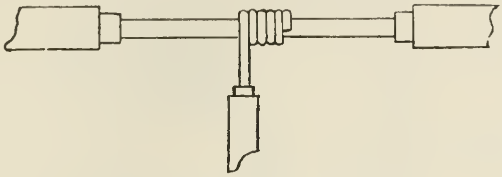
No. 64.—Joint for Small Lamp Wires.

solution should be applied over each coat, and sufficient time allowed for the spirit to evaporate before putting on another coat; this will cause the indiarubber to unite together.

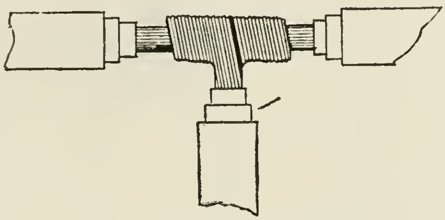
Outer Protection.—Two coverings of felt tape ($\frac{1}{2}$ in. broad) are to be laid on, in opposite directions, with strong shellac varnish between them, and finally varnished over all.

"T" Joints.

Preparing Ends.—Remove the two outside tapes for about 5 in. from the main lead. Bare the conductor of its covering of indiarubber and



No. 65.—Small "T" Joint.



No. 66.—"T" Joint

inside lapping of tape $1\frac{1}{2}$ in. Remove the braiding and tape for 6 in. from the end of the wire intended to be jointed to the main lead. The two rubber coverings and cotton serving are then to be unlapped for 3 in. and the rubber cut off. Thoroughly clean the strand, and also the solid wire with fine emery cloth.

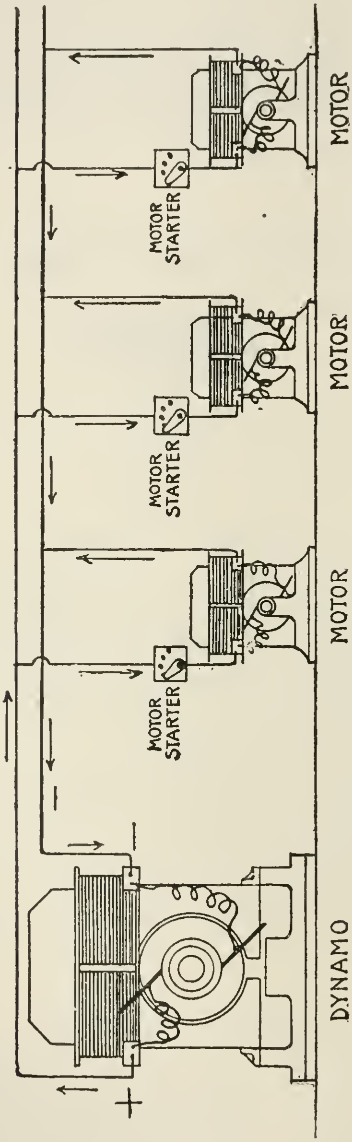
Metal Joint.—Solder the wires composing the strand together, take two or three turns of the solid wire round the main conductor, and back round itself for three or more turns, and solder only at the top of the T. Resin, and not acid, must always be used for soldering.

NOTE.—In joint making care must be taken to keep the hands, tools, and materials clean and dry.

Electric Motors.

The construction of a motor is similar to that of a dynamo—all the various parts, such as armature, magnets, commutator, brushes, &c., corresponding to the latter; but in the case of shunt motors a difference exists in the wire connections, which will be explained later.

As will readily be understood, a motor receives current from a dynamo, and the action is reversed, that is, in place of mechanical power generating electro-motive force, electro-motive force generates



No. 67.—Dynamo and Motors in Connection.

mechanical power. Put simply, a motor is a dynamo reversed in its action. Sketch No. 67 illustrates the principle of a dynamo with motors in connection.

In numbers of motors in use at present, the various parts are so fitted as to occupy very little space and to allow of the working parts of the motor being covered in or enclosed. This protects the delicate parts of the motor such as commutator, brushes, &c., from dirt and grit, and reduces the chances of breakdown occasioned by short circuits formed possibly by the dirt and grit in question. In this type of enclosed motor, four poles are often used instead of two, as being more suitable, as the four-pole arrangement lends itself better to the circular shape of the motor.

As stated before, in shunt motors an extra wire is fitted. The wire referred to is employed in the starting and stopping of the motor, and is connected between the field magnets and the supply wire through a "starting resistance" or switch.

On switching on the current at the starter it first enters the field magnet coils, and after freely exciting them, is then admitted gradually to the brushes and armature coils. The object of this is to prevent what may be called "racing" of the motor when the current is first turned on, as by first passing the current into the magnet coils the magnetic field is strengthened, and the tendency of the armature to run off at a high speed is checked by the magnetism developed opposing its too rapid rotation. This prevents mechanical shock, and possible damage to the working parts; it also checks undue variation in the current passing through the wires and supplying other motors or lamps in connection, and which might otherwise be affected.

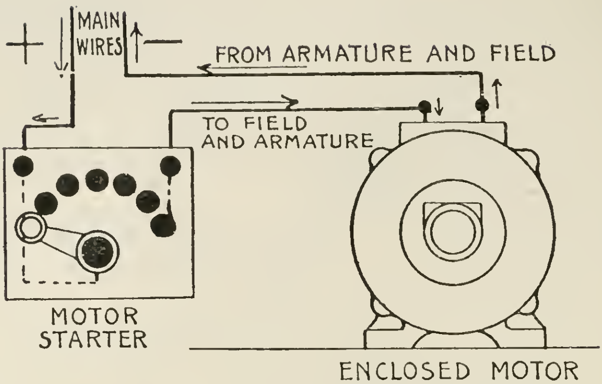
In a series wound motor the whole of the current entering the armature first passes through the field magnets, and gives the necessary strength to the field.

In a shunt wound motor only part of the current passes through the magnet coils by the fine shunt wires which are branched off from the main or supply wire.

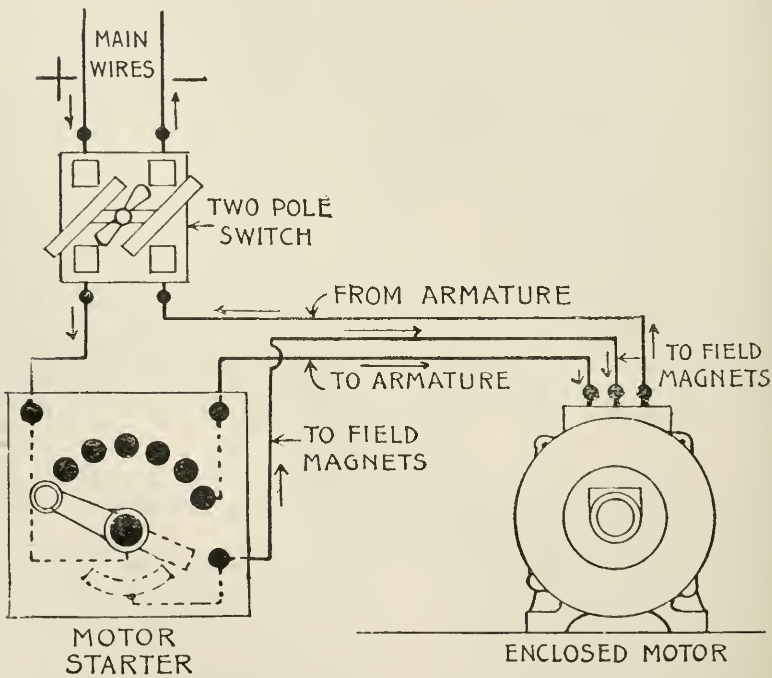
This being the case, it will be obvious that with a shunt motor the starting resistance must first freely excite the field magnets before the current enters the armature, otherwise by suddenly switching on the current the machine may be seriously damaged. It is also important that the current be only admitted to the armature by small degrees, and this is arranged for by fitting a "starting resistance" between the motor and supply wire.

Motor Starters.—A "starting resistance" consists of a box containing a number of platinoid wire coils connected together in series and to insulated earthenware bases.

Each coil has a brass or copper contact piece, and the hand lever, which is in connection at one end with the supply wire and to the contact stops at the other, can be moved over the coils in succession, so that at first all the coils are in series; but as the handle moves over each stop in rotation, one less coil is included in the circuit, and the resistance decreased in proportion. When the handle passes the last contact stop, all the resistances are cut out, and the full current

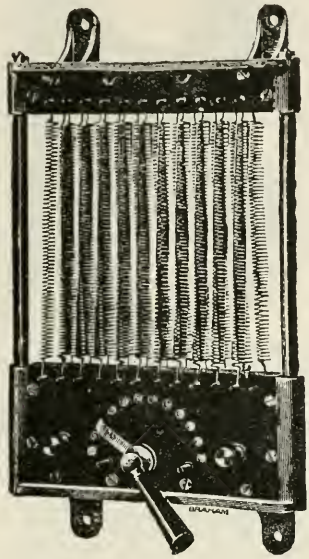


No. 68.—Motor Starter Connections ("Series Wound")



No. 69.—Motor Starter Connections ("Shunt Wound").

is then passing direct from the supply wire to the armature. The starting handle should be moved slowly over the contacts, and allowed



No. 70.—"Resistance" Regulator.

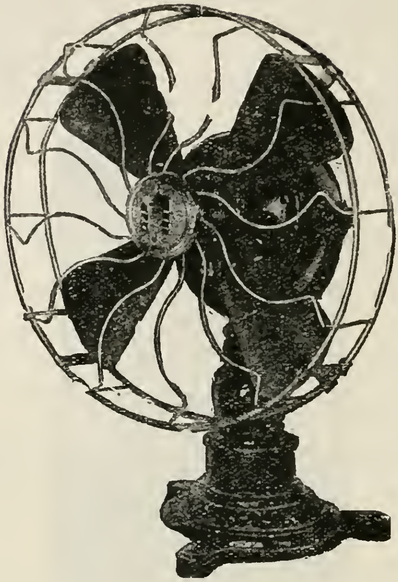
to press on each for a few seconds, as the speed of the motor gradually increases.

In stopping a motor the same precautions must be used, that is, the current must be switched off gradually by moving the resistance handle from stop to stop with an interval for each, so that each coil is inserted in rotation until the lever is in the "off" position.

The speed of the motor can also be regulated by the insertion of more or less of the resistance coils into the supply circuit by means of the handle and contact stops.

Some motor starters have small electro-magnets or bobbins arranged so that the lever is held in the "on" position so long as the proper amount of current is passing, but should anything happen to destroy the balance by excess or loss of current, the lever automatically flies over to the "off" position and cuts off the current altogether.

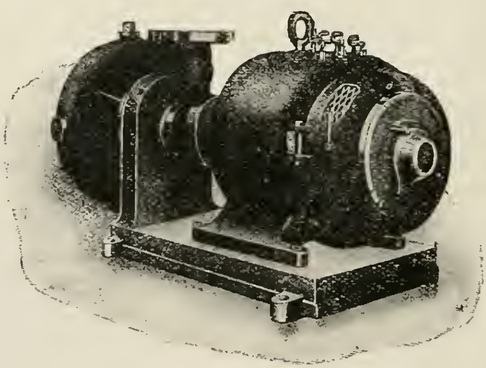
Small fan motors, as used for state-rooms, &c., are usually of the series wound type; and larger fans, for ventilation or induced draught purposes, of the shunt wound type.



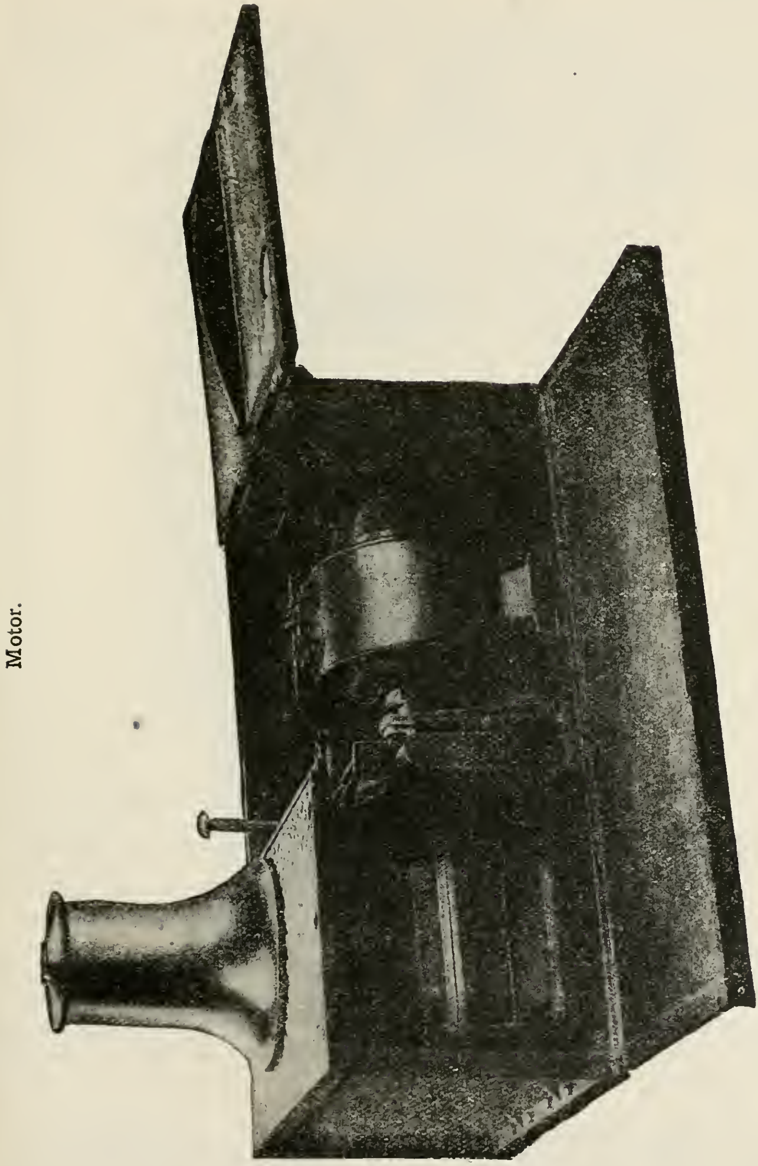
No. 71.—Type of G.E.C. "Freezor" Cabin Fan for Table or as Bracket Fan. These Fans are fitted with Three-speed Regulators in Base.

G.E.C. "Witton" Motor Driving Capstan.

One of the latest electrically driven appliances is the capstan A motor direct coupled gives complete and instantaneous control by



No. 72.—"Witton" Motor direct coupled to Centrifugal Pump.



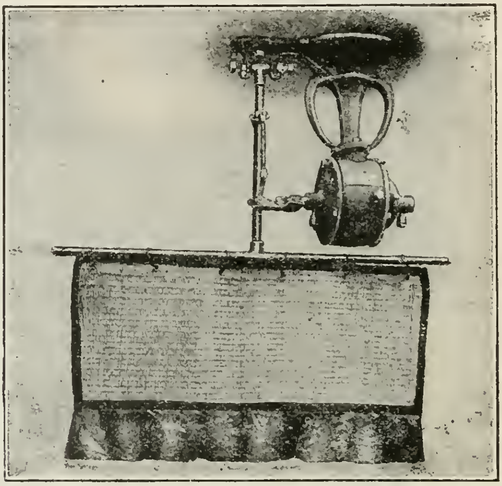
No. 73.—G.E.C. "Witton" Motor Driving Capstan.

means of a starting switch placed in a convenient position, and operated by foot only.

Pull, 4480 lbs.; speed, 100 ft. per minute; motor, series wound.

Largely used on board steamships, shipbuilding yards, and dockyards.

Circulating water for condensing, or washing down decks, and fire purposes.

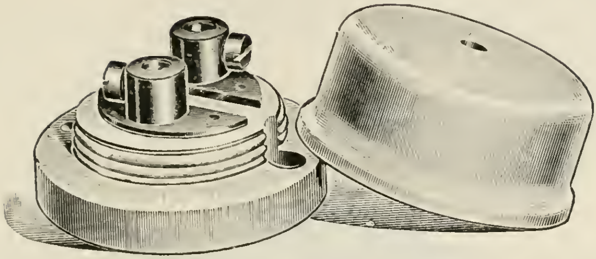


No. 74.—Electric Punkah, G.E.C. "Bandy."

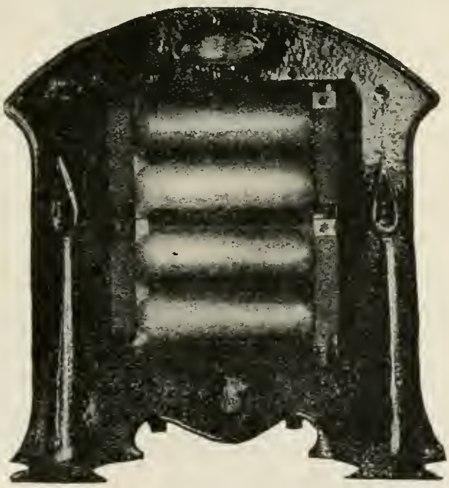
For Saloons, Cabins, &c. Installed on board many of the large liners on the Eastern Routes.

The only electrically operated punkah giving the "flick"—the distinctive feature of punkahs as compared with other cooling devices. The G.E.C. Bandy punkah effects a saving of 50 to 70 per cent. in current, when compared with an ordinary ceiling fan. The consumpt per one hour is approximately:—

Of the 2 ft. 6 in. size - - - -	27 watts.
,, 3 ft. 6 in. ,, - - - -	34 ,,
,, 4 ft. 6 in. ,, - - - -	38 ,,
An ordinary ceiling fan - - -	100 ,,



No. 75.—Extension Box.



No. 76.—Luminous Electric Glow Radiators.

For Heating State Rooms, Music Rooms, &c.

This horizontal type of glow lamp Radiator is a great improvement over the vertical type, specially for ships' use where vibration and jarring occur. The lamps are made with the poles at opposite ends, and one zig-zag filament runs through the lamp supported in the middle. Each lamp is rated at 250 watts.

Advantages of Zig-Zig Type Radiators.

Lamps firmly held by clip at each end.

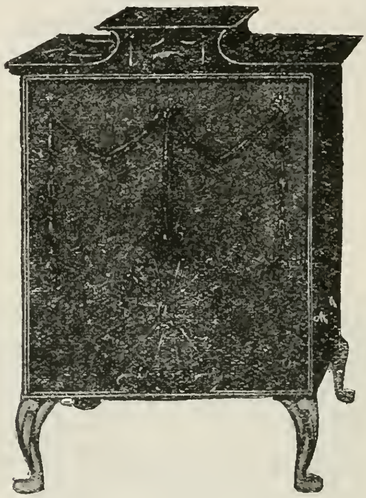
Opposite poles at opposite ends of lamp.

Increased contact surface.

Longer life of lamps.

The "Zig-Zag" filament **increases** radiation of heat.

Horizontal arrangement of lamps gives more pleasing effect.



No. 77.—Elevector.

"Archer" System of Heating by Convection, for State Rooms, Dining Saloons, &c.

The Elevector is an air warmer. The cases are supplied in various metals and finishes, and all types have interchangeable heaters or elements of 500 watts each. 1 watt for each cubic foot of space to be heated should be allowed for.

Electrical Notes.

By "**Potential**" is meant the difference of electrical tension existing between the positive and negative leads.

A **Volt** is the measure of electrical pressure or E.M.F. (Electro-Motive Force).

NOTE.—A Volt is the E.M.F. required to give one Ampere of current against one Ohm resistance.

An **Ampere** is the measure of electrical current, and is taken as the standard flow of electricity in a wire per second.

An **Ohm** is the measure of electrical resistance, and is about equal to that of one mile of copper wire $\frac{1}{4}$ in. in diameter.

Volts × Amperes = Watts.

746 watts are equal to 1 **Electrical Horse-Power.**

$$\text{Therefore,} \quad \frac{\text{Volts} \times \text{Amperes}}{746} = \text{E.H.P.}$$

E.H.P. compared to I.H.P.

It should be noted that the Electrical Horse-Power refers to *one second* of time as the ampere flow is measured for that period, whereas the I.H.P. of steam refers to one minute of time.

Therefore, $33000 \div 60 = 550$ foot-pounds per second,
and, 746 watts $= 550$ foot-pounds.
So that, $746 \div 550 = 1{\cdot}35$ watts per foot-pound.

As work and heat are equivalent, then it can be proved that to produce the same heating effect (energy), 1·35 watts are equal to 1 foot-pound.

E.H.P. per minute $= 746 \times 60 = 44760$ watts per min.

The **size of a wire** depends on the amount of current it has to carry; in other words, on the number of amperes.

The **insulation of a wire** depends more on the number of volts carried by the wire.

The Board of Trade limit is 1000 amperes per square inch of wire section.

Fuses or **cut-outs** are constructed to melt when the current becomes double the working current.

Insulating materials are composed of indiarubber, tape, varnish, vulcanised fibre, glass, cotton, earthenware, &c.

Arc lamps are usually run at from 45 to 55 volts.

Arc lamps require from 8 to 12 amperes of current.

Projector arc lamps require from 80 to 150 amperes of current.

A 16 candle-power incandescent carbon filament lamp requires about 60 watts.

A 16 candle-power incandescent lamp run at 75 volts requires ·8 of an ampere, because 60 watts $\div$ 75 volts $=$ ·8 of an ampere.

An "Osram" 16 candle-power incandescent metallic filament lamp only requires about 20 watts, as the resistance of the filament is higher and requires less current to produce the same heat.

1000 watts are equal to **1 kilowatt.**

The **positive** wire or terminal is often marked thus +, and painted red.

The **negative** wire or terminal is often marked thus –, and painted black.

Fuses in earthenware cases are placed at different parts of the wire circuits to act, if required, as automatic circuit-breakers.

Dynamos for ship-lighting usually develop from 65 volts to 100 volts.

"**In series**" means in continuation.

"**In shunt**" means branched off.

Continuous current dynamos are generally employed for lighting purposes, and alternating current dynamos for power stations.

Transformers are used in power stations to reduce the current from a high to a low voltage without serious loss.

100 amperes at 2000 volts will give 400 amperes at 500 volts if a transformer is used, because

$$\frac{\overset{\text{Amperes. Volts.}}{100 \times 2000}}{\underset{\text{volts}}{500}} = 400 \text{ amperes.}$$

NOTE.—This neglects loss of efficiency in transformer.

Accumulators are used for the storage of electricity, and consist of a number of galvanic cells joined in series. The cells are charged by the current from a dynamo, which decomposes the acid bath of the cells and reverses the chemical conditions. After charging, the electricity so stored up may be released and employed to act on an external circuit if suitable wiring is arranged, as the chemical relation of the plates causes a return to their original condition. Accumulators are often employed on yachts, where a small number of lamps may be required during the night, and in cases where the dynamo is not kept running constantly.

The quantity of current flowing past a one ampere section of wire in one second is called a "**coulomb.**"

Electricity is one form of "energy" or "force."

Induction is the magnetic or electrical effect produced on surrounding bodies or substances by an electric current.

The principle of induction is employed in transformers, where a current of high voltage in a set of fine wires is made to induce currents of a lower voltage in a set of coarser wires.

One "**megohm**" is equal to 1,000,000 ohms.

All substances offer more or less resistance to the flow of an electric current: those having least resistance are employed as "**conductors**," and those having most resistance are employed as "**insulators**."

RULES—

To find the **Current strength** in Amperes passing through an electrical circuit.

RULE—

$$\frac{\text{Volts}}{\text{Ohms Resistance}} = \text{Amperes.}$$

$$\text{and,} \qquad \frac{\text{Volts}}{\text{Amperes}} = \text{Ohms,}$$

$$\text{or,} \qquad \text{Amperes} \times \text{Ohms} = \text{Volts.}$$

EXAMPLE 1.—The voltage is 100, and the resistance of a 16 candle-power lamp 220 ohms. Find the required current in amperes.

$$\text{Then,} \qquad \text{Amperes} = \frac{\text{Volts}}{\text{Ohms}} = \frac{100}{220} = \cdot 45 \text{ Ampere}$$

EXAMPLE 2.—Find the resistance in ohms if the voltage is 100 and the amperes 300.

$$\text{Then,} \qquad \text{Ohms} = \frac{\text{Volts}}{\text{Amperes}} = \frac{100}{300} = \cdot 33 \text{ Ohm.}$$

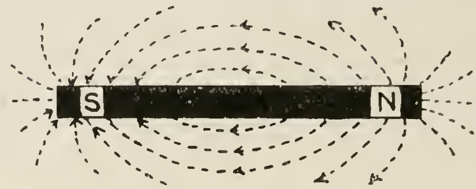
EXAMPLE 3.—The output in amperes is 250, and the resistance ·4 ohm. Find the required voltage.

$$\text{Then,} \qquad \text{Volts} = \text{Amperes} \times \text{Ohms} = 250 \times \cdot 4 = 100 \text{ Volts.}$$

EXAMPLE 4.—Find the number of watts required for a lamp taking ·6 of an ampere at 100 volts.

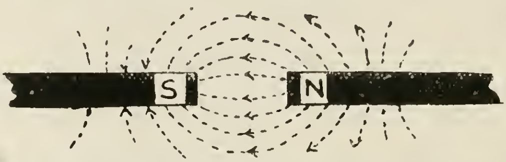
$$\text{Then,} \qquad \text{Watts} = \text{Volts} \times \text{Amperes} = 100 \times \cdot 6 = 60 \text{ Watts.}$$

GENERAL ELECTRICAL NOTES AND SKETCHES.



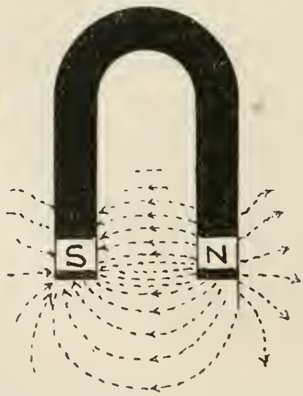
No. 78.—Lines of Force in a Bar Magnet.

Lines of force flow from N. to S. poles as shown by the arrows and dotted lines.



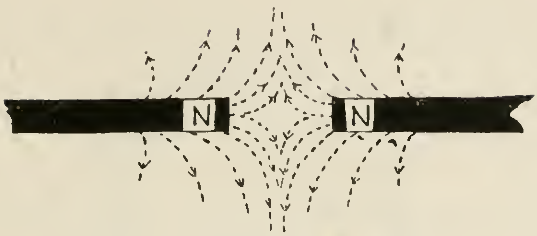
No. 79.—Lines of Force Between Two Bar Magnets.

Notice that when the N. pole of one magnet and the S. pole of another are brought together lines of force pass out from the N. pole to that of the other S. pole, that is, "unlike poles attract."



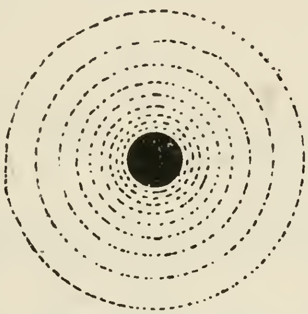
No. 80.—Horse Shoe Magnet.

As in the case of the bar magnet, lines of force pass over from the N. pole to the S. pole, the gap between being known as the "field space," or "magnetic field." A two-pole dynamo is similar to the above, the armature revolving in the space between the two magnet poles.



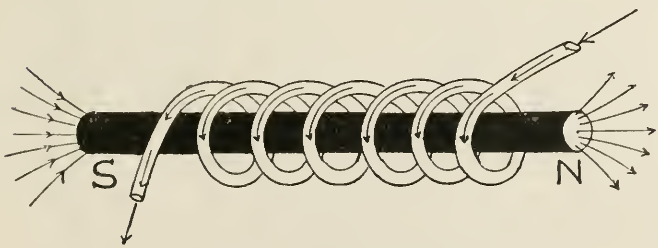
No. 81.—Opposed Magnetism.

Two bar magnets placed as shown, with like poles to like, produce repulsion of the lines of force, as, "like poles repel" and "unl ke poles attract."



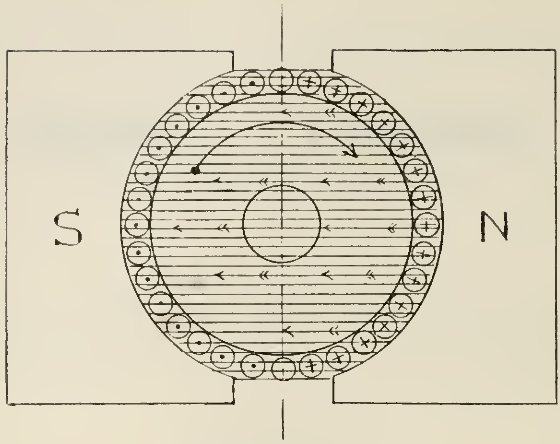
No. 82.—Lines of Force Surrounding a Conductor.

A live wire is encircled by lines of force as indicated by the dotted circles. From these are obtained the induction effects, which develop between adjacent coils under certain electrical conditions.



No. 83.—Electro-Magnet.

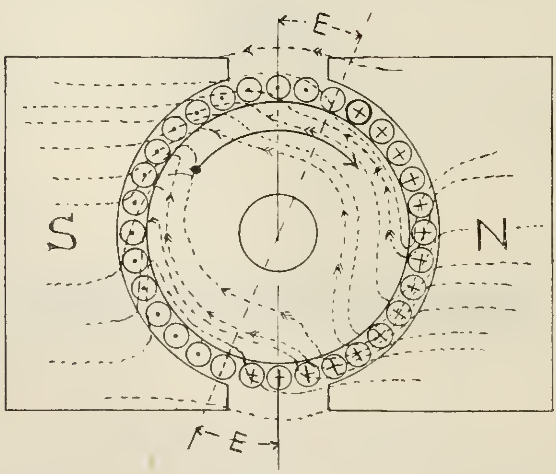
By coiling a wire round a bar of iron as shown, and passing a current through the wire, the bar becomes magnetised. The flow of current in the wire, polarity, and the lines of force produced in the bar are clearly shown by the arrows.



No. 84.

Diagram showing lines of force across field space, when undisturbed by armature magnetism. The position of the brushes would in this case be exactly at the top and bottom of the commutator circle.

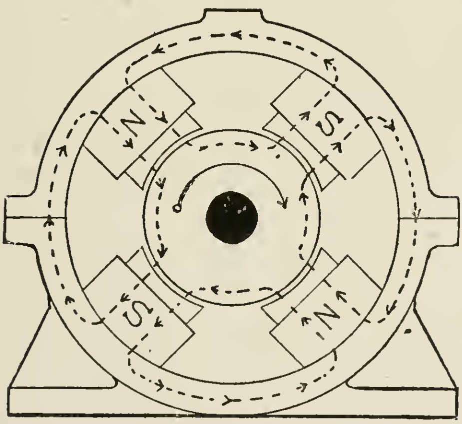
The dots on the conductor ends represent positive E.M.F.
,, crosses ,, ,, ,, negative ,,



No. 85.

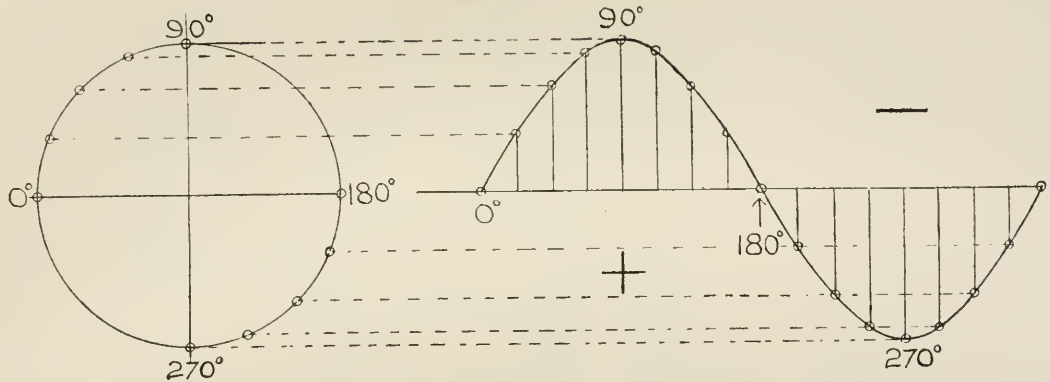
Diagram showing lines of force across field space as affected by armature magnetism. It will here be seen that the lines of force are now distorted by the effect of the armature acting as a magnet, thus pulling round the field lines in the direction of rotation, so that to strike the neutral point or sparkless position for the brushes when the load is increased, the latter require to be rocked round slightly in the direction of rotation (advanced). For an electric motor, as the E.M.F. action is reversed, the brushes require to be moved back in place of forward.

To correct this distortion of the field, dynamos are now usually fitted with "commutator poles" (see sketch 96), which neutralise the effect of the armature magnetism, and eliminate the necessity for altering the position of the brushes when the load or speed is increased.



No. 86.—Lines of Force in 4-Pole Dynamo.

The arrows show the direction of the lines of force between the poles of a 4-pole machine, the armature running clockwise. The armature conductors cut through these lines of force and generate currents by induction.



No. 88.—E.M.F. Alternation Curve of 2-Pole Dynamo.

In a 2-pole dynamo the E.M.F. alternates twice per revolution, and is represented as rising for positive and falling for negative.

The diagram shows what is called a "sine wave," and illustrates the variation of the current E.M.F. at different positions of the circle or revolution. At 0° the E.M.F. is at zero, at 90° the E.M.F. is at a maximum positive; at 180 it drops to zero again, at 270° it reaches a maximum negative, and back at 0° it again comes to zero.

The vertical division lines and the dotted horizontal lines show how the curve is developed.

To change these alternations into direct flow or continuous current a commutator becomes necessary. The number of alternations occurring per second is called the "frequency."

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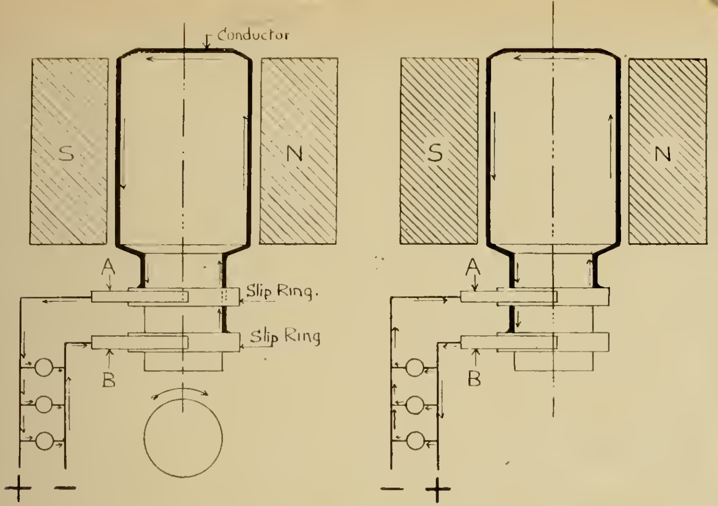
through the lamps and back again. At present position the shaded commutator bar is negative and the light one (left) positive.

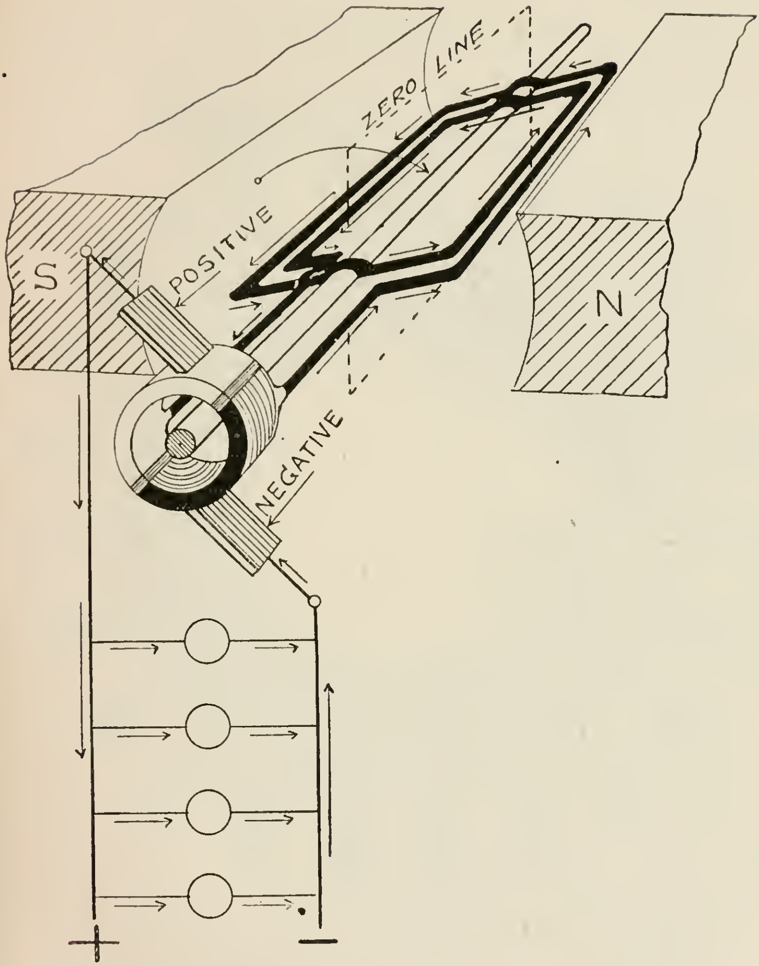


No. 87.—Simple 2-Pole A.C. Dynamo.

The arrows, together with the plus and minus signs, show clearly the reversal of E.M.F. at each half of the revolution, and it should be noted that in the left hand view the current is flowing through the lamp filaments from the brush on the slip ring nearest the armature, whereas in the right hand view the current is flowing through the lamps from the brush on the other slip ring and is thus reversed in direction. This is due to the fact that the conductor has turned half round, and the portion previously under the influence of positive current now comes under the influence of negative currents.

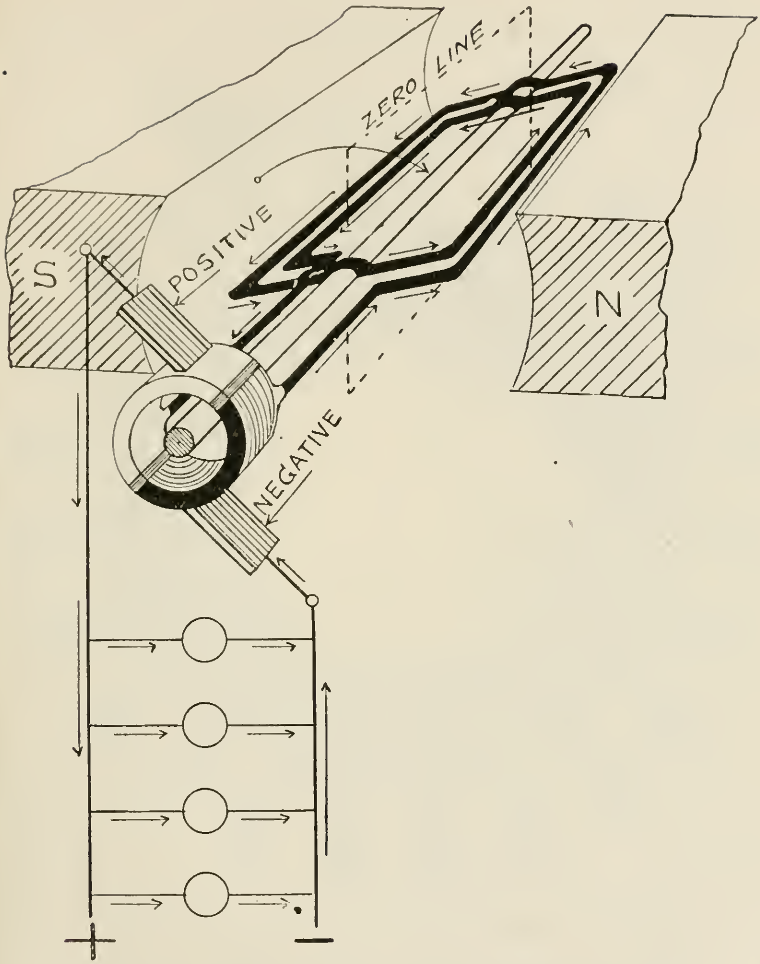
In place of a commutator two slip rings are fitted, and the brushes bear on these rings, taking up and giving back the current in the usual way. The conductor ends are connected to the two slip rings.

[*To face page* 498.



No. 89.—Simple D.C. Armature Conductor (with one conductor and two commutator bars).

The conductor, it will be noticed, is wound back on itself, and this is done to intensify the E.M.F. effect by increasing the number of turns or loops formed by the conductors, the effect being cumulative. The arrows show that the current flow is in one direction, from left to right through the lamps and back again. At present position the shaded commutator bar is negative and the light one (left) positive.



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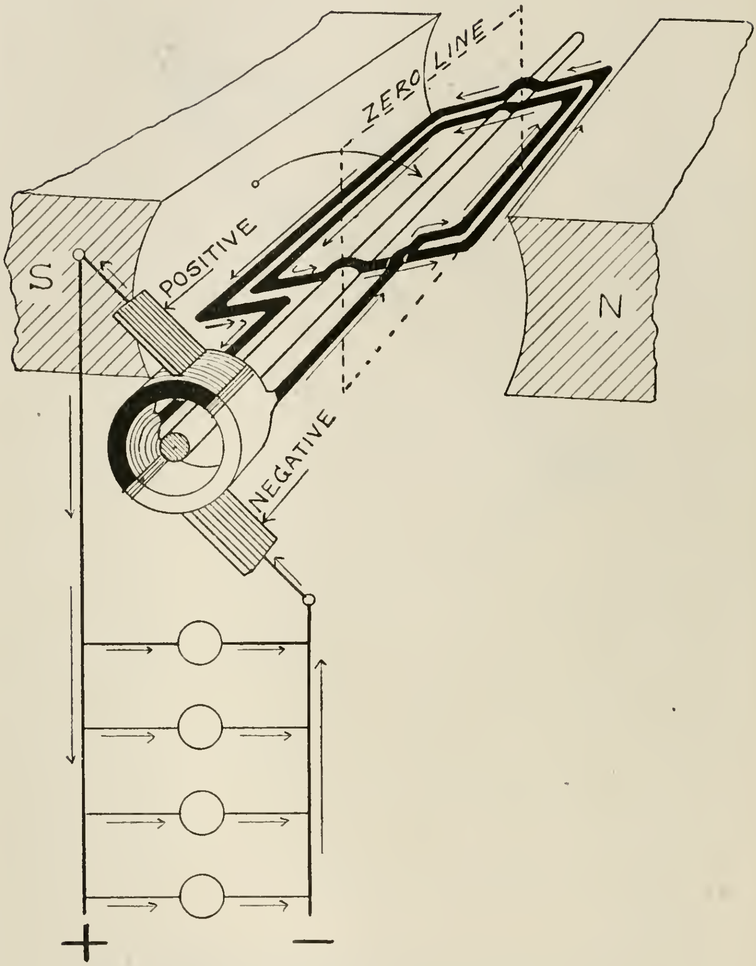
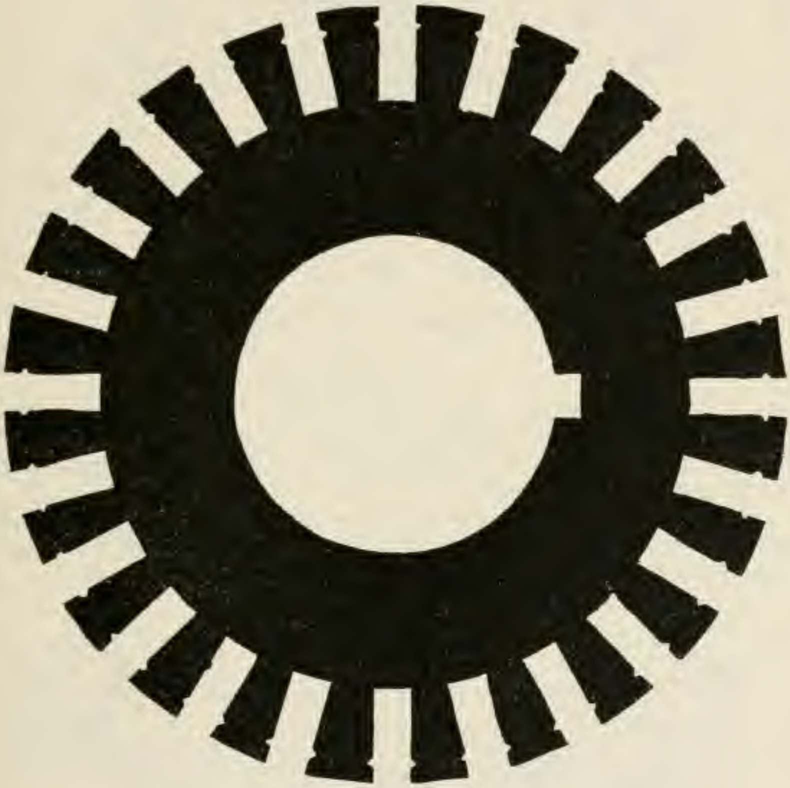


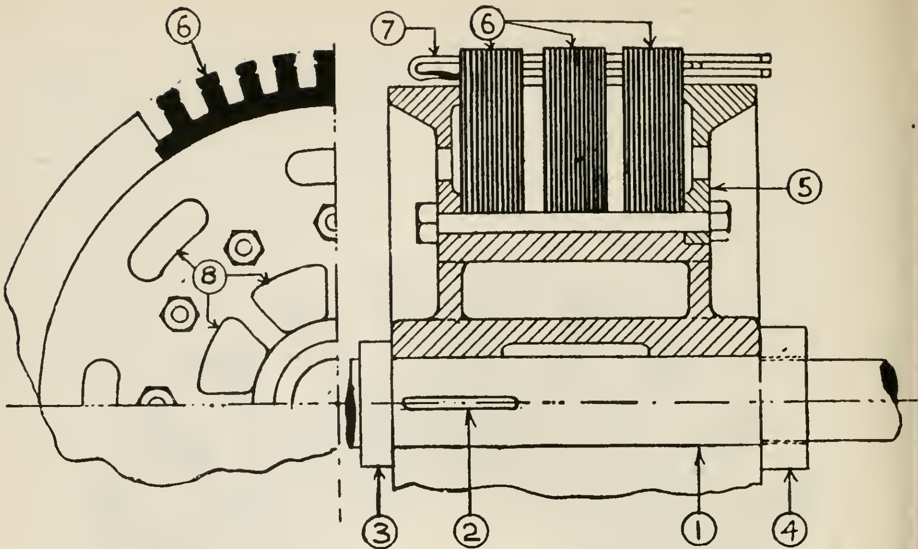
No. 90.—Simple D.C. Armature Conductor.

After making a half revolution it will be noticed that the current is still flowing in the same direction, as the dark commutator bar is now under the positive brush, and thus counteracts the reversal of current flow in the conductor. By this means the current is "commuted" to flow in one direction all the time.



No. 91.—Soft Iron Disc of Armature Body.

The armature is built up with a number of these plates formed of soft charcoal iron, insulated from each other by thick varnishing, and clamped on endways to the driving spindle, spider, and key. The slots shown are for the insulated conductors, which are held in place by check pieces and by binding straps.



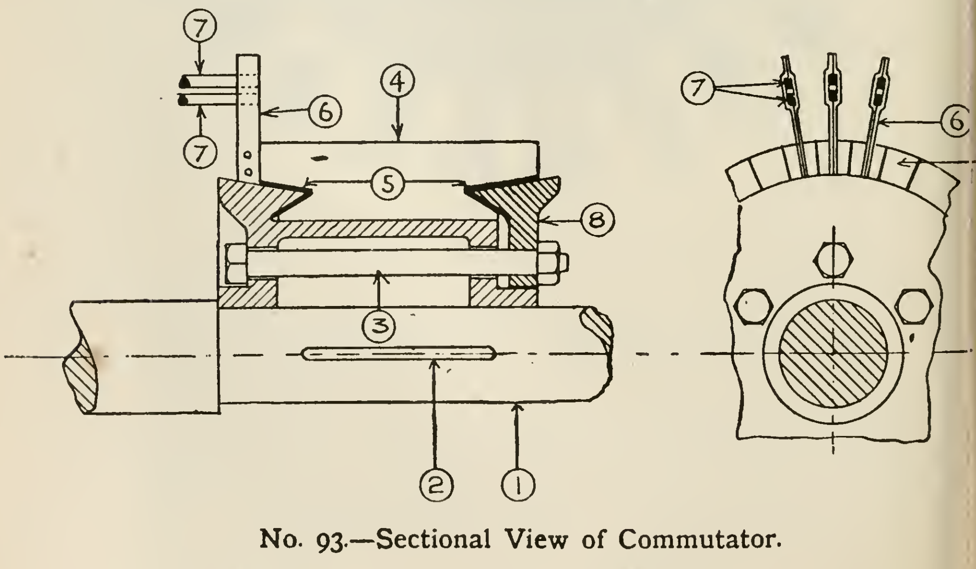
No. 92.—Sectional View of Armature.

1, Driving shaft. 2, Feather or key. 3, Collar.
4, Nut. 5, Locking plate.

6, Plates of soft iron insulated from each other by varnish, and which form the "armature core." The insulating of each plate from the next prevents the formation of eddy currents in the armature body, without in any way interfering with the flow of lines of force across the poles.

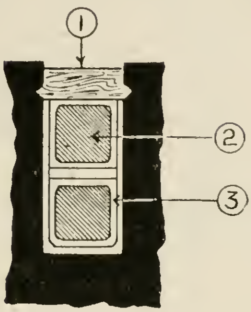
7, Conductor lying in longitudinal slotway, and connected up to the commutator bars by either the "wave" or "lap" wound system of winding.

8, Ventilation openings to prevent rise of temperature.



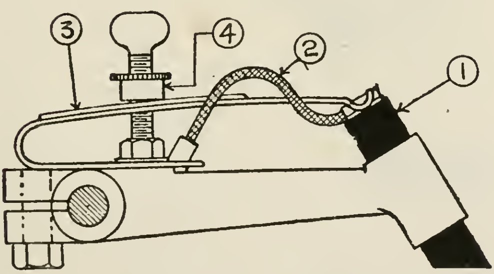
No. 93.—Sectional View of Commutator.

1, Driving shaft.
2, Feather or key.
3, Bolt for clamping copper blocks in position.
4, Copper bar.
5, Insulation.
6, Connector for conductor and copper block.
7, Armature conductor.
8, Locking clamp.



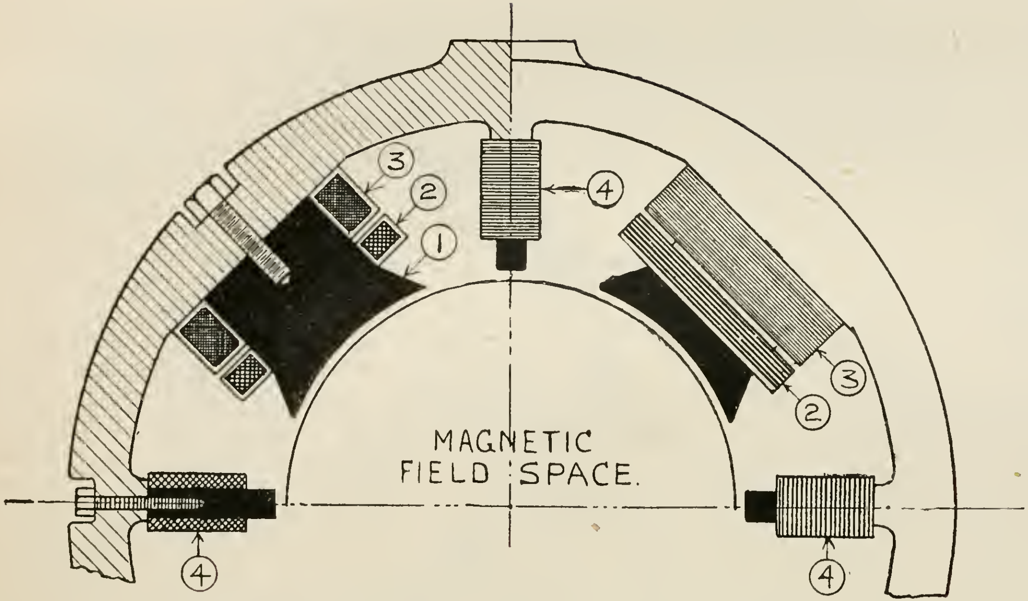
No. 94.—Slot in Armature with Conductor in Position.

1, Wooden key or locking piece.
2, Copper conductor.
3, Insulation of conductor.



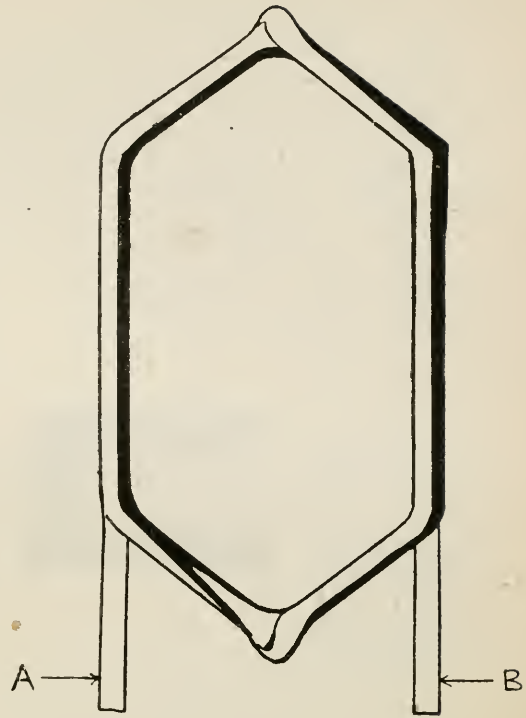
No. 95.—Box Type Brush Holder (Carbon Brushes.)

1, Carbon brush.
2, Conductor connection to brush.
3, Spring for pressure of brush.
4, Screw for spring adjustment.



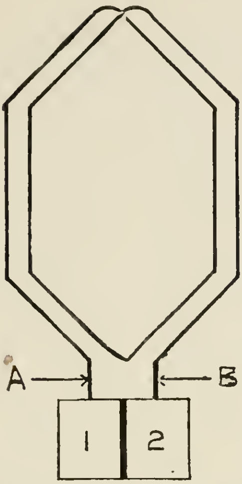
No. 96.—Field Magnets of 4-Pole Dynamo.

1, Magnet core of steel or of wrought iron.
2, Series wound coils of coarse wire.
3, Shunt ,, ,, fine wire.
4, Commutator poles, which correct the field distortion produced by the armature magnetism, and thus admit of the brush position being kept to a fixed mark for all conditions of load. When commutator poles are omitted the brushes require to be advanced when the load is increased to obtain the sparkless position.



No. 97.—Armature Conductor.

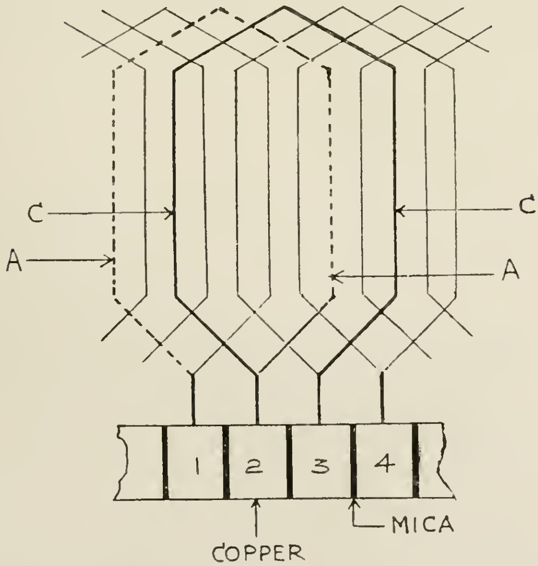
End A connects metalically to a copper block of the commutator, and, after looping round the other end B connects to the adjacent copper block.



No. 98.—Lap Winding.

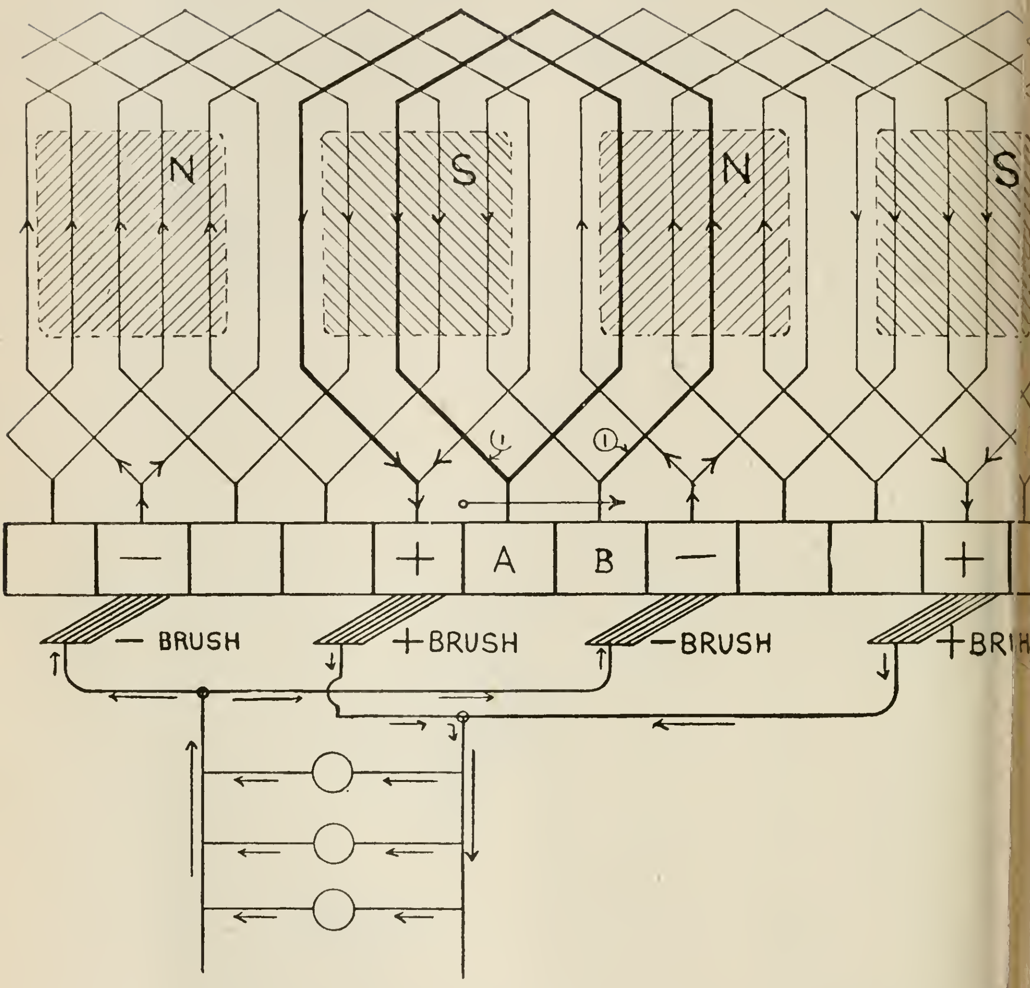
Observe that the conductor starts at A end on block 1, and after looping round finishes on block 2 at B end.

The loop is arranged to intensify the electrical effect produced in the coil.



No. 99.—Example of Lap Winding of Armatures.

Conductor A begins on commutator bar 1 and finishes on 2.
,, C ,, ,, ,, 2 ,, ,, 3.
That is, each conductor is led off from one bar and completes its path on the adjoining bar, after first passing from front to back of the armature, across the back, then back to the front again.

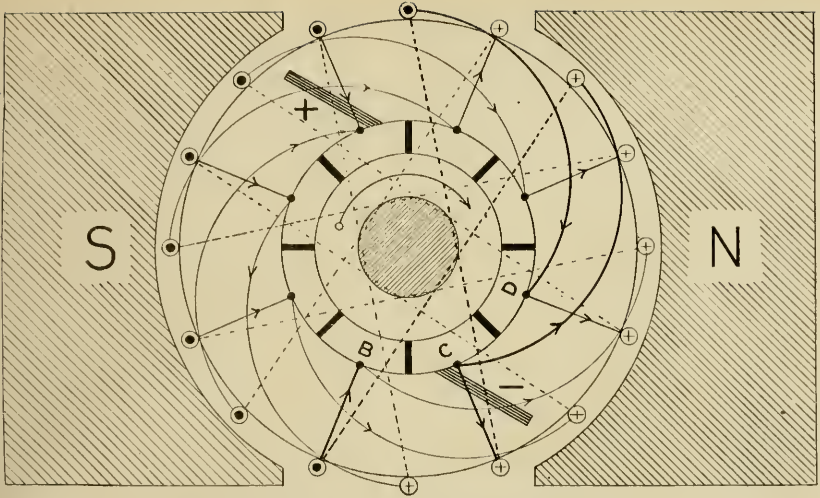


No. 100.—Diagram of 4-Pole D.C. Armature Lap Winding.

Note.—By D.C. is meant "direct current," and A.C. "alternating current."

The above sketch shows an expanded view of the armature windings, magnet poles, commutator, and brushes; the lamp circuits are also indicated, and the current flow (or E.M.F. induced) represented by the direction of the arrows. Observe that the conductor (1) starts on commutator block A, and finishes on block B; each conductor is coupled up similarly, and by suitably arranging the conductor groupings the combined flow of the current from each conductor *enters* at those blocks of the commutator where the positive brushes are placed (marked +). In the same way at other two neutral positions the current from each conductor flows *away* from the commutator blocks, and these are the positions for the negative brushes.

angles of connection back and front,
off at the same angle back and front, owing to the fact that the conductors connect to different commutator bars than those of the lap winding.



No. 101.—Diagram of 2 Pole Armature Lap Winding.

Black dots shown thus • represent positive flow E.M.F. currents.
Crosses ,, + ,, negative ,, ,,

Observe that each conductor starts out from one commutator block, passes out radially to the drum surface, runs back to the end of the armature, crosses the back, is led forward again, and finally connects up to the adjacent commutator bar.

Referring to the sketch, the coil beginning on block C completes its path on block D after being led round as described, and in the same way the conductor starting from block B finishes on block C, and so on. Tracing out the flow of E.M.F. it will be evident that the resultant current flow of each conductor connected to the copper block on which the positive brush is shown in contact is in the same direction, that is, into the block, and the same holds good at the negative brush position as the combined current flow is then outwards or negative. This action is successive, as each block comes into position under the positive or negative brush.

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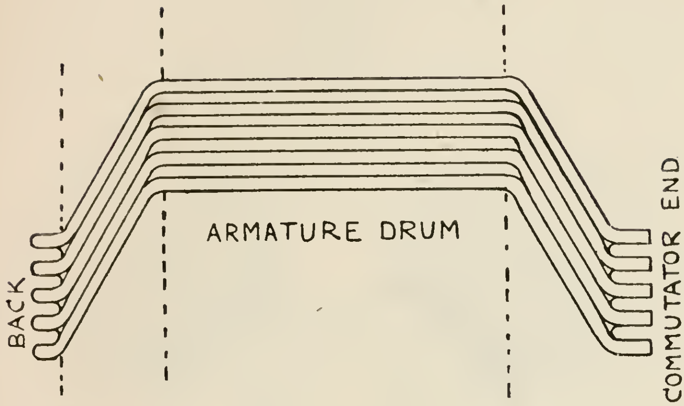
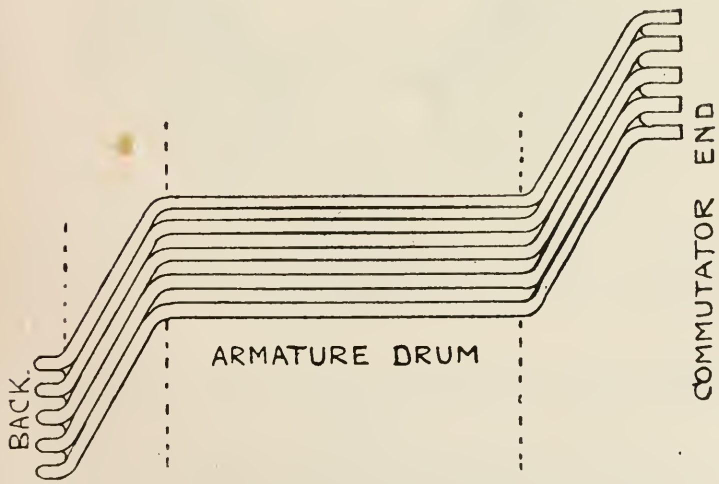
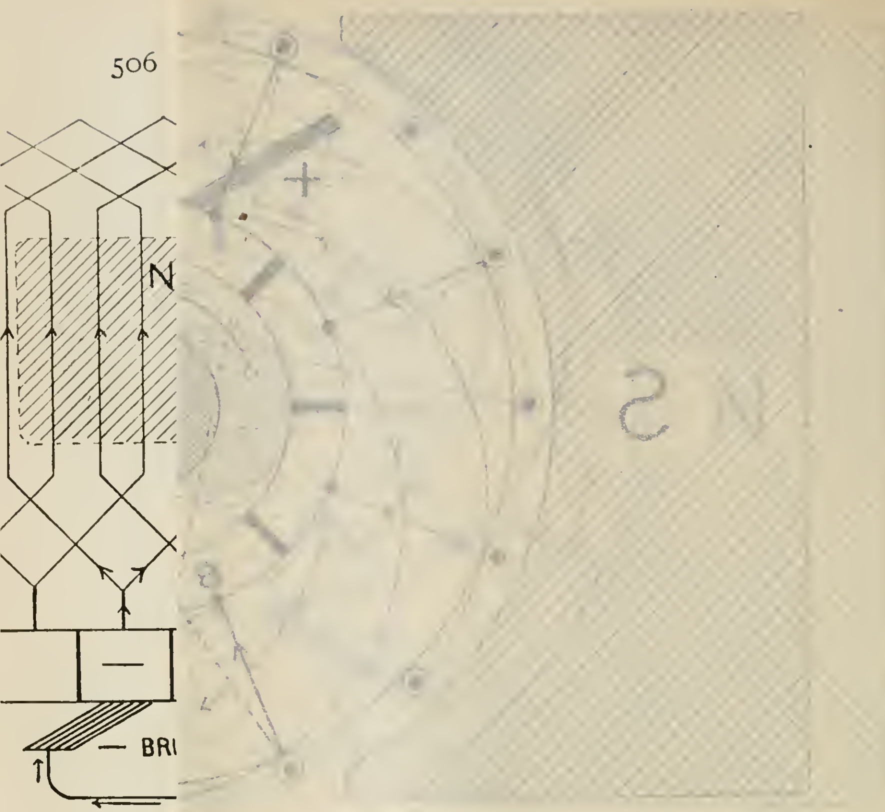


No. 102.—Lap Wound Armature Winding.



No. 103.—Wave Wound Armature Winding.

Observe the difference in the two methods of winding as seen on the drum surface, the lap method forming equal and opposite angles of connection back and front, whereas the wave winding runs off at the same angle back and front, owing to the fact that the conductors connect to different commutator bars than those of the lap winding.



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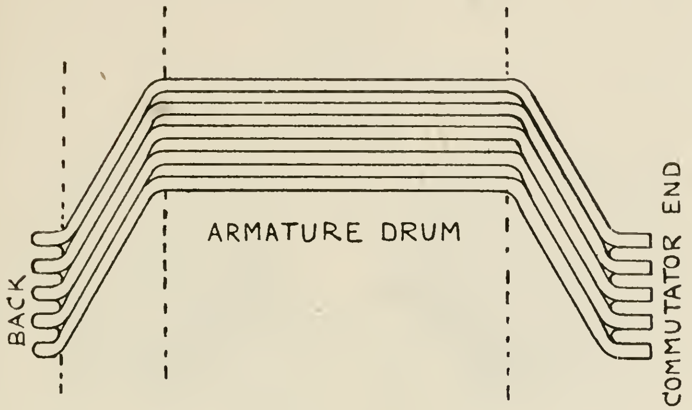
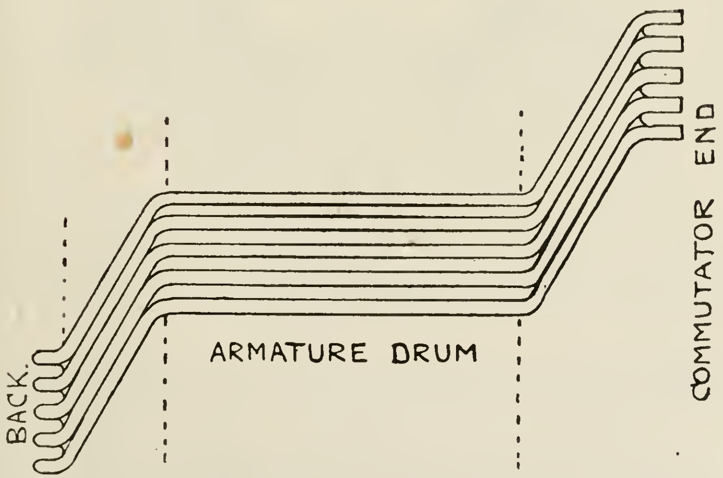
and finishes on block B; each conductor is coupled up similarly,

suitably arranging the conductor groupings the combined flow of the current from each conductor *enters* at those blocks of the commutator where the positive brushes are placed (marked +). In the same way at other two neutral positions the current from each conductor flows *away* from the commutator blocks, and these are the positions for the negative brushes.

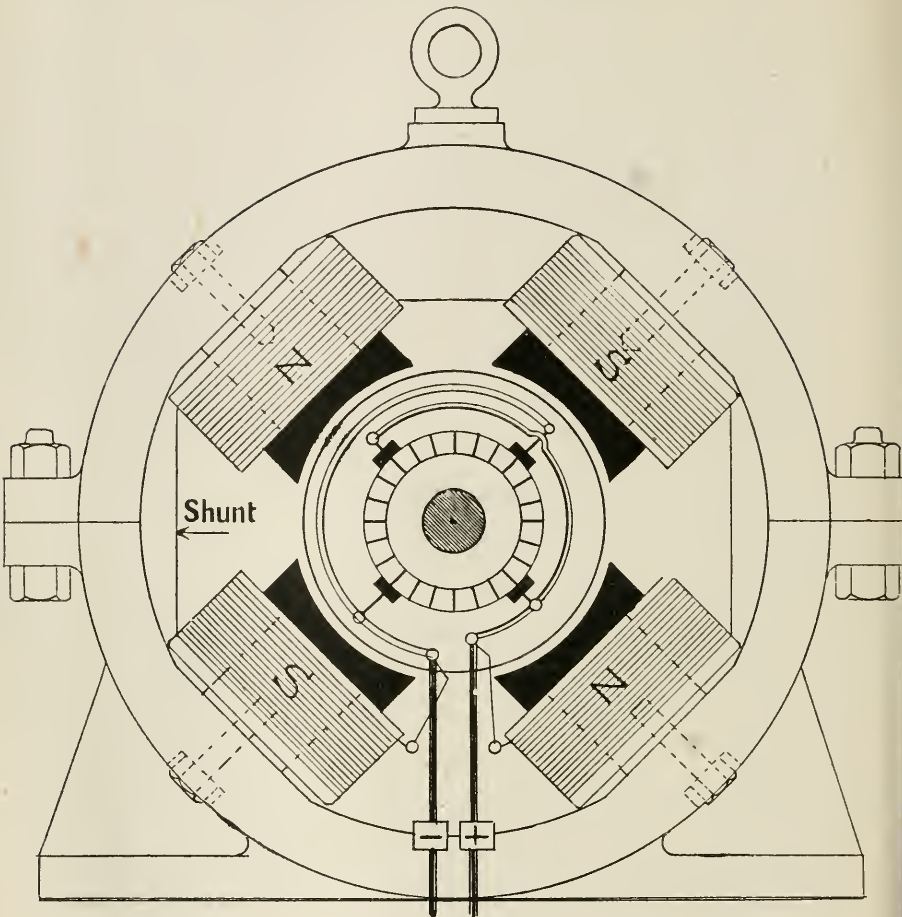


No. 102.—Lap Wound Armature Winding.



No. 103.—Wave Wound Armature Winding.

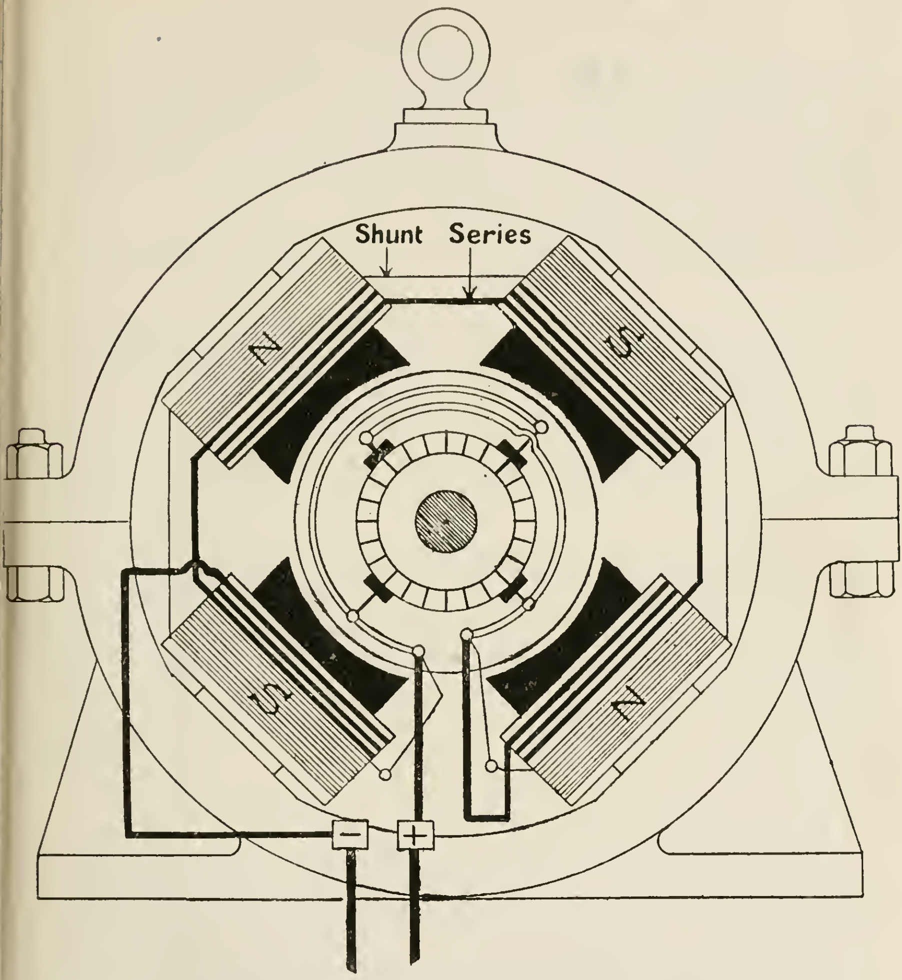
Observe the difference in the two methods of winding as seen on the drum surface, the lap method forming equal and opposite angles of connection back and front, whereas the wave winding runs off at the same angle back and front, owing to the fact that the conductors connect to different commutator bars than those of the lap winding.



No. 104.—End View of 4-Pole Shunt Wound D.C. Dynamo.

With the armature turning clockwise the brushes opposite the S. poles are positive, and those opposite the N. poles are negative. The two positive sets are coupled together by the ring shown, and the two negative sets are arranged similarly.

The objection to this type of dynamo for parallel lighting purposes is that when lamps are switched off the remaining lights would burn brighter unless the armature speed was reduced to correspond: the reverse also holds good if lamps are switched into circuit, the reason being that the current has two paths open to it, and if the resistance in one of two is increased the other will receive an increase of current, assuming that the armature speed were to remain constant.



No. 105.—End View of 4-Pole Compound Wound D.C. Dynamo.

With the armature revolving clockwise, the two brushes opposite the S. poles are positive, and it should be noted that they are connected together by a ring-piece to which is fixed the positive series winding wire.

The negative brushes are opposite the N. poles, and the two sets of brushes are also connected to each other and to the negative series winding wire.

Observe that the shunt winding starts from one brush connection, is led round the four magnet poles, and returns to the other brush connection, thus completing its path. Again it should be noted that only a few series turns are employed, whereas a great number of the fine shunt windings are necessary.

MOTOR STARTING RESISTANCE, &c.

The reader is advised to carefully study the following extract from an article entitled "Some Notes on Elementary Electrical Engineering," by W. Benison Hird, B.A., M.I.E.E., and which appeared recently in *The M. and C. Apprentices' Magazine*, as the section dealing with the principle of motor starting switch resistances is of peculiar value to the marine engineer.

The author has to thank W. B. Hird, Esq., for kind permission to use the article as here reprinted.

"In these notes an attempt is made to put the reader in the way of solving for himself such of these questions as relate to elementary electricity and magnetism and the simpler facts concerning dynamos and motors.

"There are two elementary laws or principles which, properly understood and applied, will carry him very far towards the solution of many at first sight puzzling facts in elementary electrical engineering.

"First. The current flowing in any circuit is equal to the sum of the electromotive forces in that circuit divided by the resistance.

"That is to say that a given E.M.F. being applied to the terminals of any circuit, a perfectly definite current will flow through it, and that current will be proportional to the E.M.F. divided by the resistance. If the E.M.F. is measured in volts and the resistance in ohms, the ratio $\frac{\text{E.M.F.}}{\text{Resistance}}$ will represent the current in amperes. In symbols if E represent the E.M.F. in volts, R the resistance in ohms, and C the current in amperes, then $C = \frac{E}{R}$.

"Second. The second principle to remember is the observed fact that if any conductor is moved in a magnetic field so as to cut the lines of magnetic force, an E.M.F. will be generated in the conductor.

"A magnetic field is any portion of space under the influence of a magnet. Every magnet has two poles, a north pole and a south pole, and we can imagine the whole of the space near the magnet as filled with imaginary lines running from the north pole to the south pole, and along which the force of attraction of the magnet acts. Such lines are known as lines of magnetic force, and the number of such lines in a given area depends upon the strength of the magnet and the distance they have to pass in getting from the north pole to the south pole; in fact, it is proportional to the strength of the magnetic field.

"To take a concrete example, take a two-pole dynamo or motor: the magnet consists of a horse-shoe shaped piece of cast steel, and is converted by the current circulating in the field coils into an electro-magnet with a north pole at one end and a south pole at the other; when the armature is put into this magnet a narrow air gap is left between the steel of the magnet and the iron of the armature core: this narrow gap must be imagined as filled by a large number of lines of magnetic force running from the north pole of the magnet into the armature iron and out of this into the south pole of the magnet. For the most part these lines will run radially and straight across the gap, but at the edges of the magnet they will spread out and form a fringe extending into the armature iron at some little distance from the magnets. Forming a mental picture of such a magnetic field, it is evident that any conductors laid on the armature parallel to the shaft will, when the armature rotates, cut the lines of force, and will therefore have an E.M.F. generated on them.

"Note that it is only an E.M.F. that is generated, and that there is not necessarily any current flowing in the conductors. In order that current may flow, another condition is necessary besides the cutting of lines of force, and that is that the conductors should form part of a closed circuit to carry this current, and if such is the case the current will then follow our first law, and the amount of current flowing will be equal to the E.M.F. generated divided by the total resistance of the circuit.

"Further, a conductor placed in a magnetic field and carrying a current will tend to move in such a way as to cut the lines of magnetic force, the direction of motion being such as to generate an E.M.F. opposing the current.

"In illustration of the above apply these principles to some of the simplest and most elementary questions which are likely to occur to any one just coming into contact with dynamos and motors. For instance—

Why is a Resistance put in Series with a D.C. Motor on Starting it Up?

"The resistance of a motor armature is necessarily very small: high resistance in the armature winding would mean large losses, which would be injurious in two ways. First, as all losses appear ultimately as heat, they would cause too high a temperature rise to the detriment of the insulating materials, which would rapidly perish; secondly, they would mean an unnecessarily low efficiency, for evidently these losses must be supplied from some source, and must mean that an increased power is taken from the mains to give out the same power at the pulley. This being the case, the armature resistance is kept as low as possible, and if the motor were put directly on to the mains, it would momentarily take a very large current. Take a numerical example. A 10 H.P. motor designed to run at 500 volts will have an armature resistance of about ·5 ohm. If such an armature were switched directly on to the mains at 500 volts, it is readily seen from our first principle that the current would amount to $\frac{500}{\cdot 5} = 1000$ amperes, but the normal current of the motor when giving 10 H.P. is only about 20 amperes; the mains and armature conductors are only designed to carry 20 amperes, and such a large current as 1000, even if only for a moment, is very excessive, and evidently out of the question. Even apart from any electrical consideration, the mechanical effect of such an excessive current would be damaging in the extreme to the mechanical parts of the motor, and of the machines it might be driving; to put suddenly on to any machine a force 50 times in excess of what it is intended to carry is evidently courting disaster. For this reason a resistance is inserted to check the momentary flow of current. Say a resistance of 19·5 ohms is inserted in circuit with the motor, then the current flowing through the circuit will be $\frac{500}{20} = 25$ amperes. The armature conductors are now in a magnetic field, and they are carrying a current of 25 amperes. They will tend to move in such a way as to cut the lines of magnetic force, and the armature will start to rotate. As soon as rotation begins an E.M.F. will be generated in the armature opposing the E.M.F. applied to the terminals, and this back E.M.F. will increase as the speed increases, because the E.M.F. generated by the cutting of lines is proportioned to the rate of cutting. After the armature has attained a certain speed, the back E.M.F. will be, say, 200 volts. Then the effective electromotive force in the circuit will be 500-200 volts = 300 volts, 500 volts at the mains and an E.M.F. of 200 volts generated in the motor armature, and acting in the opposite direction. Again, divide E.M.F. by resistance, and we will find that the current will now be $\frac{300}{20} = 15$ amperes. Will the armature increase in speed and the current be further cut down? This depends entirely on what work the motor is put to. Say it is driving shafting; if 15 amperes is just sufficient to drive the shaft, the speed will not increase, and the motor will go on running steadily at the same speed. If, however, a smaller power is sufficient to drive the shaft at this speed, the motor armature will go on accelerating, the E.M.F. generated will also increase, and further reduction of the current will take place. Assume that 15 amperes is just sufficient to drive the shaft, the motor will then continue running at the speed it has now reached. Now is the time to cut out some of the resistance introduced in the circuit. Say this is reduced by moving the starting switch on to the next contact to 11·5 ohms, the total resistance in circuit will be 12 ohms, and we have seen the effective E.M.F. to be 300 volts; the current will therefore rise to $\frac{300}{12} = 25$ amperes. This is more than what is required to keep the shafting in motion, and the armature will therefore increase in speed: the back electro-motive force will also increase and the current be cut down in value until it again reaches the value of 15 amperes. The process is then repeated until all the resistance is cut out and the motor is running at normal speed.

Why does the Introduction of Resistance in the Shunt Circuit of a Motor cause an Increase in the Speed?

"To many beginners in the subject it appears that one would more naturally expect the reverse to be the case, and that putting in resistance would lower the speed, taking it out would increase the speed. On consideration, it is easily seen that this view is erroneous. The shunt circuit of a motor is a separate circuit, having a definite resistance and connected across the mains, so that it has a definite E.M.F. on its terminals; it will therefore carry a definite current $C = \frac{E}{R}$ where C is the shunt current, R the resistance of the shunt circuit, and E the E.M.F. at the mains. Now, any increase in the resistance evidently decreases the current and weakens the magnetic field. There will be fewer lines of magnetic force flowing across the air gap of the motor, but in order to give the same back E.M.F. the conductors must cut the lines of force at the same rate. Since there are fewer lines to cut, the conductors must cut what there are quicker, that is, the armature will speed up. Why must the armature keep up the same back E.M.F.? Because if it does not, the effective E.M.F. in circuit is greater; the current being equal to the effective E.M.F. divided by the resistance of the armature circuit is also increased, and the armature taking a current greater than that required to overcome the load will accelerate.

SECTION VIII.

PROPELLERS.

As the majority of marine engineers are unfamiliar with the various definitions connected with the screw propeller, the author has deemed it advisable to endeavour to give clear and, if possible concise explanations of each, accompanied by suitable illustrations.

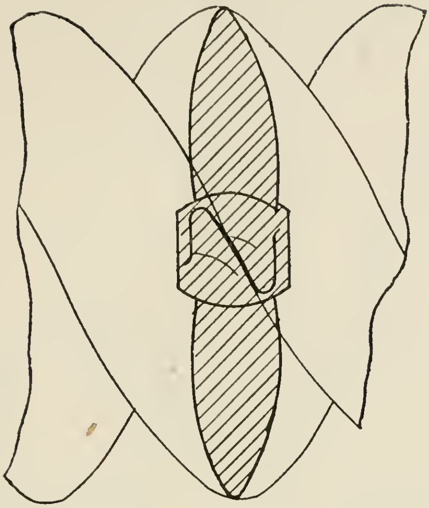
General.—The marine propeller, simply considered, is merely a common screw working in a nut.

If a screwed bolt be turned one complete revolution in a corresponding nut, the bolt will advance or travel along the nut a distance equal to that between two adjacent threads, or, as it is usually expressed, equal to one pitch, P (see sketch), so that if a bolt has, say, eight threads to the inch, in one turn of the bolt or nut the advance will be $\frac{1}{8}$ inch.

No. 1.—Common Screw.

P = Pitch.

The diameter of the propeller boss represents the bolt at the bottom of the thread, and the propeller blades the actual threads, or, more correctly, *pieces* of thread (see sketch).



No. 2.—Actual Propeller compared with Actual Screw.

A three-bladed propeller consists of three pieces of thread set on the boss, and a four-bladed propeller consists of four pieces of thread set on the boss.

The water in which the propeller is immersed, and in which it works, represents the nut, so that when the bolt or propeller shaft is revolved the screw or propeller advances in the nut, which, as before stated, is represented by the water in which the propeller works.

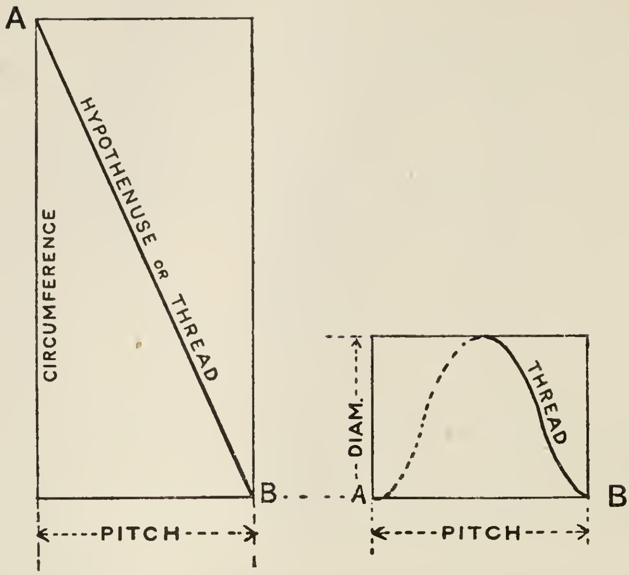
As, however, the water constitutes a yielding nut and gives way a certain amount to the blades, the actual advance of the propeller and ship is less than one pitch of the screw for one complete revolution of the shaft ; this difference of advance is known as the slip.

Thrust.—The effect of the propeller thrust is, generally speaking, twofold—(1) to drive the water aft, (2) to drive the ship forward. The reaction of driving the water aft results in the steamer being driven or propelled forward.

Pitch.—As stated above, the pitch is the longitudinal advance of a screw during one revolution, if working in a solid nut.

A screw or helix, if unrolled, forms with the pitch and circumference a triangle, of which the thread represents the hypothenuse or diagonal.

If, therefore, a sheet of paper is taken, and a line A B drawn from corner to corner, and the sheet rolled up into a cylinder, the ruled line A B will represent the edge of a screw or thread, and the length of the roll the pitch (Sketches Nos. 3 and 4).



No. 3.—Flat Sheet of Paper. **No. 4.—Paper rolled up.**

Triangle described by Screw.

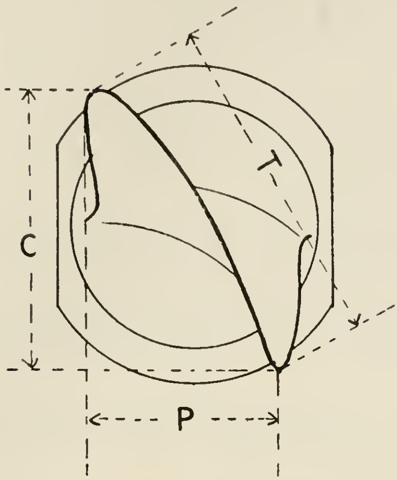
It will thus be seen that the pitch is the length between the two ends of the thread taken for one complete turn of the screw, or the advance made by any point or piece of the thread during one revolution.

In an actual propeller the breadth of blade represents the piece of thread or the hypothenuse of the triangle, the distance between the leading and after edges the piece of pitch, and the distance, measured vertically, between the lower and upper edges the piece of circumference: these all correspond, it will be noted, with the screw representation on the sheet of paper as shown above.

Right- and Left-hand Screw.—The blades of a right-handed propeller revolve from port to starboard (upper half of blade circle), and the blades of a left-handed propeller revolve from starboard to port.

Circumference.—As will readily be understood from the foregoing, the circumference is the distance round the cylinder forming the screw surface at right angle to the shaft, which, when unrolled, forms one of the shorter sides of the triangle. The diameter of the propeller multiplied by 3·1416 is equal to the complete circumference.

Thread.—As before stated, the thread is the complete length of the unrolled helix, and each propeller blade is equal in width to a piece

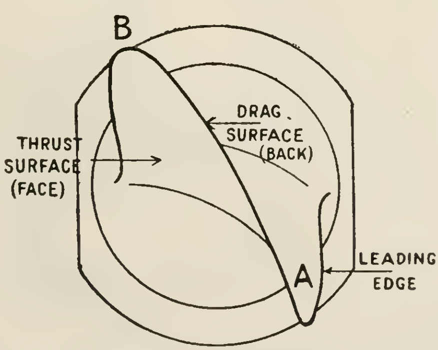


No. 5.—Right-hand Propeller Blade.

T, Piece of Thread, or Hypothenuse. P, Piece of Pitch. C, Piece of Circumference.

of the thread only. The thread forms the diagonal or hypothenuse of the pitch triangle, and therefore its longest side.

Increasing Pitch.—Propellers are sometimes designed with a varying pitch—(1) the pitch increasing radially, that is, the pitch at or near the tip of the blades is more than the pitch near the boss; or (2) the pitch may increase axially, or from forward aft, that is, the pitch near the after edge of the blade may be slightly more than the pitch near the forward edge. In the sketch shown below the pitch at B is more than the pitch at A.



No. 6.—Thrust and Drag Surfaces.

True Screw.—When the blades have no variation in pitch, either radially or axially, the blade surface is said to be that of a "true screw." Some of the most efficient propellers of present day practice, including those of turbine steamers, are of this type.

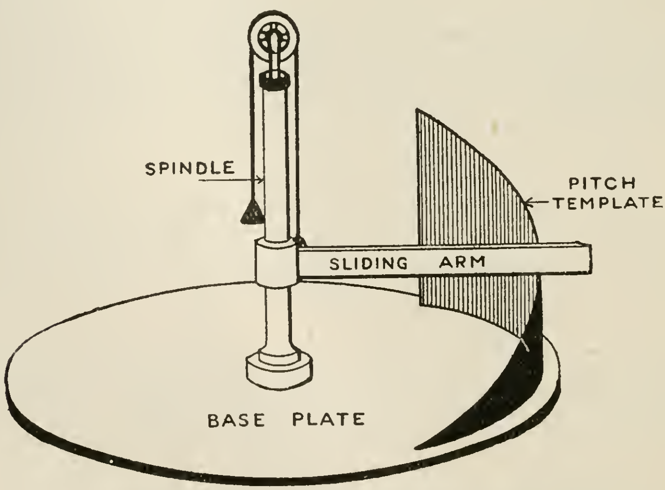
Pitch Ratio.—The propeller pitch, divided by the propeller diameter, is equal to the "pitch ratio." The pitch ratio may vary from about ·9 in small propellers to 1·4 in large propellers, in steamers with reciprocating engines.

Diameter of Propeller.—The diameter of a propeller is the diameter of the circle described by the tips of the blades.

Length of Propeller.—The length of a propeller is measured longitudinally on the shaft, and is usually about equal to the length of the boss.

Length of Blade.—The length of a blade is measured from the root at the boss to the tip of the blade.

Moulding of Blades.—In moulding the propeller blades a horizontal arm is rotated round a vertical spindle, and at the same time moved up or down the spindle, thus generating the screw surface or helix. To guide the travel of the moving arm one or more curved vertical guide templates are placed in position, and the moving arm travelling over the upper edges of the templates shapes out the blade surface for the required pitch, &c. The vertical templates referred to are triangular in shape, and are cut to the correct pitch angle for various points of radius on the blade.



No. 7.—Method of producing the Screw Surface.

Slip.—Slip is of three kinds—(1) apparent slip, (2) real or actual slip and (3) negative slip. The "log" slip found by taking the difference of the propeller speed and the ship speed is only apparent slip, the real slip being (in nearly every case) in excess of this. The real slip is found by adding together the apparent slip and the "wake speed" (if known).

Wake Speed.—Wake speed is the name given to the velocity of the stream or column of water which *follows* at the stern of a vessel.

The wake speed will be more with a bluff-lined steamer than one with fine lines, as the more square shape of the stern tends to pull the water along with the vessel, and thus give a higher wake speed.

From the foregoing it will perhaps be seen that the speed of the vessel is less relatively to the wake speed than to still water. This, therefore, has the effect of taking away, as it were, part of the actual slip.

As before stated, to calculate the real slip the wake speed must be added to the apparent slip.

Disc Area.—By disc area is meant the area of the circle described and enclosed by the tips of the propeller blades.

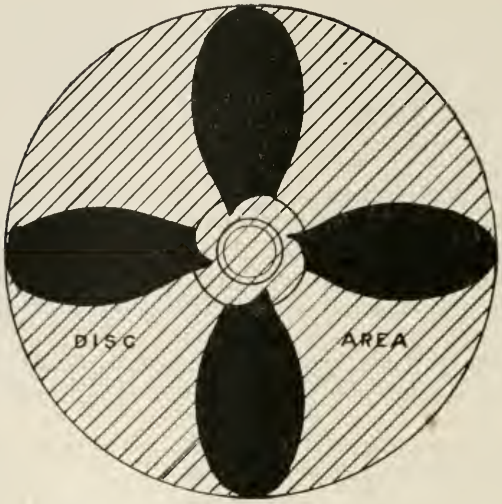
Developed Area.—By developed or expanded blade area is meant the full area of all the blades if flattened out and taken as approximate plane surfaces.

No. 8.—Expanded Blade Area.

Area Ratio.—By area ratio is meant the relative total expanded area of the blades compared with the "disc area." The area ratio varies from ·3 to ·6 of the disc area.

Projected Area.—This means the actual area of blades as projected at right angles to the line of shafting, and constitutes the effective thrusting area of the blades.

NOTE.—The blade area as seen when looking forward from behind the propeller is the "projected area."



No. 9.—Projected, or Effective Thrusting Area of Blades.

Thrust Surface or Driving Face.—The *after* surface of the blades is the thrusting surface, which acts on the water to drive forward the vessel.

Drag Surface or Back of Blade.—The forward (usually the rounded) surface of the blades is known as the drag surface.

Leading Edge.—With the blade in a horizontal position (see sketch), the "leading edge" is that edge lowest down, and which for a right-hand propeller lies on the starboard side; for a left-hand screw the leading edge will be on the port side.

Following Edge.—With the blade in a horizontal position, the after edge and that highest up is called the "following edge."

Cavitation.—By cavitation is meant the failure of water supply or "feed" to the propeller, due generally to excessive blade velocity; in other words, the blade speed exceeds the water flow speed *to* the blades, therefore the effective thrust falls off in proportion, as *cavities* form at the forward side of the blades. Cavitation is therefore caused by the ineffectiveness of the atmospheric pressure to press up the water at the back of the blades (forward side) fast enough to allow of effective thrust; this usually occurs at high revolution speeds and high blade pressures per square inch.

The phenomenon of cavitation has been very exhaustively investigated in a series of elaborate experiments carried out by the Hon.

C. A. Parsons, in connection with trials of turbine-engine propellers (see "The Marine Steam Turbine," by J. W. Sothern).

It should be noted that slip is an absolute necessity for the effective effort of a screw propeller, and if a propeller shows a very low slip percentage it indicates that the propeller fitted is evidently unsuitable for the steamer in question, as it is not delivering an effective thrust on the water. In most cases the slip should average from 5 to 15 per cent., and in some cases even more.

Apparent Negative Slip.—If *without* strong current speed a propeller shows negative slip, it may be taken for granted that (as in the case of low positive slip) the propeller is not suited for the work it has to do, and indicates the necessity for another propeller of different design being substituted, probably one of greater pitch and diameter. Negative slip is most likely to show in steamers having very bluff stern lines, and therefore giving a resultant high wake speed.

NOTE.—Apparent negative slip may occur with a strong current going with the steamer.

It is generally admitted now by authorities on the propeller that negative slip can only be apparent, and an example may be given as follows:—

EXAMPLE.—Pitch 15 feet, Revolutions 62, ship's speed 14 Knots, speed of *following current* 4 Knots, find slip.

Then, Engine Knots $= \frac{18 \times 62 \times 60}{6080} = 11$ Knots,

and, $14 - 11 = 3$ Knots apparent negative slip.

But, Actual Propeller advance through water $= 14 - 4 = 10$ Knots.

Therefore, Real Slip $= 11 - 10 = 1$ Knot.

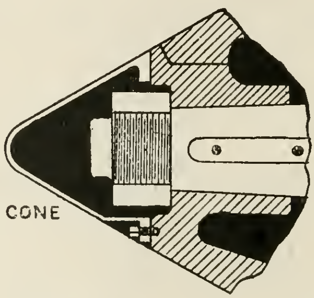
So that instead of an apparent negative slip of 3 Knots we have a positive and real slip of 1 Knot.

No. 10.—Blade "Set Back."

Set Back.—Sometimes the blades of a propeller are set with an inclination aft instead of being arranged at right angles to the shaft: this is known as "set back" or "skew."

Racing.—Racing is produced by the blades or *parts of the blades* rising out of the water when the stern lifts in pitching. The effect of this is to carry down into the water a quantity of air, and as the resulting mixture of air and water is less solid or dense than water alone, the blades meet with less resistance, and the engines "race" in consequence. It will be noted that the whole propeller does not require to come out of the water to produce racing, as part only of a blade coming above the surface may be sufficient to produce it. Less racing will occur when small propellers are fitted low down, as in the case of turbine steamers.

Cone.—In well-finished propellers the nut aft of the boss is covered with a thin metal cover, conical in shape, which continues aft the round of the boss, and prevents interruption of the "flow" or "run" of the water past the blades.



No. 11.—Nut and Cone.

Propeller Design.

The majority of the following rules for propeller design are taken from Seaton's "Manual of Marine Engineering," also Seaton and Rounthwaite's "Pocket-Book of Marine Engineering Rules and Tables," and the author would take this opportunity of recommending a copy of either of these standard works to all readers anxious to investigate more fully into the various problems of marine engineering design. The following descriptions are therefore intended to be applied in conjunction with the above-named books, a copy of which, as before stated, should be obtained for reference.

NOTE.—It must be clearly understood that no absolutely correct hard and fast rules suitable for the successful designing of propellers can be laid down on paper, as in actual drawing-office practice comparative records of previous performances, tables of "slip" factors, "area factors," Admiralty coefficients, results of tank experiments, and other data, are largely employed in arriving at the best pattern of propeller suitable for a steamer of given type, dimensions, and speed. A vast amount of investigation is yet open to experimenters in propeller efficiency and design, as at present, in a number of cases, the most suitable propeller is often only found after repeated trials of other propellers of different pitch, diameter, and area. In support of this the writer remembers once seeing nineteen propellers which had all been tried successively on a torpedo destroyer before the one giving the best results was discovered.

I. To Design and Draw a Propeller for the following:—

Given:

I.H.P.	- -	600.
Knots	- -	10.
Revolutions	-	76.
Tail shaft	- -	11 inches diameter.
Type of steamer		Cargo boat.

Propeller to be of the four-bladed, cast-iron, solid type.

NOTE.—It should be stated that it is not usual in drawing-office practice to show as many different views of the blades, &c., as here drawn in the examples given, but as this description is specially written for the use of marine engineers with little or no experience of geometrical projection drawing, it has been considered advisable, for the sake of clearness, to show each separate stage of the construction, hence the necessity for repeating some of the views which might otherwise, as will easily be seen, have been combined in one.

To find Propeller Pitch.—Allow 10 per cent. for apparent slip, and proceed as follows:—

RULE.—

$$\frac{\text{Knots} \times 6080 \times 100}{\text{Rev.} \times 60 \times \text{effective per cent.}} = \text{Pitch.}$$

Therefore, $$\frac{10 \times 6080 \times 100}{76 \times 60 \times 90} = 15 \text{ feet Pitch.}$$

NOTE.—6080 feet = 1 knot.
60 min. = 1 hour.
100 − 10 = 90 per cent. effective advance.

To find Propeller Diameter.

RULE.—

$$\text{Constant K} \times \sqrt{\frac{\text{I.H.P.}}{\left(\frac{\text{pitch} \times \text{rev.}}{100}\right)^3}} = \text{Diameter.}$$

Therefore, $$\text{K } 18 \times \sqrt{\frac{600}{\left(\frac{15 \times 76}{100}\right)^3}} = 11{\cdot}46 \text{ feet, or say } 11\tfrac{1}{2} \text{ feet Diameter.}$$

NOTE.—K = Constant 18 in present case (see Seaton and Rounthwaite's "Pocket-Book" for table of Constants).
600 = I.H.P.
15 = pitch.
76 = revolutions.

To find Total Expanded Blade Area.—The *total* expanded blade area is found as follows:—

RULE.—

$$C \times \sqrt{\frac{\text{I.H.P.}}{\text{rev.}}} = \text{total Surface.}$$

Therefore, $\quad \text{Constant } 16 \times \sqrt{\frac{600}{76}} = 44{\cdot}9$, or say 45 square feet.

NOTE.—C = Constant 16 in present case (see Seaton and Rounthwaite's "Pocket-Book" for table of Constants).
600 = I.H.P.
76 = revolutions.

To find Diameter and Length of Boss.—For a cast-iron propeller with blades and boss solid, the diameter and length of tne boss are found as follows:—

RULE.—Constant $2{\cdot}7 \times$ shaft diameter = boss diameter and boss length, therefore $2{\cdot}7 \times 11 = 29{\cdot}7$ inches, or say 30 inches diameter and length of boss.

NOTE.—11 inches = tail shaft diameter.
The boss diameter varies from $\frac{1}{4}$ to $\frac{1}{6}$ of propeller diameter.
The curve of boss radius is taken with a radius equal to boss diameter $\times \cdot 8$, therefore 30 inches $\times \cdot 8 = 24$ inches radius for curve.

To find Blade Thickness.—The blade thickness, if continued to the shaft centre line or axis, is found as follows:—

RULE.—

$$\sqrt{\frac{\text{Shaft diameter}^3}{\text{Number of blades} \times \text{boss length}}} \times \text{Constant } 4 + \cdot 5 = \text{Thickness.}$$

Therefore, $\quad \sqrt{\frac{11^3}{4 \times 30 \text{ inches}}} \times 4 + \cdot 5 = 6{\cdot}5$ inches thickness at shaft axis.

NOTE.—11 inches = shaft diameter.
4 ,, = number of blades.
30 ,, = breadth of blade at boss (roughly).

The blade thickness near the tip is found as follows:—

RULE.—

Constant $\cdot 04 \times$ propeller diameter in feet $+ \cdot 4 =$ thickness.

Therefore, $\quad \cdot 04 \times 11{\cdot}5 + \cdot 4 = \cdot 86$ inch, or say $\frac{7}{8}$ inch thick near tip.

NOTE.—11·5 = propeller diameter in feet.

To find Boss Thickn —The boss thickness at position of the blades is found as follows

RULE.—

Constant $\cdot 65 \times$ blade thickness at shaft axis = thickness.

Therefore, $\quad \cdot 65 \times 6{\cdot}5 = 4{\cdot}22$ inches, or say $4\frac{1}{4}$ inches thick.

NOTE.—6·5 inches = blade thickness at shaft axis.

D BLADED CAST IRON PROPELLER

SCALE $\frac{1}{2}''$ PER FOOT

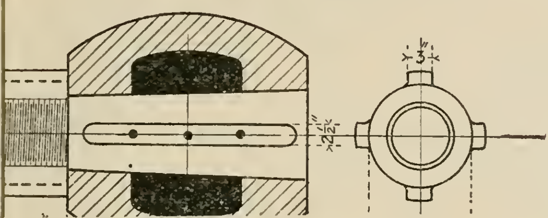
H - - - - - - - - - - - - - - - $15'-0''$

ETER - - - - - - - - - - - $11'-6''$

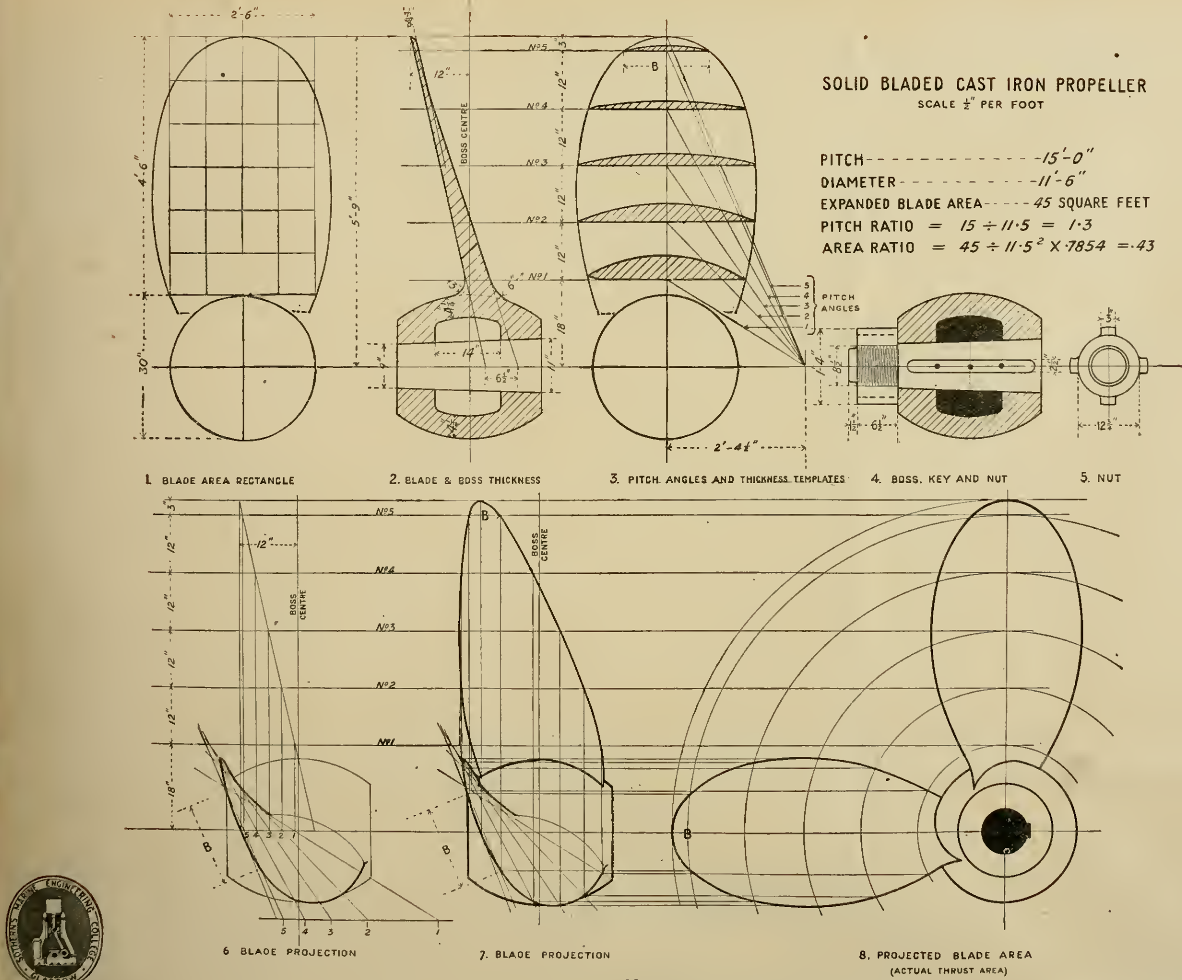
NDED BLADE AREA - - - - - 45 SQUARE FEET

RATIO $= 15 \div 11{\cdot}5 = 1{\cdot}3$

RATIO $= 45 \div 11{\cdot}5^2 \times {\cdot}7854 = {\cdot}43$



Shape of Blade.—The standard shape o blade takes the form of an llipse, but in practice various modifications of this are adopted, with nore or less satisfactory results. Experience proves that difference in lade contour affects the efficiency but slightly, provided that the area f blade is kept constant.


SOLID BLADED CAST IRON PROPELLER
SCALE ½" PER FOOT
PITCH - - - - - - - - - - - - - 15'-0"
DIAMETER - - - - - - - - - - - 11'-6"
EXPANDED BLADE AREA - - - - 45 SQUARE FEET
PITCH RATIO = 15 ÷ 11·5 = 1·3
AREA RATIO = 45 ÷ 11·5² X ·7854 = ·43
PITCH ANGLES
BOSS CENTRE
1. BLADE AREA RECTANGLE
2. BLADE & BOSS THICKNESS
3. PITCH ANGLES AND THICKNESS TEMPLATES
4. BOSS, KEY AND NUT
5. NUT
6 BLADE PROJECTION
7. BLADE PROJECTION
8. PROJECTED BLADE AREA
(ACTUAL THRUST AREA)
SOTHERN'S MARINE ENGINEERING COLLEGE
GLASGOW

No. 12.

To find Taper in Boss.—Allow the taper of shaft hole in boss to be *not less* than $\frac{3}{4}$ inch per foot of length.

Therefore, $2{\cdot}5$ feet $\times {\cdot}75$ inch $= 1{\cdot}875$ inches taper,

and 11 inches $- 1{\cdot}875$ inches $= 9{\cdot}125$ inches diameter at small end, or say 9 inches.

NOTE.—30 inches $= 2{\cdot}5$ feet $=$ length of boss.

To find Dimensions of Key.—The width and thickness of the key which secures the boss to the shaft are found as follows:—

RULE.— $\dfrac{\text{Shaft diameter}}{6} + {\cdot}6 =$ width of key.

Therefore, $\dfrac{11}{6} + {\cdot}6 = 2{\cdot}4$ inches, or say $2\frac{1}{2}$ inches in width.

RULE.—

Width of key $\times {\cdot}5 =$ thickness of key.

Therefore, $2{\cdot}5 \times {\cdot}5 = 1{\cdot}2$ inches, or say $1\frac{1}{4}$ inches in thickness.

To find Dimensions of Nut.—The diameter and thickness of the nut at back of boss are found as follows:—

RULE.—

Shaft diameter at screw $\times 1{\cdot}5 =$ diameter of nut, therefore $8{\cdot}5 \times 1{\cdot}5 = 12\frac{3}{4}$ in. diam.

Shaft diameter at screw $\times {\cdot}75 =$ thickness of nut, therefore $8{\cdot}5 \times {\cdot}75 = 6\frac{1}{3}$ in. thick.

NOTE.—$8{\cdot}5$ inches $=$ shaft diameter at screw.

To find Single Blade Expanded Area.—As there are four blades, the area of one blade is found as follows:—

Total blade area $\div 4 =$ one blade area.

Therefore, $45 \div 4 = 11{\cdot}25$ square feet surface for one blade.

NOTE.—45 square feet $=$ total blade area.

To find Blade Length.—Half of the boss diameter subtracted from half of the propeller diameter will give the blade length:—

Therefore, $11{\cdot}5 \div 2 = 5{\cdot}75$ feet, and $2{\cdot}5 \div 2 = 1{\cdot}25$;

then $5{\cdot}75 - 1{\cdot}25 = 4{\cdot}5$ feet length of blade.

NOTE.—$11{\cdot}5$ feet $=$ propeller diameter.
$2{\cdot}5$,, $=$ boss diameter.

To find Width of Blade Area Rectangle.—The single blade area divided by the blade length will give the mean width of the blade area rectangle.

Therefore, $11{\cdot}25 \div 4{\cdot}5 = 2{\cdot}5$ feet width.

NOTE.—$11{\cdot}25$ square feet $=$ single blade area.
$4{\cdot}5$ feet $=$ blade length.

Shape of Blade.—The standard shape of blade takes the form of an ellipse, but in practice various modifications of this are adopted, with more or less satisfactory results. Experience proves that difference in blade contour affects the efficiency but slightly, provided that the area of blade is kept constant.

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NOTE.—6·5 inches = blade thickness at shaft axis.

To find Taper in Boss.—Allow the taper of shaft hole in boss to be *not less* than $\frac{3}{4}$ inch per foot of length.

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and 11 inches $- 1{\cdot}875$ inches $= 9{\cdot}125$ inches diameter at small end, or say 9 inches.

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NOTE.—$11{\cdot}5$ feet $=$ propeller diameter.
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NOTE.—$11{\cdot}25$ square feet $=$ single blade area.
$4{\cdot}5$ feet $=$ blade length.

Shape of Blade.—The standard shape of blade takes the form of an ellipse, but in practice various modifications of this are adopted, with more or less satisfactory results. Experience proves that difference in blade contour affects the efficiency but slightly, provided that the area of blade is kept constant.

Set Back.—Allow a "set back" of blade equal to about 1 inch per foot of propeller diameter, or say 12 inches in all, at tip.

Radial Pitch Angles and Thickness Templates.—In a working drawing the pitch angles and thickness templates at various radial distances on the blade require to be shown for the fitting up of the shop moulding templates, and as the length of blade from centre of boss to tip is equal to the radius of the circumference circle, the corresponding reduced pitch distance will be equal to the full pitch divided by $2 \times 3{\cdot}1416$; hence,

$$15 \div 2 \times 3{\cdot}1416 = 2{\cdot}38 \text{ feet, or 2 feet } 4\tfrac{1}{2} \text{ inches.}$$

NOTE.—This distance of 2 feet 4½ inches requires to be measured horizontally from the boss centre, and all lines from radial points on the blade drawn to it.

Summary of Results.

The principal dimensions as found by the foregoing rules are then as follows:—

Propeller pitch	15 ft.
,, diameter	11 ft. 6 in.
Expanded blade area	45 sq. ft.
Single blade area	11·25 ,,
Boss diameter	2 ft. 6 in.
,, length	2 ,, 6 ,,
,, taper	11 in. to 9 ,,
Breadth of key	2½ ,,
Thickness of key	1¼ ,,
Nut diameter	12¾ ,,
,, thickness	6½ ,,
Pitch angle distance	2 ft. 4½ ,,
Length of blade area rectangle	4 ,, 6 ,,
Width of ,, ,,	2 ,, 6 ,,
Set back of blade	12 ,,
Blade thickness at shaft axis	6½ ,,
,, ,, near tip	⅞ ,,
Boss thickness	4¼ ,,

To Draw the Propeller (Sketch No. 12).

The following method, it should be noted, is not mathematically correct (particularly in the case of the "projected area" view), but is quite near enough for practical purposes as required in a working drawing. For the shop moulding of a propeller the projected area view is not required.

1. **Blade Area Rectangle.**—Set off the horizontal shaft centre line and the vertical centre line of the boss, then from the centre of boss with a 15-inch radius, describe the circle of the boss diameter, 30 inches; also measure up from the shaft centre line half of the propeller

diameter, or 5 feet 9 inches. Next measure on each side of the boss centre line half of the blade rectangle width, 1 foot 3 inches, and complete the rectangle as shown. Now proceed to sketch in by hand the approximate shape of the blade, taking care that the actual surface of blade when drawn in is at least equal to the original rectangular area. A good plan is to divide off the blade area rectangle into a number of divisions, horizontally and vertically, counting up the total number, and after the blade is shaped out as desired, arrange that the actual area of blade contains the same number of divisions.

2. **Blade and Boss Thickness.**—Set off the length of the boss, 30 inches, and the diameter of the boss, also 30 inches, then complete the boss outside curve with a 24-inch radius. Next draw in the taper of the hole in the boss 11 inches to 9 inches. Then set off aft at the propeller tip the "set back" of 12 inches, and draw a line for the face of the blade through the boss curve at the vertical centre line, and from where this inclined line cuts the shaft centre line measure forward the thickness of blade at shaft axis ($6\frac{1}{2}$ inches). At the blade tip also measure forward the thickness at that position, that is $\frac{7}{8}$ inch, and draw a line *parallel* to the face of blade line. To complete the blade section, draw another line from the thickness at shaft axis ($6\frac{1}{2}$ inches) to the tip of propeller, then run the two thickness lines into each other by a suitable curve as shown in the drawing, and join the blade at the root to the boss at the forward side by a large fillet of, say, 6 inches radius. The hollow cast part of the boss is shown as 14 inches in length, as this leaves sufficient strength of metal fore and aft.

3. **Pitch Angles and Thickness Templates.**—Begin this view by again drawing in the complete boss and blade as in view 1, and set off any convenient number of divisions from the beginning of the blade radius at root. The first is at 18 inches from the centre, and the others, Nos. 2, 3, 4, 5, are at equal distances of 12 inches. Draw horizontal lines, and where these lines cut through view 2, the thickness of blade at each division will be found. Transfer the various thicknesses to view 3, and complete the thickness sections by drawing in radial curves for the back of the blade or forward surface. Observe that near the root the after edges are slightly rounded away to allow a free flow of water past the blade. Next measure to the right the pitch angle distance of 2 feet $4\frac{1}{2}$ inches, and from each radial point draw lines to it as shown; this gives the required pitch angles at the differentradial positions marked.

4. **Boss, Key, and Nut.**—This view will be easily drawn, as the dimensions are simply taken from the "Summary of Results," and measured off. Observe that the screw for the nut is *left-handed* for a right-hand propeller, to keep the propeller hard up when the shaft is revolving.

5. **Nut.**—The nut shown has four small projections 3 inches wide, to allow of the nut being screwed on or off.

6. **Blade Projection.**—As usual set off the propeller radius of 5 feet 9 inches, and the various divisions, 1, 2, 3, 4, 5, from view 3, and draw long horizontal lines. Now complete the external view of the boss and the blade angle centre line at the set back of 12 inches from tip. At each of the divisions marked, Nos. 1, 2, 3, 4, 5, drop down vertical lines from each intersection of the blade angle line with the horizontal division lines, and where these vertical lines cut the shaft centre, set off the corresponding blade angle at that position, which requires to be transferred from view 3, so that at position 1 on shaft centre the angle is that marked 1 on view 3; at position 2 the angle is that marked 2 on view 3, and so on for each of the five radial points, which gives the projected lines of the horizontal blade as seen looking from the starboard side. On each of these angle lines set off the half width of blade, measured from view 3, and, taking No. 2 as an example, note that the width across B in view 3 is the same as B in view 6. Repeat this blade width measurement at each of the radial positions 1, 2, 3, 4, 5, and then through the points so found draw in, first by hand and afterwards with a "French curve," the blade contour.

NOTE.—" French curves" are wooden shapes used in drawing when the contour required does not readily allow of the use of radial curves, and are very convenient for use in propeller design. " French curves" are obtainable at shops where drawing instruments are sold, and the purchase of one or two is recommended.

7. **Blade Projection.**—In this view transfer the previous view of the boss and blade complete, and run up lines from the various blade widths at the radial points 1, 2, 3, 4, 5, and observe where the vertical lines so drawn cut the corresponding radial horizontal lines; mark these points with a dot or a small cross, and if this is done for each of the varying widths of blade, the curve can then be drawn in by hand, and afterwards (as before) completed more carefully with the " French curves." Notice that the width of blade at B (No. 5 radial line) in the vertical view of the blade is run up from the horizontal view of the blade also marked B, and each width is similarly treated.

8. **Projected Blade Area.**—This view is fairly simple in construction and requires very little explanation. Set off the various circles of the tip of blade, radial points of blade and boss, then project over, by lines from view 7, the various widths of the blade at the same radial points 1, 2, 3, 4, 5, and where these lines cut the radial curves of view 8, mark small dots or small crosses; if these points be then connected by hand curves, the approximate shape and width of the blade, looking from aft forward, will be shown. After completing the horizontal blade, merely transfer the widths, &c., to the vertical blade and complete it also. As before mentioned, in speaking of other views, the blade contour can be better finished by the use of "French curves." Observe that width B on radial line 5 of view 7 corresponds to width B on radial curve 5 of view 8.

To Fit on a New Propeller Blade.—If, in a loose bladed propeller, a blade is knocked off at sea, the spare blade can be placed at the correct pitch angle by the following method:—With the steamer in dry dock turn the engines until one of the remaining blades is in an upright or vertical position. Then with a straight-edge placed against the stern post, and at a convenient radius, R, mark the blade at the leading edge A; shift the straight-edge to the other side of the blade, and mark the following edge B. Now turn the shaft round until the surface of the boss to receive the spare blade is in position, and when the new blade is placed on the boss turn round the flange until the leading and following edges coincide with the marks on the straight-edge, which in the meantime must be held or fixed up against the stern post at the *same radius* R. The new blade will then have the correct pitch angle, and the studs may be screwed up.

NOTE.—The foregoing method is necessary when the stud holes in the flange are cut oval to allow of pitch variation.

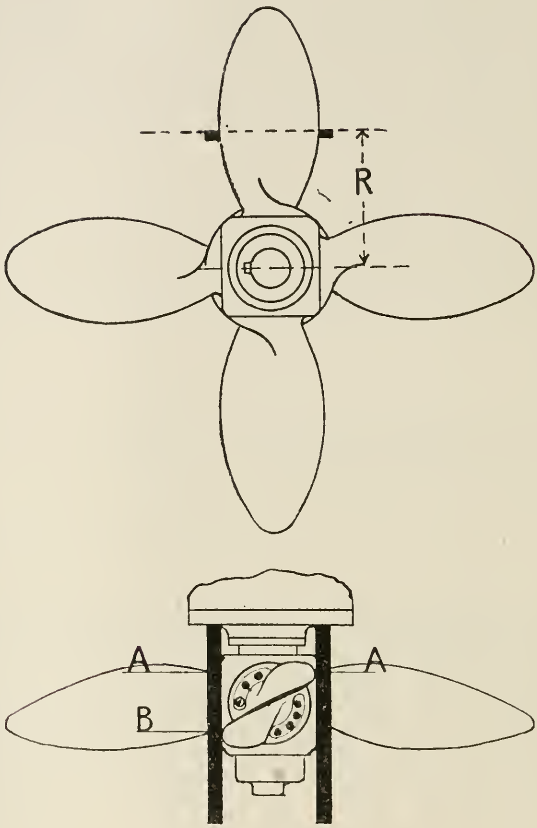
To find the Pitch when the Propeller is in the position shown (on the surface table or shop floor).—Fit up two set-squares and a horizontal piece of wood as shown, at a suitable radius R (say about two-thirds out from boss to give average pitch); then the piece of pitch P will be obtained by the vertical measurement, and the piece of circumference C by the horizontal or floor measurement.

And $R \times 2 \times 3{\cdot}1416 =$ full circumference at radius R.

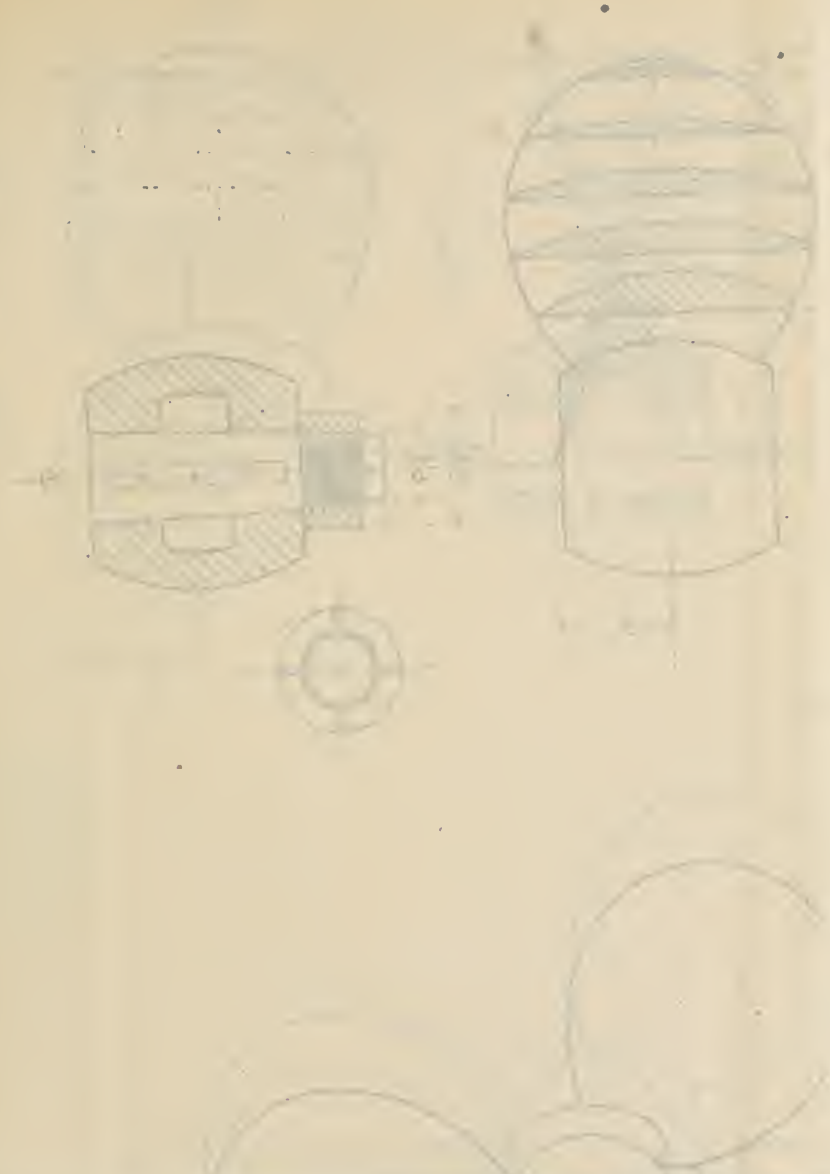
To find corresponding pitch—

As C : full circumference : : P : full pitch.

NOTE.—If the blades have a varying pitch, repeat the above at two or three radial positions, and take the mean of the two or three pitches so found as the average pitch.

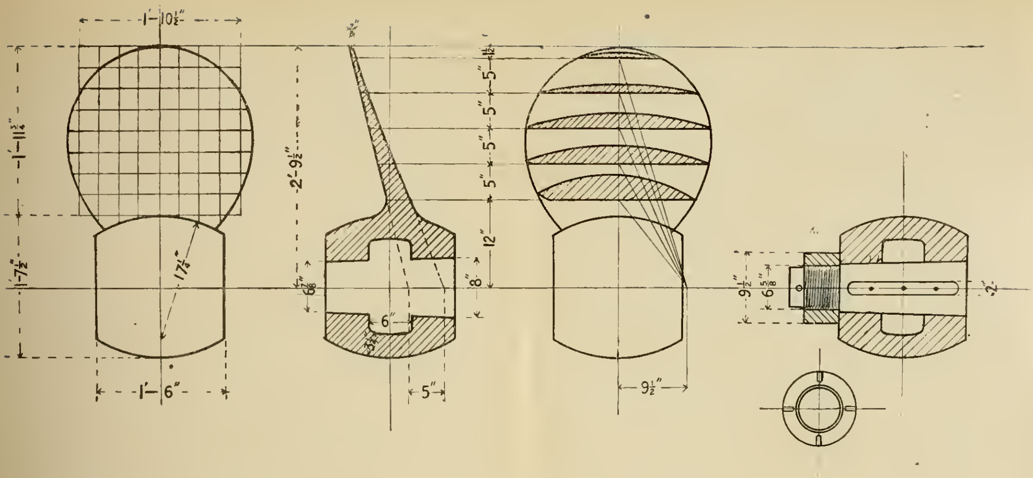


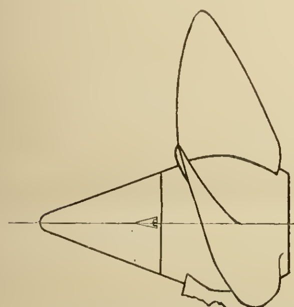
No. 13.—Fitting on a New Blade.

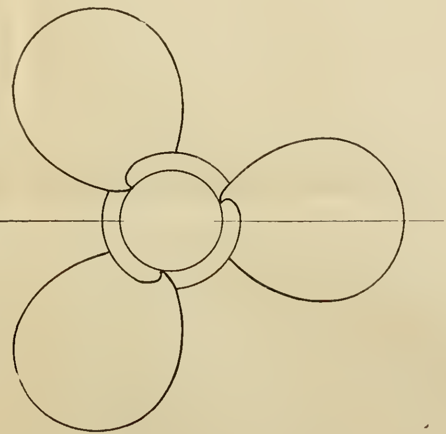


handle. It will thus be seen that the engines always run in the same direction.

Blade Interference.—By this is meant the effect produced by one blade on another blade acting on the water in such a manner as to break up the surface, and thus cut out or rob the next blade of its







TURBINE TYPE PROPELLER

SCALE ¾″ PER FOOT

PITCH . . . . 5′–0″

DIAMETER . . . . 5′–7″

EXPANDED BLADE AREA . . . 11·8 SQUARE FEET

PITCH RATIO . . . . ·89

AREA RATIO . . . . ·48

No. 15 19.

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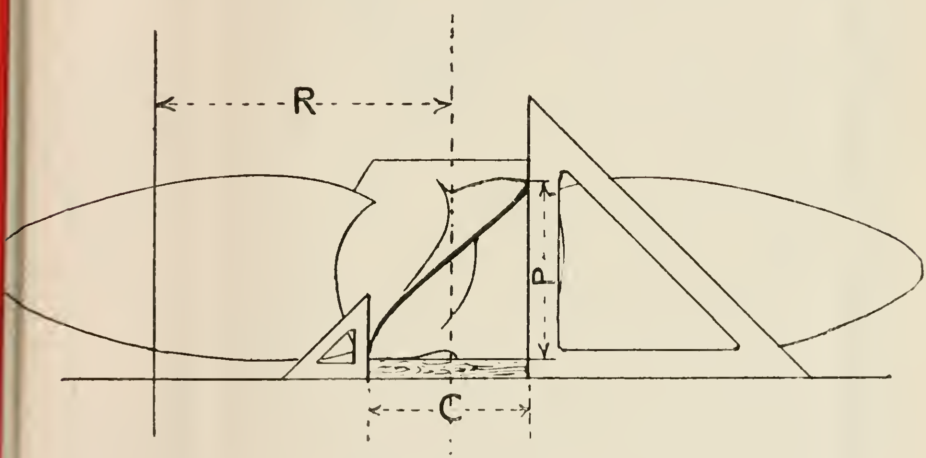
Motor Launch Propellers.—In oil motor launches the two-bladed propellers usually fitted are often of the reversible kind, that is, the blades are so arranged that their angles can be changed as desired, to "stop" or "astern" or "ahead." In the "stop" position the blades lie at right angles to the centre line of the ship, and for "ahead" the blades are moved round with the leading edge forward, while for "astern" the blades are turned round with the leading edge aft. These



No. 14.—Pitch with Propeller Horizontal.

changes in blade position are obtained by means of a rod passing through the hollow tail-end shaft, the rod being operated by a suitable handle. It will thus be seen that the engines always run in the same direction.

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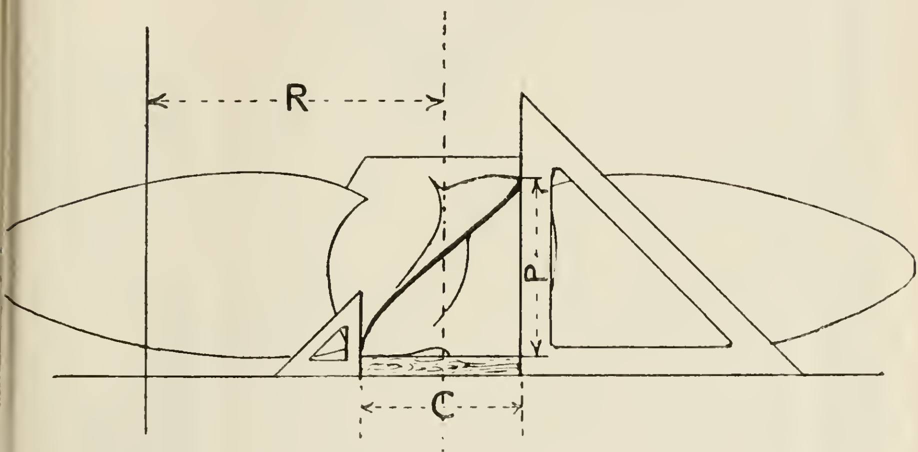

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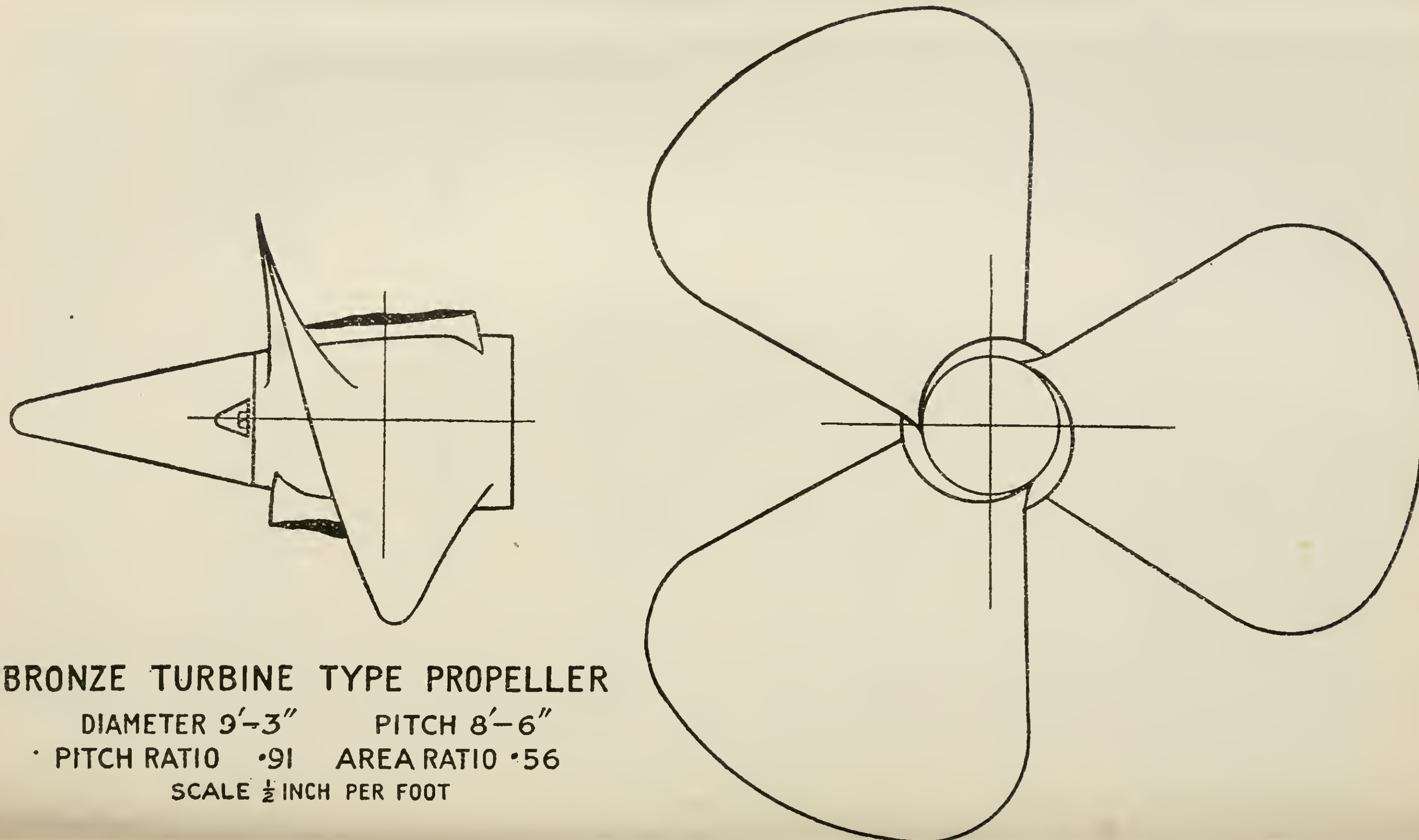


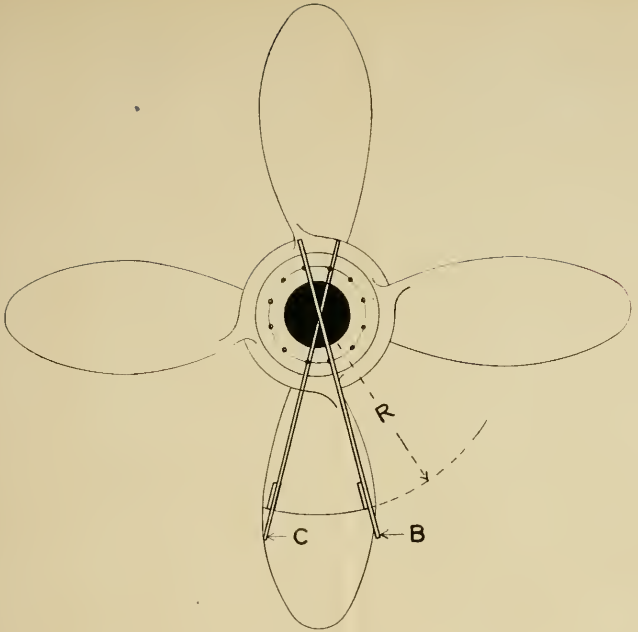
BRONZE TURBINE TYPE PROPELLER

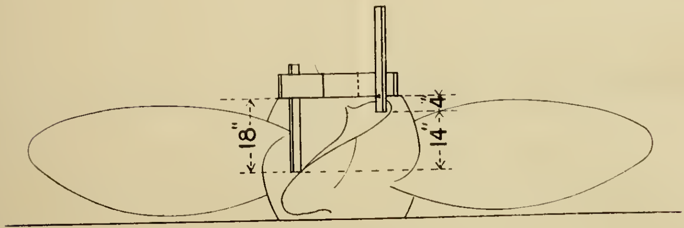
DIAMETER 9′-3″ PITCH 8′-6″

PITCH RATIO ·91 AREA RATIO ·56

SCALE ½ INCH PER FOOT

Improved Propeller.—A case which came under the writer's notice gave the following data, and clearly indicates the fact that propeller design, in many cases, is more or less a matter of "trial and error,"





Pitch of Propeller.

To measure the pitch of a propeller lying on the floor as shown, proceed as follows:—

1. Mark off the face of the boss into twelve equal divisions.

2. Take a long straightedge, and place it in line with two oppositely placed divisions, as at position B.

3. Take another straightedge (or plumb line) and measure the distance down from the long straightedge to the blade surface at a radius R, taken at, say, $\frac{2}{3}$ out from the boss centre.

4. Shift the long straightedge through *one division* of the boss marks (or $\frac{1}{12}$), as shown at position C, and again measure down to the blade surface.

5. Subtract the two measurements in inches, and the difference is the mean pitch *in feet*.

In the sketch shown the first measurement is 4 in. and the second measurement 18 in.

Therefore, 18 inches − 4 inches = 14 feet pitch.

It will easily be seen that the difference of the measurements is equal to $\frac{1}{12}$ of the pitch only.

Therefore, 14 inches × 12 inches = 168 inches pitch, and $\frac{168}{12}$ inches = 14 feet.

The 12 above the line and the 12 below the line cancel out in all cases, thus leaving the difference *in inches* exactly equal to the pitch *in feet*.

NOTE.—The above method can also be applied with the propeller in the usual position on the shaft.

[*To face page* 532.

effective thrust. For this reason an improvement is sometimes effected by changing a four-bladed propeller for one of three blades, and, it may be noted, that this change has been made in the case of the turbine steamer "Victorian."

Blade interference has not, as yet, been fully investigated by experiment, and is therefore at present largely a matter of speculation.

Twin Screws.—In twin-screw steamers it has been observed that in most cases the starboard engine runs at a higher revolution speed than the port engine, and this may be due to some effect of blade interference reducing the efficiency of one of the propellers.

Effect on Steering.—The steering is found to be improved if in twin-screw steamers the propellers revolve outwards from each other, instead of inwards, that is, the port engine propeller to be a left-hand screw, and the starboard engine a right-hand screw. The propellers also develop a more effective thrust, as the blades work in unbroken water.

Surface of Blade.—It has recently been proved beyond doubt that a polished blade surface increases the efficiency of the propeller. This is fully recognised in turbine propulsion, as nearly all propellers fitted to turbine steamers have highly polished surfaces.

In a case which came under the writer's notice, an increase of one knot was obtained, for the same power and consumption, by changing a cast-iron propeller for one of polished bronze.

NOTE.—If the propeller pitch is increased by, say, 1 or 2 feet, the mean pressure of the indicator diagrams will be more if the engine develops the same power with reduced revolutions, and the ship runs at the same speed.

Bronze Propeller Blades and Corrosion.

With bronze propeller blades the chemical conditions are much like those of an electric battery, the steel stern posts and hull forming one electrode, the bronze propeller blades the other, and the salt water the solution of the battery. As the propeller is the positive terminal and the ship the negative terminal, the current flows from the ship to the propeller, causing pitting of the steel hull. In order to overcome this galvanic action, zinc plates are fitted, in single-screw ships to the after face of the stern posts, and in twin-screw ships fitted round the after propeller bracket. This arrangement does not prevent galvanic action, but such action takes place between the propeller blades and the zinc plates instead of between the propeller blades and the steel plates of the steamer.

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Improved Propeller.—A case which came under the writer's notice gave the following data, and clearly indicates the fact that propeller design, in many cases, is more or less a matter of "trial and error,"

as notice the great improvement effected by the increase of blade area and of pitch, and the alteration in blade material, shown as follows:—

With Original Propeller.

Diameter, 18 feet 6 inches.
Pitch, 18 feet 6 inches.
Expanded blade area, 98 square feet.
Revolutions, 72 to 75.
Cast steel blades.
Consumption, 52 to 55 tons per day.
Speed, about 11·5 to 11·75 knots.
Slip per cent., 15 to 30 per cent.
I.H.P., about 3300.

With New Propeller.

Diameter, 18 feet 6 inches.
Pitch, 20 feet.
Expanded blade area, 104 square feet.
Revolutions, 64 to 66.
Bronze blades (polished).
Consumption, about 40 to 45 tons per day.
Speed, 12 knots.
Slip per cent., 5 to 10 per cent.
I.H.P., about 2800.

From the foregoing it is evident that the original propeller was not absorbing the full power of the engines, and the alteration made in increased pitch and surface utilised more effectively in propulsive effort the I.H.P. developed. Notice that the consumption, and therefore the I.H.P., developed is less in the second case, and the speed rather more.

The above rather striking results were therefore obtained by—

(1.) Increasing the pitch, with slight pitch variation.
(2.) Increasing the expanded blade area.
(3.) Changing the material of the blades from steel to that of *polished* bronze, which allows of thinner blades. The polishing of the blade surface and thinning down of the thickness both contribute to increased propeller efficiency.

It is, of course, difficult to accurately estimate how much each of the foregoing alterations contributed individually to the resulting general improvement in propulsive efficiency.

Propulsive Efficiency.

Of the total I.H.P. developed by the engines only about 50 per cent. or thereabout is applied in the effective advance of the steamer when the various losses are eliminated—that is, the actual hull resistance in lbs. at any given speed, multiplied by the advance of the hull per minute and divided by the constant 33,000 foot-pounds, will give the effective horse-power, or, as usually expressed, the E.H.P. Therefore, E.H.P. ÷ I.H.P. = Propulsive efficiency. This efficiency can only be accurately determined by model tank experiments of resistance, after which the data so obtained is converted into terms of the actual hull by a series of calculations known as the "law of comparison," and devised by the late Dr Froude. The tank experiments with the reduced scale hull models obtain progressive "tow rope" resistances which are the actual resistances at various speeds of the model hull (the propeller being omitted). These are made up as follows:—

Resistance.—1. Skin frictional resistance of the hull surface. 2. Wave making resistance of the hull body. 3. Eddy making resistance of the hull body.

The foregoing constitute what may perhaps be termed the true resistances, and to overcome these the effective horse-power is required.

Power Losses.—The losses of engine power are made up as follows:—

1. Friction (initial and load).
2. Propeller inefficiency.
3. Hull inefficiency.

The frictional losses are those occasioned by the working parts and the power absorbed by the thrust block. The propeller losses are due to excessive slip, blades friction, and other causes, and the hull efficiency is a result which may either be under or above unity, according to the difference between what is called "augmentation of resistance," due to the propeller blades at the stern, and "wake speed gain." Generally, however, the "wake speed gain" balances the augment of resistance to within a very few per cent., although an allowance of about 95 per cent. is often taken as the "hull efficiency."

The "wake speed" is produced by the water closing in on the stern as the hull advances, and this body of water acquires a forward motion or speed varying in degree with the lines of the hull body.

Utilisation of Power.—The total I.H.P. developed by the engines is therefore used up somewhat as follows, although it must be understood that the values given vary in different cases and under different conditions in the same case:—

Taking the total I.H.P. as 100 per cent.

Reciprocating Engines.

Indicated horse-power - - - - -	100 per cent.
Engine friction loss - - - - -	10 ,,
Horse-power at propeller - - - -	90 ,,
Propeller efficiency, 62 per cent.	
Then, $90 \times \cdot62 = 55\cdot8$,,	
Horse-power by propeller - - - -	55·8 ,,
Hull efficiency, 95 per cent.	
Then, $55\cdot8 \times \cdot95 = 53$,,	
Effective horse-power - - - - -	53 ,,

Therefore, propulsive efficiency $= \frac{53}{100}$, or ·53.

Turbine Engines.

Shaft horse-power - - - - - -	100 per cent.
Propeller efficiency, 60 per cent.	
Then, $100 \times 60 = 60$,,	
Horse-power by propeller - - - -	60 ,,
Hull efficiency, 95 per cent.	
Then, $60 \times 95 = 57$,,	
Effective horse-power - - - - -	57 ,,

Therefore, propulsive efficiency $= \frac{57}{100}$, or ·57.

The following explanations of propeller slip are taken from a paper on "Screw Propellers," read by T. Sidney Cockrill, Esq. M.I.Mech.E., before the Liverpool Engineering Society in April 1907 and are well worthy the attention of students.

"**Slip.**—The slip is the difference between the speed of advance of the propeller (supposing it to be working in an unyielding substance) and the actual speed of the ship. In other words, it is equal to the pitch multiplied by the revolutions, less the distance traversed by the ship. If the water did not yield to the propeller and flow sternward, the speed of the ship would be the same as the speed of the propeller, and there would be no such thing as slip; but water, being a fluid, is driven astern by the action of the propeller as the ship moves ahead. The rate at which the water is driven astern relatively to the surrounding water is usually said to be equal to the slip; but this is only true provided the pitch multiplied by the revolutions is equal to the speed of the race relatively to the ship, or, in other words, provided the propeller itself does not slip in the race. As far as the author can see, we have no means of ascertaining the truth of this.

"The above, however, only relates to apparent slip, for it does not take account of the fact that the propeller is not working in still water, but in water in motion in a forward direction owing to the influence of the ship in passing through it; and, as it is the propeller that is under consideration, the speed of the propeller and not the speed of ship through the water should be the basis in calculating the slip.

"For example, if the pitch were 15 feet, the revolutions 81 per minute, and the speed of ship 11 knots:—

Then,	Speed of propeller $=\frac{P \times R \times 60}{6080}=\frac{15 \times 81 \times 60}{6080}$	$=12$ knots.
	Speed of ship - - - -	$=11$,,
	Apparent slip of propeller - -	$=1$ knot.

But supposing the wake to have a speed of two knots,

Then,	Speed of propeller, as before $=\frac{P \times R \times 60}{6080}$	$=12$ knots.
	Advance of propeller through the water in which it works $= V - w = 11 - 2$ -	$= 9$,,
	'Real slip' of propeller - - -	$= 3$ knots.

"It is sometimes found that a propeller seems to have negative slip, that is to say that the speed of the ship is (apparently) greater than the speed of the propeller which drives it. This negative slip is only found when the speed of the ship is taken instead of the speed of the propeller through the water; it is the apparent slip, not the real slip of the propeller.

"The phenomenon of negative slip is the cause of a vast amount of ingenious, if not always scientific, hypotheses to account for its existence. Everyone agrees that negative real slip is a physical impossibility. In the author's humble opinion negative apparent slip is also a physical impossibility, and, therefore, there is no need to account for a thing which does not exist. This is on the understanding that 'apparent slip' means $\frac{P \times R}{60} - V$ while 'real slip' means $\frac{P \times R}{60} - (V - w)$; these are the usual definitions of the terms, and they are given here to prevent misunderstanding arising from the mere meaning of words. The explanation is simple enough, for, in order to calculate the slip, the pitch P must first be known; it is one of the factors in the calculation, and, in the author's opinion, no one knows what the true effective pitch of a propeller is. What is usually taken as the pitch of a propeller is the pitch of the after face of the blade, or, rather, the mean of the various pitches found at different parts of the after faces of the blades. This is certainly not the effective pitch of the propeller with respect to its action on the water, one reason at least for this being that it takes no account of the curved back of the blade, and it is certain that this increases the true effective pitch, although no one knows exactly, or even approximately, what value is to be assigned to it. There are also other reasons for supposing that we do not know what the effective pitch of a propeller is.

"The cause of negative slip often given is that the propeller is working in the wake of the ship, and, therefore, working in water which has a forward motion.

"For example of this reasoning we may take the previous example, supposing the speed of the ship to be 13 knots, and the speed of the wake 4 knots.

Then,	Speed of propeller, as before - -	12 knots.
	Speed of ship - - - - -	13 ,,
	Apparent slip of propeller - -	− 1 knot.
And	Speed of propeller, as before - -	12 knots.
	Advance of propeller through the water = 13 − 4 =	9 ,,
	'Real slip' of propeller - -	3 knots."

SECTION IX.

REFRIGERATION.

The Ammonia Compression System.

Anhydrous ammonia has found great favour as a refrigerating medium on account of its high latent heat of vaporisation and the comparatively low pressure at which it can be liquefied. The idea involved in an ammonia refrigerating plant may be explained as follows: the same, however, holds good for machines using carbonic acid, sulphurous acid, ether, &c.:—Anhydrous ammonia, *i.e.*, ammonia free from all water or moisture, is naturally a gas. Under pressure and cooling by water it may easily be condensed to liquid form. It is almost colourless, and in appearance just like water, and weighs at ordinary temperatures about 37 lbs. per cubic foot. In this form it may be purchased in steel cylinders or drums containing from 50 to 100 lbs. in weight. If the pressure be relieved from the liquid ammonia it will quickly revaporise, producing as it does so intense cold.

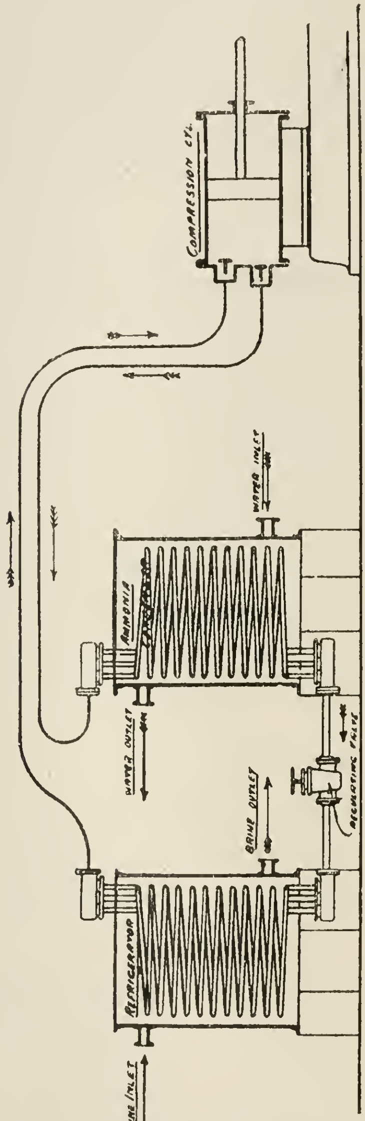
An ammonia refrigerating machine consists of the following principal parts:—

(*a.*) A **vaporiser, evaporator, or refrigerator**—a vessel in which the ammonia is allowed to vaporise, producing a low temperature and surrounded either by the air or brine to be cooled.

(*b.*) A gas pump or **ammonia compressor** which draws the ammonia gas from the refrigerator and compresses it into the condenser.

(*c.*) The **ammonia condenser** in which the gas discharged from the compressor is condensed to liquid form ready for vaporisation in the refrigerator. The diagram (No. 1) will further explain this paragraph.

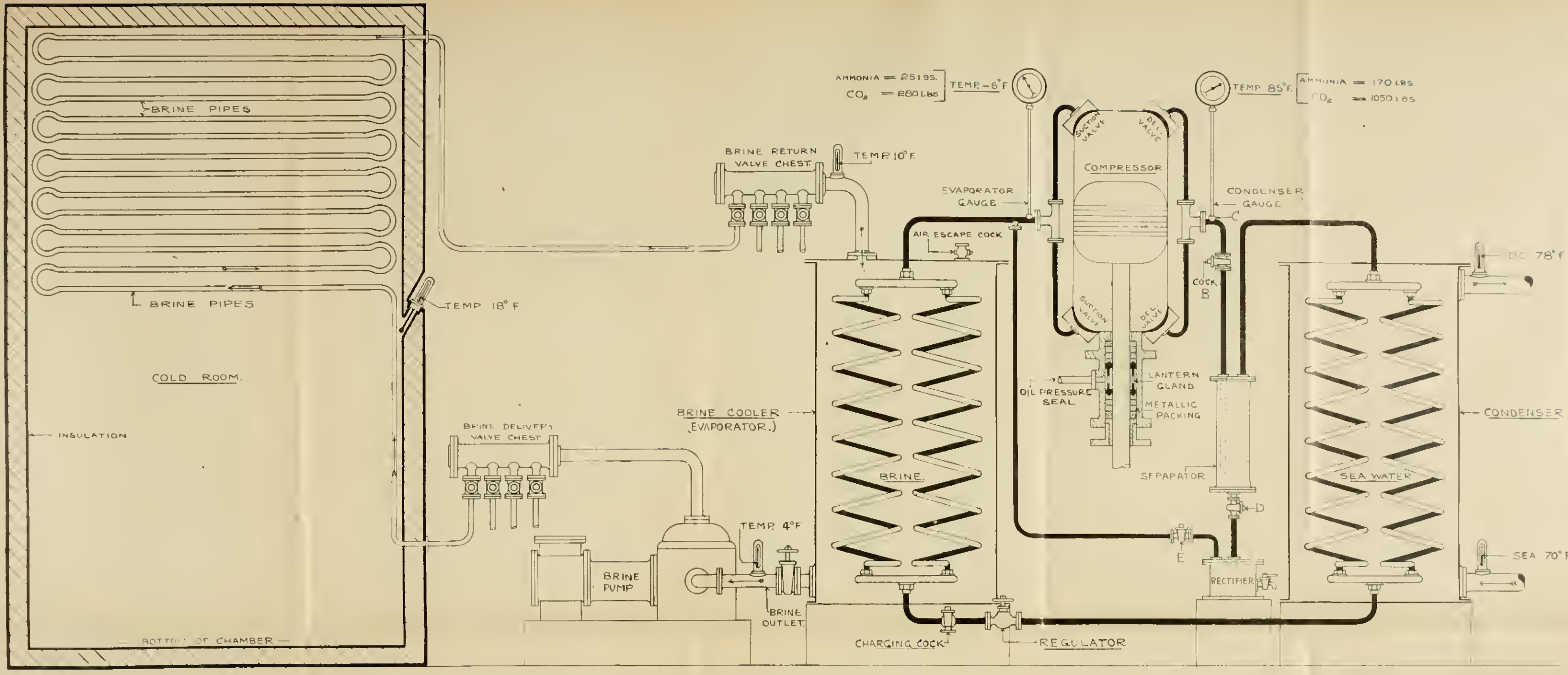
It should here be explained that the ammonia in vaporising under low pressure in the refrigerator, changing its form from liquid to gas, must absorb into itself the latent heat of vaporisation. This is taken from the brine surrounding the coils—or in the case of direct expansion, from the air itself—and in condensing to liquid form again this heat is given up to the condensing water. The pressure necessary to condensation is automatically regulated by the temperature of the condensing water, and will vary from 100 to 200 lbs. pressure per square inch. The pressure in the refrigerator is controlled by the



SECTIONAL DIAGRAM OF REFRIGERATING MACHINE

No. 1.

regulating valve, which is adjusted according to the temperature required, and varying from 70 lbs. absolute per square inch for a vaporising temperature of 40° Fahr. to 20 lbs. absolute per square inch



No. 1A Compression Systems of Refrigeration.

(Ammonia and CO_2)

CO_2 System.

Carbonic anhydride (CO_2), naturally a gas, and obtained from the following sources:—

1. Natural springs.
2. Combustion of carbon.
3. Action of sulphuric acid on calcium carbonate.
4. Fermentation of wort in brewing processes.

The gas collected from these sources is first purified and dried, then compressed in stages, next cooled to the liquid state, and finally forced into steel bottles for supply purposes. As the CO_2 is under pressure in the bottles, it remains in the liquid condition until the pressure is reduced.

When charging up the machine it evaporates back into gas by expansion, and enters the evaporation coils in that condition.

Latent heat value per lb. (at average evaporator pressures) = 125 B.T.U.
Boiling point at atmospheric pressure = −120 F
Critical temperature = 88·4 F (Latent heat falls off in value from this temperature up)

CO_2 is heavier than air

Ammonia (NH_3) System.

Ammonia, an alkali, and naturally a gas, is obtained by the heating of sal ammoniac and quicklime mixtures, or by the forced decomposition of animal substances.

The gas is collected and treated similarly to CO_2 gas as regards compression, cooling, and storing in steel bottles.

Ammonia changes red litmus paper blue, and being alkaline in nature destroys copper, brass, and leather.

Ammonia is soluble in water, one part of the latter absorbing about 800 parts of ammonia.

Latent heat value per lb. (at average evaporator pressures) = 580 B.T.U.
Boiling point at atmospheric pressure = −37 F.
Critical temperature = 256 F. (Latent heat falls off in value from this temperature up)

Ammonia is lighter than air

Average Pressures and Temperatures.

The following represent average marine refrigerating practice for a sea temperature of 70° and brine outlet of 4° F.

By the combined effects of compression and cooling, ammonia gas and CO_2 gas can be reduced to the liquid condition.

	Absolute Pressure.	Temperature Fahrenheit.	
Ammonia (NH_3)—			Cold chambers maintained at 18° to 20° for beef, and 35° to 40° for bananas and other fruits.
Brine cooler	25	−6°	
Condenser	170	85°	
Brine out	...	4°	
Brine returns	..	10°	
Sea	..	70°	
Discharge	..	78°	
CO_2—			
Brine cooler	280	6°	
Condenser	1050	85°	
Brine out	...	4°	
Brine returns	...	10°	
Sea	...	70°	
Discharge	...	78°	

Brine pressure about 10 or 15 lbs. by gauge.
" density about 40 oz. per gallon.
" composed of fresh water and calcium chloride (calcium salt).

Tests, Etc.

1. **To Test Compressor or Regulator.**—Stop machine and shut regulator; if pressure rises in evaporator gauge and falls in condenser gauge, the compressor rings are leaky, or regulator is passing gas.

2. **To Test for Shortage of Gas.**—Close down regulator, and run machine for, say, 15 revs.; if, during this period, the evaporator gauge pointer falls back it indicates insufficient gas, as the machine should hold up to the pressure if the supply is sufficient. Insufficient gas also shows by jumping of the gauge pointers, and by rise of brine temperature, loss of temperature difference, etc. If delivery pipe from compressor heats up, and still remains heated after further opening of the regulator valve, it indicates shortage of gas; the condenser gauge falling in pressure is also an indication of insufficient charging.

Temperature Differences. The following examples of difference in temperatures represent good refrigerating practice, and should be kept to as nearly as possible.

Evaporator.		Condenser.	
Brine Out Temperature.	Gas Temperature.	Sea Water Temperature.	Gas Temperature
10°	0	60°	75°
6°	−4	65°	80°
4°	6°	70°	86°
2°	8°	75°	90°
0°	−10°	80°	95°
2°	−12°	85°	98°
About 10° difference.		About 15 difference.	

To Work Rectifier System.—At intervals of, say, 30 minutes or more, open cock D, and blow out separator into rectifier chamber, then shut cock D, and open cock E, which allows the machine to draw off any ammonia which has passed into rectifier; next shut cock E, and drain off oil from rectifier into a bucket by the cock fitted for that purpose.

Air in Brine System. Air in the brine system can be got rid of by opening the air escape cock fitted on top of the evaporator chamber (shown in the sketch).

To Clear System of Air when Machine is Charged

Ammonia Machine.—After running for some time when charged, stop machine, but keep circulating, then connect a rubber tube either to condenser gauge cock fitting, previously mentioned, or to the air escape cock sometimes placed on condenser delivery valve chest, and immerse lower end of tube into a bucket of water, gently open the cock, and close same when smell of ammonia is noticed coming from the water.

NOTE.—The same method holds good for the CO_2 system, only the rubber tube connection to the water bucket is not required, as the air and gas may be safely discharged to the atmosphere

Overcharging. If the machine is overcharged it will show by a rise of the condenser gauge pressure. Air present in the system also produces the same result.

Rise of Sea Temperature.—If the sea water temperature rises the machine requires more gas, as the pressure and temperature of the refrigerant must be proportionally increased to obtain the same temperature difference and cooling effect as previously.

Oil in System.—This shows by erratic movement of evaporator gauge pointer, and variation in temperature of delivery pipe.

Air in System.—The preseoce of air is indicated by the machine falling off in efficiency, and by jumping of condenser gauge pointer.

To Clear System of Air before Charging Machine

CO_2 Machine. Break joint between condenser liquid pipe and regulator and discharge air, etc., into atmosphere.

Ammonia Machine.—Close stop cock B and disconnect gauge from delivery pipe fitting, then run machine slowly, and air will be discharged through this opening. The evaporator and condenser gauges will indicate a vacuum when the air is exhausted out of the system.

Testing of Gauges. After stopping machine, leave regulator full open and the two gauges should show identical readings, with the temperature shown corresponding to that of the liquid brine.

NOTE.—To test for leaky condenser coils or leaky evaporator coils of ammonia machines. Use red litmus paper which, if immersed in a sample of the brine, or of the circulating water, will change to blue if the coils are leaky. The air cocks on the top of the condenser and evaporator should be opened once during a watch when the machine is running, and should be left open altogether when the machine is stopped

[To face page 540]

for a vaporising temperature of minus 15° Fahr., or higher or lower as required.

Passing on now to a description of typical plants. The one described below is manufactured by the Liverpool Refrigeration Company Limited, Liverpool, and is largely used in the North Atlantic chilled beef trade. It is also equally applicable to the carriage of frozen goods, mutton, &c. As a rule, plants are fitted in duplicate. The diagram (No. 2) shows the essential parts and arrangement of this plant. We may here mention that, with few exceptions, the cold chambers on shipboard are entirely chilled by means of wrought-iron piping arranged on the ceilings, sides and ends, and through which the brine cooled in the refrigerator is circulated. The plant before us is one of this type.

Description of Plant.

The ammonia compressor (No. 2) arranged in single, and in the duplex form, is direct steam driven, and is of the horizontal double-acting type. For convenience and for saving of space, the box bed on which the engine and compressors are mounted, contains the ammonia condenser, which consists of a series of coils of wrought-iron tubing in which the ammonia is condensed, the water circulating round the outside of the tubes. After passing the condenser, the liquefied ammonia collects in the reservoir AR from whence it passes through the regulating valve RV into the refrigerator, which is of the vertical type, and contains several circular concentric coils of pipe placed in a steel shell with covers top and bottom. The ammonia vaporises inside the coils, the brine circulating round the outside. After passing through the refrigerator, the ammonia is drawn back to the pump as shown, to be compressed and discharged at the higher pressure into the ammonia condenser for recondensation. The reader will please note that the ammonia circuit is complete, and there is no loss whatever of ammonia, which goes through the cycle of vaporisation, of compression, and condensation time after time indefinitely. The condensing water is supplied either by a separate pump, or from one of the ship's donkeys, is drawn from the sea, pumped through the condenser, and overboard. The cold brine is drawn from the refrigerator by the brine pump, which is preferably one of the "Worthington" duplex or similar type, and is discharged to the distributing headers from whence, in several independent circuits, it passes through the pipes in the cold chambers, returning again to the return headers. Both distributing and return headers are fitted with controlling valves so that each brine circuit may be regulated as desired; each circuit controlling a separate portion of the cold chamber. The temperatures may thus be regulated by allowing more or less brine to pass through any particular section of piping. The thermometers index the return temperature and are useful adjuncts in the regulating of the brine. After passing through

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The following extracts from the Liverpool Refrigeration Company's Book of Instructions may be of use :—

Pressures.—All ammonia pressures are absolute. The pressure on the ammonia condenser gauge will vary with the temperature and quantity of water passing through the condenser, and should generally vary from 120 to 180 lbs. per square inch. The warmer the condensing water, the higher the pressure. The pressure on the refrigerator gauge may be regulated as desired by means of the regulating valve RV (see diagram), which should be adjusted to the brine temperature; the higher the temperature, the higher the pressure.

Evaporator Pressures.—The following table gives the approximate evaporator pressures and temperatures which should be kept to secure the best effects :—

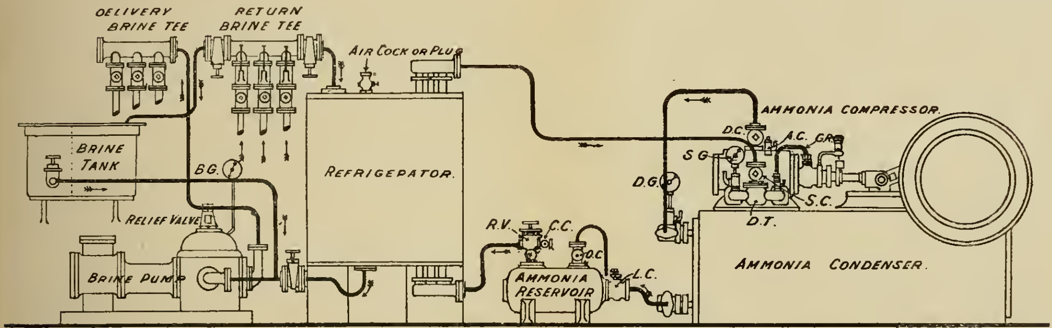
For brine temperatures of		zero	Fahr.	- - -	24 lbs.
"	"	5°	"	- - -	28 "
"	"	10°	"	- - -	32 "
"	"	15°	"	- - -	36 "
"	"	20°	"	- - -	40 "
"	"	25°	"	- - -	45 "
"	"	35°	"	- - -	55 "
"	"	50°	"	- - -	70 "

The refrigerator pressure should, while approximating to these figures, be such that the discharge pipe from the ammonia compression pump should not be warmer than can be easily borne by the hand, say roughly at a temperature not higher than 120° Fahr.; if warmer than this, open the regulating valve slightly; if colder, close same slightly. The discharge pipe of the compression pump should never be allowed to get cold or very hot, but should always be kept as stated above.

Compressor Gland. — Sufficient oil should always be kept in this for the lubrication of the cylinder, and to keep the gland ammonia tight. The packing in the gland should always be very carefully fitted; if slack and badly fitted, the oil will leak past the packing into the compressor in considerable quantities, which is undesirable. The least possible quantity of oil to keep the rods

DIAGRAM OF MARINE TYPE MACHINE SHEWING CONNECTIONS.

THE LIVERPOOL REFRIGERATION Cº LTD.



D.C. Discharge Cock on Compressor.
S.C. Suction Cock on Compressor.
D.T. Dirt Trap on Suction Branch.
A.C. Air Cock on Compressor Discharge
S.G. Ammonia Suction or Refrigerator Gauge
D.G. Ammonia Discharge or Condenser Gauge.

L.C. Ammonia Liquid Cock.
O.C. Oil Cock for Drawing off Oil
R.V. Ammonia Regulating Valve.
C.C. Charging Cock.
B.G Brine Pressure Gauge
G.R. Gland Relief Pipe

No. 2.

[To face page 542.

lubricated and the gland tight should be used; sufficient will always get through into the compressor to keep this in thorough order.

Oil Extraction.—The oil extraction cock OC (diagram No. 2) is placed on the ammonia reservoir; any oil passing through the machine will collect here. Attach a short piece of pipe with the end carried to a bucket, open the cock very gently; the cock has an internal pipe leading to the bottom of the reservoir, and the oil will be driven off by way of this pipe through the cock and into the bucket. When the machine is new and probably more oil is used, this oil should be withdrawn from the reservoir once every few days, but afterwards the intervals may be greatly lengthened. Do not open the cock carelessly.

To Charge the Machine with Ammonia.—The ammonia drum should have the end remote from the cock slightly raised; it should be connected to the charging cock CC. After this is done, close the regulating valve RV and start the compressor, water, and brine pumps. When the pressure in the refrigerator is reduced, the charging cock first, and afterwards the cock on the ammonia drum, may be opened a little, and some of the ammonia from the drum allowed to flow into the refrigerator to be pumped through into the condenser. When it is thought that sufficient has been put in, first the cock on the drum, and afterwards the charging cock, should be closed, and then if necessary the drum can be disconnected and weighed, the difference in weight before and after charging being the amount of ammonia put into the machine.

To Overhaul the Compressor.—Before overhauling the compressor, shut the suction cock SC. Run the machine a few turns to exhaust the ammonia in the pump, then stop and close discharge cock DC. The pump being now free from ammonia, may be opened and examined. After examination is complete, and before letting back the ammonia, take out the small screwed plug in the discharge valve cover, and run the machine a few turns to discharge the air from the compressor; then when no more is discharged, screw in plug and open the suction and discharge cocks.

To Make Brine.—The Calcium Chloride should be broken up into small pieces and put into the brine tank. The brine should be circulated through the brine tank by means of the pipes and valves provided. First close the suction cock on the refrigerator; the water can be added as required into the brine tank. The small air cock on the refrigerator top should be opened very frequently or left open altogether.

Density of Brine.—The density of the brine should be measured by Twaddle's hydrometer, and should be kept at from 40° to 48° to that

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Density of Brine.—The density of the brine should be measured by Twaddle's hydrometer, and should be kept at from 40° to 48° to that scale, the latter figure being for lower brine temperatures, the former for brine temperatures above 15°.

Circulation of Brine.—Each section of brine pipe should after every voyage be circulated by itself, the others being shut off. This circulation should take place through the brine tank, the refrigerator being shut off for the time being. The brine pump must be worked very slowly, and the brine pressure gauge watched during this operation Any air collected in the pipes will be got rid of by this means.

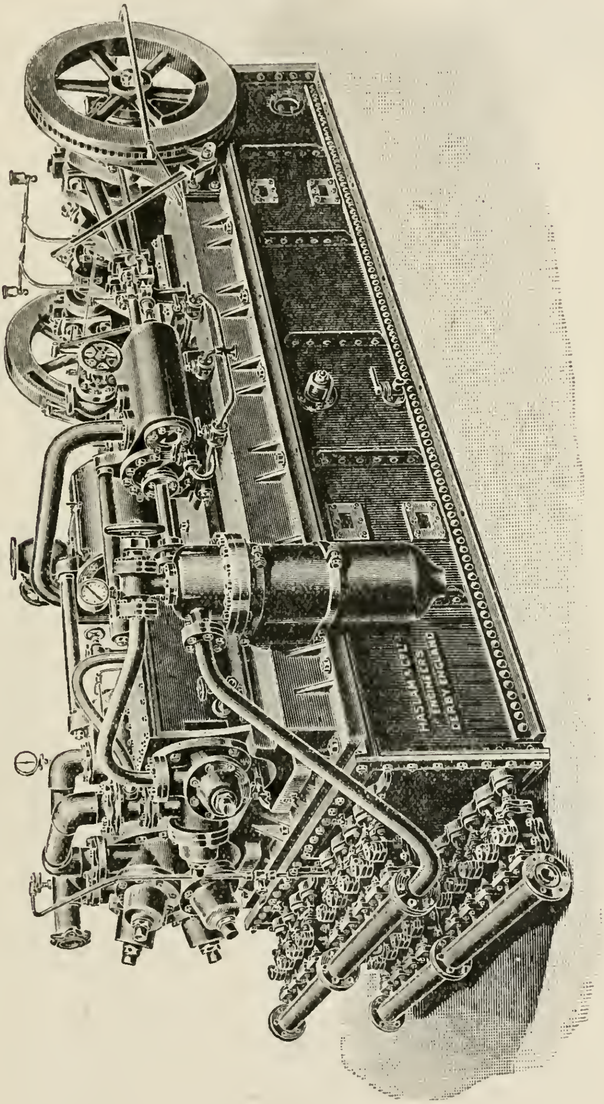
Regulation of Chamber Temperatures.—The temperature of the chamber must be regulated exactly as desired by closing or opening the gland cocks on the return tees. The engineer should make himself acquainted with the particular section of piping, and therefore of the chamber governed by each cock. The inlet cocks should be left fully open, the regulation being effected by the returns. The pressure on the brine gauge should not exceed 10 lbs. per square inch.

Air in System.—The presence of air or other gas than ammonia in the gas circuit, reduces the efficiency of the machine, and this may be detected by the machine working irregularly and by having a condenser pressure higher than that due to the temperature of condensing water. To purge of air; pump all ammonia into condenser and reservoir, stop machine, keep circulating pump going, and after standing, say, half-an-hour, remove gauge from cock on top of condenser tee and couple up small pipe provided for the purpose, the remote end of the pipe being put in a bucket of water, gently open cock and let it blow until the water begins to crackle and rise in temperature. This shows that ammonia is passing and that the air has been got rid of. If necessary, repeat the process after running the machine a few hours.

Generally.—Always take care before starting the machine that both brine and water pumps are started and working properly and all necessary cocks in ammonia circuit are open. Keep a careful eye on the pressure gauges; never let the compressor work too hot or too cold, remembering that by the regulating valve you may regulate this exactly as you wish. It is better for the cargo when thoroughly chilled down to run the machine continuously at a slow speed rather than running full speed for a short time and stopping the remainder.

Haslam Type.

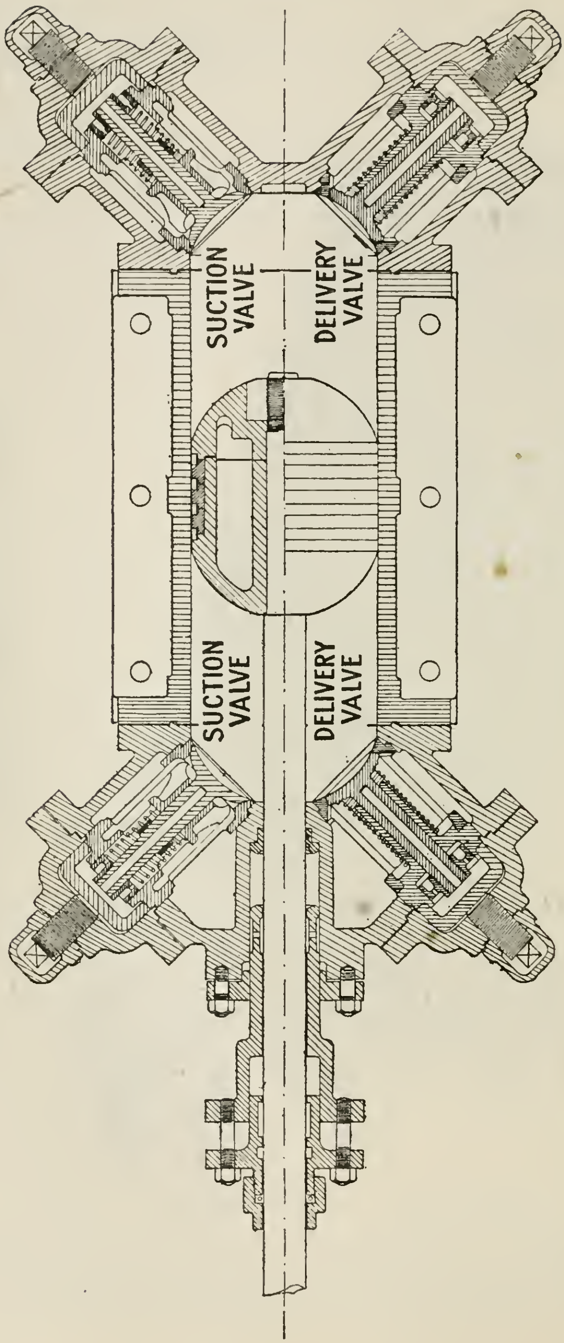
We illustrate on next page a marine type machine working on the Haslam ammonia compression system as made by the Haslam Foundry and Engineering Company Limited, of Derby. This machine has compound compressors to give greater economy when working in hot climates. These are driven by a compound steam-engine placed in front of the compressors. The lower part of the bed is a substantial wrought-iron tank, which contains the ammonia condensers—coils of galvanised wrought-iron pipes, the ends of which are fitted with cocks so that any coil can be isolated



No. 3.—Ammonia Refrigerating Machine.

(Messrs The Haslam Foundry and Engineering Company Limited, Derby.)

in case of leakage. The method of cooling the meat chambers is either by brine pipes, as has already been described, or by blowing air by means of fans over nests of coils in which the liquefied



No. 4.—Section of Ammonia Compressor.

(Messrs The Haslam Foundry and Engineering Company Limited, Derby.)

ammonia is evaporated; the cooled and dried air is then distributed throughout the meat chambers through wooden ducts or trunks.

Haslam's machinery, working on the brine pipe system, is at the present time bringing chilled beef from the River Plate, a voyage lasting thirty days, with a variation of temperature of within half a degree on each side of a fixed point, the cargo being invariably landed in perfect condition.

Haslam's double-acting ammonia compressor, of which we show a section, is fitted with suction and delivery valves in the end covers, which are made concave to give room for the valves; the piston is turned an accurate fit for the covers so that the clearance is reduced to a minimum, being in fact considerably less than if the valves were placed in the cylinder body, as is sometimes done. The special form of gland with two separate packings allows of either packing being adjusted independently of the other. The annular space between the packings is kept full of oil, and there is also a lubricator on the outer gland.

The method of working these machines is generally as has been before described.

The Carbonic Anhydride System.

The general principle of this system is similar to that of the ammonia compression machine, in regard to the cycle through which the refrigerating agent passes. This is made clear by the diagram given below, which consists of the following parts:—

1. The compressor (the only moving part), in which the gas drawn from the evaporator is compressed.

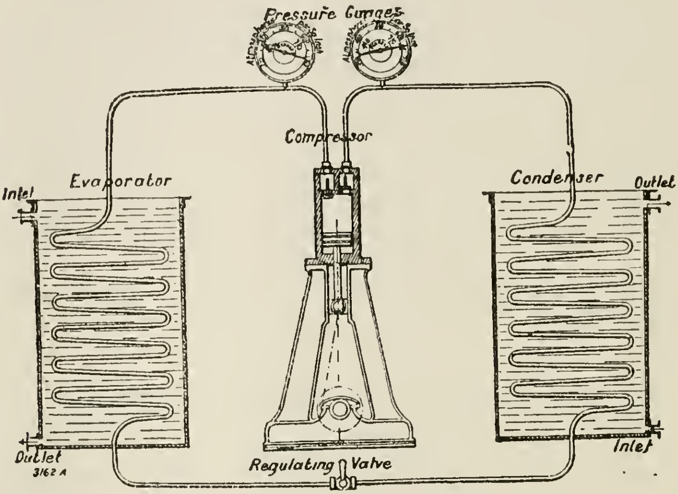
2. The condenser, consisting of coils, in which the compressed warm gas is cooled and liquefied by the action of cooling water.

3. The evaporator, consisting of coils, in which the liquid carbonic anhydride evaporates, producing any degree of temperature that may be required, down to 80° below freezing point.

We are indebted to Messrs J. & E. Hall Limited, Dartford, for the following description of their refrigerating machinery:—

The difference between the Hall carbonic anhydride machine and the ammonia machines lies chiefly in the employment of another refrigerating agent. Carbonic acid is gas which liquefies under a pressure of about 50 atmospheres under temperate conditions and about 75 atmospheres in the tropics, and the parts of the machine are constructed of sufficient strength to stand these pressures, and moreover are tested to 3-4 times the working pressure, as will be mentioned below. The efficiency of the refrigerating agent is found in these machines to be very high, and the reduction of cooling effect due to higher temperatures of condensing water is found to be, in practice, about the same as that which occurs with other types of refrigerating machines.

The charge of carbonic anhydride originally put into the machine is used over and over again, going progressively through the processes of compression, condensation, and evaporation, passing through a closed cycle. Thus a small quantity only is required to be added from time to time to replace any small losses, and for this purpose carbonic acid is sent in steel cylinders to any part of the world. The cost of



No. 5.—Diagram of Messrs J. & E. Hall's Carbonic Refrigerating Machine.

the material is only a few pence per pound. The quantity required for a complete charge is very small, the cost of a charge for a 24-ton ice plant being only about £7.

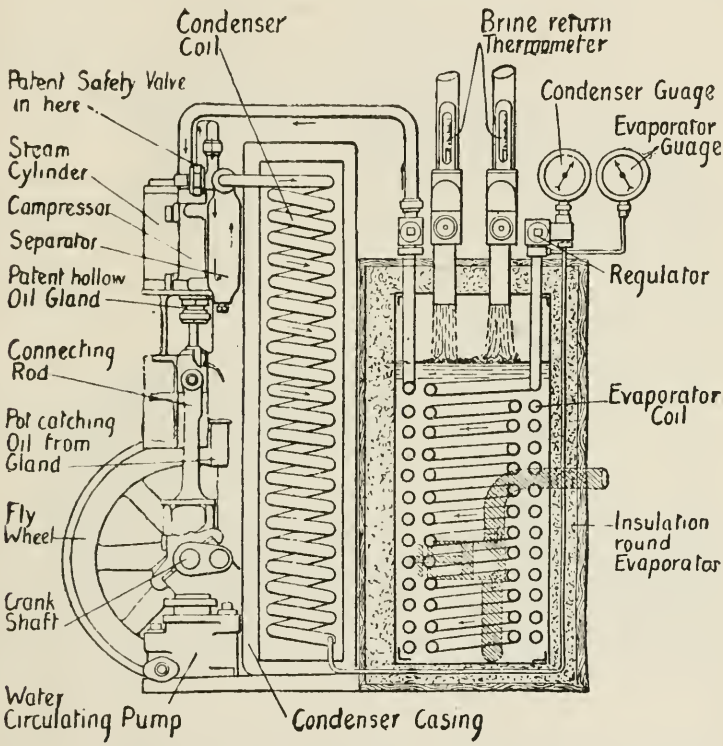
Properties of CO_2.—Carbonic anhydride is a non-poisonous gas, and a constituent of atmospheric air. To give an idea of the freedom from danger of J. & E. Hall's patent refrigerating machines, it may be stated that the entire contents of the machine might be allowed to escape into an ordinary engine-room without any disastrous results or, in most cases, even inconvenience.

Though it is not contended that an atmosphere containing only carbonic acid will support life, on the other hand, it has been found by careful experiments made by well-known scientists, that men can breathe fairly comfortably in air containing as much as 15 per cent. of carbonic acid, in which atmosphere one of the investigators remained for three-quarters of an hour. Now the entire charge in the machines usually fitted on board ship by Messrs J. & E. Hall Limited is such that the atmosphere would not contain so large a proportion of carbonic acid even if the whole charge escaped instantaneously from the machine into the main engine-room.

The details of the machines are as follows :—

Description of Hall Plant.

Compressor.—The compressors for the large machines are bored out of solid steel forgings, partly to secure strength, but principally on



No. 6.—Section of Hall Vertical Marine Type Machine.

account of greater certainty of soundness of the material, and to provide a perfect bore in which may work the cup leathers with which the pistons are provided.

Compressors of smaller machines are cast in a special bronze, which secures the two essentials of soundness and hardness. The suction and delivery valves are identical for facilities of interchange.

Gland.—The gland is made gas-tight by means of two cupped leathers on the compressor rod. A special lubricating oil is forced into the space between these leathers at a pressure superior to the greatest pressure in the compressor, so that whatever leakage takes place at the gland is a leakage of the oil either into the compressor or out into the atmosphere, and not a leakage of gas. What little leakage of the oil takes place into the compressor is advantageous, inasmuch as it in the first place lubricates the compressor, and in the second place it fills up all clearances, thereby increasing the efficiency of the compressor.

In order to replace the special oil which leaks out of the pressure lubricator, there is a small hand hydraulic pump, a few strokes of which are required to be made every two or three hours, as may be indicated by the position of the pressure lubricator piston rod. This form of gland is now in constant use on nearly 2000 machines supplied by J. & E. Hall Limited.

Separator.—Any oil which passes into the compressor, beyond what is necessary to fill the clearance spaces, is discharged with the gas through the delivery valves. In order to prevent this passing into the condenser coils, all the gas is delivered into a patent separator. The oil drains to the bottom of this vessel, whence it is drawn off from time to time; meanwhile the compressed gas passes off by an opening at the top on its way to the condenser.

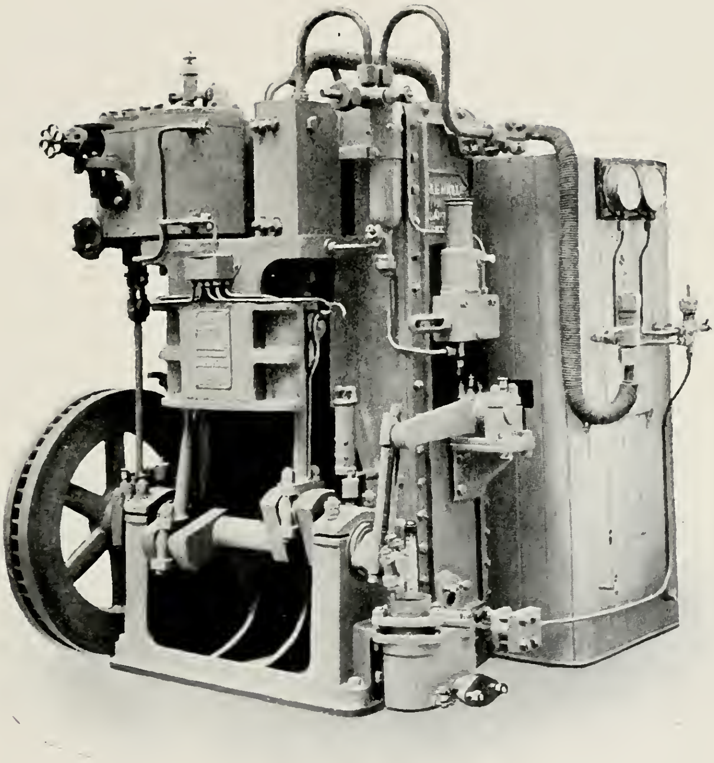
It may here be remarked that the oil has no affinity for carbonic anhydride, hence it undergoes no change in the machine, and there is, therefore, no fear of the coils becoming clogged by any small amount of oil which might be carried over in spite of the separator.

Condenser.—This consists of coils of copper tube, which are placed in a tank and surrounded by water, similar to the condenser of a steam-engine, with, however, the gas *inside* the tubes and the cooling water outside. These coils are welded together into such length as to avoid altogether any joints inside the tank, where they would be inaccessible. The welding of these pipes is all done at J. & E. Hall's works by the electrical method, which gives very good and reliable results.

In connection with the condenser, one very important advantage of carbonic acid machines is apparent, for as carbonic acid has no chemical action on copper, in the numerous cases where sea water only is available for condensing purposes, that metal is used in the construction of the coils.

Evaporator.—This also consists of nests of wrought-iron hydraulic pipes electrically welded up into long lengths, inside of which the carbonic anhydride evaporates. The heat required for evaporation is usually obtained either from brine surrounding the pipes, as in cases where brine is used as the cooling medium, or else from air surrounding the pipes, as in cases where air is required to be cooled direct.

Between the condenser and evaporator there is a regulating valve



VERTICAL MARINE TYPE CO_2 MACHINE.
By Messrs J. & E. HALL, LIMITED.

for adjusting the quantity of the liquid carbonic anhydride passing from the condenser.

J. & E. Hall's Patent Safety Valve.—In order to enable the compressor to be opened up for examination of valves and piston without loss of carbonic anhydride, it is necessary to fit a stop-valve on the suction and delivery sides, so as to confine the carbonic anhydride to the condenser and evaporator. It is, of course, possible for a careless attendant to start the machine again without opening the delivery valve, and in such case an excessive pressure would be created in the delivery pipe, from which there would be no outlet. To provide against this danger a patented safety device is adopted, consisting of an ordinary spring safety valve, at the base of which is a thin copper disc, which is designed to burst at a pressure considerably below that to which the machines are tested. This disc is made perfectly gas-tight, an object which could not be attained by the spring safety valve alone, and the latter only comes into play when the disc is ruptured. Great care is necessarily exercised in making the discs to provide against variation in strength, due to any variation either in the thickness or hardness of the copper sheets out of which the discs are made.

Joints.—With regard to the joints to withstand the pressures, those which are not subject to a high temperature can be made absolutely tight with any suitable material, such as leather, but for the hot joints, special rings are supplied which withstand the heat and still have the necessary elasticity to ensure the joint being perfectly tight when either hot or cold. The absolute tightness of all joints is effectually tested by brushing them over with soap and water, the slightest leak being thereby detected.

Testing Parts.—Very careful tests are carried out in J. & E. Hall's works to ensure perfect soundness of all parts subject to the gas pressure. The working pressure varies from about 750 lbs. per square inch in temperate climates, with water at 50° Fahr., to about 1125 lbs. with water at 84° to 90°, as is usual in the tropics. Owing to the very small diameter of all parts, even in large machines, there is no difficulty in securing a very ample margin of strength. All parts of machines subject to the pressure of the carbonic anhydride are, in the first place, tested for strength by hydraulic pressure to 3000 lbs. per square inch, and they are then again tested while immersed in warm water by air to 1350 lbs. per square inch, whereby the slightest porosity which might exist in any of the materials is at once detected by air bubbles ascending through the water.*

* Extract from Paper read before the British Association by Mr E. Hesketh, M.Inst.C.E., M.I.Mech.E., Managing Director of J. & E. Hall Limited. 14th September 1895.

Refrigeration.—As a refrigerating agent, liquefied carbonic acid is second to none. Under atmospheric pressure it evaporates from the liquid state at the particularly low temperature of 120° Fahr. below zero, or 152° below the freezing point of water. In J. & E. Hall's refrigerating machine, however, it is caused to evaporate at only a few degrees below the temperature of the material which it is desired to cool, the principle of the machine being exactly the same as that of machines using anhydrous ammonia on the compression system—viz., as water boils at 212° Fahr. under atmospheric pressure, and about 250° Fahr. at 15 lbs. pressure, fire being usually the source of heat, so liquid carbonic acid boils or vaporises at 30° Fahr. at 35 atmospheres' pressure, and thus permits cold water or colder brine to be the source from which the necessary heat to boil it is absorbed, exactly in the same manner as the heat of the fire is absorbed in boiling water.

The compressor draws the gas or vapour from the evaporator and compresses it to the liquefying pressure, which is controlled within certain limits by the temperature of the cooling water. The heat due to compression is absorbed by the cooling water in the condenser, the gas circulating within the condenser coils and becoming liquefied by the time it reaches the lower extremity of these coils.

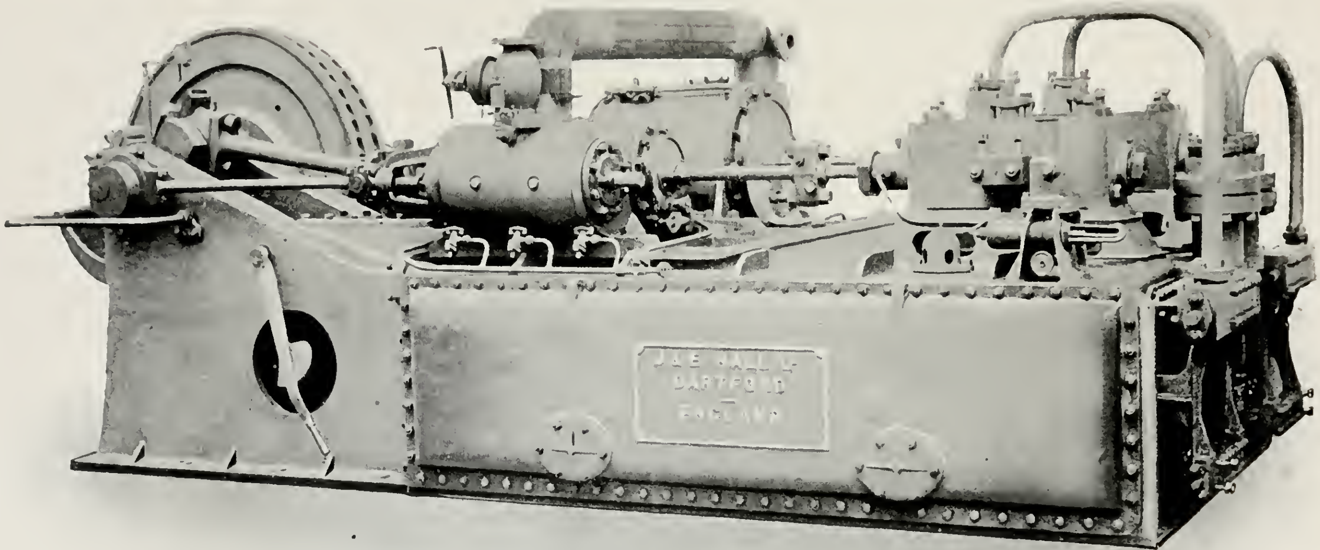
We are able, by regulating the pressure in the evaporator, to cause the liquid to boil throughout the coils of the evaporator, which act in the same manner as the heating surface in a steam boiler, and the temperature or boiling point of the liquid carbonic acid adjusts itself to that of the source of heat which is causing it to boil, whether it be water at 70° to be reduced to 40°, or brine to be maintained at +10° Fahr. or −10° Fahr.

The surfaces of the evaporator coils are so proportioned that all the liquid which enters at the lower end of the coil is evaporated by the time it reaches the top end, and thus the maximum efficiency is obtained. The compressor then draws in only gas, and compresses it up again to the pressure necessary to liquefy it, and delivers it warm to the condensing coils to continue the cycle of operation.*

The vertical marine type machine, illustrated on the preceding pages, consists of a single vertical steam cylinder, with the compressor arranged alongside of it, both secured to a casting containing the condenser coils, which are made of copper, and behind this casting is another secured to it containing the evaporator coils, the whole making a very compact and accessible design.

These small machines are perfectly simple, work quite noiselessly, and can easily be worked by a person of ordinary intelligence from the printed instructions supplied with each machine. They need very little attention, and the wear and tear and consequent need of repairs is almost *nil*. No increase in the staff of engineers is necessitated, as

* Extract from Paper read before the Institute of Brewing, by Mr Alex. Marcet, A.M.Inst.C.E., Managing Director, J. & E. Hall Limited. May 1894.

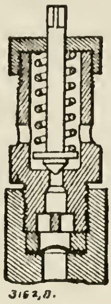


HORIZONTAL TYPE CO_2 MACHINE.
By Messrs J. & E. Hall, Limited, Dartford.

[To face page 552.

they can be placed in any available corner of the engine-room, where they come under the eye of the engineer on watch.

The complete charge of carbonic anhydride in the machine is so small that it may be allowed to escape into the engine-room without the slightest inconvenience. A patent safety valve is fitted so that no



No. 7.—Patent Safety Valve, Hall Machine.

mistake or neglect on the part of the attendant can cause anything like an accident.

Instructions for Charging and Working.

Before Charging.

Pressure Lubricator.—Fill receptacle above hand pump with Vacuum Dartford Refrigerating Oil, and pump this into the pressure lubricator by means of hand pump till piston is at inner end of stroke.

Compressor.—In a vertical machine put piston at bottom end of stroke, take off cover and about one-fourth fill compressor with Vacuum Dartford Refrigerating Oil. Replace cover. In a horizontal machine, take out one of the delivery valves and put piston at the other end of the compressor. Then about one-fourth fill compressor with Vacuum Dartford Refrigerating Oil. Replace the valve. Pull machine round twice, then run machine for a quarter of an hour, all screw-down valves being open. After this commence charging.

Charging.—The CO_2 is supplied in steel flasks of various sizes containing from 22 lbs. to 40 lbs. each, as stamped on flask. About * lbs. constitutes a charge.

Suspend the flask, valve upwards, on the spring balance, and connect by copper pipe provided to small screw-down valve at end of evaporator coil. Note the weight. Open valve on flask and on

* Varies according to size of machine.

machine. See that the connecting joints, which are made with leather washers, are tight. After the CO_2 has passed into the system, note weight again, and the difference is the weight of CO_2 passed into the machine.

When flasks have had 10 lbs. taken out (**not before**) they may be warmed by pouring on hot water to assist in driving out further gas. While this proceeds the lower end of flask will remain cold so long as liquid CO_2 remains, but when the whole flask has become heated, no more liquid CO_2 remains, and valve should be closed **while flask is still warm.** To get the utmost out of the flask before closing its valve the evaporator should be pumped down to say 15 atmospheres (on outer circle of gauge) by running the compressor for a few minutes with closed regulator.

When first charging, blow the air out of the system by breaking the joint between the regulator and pipe leading to it from condenser, the regulator being closed, and all other valves open, and blow some CO_2 to waste according to size of machine, thus :—

Up to No. 9 Machine - - - -	2 lbs.
,, ,, 13 ,, - - - -	5 ,,
,, ,, 16 ,, - - - -	10 ,,
,, ,, 20 ,, - - - -	20 ,,

As the pressure, while charging, rises, carefully examine all joints. The slightest leaks become visible when painted with soap and water lather.

Gauges.—The CO_2 gauges on condenser and evaporator show on outer circle the pressure in atmospheres, and on inner circle the corresponding temperature of CO_2. (When logging, the inner circle only should be recorded, figures in red on the gauge being entered in log with a minus sign thus:

$$-15^\circ$$

denoting 15° below zero.)

Working Conditions.—Having fully charged, start the machine, and adjust the regulator (*i.e.*, inlet valve of evaporator coil) so that the evaporator gauge indicates on inner circle 10° to 15° Fahr. (6° to 9° Cent.) below the temperature of the brine leaving the evaporator. If the machine be sufficiently charged, the condenser gauge will indicate usually some 15° Fahr. (8° Cent.)* above the temperature of the water entering the condenser.

* (Note.—This varies with the quantity of water passing condenser, and is correct if the water outlet is 10° Fahr. higher than the inlet. If the rise in temperature of water is only 5° Fahr. the gauge should stand about 12° above water inlet. If the rise in temperature of water is 20° Fahr., the gauge should stand about 20° above water inlet.)

An excessive charge is indicated by the gauge standing higher, and a very excessive charge by a considerable fluctuation of the pointer.

Under ordinary working conditions, the compressor should be cold or partly covered with snow, and the delivery pipe from it should be rather warmer than the hand can comfortably bear. If the delivery pipe is not hot enough, slightly close the regulator, when the temperature will quickly rise. If compressor becomes warm, it points to regulator being insufficiently open.

If unable to obtain the indications given above on condenser gauge, then the system is short of gas.* As a further test of this close the regulator: if sufficient gas is present, the evaporator gauge should hardly fall for some fifteen or more revolutions of machine. If the gauge falls immediately, without any pause, it indicates shortness of CO_2.

If in doubt as to sufficiency of charge add more; some extra gas in the system, up to one-fourth of a charge, will be far more beneficial than a slight undercharge. **If machine is short of gas, the refrigerating work done will be but a fraction of its proper duty.**

A rise of several degrees of temperature of inlet cooling water will necessitate the addition of CO_2 in order to keep machine fully charged, and similarly a fall in temperature may cause indications of overcharge.

Brine. — The density should be regulated by the addition of calcium chloride till the densimeter supplied floats at central mark, which is 40° on Twaddell's Densimeter, or till one gallon weighs about 12½ lbs., that is, a specific gravity of about 1·25. The brine should be made in a separate vessel, and lumps of calcium should not be thrown into the evaporator tank for fear of choking the pipes. **It is important that sea water should not be used.** Common salt (sodium chloride) may be used instead of calcium chloride, but to prevent corrosion, for every 100 lbs. of salt used 1 lb. of caustic soda is to be added.

In machines fitted with open evaporators there should be sufficient brine in the system to ensure the evaporator coils being entirely covered by about 6 inches when machine is running. In machines fitted with closed evaporators the cock on air pipe must be kept slightly open to prevent air collecting in evaporator.

Compressor Piston and Rod.—The machine is fitted with a double or single acting compressor. Whenever replacing piston observe the following instructions:—

* (NOTE.—In above instructions as to working conditions it is assumed that the valves and piston leathers are in good order. If doubt exists as to this, proceed as explained under heading "TEST.")

For Double-Acting Compressor.—*As the clearances between the piston and the ends of compressor are very small they must be maintained equal at both ends.*

Always bar round before starting after replacing piston.

The piston is packed with hydraulic leathers which will require examination and renewal occasionally. The compressor rod must be kept in a highly polished state and free from any marks, and if machine is lying idle, the rod should be removed and kept well greased in a dry place.

It is very necessary that the nut securing the piston leathers should be well screwed up and locked; when new leathers are put in, it is advisable, a few hours after starting, to tighten the nut up again. (See instructions under "TO EXAMINE COMPRESSOR.")

Compressor Valves.—The suction and delivery valves will require occasional examination and cleaning. A set of spare ones should be kept ready for use.

* *The valve seats are separate from the compressor and make double joints: see that both copper rings are equally crushed by the valve casing. Leakage at the outside joint will indicate itself outside, but at the inner joint will not be perceptible except in reducing the work done by the machine.*

Test.—To test the working of the compressor, close the regulator, when evaporator gauge should be pumped down from say 25 atmospheres to 5 atmospheres in about 200 revolutions. If slower, either the valves or the piston leathers are faulty, or the regulator may be leaking.

Compressor Gland.—This is packed with two hydraulic leathers and compressible rings, between which a pressure of Vacuum Dartford Refrigerating Oil is maintained by the patent automatic pressure lubricator provided. The gland should be screwed up hard enough to compress the rings, and will require tightening up occasionally, **but this should not be done while running.** The pressure lubricator will require pumping back when its piston has moved 4 inches, but this should not occur at least under three hours if the gland leathers and compressor rod are in good order. **The pressure lubricator valves must be full open**, otherwise, though not easily observable, the gland **cannot** be gas-tight. The oil which leaks from the gland should be caught, and after filtering and separating any water from it, used over again.

Separator.—Any oil passing into the compressor will be caught in the separator and must be drawn off every second time of pumping

* For steel block compressors only.

up pressure lubricator, or oftener if much oil is passing in, by slackening drain plug, and after filtering it may be used over again.

CO_2.—This must be pure and free from water and air. If the gas cannot be obtained dry, a CO_2 dryer should be fitted to the machine.

As a precaution against moisture each flask should be suspended valve down for some twenty-four hours before using, and then by *very slightly* opening valve any water present will escape.

Gland Oil.—As an improper oil may cause trouble, it is strongly recommended that only **Vacuum Dartford Refrigerating Oil** should be used. This is obtainable from the Vacuum Oil Co., York House, Norfolk Street, London, W.C., and its branches.

Strainer.—On suction side of compressor is a strainer, which should, with a new machine, be taken out and cleaned after the second day's working, and afterwards occasionally if required.

Stopping and Starting.—When stopped for some days, the screw-down valves on suction and delivery of compressor should be closed, but no other valves. For shorter stoppages, no valves need be closed. The gauges will then equalise, standing at temperature of brine in evaporator. Before starting, care should be taken that any valves closed are reopened, but should this be neglected a safety valve is provided to relieve the pressure. If machine is run at constant speed the regulator should require very little alteration after being once adjusted.

Speed.—The machine should run* revolutions per minute.

Leakages.—It is very necessary that all pipe joints and glands of valve spindles should be carefully examined with soap lather and kept tight. For the first few days especially they should be examined daily and all bolts and gland nuts screwed hard up. The most minute leak must instantly be stopped.

To Examine Compressor.—Close the suction and delivery screw-down valves, also the valve between pressure lubricator and gland, and slack off a joint to let the gas escape. **Make sure all pressure is gone before opening up.**

* According to size and type of machine.

Stores and Spares.—It is recommended that a supply of the following be kept on hand:—

Flasks of CO_2.
Vacuum Dartford Refrigerating Oil.
Calcium Chloride.
Compressor Piston Leathers.
Compressor Gland Leathers.
Pressure Lubricator Leathers (two sizes).
Compressible Rings for Gland.
Set of Delivery Valves.
Set of Suction Valves.
Set of Bronze Joint Rings for Compressor.
Compressor Piston Rod, highly polished.
Safety Valve Discs.

Possible Causes of Trouble.

1. Owing to leakage, or to a rise in temperature of condensing water, the machine may become insufficiently charged. Remedy:—Add more gas until CO_2 gauges show indications under heading "WORKING CONDITIONS."

2. Gland leathers may be worn out. This will be indicated by pressure lubricator piston working out to its full stroke in one hour or less. Remedy:—Renew gland leathers, carefully examining piston rod for roughness; if necessary use spare rod, and repolish rod taken out.

NOTE.—This may also rarely be caused by a defective piston leather in the pressure lubricator itself.

3. Piston leathers may be slack at nut or worn out. This will be indicated by compressor failing to pump out evaporator as indicated under heading "TEST." Remedy:—Tighten piston nut, or, if necessary, change piston leathers.

NOTE.—The same indications may be caused by the valves being worn or stuck up, in which case they must be examined, cleaned, and, if necessary, repaired.

4. Irregular action of evaporator gauge and also the delivery pipe from compressor changing from hot to cold frequently without apparent reason, indicates that there is some foreign liquid present in the system, probably oil from pressure lubricator which has not been drained from separator. Remedy:—Slack joint between liquid pipe and condenser outlet coil box, and allow foreign matter to escape. Also open the valve on evaporator coil used for charging machine and blow out any oil, &c., present. Also drain separator frequently till trouble ceases, and then drain it according to the instructions.

As an alternative remedy when brine is not at a low temperature, run the machine with regulator full open for ten minutes, then close regulator to working position, and, as soon as compressor delivery is warm, drain separator.

5. Evaporator gauge may register a much lower temperature than stated under "WORKING CONDITIONS" in spite of further opening of regulator. This may be due either to the evaporator casing being partly empty or to the formation of ice on the outside of the coils owing to the brine density being insufficient.

6. Gauges sometimes give false indications—To test this, both gauges should indicate alike when the machine is stopped, regulator remaining open. They should then both indicate the temperature of the brine surrounding the evaporator coils.

7. Condensing water pump may be out of order, or the supply of water may be insufficient. This will be indicated by unusually high condenser pressure, and high temperature of overflow water. Remedy:—Examine water pump or increase water supply.

8. Do not run machine faster than these instructions state. If not effecting usual refrigerating work, ascertain fault and apply remedy.

9. Hand pump on pressure lubricator may refuse to work. Remedy:—Examine and clean the valves of hand pump and small strainer covering suction. In case CO_2 has got in between the gland leathers, slack joint of guard round pressure lubricator piston rod, allowing gas to escape.

10. When cooling chambers, loss of efficiency is sometimes due to the chamber doors (1) being open too often, or (2) being left open, or (3) shrinking and becoming a bad fit. The cold air will then flow out and necessitate machine running many more hours than necessary. Doors can be tested for tightness by closing them on slips of paper which will then be nipped if door fits tightly.

Log.—It is advisable to keep a log, recording especially speed, indication of gauges, temperature of condensing water, in and out, and brine, in and out. Compare present log with past logs, and if any falling off is indicated, ascertain the cause and apply remedy. As a guide, a form of log is appended.

Assistance.—J. & E. Hall Ltd. gladly give advice to users of their machines, but letters explaining any difficulty experienced should always be accompanied by a log of actual working in the form appended.

Important.—*Whenever consulting makers as to any point in working of machine, send them a log giving the following particulars so far as they apply, and always give number cast on machine.*

Time.	Hours run per Day.	Revolutions.	Steam.	Vacuum	Brine.		Gauges Inner Circle		Cooling Water.	
					Return to Evap.	Outlet from Evap.	Evaporator.	Condenser.	Inlet.	Outlet.
Reading of Gauges fifteen minutes after stopping										

Haslam CO_2 Machines.

The Haslam Foundry Company Ltd., of Derby, are also large makers of CO_2 machines, so that the following description of this firm's machines will be of interest:—

The arrangement and general construction is practically similar to that already described, the smaller machines being made vertical and self-contained, and either steam or belt driven. The condenser for use on land is a coil of heavy wrought-iron pipe, but on board ship, copper pipe is used. The larger machines are made horizontal, and the marine type duplex, as shown by illustration on page 551.

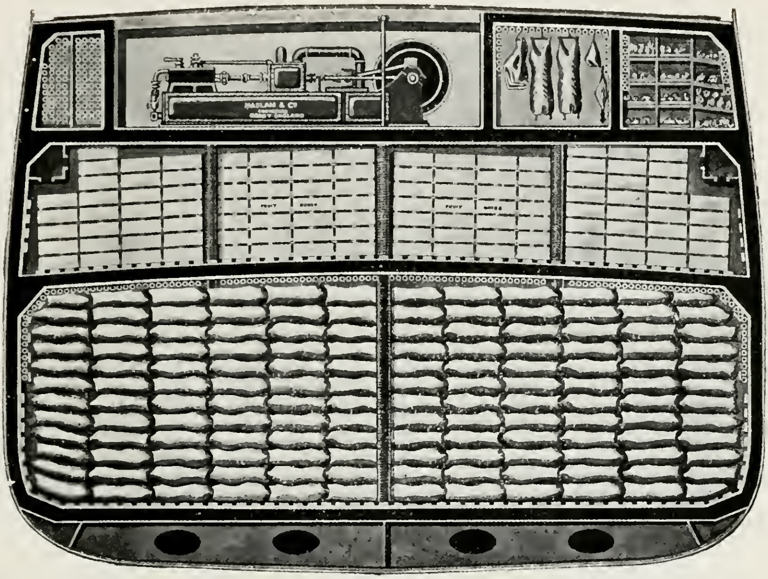
The condenser is here contained in the bed, but in the large machines it is in a separate casing; the evaporator is also in a separate casing. Messrs Haslam fit a safety valve to all their machines.

These machines work either with the brine pipe or air circulation system.

NOTE.—See page 554 for notes on CO_2 and Ammonia, and method of charging machine with gas.

Arrangement of "Haslam" Plant on the CO_2 System.

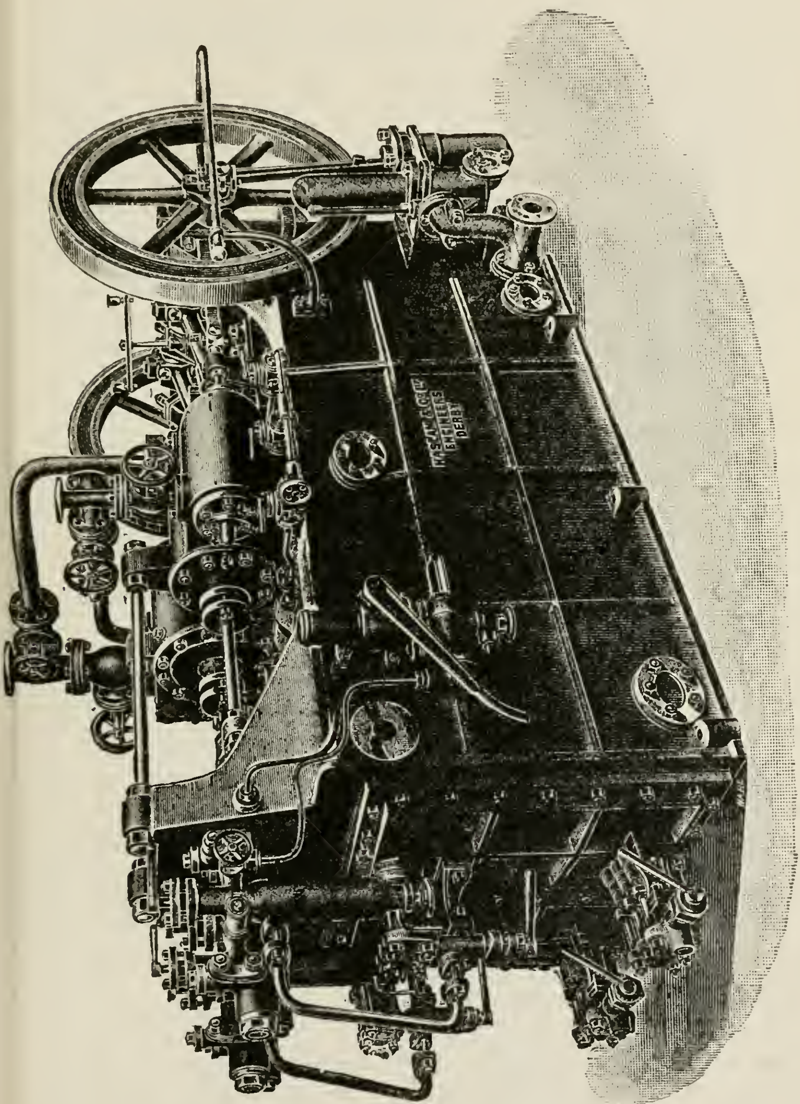
The plate opposite represents a cross section through a ship, showing the arrangement of a Haslam refrigerating plant on the CO_2 system. The ships are so arranged that they are capable of



CO_2 SYSTEM ON SHIPBOARD.

By Messrs The Haslam Foundry and Engineering Co., Limited, Derby.

carrying a mixed cargo, such as frozen meat in the holds and fruit in the 'tween decks, or a general cargo, such as butter, eggs, cheese, poultry, fish, &c. The holds are fitted with brine pipes, under the



No. 8.—CO_2 Refrigerating Machine.
(Messrs The Haslam Foundry and Engineering Company Ltd., Derby.)

deck and on the sides, for maintaining a low temperature. The 'tween decks are fitted with collapsible air trunks, through which the cold dry air is circulated at a suitable temperature for preserving fruit. The air is drawn through the suction trunks by means of a steam-driven fan; it is then passed through the air cooler, consisting of nests of pipes through which the cold brine is circulated. The air is thus cooled to any desired temperature, all moisture being deposited on the pipes in the form of snow. The trunks are so designed that the foul air can be discharged into the atmosphere and fresh air supplied as required. The 'tween decks can also be fitted with brine pipes. When they are not in use the collapsible air trunks are hinged up out of the way. The two small chambers which are intended for holding passengers' provisions are fitted with brine pipes, the larger one being maintained at a temperature of 20° Fahr. for meat, and the other at a temperature of 35° Fahr. for vegetables. A small ice tank for producing about 3 to 4 cwt. of ice for table use and water and wine coolers is supplied when required.

The CO_2 refrigerating machine can either be fitted in the tunnel (when a twin-screw ship), in a corner of the main engine-room, or on one of the upper decks, thus taking up little or no valuable space.

Liverpool Refrigeration Co. Ltd. CO_2 Machines.

The machine generally consists of a steam cylinder and gas compressor, arranged side by side, and coupled to a double throw crank-shaft which runs in three bearings. The shaft has a heavy fly-wheel at one end with safety barring lever.

The steam cylinder is fitted with a piston valve. Crossheads are of the open type, and all bearings are adjustable.

The compressor embodies a number of patented improvements. It consists of an outer steel casing enclosing an easily withdrawable liner of special metal, forming the bore of the cylinder. At either end are heads of forged steel which carry the valves, and back and front covers.

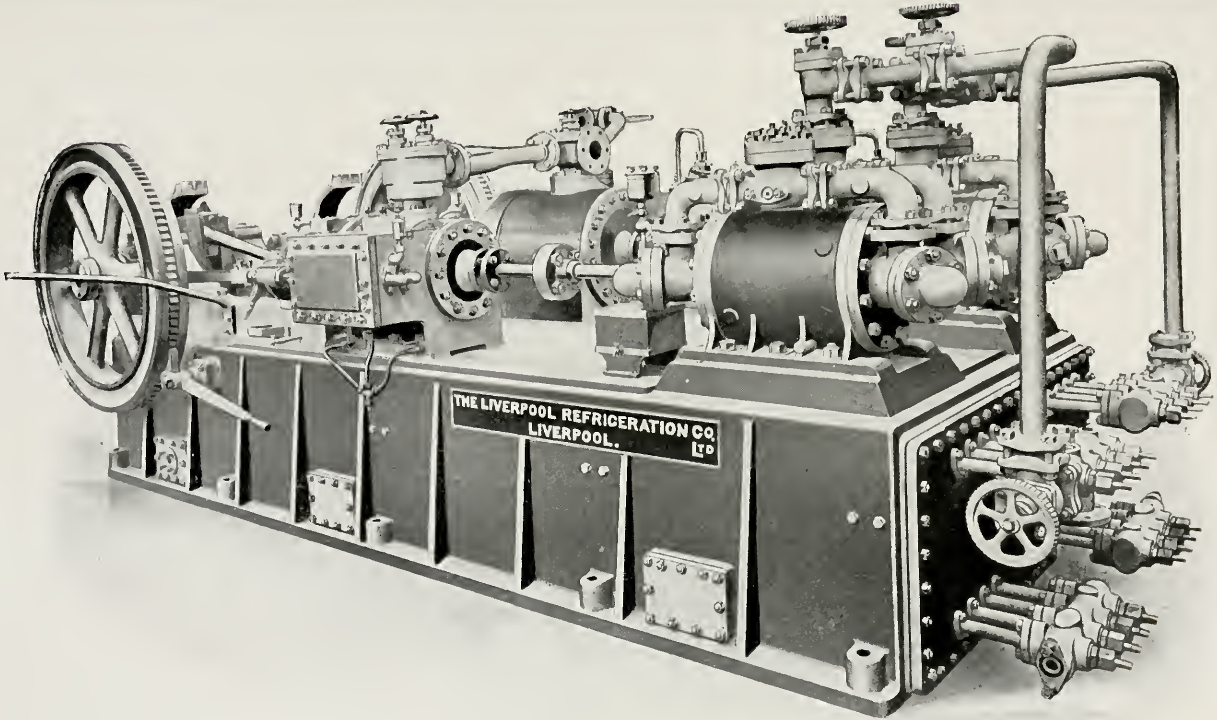
The usual leather cups for packing the piston and glands are entirely absent, and there is no forced lubrication.

The piston is a metallic packed piston of considerable length, and having a number of special cast-iron packing rings held in carriers of hard bronze, the whole being kept in place by a special split collar, so arranged that it is impossible for the piston head to come slack so long as the piston is within the cylinder; there are no screws or pins whatever.

The compressor valves work vertically, and are of very large area and small lift.

The gland consists of an inner and outer stuffing box, the former being packed with a special form of metallic packing. This stuffing box and gland does most of the work, and, in addition, a couple of woodite washer rings are fitted to the outer gland to stop any small leakage past the inner packing.

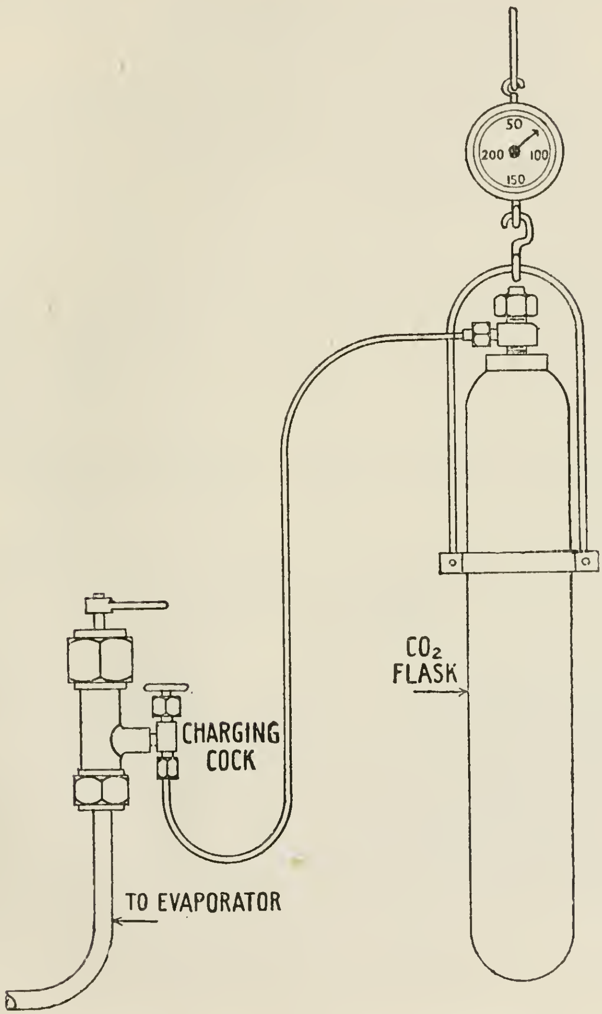
The whole is arranged for long continuous runs, and avoids



DUPLEX COMPOUND MARINE REFRIGERATING MACHINE.
THE LIVERPOOL REFRIGERATION CO., LIMITED.

[*To face page* 562.

altogether the difficulties experienced with leather cups. In the box-bed underneath the machine are arranged both the condenser and evaporator coils. The bed is divided into two portions by a



No. 9.—Method of Charging Machine with CO_2 Gas.

watertight bulkhead, the condenser coils being placed in one portion, and a cast or wrought iron cartridge or case heavily lagged, and containing the evaporator coils, is placed in the other portion. The evaporator casing is hermetically sealed inside the box-bed so that there can be no condensation of moisture, and the lagging is always kept perfectly dry and highly efficient. The coils, however, can be readily withdrawn by removing the covers and without disturbing the insulated cartridge case.

There are no internal joints whatever, and the headers, regulating and control valves are all easily accessible, and are neatly and handily arranged.

All connections subject to CO_2 pressure are of steel.

The brine and water pumps are independent of the machine, and in one form consist of a duplex, double-ended pump having the brine cylinders at one end, the water cylinders at the other end, and the steam cylinders in the middle.

Charging Refrigerator with Gas.—The balance overhead indicates the amount entering the evaporator by showing a difference of weight, and as the flask becomes emptied of its contents, slightly heating it with warm water will quicken evaporation and thus produce complete (or nearly so) evacuation.

NOTE.—The steel flask contains liquid CO_2 (or Ammonia) under pressure, but when the pressure is decreased as described, evaporation instantly commences, and the CO_2 then passes off as a gas.

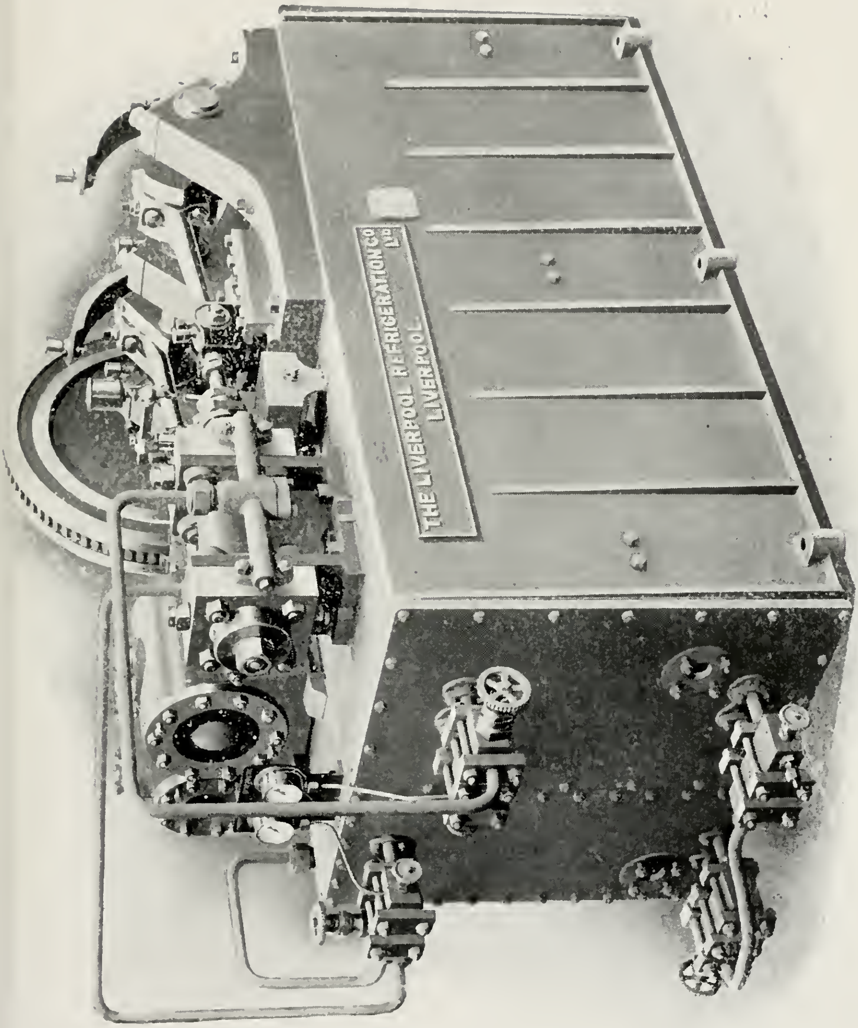
When the connection is opened between the flask and the evaporator, evaporation begins owing to decrease of pressure, and CO_2 gas passes from the flask into the machine.

Latent Heat of NH_3 and CO_2.—At atmospheric pressure the boiling point of Ammonia (NH_3) is $-37{\cdot}5°$ Fahr., and of Carbonic Acid (CO_2) $-125°$ Fahr. Notice that both of these are below zero.

The Latent Heat of Evaporation of Ammonia (NH_3), at a pressure of 30 lbs. and temperature of 0° Fahr., is 555 units of heat. The Latent Heat of Evaporation of CO_2, at a pressure of 310 lbs. and temperature of 0° Fahr., is 124 units of heat. This means that 1 lb. of Ammonia, in evaporating in the evaporator coils, absorbs 555 units of heat from the surrounding brine, which is therefore lowered in temperature correspondingly; and that 1 lb. of CO_2, in evaporating in the evaporator coils, absorbs 124 units of heat from the brine, which is therefore lowered in temperature correspondingly.

NOTE.—Water absorbs Ammonia in the proportion of 600 volumes of Ammonia gas to 1 volume of water. From this it will be evident that the best means of getting rid of Ammonia gas, should a serious escape occur in the engine-room, would be to play a jet of steam into the Ammonia fumes.

NOTE.—CO_2, being a non-supporter of combustion, can be employed to put out fire in a steamer's hold, if allowed to escape into it.



HORIZONTAL MARINE TYPE CO_2 MACHINE.
By The Liverpool Refrigeration Co., Limited.

CO_2 and NH_3 Pressures and Temperatures, &c.

Chemical.	Temperature.	Pressure.	Latent Heat of Vaporisation.
Anhydrous ammonia (NH_3) -	$-6°$	20 lbs.	580 B.T.U.
Carbonic acid (CO_2) - -	$-6°$	280 ,,	125 ,,
Anhydrous ammonia (NH_3) -	$85°$	170 ,,	530 ,,
Carbonic acid (CO_2) - -	$85°$	1050 ,,	25 ,,

From the foregoing table it will be noticed that for equal temperatures the CO_2 system requires much higher pressures than the NH_3 system, but it should be remembered that the working medium is safer in case of escape, and the size of compressor required is much less. On the other hand the latent heat of vaporisation is higher for the ammonia, which means that less gas will be required to produce a given amount of cooling.

Critical Temperature of CO_2 and NH_3.—By this is meant the temperature beyond which the gas cannot be reduced to liquid form.

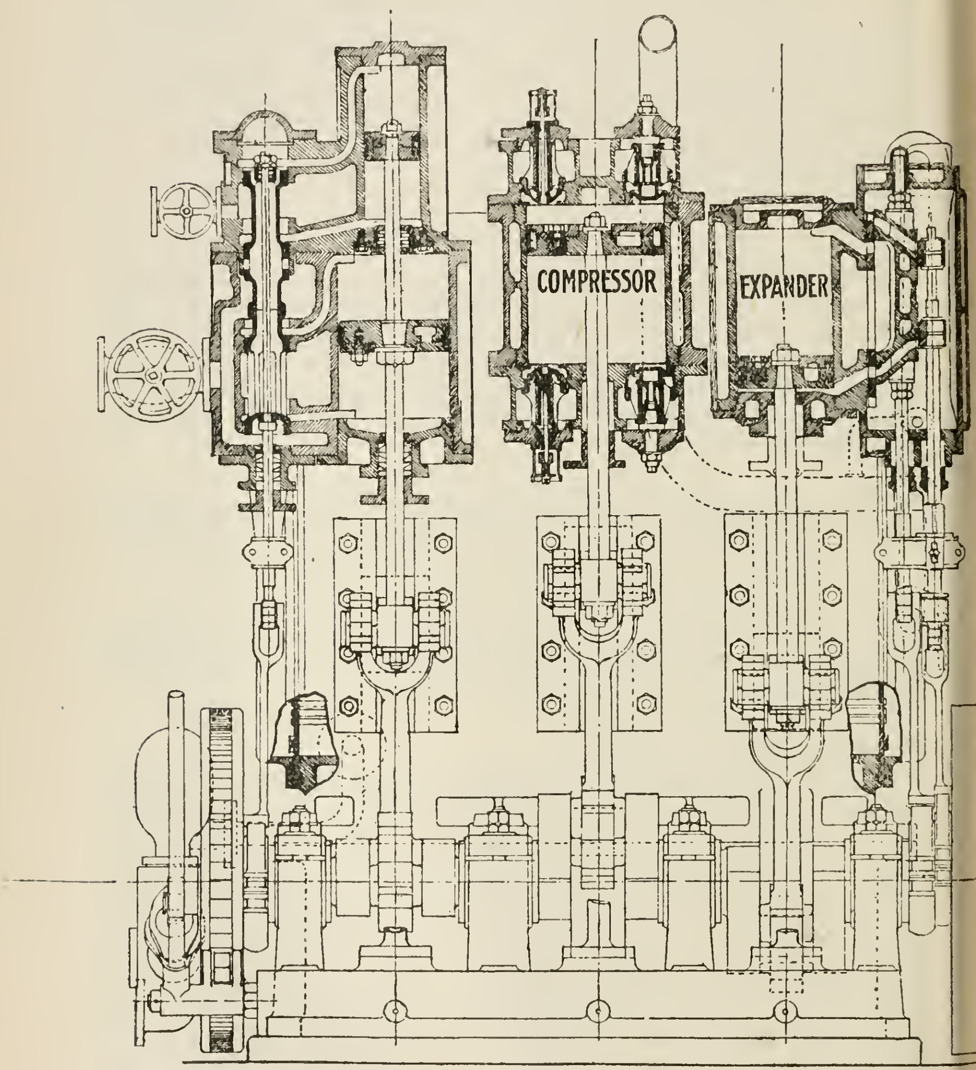
For Ammonia (NH_3) the critical temperature = 256° Fahr.
For Carbonic Acid (CO_2) ,, ,, = 88° ,,

If, therefore, in a CO_2 machine the water circulation temperature rises above 90°, the efficiency of the machine will fall off, owing to the disappearance of the latent heat.

The Compressed Air System.

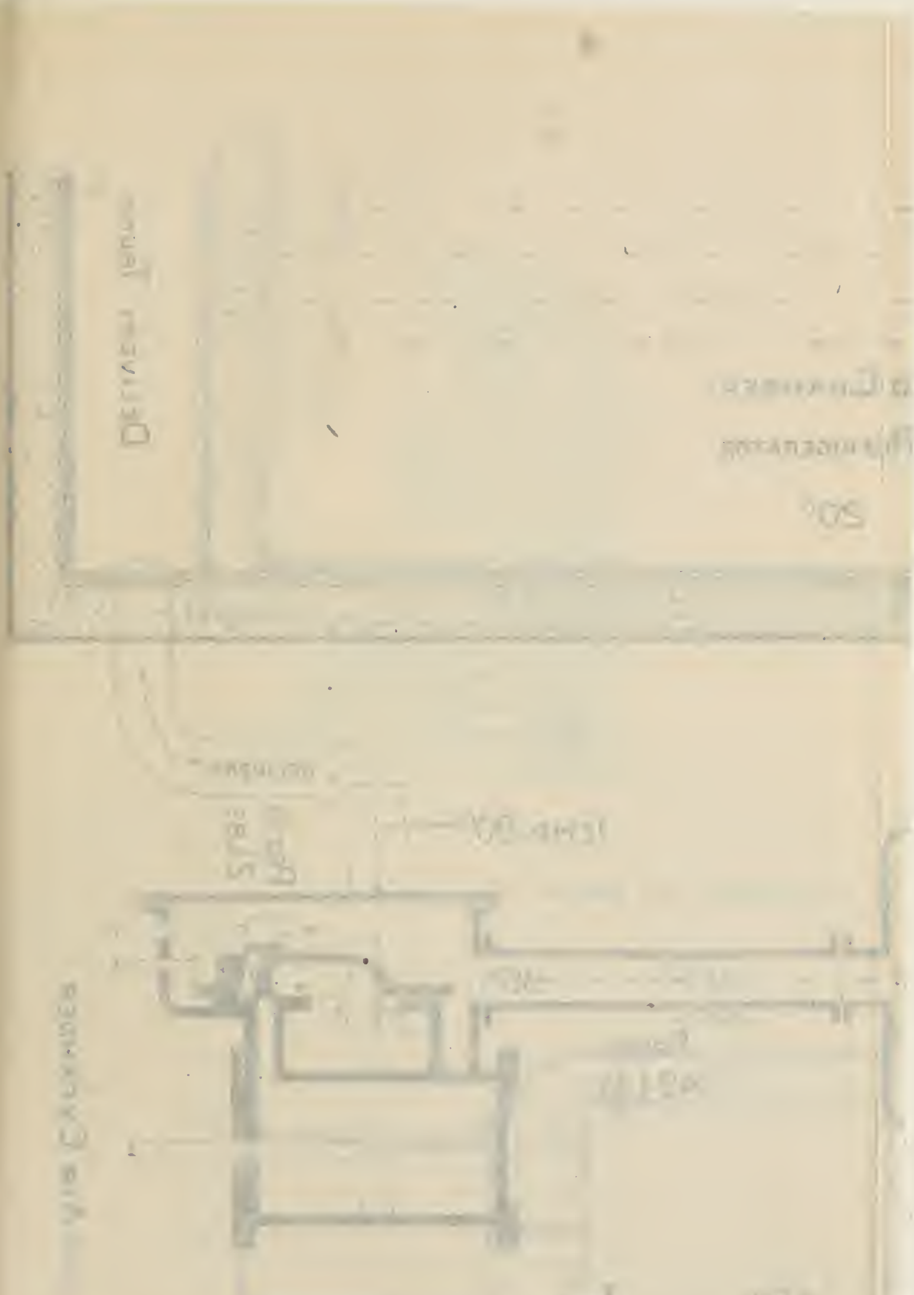
One of the simplest forms of refrigerating machine is that commonly known as the dry air machine, which works by the compression, cooling, and expansion of atmospheric air. These machines have found great favour for use on shipboard, owing to the low temperature which can be obtained, the dryness of the cooled air, the low pressure at which they work, and the absence of any chemicals—the working medium, air, being always available. Their chief disadvantage as compared with the chemical machines is that no use can be made of the latent heat of the refrigerating medium; therefore a much larger quantity has to be dealt with, necessitating larger compressors and increased power to drive them. Had it not been, however, for the compressed air machine, our meat trade with the colonies could not have assumed the dimensions it has at present, and the fact that some of the leading steamship companies are still fitting their ships with this class of machine speaks well for its reliability and usefulness.

Referring to the diagram showing the working of a compressed air machine. The air compressor is driven by any independent means, such as an engine and crank-shaft, and the suction pipe, being connected to the cold chamber, draws in the warmest air from the top of the



No. 10.—Vertical Type Haslam Dry Air Machine.

room. This air is compressed to 50 lbs. pressure, thus increasing its temperature to about 280°, and it is then delivered into the air cooler, consisting of a number of tubes surrounded by water, which is circu-



"Verbal" Notes and Sketches.

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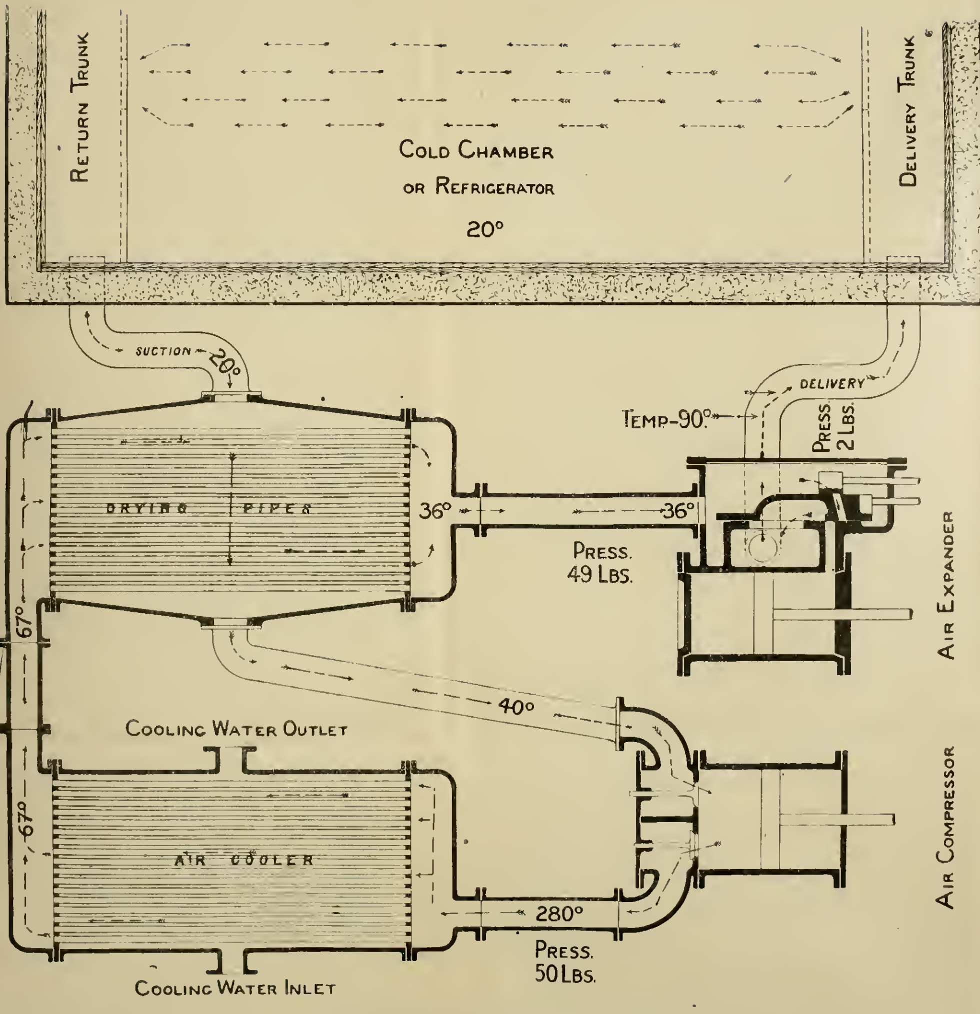

No. 11. Diagram of Compressed Air System.

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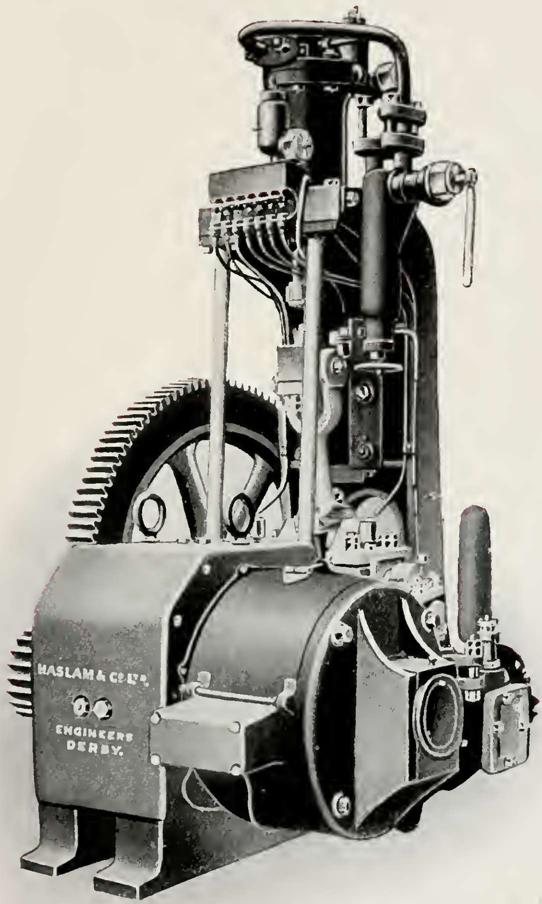
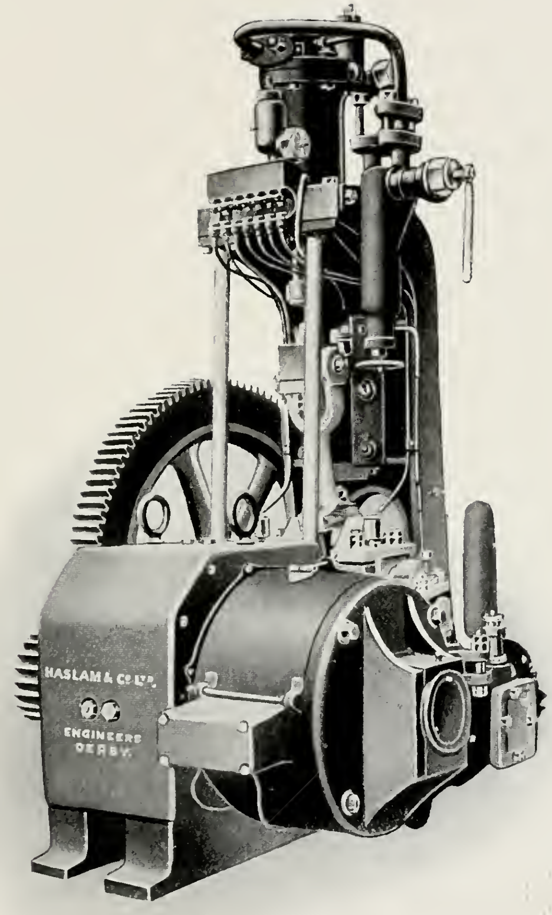


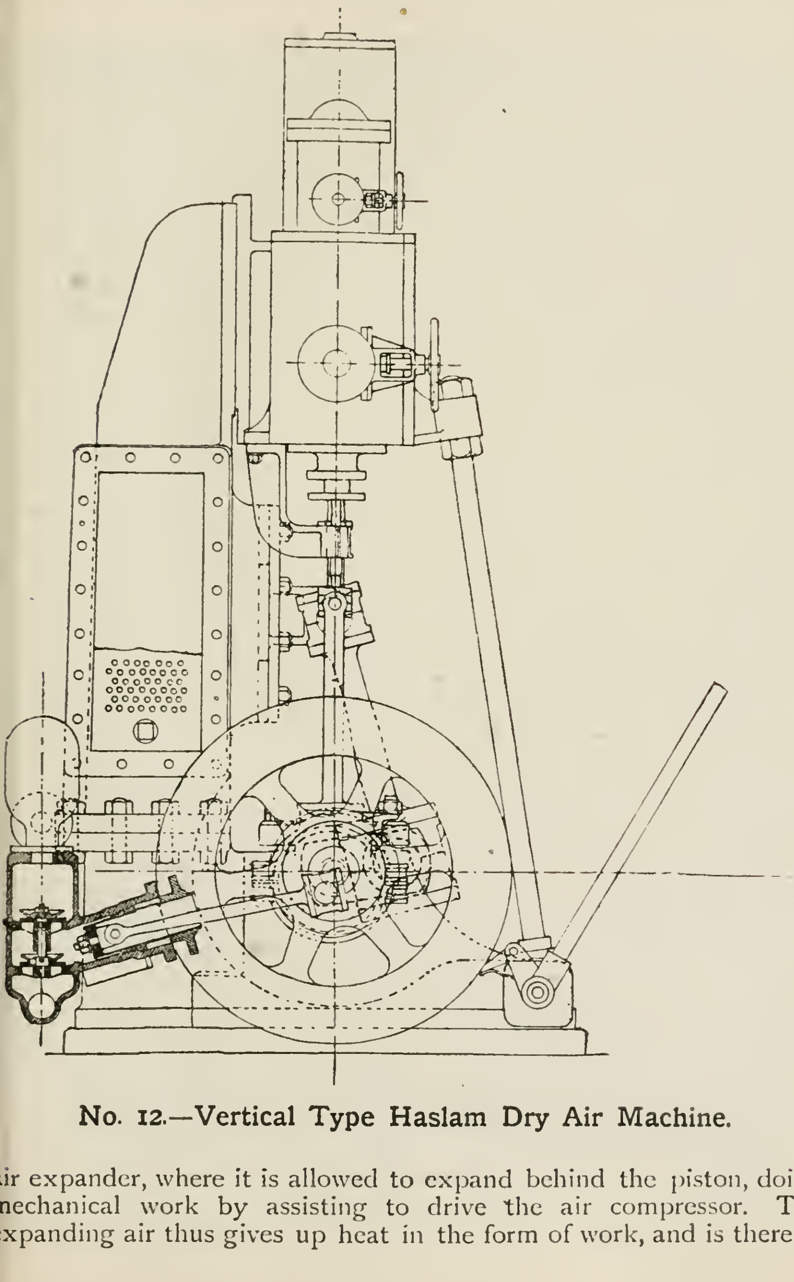
VERTICAL ELECTRICALLY-DRIVEN CARBONIC ANHYDRIDE REFRIGERATING MACHINE, ADMIRALTY TYPE.

tempe...
consisting of a number of tubes surrou...



VERTICAL ELECTRICALLY-DRIVEN CARBONIC ANHYDRIDE REFRIGERATING MACHINE, ADMIRALTY TYPE.

lated by a separate pump. Here the air is cooled down to a few degrees above the circulating water temperature and delivered to the



No. 12.—Vertical Type Haslam Dry Air Machine.

air expander, where it is allowed to expand behind the piston, doing mechanical work by assisting to drive the air compressor. The expanding air thus gives up heat in the form of work, and is thereby reduced in temperature to about −90° Fahr. The air is then delivered into the cold chamber at one side near the top, and, circulating through the room to the return trunk, is drawn again to the compressor, the cycle thus being complete.

In most cases patent drying pipes are fitted in connection with the above apparatus, their function being to dry the compressed air before it is expanded. This is done by passing it through tubes round which is circulated the returning air from the cold rooms. By this method the compressed air is cooled, and any moisture is deposited in the "dryer" and discharged through cocks or valves at the bottom.

Sir A. Seale Haslam has developed the compressed air machine from its experimental stage, and machines built by his firm were the first to carry successfully large cargoes of frozen meat from the colonies. We now give illustrations and descriptions of machines as built by the Haslam Foundry and Engineering Company Ltd., of Derby.

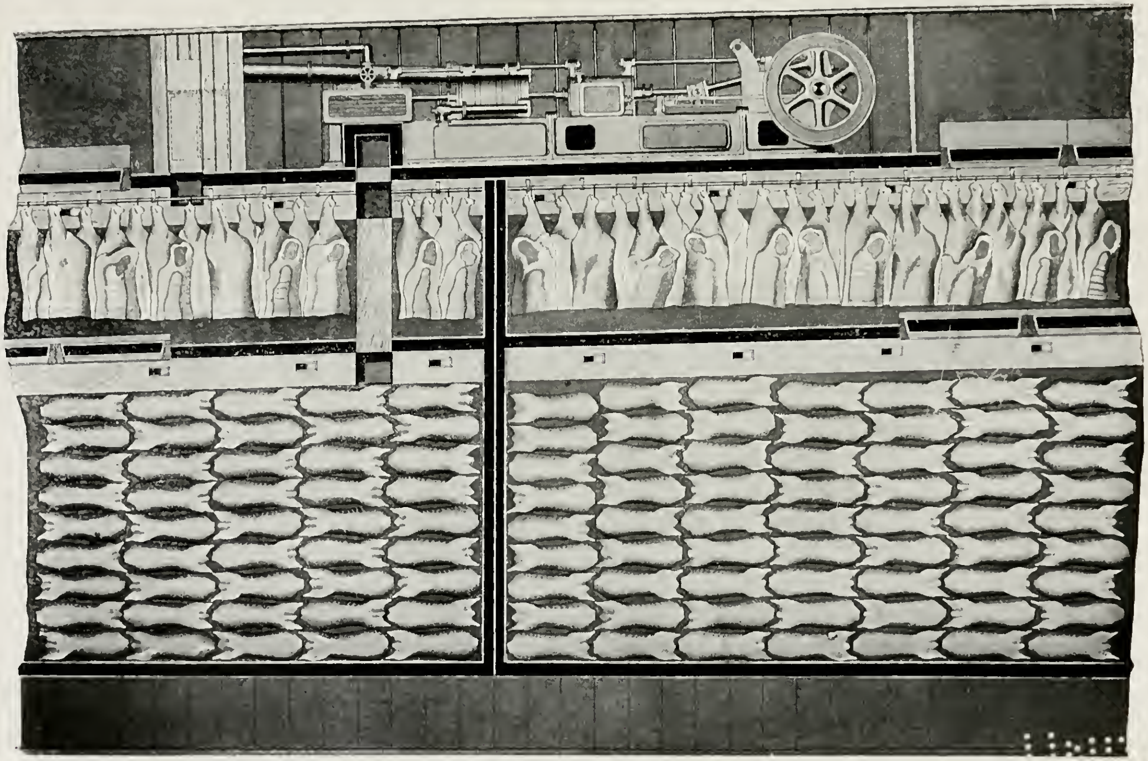
The machine here illustrated is of the vertical type, and is capable of circulating 8000 cubic feet of air per hour. It is driven by a compound steam-engine, the steam cylinders being arranged tandem; the steam supply is controlled by a piston valve in each cylinder, actuated by a single eccentric. The compressor, arranged alongside the steam cylinders, is double acting and water jacketed, and has hollow covers of brass in which are the suction and delivery valves of phosphor bronze. The air expander, placed alongside the compressor, is also double acting and has main and cut-off valves, the latter being adjustable. The ports of this cylinder are made as short and direct as possible to avoid choking with snow, which in small machines not fitted with drying pipes would otherwise give trouble.

Across the back of the machine is arranged the air cooler, similar in construction to an ordinary steam surface condenser, in which the compressed air on its way to the expansion cylinder is cooled by the sea water surrounding the tubes. A water-circulating pump is fixed to bed and driven from crank-shaft.

The next illustration shows Haslam's duplex compressed air refrigerator, capable of circulating 180,000 cubic feet of air per hour at 90° below zero (Fahr.). It is specially arranged to work in the 'tween decks of a ship, and is duplicated so that either side of the machine, consisting of one steam cylinder, one compressor, and one expansion cylinder, can be worked separately in case of accident to the other side, thus giving greater security to the cargo. The machine is fitted with steam surface condenser, air, circulating, and feed pumps, contained within the soleplate, the pumps being driven by a rocking shaft worked from either crosshead pin. When worked as one machine the steam cylinders work compound.

The drying pipes are separate nests of tubes, and are not shown in the illustration.

The plate illustration opposite shows the usual arrangement of the refrigerating machine, drying pipes, and air trunks as fitted on board a steamer, and is a good type of a large cargo installation. The holds



ARRANGEMENT OF AIR TRUNKS, ETC., HASLAM COLD AIR MACHINE.

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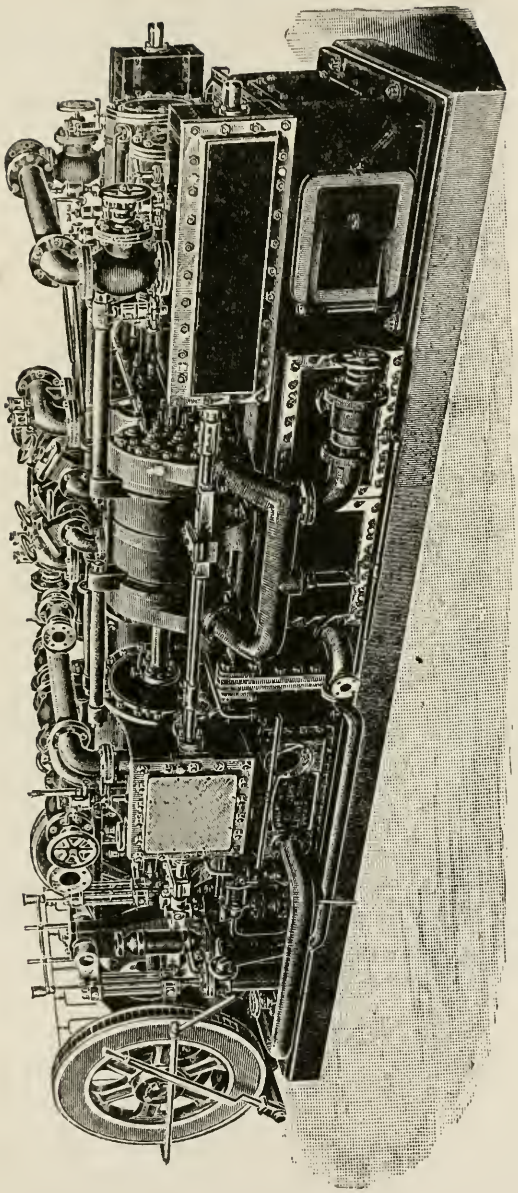


No. 13.—Duplex Horizontal Marine Type Dry Air Machine.

(Messrs The Haslam Foundry and Engineering Company Ltd., Derby.)

and upper and lower 'tween decks are insulated. The air trunks are made collapsible when not in use, and are arranged on both sides of the ship so that a perfect circulation is obtained, and by an arrangement of doors the direction of the air current may be reversed at will. The 'tween decks may be used for carrying cheese or other produce not requiring so low a temperature as the meat, and are then cooled by the air returning from the holds to the machine. The vertical trunks on each side of bulkhead connect to the longitudinal trunks at sides of ship; the connection is made separately on each side of bulkheads to obviate the necessity of cutting the bulkhead and fitting watertight doors. Two duplex machines are fitted side by side, and on the upper deck, so as to be directly above their work.

Haslam Dry Air Machines.

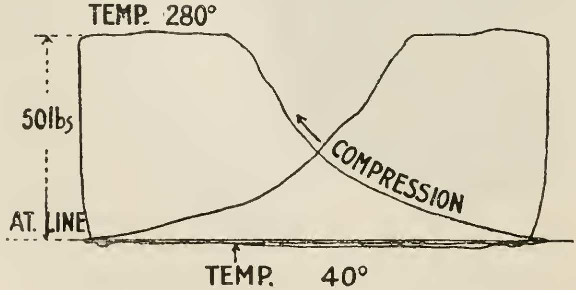
PRESSURES AND TEMPERATURES.

	Gauge Pressure in lbs.	Temperature. Degrees Fahr.
Air entering Compressor - - -	0	40° to 45°
,, leaving ,, - - -	50	280°
,, entering Cooler - - - -	50	280°
,, leaving ,, - - - -	50	67°
,, entering Dryers - - - -	50	67°
,, leaving ,, - - - -	49	32° to 36°
,, entering Expander - - - -	49	32° to 36°
,, leaving ,, - - - -	1 to 2	−90° to −100°

Cut-off in Expander, $\frac{3}{8}$ stroke.
Sea water, 64° Fahr.

Air leaving chamber at 20° is raised to 32° to 36° on entering compressor usually, but if the return pipe in engine-room is not sufficiently lagged, the temperature rises, as shown in the above log, to 40° or 45°.

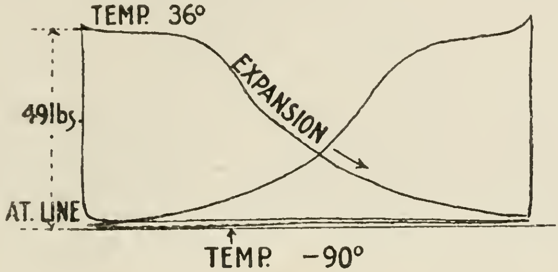
Compressor and Expander Cylinder Diagrams.—Observe that the pressure entering the compressor is rather less than that of the atmosphere, as the initial pressure line is just below the atmospheric



No. 14.—Diagram from Compressor.

line; also that the pressure rises to 50 lbs. (gauge) and the temperature to 280° Fahr.

The air at 50 lbs. pressure and 280° temperature is discharged into the cooler, where most of the heat (work) is carried off by the circulating water passing through the cooler overboard, so that the air, still at 50 lbs. pressure, is now reduced to a temperature of 67° Fahr.



No. 15.—Diagram from Expander.

This necessary loss of heat in the cooler accounts for the difference in area in the compressor and expander diagrams, and means that work at least equal to the loss must be given out by the steam cylinders of the machine.

In the expander diagram, which is of less area than the compressor diagram, notice that the compressed air enters the expansion cylinder at a pressure of 49 lbs. and a temperature of only 36° Fahr., so that when the air is cut off at three-eighths stroke it expands and falls in pressure to about 1 or 2 lbs. above the atmosphere; but the fall in temperature is very much more in proportion, as work is being done by the air in assisting to drive the expander piston. Briefly, by the combined effects of low initial temperature, expansion, and work done, the exhaust temperature is reduced to −90° Fahr. or to −100° Fahr.

NOTE.—If the air merely expanded without doing work the pressure would drop, but the temperature would remain constant; as, however, work is done on the expander piston by the air, heat is lost and the temperature lowered. One B.T.U. disappears for every 778 foot-pounds of work done.

General Notes on Refrigeration.

Pressures and Temperatures.

For a sea temperature of about 70° the following are the average pressures and temperatures required for the three systems of refrigeration obtaining in ordinary marine practice—cold air, ammonia, and CO_2:—

Medium Employed.	Absolute Pressure.	Temperature Fahr.
Cold Air - - -	17 lbs. (Expander exhaust)	− 90°.
,, - - -	65 ,, (Compressor) -	280°.
Ammonia (NH_3) - -	25 ,, (Evaporator) -	− 6° (Brine − 4°).
,, ,, - -	170 ,, (Compressor) -	85° (Sea 70°).
Carbonic Anhydride (CO_2)	280 ,, (Evaporator) -	− 6° (Brine − 4°).
,, ,, ,,	1050 ,, (Compressor) -	85° (Sea 70°).

NOTE.—In the Ammonia and CO_2 systems with brine circulation the brine temperature would be somewhere about − 4° Fahr. for the pressure and temperature of the gas as given in the Table. Notice that the brine temperature is above that of the Ammonia or CO_2 in the evaporator coils, and that the sea water temperature is below that of the Ammonia or CO_2 in the condenser coils (from 12° to 18° difference in each).

Temperature Difference.— For the cooling effect required, it is necessary that a difference of temperature should exist between the gas in the condenser coils and the circulating sea water, the latter being the lower temperature of the two, so that the excess heat picked up by the refrigerant from the brine in the evaporator may be transferred to the circulating water and so carried over the side.

It will thus be obvious that if the sea water rises to a temperature of, say, 80° Fahr., then the temperature of the Ammonia or CO_2 must be in *excess* of this by 8° or 10° to allow of heat transfer, and to obtain this difference of temperature the pressure of the gas must be increased in due proportion.

For a gas temperature of 90° the ammonia pressure would require to be 180 lbs., and the CO_2 pressure 1140 lbs., and if the sea temperature rose to 85° and the gas temperature is to be, say, 93°, the ammonia pressure would require to be 200 lbs., and the CO_2 pressure 1180 lbs. per square inch; so that the higher the sea temperature the higher the pressure required in the compressor to still maintain the necessary temperature difference.

Ammonia (NH_3) evaporates at a temperature of − 37° when the pressure is 14·7 lbs. (atmospheric) and has a latent heat of evaporation of 555 B.T.U.

Leaky Compressor Piston and Valves.—1. Leaky suction valves show by a rise on the evaporator gauge.

2. Leaky delivery valves show by a rise on the evaporator gauge, and a variation in pressure on the condenser gauge.

3. Leaky compressor pistons show by a rise on the evaporator gauge and a fall on the condenser gauge.

To Test if Brine is Corrosive.—Immerse a piece of bright iron in the brine for a period of, say, two days, and the iron will remain unchanged if the brine is non-corrosive.

Air Extraction (Ammonia System).—In extracting air from the system by means of the air cock on top of condenser coils, for safety it is advisable to connect up a flexible length of tubing from the cock, the other end of the tube being immersed in a bucket of water. When the air has all passed out any ammonia which follows will be absorbed by the water, and the smell of which will indicate when to shut off the cock. Previous to repacking the gland or piston the ammonia contained in the compressor can be got rid of by the same method, first closing the hand suction and delivery valves.

Overhauling Compressor (Ammonia System).—In opening up the compressor or any other working part of an ammonia machine the gas should first be got rid of by thorough ventilation, otherwise a light brought near may produce an explosion.

Brine Temperature Difference.—The difference in temperature between the outgoing and return brine should be from 3° to 5° Fahr.

CO_2 evaporates at a temperature of $-120°$ at an atmospheric pressure of 14·7 lbs., and has a latent heat of evaporation of 130 B.T.U.

Cold Air System.—For good efficiency the following points require attention :—

1. Tightness of pistons, valves, of compressor and expander, also Cooler and Dryer tubes.
2. When starting up machine after standing idle for some time, all relief cocks should be kept open, and the machine run for some time before drawing the air direct from the cold chambers.
3. The following temperatures should be regularly taken :—

    A. Air temperature before entering compressor.
    B. ,, ,, after leaving compressor.
    C. ,, ,, before entering expander.
    D. ,, ,, after leaving expander.
    E. Cooling water temperature.

4. Drains on Cooler and Dryer to be opened at regular intervals.

Joint Testing.—To test for gas leakage at joints or connection soap lather is employed, and bubbles form if leakage exists.

Brine Temperature.—The brine temperature kept is about 8° or 10° lower than the temperature of the cold chamber, so that if, say, fruit is to be maintained at a temperature of 16° Fahr., then the brine temperature should be 8° Fahr.

SECTION X.

INTERNAL COMBUSTION ENGINES.

General.

The gas engine is now rapidly coming to the front in marine practice, and (on the Continent particularly) is being effectively developed for use at high powers. The chief difficulty, so far, is that of reversing, but this has been overcome to some extent by the use of compressed air, which, forced into the cylinder, changes the direction of piston travel and thus reverses the engine. The four-stroke engine is still more in evidence than the two-stroke type, but the latter system has been much improved and perfected of recent years. Some foreign makers build gas engines of the double-acting type, but the reliability and efficiency of this system has yet to be proved.

Advantages.

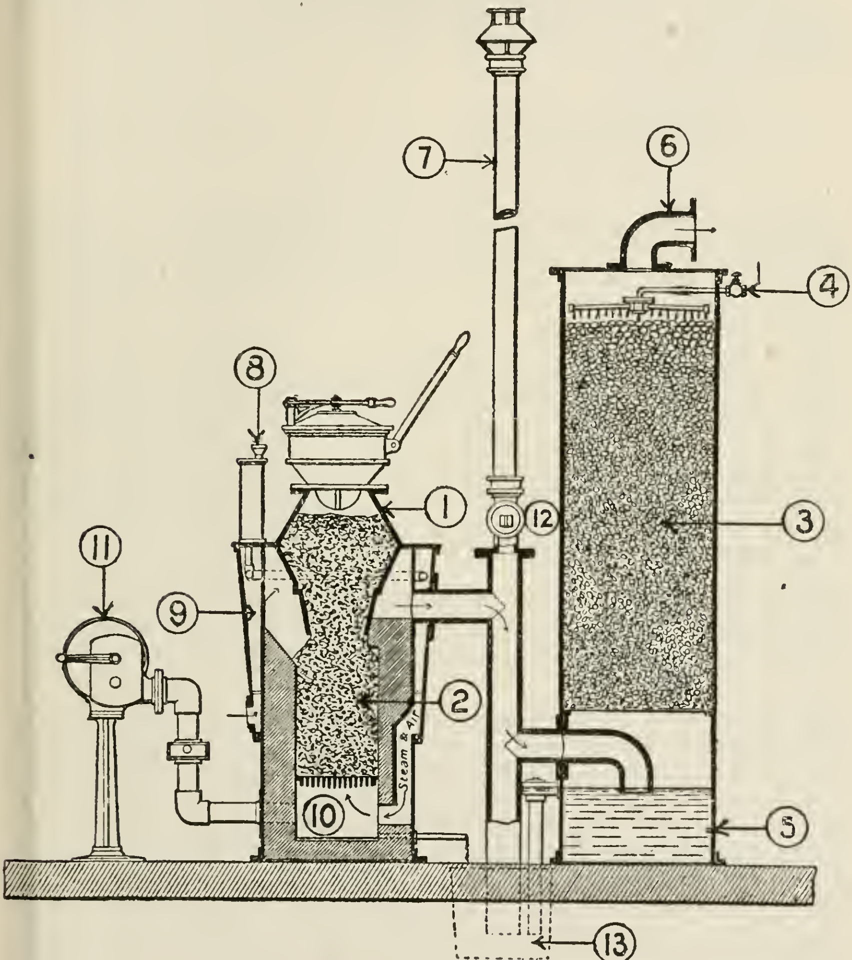
1. Boilers deleted.
2. Smaller bunker space (petrol or petroleum tank) required.
3. Instant starting (with petrol).
4. Ease of manipulation.
5. Cleanliness.
6. Economy.
7. Reduced staff.

Disadvantages.

1. Ignition and other troubles.
3. Danger from petrol and petroleum vapour.
6. Difficulty of reversing.
7. Complication of machinery with engines of large power.
8. Deposits of carbon in cylinder heads.
9. Difficulty of revolution speed control.

Producer Gas.

In land installations, gas obtained by the "Producer" system is used, and this has also been experimented with in marine practice. It is the opinion of many eminent engineers that this is the best method of running internal combustion engines. In the producer system the gas instead of being obtained from oil, is produced direct from coal by heating, the appliance being known as a "producer." This, of course, necessitates the carrying of coal and the use of furnaces, &c., whereas in the case of oil, the oil only is carried in the tanks, and is



No. 1.—Dowson Suction Gas Producer.

1, Coal hopper.
2, Incandescent coal.
3, Coke scrubber.
4, Water supply to scrubber.
5, "Water seal."
6, Gas outlet to engine.
7, Funnel.
8, Water inlet to boiler.
9, Boiler (low pressure).
10, Ashpit.
11, Hand fan for starting up.
12, Funnel shut-off valve.
13, Water overflow pit.

supplied direct to the cylinders, after heating by means of a heating lamp to produce vaporisation.

Producer System.

The gas is obtained by passing a jet of low pressure steam and air through a mass of incandescent fuel. The principal parts of the appliance are:—

1. Producer.
2. Cooler and scrubber.
3. Small steam generator.
4. Hand-driven fan for starting up fire.

Producer.—This is a vertical cylinder chamber with a coal feed hopper at the top, into which the coal (usually Anthracite) is tipped: at the bottom is a fire-box with the ordinary fire-bars and furnace door.

Cooler and Scrubber.—The scrubber is filled up with water ("water seal") for a short distance from the bottom, and the gas entering from the producer enters below the water level, thus forcing the gas through the liquid for cleaning purposes; the upper part of this chamber contains lump coke, and has a water drip from the top; the gas passing up and through the coke is purified or scrubbed of the tarry substances which would otherwise cause trouble in the engine by depositing in the valves, cylinders, &c.

Steam Generator.—This is a smaller boiler fitted on the producer to obtain steam for admission to the incandescent fuel, so that the gas may be produced by the chemical action resulting.

Action.—The general action of the producer is as follows:—After lighting the fires and using the hand fan to create a draught, when the heat is sufficiently strong, the funnel outlet valve is closed and the coal gases pass into the water of the cooler, thence to the scrubber, and from there by suction to the oil engine cylinders, to be compressed and fired in the usual way. The steam generated in the small boiler together with air passes down a pipe and enters the producer among the fuel; in doing so the steam (H_2O) is decomposed by the heat of the coal, and Hydrogen and Oxygen are set free. These gases combine with the carbon of the coal to give CO or Carbonic Oxide, and this, together with the Hydrogen, passes to the top of the producer chamber, then to the cooler and scrubber as previously described.

Therefore, producer gas is obtained by passing air and steam through the heated fuel, and the chemical reactions which take place are as follows:—

$$2C + O_2 = 2CO.$$
$$C + H_2O = CO + H_2$$

In other words, Oxygen of the air combines chemically with Carbon of the fuel to produce CO (Carbonic Oxide), and the water (steam) is decomposed and forms more CO, also free Hydrogen. CH_4 (Marsh Gas) and CO_2 are also formed during the heating process, so that we have as a result—

Combustible gases = CO, H_2, CH_4.
Incombustible ,, = CO_2, N_2.

The average chemical composition of producer gas is as follows:—

CO (Carbon Monoxide) - - - -	about 30 per cent.
H (Hydrogen) - - - - - -	,, 15 ,,
CH_4 (Marsh Gas) - - - - -	,, 1 ,,
CO_2 (Carbonic Acid) - - - - -	,, 6 ,,
N (Nitrogen) - - - - - -	,, 48 ,,

Efficiency.—The efficiency of producer plant is about 80 per cent.

Consumption.—The coal consumption per I.H.P. per hour varies from about ·9 lb. coal at low powers to about $1\frac{1}{4}$ lbs. coal at high powers.

Heat Value.—The heat value of producer gas varies from 140 to 180 B.T.U. per *cubic foot* of gas.

Test Burner.—A testing burner is fitted on the pipe leading from the producer to the cooler, and this when lighted indicates the quality of the gas which, if satisfactory, is allowed to enter the cooler, but which, if not satisfactory, is allowed to pass up the waste gas funnel.

NOTE.—It should be observed that when the engine is running, the regular suction of the cylinders acts like a draught to keep the producer working, so that the supply is equal to the demand, and this being so, the hand fan is stopped as soon as the engine is started running.

It should also be noted that the producer is worked at a pressure slightly under that of the atmosphere, owing to the suctional action of the engine in drawing off the gas generated. This also explains how the atmospheric air pressure enters the producer.

After the gas passes from the producer to the engine it is mixed with the required amount of air in the carburetter or vaporiser to form a more highly explosive mixture for rapid combustion (explosion) in the cylinders.

Explosion Systems.—The oil motor is an explosive engine, the cylinder head constituting the explosion or combustion chamber, and to allow of this the clearance space is equal to about 30 per cent. of the cylinder volume. When the explosion is produced by a spark, or, as in the case of paraffin motors, by a hot tube or lamp, the engine is then really of the "explosive" type; but if, as in some cases, the explosion is produced by the gases being compressed sufficiently to ignite spontaneously by the heat left in the cylinder head, the motor is then of the "internal combustion" type, although the term internal combustion is generally applied in both cases.

In the typical modern motor boat using petrol, the energy is stored in the petrol; this is transformed into gas and mixed with air in the carburetter, while the energy is liberated as heat in the engine cylinder, and then directly transformed into mechanical work by the piston, and into propulsive work by the propeller.

The oil or explosive engine is of course a heat engine, and as such develops concentrated energy at the moment of ignition of the gases, whereas with steam the energy is distributed more over a longer period. The gas entering the cylinder represents potential energy, which on ignition changes almost instantly into kinetic or active energy, the fly-wheel receiving and storing up some of this power or energy, which is afterwards employed in completing the three other *powerless* strokes. In one of these, the compression stroke, a large proportion of the power previously developed is absorbed or used up in compressing the gases to the necessary pressure required for effective ignition and explosion. This work, then, is done *by* the piston *on* the gas, and may be called negative work.

As heat and work are equivalent, the intense heat obtained by the explosion of the charge produces work, each unit of heat being equal to, or giving out, 778 foot-pounds of work.

Paraffin and Petroleum.—The paraffin or petroleum motor is certainly the most convenient in many ways, as being usually ignited by a heating lamp less trouble is experienced than with electrical connections, magneto, &c., all of which require careful attention and regular overhaul.

Paraffin can always be obtained, and is fairly cheap; it is also safe, as vaporisation does not readily take place, hence the necessity for the heating lamp previously referred to. This at the same time, however, constitutes a disadvantage, as time is required to heat up the vaporiser before the engine can be started. After once starting, however, the heating lamp is usually turned off, as the carburetter is then kept hot by the exhaust gases. The flash point of paraffin, which is known as a "heavy" oil, is about 84° Fahr.

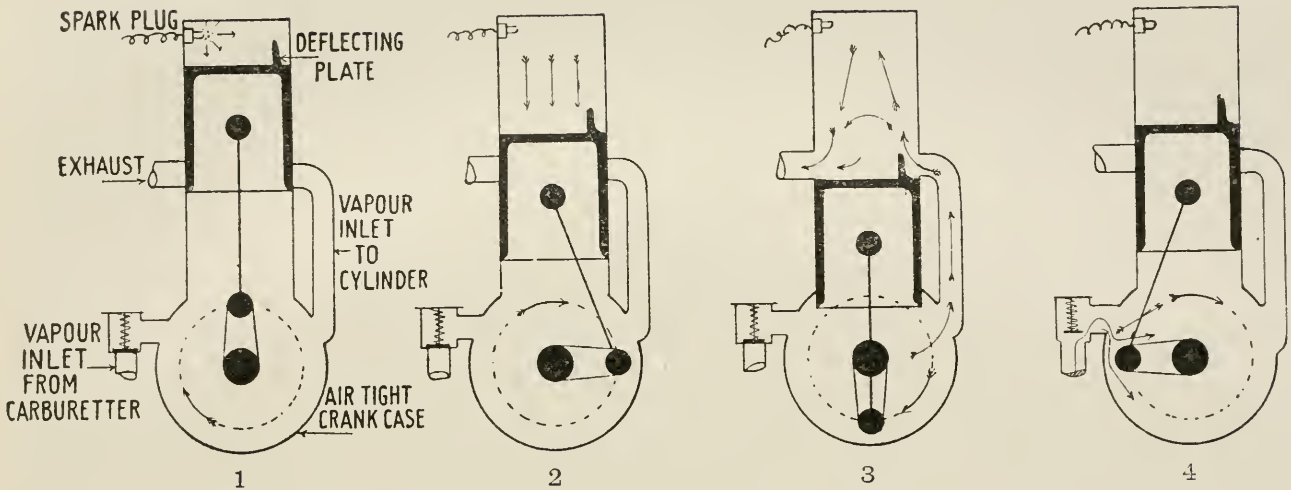
Petrol or Gasoline.—Petrol is the spirit obtained from the crude petroleum, and is chiefly composed of carbon and hydrogen, about 85 per cent. carbon and 14 per cent. hydrogen. The flash point is low, about 35° Fahr., so that evaporation takes place very easily under atmospheric pressure, which is an advantage in the matter of instant starting, but which at the same time constitutes a danger, as if leakage of petrol takes place explosion readily occurs when the vapour formed becomes mixed with atmospheric air. The heat units in 1 lb. of oil fuel are usually about 20,000 units, or roughly about one-half more than that in 1 lb. of ordinary coal, so that for the same power about two-thirds the amount of petrol or paraffin is sufficient. As the fuel is in liquid form it also occupies less bunker space, and, as before stated, for the same weight contains half as much again power or energy, which still further reduces the fuel space required for a given distance to be run.

Two-Cycle.—In small single-cylinder motor sets up to about 7 Brake Horse-Power, and two-cylinder 10 B.H.P., the two-cycle system is usually adopted, and is carried out as follows:—

The crank case is air-tight, and is fitted with a small port, which is opened and closed by the piston as it travels up and down. Through the opening or port referred to, the charge of oil vapour and air is drawn in from the carburetter on the up-stroke, and compressed on the down-stroke, until released by the piston uncovering a connection between the crank case and the top of the cylinder: the charge then rushes into the cylinder, and at the same time drives out the exhaust or burnt gases through the exhaust port which has just opened: when the piston comes up to the top the spark occurs and ignites the compressed mixture of oil vapour and air, and explosion follows, driving the piston down, and, as before stated, when near the bottom the exhaust port is uncovered, and shortly afterwards the connection between the crank case and cylinder. Observe that the exhaust port is opened just *previous* to the inrush of the fresh charge, which being at a slight pressure assists in forcing out the burnt gases left from the previous stroke.

The two-cycle motor, it will be noticed, gives one impulse or power stroke to each revolution. A disadvantage exists in the confusion of the intake gases with those of the exhaust, which to a certain extent reduces the effective energy developed, and which increases with the number of revolutions per minute. This type of motor, however, requires no valves or cams of any kind, as ports only are required to admit and exhaust the gas and air. If two cylinders are employed, the cranks are placed opposite to give good balance and an even turning moment on the shaft. Both two-cycle and four-cycle motors are single acting—that is, the power stroke only occurs on one side (top) of the piston.

Four or "Otto" Cycle.—The action of a "four-cycle" oil motor is as follows:—(1) On the down-stroke of the piston the air and oil vapour are drawn into the cylinder through the inlet valves; and (2) on the following up-stroke the gas is compressed; then (3) fired by an electric spark, or a hot tube, set to act when the piston is near or at the top centre; the explosion which follows drives the piston down, and on the *next* up-stroke (4) the burnt gases are expelled through the exhaust valves, and out into the atmosphere by way of the "silencer." The inlet valves are sometimes worked by the piston suction and springs, and sometimes mechanically by a cam fixed on a special shaft. The exhaust valves are opened by cams



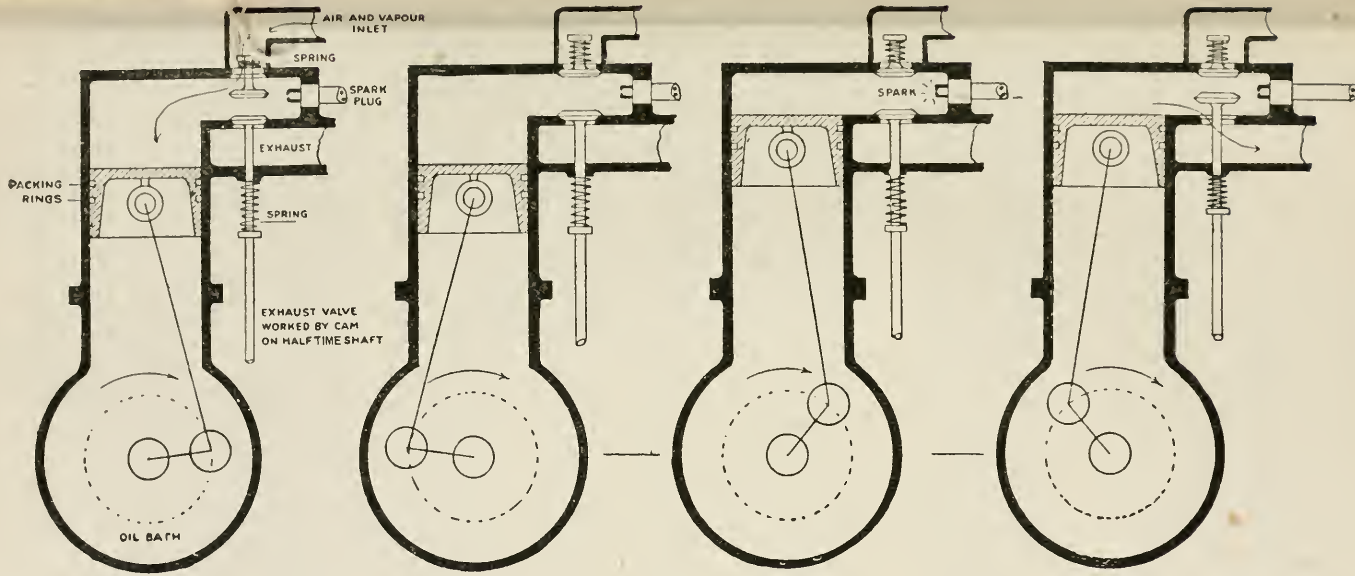
1, Piston passing top centre. Explosion taking place above piston.

2, Down-stroke. Expansion of exploded gases above piston, and compression of vapour charge in crank case to a few pounds pressure.

3, Piston at bottom. Opening of exhaust port (left), and fresh charge passing from crank case to top of piston (right).

4, Piston on up-stroke. Admission of fresh charge into crank case, and compression of previous charge above piston.

No. 2.—Action of "Two-Cycle" Oil Motor.



1 2 3 4

1, *Down-stroke.* — Air and oil vapour entering cylinder by inlet valve.

2, *Up-stroke.* — Mixture of air and oil vapour being compressed by piston to about 80 lbs. pressure. Inlet and exhaust valves closed.

3, *Down-stroke.* — Compressed gas exploded by an electric spark set to act when piston near top centre. Piston being forced downwards by explosion. Inlet and exhaust valves closed.
Note.—This is called the "impulse" or "power" stroke.

4, *Up-stroke.* — Burnt gases being forced out of cylinder by piston, the exhaust valve opened by a cam on the "lay" or half-time shaft. This completes the cycle.

No. 3.—Action of "Four-Cycle" Oil Motor.

on the shaft referred to, and are kept shut by strong springs as shown in the sketches.

Steam-Engine and Oil Motor compared.

Steam-Engine.—Coal stored in bunkers and containing heat energy is supplied to the furnaces, and, combustion more or less complete being effected, the heat obtained is transmitted through the furnace metal, and evaporation of the water in the boiler results. The imperfect steam gas thus produced passes along the steam pipe and enters the cylinder, where the heat energy is employed to drive the pistons up and down, and so produce rotation of the shaft. After doing work in the various cylinders, the steam finally exhausts to the condenser, where condensation takes place, and the water thus obtained is put back into the boilers as feed. During the process of condensation most of the heat in the steam is absorbed by the circulating water, and is thus lost overboard.

Oil Motor.—In the oil motor the boilers are deleted and the fuel contained in the oil tank or bunker is supplied direct to the cylinder head, which also forms the combustion chamber. Combustion is effected by a spark, and the energy thus developed is applied direct to the piston, and acts on the down-stroke only. The waste gases produced by the rapid combustion are then exhausted out to the atmosphere, and the whole of the heat contained is thus rejected. Notice that a condenser cannot be fitted, so that no return of heat is possible.

Again, it will be seen that to obtain the same impulse effect on the crank-shaft *four* cylinders are required if the motor is of the "four-cycle" type, and two cylinders if of the "two-cycle" type, to be equal in effect to one double-acting steam cylinder. The chief difference then exists in the fact that with steam as the motive power ordinary combustion takes place in the furnaces of the boilers, whereas with oil *rapid* combustion takes place direct in the cylinders, and the heat energy developed during explosion does the work of turning the shaft and propelling the boat.

In the steam-engine steam gas generated in the boilers is admitted to the H.P. cylinder, and is cut off at say one-third stroke. This gas expands, and after doing work in the H.P. is admitted to the larger M.P. cylinder, where more work is done and further expansion takes place; the gas then expands into the still larger L.P. cylinder, where the last stage of work and expansion is completed, the gas then passing into the condenser where condensation is effected. The steam gas charge, be it noted, is only supplied to the H.P. cylinder, and by expanding about twenty-two times or so altogether in the other two larger cylinders, most of the effective energy is extracted and useful work done.

The oil-gas engine, on the other hand, is a single-acting, and, of course, non-condensing engine, each cylinder requiring its own independent charge of gas, and, as before stated, combustion is effected in the cylinder head. The cylinders are therefore all of equal diameter, and the gas only expands in one cylinder. No condenser is possible, so that after driving the piston down the expanded gases at a high temperature are exhausted direct into the atmosphere through the "silencer." The back pressure is therefore somewhere about 18 lbs. per square inch absolute, whereas in the steam-engine the back pressure is usually only about $2\frac{1}{2}$ lbs. absolute, owing to the condenser vacuum.

Number of Cylinders, &c.—Any number of cylinders may be employed in an oil motor, but for small launches two or four is the usual number. The number of cylinders, in fact, depends on the power required, and as many as six, eight, twelve, and eighteen cylinders are occasionally fitted, according to the H.P. to be developed by the motor.

The cylinders are of cast iron, and are occasionally fitted with liners, which allow better for the expansion of the upper end of the cylinder due to the intense heat of explosion. One or two cases have come under the writer's notice of cylinder heads cracking, due to the unequal expansion of the upper and lower portions.

Pistons.—The pistons, of cast iron, are kept tight by means of four or six light Ramsbottom cast-iron rings, which are cut as usual.

The pistons are of the trunk type, being very deep in section, and open at the bottom to the crank case, which is often arranged to form an oil bath, thus supplying lubrication to the working parts by the "splash" system.

Revolutions.—Developing full power, the revolutions vary in oil motors from 700 to 1000 per minute, and therefore, neglecting missfires and premature explosions, the average number of power strokes in a "four-cycle" motor will be equal to one-half of the revolutions.

Water-Jacket.—Steam cylinders are sometimes fitted with steam-jackets to keep up the temperature, whereas in oil motor cylinders cooling water-jackets are required to extract the intense heat produced by explosion, and thus keep down the temperature. The water-jacket is a very important detail, and is indispensable. The temperature of the waste gases may be anything from 1000° to 1500° Fahr., which fact in itself indicates the great waste of energy which goes on, and which is evidently unavoidable. Sometimes the valve pockets are also water-jacketed, and occasionally the exhaust gases are arranged to pass round the carburetter to raise the temperature and allow of quick vaporisation of the oil inside.

Pressures and Temperatures.—The pressure of compression just previous to explosion varies from 50 lbs. to 70 lbs. or 80 lbs. per square inch, and the pressure of explosion from 150 lbs. to 300 lbs. per square inch.

The temperature of the gases at explosion is estimated as being somewhere about 1500° Fahr. and upwards.

Carburetter.—The carburetter is a small chamber employed to vaporise the petrol and at the same time mix with it the proper proportion of air. The openings are arranged to give from five to ten parts of air to one of petrol vapour, but of course this varies greatly with running conditions. Throttle valves are also fitted on the carburetter to regulate both the amount of air supplied and the amount of vapour admitted. A float and needle valve regulates the oil supply to the mixing chamber of the carburetter.

A single carburetter is sufficient for any number of cylinders, as by suitable pipe connections the vapour can flow into each as required.

Valves.—The valves are usually of the cam and poppet type, and are opened and shut by means of a "half-time" shaft. This shaft is connected up by gear wheels with the engine shaft so as to only give one revolution for every two revolutions of the engine. It will be noticed that the valves employed are similar to those of the historic Smeaton and Watt engine, and were in use before the slide valve was introduced. Recently, however, the "sleeve" type of valve has come into more general use.

"Sleeve" Valve.—The "sleeve" valve consists of two cylindrical casings fitted round the cylinder with ports cut in each; the outer sleeve is actuated by a link from the connecting rod, and when the ports coincide the gas is admitted to, or exhausted from, the cylinder. This type of valve is particularly silent in running, and gives very little wear.

Fly-Wheel.—A fairly heavy fly-wheel is fitted to motors of both the two-cycle and four-cycle types, the latter requiring the heavier wheel. The fly-wheel makes for steady running balance, and also stores up energy, which is released when the power strokes become irregular or intermittent. The wheel is also convenient for use in starting up by hand.

Firing or Sparking Plug.—This appliance is screwed into the cylinder head, and, when possible, is placed direct in the path of the incoming charge of vapour. One wire from the contact maker, induction coil, and battery (or from the magneto contact brush of this system) passes through the centre, and forms a point from which the spark passes to the other point, which makes metallic contact with the body of the plug, and so to the cylinder, engine frame, and battery, &c. The current, therefore, is "earthed," and after sparking returns to the battery, &c., or magneto, by means of the engine metal (Single Wire System).

Half Compression.—This is fitted in the larger engines for convenience in starting, and consists of a lip on exhaust cam extending half its width. The whole cam is capable of sliding on its axis when the half-compression lever is moved. Thereby, in one extreme position the lip lifts the exhaust valve during the compression stroke and releases the compressed air, making it much easier to pull the engine over the centre; in the other extreme position it clears the valve tappet rod, and the exhaust valve is only opened on the exhaust stroke.

"**Cranking.**"—A petrol motor is started by having the engine disconnected from the propeller, and the handle connected to the shaft or fly-wheel turned smartly round, with the half-compression valve open to reduce the opposing back pressure. After a few revolutions the spark acting at the right time catches up the running, and the motor runs up to the ordinary revolution speed. The spark gear is also "retarded," that is, set to fire when the piston has just passed over the centre. The half-compression valves are, of course, shut down once the motor takes up the speed, and the spark "advanced" or set so as to fire just before the piston reaches the top centre.

In starting, care has to be taken in "cranking up" not to set the motor running in the wrong direction, as this is very easily done, and in some cases may be, to put it mildly, awkward. In one motor boat in which the writer had an experimental run, the engine unfortunately went off in the reverse direction, and as a counterbalance, rather than stop (the motor being of the paraffin type and slow to heat up), the angles of the propeller blades were simply reversed to suit the direction of rotation of the engine, and the run completed under these conditions, which were certainly not the best.

During another run the motor stopped dead owing to wrong adjustment of the oil or choking up of the supply pipe, and as before, being a paraffin motor, the boat had to be allowed to drift about or the oars employed for about ten minutes, until the pressure heating lamp was set away and the carburetter heated up sufficiently to allow of the motor starting again. This little experience indicates very clearly one of the drawbacks to paraffin, as, being what is known as a "heavy" oil, it requires some considerable heating up before vaporisation takes place. At the same time, it should be stated that quite recently a patent paraffin carburetter has been devised which is said to "start from cold." Given a reliable ignition gear, the petrol motor of the four-cycle type is certainly the sweetest-running motor of any; this, at least, is the writer's experience after many trials of other types. At the same time, it is only right to state that paraffin-type motor launches give every possible satisfaction, being steady and cool in running, also easily and quietly stopped and reversed.

Ignition.—In petrol motors the greatest difficulty is experienced in obtaining a reliable ignition gear, and the most frequent cause of stoppage is undoubtedly that due to defective ignition. The best method of firing is at present a matter of much discussion and difference of opinion.

Some makers prefer the battery and "jump spark" system, others the magneto or "make-and-break" method of producing the firing spark.

The "jump spark" system offers the greatest complication of parts with correspondingly increased risks of breakdown; it also necessitates the charging of chemical batteries, and some knowledge of electrical science, while the magneto, on the "low tension" system, has fewer working parts, and being a mechanical device, driven by the shaft, presents less danger of breakdown.

Jump Spark.—In this ignition system the sparking points are both fixed, and a current of high intensity being produced in the circuit, a spark is generated between the two points referred to, and is therefore called a "jump spark."

The ignition gear is made up of four separate and independent parts.

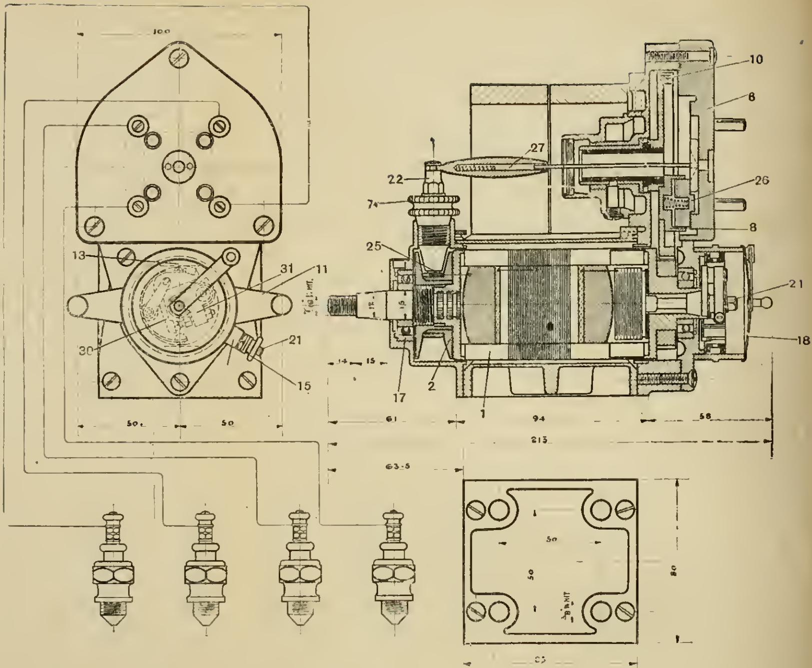
1. Chemical cells or battery.
2. Induction coil or "intensifier."
3. Contact maker.
4. Firing plug.

All of the foregoing parts are, of course, liable to derangement and breakdown, as they are affected seriously by damp and sea water.

Make-and-Break.—In the "make-and-break" system a small lever rests against a pin and at the required time is drawn away from the pin, thus leaving a small gap of about $\frac{1}{16}$ inch. The current endeavours to follow up the moving lever, and as before, a spark results. In both systems, then, the explosion is produced by means of an electric spark arranged to flash out at the correct instant, so that in the case of a four-cycle motor running at 800 R.P.M., about 400 sparks are produced per minute in each cylinder head. This means, then, a constant shower of sparks require to be produced when the motor is running.

Magneto and Spark Plug Ignition.

This system of firing is the best, being, as a rule, more reliable than the "make-and-break" or coil and battery, and lasting longer without



No. 4.—Simms Magneto (four-cylinder type).

2, Slip Ring.
6, Distributer.
7, Collector Carbon Holder.
8, Distributer Carbon Holder.
10, Half-Speed Wheel and Spindle.
11, Timing Lever.
13, Segment.
15, Milled Nut.
17, Ball Bearing.
18, Dust Cap.
21, Long Contact Breaker Screw.
22, Terminal.
25, Collector Brush and Spring.
26, Distributer Brush and Spring.
27, Central Connection.
30, Contact Breaker Lever.
31, Contact Piece.

NOTE.—The dimensions on the diagram are expressed in millimetres, 25 of which are about equal to 1 inch, therefore 50 millimetres = 2 inches (approx.).

[*To face page* 587.

requiring attention or repair. The magneto wiring is connected direct to the sparking plug, and the spark passes across the platinum points of the latter as in the coil and battery system. While the engine is running the magneto is generating the current for the spark, and, of course, when the engine is stopped the current ceases to flow, no loss of current taking place. The magneto is usually driven by gear wheels off the main shaft, and the current is taken from the small carbon brushes of the commutator by the positive wire direct to the plug, the return or negative being made by means of the engine metal back to the magneto, on the "single wire" system. The magneto spark is much stronger than that of the coil and battery system, and produces a much more powerful explosion of gas in the cylinder, giving greater power.

The Simms Magneto (Four-Cylinder Type).

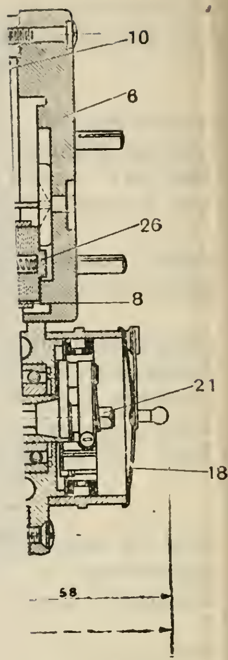
General Description.

The armature carries two windings, viz., the low-tension winding (formed of a comparatively small number of turns of thick wire) and the high-tension winding (composed of a very large number of turns of a very fine wire). This arrangement entirely dispenses with the necessity for the ordinary high-tension coil, such as is used in connection with accumulators, owing to the fact that both the high and low tension windings form an integral part of the magneto itself.

The low-tension circuit is interrupted by the contact breaker twice per revolution, thus giving two sparks per revolution, and the high-tension current is led from the carbon holders or the distributer to ordinary high-tension sparking plugs on the engine by means of the usual insulated cable.

The diagram shows the connections of the "four-cylinder" type.

One end of the low-tension armature winding is connected to the armature core, and thus to the frame of the magneto, and the other end is led through the hollow spindle of the armature to the contact breaker, the fixed portion of which, 31, is insulated, and carries a platinum pointed screw, and the bell crank lever, which carries a second platinum pointed screw, is connected to the frame of the magneto. Both the bell crank lever 30 and the contact piece 31, together with the contact breaker disc, revolve solid with the armature, and as the fibre heel of the bell crank lever comes into contact with the segments 13 the two platinum points are caused to break, owing to the bell crank lever rocking about its pivot. This sudden interruption of the primary or low-tension circuit induces a current of very high potential in the secondary or high-tension circuit, and gives a flaming spark of great heat at the points of each of the sparking plugs in turn. On the timing lever 11 is a milled nut and terminal for making connection to the insulated terminal of a single pole switch. One terminal of the switch should be connected to the frame of the car. Connected to the terminal is a spring 12, which makes contact



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The Simms Magneto (Four-Cylinder Type).

General Description.

The armature carries two windings, viz., the low-tension winding (formed of a comparatively small number of turns of thick wire) and the high-tension winding (composed of a very large number of turns of a very fine wire). This arrangement entirely dispenses with the necessity for the ordinary high-tension coil, such as is used in connection with accumulators, owing to the fact that both the high and low tension windings form an integral part of the magneto itself.

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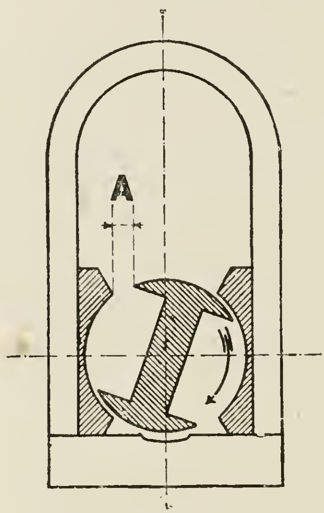
The high-tension winding of the armature has one end connected with the primary, or low-tension winding, and through this winding to earth, the other end of the secondary, or high-tension winding, being connected to the slip ring 2, from which the current is collected by the carbon brush 25. In the "four-cylinder" type, as shown, the



No. 5.

current leaves the carbon holder and passes along the central connection 27 to the distributer. A safety spark gap is formed between the hollow spindle surrounding it, the central rod passing through a glass disc 40. The central connection is held in contact with the central disc of the distributer by a spring. Connection is made between the central disc and the distributer segments by a carbon brush, which rotates with, and is insulated from, the half-speed wheel.

The safety gap is made sufficiently long to prevent the spark passing here in preference to jumping the plug gap inside a high compression cylinder.

To ensure satisfactory working, care must be taken that clean contact is made between the slip ring 2 and the carbon brush, and also between the distributer carbon 26 and the central contact piece and segments in the distributer.

Setting the Magneto.—The four segments of the distributer 6 are connected to the four terminals on the front of the distributer. The terminals are numbered 1, 2, 3, 4, in the order in which they fire. The distributer carbon holder runs in the opposite direction to that of the armature.

When viewed from the distributer end, the distributer carbon holder rotates clockwise in a right-hand machine, and anti-clockwise in a left-hand machine.

1. Set the piston of No. 1 cylinder of the motor exactly at the top of the firing stroke.

2. Fully retard timing lever, which is done by moving the lever as far as possible in the same direction as the armature rotates.

3. Rotate the armature in its proper direction until the distributer carbon rests on No 1. segment, and continue rotating slowly until the distance A (see diagram on page 578) measures 7 millimetres, then tighten all up.

4. Connect the magneto to the motor when both are in the above-described positions, taking care there is no movement. It will be found that the motor can be started more easily if the timing lever is advanced as far as possible (usually about two-thirds) without causing back-firing.

Motor Troubles. — Overfeeding with oil or petrol is a common source of trouble, as this produces a mixture too heavy for the digestion of the motor, resulting in miss-fires, premature explosions, and slowing down or actual stoppage of the engine. The majority of the owners of private motor launches have the fixed idea that if the machine is not running satisfactorily it needs more fuel, with the result above mentioned. It should also be noted that when the charge is too heavy some of the oil or petrol is not vaporised at all, and remains in the cylinder to perhaps form sooty deposits which, remaining red-hot, may afterwards produce premature explosions, or may form a carbon bridge across the points of the sparking plug and thus do away with the spark altogether. The same remark holds good if oil from the crank-case or "splash" lubrication system is carried up into the cylinder head, as similar troubles may result.

Loss of Power in Engine.—Should there be a loss of power in the engine, the following points should be examined for the cause:—

Leakage at either the exhaust or inlet valve, sparking plug, or piston rings.

Weak accumulators.

Dirty sparking plugs (or contact breakers with low tension).

Imperfect contact at the contact breaker, caused by weak spring at contact hammer, or imperfect connection of high-tension wire to plug screw, due to screw being slack.

Carbonised oil on the fibre disc and contact pieces.

Fusing or burning of the platinum on the trembler of the induction coil.

The formula very commonly quoted with reference to the economy of internal combustion motors using gasoline is, "one pint per horse-power hour." This is better than two-cycle engines can be depended on to give, and small four-cycle engines will not give any such figures except with care and under the best conditions. One pint per horse-power hour means good conditions with the four-cycle engine of moderate large size. Economy with an internal combustion engine depends chiefly on the following conditions :—

(1.) Correct proportion of air to petrol vapour.
(2.) Correct degree of compression.
(3.) Prompt ignition and complete combustion.

The first step is combustion, or union with oxygen, and as the gas itself will not burn, it must therefore first be mixed with a suitable amount of oxygen. This is accomplished most cheaply and conveniently by using air, but even then such gases, when mixed with a suitable amount of air, will not burn readily at ordinary pressures, and it is not until they are compressed to a high degree that combustion once started will act with sufficient vigour and rapidity to produce the action desired.

Leaky Pistons.—It is of the utmost importance that the piston is kept absolutely gas-tight, as should leakage occur, as will easily be seen, the motor will not develop the full power. With a leaky piston, compression will be less on the up-stroke, and the resultant explosion on the down-stroke weakened in due proportion.

Colour of Exhaust Gases.—The colour of the exhaust gases leaving the silencer affords a fair indication of the completeness of combustion in the cylinders, as if the colour is strong it indicates incomplete combustion. The more colourless, therefore, the waste gases appear, the more complete is the combustion, and only a very faint tinge of grey colour should be visible if the motor is working satisfactorily. If any of the lubricating oil passes into the cylinder it becomes burnt and gives out a disagreeable smell, and also colours the waste gases; a similar result may be produced by overfeeding with petrol or paraffin, as incomplete combustion of the gas may take place in the cylinder head.

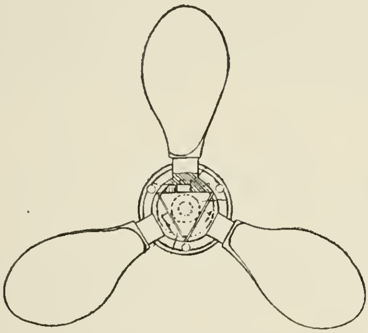
Composition of Exhaust Gases.—The following analysis of the exhaust gases of an oil engine, using Russian Petroleum, is of interest :—

$$\text{Composition of Exhaust Gases} \begin{cases} CO_2 = 9\cdot2 \text{ per cent.} \\ CO = 5\cdot5 \quad ,, \\ O = 1\cdot3 \quad ,, \\ N, \&c. = 84 \quad ,, \end{cases}$$

Reversing.—Reversing is usually effected by one of two methods.

(1.) Reversible propeller, the blades of which can be rotated to any angle by means of suitable gear contained in the hollow boss, and actuated by a lever and rod, the rod passing through the hollow tail shaft. This method is very much in favour at present, although it is not the best as regards propeller efficiency.

(2.) By a reversing clutch, which is commonly arranged to give a direct drive from engine shaft to propeller when in the "ahead" position, but which, by means of a gear box, reverses the propeller shaft rotation when in the "astern" position, the gear box in that case also revolving. It should be noted that in this case the propeller speed for astern is only about half that for ahead. When in the



No. 6.—Gaines' Reversible Type Propeller.

intermediate position, both engine and propeller shafts are disconnected, and the engine thus running free, and with the load off, requires a governor to check the speed. There are many types of patent reversing gear now in the market, all claiming, of course, certain advantages over the others.

In cases where the engine is connected to the propeller shaft through a reversing clutch, and where in consequence the engine may be running without load, the throttle, in the best designs, is placed under the control of an automatic governor of the centrifugal type, thus placing the speed under limitation with any and all conditions.

Crank Arrangements.—It should be noted that to obtain the same power, other conditions being the same, a *four-cylinder* "four-cycle" motor would be necessary to give the same power per revolution as that of a single double-acting steam-engine cylinder.

NOTE.—When four cylinders are employed in the four-cycle system, the cranks are generally arranged as follows, to produce two impulses per revolution:—

No. 1 cylinder, crank on top explosion taking place.
No. 2 cylinder, crank on bottom compression taking place.
No. 3 cylinder, crank on top admission taking place.
No. 4 cylinder, crank on bottom exhaust taking place.

Starting, &c.—Before starting a paraffin or petroleum motor, the lamp burner is set away to heat up the oil vapour, and the oil supply seen to be free to flow to the carburetter from the tank. The exhaust valves are also eased back, and the engine turned round by hand to force out any gas left behind in the cylinders. After the vaporiser is sufficiently heated up, the supply cock to carburetter is opened and the engine turned quickly by hand to start. If starting prove difficult, it might be caused by insufficient heating up of the vaporiser or by dirty or defective sparking gear. The oil supply should also be examined in case the petrol is not flowing into the carburetter. Piston speed is varied by means of the governor, which regulates the petrol supply and sometimes the time of ignition. The fewer the number of ignitions per minute, the slower the piston speed, and *vice versâ*, the speed of the vessel varying in proportion. A petrol motor can be started at once by merely "cranking" and adjusting the spark, as the petrol spirit vaporises instantly on passing the spray nozzle of the carburetter.

Speed Regulation.—The speed of the engine is varied by one of the following methods:—

(1.) Advancing or retarding the period of sparking, as advancing the spark increases the speed, and retarding the spark decreases the speed.

(2.) Shutting down the petrol supply to the carburetter.

(3.) Throttling the amount of the charge passing into the cylinder.

(4.) Increasing the proportion of air to oil vapour, and thus weakening the charge.

(5.) Easing the compression, and thus reducing the force of the explosion.

Sometimes a combination of two of the above systems is employed, and the governor is often arranged to both reduce the petrol supply and retard the spark simultaneously.

Internal Combustion Engine Troubles.

Symptom.	Cause.	Remedy.
Engine refuses to start.	1. Faulty ignition. 2. Weak compression. 3. Want of petrol or water in petrol. 4. Spark plug broken. 5. Weak mixture. 6. Spark plugs screwed into wrong cylinders.	Examine wiring, test plugs, test for compression, put petrol into cylinder through compression cock, take out plugs and test for spark in each cylinder.

Internal Combustion Engine Troubles—*continued.*

Symptom.	Cause.	Remedy.
Engine stops suddenly.	1. No spark. 2. No petrol. 3. Choked carburetter jet. 4. Fault in coil or in magneto. 5. Wiring short circuited to metal of engine.	Test spark plugs, test for petrol supply, clear carburetter jet, examine and test battery, coil, and wiring for short circuit or broken wire, &c.
Engine stops gradually and miss-fires.	1. Weak spark. 2. Want of petrol. 3. Carburetter partly choked. 4. Spark plugs fouled up with oil. 5. Wrong mixture.	Test spark plugs and examine coil contacts, examine carburetter, examine petrol tank to see if air-bound, &c.
Overheating.	1. Faulty circulation. 2. Pump broken down. 3. Steam lock in pipes. 4. Mixture too rich in petrol. 5. Spark retarded. 6. Exhaust throttled. 7. Valve timing wrong. 8. Silencer choked up with carbon.	Test pumps and pipes for water circulation, increase air supply to carburetter, test timing of valves, examine silencer.
Engine knocking.	1. Faulty lubrication. 2. Ignition advanced too far. 3. Premature ignition. 4. Worn bottom, end, or main bearings. 5. Cylinders loose on bed plate.	Examine and test lubrication system, put back ignition lever, overhaul engine for worn bearings, &c.
Hissing noise from engine.	1. Spark plug broken, joint blown out between engine and exhaust or between engine and carburetter. 2. Compression cock partly open.	Test spark plugs, examine joints mentioned, &c.
Loss of engine power.	1. Loss of compression due to leaky piston rings. 2. Too much petrol. 3. Want of lubrication. 4. Throttled exhaust. 5. Wrong adjustment of valves. 6. Faulty wiring, coil, battery or magneto.	Test pistons for compression, reduce petrol flow, examine lubricator, test timing of valves, examine and test battery, coil, wiring, and magneto.

Internal Combustion Engine Troubles—*continued.*

Symptom.	Cause.	Remedy.
Hot crank-case.	1. Leaky pistons allowing gas to enter crank-case. 2. Cracks in cylinder.	Test for compression, examine for cracks.
Explosions in silencer.	Mixture too weak to fill in cylinder, one cylinder missing and charge afterwards exploding in silencer.	Increase petrol supply, test plugs for spark and for timing of spark.
Red-hot silencer.	1. Using excessive gas. 2. Spark too far retarded. 3. Exhaust throttled. 4. Faulty adjustment of valves. 5. Choked up silencer.	Give more air, advance spark, test for timing of valves, examine silencer.
Engine refuses to stop.	1. Short circuit of wiring. 2. Carbon deposit in cylinder head or on plug spark points (red hot).	Test wiring, examine plugs and cylinder heads for carbon deposits.
Engine refuses to start.	Cylinders full of water owing to leaky silencer or cracked cylinder.	Examine silencer and cylinder heads, &c.
Blue smoke from silencer.	Overheating due to faulty lubrication or defective water circulation.	Examine and test lubrication system, and water circulation.
Engine stops when reversed.	Clutch jammed owing to dirt, grinding of metal, or want of lubrication.	Clean clutch and lubricate.
Engine refuses to take up drive.	Clutch slipping.	Examine clutch for wear, clean clutch if over lubricated, fit new collar if required.

Carburetter.—The carburetter needle valve should be ground in regularly, as if not the mixture will become too rich, and in some cases the carburetter may become flooded, resulting in loss of power and probable stoppage of the engines.

Tests.

Spark Plugs.—Screw out the plug to be tested, and lay it on the cylinder top with the wire connected up; now turn engine round by hand with switch on, and if there is no fault a spark will show across the points.

NOTE.—With the spark retarded the spark should show just as the piston is passing the top centre, but if the spark appears when the piston is at half stroke or thereabout, it indicates that the plugs have not been put into the right cylinders, or that the timing of the spark is wrong. This fault shows when starting up by back firing out of the air inlet.

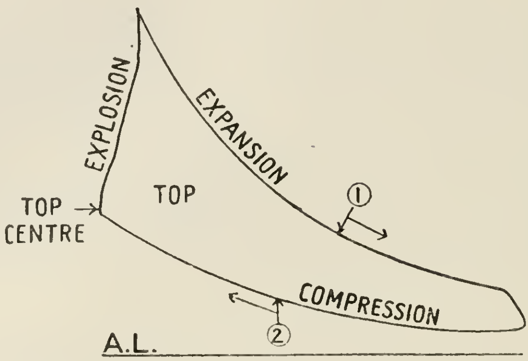
Valves.—To test the timing of the valves, open compression cock on cylinder top, and fit in a straight copper wire, now move engine by hand and see if the inlet valve lifts just as the piston comes to the top, and if the exhaust valve lifts just *before* the piston comes to the bottom (every second stroke for a four-cycle motor); if not, alter the washers on ends of valve lifting rods to either increase or decrease the clearance as required. The exhaust valves should be set to close down just before the piston commences the down (inlet) stroke.

Battery.—For battery testing a small pocket voltmeter is required, and when connected up with the positive and negative terminals of the cell, the voltage indicated should not be less than 4·2. The cells should be recharged every four or six weeks.

Diagrams from Oil Motors.

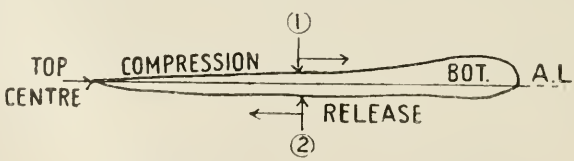
Diagrams Nos. 7 and 8 show clearly what goes on above and below the piston in this type of motor, as on the down-stroke simultaneous expansion of gas after explosion above the piston and compression of the next charge below the piston is taking place, and on the up-stroke simultaneous admission of oil vapour below the piston and compression of the previous charge above the piston. Observe that line 1 in A and line 1 in B are described at the same time, also line 2 in A and line 2 in B at the same time. As the bottom card represents work done *on* the air and oil vapour by the piston, the actual work done is equal to the difference in area of the two diagrams, A and B, and the mean effective pressure is found by making this allowance.

Diagram No. 9, taken with a light spring, shows clearly the four operations which constitute the "Otto" cycle, so named from Dr Otto, who first applied this principle to gas engines. (1.) The lowest line shows the admission of air and oil vapour at a pressure rather



No. 7.—Diagram from Cylinder of Two-Cycle Motor.

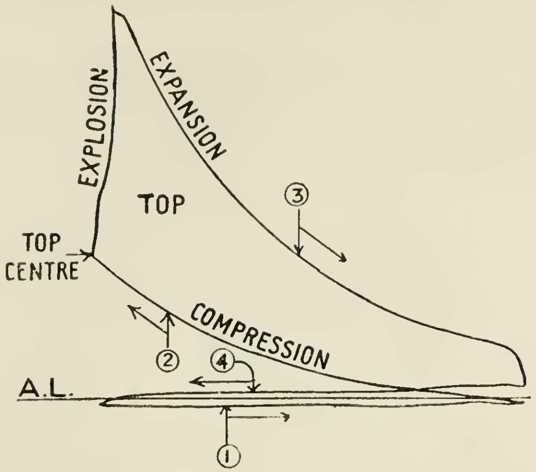
below that of the atmosphere, the indicator pencil travelling from left to right. (2.) The line rising and going from right to left shows the compression of the gas on the up-stroke, the pressure increasing to about 80 lbs. per square inch or so. (3.) At, or just before, the top



No. 8.—Diagram from Crank-Case of Two-Cycle Motor.

centre, the charge of air and oil vapour is fired and explosion follows, as shown by the sudden rise of pressure to about 240 lbs. or more; the effect of the explosion is to drive the piston down, the pressure falling at the same time by expansion and loss of heat. (4.) On the next up-stroke the exhaust valves open and the burnt gases are expelled at a pressure just above that of the atmosphere, which is seen by the sloping line crossing that of the compression curve. After this the cycle begins again and repeats itself. The arrows show the direction of indicator pencil travel.

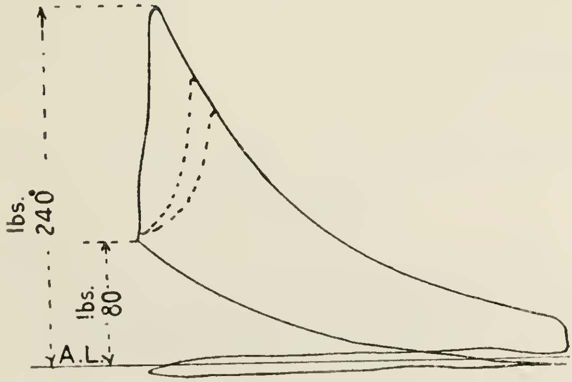
The small numbers shown in the sketch indicate the successive stroke in the same rotation as follows:—



No. 9.—Diagram from Cylinder of Four-Cycle Motor.

1, Admission. 2, Compression. 3, Explosion and Expansion. 4, Exhaust.

(1.) Down-stroke, air and oil vapour admission to cylinder.
(2.) Up-stroke, air and oil vapour compression in cylinder.
(3.) Down-stroke, air and oil vapour explosion in cylinder.
(4.) Up-stroke, burnt gases (CO_2 and N) expelled from cylinder.

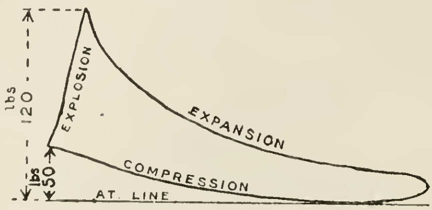


No. 10.—Typical Four-Cycle Diagram.

No. 3 stroke is the only working stroke of the four, and is known as the "impulse" stroke, so that there is only one *power stroke* in every four strokes, or in two revolutions. So that 800 revolutions per minute require 400 sparks in each cylinder of the motor.

Diagram No. 10 shows the average pressures obtained in ordinary petrol motors, and the dotted lines show the effect of retarding the spark or ignition, which, it should be noted, has the effect of slowing down the engine.

The loop shown in both the foregoing diagrams, and caused by the crossing of the admission and compression lines, represents negative work, and must be deducted from the diagram area to obtain the effective work done, and to measure the mean effective pressure in calculating the I.H.P.



No. 11.—Heavy Spring Diagram.

NOTE.—The scale of the diagram being small, the difference between the admission line, atmospheric line, and exhaust line is imperceptible, as the three lines merge more or less into one.

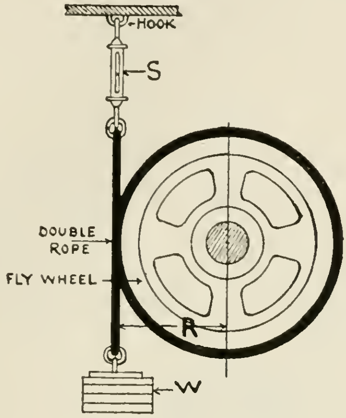
Mean Pressure.—As an oil engine is single acting, the mean effective pressure is calculated from a *single* diagram, the method adopted being similar to that for a steam-engine. The card is divided into, say, ten divisions, or nine whole divisions and two half divisions, and the pressures measured by the scale of the diagram on each line. The ten pressures are then added together, and the result divided by ten gives the mean effective pressure.

Indicated Horse-Power.—The I.H.P. is found by the following formula, viz. :—

$$\frac{A \times S' \times N \times P}{33000} = \text{I.H.P.}$$

Where A = piston area in square inches.
S′ = stroke in feet.
N = number of explosion strokes per minute.
P = mean effective pressure (from card).
33000 = foot-pounds per minute per I.H.P.

Brake Horse-Power.—In high-speed engines the I.H.P. as found by calculation cannot always be relied upon; it is therefore more satisfactory to state the B.H.P., or actual power transmitted to the propeller. The friction brake required to obtain this consists of a double rope passed once round the fly-wheel, one end being secured to a spring balance hung overhead, and the other end supporting the required amount of weights to balance the spring, when the engine is running the average number of revolutions.



No. 12.—Friction Brake.

S, Spring Balance. W, Weights. R, Rope Radius in feet.

NOTE.—The shaft is revolving from left to right.

The B.H.P. is found as follows :—

$$\frac{(W - S) \times R \times 2 \times 3{\cdot}1416 \times \text{Revolutions}}{33000} = \text{B.H.P.}$$

NOTE.—The difference in pounds of the weights and spring is the actual pull.

W = Pounds weight on rope.
S = Pounds shown by spring balance.
R = Radius of rope from shaft centre.
2 = Twice radius for diameter.

Types of Motors, &c.

The following descriptions of modern marine motors and motor details are reprinted from various issues of "**The Motor Boat**," to the proprietors of which journal the author's thanks are due for permission to reproduce both text and illustrations.

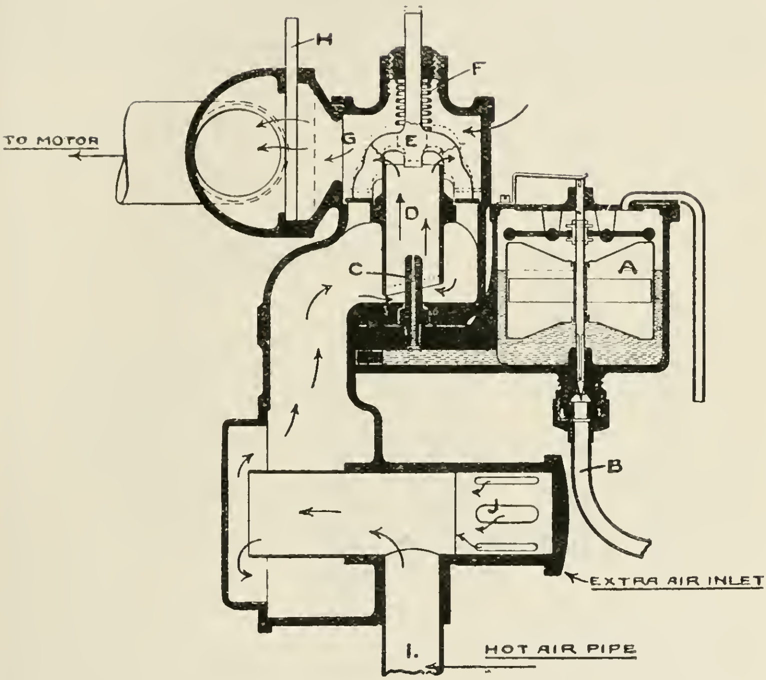
The Wolseley Carburetter.

This carburetter is of the float feed automatic type, fitted with an auxiliary hand-controlled extra air inlet ; the float chamber and float (A) are of the usual construction, and serve the customary purpose of maintaining a constant level of the petrol, which is supplied through the pipe (B) in the spraying nozzle (C). Surrounding this nozzle is a choke tube (D), supported by arms (E) fixed to a disc, the section of which may be seen in the illustration, the whole being capable of a vertical movement against the action of the helical spring (F). The passage of the air when the engine is running is clearly shown by the arrows, entering through the pipe (I), which leads from the neighbourhood of the exhaust system in order that warm air may be obtained; it passes up through the choke tube, and drawing the petrol from the nozzle (C) in the form of a spray forms the explosive mixture which issues from the top of the tube and reaches the motor through the throttle (H) and the induction pipe. When the speed, and thus the suction of the engine, increases, the disc attached to E is raised from its seating, lifting with it the choke tube to the position shown by the dotted lines, and air is allowed to travel through the ports normally covered by it to the chamber (G) without passing over the spraying jet, thus automatically adjusting the proportion of vapour and air which is supplied to the motor. The pressure of the spring (F) tending to keep the disc on its seating may readily be varied by altering the position of the screw cap which may be seen in the illustration, thus enabling very accurate adjustment to be made. J is an extra air inlet which may be moved by hand when desired.

This carburetter has proved itself to be thoroughly satisfactory in use, the moving parts are very accessible, and, as has been mentioned above, accurate adjustment can be made without dismantling any part of the apparatus.

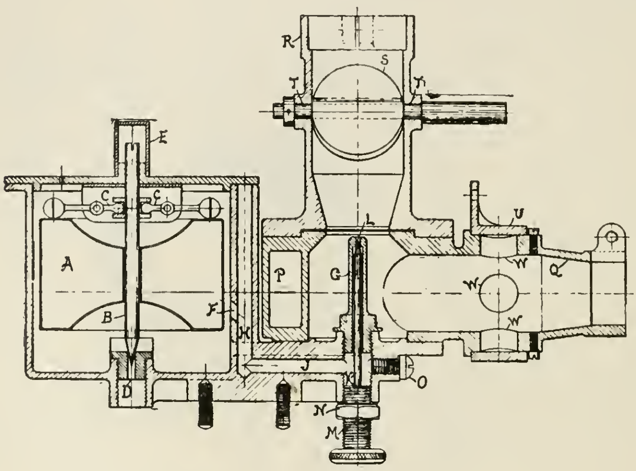
Thornycroft Type Carburetter.

It will be seen upon reference to the figure that the carburetter consists of a float chamber and a vaporising chamber located in very close proximity to one another. When the float (A) sinks, the needle valve (B) is lifted by the balance levers (C), and petrol is allowed to enter the chamber through the inlet (D). The top of the spindle (B) is protected by a cap (E), which may be removed and the valve (B) lifted when it is desired to flood the carburetter. Leaving the float chamber by the port (F), the petrol gains access to the spray nozzle (G) through the ducts (H and J). K is an adjustable spindle, enabling the spray at L to be regulated by the movement of the screw (M), a lock nut (N) maintaining the correct position of this screw when the adjustment has been made. Should the duct (J)



No. 13.—Wolseley Petrol Carburetter.

become obstructed it may be readily cleared upon removing the screw (O), whilst the duct (H) is continued to the top of the float chamber for the same purpose. The vaporising chamber is warmed by a hot-water jacket (P), which is connected with the engine-cooling system. This carburetter is of the hand-controlled type, no automatic air valve of any kind being relied upon. The main air supply is drawn in through the pipe (Q), which has its intake in proximity to the exhaust piping in order that warm air may be obtained, and,



No. 14—Thornycroft Carburetter.

passing over the jet (L), forms the explosive mixture which is carried to the engine through the induction pipe (R). The throttle valve (S) is of the ordinary disc type, pivoted in the induction pipe at T. The auxiliary air supply is controlled by a revolving collar (U), having four holes bored in it, which may be made to coincide with four holes —three of which may be seen at W, bored in the pipe Q ; thus, by moving the collar, the amount of air drawn in through the ports (W) may be regulated.

The chief feature of this carburetter is its simplicity, whilst its substantial construction should enable it to withstand a considerable amount of rough usage.

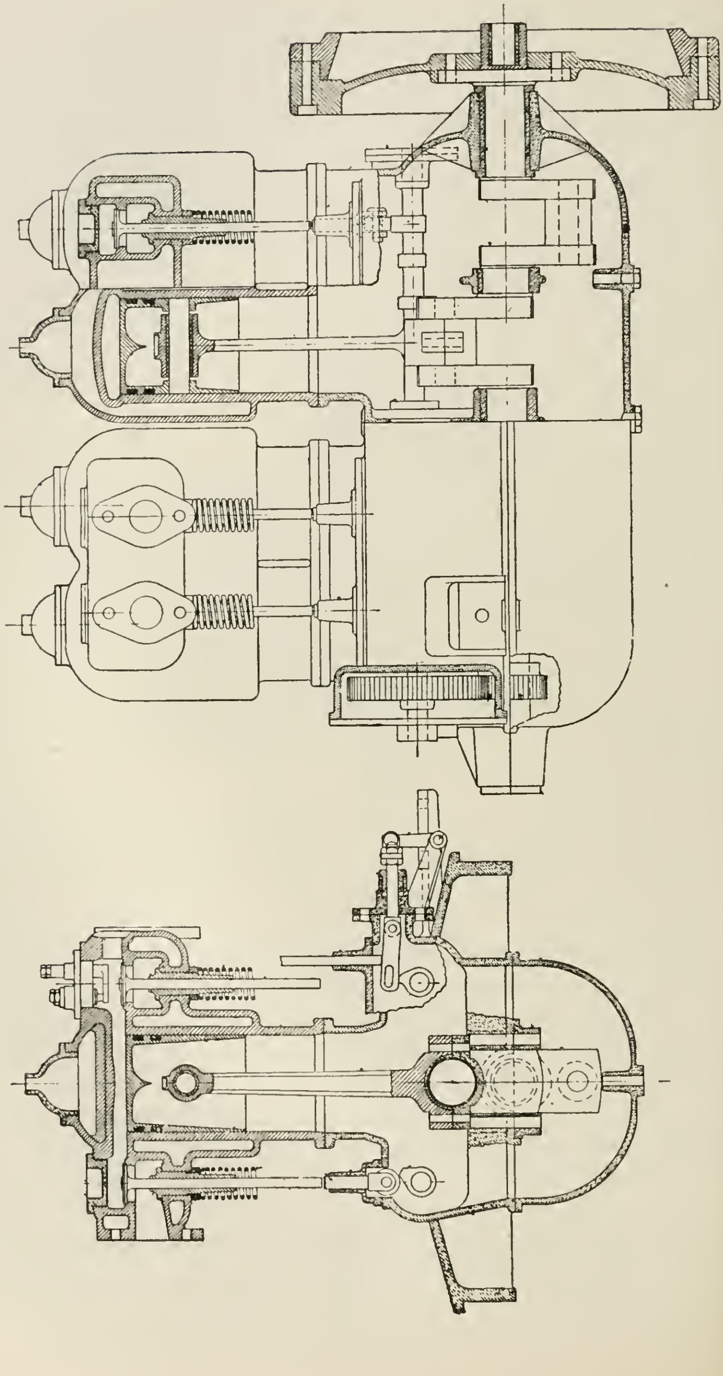
18 H.P. Brooke Motor.

The 45 H.P. four-cylinder engine has all cylinders separately cast, and in one piece with their combustion heads, the bore being $5\frac{1}{2}$ inches and the stroke 6 inches, with a normal running speed of 900 revolutions per minute. Water-jacketing of the valve pockets has been very well carried out, as may be seen from the cross section, and they are provided with extra large screw-in valve caps to simplify the operation of removing the valve stem guides should this be necessary. Except that the ends of the valve springs are hooked through the valve stems instead of the more complicated collar and cotter arrangement, there are no special points in the valve and tappet gear, the tappets being of the ordinary roller type.

Both cam-shafts are carried by ordinary end bearings bolted to the ends of the crank-case, and by split intermediate bearings supported by the cross-webs and held in position by set-screws. For lightness the crank-shaft is hollow, as also are the crank-pins, one end of the shaft carrying the pinion of the two-to-one gear, the pinion and both spur wheels being of phosphor-bronze, as the firm find that quieter running is possible than with steel, while the bad wearing qualities of fibre wheels are avoided. On the other end of the crank-shaft is the combined fly-wheel and clutch, which is of a special type, very ingeniously arranged, so that the thrust due to the clutch is entirely self-contained.

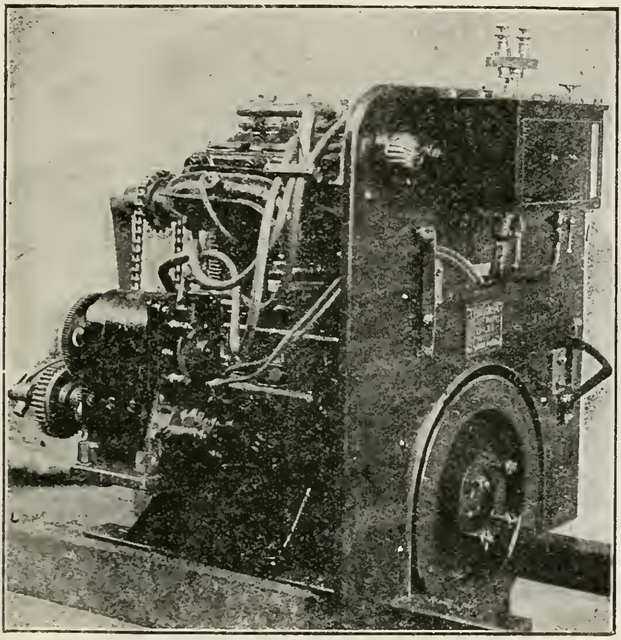
The entering member of the clutch is free to slide on the squared end of the tail shaft, and is normally kept in engagement by a spiral spring bearing at its after end on a collar of the tail shaft. The extreme forward end of the tail shaft is enlarged to form a flange (or there may be a disc bolted to the end face of the shaft), and this flange is inside a flanged ring bolted to the after face of the engine fly-wheel. Between these two flanges is a row of ball bearings, and from the illustration it will be evident that these balls take the whole thrust due to the clutch spring, which is tending to pull the tail shaft aft, and also tending, with an equal force, to push the entering member of the clutch forward. This thrust is transmitted to the other clutch member, and then falls on the ball bearing between the two flanges already described, so that the two stresses exactly neutralise each other.

Returning to the crank-shaft. There is a main bearing between each pair of cylinders, and these bearings, together with the big end and cam-shaft bearings, are white metal lined, it being considered that a really badly-heated bearing causes less damage by running out



No. 15.—Cross Section and Side Elevation of Brooke 18 H.P. Engine.

the metal and starting a knock that makes it instantly necessary to stop the engine, than would be done by the constant scoring that is liable to occur with phosphor-bronze, which will continue to run until it seizes up. With a view to reducing vibration as far as possible, the pistons are made very light, with thin strengthening webs; they each have only two grooves for piston rings, but there are two rings in each groove, the idea being, of course, that gas getting through the slot in one ring does not have an annular space (caused by the comparatively loose fit of the piston in the cylinder) by which to reach the slot in



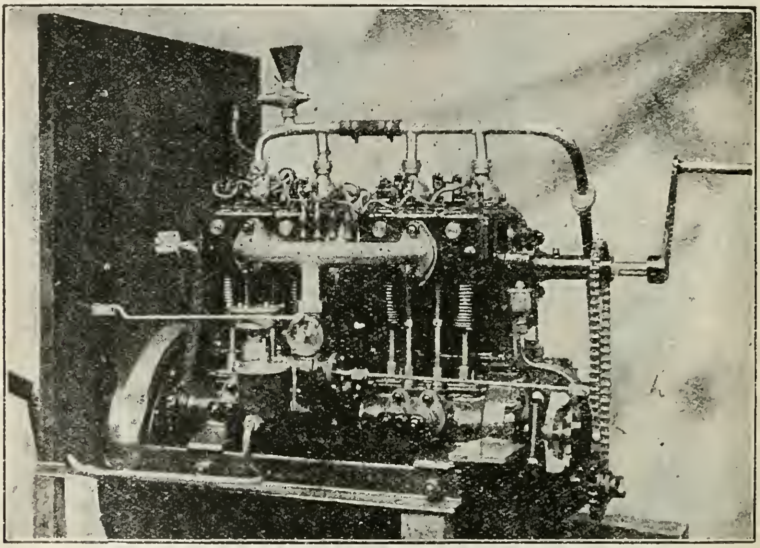
No. 16.—Brooke Engine with Control Board.

the next ring, but is stopped at once by the second ring in the slot; it is for this reason that only two grooves are necessary. With mention of the fact that the connecting rods are of H sectioned steel stampings, the description of the engine proper is concluded.

Water circulation is maintained by a rotary pump of the eccentric type driven off the cam-shaft, and the direction of the flow of water in the jackets is, to a certain extent, guided by a web inside the jackets, serving the double purpose of a baffle plate and of giving extra strength to the cylinders.

Splash lubrication is relied upon entirely, oil being fed in at the forward end of the crank-case from a sight-feed lubricator. There is no separate feed to the pistons, as Messrs Brooke & Co. consider that this system leads to sooting-up of the cylinder and a dirty exhaust; the piston truck, however, gets a good dose of oil at the bottom of the stroke, since it projects a little below the cylinder, and there is a point on the bottom of the internal web of the pistons which catches oil and allows it to drip in a little pocket on the top of the connecting rod to feed the gudgeon-pin bearing.

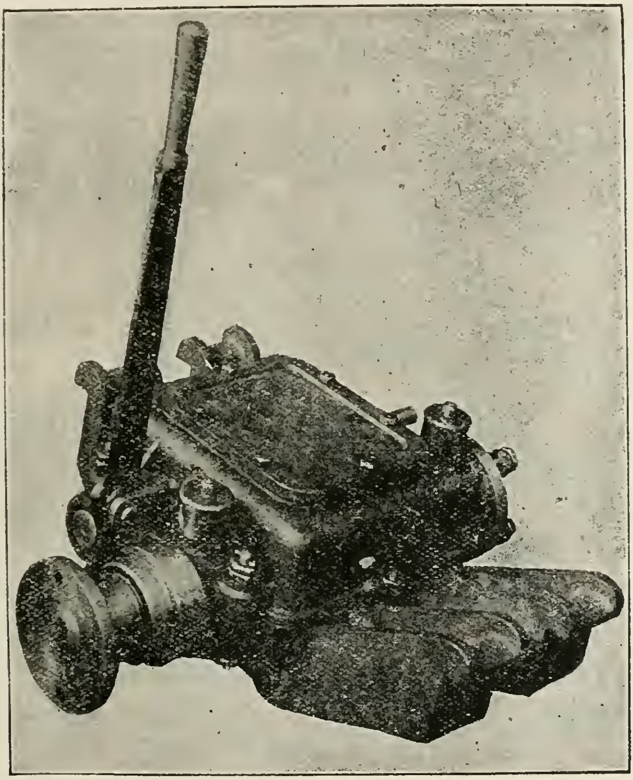
Little need be said about the Brooke carburetter, as it was recently described in this journal, but it may be repeated that the throttle governor attached to the carburetter is one of the special features



No. 17.—18 H.P. Brooke Engine.

with which the Brooke engine is fitted. As regards ignition, there is a tendency to abandon the low-tension magneto, which was at one time much favoured by this firm, in favour of the latest Simms high-tension pattern, which gives an extremely hot spark of very high frequency, and is not therefore nearly so much affected by wet as the ordinary high-tension electric circuit. Another point in favour of this magneto is that starting is as easy as with coil and battery, so low is the speed at which a spark is obtained.

The Meissner reversible propeller is supplied with a considerable number of Brooke engines, but to those who prefer reversing gear a very neat type has been evolved from the Adrian works. It is of the sliding gear type, giving a direct drive ahead through a dog clutch no wheels being in motion except those which are out of mesh. On the reverse the drive is, of course, through two countershafts. The gear is very compact and efficient, in addition to which it can be



No. 18.—Brooke Reverse Gear.

got very close to the bottom of the boat, is perfectly silent going ahead, and far quieter astern than is usually the case.

OIL FUEL.

This method of firing has recently come very much to the front, particularly in naval practice, and a brief description of the system will not be out of place The chief drawbacks to the use of oil fuel at present are those of supply and of cost; oil supply ports being few in number, although in time this matter will be remedied. The cost is also a consideration, but this will also be adjusted to meet the requirements of the demand. It may safely be stated that as fuel, oil has a great future before it in marine practice.

Advantages of Oil Fuel over Coal.

1. Less bunker space required (about 36 cubic feet against 44 cubic feet per ton).
2. Greater heat per pound (20,000 B.T.U. for oil against 14,500 B.T.U. for coal).
3. Cleanliness both in working and bunkering.
4. Reduced stoke-hole staff.
5. Greater control of fires.
6. More complete combustion obtained.

Disadvantages.

1. Difficulty of obtaining oil supplies.
2. Cost.
3. Danger from inflammable vapour caused by leakage into bilges, &c.
4. Danger of oil leaking into steam side of heater and finally entering boilers.

Oil and Coal Compared.

Fuel.	Heat units per pound.	Bunker space per ton.
Coal - - -	14,500	44 cub. ft.
Oil - - -	19,000	36 ,, ,,

NOTE.—Taking into account both heating value and bunker space, one ton of oil is equivalent to 1·6 tons of coal.

The U.S. Naval Department Committee report on the advantages of oil fuel as follows:—

1. A greater evaporative efficiency in ratio of about 14 to 9.
2. A reduction in the fire-room force.
3. Elimination of ashes and dirty fires.
4. Convenience of storage. Oil tanks may be located in double bottoms, in spaces now useless.
5. Rapidity and ease of taking aboard and handling. The manual labour in this connection is eliminated.
6. Easy control of fires, permitting sudden variations in power developed by boilers.

7. Facility of controlling proportions of the air and fuel, thus ensuring good combustion. There is no opening or shutting of furnace doors of varying thicknesses as is the case with coal.

8. Elimination of cinders and of smoke, except at full power.

9. The reduction of fire room, there being no space required to permit working of the fires.

10. As there is still a much better distribution of coal among the seaports of the world than oil, this is said to be one of the principal disadvantages of oil.

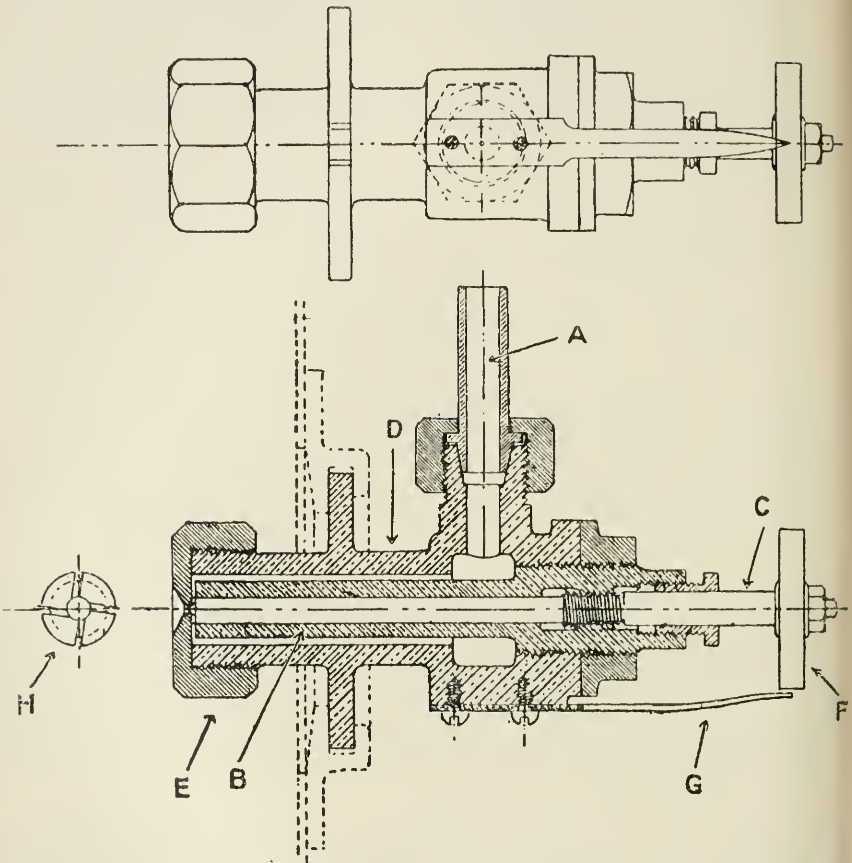
Composition of Oil.—The average composition, &c., of the oils used as fuel are as follows :—

Class.	Flash Point.	Specific Gravity.	Carbon (C).	Hydrogen (H).	Oxygen (O).	Heat Units per Lb.
			Per Cent.	Per Cent.		
Burmah - -	200°	·92	86	12	1·5	18,800
Shale - - -	125°	·81	86	13	1	19,000
Russian Petroleum	120°	·822	86	14	0	20,000

Methods of Working.—Oil is pumped into the tanks which act as bunkers, and is afterwards pumped into direct supply tanks known as "settling tanks." These tanks are generally placed at some height above the floor plates, and are intended to allow any water which may have become mixed with the oil to settle to the bottom, leaving the pure oil on top. This oil, after being heated and filtered, is then pumped under pressure direct to the burners which are fitted on the front of the furnaces.

Oil Spray.—The oil, under pressure, is sprayed into the furnace through the nozzle and needle valve of the former, and this results in atomising of the oil which flashes up just after leaving the nozzle point, the effect produced being that of a mass of gas at white heat filling up the entire space of the furnace or combustion chamber.

Burners.—In one system the oil jet is mixed with steam as it enters the burner, and in another system air only is mixed with the oil jet and forced through the burner. In the Navy, however, the oil, at a pressure of about 100 lbs. and temperature of 200° Fahr., is forced through the burner, and the air (of usual forced draught pressure) enters the slots of a specially designed cone, and so mixes with the rotary spray. The burner itself sits inside the air cone at a slight upward angle, and the air enters first by air doors on the boiler front, and from these into the slots of the cone previously mentioned. Each cone is boxed in by division plates so that the air supply is localised to the corresponding burner.



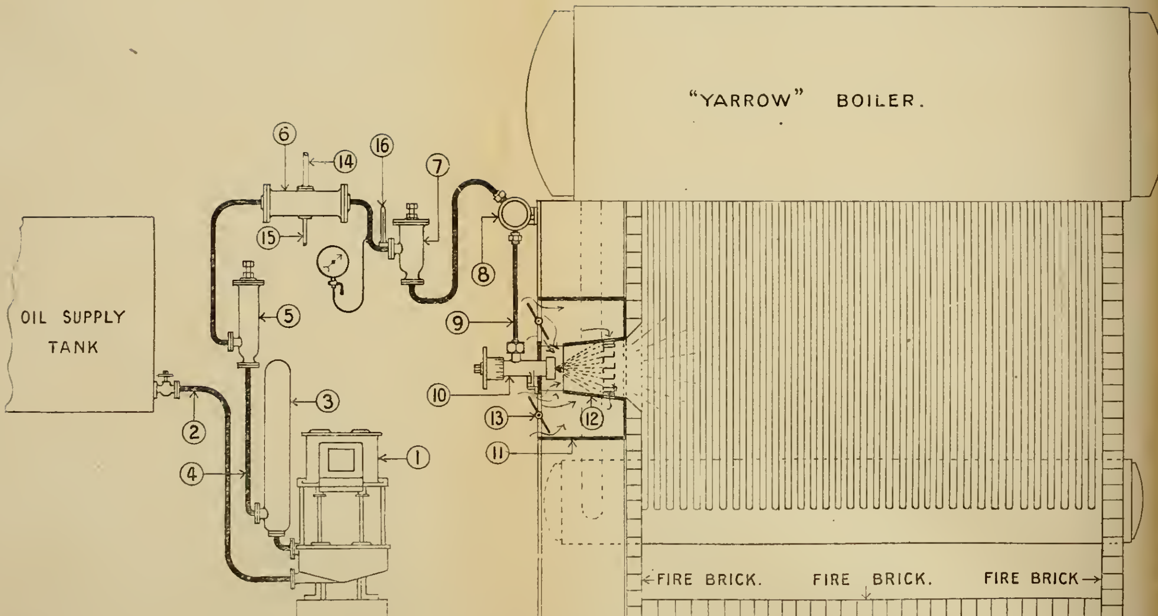
No. 19—"Kermode" Type Oil Burner.

The Admiralty type burner is similar in principle to the above, but is of much improved design, giving higher efficiency.

A, Oil feed to burner.
B, Nozzle.
C, Regulating spindle.
D, Burner body.
E, Cap nut.
F, Graduated wheel.
G, Pointer.
H, Grooves in nozzle end.

Shale Oil.—With Scotch shale oil a heating temperature of about 125° Fahr. is generally sufficient.

Specific gravity of CO = ·96.



No. 20.—Oil Fuel System.

1, Fuel pumps.
2, Suction from supply tanks.
3, Air vessel.
4, Discharge from pump to cold filter.
5, Cold filter or strainer.
6, Heater.
7, Hot filter or strainer.
8, Distribution chest or header.
9, Oil fuel to burner.
10, Oil burner.
11, Air-tight box.
12, Air cone
13, Air doors.
14. Steam to heater.
15, Drain from heater to feed tank.
16, Thermometer for oil temperature.

The above is the system in use in the Navy.

[To face page 61..

Working Oil Fuel.—The oil fuel is pumped from the supply tanks by the oil pumps, and forced through a cold filter, then through a steam heater where the temperature is raised, next through a second or hot filter, and from there to the distribution header on the boiler front, from which valves and pipes connect to the several burners fitted. In Yarrow type boilers from eight to eleven burners are supplied, but of course all of these may not be required except when steaming under full power conditions. The pressure of the oil is about 100 lbs., and the temperature 200°, with "shale" or "Texas" oil as fuel.

Control.—The regulation of the fires is controlled by the needle valve of the burner, which can be altered to increase or decrease the angle of discharge, and therefore the output or consumption; one small wheel on the burner constituting the entire control gear. The combustion of the oil is therefore regulated by the oil needle valve and the air supply doors.

The oil fuel is pulverised by being forced, under pressure, through the restricted opening of the burner end, which, by means of the grooves, imparts a rotary motion to the jet, the latter being distributed in a cone-like spray or cloud of pulverised oil particles.

In the Navy the closed stoke-hole forced draught system is employed, which means a constant and steady air pressure on the oil fuel.

Starting Up.—In starting up the fires, a hand pressure pump is employed, which forces a small quantity of oil through a U-shaped and flexible tube which is placed inside the air cone; and, having previously set fire to a quantity of oily waste, the heat so produced raises the temperature of the oil flowing through the U tube on its way to the starting burner. After the starting burner has been the means of raising sufficient steam to set away the oil pumps, the starting device described is withdrawn, and the other burners are set away. About three hours is allowed to get up steam in water tube boilers.

Leakage Test.—Before the fires are started a leakage test of the oil system is generally made by raising a pressure of 50 lbs. or more with the hand pump, and examining for leaks at the tanks, pipes, joints, bilges, &c.

Colour of Gases.—The colour of the gases in the combustion chamber space indicates the efficiency of the combustion taking place inside, and the pressure and temperature of the oil, in addition to the output required, is regulated accordingly. A very small tail of smoke at the funnel top indicates that combustion is practically complete. The colour of the gases can be observed by means of sighting holes in the boiler casings.

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Starting Up.—In starting up the fires, a hand pressure pump is employed, which forces a small quantity of oil through a U-shaped and flexible tube which is placed inside the air cone; and, having previously set fire to a quantity of oily waste, the heat so produced raises the temperature of the oil flowing through the U tube on its way to the starting burner. After the starting burner has been the means of raising sufficient steam to set away the oil pumps, the starting device described is withdrawn, and the other burners are set away. About three hours is allowed to get up steam in water tube boilers.

Leakage Test.—Before the fires are started a leakage test of the oil system is generally made by raising a pressure of 50 lbs. or more with the hand pump, and examining for leaks at the tanks, pipes, joints, bilges, &c.

Colour of Gases.—The colour of the gases in the combustion chamber space indicates the efficiency of the combustion taking place inside, and the pressure and temperature of the oil, in addition to the output required, is regulated accordingly. A very small tail of smoke at the funnel top indicates that combustion is practically complete. The colour of the gases can be observed by means of sighting holes in the boiler casings.

Flash Point and Firing Point.—It should be noticed that the spray of atomised oil at a flash point of 200° is below the "firing point," but on striking the cone ring, the temperature of which exceeds the flash point, ignition instantaneously takes place.

Firing Point.—The firing point of oil is above the flash point, and means that the oil itself (instead of the vapour) ignites.

Flash Point.—By this is meant the temperature at which the vapour formed from heated oil flashes into flame when brought in contact with a light. The flash point varies from 70° in light petrol spirit to about 240° in heavy burning oils. The Board of Trade require a flash point of not less than 185° Fahr.

Sand.—As a safeguard against fire, boxes of sand are placed ready for use in the stoke-holes, as the sand thrown on oil flame quickly extinguishes the same.

Black Smoke.—This is caused by the temperature or pressure of the oil fuel being too low for complete combustion.

White Smoke.—This may be caused by excessive air supply or by faulty oil feed through the burner.

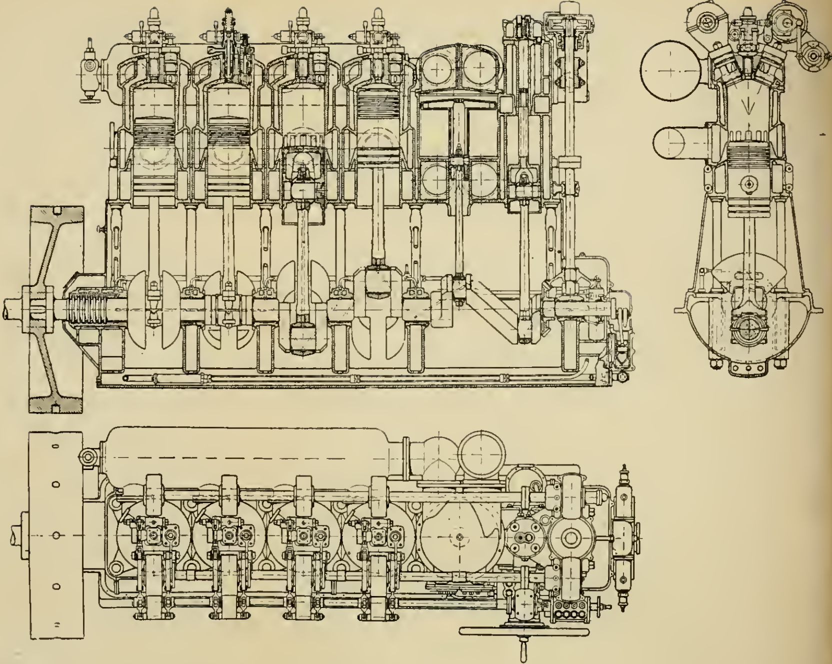
Ventilation Pipes.—To allow for the escape of oil vapour "swan-neck" pipes should be led from the top of the oil tanks to the deck, the ends of the pipes being open, but covered with gauze wire to reduce risk of explosion by a naked light. When the tanks are empty, however, the vapour formed by slow evaporation is much heavier than the atmosphere, and will therefore occupy the *lowest* positions in the tanks, and the gases thus formed are best removed by means of exhausting fans.

Air Vessel.—To maintain a steady oil pressure in the sprayers an air vessel is fitted on the discharge side of the oil supply pump.

Settling Tanks.—Sometimes these tanks are fitted with a steam coil to heat up the oil, the effect of which is to separate more quickly the water; the heating up causing a greater difference in the respective densities of the two liquids. The heating is done by exhaust steam of low pressure and temperature, so that there will be no danger of the oil vaporising. A gauge glass is fitted on the tanks, and the water shown can be drained off by suitable drain pipes.

A temperature of 180° is required to produce separation of the water from the oil in the settling tanks.

Air Cone.—The air cone (Sketch No. 29) is fitted with small air openings round the shell, these being formed by three-sided cuts, the



No. 21.—Sulzer Marine Diesel Engine, 200 I.H.P.

Dial Run	Speed	10·6 knots.
	Revolutions per minute	300.
	I.H.P.	174.
	Fuel per I.H.P. per hour	·40 of a pound.
	Cost of fuel per mile	1·5 pence.

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fourth side being bent outwards to form the air opening. This arrangement gives a centrifugal motion to the entering air and to the oil spray, which effect chiefly accounts for the efficiency of this system.

Evaporation of Oil.—1 lb. of oil fuel evaporates about 15 lbs. of water into steam (from and at a temperature of 212°).

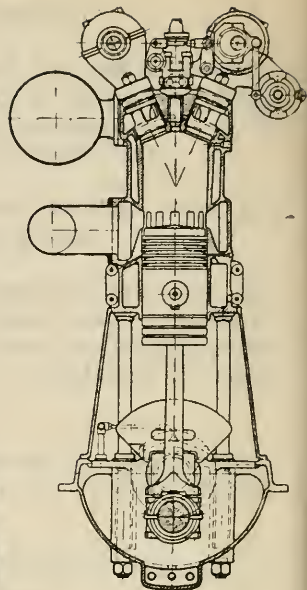
Water in Oil.—In burning oil fuel, water shows by the oil forming a brown coloured foam near the burner nozzle. Sputtering also occurs at the burner, and if the water present is excessive, the burner flame may go out altogether.

White Vapour.—In burning oil fuel white vapour at the funnel top indicates that the oil vapour is passing off unconsumed owing to excessive air supply, which lowers the temperature of combustion, with the result stated.

Diesel Oil Engine.

Regarding the development and application of the Reversing Internal Combustion Engine on a large scale, it is the opinion of most experts that a great future exists for this type of engine, and it would appear that this future is by no means far distant, since all the more important yards are preparing to supply Diesel Marine Engines, and, in many instances, the construction has already been begun.

It was apparent, however, that not being direct reversing, its application to large steamers was impossible, inasmuch as the systems employed for reversing the screw by means of revolving blades or reversing the propeller shaft by special gear, could never offer the same amount of safety in a large ship, and for this reason underwriters considered it a greater risk than the steam marine engine. These difficulties were overcome, however, in the year 1906 by the invention and introduction by Messrs Sulzer Brothers of Winterthur and Ludwigshafen-on-the-Rhine of the direct reversing engine which transformed the Diesel engine into an actual marine engine suitable for the very largest vessels in which the shafts remain coupled direct to the engine, as is invariably necessary in large boats. After an experience extending over three years with marine engines of the above mentioned firm, which had been supplied for a number of boats still in service, and in view of the fact that substantial improvements have been introduced into the details of construction, the advantages possessed by this engine are now easy to prove. Compared with a boat driven by steam, a saving of about one-third in the length of engine-room is effected, while the weight of the entire plant is about one-fourth that of a steam-engine plant of equal power, so that considerably more cargo can be carried with a corresponding



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In the Diesel engine all physical processes for converting the fuel into energy take place inside the working cylinder. Combustion of the liquid fuel, which is introduced by means of compressed air, takes place automatically in the hot air obtained by compression in the working cylinder. The combustion is gradual, there is no increase of pressure and, consequently, no explosions. The motor is started by means of compressed air, which is stored at a pressure of about 800 lbs., and the supply is sufficient for twenty starts without replenishing. Compressed air is admitted to the cylinders by simply turning a wheel, a further turn puts the starting valve out of gear, and operates the fuel valve, and the engine then begins to work upon the introduction of the fuel. In the same way the engine is brought to a standstill, restarted, and reversed. These engines require much less attention than a steam-engine, the supply of liquid fuel to the cylinders is automatic, and, therefore, one engineer is sufficient to look after the engine. One great advantage in favour of the Diesel motor as a marine engine and deserving of special mention is the fact that it is always ready for use, and being on the two-stroke principle, only the starting and fuel valves on each cylinder require reversing.

The following descriptions of the action of the Diesel type marine engine are taken from a paper by J. T. Milton, Esq. (Vice-President), read before the Institution of Naval Architects, 6th April 1911 :—

"It may be well to state here what is claimed for the Diesel engine in the way of consumption. In an ordinary steam-engine the power is generally reckoned as indicated horse-power. This is the work performed by the steam on the piston, and is the gross power obtained. It has to overcome the friction of the mechanism, work the slide-valves and the pumps, and only about 85 per cent. in round numbers is transmitted to the shaft.

"In the Diesel engine the indicated horse-power has similarly to overcome the friction of the mechanism, it has to work the fuel-pump, the mechanism for actuating the valves, and to supply the compressed air necessary for injecting the fuel. In the two-stroke cycle also it has to work the scavenging pump. These take up more of the gross power than do the accessories in a steam-engine, and hence a less proportion of the gross, or indicated power, is transmitted to the shaft than in a steam-engine. For this reason the power of a Diesel engine is more usually expressed as its brake horse-power—that is, the power usefully exerted outside itself.

"It is usually claimed that the oil consumption per brake horse-power per hour is 0·4 lb. when the engine is working at full power, and when working at somewhat lower powers the rate of consumption is not much increased.

"If one assumes that in a modern steam-engine the consumption of coal is 1·25 lbs. per indicated horse-power per hour this corresponds to about 1·47 lbs. per brake horse-power, so that the weight of fuel to be carried for the same voyage in a vessel fitted with Diesel engines would be only 28 per cent. of that of the coal necessary with ordinary steam-engines.

"We will now turn to the engine itself. Its principle of working is generally known. It is made in three forms for marine purposes—viz., as a four-stroke cycle single-acting engine, a two-stroke cycle single-acting engine, and a two-stroke cycle double-acting engine. An essential feature of these engines is that they require, besides their own cylinders and pistons, an auxiliary air-compressor capable of producing a pressure of about 700 lbs. per square inch.

"In the four-stroke cycle-engine the cylinder-cover contains a fuel-valve, a compressed-air admission-valve, one or more ordinary air-admission valves, and one or more exhaust-valves. All these valves are actuated—that is, opened—by means of cams fixed to a cam-shaft, and are kept closed by powerful springs when the cams are out of action. The cam-shaft is driven by a two-to-one gear—that is, it makes only one revolution for two revolutions of the engine crank-shaft. Broadly speaking, the cams are so arranged that the air-admission valves are open during one whole down-stroke, and the exhaust-valves during one whole up-stroke, but actually a little 'lead' is necessary. The cams for the fuel-valve and the compressed-air valve are so arranged that only one of these can be in operation at a time, so that when either is in use the other is entirely inoperative. In ordinary running, the fuel-valve is opened at the proper time when the piston is at the top of its travel, and is closed again when about one-tenth of the downward stroke has been made. The compressed-air valve is only used for starting purposes, and it is kept open for a longer period—say, for half or even more of the stroke—its range of opening being made to depend upon the number of cylinders used, so that when these valves are in gear there is no position of the engine in which there is not at least one of them open.

"In starting the engines these valves are put into gear, and the fuel-valves are consequently put out of action. When the engine has made one or more complete cycles, the compressed-air valves are put out of gear, the fuel-valves commence their work, and the engine then continues its motion, working under 'fuel' conditions. As the air-admission valve-gear is in full operation during the starting operations, the full compression would have to be overcome in each cylinder in turn, if it were not for a special arrangement made to relieve part of the pressure in order to facilitate starting. This is put out of action when the fuel admission is put into gear.

"Commencing with a piston at the top of the cylinder, the four-stroke cycle is as follows:—

"*First Down-Stroke*.—The ordinary air-admission valve is opened during the whole stroke, and the cylinder becomes filled with atmospheric air at the ordinary atmospheric pressure.

"*Second Stroke*.—The air-valve is closed, and the piston returns to the top of the cylinder, compressing the air which has been drawn in during the previous stroke. The clearance is so proportioned that in ordinary working at full speed the pressure becomes about 500 lbs. per square inch, and the temperature is, at the same time, very much raised. The compression is not quite adiabatic, as the cold cylinder walls must abstract a little of the heat from the air. If it were truly adiabatic the temperature of the air would be raised from, say, 60° Fahr. to 1000° Fahr.

"During this stroke a quantity of fuel has been pumped by the fuel-pump into an annular space round the fuel-valve. When the piston is at the top of the stroke, the fuel-valve is raised, and, at the same time, cold air from the air-compressor reservoir at a pressure of 700 lbs. per square inch blows the fuel into the cylinder, which contains hot air, at a pressure of 500 lbs. per square inch. The construction of the fuel-valve is such that the oil is pulverised or atomised—that is, it is divided up into a spray of very fine particles. These, upon coming into the very hot air in the cylinder, ignite, and the heat produced by the combustion increases the volume or the pressure of the air. When the adjustment of the valve is correct, the admission of the fuel and the combustion proceed at such a rate that they are almost completed during the time taken for the piston to travel one-tenth of its stroke, and during this period the pressure of 500 lbs. per square inch is maintained.

"*Third Stroke*.—The third stroke of the cycle commences with the combustion of the fuel as mentioned above, after which, during the remainder of the stroke, the hot gases in the cylinder expand until the end of the stroke is reached.

"*Fourth Stroke*.—The return of the piston constitutes the fourth stroke, and during this time the exhaust-valves are open, and the burnt gases are expelled from the cylinder. After this the cycle commences afresh.

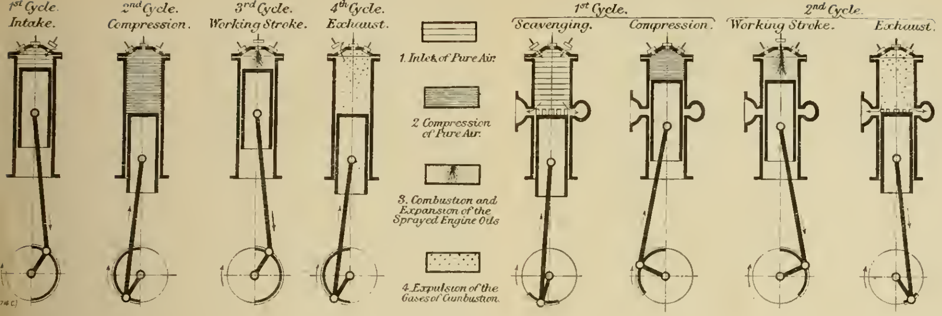
"In the two-stroke cycle single-acting engine, the cylinder covers are similarly fitted with fuel valves and compressed-air valves for starting purposes, but the ordinary air-inlet valves and exhaust valves are replaced by scavenge air-valves. All these valves are actuated by cams; the cam-shaft, however, in these engines rotates at the same speed as the main engine shaft.

"The pistons are made somewhat deeper than the total length of stroke. At the lower end of the part of the cylinder barrel uncovered by the movement of the piston, there are numerous ports leading into the exhaust passage. These ports have a vertical dimension of about one-seventh of the stroke.

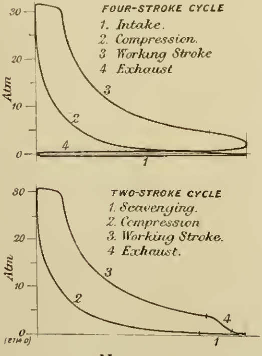
1st Cycle
Intake

WORKING DIAGRAMS OF SINGLE-ACTING DIESEL ENGINES.

FOUR-STROKE CYCLE. TWO-STROKE CYCLE.



INDICATOR DIAGRAMS OF SINGLE-ACTING DIESEL ENGINES.
(TAKEN FROM ORIGINAL DIAGRAMS.)



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"The two-stroke cycle is as follows:—

"*First Stroke.*—When the piston is at the bottom of the stroke the cylinder is full of pure air at atmospheric pressure, which air has just been admitted through the scavenge-valve. During the up-stroke the air is compressed up to 500 lbs. pressure per square inch, precisely as in the compression stroke of the four-stroke cycle engine.

"*Second Stroke.*—The second stroke commences at the top centre by the admission of fuel sprayed in by compressed air precisely as in the previously described engine, and the stroke proceeds in exactly the same way until the piston has travelled about six-sevenths of the stroke. At this point it commences to uncover the exhaust ports through the cylinder sides. So much of the hot gases escape through these ports as to reduce the pressure in the cylinder to about that of the atmosphere. The scavenge-valves are then opened, and fresh air under pressure is admitted into the cylinder, blowing out the remainder of the burnt gases into the exhaust passages. With these gases some of the scavenge air also passes into the exhaust. By the time the piston has reached the bottom of its stroke the scavenge-valves are closed, but the cylinder is left full of clean air ready for the compression stroke to commence.

"There are different arrangements made by different makers for supplying the scavenge air. In some designs each main cylinder has its own air-compressing arrangement and receiver. In other designs sometimes one, and sometimes two, air-pumps are provided, sometimes worked by cranks from the crank-shaft, and in some cases by levers similar to the method of working air-pumps in ordinary marine steam-engines.

"There are also different methods of applying the scavenge-air, one allowing it to enter through special valves in the cylinder cover, another by admitting it at one side of the bottom of the cylinder through ports uncovered by the piston in the same way as the exhaust ports are uncovered. In this latter case the scavenge ports are on one side and the exhaust ports on the other.

"The volume of the scavenge air-pumps is considerably greater than that of the cylinders, the proportion being in general not less than 1·8. This is necessary to ensure that all the burnt gases will be swept out of the cylinder. As the full quantity of air dealt with by the pump must pass from the reservoir into the cylinders every revolution, the pressure to which the air in the reservoir attains depends upon the scavenge-valve openings. When these are large a less pressure is required to force the air through them than when they are small. Hence the larger the openings of these valves the less load there is thrown on the scavenge air-pump.

"In some designs the scavenge air-pressure is as much as 7 lbs. to 8 lbs. per square inch, in others it is as low as 3 lbs. to 4 lbs. above the atmosphere.

"In the preceding engines the pistons are of the trunk form. In the double-acting two-stroke cycle engine they are necessarily made

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"In the preceding engines the pistons are of the trunk form. In the double-acting two-stroke cycle engine they are necessarily made of box form, and are fitted with piston-rods, which, as they pass through the burning gases at the lower side of piston, have to be specially cooled. The pistons also, in general, have to be cooled by either oil or water circulation.

"Water is the best cooling medium, as its specific heat is about three or four times that of oil, but some prefer oil, as any leakage from the water circulation washes off the lubrication of any of the rubbing parts which it touches.

"The admission, fuel, &c., valves, are designed for both the top and bottom of the cylinder. The exhaust ports are in the middle of the length of the cylinders, and the pistons, as in the single-acting engines, uncover the ports at nearly the end of the stroke.

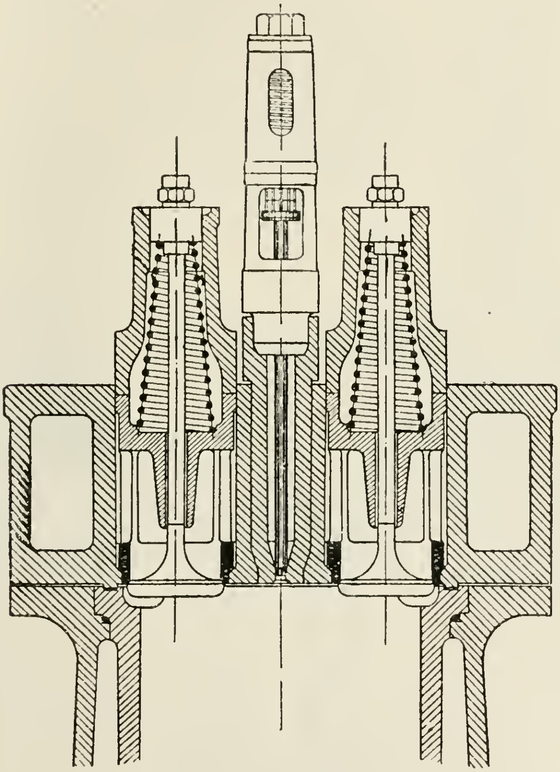
"In small engines of the single-acting type the pistons are not water or oil cooled, as it is found they may be kept at a sufficiently low temperature by their contact with the cylinder sides, which are water-cooled. A heated piston is not so objectionable as it is in an ordinary gas or oil engine with timed ignition, because, in the Diesel engine, pre-ignition cannot occur. The main objections are that, with a large piston, overheating of the crown may be the cause of structural weakness, and that the expansion of the metal by its high temperature renders it necessary to make the piston crown initially smaller than the cylinder bore. This has to be arranged in all engines, and the exact amount of allowance is one of those points in which experience is the only guide. It may be said here that this is a matter of extreme importance in those engines in which there are no piston-rods. This will be again referred to further on.

"Experiments are being made in the case of an engine with a large diameter of cylinder, to ascertain whether it is practicable to run it without special piston cooling.

"In all the types of engines highly-compressed air is needed for starting purposes, and also for the fuel injection. This has to be supplied by an air-compressing plant worked by the main engine. The compression is sometimes performed in two stages, although a three-stage arrangement is generally used. The compressed air is cooled at each stage. The volume of the compressor is such as to provide a small surplus each revolution over that required for the fuel admission in continuous working. This surplus is stored in a reservoir constructed usually as a battery of seamless steel bottles. These are tested by hydraulic pressure to 120 atmospheres. A safety valve is provided loaded to 60 atmospheres. The compression of the air is attended by the deposition of moisture from it, so that means of draining the steel bottles should be provided.

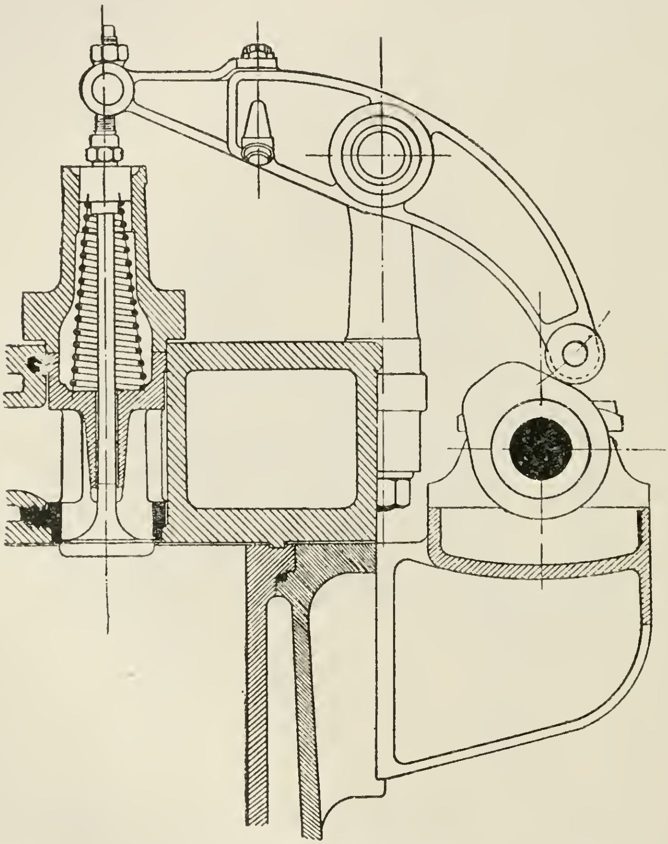
"Naturally there are advantages and disadvantages with each type of engine, and a judicious consideration of these should determine which is the more suitable type to use in any particular case. The two-stroke double-acting engine will be higher than a single-acting one with the same diameters of cylinders and stroke, but the power will be obtained with a less number of cylinders. On the other hand, there is considerably more complexity in the valve arrangements, and a probability of difficulty with piston-rod stuffing-boxes, to say nothing of the trouble of cooling the pistons and rods by water or oil circulation. There is also likely to be considerable difficulty, owing to want of access for overhauling. It should be stated that no experience has been had, as yet, with large engines of this type.

"Comparing single-acting engines of the two-stroke and four stroke types, the former require only half the number of cylinders



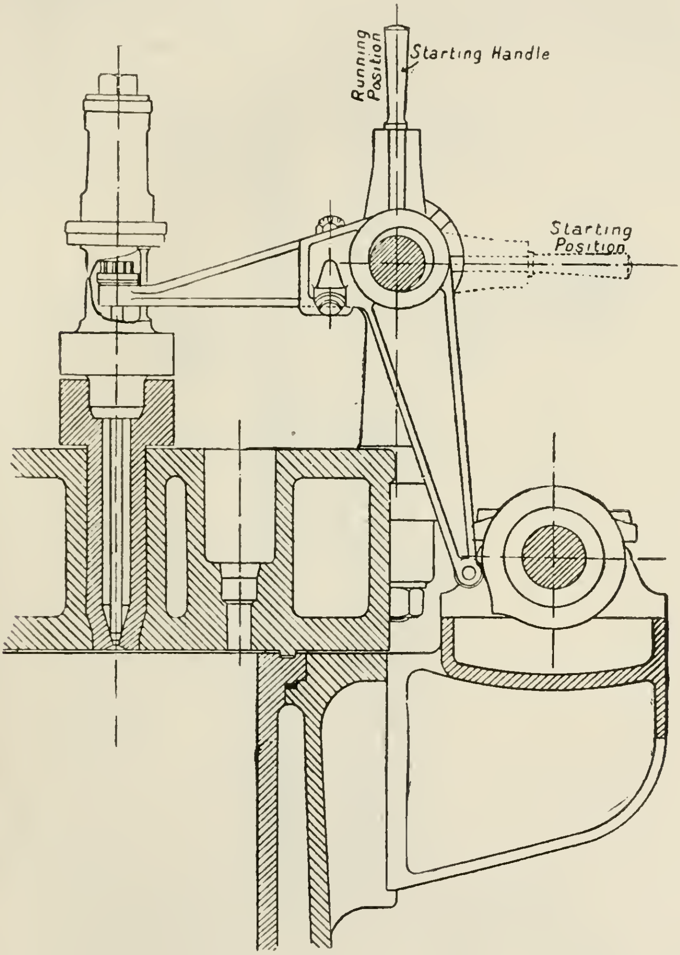
No. 23.—Air Inlet Valve and Exhaust Valve.
(Fuel Valve in Centre.)

which are requisite in the latter, either to produce the same power or the same degree of uniformity of turning moment. The four-stroke therefore means a longer engine, and necessarily a heavier one also. The valve-gear of the two-stroke engine, being actuated by a shaft with the same rotational speed as the crank-shaft, is simpler than that in a four-stroke engine, and the reversing arrangements are much less complicated. The two-stroke, however, requires the addition of the scavenge arrangements which are absent from the four-stroke, and the necessity for supplying the energy for working these makes the mechanical efficiency less. On the question of efficiency, however, it may be urged that the four-stroke engine has to overcome the friction of the piston, &c., for what may be called two idle strokes out of every four, and this must, to some extent, counterbalance the energy necessary to work the scavenging pumps. In the four-stroke engine all the hot used gases have to escape past the exhaust-valves, which thus may



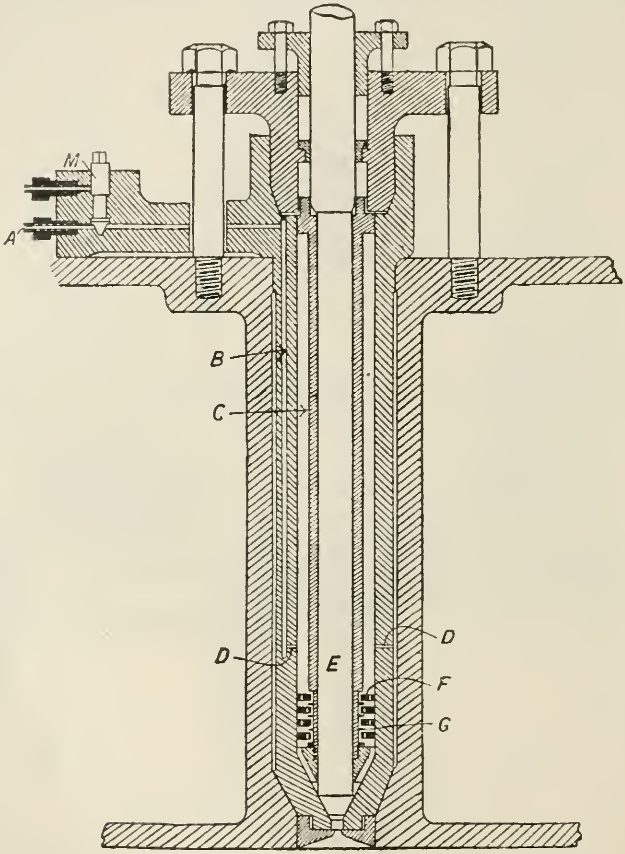
No. 24.—Exhaust Valve, Lever, and Cam.

become abnormally heated. On the other hand, in the two-stroke engine they have to pass the bars between the exhaust-ports, and it is thought by some that although these parts of the cylinder are water-jacketed, they must become over-heated and lose their accuracy of surface, and it must be remembered that all the piston-rings have to pass these bars every stroke. Extended experience will be required to settle all these points. It may be mentioned that an engine is being made on the four-stroke system in which the major portion of the exhaust passes out of the cylinder through ports precisely as in the case of the two-stroke engine, leaving only a part of the burnt gases to be pushed out of the cylinder through the ordinary exhaust-valves.



No. 25.—Fuel Inlet Valve, Lever, and Cam.

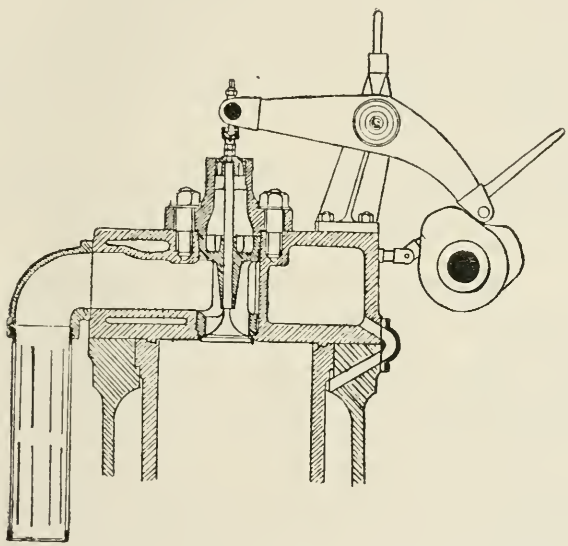
Fuel Valve.—The oil from the fuel pump enters through the pipe A, the amount being regulated by the pump governor to suit the engine load. The oil flows down the cylindrical space B, and enters the receiver C through the small opening D near the bottom, and just above the spray nozzle. The sprayer consists of four metal rings F, each containing over twenty small holes, which are arranged in "staggered" form, the holes being about $\frac{1}{12}$ inch diameter. Below the rings is a conical shaped fitting supplied with narrow channels (about $\frac{1}{16}$ inch deep), which form a series of nozzles through which the oil travels after passing the holes in the rings. The oil then enters the cylinder head by the expanding shaped opening below the needle valve. The annular space C is in direct communication with the air-blast pressure, so that whenever the needle



No. 26.—Fuel Inlet Valve and Pulveriser.

valve lifts, the air pressure forces the oil accumulated in the space C into the cylinder in the form of a spray. Ignition by the hot air follows, and the resultant expansion of the gases generate force down the piston, and constitute the working or power stroke of the cycle.

Fuel Inlet Valve and Cam.—As will be seen from the illustration, when the nose of the cam comes in contact with the valve lever the latter is forced outwards, and the valve is opened against the pressure of a spring which normally keeps the valve tight in its seat. The starting handle when horizontal causes the starting valve lever to come in contact with the nose of its cam as the cam shaft rotates, while the fuel-valve lever is held clear of its cam at the same time. With the starting handle in the vertical position the starting valve lever is clear of its cam while the fuel-valve lever then comes into operation. The actual opening allowed the fuel valve is very small, as the limited opening area increases the spraying action of the oil injection. The air admission valve, exhaust valve, and starting valve all open downwards or into the cylinder, but the fuel valve opens upwards or out of the cylinder, as shown by the various sketches.



No. 27.—Air Inlet Valve, Lever, and Cam.

In the **four-cycle** type the cylinder cover contains four valves, thus :—

1. Air Inlet Valve.
2. Fuel Valve.
3. Exhaust Valve.
4. Starting Valve (air blast).

In the **two-cycle** type "scavenge valves," one or two in number, take the place of the air inlet valve and act similarly, with the difference that the air is in this case under pressure when admitted to the cylinder, and is intended to "scavenge" the cylinder of the exhaust gases before compression of the pure air.

DIESEL ENGINE NOTES AND SKETCHES.

Action of 4-Stroke Diesel Engine.

1. **Down-Stroke.**—Atmospheric air at engine-room temperature is drawn into the cylinder through the suction silencer. Air admission valve open.

2. **Up-Stroke.**—The air is compressed to about 500 lbs. pressure and somewhere about 1200° Temp. Fahr. All valves shut.

3. **Down-Stroke.**—Just about the top centre the fuel valve is opened by the cam and lever gear, and crude oil is blown in through a sprayer on the end of this needle valve by means of the injection air blast of about 750 lbs. pressure. The oil then ignites, combustion takes place, and the expansion of the gases formed force down the piston. The fuel valve remains open for about $\frac{1}{10}$ of the stroke. This is the *power or impulse stroke* of the cycle.

4. **Up-Stroke.**—The dead gases of combustion are forced out of the cylinder through a water-cooled silencer to the atmosphere. Exhaust valve open.

Action of 2-Stroke Diesel Engine.

1. **Down-Stroke.**—(A) With the cylinder full of compressed air at about 500 lbs. and 1200° Temp. (as in the case of the second stroke of the 4-cycle type), the oil is blown in as before described, ignites, and combustion takes place, the expansion of the gases driving the piston down.

   (B) At about $\frac{7}{8}$-stroke the piston uncovers the exhaust belt port openings, and the dead gases begin to pass away to the atmosphere.

   (C) At about $\frac{9}{10}$-stroke the two small scavenge valves on the cylinder cover are opened by their cam levers, and low pressure air of about 6 lbs. pressure is blown into the cylinder, clearing out the remainder of the dead gases, and at the same time filling up the cylinder with pure air in readiness for the next compression stroke.

2. **Up-Stroke.**—The air admitted by the scavenge valves is compressed to about 500 lbs. and 1200° Temp.

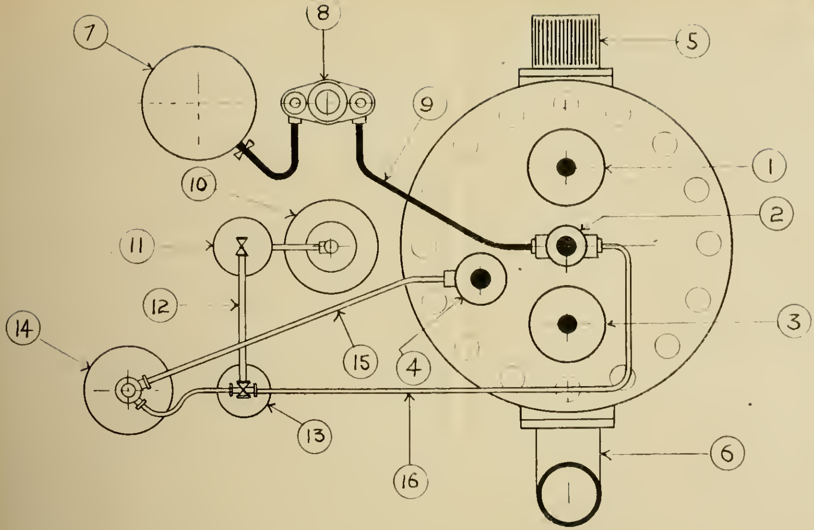
It will be seen that in the 2-stroke cycle, three operations are effected on the down-stroke, which in the 4-stroke type require three separate strokes to perform. Again, in the 4-stroke cycle, the atmospheric air is merely drawn into the cylinder, whereas in the 2-stroke cycle the air is *forced* in under pressure, the process only occupying a fraction of the stroke travel.

14

14

(1) Low pr
(2) Fuel va
3, 3) Scaven;
(4) Safety '
(5) Starting
(6) Exhaust from exhaust belt to silencer
(7) Oil fuel tank.
(8) Fuel pump.
(9) Fuel oil pipe to fuel valve.

(14) Lower pressure air bottle for starting purposes (500 lbs. ☐).
(15) Starting air pipe to valve.
(16) High pressure blast air pipe to fuel valve.



No. 28.—Plan of 4-Cycle Diesel Oil Engine Cylinder.

1. Air inlet valve.
2. Fuel valve.
3. Exhaust valve.
4. Starting valve.
5. Suction air silencer.
6. Exhaust pipe to silencer.
7. Fuel oil tank.
8. Fuel oil pump.
9. Fuel oil pipe to fuel valve.
10. 3-Stage air compressor.
11. After-cooler.
12. High pressure air delivery to high pressure bottle.
13. High pressure air bottle for fuel injection purposes (800 lbs. □″ or more).
14. Lower pressure air bottle for starting purposes (500 lbs. □″).
15. Starting air pipe to valve.
16. High pressure blast air pipe to fuel valve.

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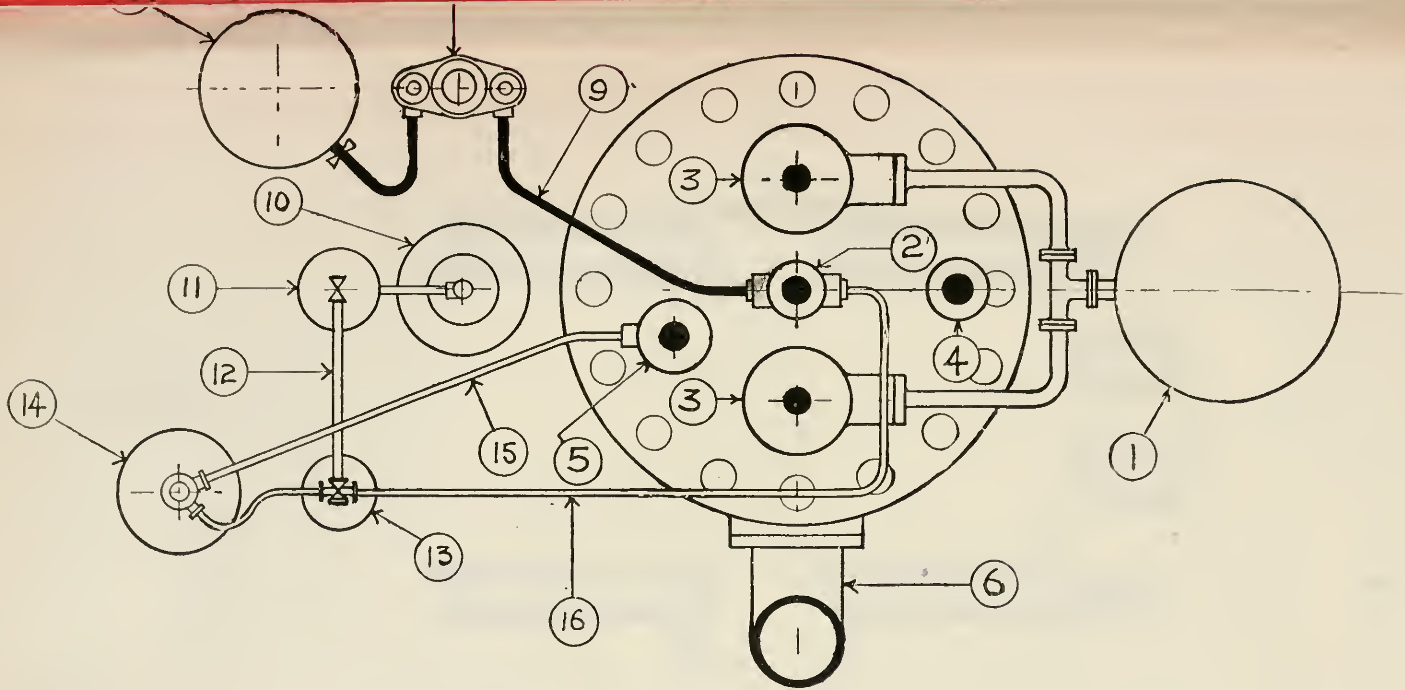
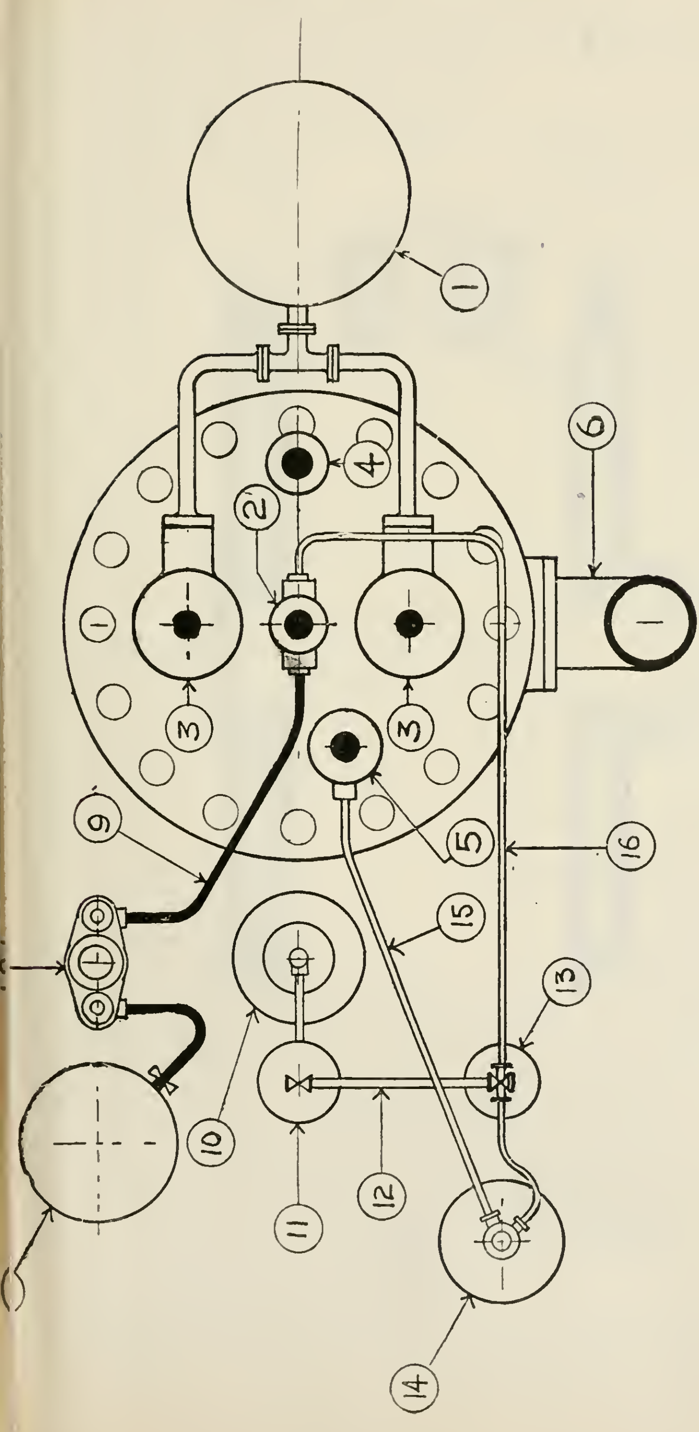


No. 29.—Plan of Two-Cycle Diesel Oil Engine Cylinder.

(1) Low pressure air pump for scavenge air (about 6 lbs. □″).
(2) Fuel valve.
3, 3) Scavenge air valves.
(4) Safety valves (loaded to about 1200 lbs. □″).
(5) Starting valve.
(6) Exhaust from exhaust belt to silencer
(7) Oil fuel tank.
(8) Fuel pump.
(9) Fuel oil pipe to fuel valve.

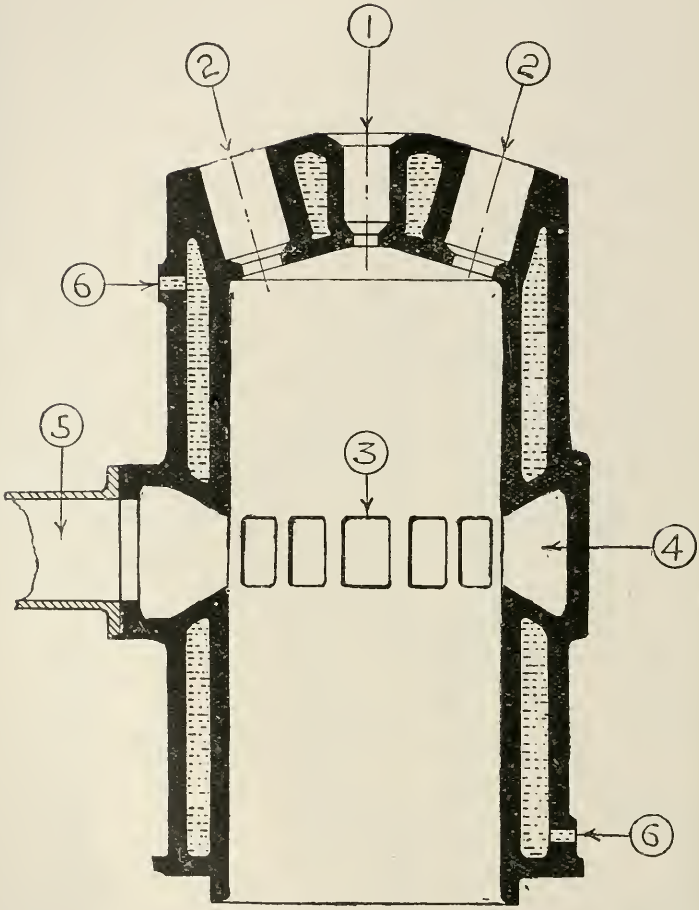
(10) Three-stage air compressor.
(11) After cooler.
(12) High pressure air delivery to high pressure bottle.
(13) High pressure air bottle for fuel injection purposes (800 lbs. □″ or more).
(14) Lower pressure air bottle for starting purposes (500 lbs. □″).
(15) Starting air pipe to valve.
(16) High pressure blast air pipe to fuel valve.

a fraction of the stroke travel.



No. 29.—Plan of Two-Cycle Diesel Oil Engine Cylinder.

(1) Low pressure air pump for scavenge air (about 6 lbs. □″).
(2) Fuel valve.
(3, 3) Scavenge air valves.
(4) Safety valves (loaded to about 1200 lbs. □″).
(5) Starting valve.
(6) Exhaust from exhaust belt to silencer
(7) Oil fuel tank.
(8) Fuel pump.
(9) Fuel oil pipe to fuel valve.
(10) Three-stage air compressor.
(11) After cooler.
(12) High pressure air delivery to high pressure bottle.
(13) High pressure air bottle for fuel injection purposes (800 lbs. □″ or more).
(14) Lower pressure air bottle for starting purposes (500 lbs. □″).
(15) Starting air pipe to valve.
(16) High pressure blast air pipe to fuel valve.



No. 30.—Section through Cylinder of 2-Cycle Diesel Engine.

1, Fuel valve seat (water jacketed).
2, Scavenge valve seats (two).
3, Exhaust ports leading to exhaust belt.
4, Exhaust belt.
5, Exhaust to silencer.
6, Water circulation of jacket.

Cooling Circulation.—The cylinders, valves in the cover, and the compressors are all water jacketed. The pistons are usually cast hollow, and either oil or water cooling is provided for by means of telescope piping (see sketch No. 45).

Lubrication.—Forced lubrication is employed, a pressure of about 10 lbs. at the pump giving 4 or 5 lbs. in the bearings.

Starting.—Having first pumped up the fuel valve by hand and opened up the injection blast air bottle connection to the cylinder, starting is effected by placing the starting lever or wheel in position for the starting valve cam, the fuel valve lever being then out of acting position. The valve on the starting air bottle is next opened and the engine begins to move round under compressed air conditions, the pressure being somewhere about 450 lbs. or higher. After a few revolutions, the starting wheel is moved round so that the starting air valve cam, as shown by the index pointer, is cut out and the fuel valve cam slipped into gear position for fuel admission ; the engine then continues running on fuel which should be admitted by degrees in quantity to avoid rapid heating up of the cylinder.

Reversing.—The cam shaft is usually supplied with a double set of cams for each fuel, air, starting, and exhaust valve, one set for ahead running, and the other set for astern running. After stopping, the valve levers resting on the cams are all lifted clear (either by hand or by suitable gear) and the shaft is moved along laterally until the astern cams are in line with the levers, which are then lowered again into working contact position.

Air is next admitted to the starting valve, and the engine begins to revolve on compressed air exactly as in starting ; the air is then cut out and the fuel admitted.

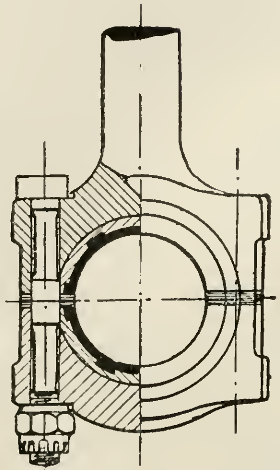
The operation of stopping and reversing may be described as follows :—

1. Stop engines by shutting off fuel supply by hand control lever. 2. Lift valve levers off cams. 3. Throw out ahead cams and put in astern cams by moving cam shaft laterally. 4. Give starting air. 5. Cut out starting air and give fuel gradually to avoid rapid heating up.	These movements are, in most cases, carried out automatically by the reversing wheel gear.

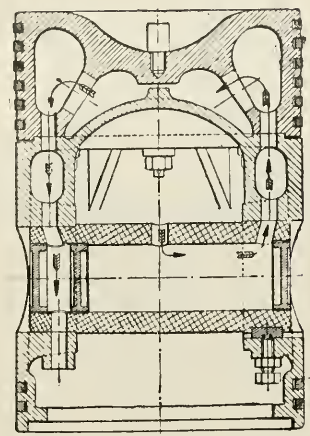
Power Control.—If required, certain of the cylinders can be run on air and the remainder on fuel, and in some cases the gear is also arranged to allow of the cutting out altogether of some of the cylinders, which admits of running under reduced power conditions.

Tests.—Pet cocks are fitted on the fuel valve, fuel pump, and other connections to allow of testing previous to starting up, for pressure and atomisation of the oil spray, etc., all of which tests should always be most carefully carried out beforehand.

The air blast employed should always be kept as low as possible, consistent with the power required. The correctness of the firing in each cylinder can be tested by opening the indicator cock.

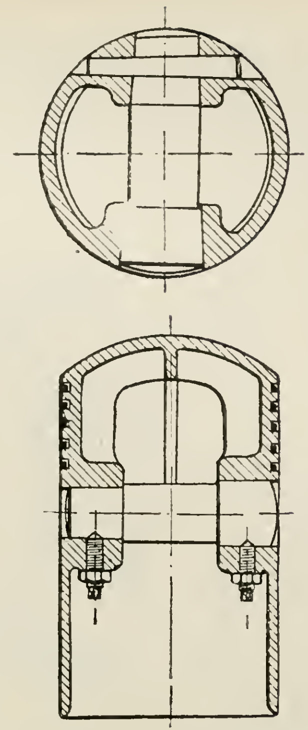


No. 31.*—Bottom End of Diesel Engine Connecting Rod.



No. 32.*—Diesel Engine Piston.

This type of piston is cast hollow to allow of oil or water cooling. Notice that double sets of Ramsbottom rings are provided, 6 rings above, and 2 rings below the cross pin. The arrows show the circulation flow.



No. 33.*—Diesel Engine Piston.

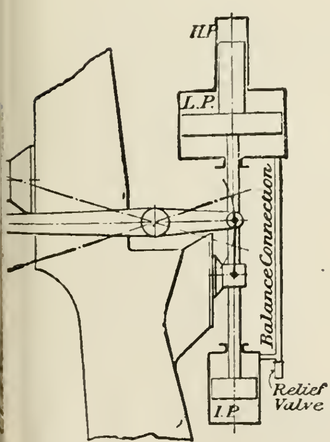
This type of piston is fitted for small sizes and powers.

Wear of Valves.—The exhaust valves usually show signs of wear first, as the intense heat of the gases produce pitting on the faces. As leaky valves mean loss of compression, and therefore reduced power, these valves require regular overhaul and attention. The fuel valve also tends to wear away at the sprayer end, and therefore requires to be examined and ground in at regular intervals, as any difference in setting due to pitting, etc., may seriously upset the timing of the oil injection, and throw the cylinder out of power balance with the other cylinders of the engine. Premature ignition may also take place. If dirt enters with the fuel oil, choking up of the pulveriser may also happen, resulting in falling off of power.

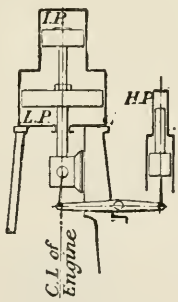
Fuel Valve Lift.—The lift of this valve seldom exceeds $\frac{1}{16}$ in., and the period that the valve remains open is usually about $\frac{1}{10}$ of the down stroke. Assuming the revolutions to be, say, 120 p.m., then in one minute the strokes $= 120 \times 2 = 240$. So that time per stroke $= \frac{60}{240} = \frac{1}{4}$ sec. Time fuel valve remains open $= \frac{1}{4} \times \frac{1}{10} = \frac{1}{40}$ sec.

Smoke.—Heavy smoke and deposit of carbon is due to one of the following causes :—

1. Weak compression.
2. Fuel oil not atomising properly.
3. Excessive fuel supply.
4. Excessive piston lubrication.



No. 34.



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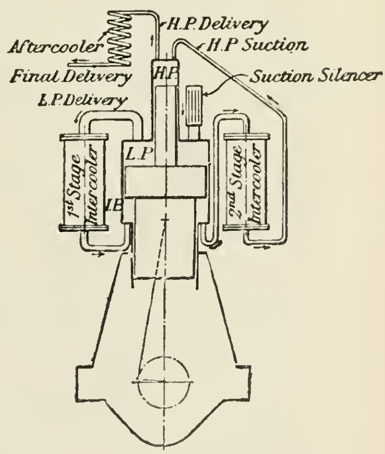

No. 36.

Compressor Arrangements.*

No. 34.—3-stage compressor driven from crosshead by levers.

No. 35.—3-stage compressor, L.P. and I.P., driven from one rod, and H.P. driven by lever from I.P. and L.P. The whole worked by means of links and crank from main shaft.

No. 36.—3-stage compressor driven by crank on main shaft.

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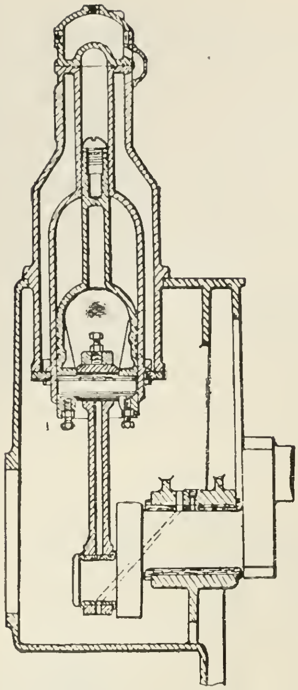
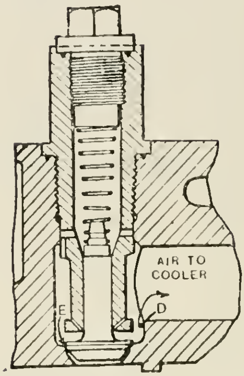


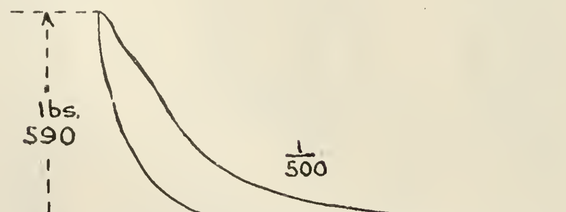
No. 37.*—Two-Stage Compressor.

Driven from shaft by crank and fitted with water circulation jacket.



No. 38.*—Compressor Delivery Valve.

After compression the hot air passes to an intercooler which is provided with water circulation, and is reduced in temperature before entering on the next stage of compression.



No. 39.—2-Stroke Diesel Engine Diagram.

M.E.P.=95·6 lbs. per square inch.

Notice that in this case the pressure of compression is 590 lbs. per square inch, also that no further increase of pressure takes place after ignition.

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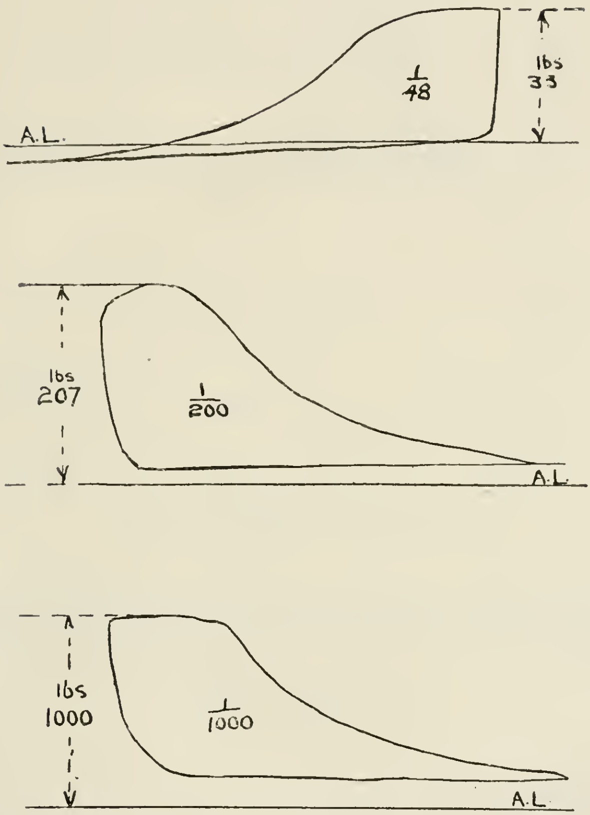
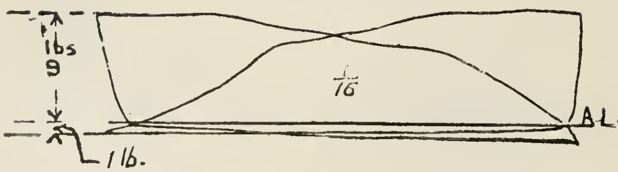
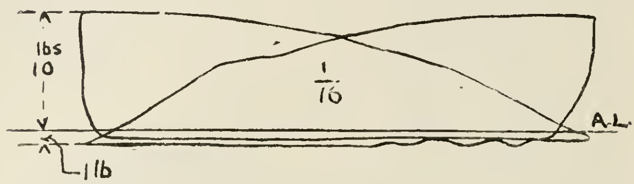


No. 40.—Diagrams from 3-Stage Compressor.
(With pressures.)

L.P. compressor,	M.E.P.	= 13·2 lbs. per square inch.
,,	I.H.P.	= 6·8.
I.P. compressor,	M.E.P.	= 80 lbs. ,, ,,
,,	I.H.P.	= 10·2.
H.P. compressor,	M.E.P.	= 345 lbs. ,, ,,
,,	I.H.P.	= 9·6.





No. 41.—Diagrams from Scavenge Pump Cylinders; (2) of 2-Stroke Diesel Engine.

Observe that the scavenge pump suction pressure is 1 lb. below atmosphere, and the delivery pressure 9 or 10 lbs. gauge, which falls to about 6 or 7 lbs. on entering the cylinder through the scavenge valves.

Average M.E.P. = 6·8 lbs.
,, I.H.P = 20·5.

Consumption of Fuel.—The consumption of fuel per B.H.P. per hour ranges from ·4 to ·5 of a lb.

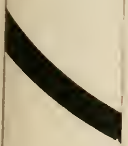
Heat Efficiency.—Taking the oil consumption per B.H.P. as, say, ·45 of a lb. per hour, and the heat value of 19000 B.T.U., then the efficiency works out as under :—

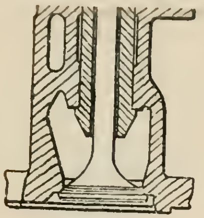
B.H.P. = ·85 of I.H.P. (average).
Then, $\cdot 85 \times \cdot 45 = \cdot 382$ lbs. oil per I.H.P. per hour.

$$\text{Efficiency (on I.H.P. basis)} = \frac{33000 \times 60}{\cdot 382 \times 19000 \times 778} = \cdot 347, \text{ or } 34\cdot 7 \text{ per cent.}$$

The efficiency sometimes reaches the high value of 36 per cent., and even more, in cases of land installations.

The compressors usually absorb about 10 per cent. of the power developed.





No. 44—Exhaust Valve.

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2. Fuel
3. Air a
by
4. Admi
5. Air b
incl

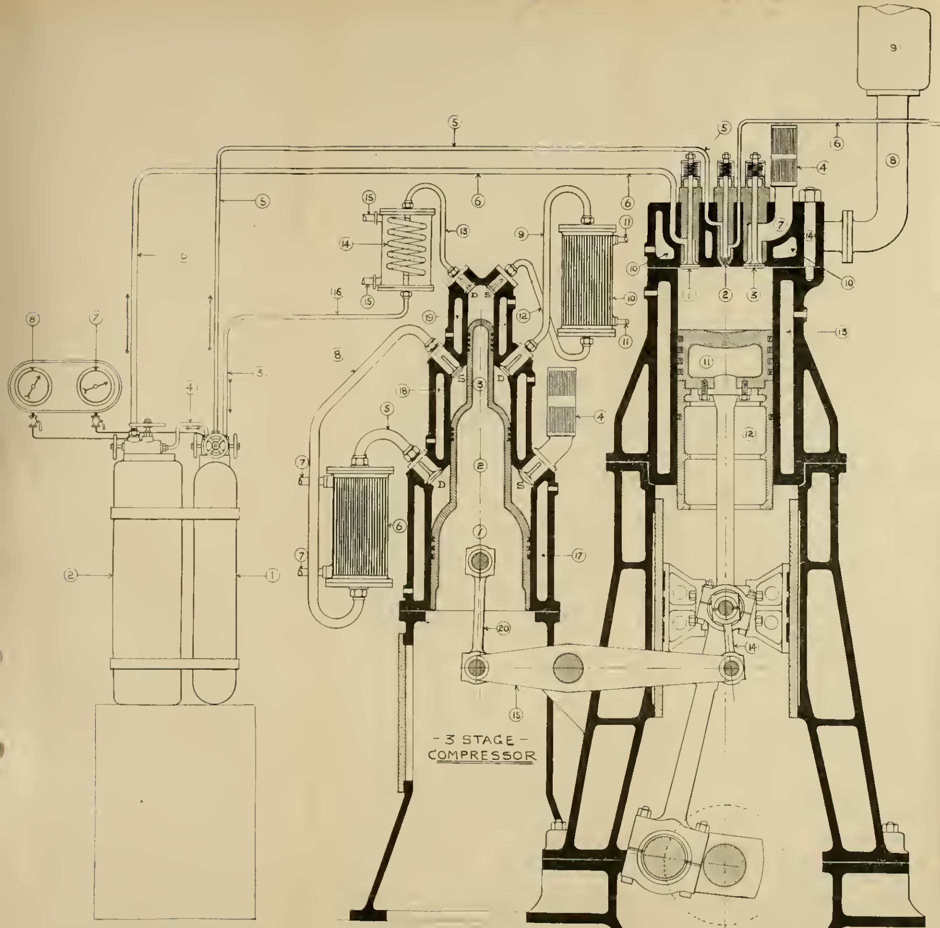
Fuel Valve.

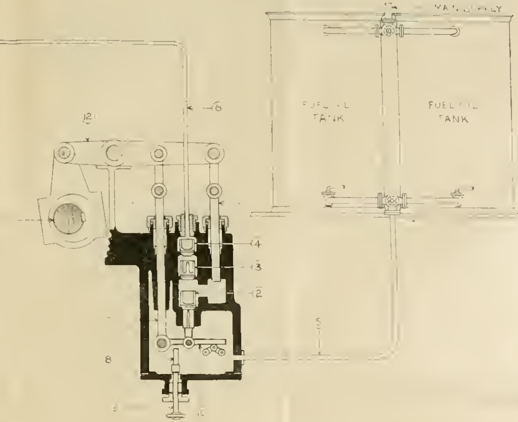
6 (left
pre let from fuel pump, $\frac{3}{16}''$ bore (850 to 1200 lbs. pressure).
6 (right nlet from high pressure air bottle. $\frac{3}{8}''$ bore.
fror
7. Air a assage of valve.
8. Exha assage of valve.
9. Exha
10. Wate ve.
11. Upper
12. Lowe for actuating the valve.
13. Water on.
14. Crossl
15. Lever

of 4-S

[*To face page* 632.

requ
of each piston a ring is pinned on to wipe the lubricating oil off the cylinder walls. This oil is collected in a circular tray to prevent it from mixing with the engine lubricating oil, and it is claimed that when filtered 60 per cent. of the original amount supplied may be recovered for use over again.

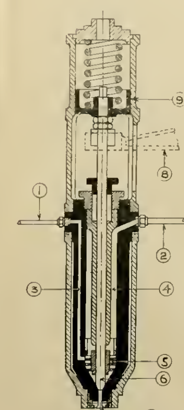




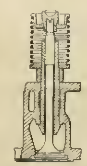
Index for Fuel Oil Pump Only.

Note.—This pump is often driven by an eccentric fixed to the cam or half time shaft, as shown in [illegible] For the sake of clearness of explanation the pump is here shown detached from the engine altogether [illegible] will be seen, the scale to which the pump is drawn is much larger than that of the other parts

1 Fuel pump plunger
2 Suction valve
3 Lower delivery valve } Two delivery valves are fitted to secure high pressure discharge of the oil
4 Upper delivery valve }
5 Oil supply to pump suction from main tanks
6 Oil delivery pipe leading to fuel valves (pressure of oil may be anything between 900 to 1200 lbs per square inch, as the oil pressure must be sufficient to overcome the resistance of the air blast employed for injection purposes, and thus ranges from 800 to 1000 lbs per square inch)
7 Governor connection to hook on bottom of suction valve which is held open when the speed is to be reduced thus controlling the quantity passing through the delivery valves to the cylinder
8 Hand gear rod for direct speed or power control applied as described for governor connection
9 connection for direct power control
10 Hook or lever which, when lifted (either by hand or governor), holds up suction valve and thus reduces fuel supply and power developed
11 Cam shaft worked by gear wheels from main shaft, and to which the driving eccentric is keyed
12 Lever from eccentric to fuel pump plunger



No. 43. Fuel Valve.



No 44 Exhaust Valve

Index for Air Bottles Only.

1 High pressure air supply bottle (pressure from 800 to 1000 lbs per square inch)
2 Lower pressure air supply bottle for starting or reversing purposes (pressure about 500 lbs per square inch)
3 Air delivery pipe from 3rd stage compressor to bottle
4 Connection between high pressure and lower pressure bottles to which suitable regulating valves are fitted.
5 Air blast pipe leading to fuel oil valve for spray injection at end of compression stroke.
6 Starting (or reversing) air blast pipe leading to starting air valve
7, 8 Pressure gauges for bottles

Index for 3-Stage Compressor Only.

1 L P stage (pressure about 45 lbs per square inch)
2 I P „ „ 240 „
3 H P „ „ 900 „ (Temperature 110
4 Admission air silencer Air enters compressor at atmospheric pressure
5 Air delivery to 1st intercooler
6 Intercooler with water circulation over the tubes, the air being inside the tubes
7 Water circulation.
8 Delivery from intercooler to suction valves of 2nd stage compressor
9 Delivery from 2nd stage compressor to 2nd intercooler
10 Intercooler with water circulation
11 Water circulation
12 Delivery from 2nd intercooler to suction valves of 3rd stage compressor
13 Delivery from 3rd stage compressor (pressure, 850; temperature about 180) to cooling coil of aftercooler
14. Aftercooler with water circulation
15 Water circulation of aftercooler
16 Delivery of H P air to storage bottle (Temperature 110°)
17 Water jacket of 1st stage.
18 „ „ 2nd „
19 „ „ 3rd
20 Driving link of compressor

Index for Engine Parts Only.

1 Air starting (or reversing) valve (blast pressure about 500 lbs per square inch).
2 Fuel oil injection valve.
3 Air admission valve Air enters cylinder at atmospheric pressure by piston suction on down stroke
4 Admission air silencer.
5 Air blast for oil injection (pressure 800 to 1000 lbs per square inch
6 (left of cylinder) Starting air from bottle at about 500 lbs. pressure per square inch
6 (right of cylinder) Oil supply from fuel pump (pressure of oil from about 800 to 1000 lbs per square inch)
7 Air admission to cylinder, first down stroke of cycle
8 Exhaust pipe from exhaust valve on opposite side of cylinder
9 Exhaust silencer (usually water cooled) leading to atmosphere
10 Water jacket round cylinder head
11 Upper hollow cast piston for cooling
12 Lower piston, trunk type and open below
13 Water jacket round cylinder walls
14 Crosshead link connecting to compressor lever
15 Lever for driving the compressor

Fuel Valve.

1. Fuel oil inlet from fuel pump, $\frac{1}{8}$" bore (850 to 1200 lbs pressure [illegible]).
2. Blast air inlet from high pressure air bottle, $\frac{1}{4}$" bore
3. Fuel oil passage of valve
4 Blast air passage of valve
5 Pulveriser
6 Needle valve
7. Sprayer
8 Cam lever for actuating the valve
9 Guide piston

No. 42.—Working Diagram of 4-Stroke Marine Type Diesel Oil Engine.

To face page 632.

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the usual shallow type, with nine rings, and is supported in the centre on the dished-out piston rod. The pistons are cooled by salt water, supplied through telescoping pipes kept tight by metallic packing, as with pistons of this large size the subsequent cooling of the oil required by oil cooling of the pistons is too difficult. On the bottom of each piston a ring is pinned on to wipe the lubricating oil off the cylinder walls. This oil is collected in a circular tray to prevent it from mixing with the engine lubricating oil, and it is claimed that when filtered 60 per cent. of the original amount supplied may be recovered for use over again.

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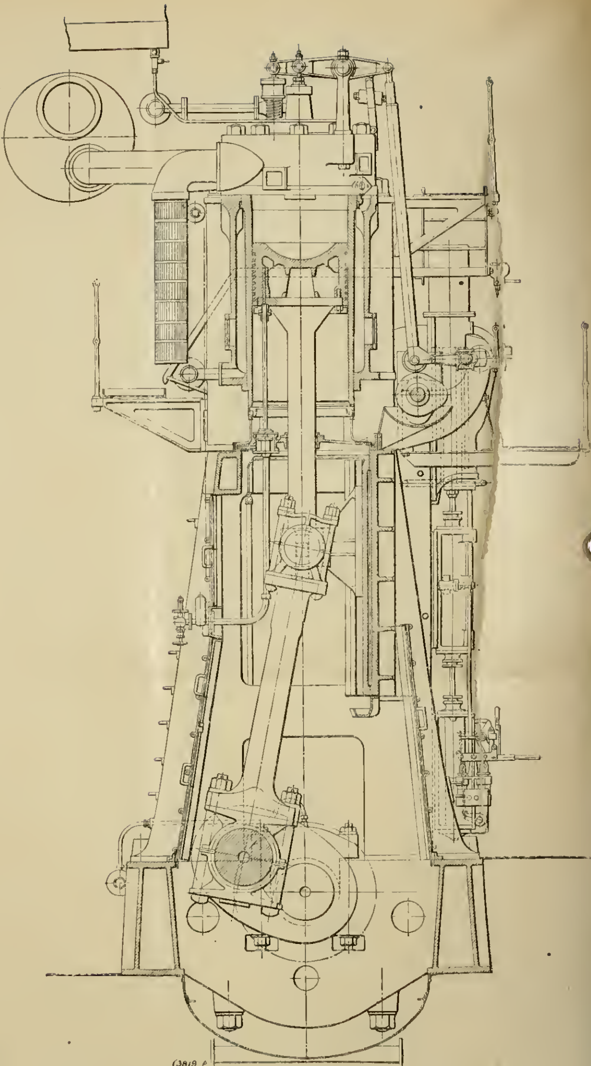

No. 45.*—Four-Cycle Diesel Marine Oil Engine.

Motor-Ship "Fionia."

Constructed by Messrs Burmeister and Wain, Copenhagen

* Reprinted from *Engineering.*

To face page 633.

THE MOTOR SHIP "FIONIA."

General Description of Engines.

The six cylinders are arranged in two sets of three, and the cast-iron bed-plate is in two interchangeable pieces, with deep box-girders unsupported in the centre, an arrangement justified owing to the through bolt system of construction adopted. The sump for the lubricating oil is formed by welded-steel plates. The design of the crankcase and the means adopted for carrying the piston load are departures from earlier engines, and give greater accessibility to the working parts. Between the cylinders, and at the ends of each group of three cylinders, there is an A frame formed by box columns at the front and at the back, to the tops of which the cylinder feet are bolted. At the front of the engine, between these columns, the crosshead guides are bolted, and for stiffening purposes box cross-girders connect the tops of the columns at the back of the engine. By this means the back of the engine is enclosed by sheet-iron oil-tight doors, and at the front sheet-iron oil-tight doors encase the crank-chamber beneath the guides. This gives excellent accessibility for the adjustment and examination of the top end, bottom end, and main bearings. The guides are arranged in the front of the engine, so that the working stroke-guide thrust may be taken in the largest guide area. Forced lubrication is adopted to all the main bearings.

The crank-shaft is built up in two interchangeable pieces, of three throws, and is drilled out for the forced lubrication system. The connecting rod is of the usual marine design, and the absence of keeps for both the top and bottom ends will be noted. The only tension load to be taken is that due to inertia, which, with a normal speed of revolution of 100, and the light piston made possible with engines of the four-cycle type, must be very small.

The cylinder jackets for each set of three cylinders are cast in one piece provided with feet, to which reference has already been made, for bolting to the A frames. Between the cylinders, and at the ends of each group, the cylinder jacket casting is swelled out to carry through bolts from the cylinder tops to the bottom of the bed-plate underneath the main bearings, so relieving all the engine structure of tension stresses. This jacket casting is provided with inspection doors, and liners are pressed in for each cylinder. Each piston is of the usual shallow type, with nine rings, and is supported in the centre on the dished-out piston rod. The pistons are cooled by salt water, supplied through telescoping pipes kept tight by metallic packing, as with pistons of this large size the subsequent cooling of the oil required by oil cooling of the pistons is too difficult. On the bottom of each piston a ring is pinned on to wipe the lubricating oil off the cylinder walls. This oil is collected in a circular tray to prevent it from mixing with the engine lubricating oil, and it is claimed that when filtered 60 per cent. of the original amount supplied may be recovered for use over again.

The compressors usually absorb about 10 per cent. of the power developed.

THE MOTOR SHIP "FIONIA."

General Description of Engines.

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In each cylinder-head, which is a separate casting, are the usual valves—the inlet, exhaust, fuel injection, and starting air-valves—operated by cams, push-rods, and levers. The camshaft is not driven, as in former engines by the same firm, by spur-gearing and connecting-rods, but by large spur-wheels, to give a more rigid connection between the crankshaft and the camshaft. Longitudinal movement of the camshaft to bring the ahead or astern cams under the various push-rods, and rotation of the manœuvring shaft to lift and replace the rollers upon their cams, are effected by a compressed air servo-motor of a similar type to Brown's reversing gear. The starting of the engine is by compressed air at a pressure of 360 lbs. per sq. in.

Separate fuel injection pumps are now provided for each cylinder, and these are driven by levers and links from one of the intermediate shafts driving the camshaft. The control of the engine is substantially the same as in previous ships.

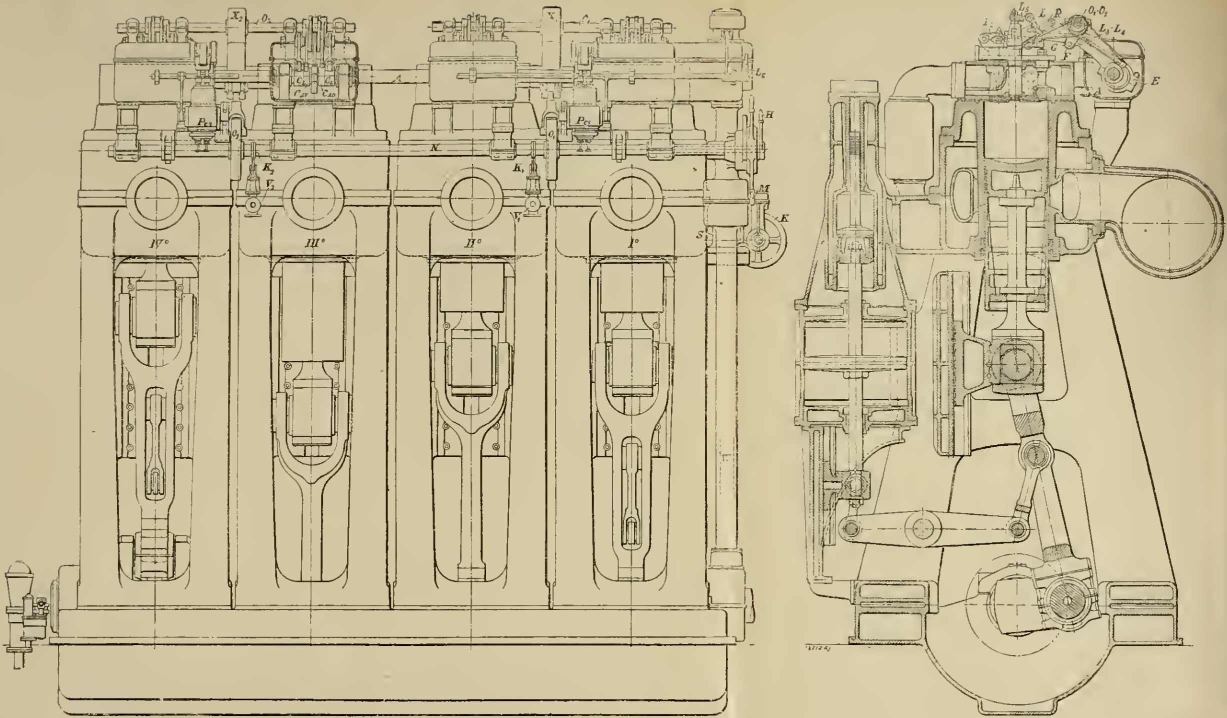
Reavell compressors for fuel injection and starting air are driven from the forward end of the main engine crankshaft.

Starting air is taken from the two main-engine compressors at about 360 lbs. per sq. in., and is stored in two reservoirs, each of 800 cub. ft. cubic capacity. When manœuvring a Reavell auxiliary compressor driven by an electric motor of 200 brake horse-power is used, and when at sea this compressor serves as a stand-by for either of the main engine compressors, being of the same size. Fuel-injection air at about 850 lb. per sq. in. is stored in bottles placed in the centre of the engine room platform.

The auxiliary machinery, all of which is electrically driven, consists of two rotary cooling water pumps, two rotary lubricating oil pumps, and two sets of plunger bilge, sanitary, and piston-cooling pumps. The plunger piston-cooling water pump draws the water from the pistons and delivers it overboard. The water is forced to the pistons by the main cooling water pumps. These pumps are all in duplicate, one of each type serving as a stand-by. The remaining engine-room auxiliaries are:—One rotary ballast pump, a donkey pump, fresh water and fuel-oil pumps, and a refrigerating machine. For supplying electric current to these auxiliaries when the vessel is at sea, and to drive the twelve electric 3-ton to 5-ton winches when in port, two four-cylinder four-cycle oil-engines are provided. Each of these engines is sufficient for the work, and develops 200 brake horse-power at a normal speed of 225 revolutions per minute, and both are of Messrs. Burmeister and Wain's usual design, being in every way self-contained and provided with their own compressors, air-storage, etc. The dynamos work at 220 volts, and for lighting the voltage is transformed to 110 volts. To light the ship when in port, and when the winches are not working, a small crude-oil engine and dynamo set is provided, and by this engine a compressor for bottle-charging can also be worked. In the engine room there are two daily service fuel-oil tanks, each of sufficient capacity for twelve hours' running.

lever E by the pin F, as shown, so that by rotating the shaft O_2 the arc described by the pin F is displaced, and so varies the time of opening given to the valve.

In starting up the engine with compressed air the valve gear is first set in the desired position by rotating the lay shaft A with regard to the vertical shaft W, by which it is driven from the crank shaft. This operation is con-



No. 46.*—Two-Cycle Marine Diesel Engine.

Revs. 170. 4 Cylinders. H.P. 500.

Constructed by Mr Franco Tosi, Legnano.

Index to Parts.

A, Lay or half-time shaft from which the fuel valve is operated by an eccentric.
W, Vertical shaft which by gear wheels from main shaft actuates the lay shaft W overhead.
E, Lever on eccentric for operating fuel valve.
R, Roller for connecting to bell crank lever L_5.
L_5, Bell crank which engages with fuel valve spindle to lift same.
G, Link fixed to eccentric of shaft O_2 and which connects to lever E by pin F.
F, Pin on link G which is displaced in position when shaft O_2 is rotated a given amount: by this means the time of opening of the fuel valve can be varied.
L, L_2, Levers for operating scavenge valves.
C_L, Cams for scavenge valve levers L, L_2.
L_3, Lever for working starting valve in ahead running.
C_{AV}, Cam ,, ,, ,, ,,
L_4, Lever for working starting valve in astern running.
C_{AD}, Cam ,, ,, ,, ,,
O_2, Reversing shaft by rotating which, either ahead or astern cams are put into gear as required.
Pc_1, Fuel pump driven by eccentric from lay shaft A.
Pc_2, ,, ,, ,, ,,
L_6, Hand adjustment lever for fuel valves.
H, Reversing lever for working the air servo-motor which, in turn, operates the vertical shaft, and through it the lay shaft.
S, Servo-motor which, when set in action by the reversing lever H, raises or lowers the vertical shaft W, and cants over the lay shaft to the required position for the valve cam positions.
K, Hand gear for use in emergency if the servo-motor gives out.
V_1V_2, Air supply valves to starting valves.
K_1K_2, Cams for opening or closing valves V_1V_2.
X_1X_2, Reversing discs for operating reversing cam shafts O_1O_2.
N, Intermediate shaft for distributing lever M.
M, Distributing lever for all valves, and which automically unlocks when the cams come into the correct running position either ahead or astern.

* Reprinted from *Engineering*.

[To face page 635.

General Description of Two-Cycle Diesel Engine (No. 46).

The three-stage air compressor, situated above the scavenging pump, has two cylinders, and is driven from the rod of the scavenging pump. The air is first compressed below the lower cylinder in the chamber shown, and thence passes successively to the upper end of this cylinder and to the high pressure cylinder. The pump is water-jacketed, and between the stages the air is passed through a cooler of ample capacity, and thus the increase in temperature at each successive compression stroke is limited. This precludes the possibility of temperature being reached sufficient to lead to the ignition of the cylinder lubricating oil which the air may carry away with it.

The engine is provided with a separate fuel pump for each cylinder, these pumps, P_{C1} and P_{C2}, being mounted in pairs, and driven by eccentrics from the lay shaft A. The control of the fuel supply is carried out in the usual manner, the suction valve being held off their seats during the required portion of the delivery stroke by means of the gear shown. The fuel supply is automatically controlled by the governor, and provision for hand adjustment is also given by the lever L_6; a separate adjustment, worked in connection with the starting gear, is also provided, and will be dealt with more fully later, in connection with the description of the distribution arrangements. Either of the four pumps can be put out of action, as desired, by raising the spindle at the bottom of the pump casting. The spindle in its raised position prevents the closing of the suction valve, and thus prevents the delivery of fuel to the cylinder corresponding to the pump affected.

The lubricating and water-circulating pumps, which are of the ordinary piston type, are driven from the beam of one of the scavenging pumps, while a separate pump, driven in the same manner, is used for raising the oil fuel from the storage tanks to a fuel tank placed at a suitable height above the engine.

As will be seen, the valves are placed in the cylinder covers, the four scavenging valves and starting valves being operated in the usual manner, by simple cam and lever gear. The fuel valve, which is placed in the centre of each cover, however, has a separate gear, since it is considered preferable to provide it with adjustment entirely independent of the other valves. Although a common drive for all valves may appear more simple, it renders the adjustment of the mechanism a more difficult matter, and on account of the general interconnection, makes it practically impossible to adjust any one valve without affecting the operation of the others. The scavenging valves are operated in pairs by means of the levers L and L_2, from the cams C_L. Each starting valve is provided with two sets of driving gear, the lever L_3 and cam C_{AV} being brought into operation for running ahead, whilst the lever L_4 and cam C_{AD} are used for going astern. The levers L_3 and L_4 are mounted on eccentrics keyed to the shaft O_2 in such relation that by the suitable rotation of this shaft either set of gear comes into operation. The fuel-valve is operated from an eccentric on the lay shaft A by means of the lever E, which carries a roller R; this roller engages with the lower end of the bell-crank lever L_5, the other end of which engages with the valve spindle. A link C, which is mounted on an eccentric on the shaft O_2, is connected to the lever E by the pin F, as shown, so that by rotating the shaft O_2 the arc described by the pin F is displaced, and so varies the time of opening given to the valve.

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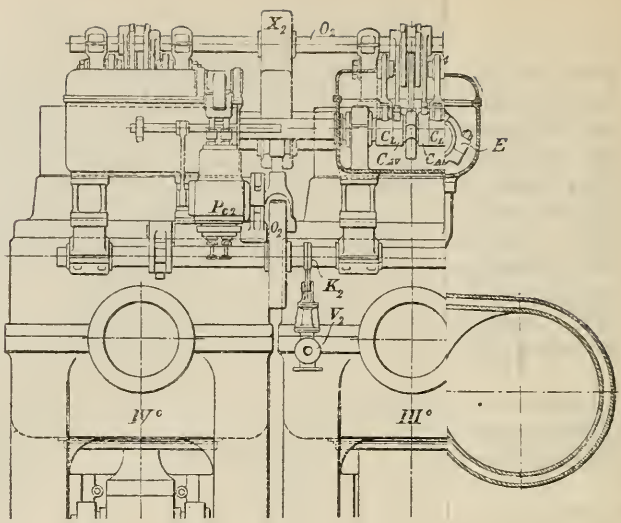

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In starting up the engine with compressed air the valve gear is first set in the desired position by rotating the lay shaft A with regard to the vertical shaft W, by which it is driven from the crank shaft. This operation is con-

trolled by the reversing lever H, which actuates an air servo-motor S, which in turn raises or lowers a sliding portion of the vertical shaft according to the direction in which the lever is moved. The lay shaft A being driven by helical gearing, this vertical movement of the shaft w results in a relative motion of the two shafts, and since the vertical shaft is prevented from rotating by the main shaft, the lay shaft itself is turned and places the valve gear in the desired position. The vertical shaft runs in a thrust-bearing of ample proportions, and it is claimed that the method adopted of raising the shaft in place of using the more general system of a sliding gear wheel results in considerable reduction of wear. An alternative hand gear, controlled by the hand wheel K, is provided for this setting operating, which is available in the event of the servo-motor being out of operation. The movement of the lever H also disengages a locking device, which prevents the accidental movement of the vertical shaft whilst the engine is running; this, however, would appear to be a remote contingency, since the action of the driving gear tends to maintain the shaft automatically in the desired position.

A further function of the reversing lever is the operating of a valve, placed in the supply pipe to the servo-motor, which it opens immediately before the distributing valve comes into action, and closes towards the end of the reversing operation. Thus the air supply is entirely cut off from the motor except during the time the latter is actually in use.

As soon as the desired adjustment of the lay shaft is completed the distribution lever M is automatically unlocked; this lever, which is mounted on the shaft N, operates the reversing shafts O_1 and O_2 by means of the reversing discs on the shaft N and the gears X_1 and X_2. The discs, however, are so arranged that the motion of the shaft N may be either transmitted to one or both of the shafts O.

The movements effected in starting the engine in the ahead direction are as follow:—

1. The air supply valves V_1 and V_2 are opened by means of the cams K_1 and K_2, thus admitting air to the starting valves.

2. The starting valves, operated by the cams C_{AV} and the lever L_3, then admit air to the cylinders, and since at least one of the valves will be open, whatever the point of the crank shaft, and the valves remain open during crank movement of 120°, the engine starts running.

3. The starting valves on cylinders I. and II. are then put out of action and the air valve V_1, automatically closed, thus preventing any leakage of air into the cylinders in the event of the starting-valves being out of repair. The engine is now running on cylinders III. and IV. only, and thus the quantity of air used in starting up is reduced to a minimum.

4. The position of the link G controlling the fuel valve on cylinders I. and II. has now been brought to such a point that the roller R engages with the lever L_5, and causes the fuel valve to lift at an angle of advance of 15°, and holds it open during a crank movement of 35°. At the same time a cam on the shaft N operates the gear, closing the suction valves on the fuel pump P_{cr}, and thus cylinders I. and II. commence running on the oil-fuel, whilst the other two cylinders continue to work by air.

5. The air supply can now be cut off from the other two cylinders, and the supply of fuel adjusted by means of the lever L_6 to give the desired speed. The engine can continue to run on two cylinders only, and this will be found a great convenience in manœuvring and for running at reduced speeds,

6. The fuel supply can next be admitted to the remaining two cylinders. and the fuel valves adjusted to continue running at the reduced speed.

7. Finally, the valves are adjusted to give the normal speed of the engine by means of the lever already referred to.

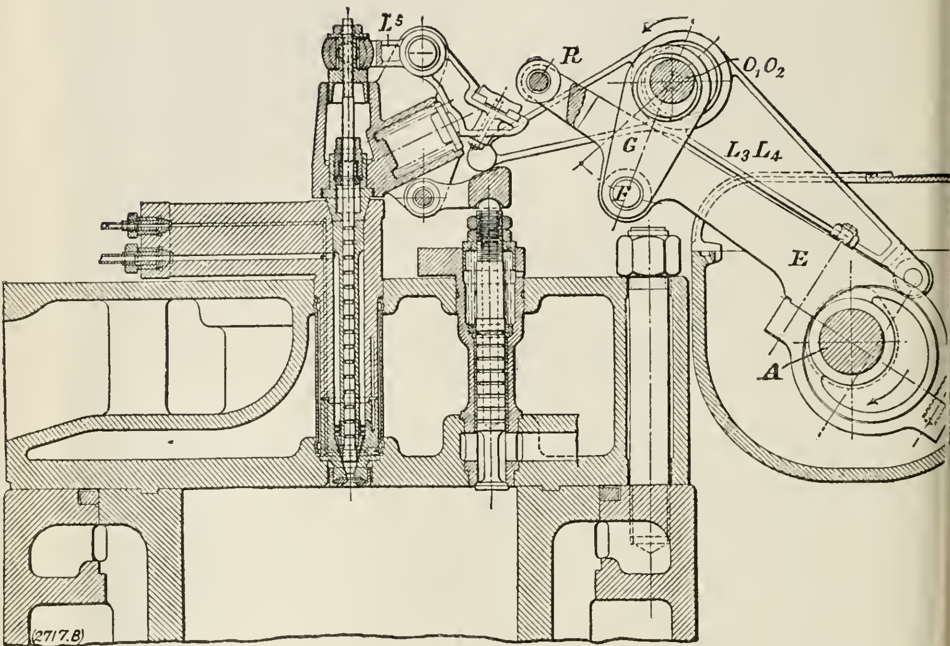
For starting the engine in the reverse direction a similar cycle of movements is carried out, with the difference that the reversing lever H and the distributing lever M are both moved in the opposite direction. The correct adjustment of the scavenging valve gear is carried out by the vertical movement of the shaft W, and when running in the reverse direction the engine develops the same power as under normal conditions.

From this description of the starting and reversing of the engine it will be seen that the whole of the operations are controlled by the reversing and distributing levers H and M. These are interlocked, so that it is only possible to move the reversing lever when the other one is in its central or "stop" position, and similarly, the distributing lever can only be operated when the reversing lever is placed completely over into the ahead or astern position. The arrangement therefore enables reversals to be carried out rapidly with a minimum consumption of air, and eliminates the risk of accidents due to the engineer attempting to carry out the movements out of their correct sequence. The centrifugal governor fitted to the engine is adjusted to cut the fuel pumps out of action should the speed rise 15 per cent. above the normal.

When running at low speeds, as described above, the regulation is effected in such a way that as the speed is reduced the angle of advance and the duration of the fuel-valve opening, measured in relation to the movement of the engine crank, are also decreased, as also is the lift of the valve. Thus, in spite of the fact that the engine is running slower, the actual time that the fuel valve remains open is only slightly increased, and since the lift is decreased, the amount of fuel entering the cylinder is reduced, and ignites gradually without allowing an excessive amount of air to pass. Further, by throttling the exhaust at the silencer a higher pressure can be obtained in the cylinder at the end of the scavenging stroke than occurs under normal conditions, and thus on compression a sufficiently high temperature to produce combustion can be secured, in spite of the increased loss of pressure due to the slower travel of the piston. A further increase in the temperature of the air can be obtained, if necessary, by reducing the amount of circulating water to the scavenging pump. For running at reduced speeds also a lever controlling all the fuel valves causes the oil to enter the cylinders immediately on the lift of the valve and without passing through the atomiser. Thus ignition takes place before any air gains admission, which otherwise might sufficiently reduce the temperature to prevent combustion ensuing. If this lever should not be replaced on resuming normal speed, the only consequence would be imperfect combustion, which would be immediately detected, but the power of the engine would not be affected.

The small cooled area of the combustion chamber relative to its volume, in addition to its general design assisting in the general diffusion of the fuel throughout the air, result in perfect combustion being attained. The scavenging is also very effective, since the provision of four comparatively large valves and the large ratio of the length of cylinder to the diameter prevent the creation of eddies.

As explained below, the pistons are made in two parts, and special arrangements are made whereby they can be withdrawn from the cylinders from below without in any way interfering with the valve gear.



No. 47.*—View of Cylinder Head with Fuel Valve, Starting Valve, Cams, etc.

A, Lay or half time shaft from which the fuel valve is operated by an eccentric.

E, Lever on eccentric for operating fuel valve.

R, Roller for connecting to bell crank lever L^5.

L^5, Bell crank which engages with fuel valve spindle to left same.

G, Link fixed to eccentric of shaft O_2, and which connects to lever E by pin F.

F, Pin on link G which is displaced in position when shaft O_2 is rotated a given amount, and in this way varies the time of opening of the fuel valve.

L_3, Lever for working starting valve in running ahead.

L_4, ,, ,, ,, ,, astern.

O_1O_2, Shaft for placing cams in either ahead or astern running positions.

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* Reprinted from *Engineering*.

General Data of Marine Diesel Engines. (Four-Cycle Type.)

Name of Vessel.	Total Brake Horse-Power.	Number of Engines.	Number of Cylinders per Each Engine.	Diameter of Cylinders in Inches.	Stroke in Inches.	Revolutions per Minute.	Mean Pressure in Lbs.
				Inches.	Inches.		
"Fiona" -	3250	2	6	29·1	43·3	100	91
"Selandia" -	1000	2	6	20·8	25·6	140	53·4
"Abelia" -	1200	2	4	17·25	33	120	98
"Kangaroo"	2250	2	6	22	29·9	140	93·3
"Mississippi"	2900	2	6	27·1	40·5	115	71

Petroleum for Diesel Engines.

Name.	Specific Gravity.	Flash Point.	B.T.U.'s per Lb.
American (light) -	·89 to ·91	240° F.	19000
Texas - - -	·92	240° F.	19000
Shale (Scotch) -	·85 to ·86	230° F.	19500

The oils given in above table are chiefly composed of carbon (85 per cent.) and hydrogen (about 14 per cent.), together with a small proportion of oxygen, about 1 per cent.

Working Data of Diesel Engine.

The following data referring to a modern Diesel two-cycle engine gives a good general idea of the various pressures carried in the cylinders, compressors, scavenge lubrication, and cooling systems, and a careful study of the figures will be found of value to the student.

Engines.

No. of Cylinders.	Diameter of Each.	Stroke.
4	14 in.	21 in.

Cylinder.	M.E.P. (Lbs. □″).	Revolutions.	I.H.P.	Total I.H.P.
No. 1 - -	102	180	149	576
,, 2 - -	98	180	144	
,, 3 - -	94	180	138	
,, 4 - -	99	180	145	

Compressor (Three-Stage).

	M.E.P. (Lbs. □″).	Revolutions.	I.H.P.
1st stage - -	14·2	180	7·6
2nd ,, - -	80	180	10·2
3rd ,, - -	350	180	9·2

Scavenge.

	M.E.P. (Lbs. □″).	Revolutions.	I.H.P.
No. 1 - - -	7·5	180	22
,, 2 - - -	7·2	180	21

Horse-Power of Diesel Engines.

The power of Diesel engines is usually expressed as brake or shaft horse (B.H.P.), and the consumption is referred to this standard. The B.H.P. is measured either by friction brakes for small power engines or, in large engines, by first testing for the actual friction horse-power, and deducting this from the calculated I.H.P., the difference being equal to the B.H.P. The power required to drive the compressors, and to overcome the friction and weight of the moving parts, can be determined by coupling up the engine to a dynamo of known resistance and efficiency, and this constitutes the "friction horse-power."

So that, B.H.P. = I.H.P. — Friction H.P.

The friction H.P. varies from about 25 per cent. to 10 per cent. under various speed and power conditions, therefore the B.H.P. varies from 75 per cent. to 90 per cent. of the I.H.P.

Four-Stroke Engine.

$$\text{I.H.P.} = \frac{\text{Piston area } \boxed{''} \times \text{Stroke in feet} \times \frac{\text{Revs.}}{2} \times \text{M.E.P.}}{33000}.$$

Two-Stroke Engine.

$$\text{I.H.P.} = \frac{\text{Piston area} \times \text{Stroke in feet} \times \text{Revs.} \times \text{M.E.P.}}{33000}.$$

NOTE—

In the four-stroke there is one impulse stroke in every four-stroke or 2 revs.
,, two ,, ,, ,, ,, two ,, 1 rev.

Forced Lubrication System.

Pressure of Oil in Lbs. $\boxed{''}$ at Pump.	Oil Inlet Temperature at Cooler.	Oil Outlet Temperature at Cooler.
12	136°	93°

Water circulation pressure for jackets = 10 lbs.
Water circulation outlet temperature from jackets = 134°.

Pressures and Temperatures of Three-Stage Compressor.

Number of Stage.	Suction Air Pressure and Temperature.		Delivery Air Pressure and Temperature Before Cooling.		Delivery Air Pressure and Temperature After Cooling.	
	Gauge Pressure in Lbs.	Temperature Fahr.	Gauge Pressure in Lbs.	Temperature Fahr.	Gauge Pressure in Lbs.	Temperature Fahr.
No. 1 -	0	62°	45	175°	45	126°
,, 2 -	45	126°	280	320°	280	128°
,, 3 -	280	128°	900	165°	900	104°

The cooling of the air before being used for oil injection purposes is necessary to prevent pre-ignition taking place in the fuel valve pocket above the sprayer ; for the same reason a high flash point oil is required for Deisel engines. The flash point of the various more or less crude or semi-crude oils in general use ranges from about 220° to 280° Fahr.

APPENDIX.

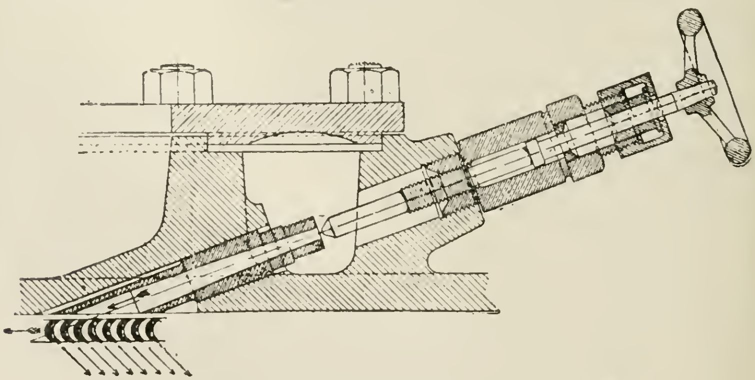
Marine Steam Turbines.*

Principle of Turbine.—The steam turbine is a machine designed to convert the kinetic energy of steam into direct rotary motion. The two principal types of turbine are—(1) Impulse Turbines, those arranged with expanding nozzles in which the high velocity of discharge impinges against a series of small buckets secured on the circumference of a large wheel keyed to the driving shaft, the De-Laval turbine being an example of this type; and those (2) Impulse-reaction Turbines, in which the steam passes through a number of rings of fixed blades and of moving blades, expanding as it travels, an example of which is found in the Parsons turbine

Work by impulse is produced by high velocities, and as the work is done at the expense of the internal heat, water is formed which thus diminishes the heat left.

The Parsons turbine is generally called a reaction turbine, although the correct term should be "impulse-reaction" turbine, as the steam actually does act first by impulse from the guide to the moving blades and afterwards works by reaction from the moving to the guide blades

De-Laval Turbine.—In the specially shaped diverging nozzle of the De-Laval turbine shown in the sketch, the steam expands down

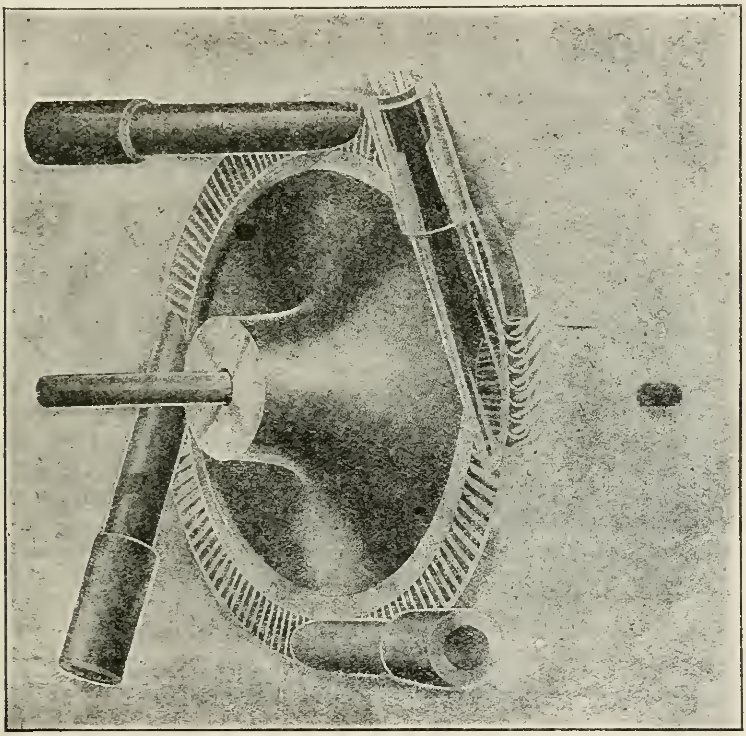


ARRANGEMENT OF NOZZLE AND SHUTTING-OFF VALVE.

No. 1.—De-Laval Turbine.

* For more exhaustive information on turbine practice, see author's "The Marine Steam Turbine" (published by Messrs Crosby Lockwood & Son, London).

to the required exhaust pressure, and the resultant kinetic energy acquired is applied direct to the small buckets or vanes, the steam being in consequence at a very high velocity. To obtain the best efficiency the circumferential velocity of the turbine blades should be equal to about half the velocity of the steam, and this, of course, demands a very high revolution speed. In the De-Laval turbine the speed is often as high as 20,000 revolutions per minute; this can, however, be reduced by suitable gearing to about 2,000 revolutions



No. 2.—View of De-Laval Turbine in Action.

per minute, but as even this is too high for the shafting of marine engines, the non-adaptability of this turbine for marine purposes will be obvious. The steam is admitted to the nozzles (usually four or six in number) and controlled by regulating hand valves.

It is worthy of notice that in this type of turbine the turbine wheel is rotated by steam at the expanded or lowest pressure, as the actual expansion takes place in the nozzle (and not within the vanes or buckets), which is specially designed for that purpose.

The De-Laval type of turbine is much in use for the driving of dynamos, and many steamers are supplied with this turbine for the lighting set of the steamer.*

Parsons Turbine.—In this, the latest and most successful development of marine engineering, steam is admitted direct from the boilers to blades on the shaft drum, thus doing away with the necessity for piston valves or slide valves, cylinders, pistons, piston rods, crossheads, connecting rods, cranks, eccentrics, eccentric rods, and links, &c.

The power to rotate the shaft is therefore applied direct, which in itself constitutes one of the conditions of an ideal engine. The inventor, the Hon. C. A. Parsons, M.A., F.R.S., gives the following brief description of the turbine: —

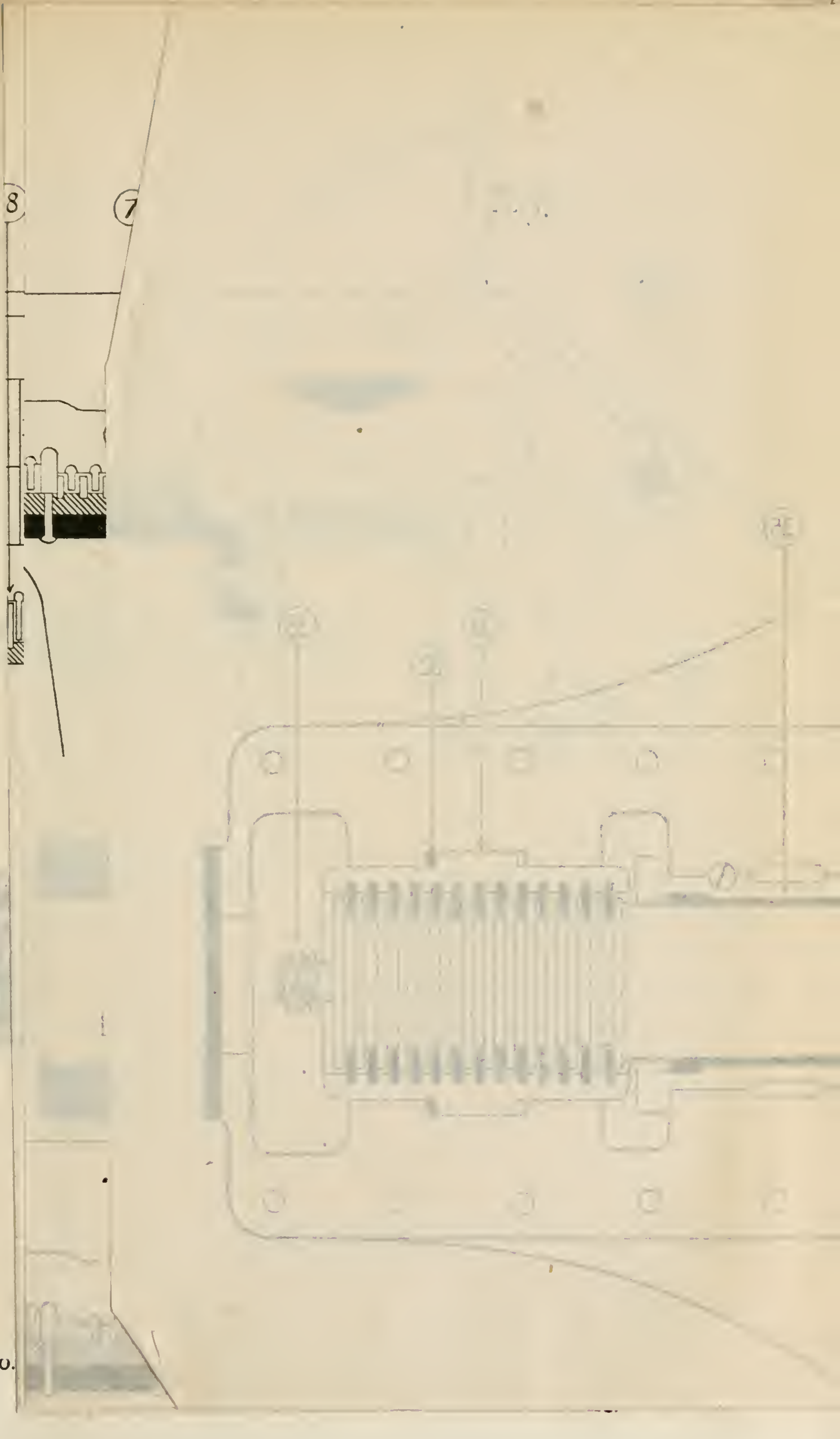
"The Parsons turbine consists of a cylindrical case with numerous rings of inwardly projecting blades. Within this cylinder, which is of variable internal diameter, is a shaft or spindle, and on this spindle are mounted blades, projecting outwardly, by means of which the shaft is rotated. The former are called fixed or guide blades, and the latter revolving or moving blades. The diameter of the spindle is less than the internal diameter of the cylinder, and thus an annular space is left between the two. This space is occupied by the blades, and it is through these the steam flows. The steam enters the cylinder by means of an annular port at the forward end; it meets a ring of fixed guide blades which deflect it so that it strikes the adjoining ring of moving blades at such an angle that it exerts on them a rotary impulse. When the steam leaves these blades it has naturally been deflected. The second ring of fixed blades is therefore interposed, and these direct the steam on to the second ring of rotating blades. The same thing occurs with succeeding rings of guide and moving blades until the steam escapes at the exhaust passage."

Steam from the boiler is admitted by suitable hand valves to the forward end of the casing surrounding the blades, and after passing through a ring of guide blades fixed to the casing, strikes the first ring of shaft or rotor blades; it next passes through the second ring of fixed blades, then the second ring of rotor blades, and so on, passing alternately ring after ring of guide and rotor blades, and so rotating the shaft, until it finally exhausts at the other end of the turbine casing at a reduced pressure.

Parallel Flow.—Parsons' marine turbine is known as that of the impulse and reaction "parallel flow" type, as the steam enters the guide vanes in lines parallel more or less to the shaft axis, and in this way passes from end to end of the turbine, reacting, expanding, and falling in pressure as it travels.

Action of Steam.—As will be seen from the foregoing, the steam striking the blades imparts a turning movement to the shaft, and after reacting and passing through the series of rings of vanes of the H.P.

* For further information on this subject see author's "Marine Steam Turbine."



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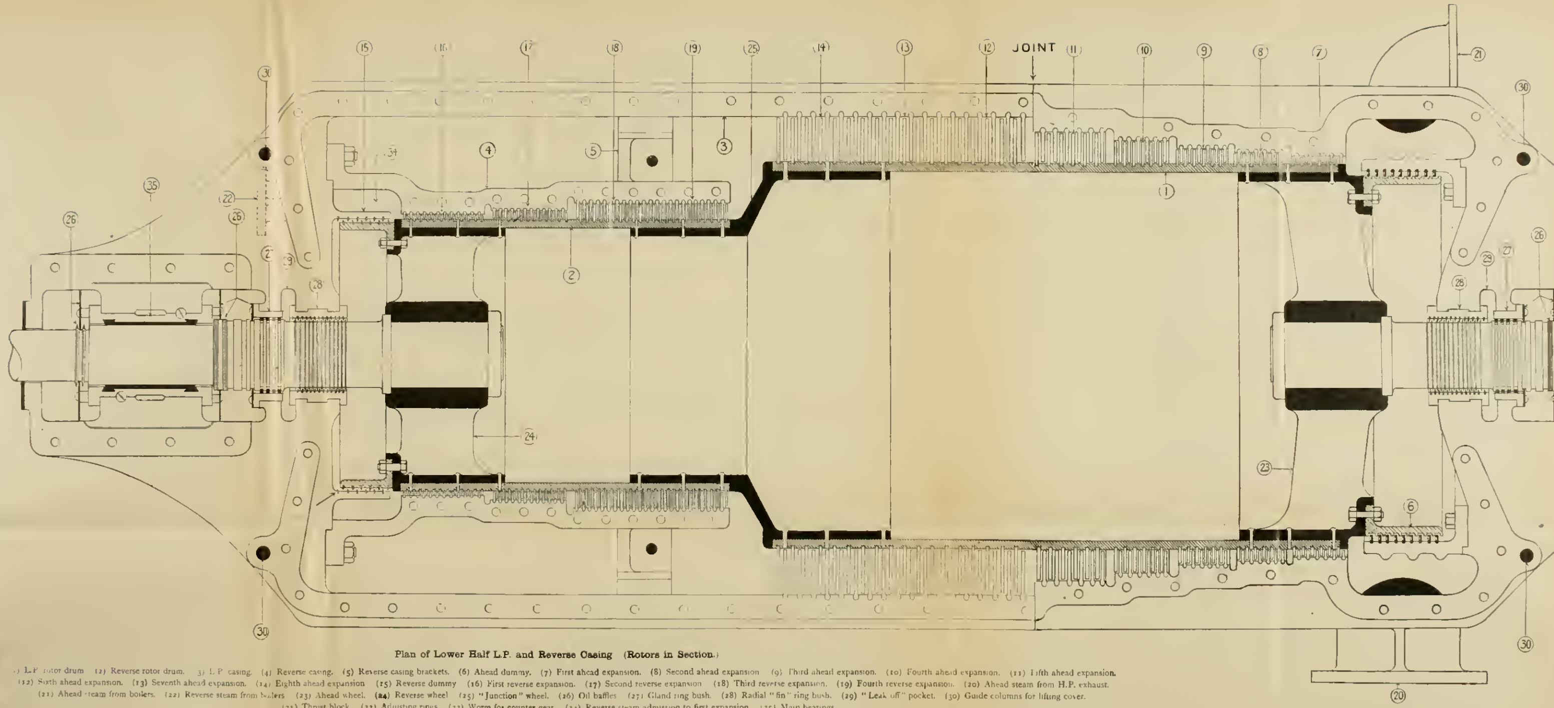
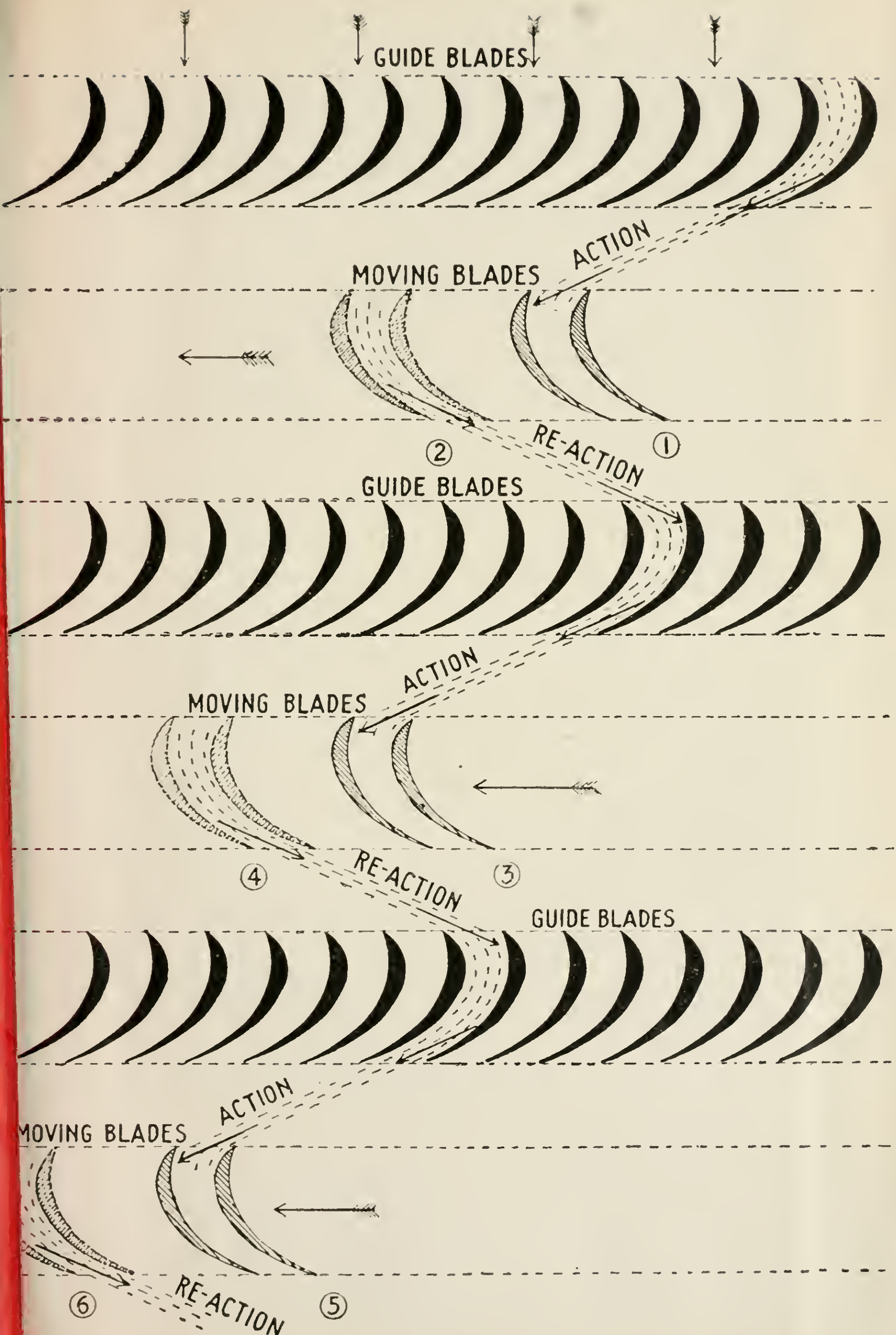


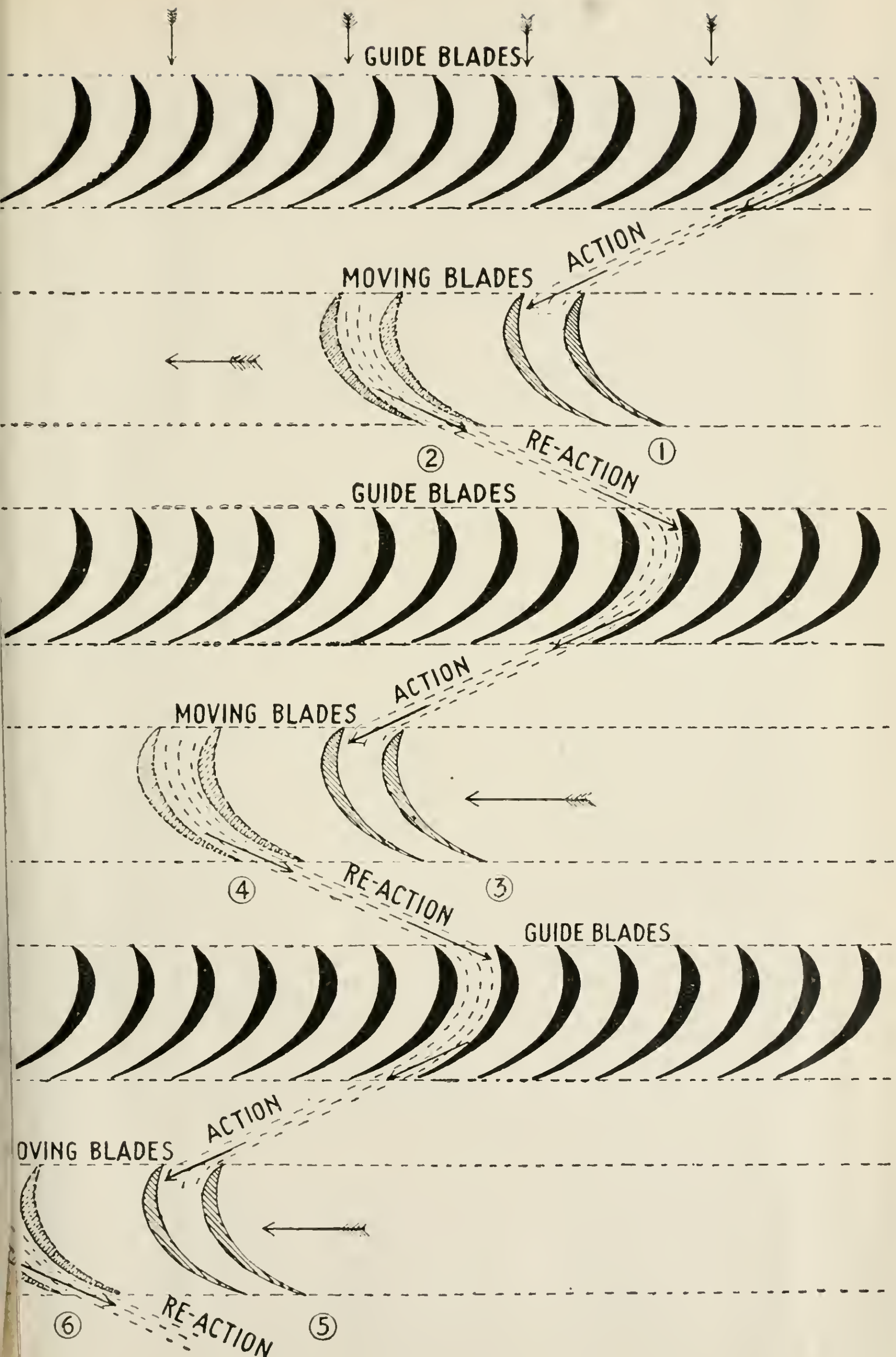
Plan of Lower Half L.P. and Reverse Casing (Rotors in Section.)

(1) L.P. rotor drum. (2) Reverse rotor drum. (3) L.P. casing. (4) Reverse casing. (5) Reverse casing brackets. (6) Ahead dummy. (7) First ahead expansion. (8) Second ahead expansion. (9) Third ahead expansion. (10) Fourth ahead expansion. (11) Fifth ahead expansion. (12) Sixth ahead expansion. (13) Seventh ahead expansion. (14) Eighth ahead expansion. (15) Reverse dummy. (16) First reverse expansion. (17) Second reverse expansion. (18) Third reverse expansion. (19) Fourth reverse expansion. (20) Ahead steam from H.P. exhaust. (21) Ahead steam from boilers. (22) Reverse steam from boilers. (23) Ahead wheel. (24) Reverse wheel. (25) "Junction" wheel. (26) Oil baffles. (27) Gland ring bush. (28) Radial "fin" ring bush. (29) "Leak off" pocket. (30) Guide columns for lifting cover. (31) Thrust block. (32) Adjusting rings. (33) Worm for counter gear. (34) Reverse steam admission to first expansion. (35) Main bearings.

[To face page 644]



No. 3.—Path traced out by Steam in Parsons Turbine.



No. 3.—Path traced out by Steam in Parsons Turbine.

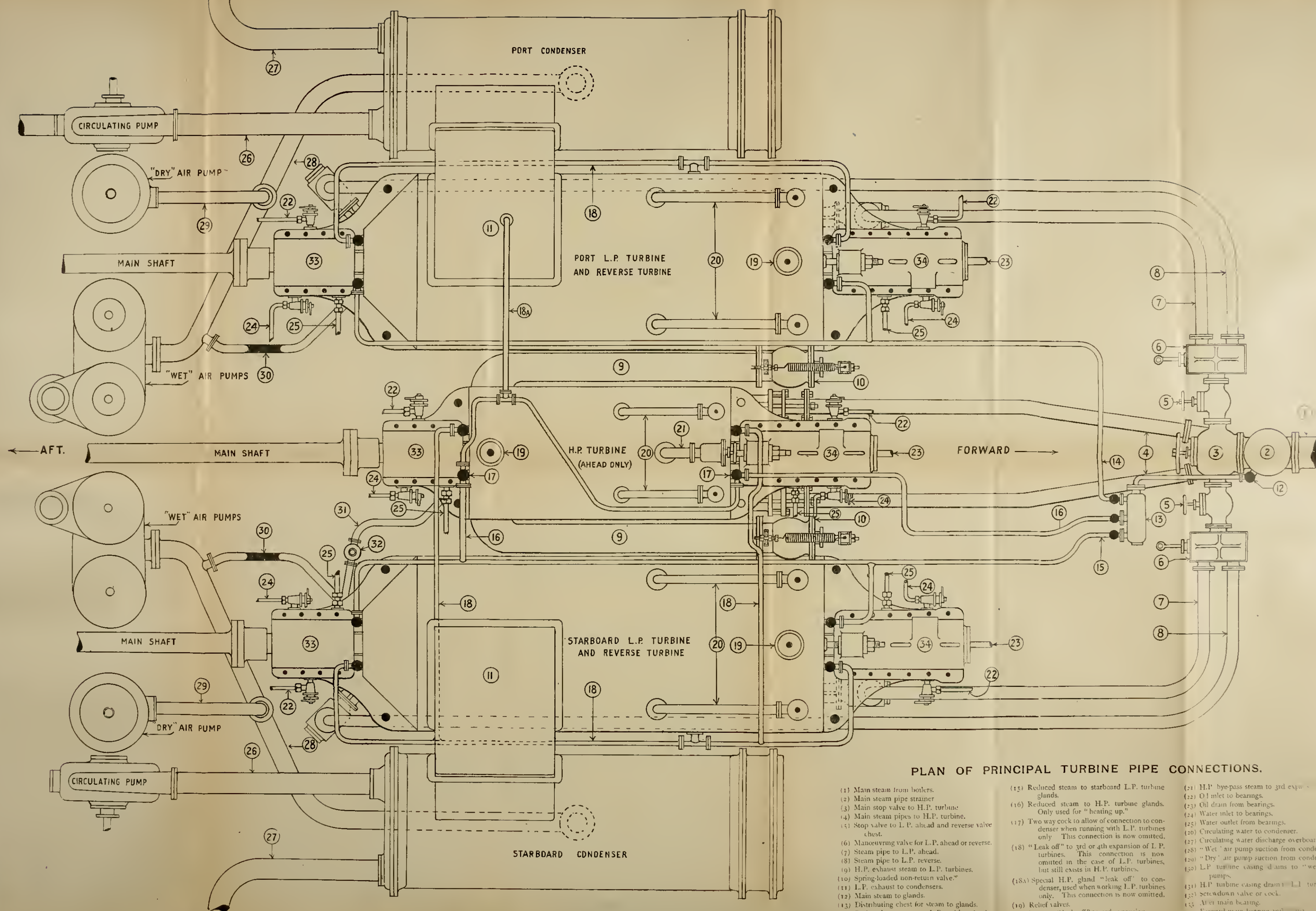
turbine exhausts simultaneously into the two L.P. turbines, one on either side, and expanding through the longer casing and shaft blades of these turbines, finally exhausts, at a low absolute pressure of from $1\frac{1}{2}$ to 2 lbs., into the condensers, one for each L.P. turbine.

Flow of Steam through Blades.—The diagram on page 629 shows graphically the path followed out by the steam as it passes through each successive ring of fixed or moving blades. Observe that the steam, after passing through the first ring of guide blades, strikes the first ring of rotor or moving blades and by the action set up assists in rotating the shaft; by the time the steam has changed its direction the rotor has moved round a certain distance (from 1 to 2), and the reaction of the steam, due to its somewhat sudden change of direction, still further assists in rotating the shaft. The steam then leaves the rotor blades and enters the next ring of guide blades, where, after again being deflected in its path, it enters the next ring of moving blades, where the action and reaction process is again repeated; leaving the second ring of moving blades at position 4 the steam enters the third ring of guide blades at a point 5 still farther round the circumference, and so on for each of the following rings.

It should be noted that the steam leaving the moving blades is deflected by the blade curvature, and strikes the casing blades, which, if free to revolve, would be acted on by the steam and moved round similarly to the rotor blades, but in the opposite direction; instead of this taking place, however, the casing blades being fixed resist the impact, and the resulting reaction throws back the steam, the velocity of which is thus increased. The pressure is therefore utilised in augmenting the steam speed, hence the statement that "in the guide blades the steam does work on itself to increase its own velocity." The steam thus describes a somewhat zigzag path in passing along the rotor, its direction being not unlike that of a screw thread. Work is done at each ring of blades and heat given up, expansion of the steam taking place in due proportion, so that the velocity of flow increases, and to allow for this the lengths and spacings of the blades must be increased to maintain the same ratio between the blade velocity and the steam velocity, upon which the turbine efficiency depends.

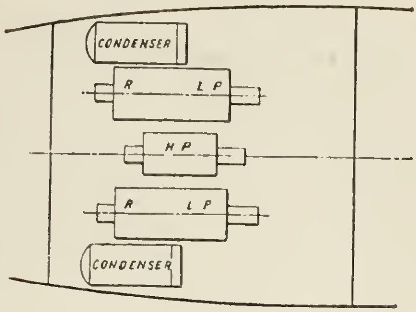
The diagram shows the imaginary path described by a small portion of steam, and the dotted blades show the circumferential advance of the rotor blades at each ring, which produces the thread-like path traced out by the steam.

Turbine Arrangements.—In steamers of normal size for either channel or deep-sea service, the standard arrangement consists of five turbines, three for ahead and two for reverse running; three shafts are fitted with one propeller on each, the reverse turbines being placed within the L.P. turbine casings aft. In exceptionally large steamers, such as the "Lusitania" and "Mauretania," four lines of shafting are arranged, with two ahead H.P. turbines and two ahead L.P. turbines,

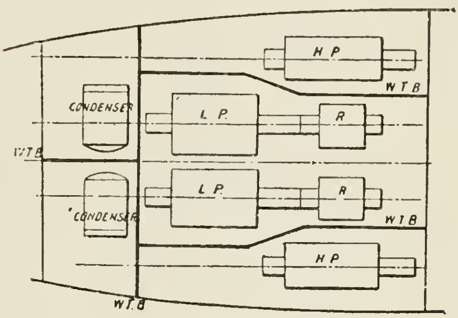


PLAN OF PRINCIPAL TURBINE PIPE CONNECTIONS.

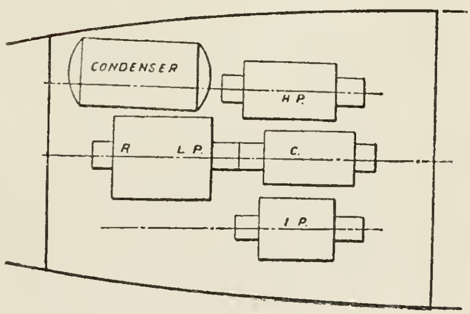
(1) Main steam from boilers.
(2) Main steam pipe strainer
(3) Main stop valve to H.P. turbine
(4) Main steam pipes to H.P. turbine.
(5) Stop valve to L. P. ahead and reverse valve chest.
(6) Manœuvring valve for L.P. ahead or reverse.
(7) Steam pipe to L.P. ahead.
(8) Steam pipe to L.P. reverse.
(9) H.P. exhaust steam to L.P. turbines.
(10) Spring-loaded non-return valve.*
(11) L.P. exhaust to condensers.
(12) Main steam to glands.
(13) Distributing chest for steam to glands.
(14) Reduced steam to port L.P. turbine glands.
(15) Reduced steam to starboard L.P. turbine glands.
(16) Reduced steam to H.P. turbine glands. Only used for "heating up."
(17) Two way cock to allow of connection to condenser when running with L.P. turbines only. This connection is now omitted.
(18) "Leak off" to 3rd or 4th expansion of L. P. turbines. This connection is now omitted in the case of L.P. turbines, but still exists in H.P. turbines.
(18A) Special H.P. gland "leak off" to condenser, used when working L.P. turbines only. This connection is now omitted.
(19) Relief valves.
(20) Dummy "leak off" to 3rd expansion.
(21) H.P. bye-pass steam to 3rd exp[illegible]
(22) Oil inlet to bearings.
(23) Oil drain from bearings.
(24) Water inlet to bearings.
(25) Water outlet from bearings.
(26) Circulating water to condenser.
(27) Circulating water discharge overboard.
(28) "Wet" air pump suction from condensers.
(29) "Dry" air pump suction from condensers.
(30) L.P. turbine casing drains to "wet" air pumps.
(31) H.P. turbine casing drain [illegible] L.P. [illegible]
(32) Screwdown valve or cock.
(33) After main bearing.
(34) Forward main bearing and [illegible]



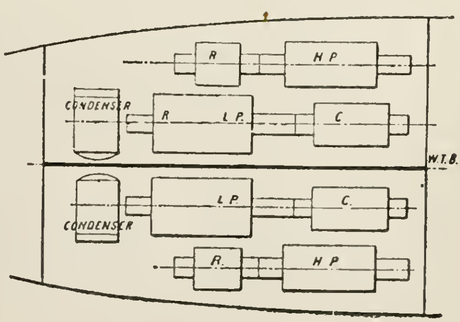
(1) **Standard arrangement,** one H.P. and two L.P. turbines, three shafts, one propeller to each.



(2) **Large passenger steamer arrangement,** two H.P., two L.P., and two independent reverse turbines, four shafts, one propeller to each.

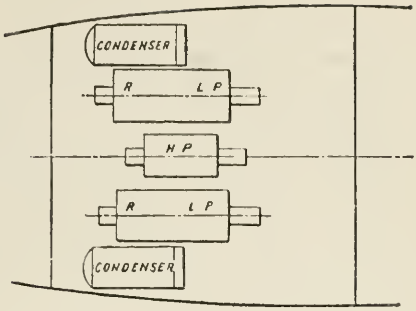


(3) **Torpedo craft arrangement,** one cruising, one I.P., one H.P., and one L.P. turbine, three shafts, one propeller to each.

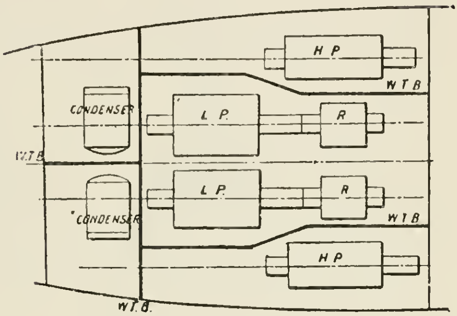


(4) **Battleship or cruiser arrangement,** two cruising ahead, two H.P. ahead, two L.P. ahead turbines, also two H.P. reverse, and two L.P. reverse turbines, four shafts, one propeller to each.

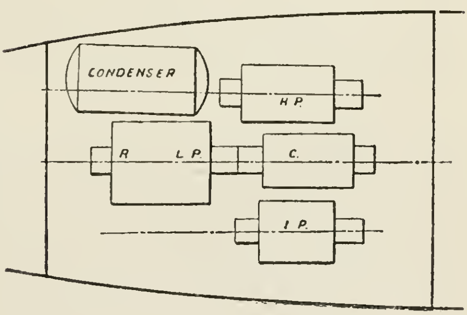
(1) **Standard arrangement,** one H.P. and two L.P. turbines, three shafts, one propeller to each.



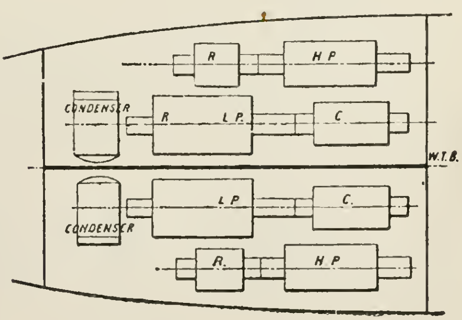
(2) **Large passenger steamer arrangement,** two H.P., two L.P., and two independent reverse turbines, four shafts, one propeller to each.



(3) **Torpedo craft arrangement,** one cruising, one I.P., one H.P., and one L.P. turbine, three shafts, one propeller to each.



(4) **Battleship or cruiser arrangement,** two cruising ahead, two H.P. ahead, two L.P. ahead turbines, also two H.P. reverse, and two L.P. reverse turbines, four shafts, one propeller to each.



also two independent reverse turbines on the inner shafts. This arrangement, with the further addition of other two reverse turbines and two cruising turbines, is carried out in the case of large battleships and cruisers; sometimes the cruising turbines are compound, one H.P. and one M.P., but generally both are of the same size, and receive direct steam from the boilers simultaneously. It should be noted that the Admiralty have decided to discard cruising turbines altogether in future, as in most cases the consumption at the low powers developed by these turbines does not justify their existence, in addition to the loss of power produced by the turbine blade resistance when running ahead or astern with the main turbines. Cruisers of the "Inflexible"-"Indomitable" type have ten turbines fitted, four ahead turbines—two H.P. and two L.P.—and four reverse—two H.P. and two L.P.—also two cruising turbines fitted, one on each H.P. turbine shaft, and intended for low cruising speeds and powers.

In torpedo craft the three-shaft arrangement is often carried out, but the turbines are arranged in triple series, one H.P. (centre), one M.P. (wing), and one L.P. (wing). Sometimes cruising turbines are fitted in addition to these in the case of large high-speed destroyers. The foregoing are the arrangements of turbines in present practice, but other arrangements have been proposed by the Parsons Company.

As regards the new combination arrangement of reciprocating engines and turbine, the steamers at present under construction are fitted with two wing triple or quadruple engines, both exhausting at a pressure below the atmosphere into the turbine on the centre shaft. An alternative design consists of one centre reciprocating engine exhausting into two wing turbines.

Steam Flow through Turbines.—In the standard turbine arrangement of five turbines—three ahead and two reverse—the steam, after expanding through the H.P. turbine, exhausts to both I.P. turbines simultaneously, and then to the two condensers. In the "Lusitania" design, the steam expands through each H.P. turbine, then through each L.P. to the condensers of each respective side.

In the "Inflexible"-"Indomitable" class turbine arrangement, at full ahead power, the steam expands through each H.P., then each L.P., and then to the condensers of each side. At reduced ahead power, the steam first expands through each cruising turbine, then through each H.P. and L.P. turbine of each side, finally exhausting to the condensers. In running astern the steam first expands through the H.P. reverse turbine, then the L.P. reverse turbine, and finally exhausts to the condenser.

In the destroyer triple arrangement at full power, the steam first expands through H.P. turbine, then M.P. turbine, and L.P. turbine to the condenser, and at reduced power or cruising speed, the steam first expands through the cruising turbine, H.P. turbine, M.P. turbine, and L.P. turbine to the condenser.

O MARINE WORK.

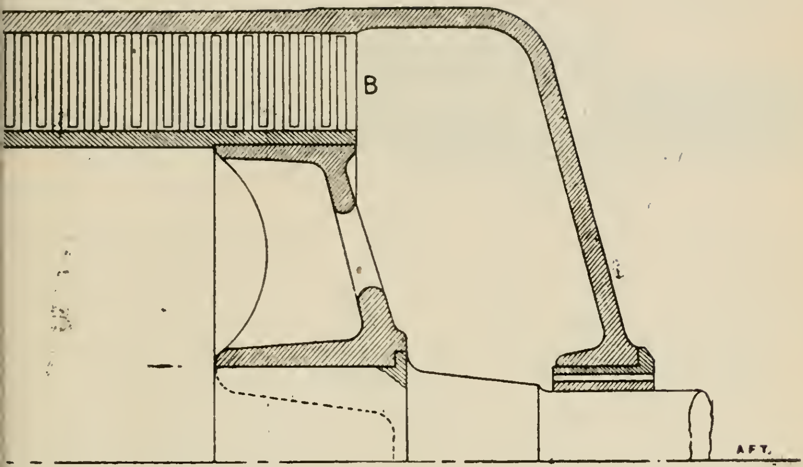
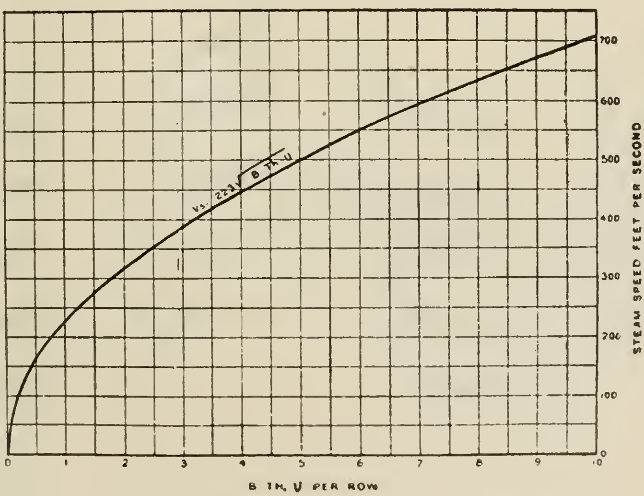
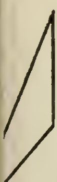


Fig. 8.





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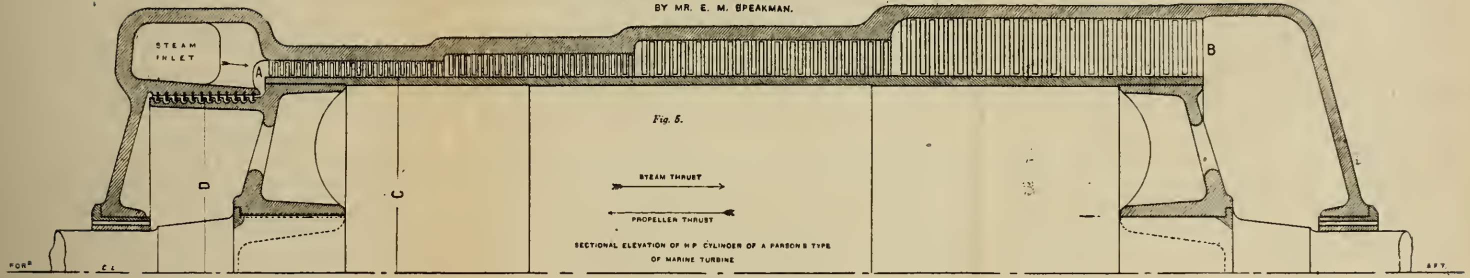
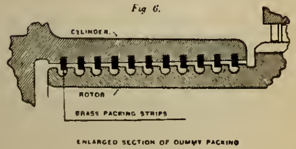
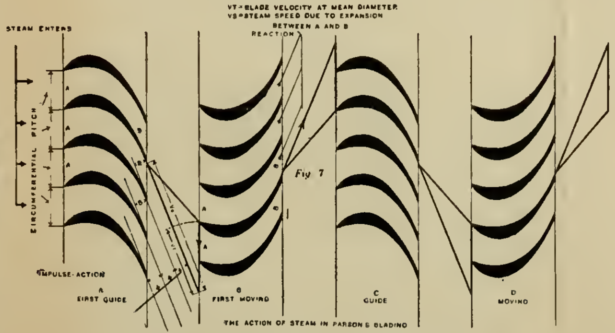
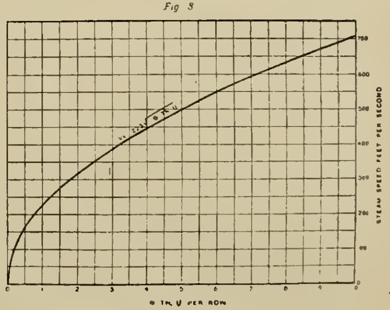
THE DETERMINATION OF THE PRINCIPAL DIMENSIONS OF THE STEAM TURBINE, WITH SPECIAL REFERENCE TO MARINE WORK
BY MR. E. M. SPEAKMAN.









No. 4.—Sectional View of H.P. Turbine.

(Also Blade Angles and Steam Velocities.)

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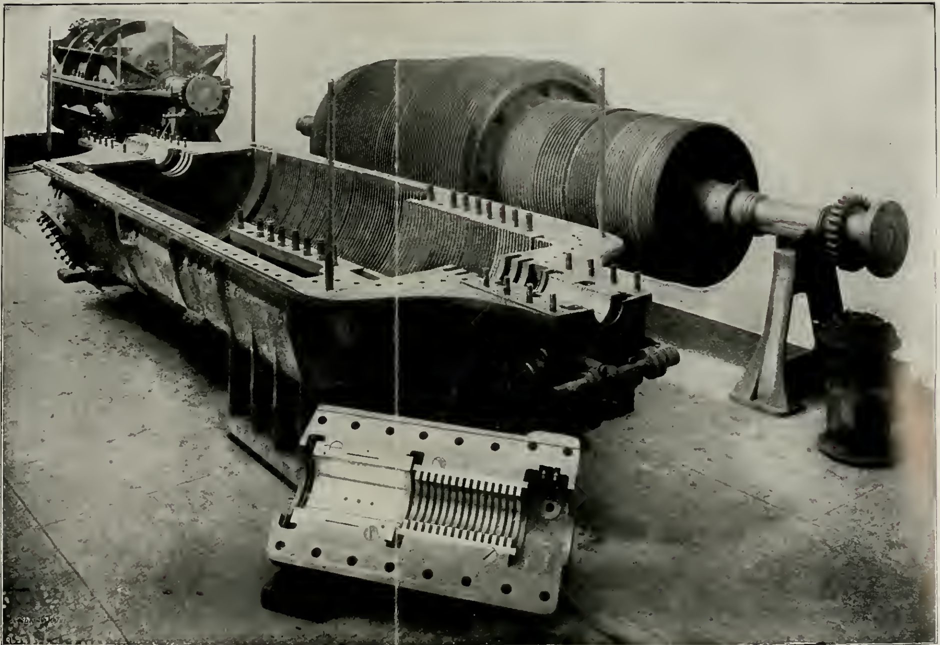
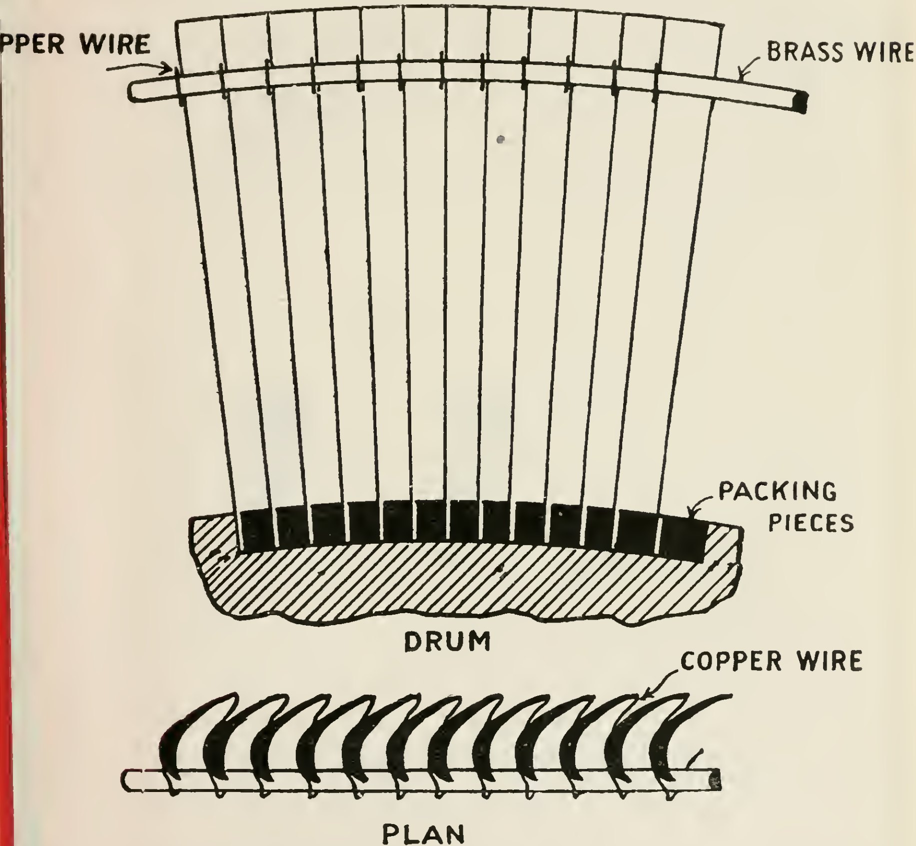


No. 5.—L.P. Ahead and Reverse Turbines.

Messrs A. & J. Inglis Ltd. (The H.P. Turbine shown in background, Khedive's Yacht "Mahroussa.")

[To face page 649.

Increase of Steam Volume.—To allow of the steam increasing in volume, as fall of pressure takes place, the various sets of blades increase in length from the forward to the after end, the clearance spaces between the blades also increasing in proportion, which necessitates packing pieces of a larger size being employed. The blades also vary in shape or curvature, being flatter in section aft



No. 6.—Elevation and Plan of Rotor Blades in Position, showing how secured (full size).

than forward Each set of blades for each expansion requires its own allowance for expansion of metals by heat, so that the working clearance between the blades and casings or drums increases slightly throughout the turbine from forward aft. One of the practical difficulties met with in turbine construction at present is the correct adjustment for this expansion, as slight mishaps have occurred in

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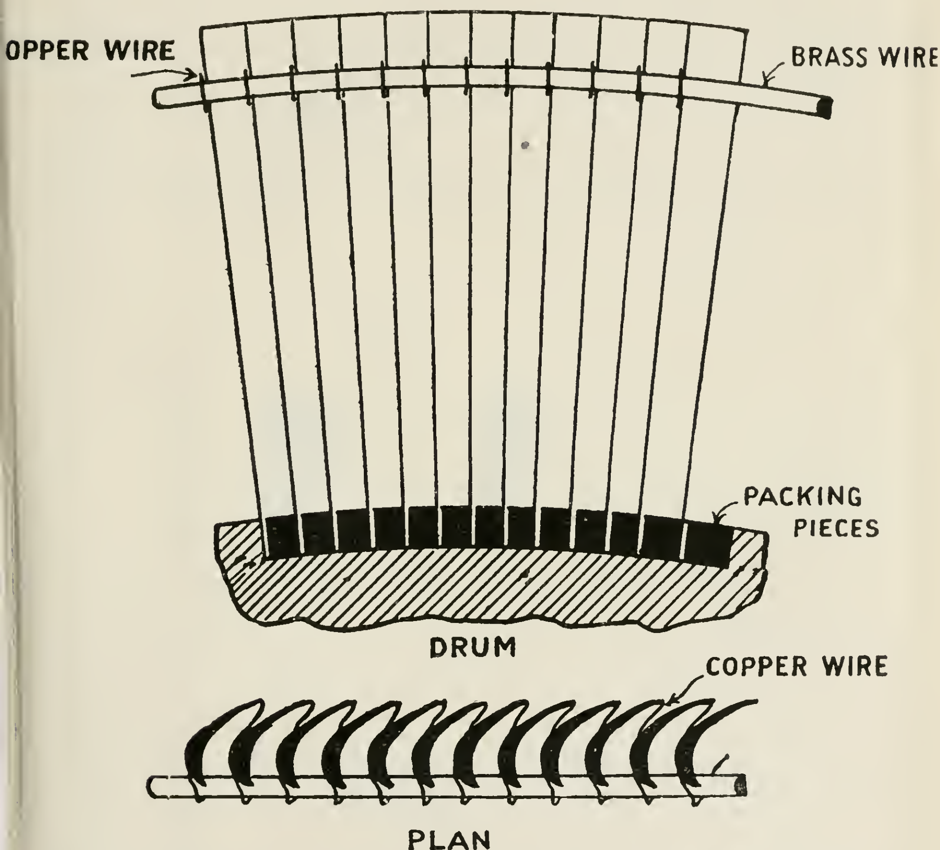

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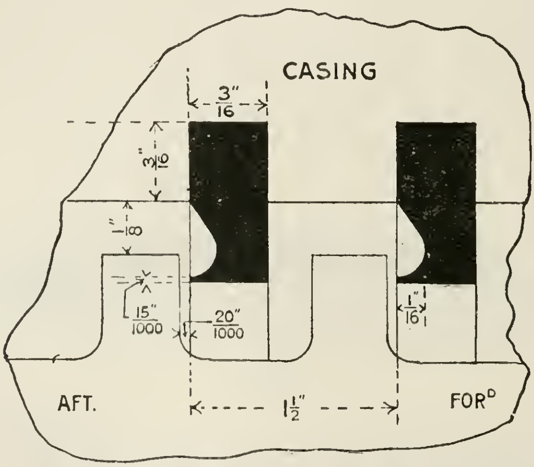
than forward Each set of blades for each expansion requires its own allowance for expansion of metals by heat, so that the working clearance between the blades and casings or drums increases slightly throughout the turbine from forward aft. One of the practical difficulties met with in turbine construction at present is the correct adjustment for this expansion, as slight mishaps have occurred in one or two instances owing to fouling of the parts when heated up, the clearance allowance being insufficient.

Strictly, each successive ring of blades should be either of a wider pitch or greater height than the preceding one, as the steam is continuously falling in pressure and expanding in volume.

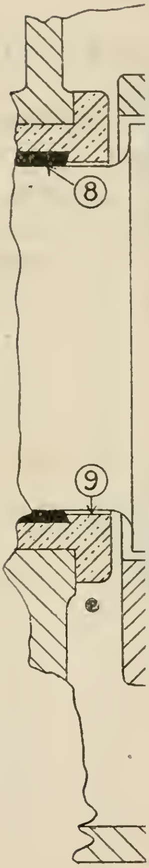
Dummies.—The dummies are placed at the steam admission end of each turbine, and two kinds of dummies, known respectively as radial and facial, are employed: the facial dummy is usually fitted in the ahead turbine, and the radial dummy in the astern turbine. The principle of the dummy is to prevent the steam from escaping through the interior of the rotor to the exhaust end of the casing instead of doing its legitimate work in passing through the blades of the turbine. Another reason is that if no dummies were fitted, the full initial pressure would be on the glands instead of exhaust or terminal pressure. A facial



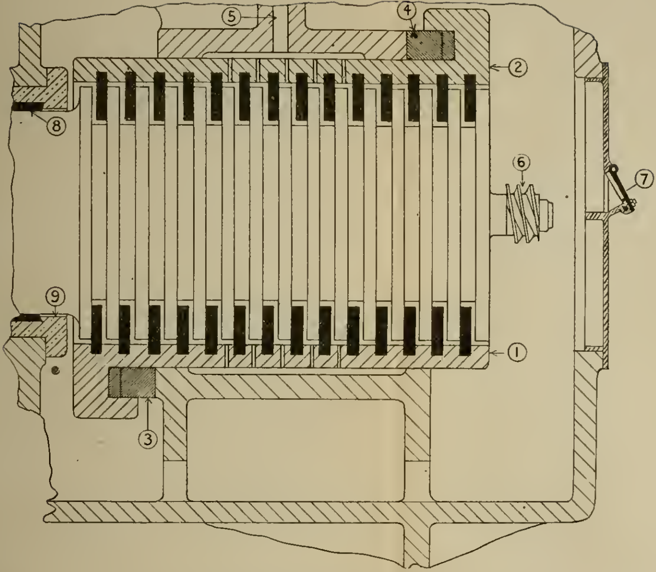
No. 8.—Ahead H.P. Dummy Facial Rings.

(With average dimensions.)

dummy consists of two parts called the casing dummy and the rotor dummy. The casing dummy is a cast-iron cylinder, which is in two halves, bolted together at the horizontal joint. This cylinder is bored out and grooved, the grooves being usually $\frac{1}{8}$ in. wide and $\frac{3}{16}$ in. deep, and into these grooves are driven brass strips. The brass strips are bent to the radius of the cylinder, and a serration made in them, so that the serration is just flush when the strip is driven into groove. After the blades are in place, the metal of the cylinder is caulked into the serration, thus binding the strips. The strips are cut in lengths of about 6 in.



(1) Ahead thr
(2) Astern thr
(3) Taper key
(4) Taper key



No. 7.—Turbine Thrust Block.

Lower Half for Ahead Thrust; Upper Half for Astern Thrust.

(1) Ahead thrust.
(2) Astern thrust.
(3) Taper key for adjustment of lower half.
(4) Taper key for adjustment of upper half.
(5) Oil inlet.
(6) Counter gear worm.
(7) Inspection door.
(8) White metal of mean bearing.
(9) "Reliefs" for wear down.

[*To face page* 650.

"BLADING LIST."

The following example of a "blading list" from actual practice will afford the student a fair idea as to the blade heights, &c., generally fitted :—

Type—Fast Channel Steamer.

Speed, 22 knots.
Equivalent I.H.P., 8,500 (approximate).

Turbine Data.

H.P. Turbine. (Drum, 2 ft. 6 in. Diam.)

Expansion.	Number of Blade Rows.	Blade Heights.
1	13	$1\frac{1}{2}$ in.
2	13	$2\frac{1}{4}$ "
3	14	3 "
4	14	$4\frac{1}{2}$ "

L.P. Turbines (two). (Drums, 3 ft. 9 in. Diam.)

Expansion.	Number of Blade Rows.	Blade Heights.
1	7	$1\frac{1}{2}$ in.
2	7	$2\frac{1}{4}$ "
3	7	3 "
4	7	$4\frac{1}{2}$ "
5	7	6 "
6	7	8 "
7	7	8 "
8	7	8 "

Astern Turbines. (2 ft. 6 in. Diam.)

Expansion.	Number of Blade Rows.	Blade Heights.
1	10	$\frac{11}{16}$ in.
2	10	$2\frac{1}{2}$ "
3	10	$2\frac{7}{8}$ "
4	10	$2\frac{7}{8}$ "
5	10	$2\frac{7}{8}$ "

"BLADING LIST."

The following example of a "blading list" from actual practice will afford the student a fair idea as to the blade heights, &c., generally fitted :—

Type—Fast Channel Steamer.

Speed, 22 knots.

Equivalent I.H.P., 8,500 (approximate).

Turbine Data.

H.P. Turbine. (Drum, 2 ft. 6 in. Diam.)

Expansion.	Number of Blade Rows.	Blade Heights.
1	13	$1\frac{1}{2}$ in.
2	13	$2\frac{1}{4}$,,
3	14	3 ,,
4	14	$4\frac{1}{2}$,,

L.P. Turbines (two). (Drums, 3 ft. 9 in. Diam.)

Expansion.	Number of Blade Rows.	B'ade Heights.
1	7	$1\frac{1}{2}$ in.
2	7	$2\frac{1}{4}$,,
3	7	3 ,,
4	7	$4\frac{1}{2}$,,
5	7	6 ,,
6	7	8 ,,
7	7	8 ,,
8	7	8 ,,

Astern Turbines. (2 ft. 6 in. Diam.)

Expansion.	Number of Blade Rows.	Blade Heights.
1	10	$\frac{11}{16}$ in.
2	10	$2\frac{1}{2}$,,
3	10	$2\frac{7}{8}$,,
4	10	$2\frac{7}{8}$,,
5	10	$2\frac{7}{8}$,,

the casing in one row a 9-in. piece is put in, and in the succeeding row a $4\frac{1}{2}$-in. piece is put in, so that the joints in each row are not in line. The strips are left "·012" clear of each other at the ends so as to allow for expansion. After the blades are all in place and caulked, the dummy is put into lathe, and the blades turned up. The blades have a face bearing of "·015" so as to ensure that if the rotor dummy should touch, the friction caused thereby would be reduced to a minimum. The rotor dummy is of steel, and usually cylindrical, and is sometimes made in two halves. The dummy is rigidly bolted to the rotor, and turned up in place. A series of grooves corresponding to the brass strips in the casing dummy are turned out, having a fillet in both sides of groove, the grooves being $\frac{3}{16}$ in. deep. The brass strips in the casing dummy project into the groove in the rotor dummy $\frac{1}{8}$ in. When the rotors are set to position in the casing, the factor which determines this position is the dummy clearance, this varying according to the size of turbines. For average sizes the clearance is usually as follows :—

·025 to ·040 in the high-pressure turbine,
and ·025 to ·060 in the low-pressure turbine.

The working clearance allowed between the tips of the rotor blades and casing, also between the casing blades and rotor, are given in the following table taken from an actual case :—

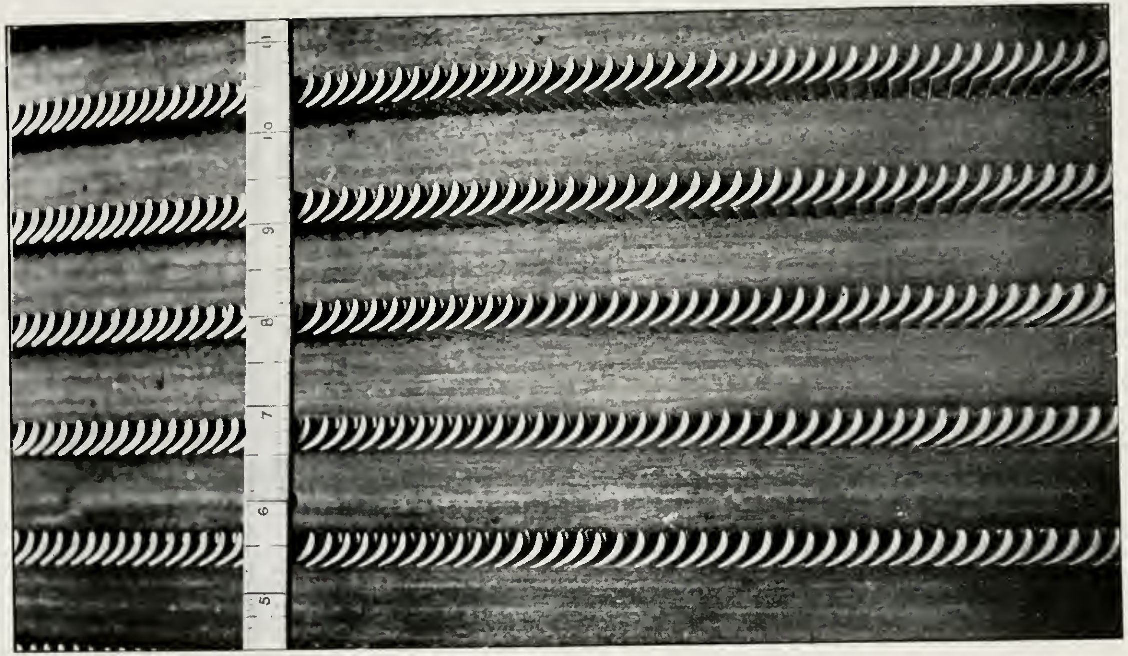
BLADE TIP CLEARANCES.

Taken when cold.

H.P. Turbine.

Rotor Drum, 48 in. Diameter.

Expansion.	Radial Clearance (Port).		Radial Clearance (Starboard).		Longitudinal Clearance.			
	Rotor Blade Tips.	Casing Blade Tips.	Rotor Blade Tips.	Casing Blade Tips.	Port Forward Side.	Port Aft Side.	Starboard Forward Side.	Starboard Aft Side.
					Inch.	Inch.	Inch.	Inch.
No. 1	·041	·041	·052	·043	$\frac{1}{4}$	$\frac{3}{16}$	$\frac{15}{16}$	$\frac{7}{32}$
" 2	·051	·055	·059	·049	$\frac{9}{32}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$
" 3	·063	·055	·070	·061	$\frac{9}{32}$	$\frac{1}{4}$	$\frac{9}{32}$	$\frac{1}{4}$
" 4	·049	·055	·054	·051	$\frac{23}{64}$	$\frac{5}{16}$	$\frac{11}{32}$	$\frac{11}{32}$



VIEW OF BLADES LOOKING DOWN ON ROTOR.

NOTE.—The scale of inches shown gives a fair idea of the blade widths and the distance apart of the various rows.

NOTE.—In recent practice the longitudinal clearance between the blade rows has been considerably reduced, as this alteration is expected to result in increased turbine efficiency.

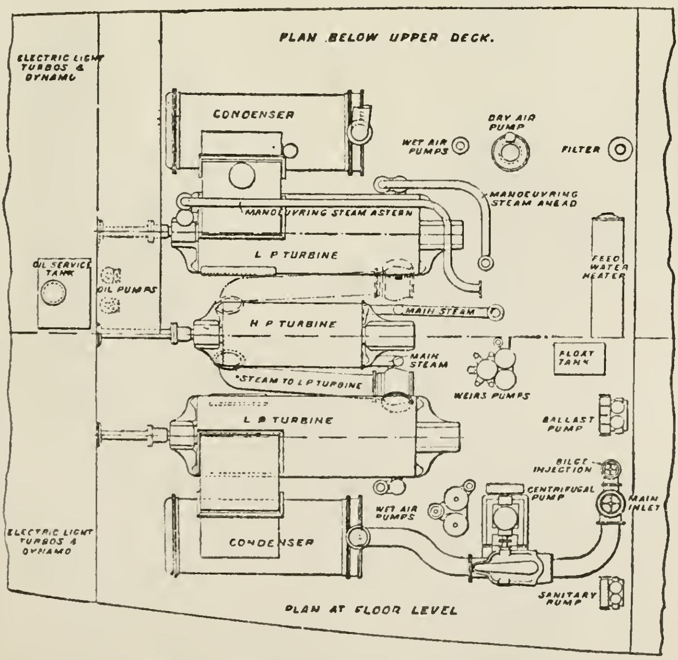
[To face page 652.

Starboard L.P. Turbine.

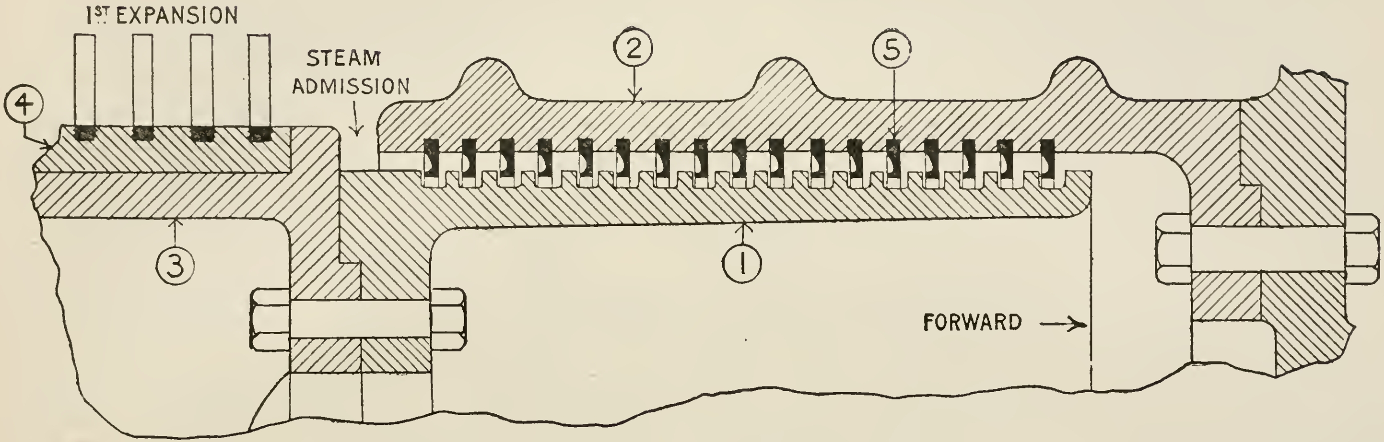
Rotor Drum, 68 in. Diameter.

Expansion.	Radial Clearance (Port).		Radial Clearance (Starboard).		Longitudinal Clearance.			
	Rotor Blade Tips.	Casing Blade Tips.	Rotor Blade Tips.	Casing Blade Tips.	Port Forward Side.	Port Aft Side.	Starboard Forward Side.	Starboard Aft Side.
					Inch.	Inch.	Inch.	Inch.
No. 1	·070	·080	·068	·070	$\frac{7}{32}$	$\frac{7}{32}$	$\frac{13}{64}$	$\frac{17}{64}$
„ 2	·072	·085	·070	·085	$\frac{5}{16}$	$\frac{7}{32}$	$\frac{1}{4}$	$\frac{5}{16}$
„ 3	·078	·090	·082	·092	$\frac{11}{32}$	$\frac{9}{32}$	$\frac{1}{4}$	$\frac{17}{64}$
„ 4	·082	·092	·082	·085	$\frac{3}{8}$	$\frac{11}{32}$	$\frac{5}{16}$	$\frac{3}{8}$
„ 5	·085	·093	·085	·088	$\frac{7}{16}$	$\frac{11}{32}$	$\frac{11}{32}$	$\frac{11}{32}$
„ 6	·095	·123	·098	·115	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{13}{32}$	$\frac{7}{16}$
„ 7	·102	·125	·105	·112	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{7}{16}$	$\frac{15}{32}$
„ 8	·102	·115	·105	·113	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{7}{16}$	$\frac{1}{2}$

NOTE.— ·050 inch $= \frac{50}{1000}$ inch, ·092 inch $= \frac{92}{1000}$ inch, &c., &c.



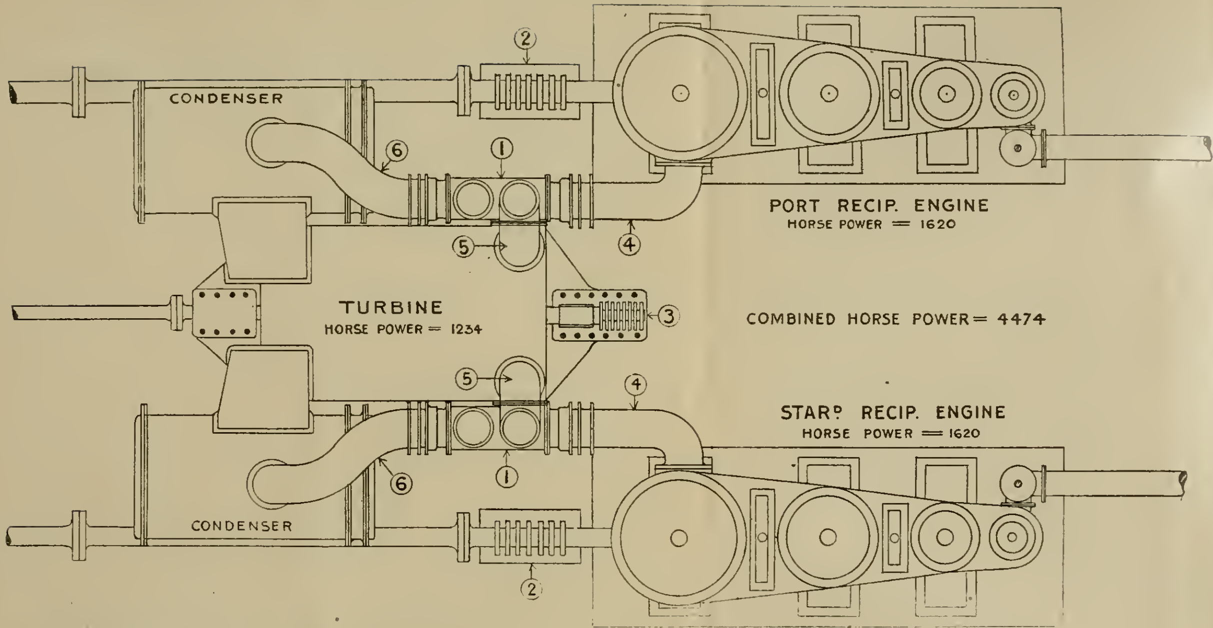
No. 9.—Plan of Turbine Room.



No. 10.—Ahead Type of Dummy (Facial).

1. Rotor Dummy.
2. Casing Dummy.
3. Wheel Drum.
4. Rotor Drum.
5. Undercut Brass Rings bedded into Casing.

The difference in diameter between the rotor dummy and drum, as shown, forms the annulus upon which the steam acts to partly counterbalance the propeller thrust.



No. 11.—Plan of Combined Reciprocating and Turbine Arrangement.

With Theoretical Horse Powers.

(1) Change valve, giving steam either to turbine or condenser direct.
(2) Reciprocating engine thrust block.
(3) Turbine thrust block.
(4) Exhaust from reciprocating engine to turbine or to condenser.
(5) Branch to turbine.
(6) Branch to condenser.

[To face page 655.

Combined Reciprocating Engines and Turbines.—The most recent practice in mercantile steamers is the combination of reciprocating engines and turbines for vessels of moderate or low speeds. Several steamers of this class are at present under construction, and the arrangement consists of two wing triple or quadruple expansion engines exhausting into a central low-pressure ahead turbine, driving a third shaft and propeller, the revolution speed of the centre turbine shaft being much higher than that of the wing shafts. An alternative design is that of two wing low-pressure turbines and one centre reciprocating engine.

The engines are arranged to be run as follows:—

(1) Boiler steam to both H.P. cylinders, and exhaust from these to turbine, the exhaust from the turbine being divided and led into two separate condensers.

(2) Boiler steam to both H.P. cylinders, and exhaust from these to condensers direct, the turbine being then cut off. This is required when running astern as the turbine is for ahead running only, and may be used for ahead running with two propellers only.

(3) Boiler steam to either H.P. cylinder, and exhaust from L.P. to centre turbine, then into one condenser only.

These combinations are obtained by the use of "change valves" fitted on the L.P. exhaust pipes, and by large butterfly valves fitted in the turbine exhaust branches. The change valves admit the reciprocating exhaust steam of either side to the turbine, or to the condenser as required, and the large valves in the turbine exhaust pipe shut off the condenser on either side as may become necessary should one reciprocating engine require to be disconnected through breakdown.

Benefits of the System.—As, broadly speaking, the economy of the reciprocating engine depends chiefly on high-pressure steam, and the turbine on low-pressure steam, the judicious combination of the two ought to result in higher efficiency results.

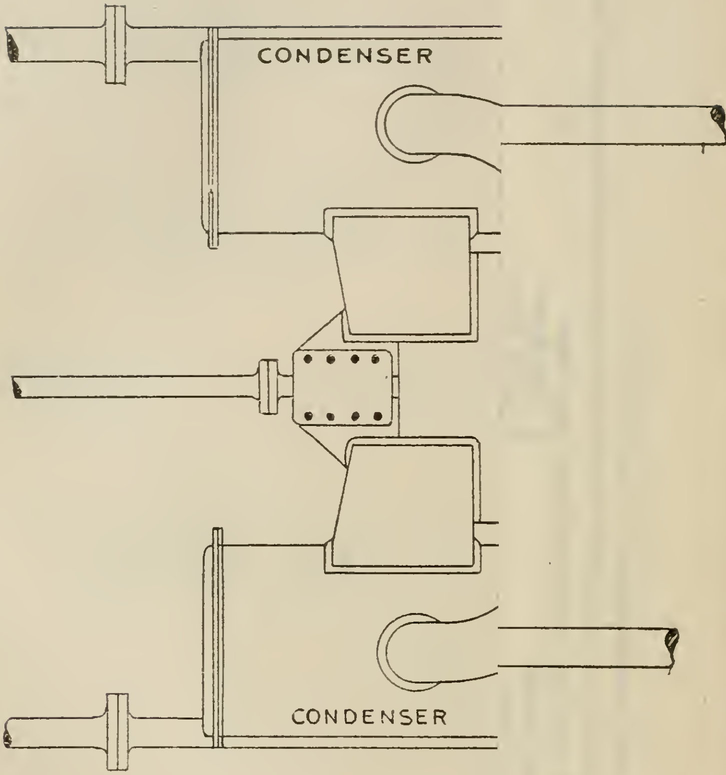
The turbine is therefore most effective in dealing with steam of a pressure which cannot be utilised with benefit in a triple or quadruple expansion engine, owing more particularly to the huge volumes involved, and requiring increase of weight, space, and frictional losses.

The L.P. exhaust pressure to the centre turbine is usually 7 or 12 lbs. absolute. This will produce a difference in the usual L.P. cylinder diagram cards, bringing up the exhaust line to a position much nearer the atmospheric line than usual.

The loss of work energy so represented by the reduced indicator card area in the L.P. engine will be more than balanced by the increase of power developed in the turbine.

The economical result of the combination arrangement is, to all appearance, beyond question, and may in time, with suitable improvements which experience suggests, prove adaptable for the usual tramp steamer speed of from 8 to 10 or 11 knots.

An innovation has been made in the case of the turbine glands,



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To face page 655.

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The economical result of the combination arrangement is, to all appearance, beyond question, and may in time, with suitable improvements which experience suggests, prove adaptable for the usual tramp steamer speed of from 8 to 10 or 11 knots.

An innovation has been made in the case of the turbine glands, which, instead of the frictionless steam packing hitherto adopted, have in one case been changed for the usual marine type of piston-rod gland, consisting of rack and pinion screwing-up gear with soft packing inside, this type of gland being fitted at both ends of the turbine.

Description of the Propelling Machinery of the Q.SS. "Reina Victoria-Eugenia."

Constructed by Messrs Swan, Hunter, & Wigham Richardson Ltd., Newcastle-on-Tyne.

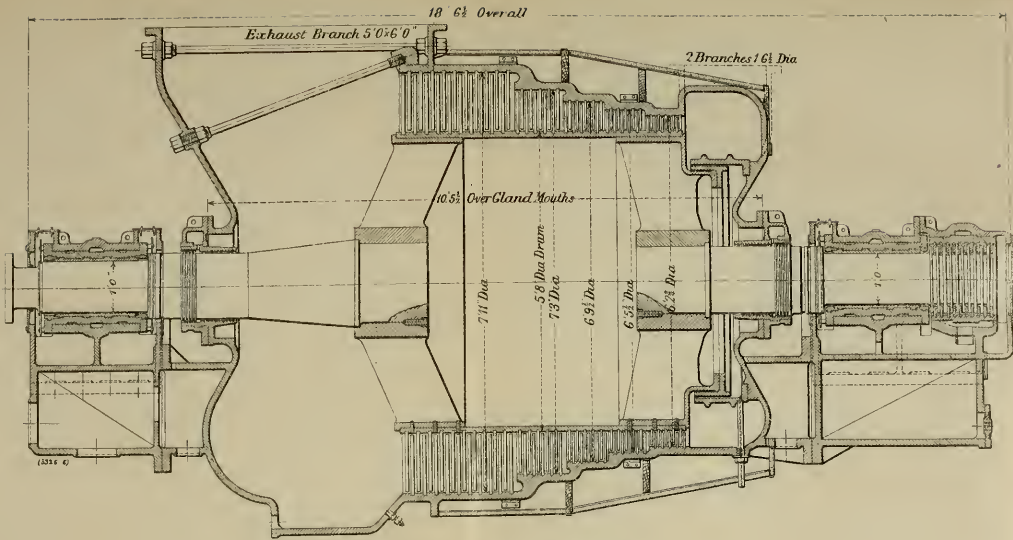
"The propelling machinery consists of two complete units of reciprocating engines and turbines, driving four screws in all.

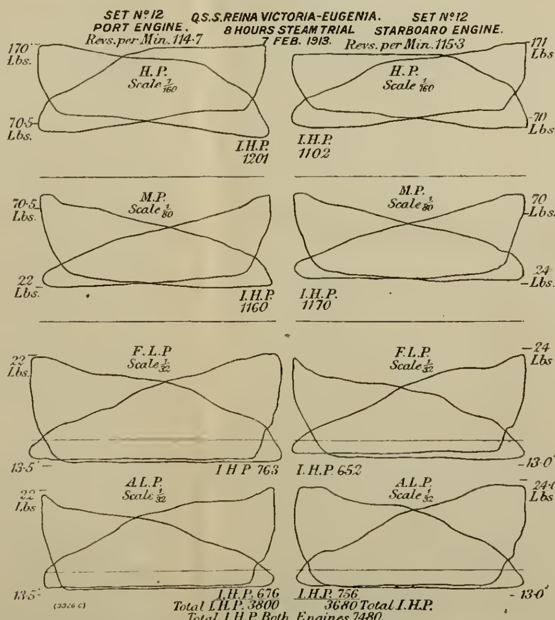
"The reciprocating engines are of the well-known four-crank triple-expansion type, balanced on the Yarrow-Schlick-Tweedy principle, with cylinders 29 in., 43 in., 45 in., and 47 in. in diameter, and a stroke of 42 in. The low-pressure cylinders at each end of the engines are designed to develop less power than the other two cylinders, so that a very uniform turning moment is obtained, in addition to good balancing.

"Each engine exhausts into a steam turbine through special manœuvring-valves mounted at the back of each low-pressure cylinder, and operated by levers on the reversing-shaft. These valves are so arranged that the exhaust steam is passed direct to the condenser when the reversing-gear is in the astern position. Thus all manœuvring is done by the reciprocating engines on the inboard shafts. Provision is made, by means of screw and hand-wheel, whereby the turbines can be cut out entirely by passing the steam directly into the condenser under all conditions.

"The turbine installation comprises two low-pressure Parsons turbines driving the wing-shafts, and designed for working ahead only. Steam enters each turbine at the forward end through two 18½-in. diameter pipes—*i.e.*, one from each low-pressure cylinder—with a suitable strainer in each branch, the admission pressure being about 10 lbs. per square in. absolute, with a maximum steam consumption of 135,000 lbs. per hour. The exhaust branch, having a cross-sectional area of 30 sq. ft., is designed for a condenser pressure of 1 lb. absolute. The rotor drum is parallel, 68 in. in diameter, and carries thirty rows of blades. The dummy at the forward or steam end is fitted with labyrinth-packing strips of the radial type—ten strips in the rotor and ten in the casing. The diameter of the dummy is such that the total axial steam thrust on the rotor substantially balances the propeller thrust. The rotor shaft glands are of Parsons combined labyrinth and ring type, each being fitted at the inner end with ten rows of moving, and ten rows of fixed, rings, and with four Ramsbottom rings at the outer end. The glands are so designed as to be easily overhauled without lifting the turbine-cover. The rotor journals at each end are 12 in. in diameter, working in white-metal bearings, each 23½ in. long. A rotor adjusting-block is fitted at the recorded on a double run was 18·6 knots in shallow water (about 75 ft. deep), as, owing to the foggy weather, the mile-posts could only be seen a small distance from the shore.

"A set of indicator-cards taken during the eight hours' trial is shown, and the results are of special interest in view of the high pro-





No. 12.—Parson's Exhaust Turbine for the Q.SS. "Reina Victoria Eugenia" (and Set of Diagram Cards).

[To face page 657.

forward end for the purpose of adjusting the axial position of the rotor, and contains ten rings, which bear upon the faces of corresponding collars on the rotor shaft. Lubricating oil is supplied under pressure to the bearings and adjusting-blocks by a duplex oil-pump. The oil draining from the bearings, &c., is collected in a tank and cooled before being discharged again to the bearings. One of the turbines is illustrated on opposite page.

"The condensers are of the 'Uniflux' type, and have each a cooling surface of 5100 sq. ft., designed for a vacuum of 28 in. with an average sea temperature of 75°. The 19-in. circulating-pumps supplied by Messrs H. Watson & Co. have proved capable of discharging 8500 gallons per minute against 23-ft head.

"The air-pumps are of Weir's latest 'Dual' type, of 13 in. by 24 in. by 17 in. each. There are further installed two pairs of Weir's feed-pumps, with cylinders 13½ in. by 10 in. by 24 in., two 35-ton evaporators, two 90-in. Howden fans, three 60-kw. dynamos, a 12-ton distiller, a ballast donkey, two general service donkeys, various smaller pumps, a refrigerating plant, and a Clayton fire-extinguishing machine, which has already been illustrated and described in *Engineering*. The exhaust steam from all auxiliaries is utilised for heating up the feed-water, a 'Neptune' surface-heater being installed for this purpose.

"There are seven single-ended boilers of 16 ft. 3 in. outside diameter and 11 ft. 6 in. in length, working on Howden's system of forced draught. Each boiler has three large furnaces, yielding a total grate surface of 480 sq. ft.: the total heating surface is 20,965 sq. ft., and the working pressure 180 lbs. per sq. in.

"On trial the ship half laden was required to steam 17·5 knots for eight consecutive hours, and when fully laden at a speed of 16 knots for twenty-four consecutive hours. The actual results obtained are given in the table on p. 642, and show that on the eight hours' trial a speed of 18·12 knots was obtained in adverse weather, while on the twenty-four hours' trial the speed obtained was 16·10 knots. In good weather conditions and in deep sea, it is certain that higher speeds would have been secured, and we are informed that off Cadiz, in a deep sea and fine weather, the speed obtained has been greatly in excess of that realised at the trials carried out off Newcastle.

"The steam consumption was measured by means of standard nozzles regularly employed by the builders for this purpose; the figures given in the table include make-up feed (about 1·5 per cent.). Progressive runs over the measured mile at St Mary's, on the North-East Coast, were made during each trial in order to determine the speeds obtained on the respective trials. The maximum speed recorded on a double run was 18·6 knots in shallow water (about 75 ft. deep), as, owing to the foggy weather, the mile-posts could only be seen a small distance from the shore.

"A set of indicator-cards taken during the eight hours' trial is shown, and the results are of special interest in view of the high pro-

combined labyrinth and ring type, each being fitted at the inner end with ten rows of moving, and ten rows of fixed, rings, and with four Ramsbottom rings at the outer end. The glands are so designed as to be easily overhauled without lifting the turbine-cover. The rotor journals at each end are 12 in. in diameter, working in white-metal bearings, each $23\frac{1}{2}$ in. long. A rotor adjusting-block is fitted at the forward end for the purpose of adjusting the axial position of the rotor, and contains ten rings, which bear upon the faces of corresponding collars on the rotor shaft. Lubricating oil is supplied under pressure to the bearings and adjusting-blocks by a duplex oil-pump. The oil draining from the bearings, &c., is collected in a tank and cooled before being discharged again to the bearings. One of the turbines is illustrated on opposite page.

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"A set of indicator-cards taken during the eight hours' trial is shown, and the results are of special interest in view of the high pro-

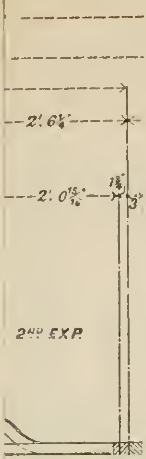
pulsive efficiency indicated by the low consumption of power. It should be mentioned in connection herewith that the experiments made by the builders some years ago with a self-propelled mo el have induced them to choose the four-screw arrangement in preference to the three screws, generally applied with the combined system."

Results of Trials of "Reina Victoria-Eugenia."

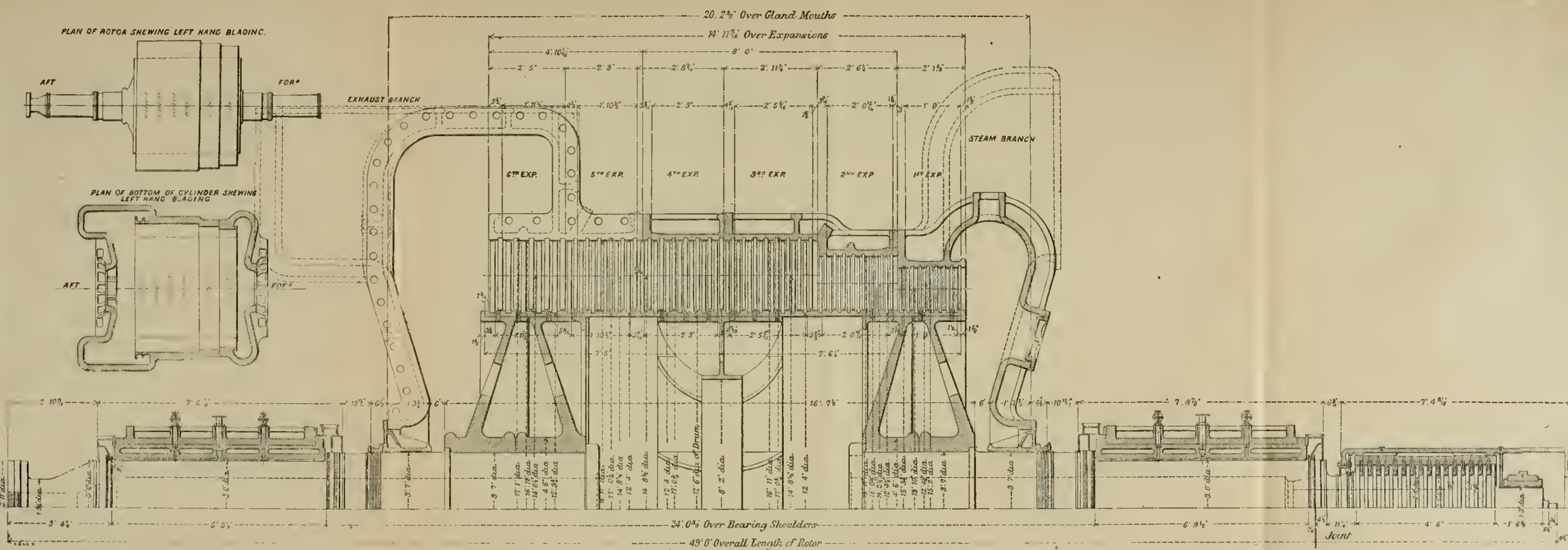
Trial	8 hours, half-loaded	24 hours, fully loaded
Date	February 7, 1913	February 15 and 16, 1913
Mean draught	19 ft. 10 in.	24 ft. $8\frac{1}{2}$ in.
Displacement (moulded)	10,181 tons	13,229 tons
Sea (waves)	0 ft. to 6 ft.	Smooth
Wind	18 knots ahead to 30 knots starboard bow, 30 knots port quarter to 40 knots port beam	Varying between 0 and 10 knots
Barometer	29·7 to 29·5 in.	30·4 in.
Distances of runs in each direction	2 × 71 knots	4 × 89 knots
Depth (average)	130 ft.	150 ft.
Speed	18·12 knots	16·10 knots
Reciprocating engines	112·6 r.p.m.	102·9 r.p.m.
,, ,,	1 per cent. slip	4 per cent. slip
Turbines	481 r.p.m.	395 r.p.m.
,,	20 per cent. slip	13 per cent. slip
Indicated horse-power	7340	5760
Shaft horse-power	3500	2157
Total horse-power	10,840	7910
$\frac{S^3 D^{\frac{2}{3}}}{H.P.}$	257	294
Pressure, H.P. chest	170 lbs. per sq. in.	170·5 lbs. per sq. in.
,, turbine	7·8 lbs. per sq. in. absolute	5·4 lbs. per sq. in. absolute
,, condenser	0·5 lbs. per sq. in. absolute	0·56 lbs. per sq. in. absolute
Temperature, sea	43 deg. Fahr.	45 deg. Fahr.
,, discharge	70 ,,	64 ,,
,, hot-well	70 ,,	62 ,,
,, feed	...	183 ,,
Feed-water consumption for main engines	114,000 lbs. per hour	85,000 lbs. per hour
Feed-water consumption for auxiliaries	...	14,000 ,, ,,
Air pressure in ash-pits	0·4 in. water	0·35 in. water
Coal heating value by calorimeter	14,200 B.Th.U.	14,400 B.Th.U.
Ashes by calorimeter	...	3 per cent.

GEARED DOWN TURBINES.

To allow of combined high turbine speeds and low propeller shaft speeds geared down turbines have recently been introduced. This arrangement admits of economy at low ship speeds, owing to the fact that turbines are most efficient at high revolution speeds, and propellers most efficient at low revolution speeds,



of a crank and connecting rod coupled to the forward end of gear-wheel shaft. The turbine and pinion shaft bearings are under forced lubrication, similar to ordinary turbine practice. The teeth of the



Centre Shaft Exhaust Turbine of White Star Liner "Britannic."

(Reproduced by permission from "Engineering," Feb. 27, 1914)

White Star Liner "Britannic."

(By Messrs Harland & Wolff Ltd.)

RECIPROCATING ENGINE DATA—

The two-wing reciprocating engines are of the four cylinder triple expansion type with two L.P. cylinders placed one at either end, and balanced on the Yarrow-Schlick Tweedy system

Diameter of cylinders	54 in., 84 in., 97 in., 97 in
Stroke	75 in.
Type of valves fitted	Piston valves in all cylinders.
Type of valve gear	Stephenson link motion.
Type of piston rings fitted	Lockwood and Carlisle rings on all cylinder pistons, except H.P., and on all piston valves. The H.P. rings are of the Ramsbottom type.
Diameter of H.P. and I.P. piston rods (high tension steel)	14 in.
Diameter of L.P. piston rods	11¾ in.
„ H.P. and I.P. connecting rods	From 13½ in. to 15 in.
„ L.P. connecting rods	From 11½ in. to 13 in.
Diameter and length of top ends (H.P. and I.P.)	16½ in. by 17¼ in.
„ „ „ (L.P.)	13½ in. by 14¼ in.
Diameter of crank pins	27¾ in. with 9 in. hole.
Length of crank pins (H.P. and I.P.)	35 in.
„ „ (L.P.)	24 in.
Diameter of crank shaft	27 in. with 9 in. hole.
„ thrust shaft	27 in. with 9 in. hole.
Number of collars	14 in
Total thrust surface per block	6860 sq. ft.
Diameter of line shafting	26½ in. with 12 in. hole
„ propeller shaft	28¾ in. with 12 in. hole, reduced to 6 in. hole at after end.

Reciprocating Engine Data—*continued.*

Propellers of Manganese-bronze with cast steel boss—

Diameter of wing propellers	23 ft. 9 in.
Pitch „ „	About 30 ft. 6 in.
Revolutions per minute	77.
I.H.P. of combined wing reciprocating sets	32000.

EXHAUST TURBINE DATA

Initial pressure (L.P. exhaust from wing reciprocating engines)	10 lbs. absolute.
Vacuum (turbine exhaust)	28 in. (with 30 in. barometer).
Over all length of turbine	50 ft.
Weight of rotor complete with blades	150 tons.
Shaft horse power of turbine	18000.
Revolutions	170 per min.
Diameter of rotor drum	12 ft. 6 in.
Length „ „	14 ft. 11½ in.
Diameter of turbine spindles	3 ft. 7 in., reduced to 1 ft. 9½ in. at shaft end, and with hole 23 in. diameter, reduced to 11 in. diameter at shaft end.
Number of collars on thrust (placed forward)	16.
Total thrust surface	5000 sq. ft.
Number of blade expansions	6.
Blade heights (1st expansion)	16 in.
„ „ (2nd „	21½ in.
„ „ (3rd „	26½ in.
„ „ (4th „	26½ „ Semi wing and wing blades.
„ „ (5th „	26½ „
„ „ (6th „	26½ „

Steam glands contain 20 radial or fin rings and 4 Ramsbottom rings.

Size of turbine bearings	3 ft. diameter and 6 ft. 9½ in. in length.

Exhaust Turbine Data—*continued.*

Diameter of turbine tunnel shaft	20½ in. with 10 in. hole.
„ „ propeller shaft	22½ „ „ „ reduced to 5 in. at after end.

Propeller of solid type and of Manganese bronze—

Diameter of propeller	16 ft. 6 in.
Pitch of propeller	15 ft. (about).

Condensers.—Two main condensers and one auxiliary condenser are fitted—

Total cooling surface of main condensers	50000 sq. ft.
Diameter of tubes	¾ in.
Cooling surface of auxiliary condenser	3600 sq. ft.

A "change valve" is fitted and connected to the reversing gear, so that when running ahead the two-wing reciprocating engine exhausts enter the centre turbine as initial steam, and from thence to the two condensers; but when running astern, the reciprocating engine exhausts are changed over and led direct to the condensers, thus cutting out the turbine. This arrangement is required as the turbine is designed for ahead running only. It will thus be obvious that when running astern the turbine will be turning idly, which action results in a slight loss of power due to the "windage" action of the turbine blades when revolving inside the casing.

NOTE.—As proving the efficiency of the turbine to utilise low pressure steam, the student should note that the two-wing reciprocating engine, each taking initial steam at about 200 lbs. gauge pressure, and expanding down to a terminal pressure of 10 lbs. absolute, developes about 16000 I.H.P., whereas the centre turbine taking initial steam at 10 lbs. absolute pressure, and expanding down to a terminal pressure of about 1 lb. absolute, developes about 18000 S.H.P.

It should be noted, however, that the turbine receives to itself the steam of both reciprocating engines combined, and running, as it does, at fully twice the revolution speed of these latter, can dispose of the large steam volumes at low pressure much more conveniently than could be done by ordinary engine cylinders.

[To face page 658.

Arrangement.—In the single propeller shaft arrangement one side H.P. ahead turbine and one side L.P. ahead turbine are connected by two small gear wheels to two large gear wheels secured to the centre driving shaft. The turbines run at about 1200 revolutions per minute and the propeller shaft at 60 revolutions per minute, the gear-down ratio thus being as 20 : 1, because $1200 \div 60 = 20$. In the two propeller shaft arrangement the above system is usually duplicated, two H.P. and two L.P. turbines being fitted and connected up similarly. The turbines and gear wheels are joined up by flexible type couplings. The gear wheels are enclosed in a casing, and an oil service under pressure is sprayed in jets on to the contact surfaces of the wheels. Thrust blocks are fitted near the forward end of the propeller shaft, to take up the thrust, and the turbines are balanced by steam pressure acting on differential type dummy pistons. The helical-toothed gear wheels are very accurately cut out of hard steel by a machine specially designed for the purpose. The astern turbines are arranged as in ordinary turbine practice, being inside the ahead turbine casing, and fitted with separate steam connections, &c.

It may be pointed out that the geared down arrangement can be adapted for either low or high ship speeds, but is not so necessary at high speeds as it is at low speeds. The gearing down allows of speeds of 10 or 12 knots with reasonable economy, whereas at these speeds and direct turbine drives the economy would fall off for the reasons mentioned previously.

The "Vespasian" was built in 1887 by Messrs Short Brothers, of Sunderland. Her dimensions are:—Length on load water line, 275 ft.; breadth, moulded, 38 ft. 9 in.; depth, moulded, 21 ft. 2 in.; mean loaded draught, 19 ft. 8 in., and displacement, 4350 tons. The boilers—two in number—are 13 ft. diameter by 10 ft. 6 in. long, with a total heating surface of 3430 sq. ft., and grate area of 98 sq. ft., working under a pressure of 150 lbs. with natural draught. The propeller is of cast iron, and has four blades, having a diameter of 14 ft., pitch 16·35 ft., and expanded area of 70 sq. ft.

Description of Geared Turbines, SS. "Vepasian."

The propelling machinery consists of two turbines in "series," viz., one high-pressure and one low-pressure, the high-pressure turbine being placed on the starboard side of the vessel and the low-pressure on the port side. At the after end of each of the turbines a driving pinion is connected, with a flexible coupling between the pinion shaft and the turbine, the pinion on each side of the vessel being geared into a wheel, which is coupled to the propeller shaft.

A reversing turbine is incorporated in the exhaust casing of the low-pressure turbine. The air, circulating, feed, and bilge pumps are of the usual design for tramp steamers, and are driven by means of a crank and connecting rod coupled to the forward end of gear-wheel shaft. The turbine and pinion shaft bearings are under forced lubrication, similar to ordinary turbine practice. The teeth of the arrangement admits of economy at low ship speeds, owing to the fact that turbines are most efficient at high revolution speeds, and propellers most efficient at low revolution speeds,

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The high-pressure turbine is 3 ft. maximum diameter by 13 ft. over all length, and the low-pressure 3 ft. 10 in. by 12 ft. 6 in. length. The turbines are similar in design to a land turbine, being balanced for steam thrust only, the propeller thrust being taken up by the ordinary thrust-block of the horse-shoe type which is fitted aft of the gear wheel.

The cooling surface of the condenser is 1165 sq. ft.

The gear wheel is of cast iron, with two forged steel rims shrunk on. The diameter of the wheel is 8 ft. $3\frac{1}{2}$ in. pitch circle, having 398 teeth—double helical—with a circular pitch of ·7854 in. The total width of face of wheel is 24 in.; inclination of teeth 20° to the axis.

The pinion shafts are of chrome nickel steel, 5 in. diameter pitch circle, with 20 teeth ·7854 circular pitch. The ratio of gear is 19·9 to 1.

On the first four voyages careful measurements of water consumption were taken. The following table gives the data and results obtained on these voyages:—

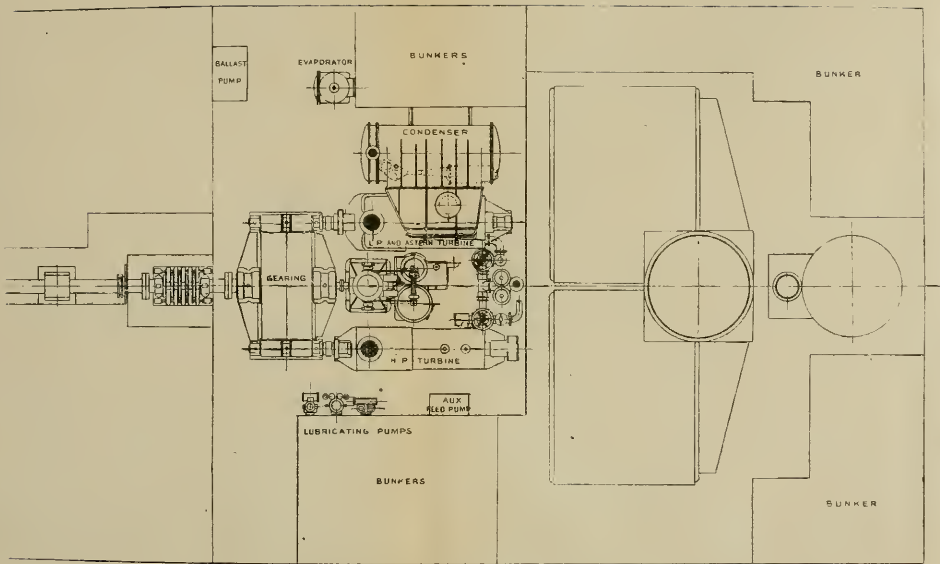
Geared Turbines of SS. "Vespasian."

RESULTS OF FIRST FOUR VOYAGES.

Date	9/6/10	16/6/10	16/6/10	22/6/10	22/6/10	22/6/10	29/6/10
Speed by log, knots	9·35	9·22	10·58	9.61	9·27	10·22	9·37
Revs. per minute	65·0	64·9	73·0	64·8	63·85	70.6	62·9
Boiler pressure, lbs. per square inch	137	135	145	135	135	140	135
High-pressure turbine (initial pressure, lbs. per square inch)	86	86	121	86	86	111	81
Vacuum (in inches)	28·5	29·1	28·6	28·55	28.4	28·4	28·3
Barometer	30·01	30·5	30·52	29·9	29·9	29·88	29·6
Water, main engines lbs. per hour	12,140	12,300	15,680	11,890	11,730	14,510	11,100
Shaft horse-power	740	736	1,080	735	710	960	668
Water consumption, lbs. per S.H.P.	16·4	18·0	14·5	16·2	16·5	15·1	16·6

Geared Turbines of the "King Orry."

The machinery is on the twin-screw system, and includes two high-pressure and two low-pressure turbines of the latest Parsons combined impulse and reaction type. The arrangement of the turbines and gear is shown on the two-page plate. The two high-pressure turbines are in the centre, and the two low-pressure turbines, with



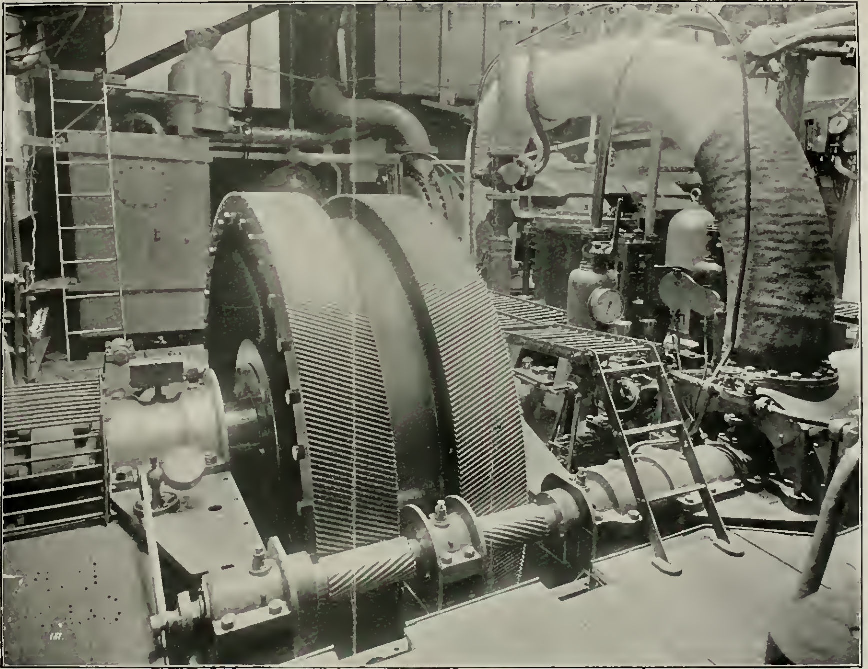
No. 13.—Geared Turbines, S.S. "Vespasian."

Plan of Engine Room and Stoke Hole.

[To face page 660.

being—high-pressure 13·58 to 1 and low-pressure 5·39 to 1. The circular pitch of the teeth is 0·815 in., and the spiral angle 44° 2½′.

being—high-pressure 13·58 to 1 and low-pressure 5·39 to 1. The circular pitch of the teeth is 0·815 in., and the spiral angle $44° 2\frac{1}{2}'$.



No. 14.—Gear Wheels of "Vespasian's" Turbines.
(Cover removed.)

[*To face page* 661.

which the astern turbines are incorporated, in the wings. The conditions of the service demand ample astern power, and, in order to provide for these requirements, this has been arranged to be not less than 60 per cent. of the maximum ahead power.

The high-pressure rotors are solid steel forgings, and bladed on two diameters, 14 in. and 23 in. Three expansions of eleven rows each are fitted on the smaller of these, and three expansions of four rows each on the larger. The mean diameter of the blading ranges from 1 ft. $3\frac{5}{16}$ in. on the high-pressure end to 2 ft. $2\frac{1}{4}$ in. at the other, in each turbine. As it is not possible to utilise the propeller thrust to balance the steam thrust, owing to the presence of the gear-box and its fittings, the whole of the steam thrust in this type of turbine is designed to be taken on the dummies. As the rotor is solid, equalising pipes are led from the end of the third expansion to the forward side of the after dummy, and from the exhaust end to the forward side of the forward dummy, thus keeping the rotor in a practically floating condition axially. A small thrust bearing, as is usual, is fitted at the forward end of each turbine to take the pressure due to any variation in the speed. The arrangement is clearly shown in the illustration.

The low-pressure turbines have six expansions of reaction blading for going ahead, each having three rows of blades. The diameter of the drum is 3 ft. 1 in., and the mean diameter of the blading ranges from 3 ft. $3\frac{3}{16}$ in. to 3 ft. $7\frac{1}{4}$ in. The astern turbines have each three rows of impulse blading, followed by four expansions having four rows each of reaction blading. The mean diameter of the impulse-wheel is 3 ft. 4 in., and in the case of the reaction blading the mean diameter ranges from 2 ft. $1\frac{1}{16}$ in. to 2 ft. $4\frac{1}{4}$ in. on a drum 2 ft. in diameter. The low-pressure rotors are of the more usual hollow type, and therefore do not need the equalising pipes as fitted in the case of the high-pressure rotors.

Each high-pressure turbine is designed to run at 2210 revolutions per minute, and each low-pressure at 1617 revolutions per minute; each is connected up with its pinion shaft through a flexible coupling to correct any small want of perfect alignment. The gearing is of the usual type, in two parts, with oppositely cut helices to neutralise end thrust, and the pinion-shaft bearings are arranged in this case to be of floating type with the object of equalising the pressures between the working teeth, and preventing objectionable noise. The pinion shafts, on which the teeth are cut from the solid, are of special nickel steel, and the wheels into which they gear are of forged steel. All these forgings were made at the Sheffield works of the builders, and the gearing was cut by the Parsons Company. The high-pressure pinions have thirty teeth, the low-pressure pinions forty-one, and the wheels 221, the arrangement thus providing for the essential hunting teeth. The ratios between turbines and propellers being—high-pressure 13·58 to 1 and low-pressure 5·39 to 1. The circular pitch of the teeth is 0·815 in., and the spiral angle $44° 2\frac{1}{2}'$.

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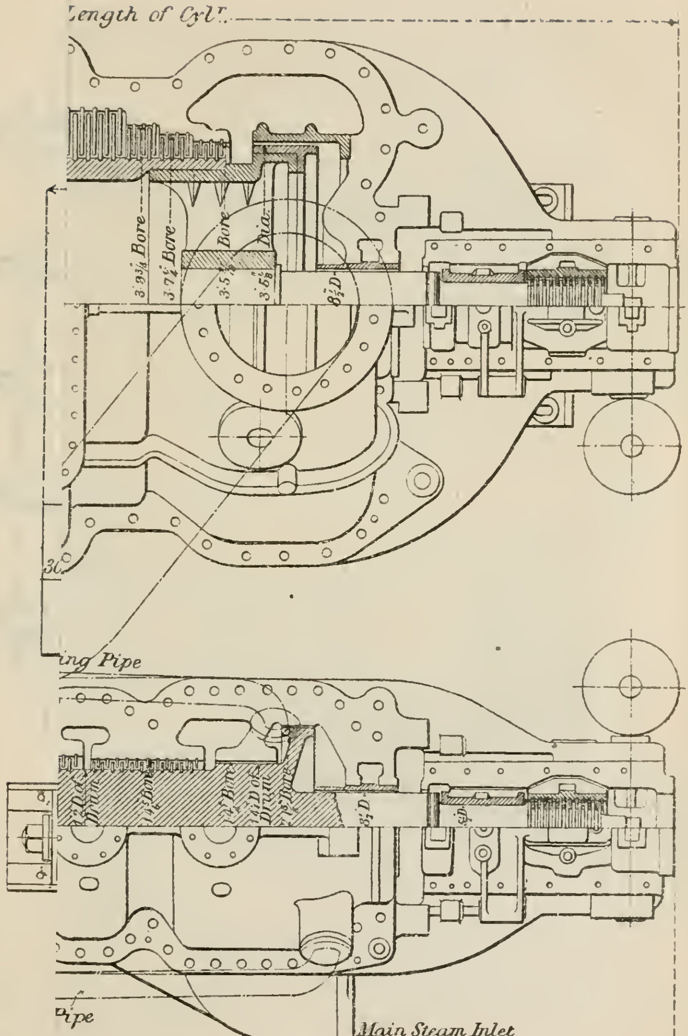
Each pinion-shaft is supported by three bearings of an aggregate length of 3 ft. 0½ in. by 5½ in. in diameter, the width between the teeth at each side of the centre bearing being 15½ in. The usual thrust block for taking the thrust from the propeller is fitted immediately aft of the gear-box, as may be seen from the engravings. The whole of the gear is enclosed in an oil-tight casing, and lubrication arranged for by sprays designed to maintain a film of oil continually between the working surfaces of the teeth. Messrs B. R. Vickers & Sons' patent frictionless stern-glands are fitted to the propeller shafts to ensure that nothing in the way of preventable loss may take place here.

The condensers are of the Weir "Uniflux" type, having steel shells and cast-iron doors, and are guaranteed to maintain a vacuum of 28¼ in. with inlet water at 60° Fahr. Two of Messrs Weir's dual air-pumps are fitted, 24 in. in diameter by 15 in. stroke, and the same makers have supplied the main feed and forced-lubrication pumps. The circulating water for the condensers is supplied by two of Messrs Allen's 18-in. "Conqueror" centrifugal pumps, and they have also supplied the two 6-ft. diameter forced draught fans, and the dynamo and electric-light engine. The vessel is fitted with two bilge and ballast pumps, a water service fresh-water pump, and a sanitary pump. An exhaust feed-water heater is fitted for utilising the exhaust from the auxiliary engines.

There are three boilers—two double-ended and one single-ended, with a total collective heating surface of 14,385 sq. ft., and total grate area of 383 sq. ft. The working pressure is 170 lbs., with a closed stokehold system of forced draught. The removal of the ashes from the stokehold is effected by two "Sentinel" ash-hoists, which are fitted in the usual manner in the ventilators. A vertical auxiliary boiler, by Messrs Cochran, of Annan, is installed to run the electric light engine, &c., when the vessel is in port.

Very careful consideration has been given to the choice of speeds for turbines and propellers, and it is confidently expected that the arrangement will prove to be a highly efficient combination, showing considerable economy in working compared with direct-driven types working in similar vessels.



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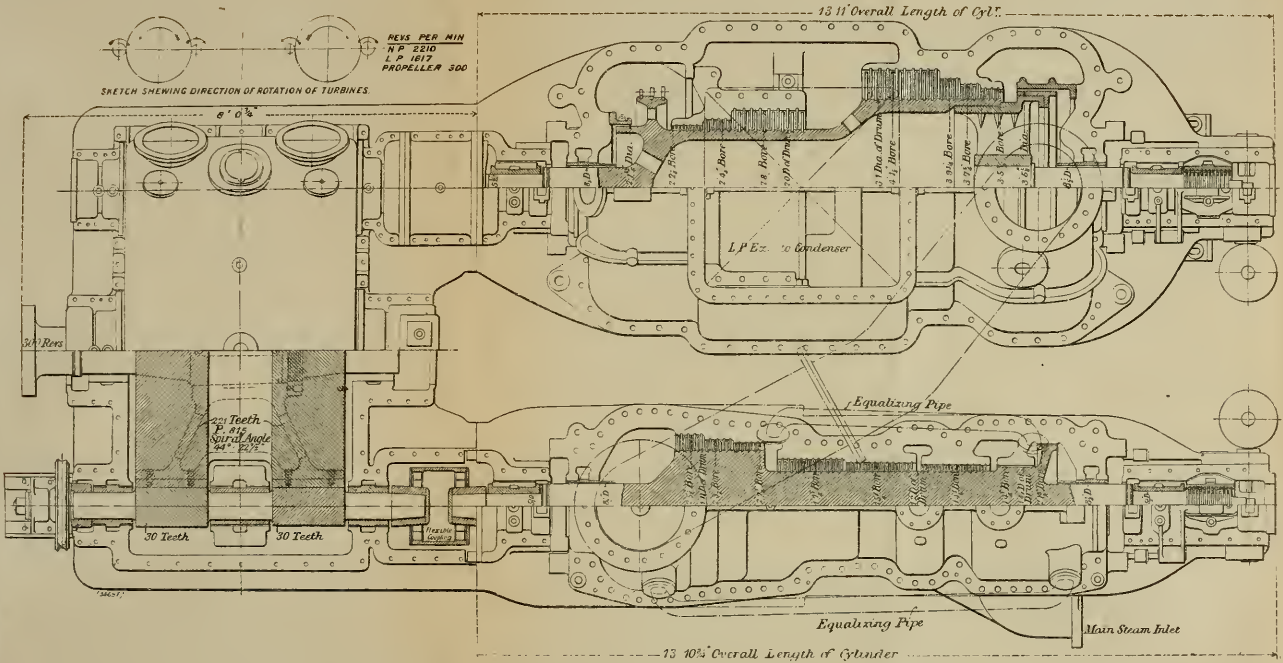

No. 15.—Geared Turbines for the Isle of Man Twin-screw Steamer "King Orry."

Constructed by Messrs Cammell Laird & Co., Limited, Shipbuilders, Birkenhead.

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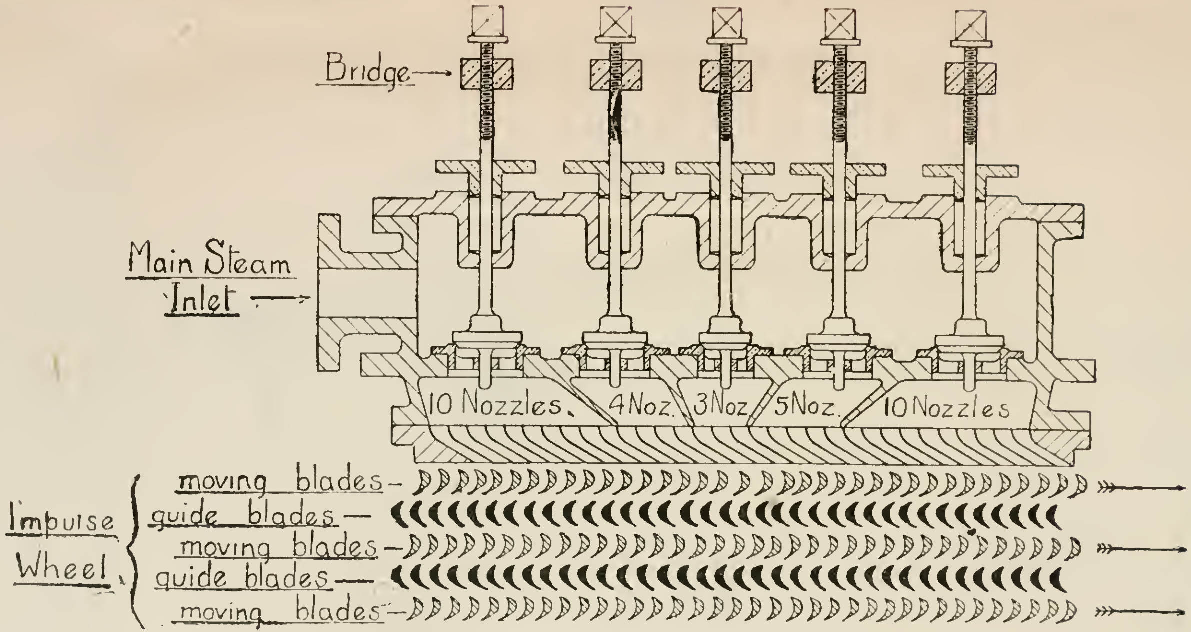
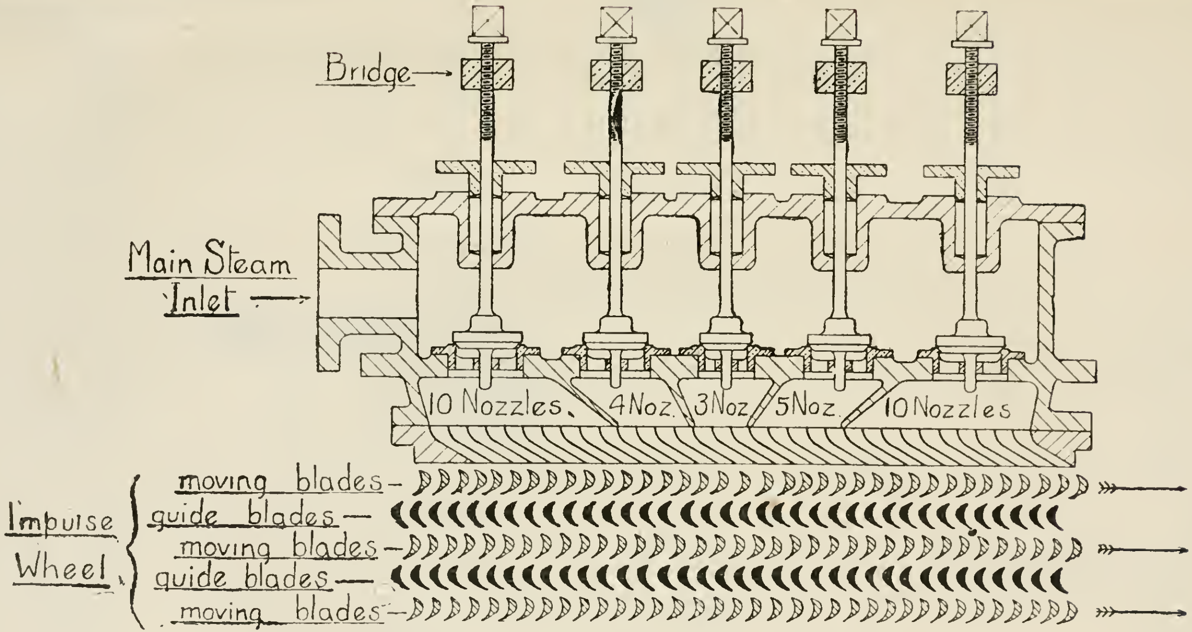


No. 16.—Control Valve Box of Impulse Turbine.

Observe that each separate valve controls a certain number of nozzle openings.

For any given speed or power the minimum number only of nozzle valves should be employed, which, when opened full, give the required steam.



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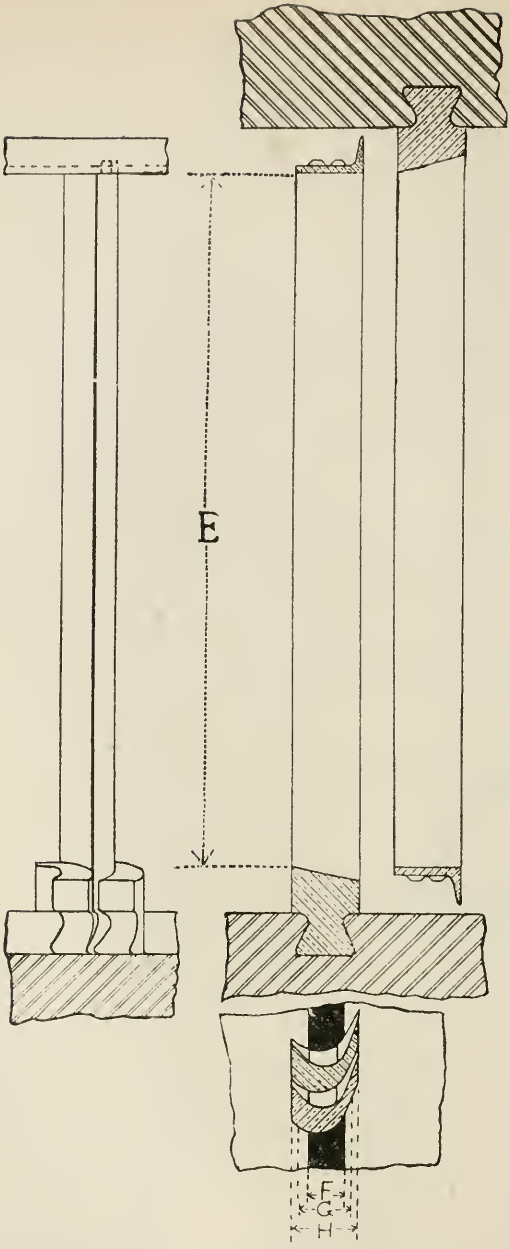
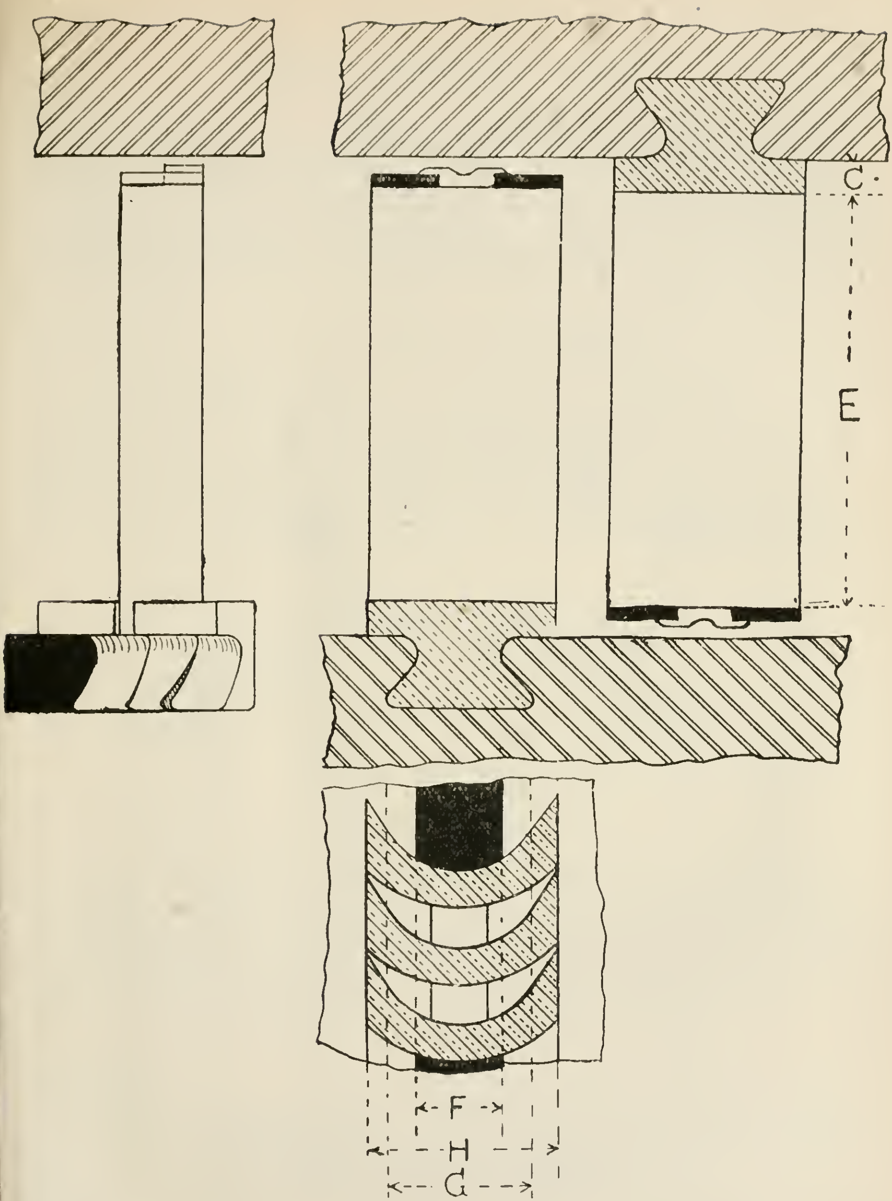


No. 17.—Brown-Curtis Type Dovetail Reaction Blading.
(As fitted on Drum Stage.) Two blades and two packers shown.

E. Effective blade height. **F.** Width of groove opening at surface of drum.
G. Width of groove at bottom. **H.** Width of packers and blades.

Observe that the packers are bevelled away at the admission edges to reduce blade tip leakage by deflecting the steam flow away from the blade tips. The shroud rings of the reaction blading are flanged, and the flanges turned away by machine to an actual "knife edge" sharpness at the tip. If, therefore, the blades come into mechanical contact with the casing or rotor, bending over or grinding away of the flange will, in most cases, be the limit of damage done to the blading. The unequal expansion of the rotor and casing blading not being correctly allowed for in the tip clearance, is a frequent cause of blade fouling.



No. 18.—Brown-Curtis Type Dovetail Impulse Blading.

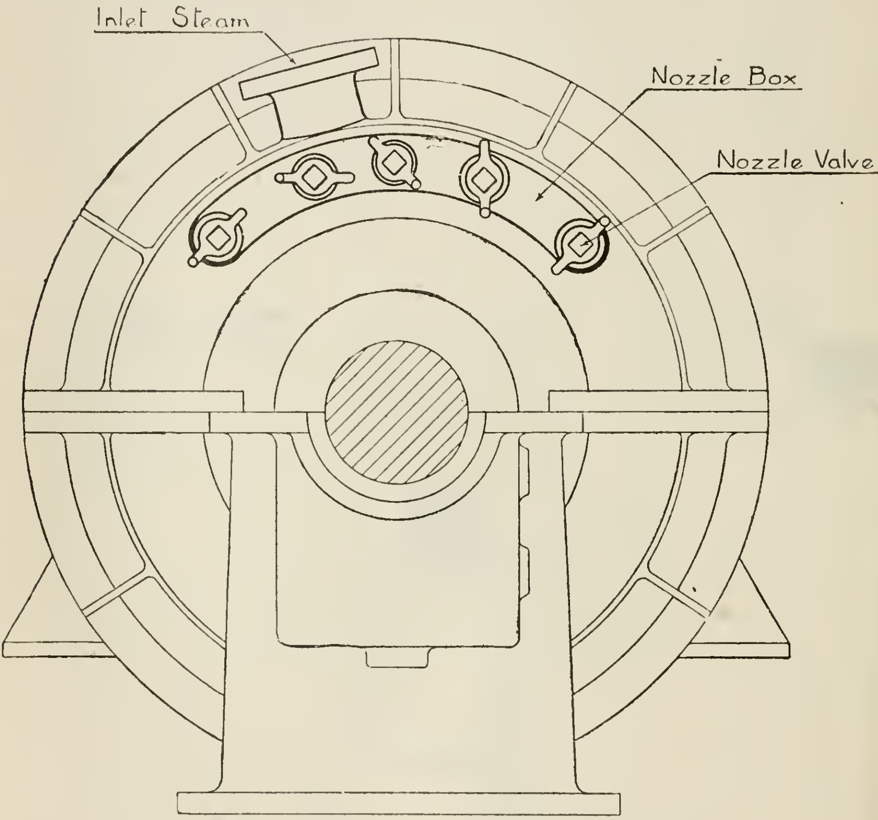
(Two blades and three packers shown in plan.)

C. Depth of packer clear above drum and casing surface, and intended to reduce tip leakage.
E. Effective blade height.
F. Width of groove opening at surface of drum.
G. Width of groove at bottom.
H. Width of packers and blades.

With impulse blading the shroud ring consists of a plain band as shown above, the tip clearance allowed varying from $\frac{1}{16}$ in. to $\frac{1}{8}$ in., or even more.

Combined Impulse and Reaction Blading.

Parson's turbines are now generally fitted at the initial end with a wheel and a set of impulse blading similar to that of the Curtis turbine, the remainder of the blading being of the reaction type. In some cases the astern turbines only are supplied with impulse blading so as to allow of quick picking up of power when running



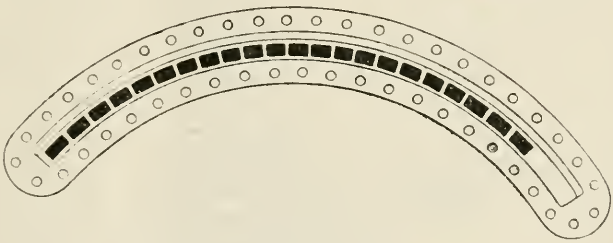
No. 19.—End View of Turbine Casing.
Showing Nozzle Control Valves.

astern. In other cases both reverse and ahead turbines are fitted with impulse blading and wheel at the first stage expansion, the other expansions being of the usual reaction type blading.

Impulse blading has the advantage of allowing the full power to be developed more quickly than is possible with reaction blading alone, a point of considerable value in naval work. The steam is admitted to the impulse wheel blades through expanding nozzles, the quantity being regulated by means of hand valves, each one controlling a certain number of nozzle openings. These area values are marked on the hand wheels of the controlling valves.

Nozzle Box.

The nozzle plate forms a segment only of the turbine casing at the top and does not, it should be noted, extend right round; the wheel is, of course, complete and the impulse blades of the wheel also complete the circle. It should, however, also be noted that the impulse guide blades which are fixed in the casing shell extend round the wheel circle for the **top half only.** The nozzle box is bolted on to the end of the turbine casing, and is fitted with a stop valve which admits steam to the box, also with a number of hand controlling valves which allow the steam to pass from the nozzle box to the nozzles, and so to the impulse blades of the wheel. After expanding through the impulse nozzles and blading, and dropping in pressure, the steam enters the first expansion of the reaction blading, which are very much less in height than those of the impulse wheel. This will easily be understood, when it is remembered that the nozzle openings only extend round the casing for a very small part of the circumference, whereas the reaction blades



No. 20.—Nozzle Openings.

complete the circle, and in this way balance up the area of steam flow at the reduced pressure. For given powers and speeds certain of the control valves are opened, and at full power all may be opened up to give full steam flow through the nozzle openings. Economy is therefore obtained at all powers, as the initial pressure is constant for all nozzle opening areas, whether small or large in number.

Pressures.

After the steam passes through the impulse nozzles and blading a considerable pressure drop takes place, and a corresponding increase in steam velocity, and this change is allowed for in the reaction blade expansions, which are designed accordingly.

Advantage of Impulse Blading.

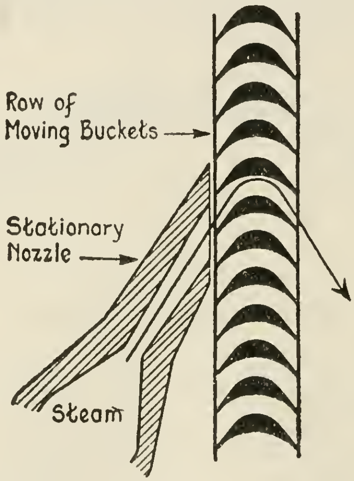
The impulse blading, being much heavier and stronger than the reaction blading, allows of greater steam flow when starting up without risk of damage to blades, and results in a reduced steam pressure admission to the first stage of the reaction blading of the turbine, which is thus less subject to shock or vibration when steam is first turned on for either ahead or astern. It will thus be evident that the maximum steam pressures are exerted on the impulse blading only, the reaction blading receiving the steam at greatly reduced pressure.

CURTIS TURBINES.

THE British Thomson-Houston Co. Ltd., Rugby, own the patent for the Curtis steam turbine, which they have manufactured at their Rugby works for a considerable time, and which they apply to their electric generators. By the courtesy of the British Thomson-Houston Co. Ltd., the author is enabled to reproduce the following notes on turbines.

Impulse Turbine.

An impulse turbine is distinguished by the fact that the potential energy of the steam is transformed into kinetic energy by expansion of the steam through specially shaped stationary nozzles. As a result



No. 21.—Diagram of Simple Impulse Turbine.

of this the steam acquires great velocity, whereupon it is brought in contact with suitable blades or buckets mounted on the periphery of a wheel or wheels free to rotate, the buckets being shaped so as to turn the issuing jet of steam gradually and without shock in a backwards direction. This is shown diagrammatically in No. 21.

The impact of the steam on the buckets results in the steam giving up the whole or part of its velocity and consequent energy to the buckets, thus causing rotation of the wheel.

It is sometimes argued that the velocity of the steam in an impulse turbine is liable to cause damage to the moving buckets through erosion, but this has been proved not necessarily to be the case, as the buckets of Curtis turbines which have been running for some years show on examination only the very slightest traces of wear.

Principle of Impulse Turbine.

A mass which has been moving at a certain velocity will, in coming to rest, give up the same amount of energy as was required to give it the velocity it previously possessed. The object to be attained in the design of an impulse turbine, therefore, is to arrange the buckets so that they will bring the jets of steam issuing from the nozzles to as near rest as possible, that is to say that the steam, after passing through the buckets of the revolving wheel, shall possess no motion and consequently no kinetic energy.

Best Bucket Speed.

It is easy to see that if a bucket is moving as fast as the steam jet it will offer no opposition to the jet, nor will it extract any of the velocity.

If, on the other hand, the wheel is so secured as to be immovable, the steam jet will be directed backwards and will rebound from the bucket with the same velocity with which it entered it, neglecting friction in the bucket.

If, for the sake of example, however, the jet is assumed to be travelling at the rate of 1000 ft. per second, and the bucket at 500 ft. per second, then the steam will strike the bucket with a relative velocity of 500 ft. per second, and on issuing from the bucket will be directed backwards with a velocity relatively to the bucket of 500 ft. per second.

Since, however, the bucket is moving forward at the rate of 500 ft. per second, the backward velocity of the jet is equal and opposite to the forward velocity of the bucket, so that the resultant velocity of the steam relatively to a fixed point in space, as it emerges from the bucket, is zero. The emerging steam accordingly is inert, having given up all its kinetic energy to the bucket.

It follows from this that for the best efficiency the buckets should move at half the velocity of the steam jet.

As a matter of fact, steam when expanded from 150 lbs. pressure per square inch to atmospheric pressure attains a speed of 2950 ft. per second, while, if expanded into a 28 in. vacuum, it can attain a velocity of 4010 ft. per second, or nearly twice the speed of a modern rifle bullet, so that it is evidently impracticable to construct a wheel which can run at half the velocity of such a jet.

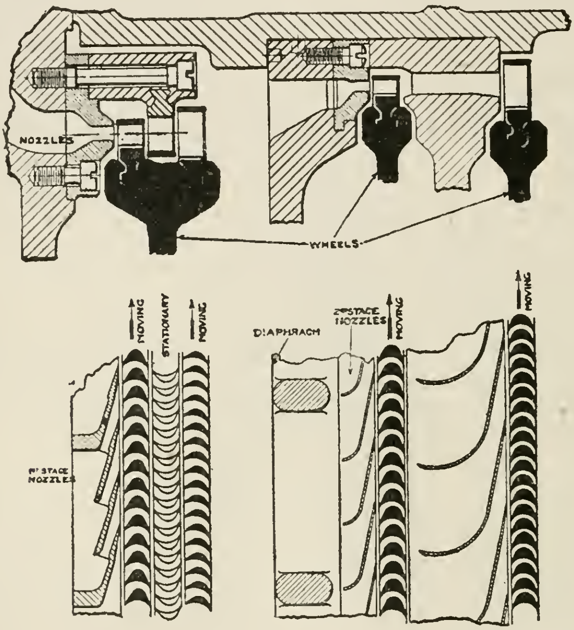
Curtis Turbine.

The Curtis turbine, while utilising the principle of expanding the steam through specially-shaped nozzles, has the great advantage that it can be run with high efficiency at a relatively low speed. This is accomplished in two ways, by means of—(1) "velocity stages," and (2) "pressure stages."

Velocity Stages.

If a turbine such as just considered be run comparatively slowly, the buckets on the rotating wheel will absorb less of the velocity of the steam jet, but the steam emerging from the buckets will still have considerable velocity and will be available for use over again.

If, therefore, the steam on emerging is redirected to a second row of moving buckets fixed to the same wheel as the first row, it may be



No. 22.—Arrangement of Moving and Stationary Elements in a Curtis Turbine.

compelled to part with its remaining velocity in the second row of moving buckets.

This method of fractionally extracting the velocity of the steam, and known as a "velocity stage," is used to a differing extent in the design of both Compound Impulse, and Combined Impulse, Curtis turbines.

Steam entering the diverging nozzles (No. 22) at working pressure is expanded during its passage through them, and issuing at great velocity enters the first row of moving buckets. The steam leaves these buckets in a backwards direction and enters the ring of stationary buckets, in which it has its direction reversed, so that it enters the second ring of moving buckets in the same direction as at first.

The nozzles are designed so that practically all the potential energy of the steam is converted into kinetic energy in the jet. In other words the steam in passing through the nozzles is expanded down to the pressure of the chamber or stage in which the wheels are revolving, so that after leaving the orifice of the nozzle there is practically no tendency for any further expansion, the steam acting on the buckets in that stage simply by impact.

As a result of this action there is no appreciable difference of pressure between the points of entrance and exit of the various rows of buckets in a stage, and consequently no end thrust nor tendency for the steam to leak across the clearance space, so that the moving buckets of the Curtis turbine need no special minimum clearance from the casing.

In the above respects the Curtis turbine shows the most marked advantage over the reaction machine, as no compensating devices for end thrust are required, and large bucket clearances are utilised without loss of efficiency.

Pressure Stages.

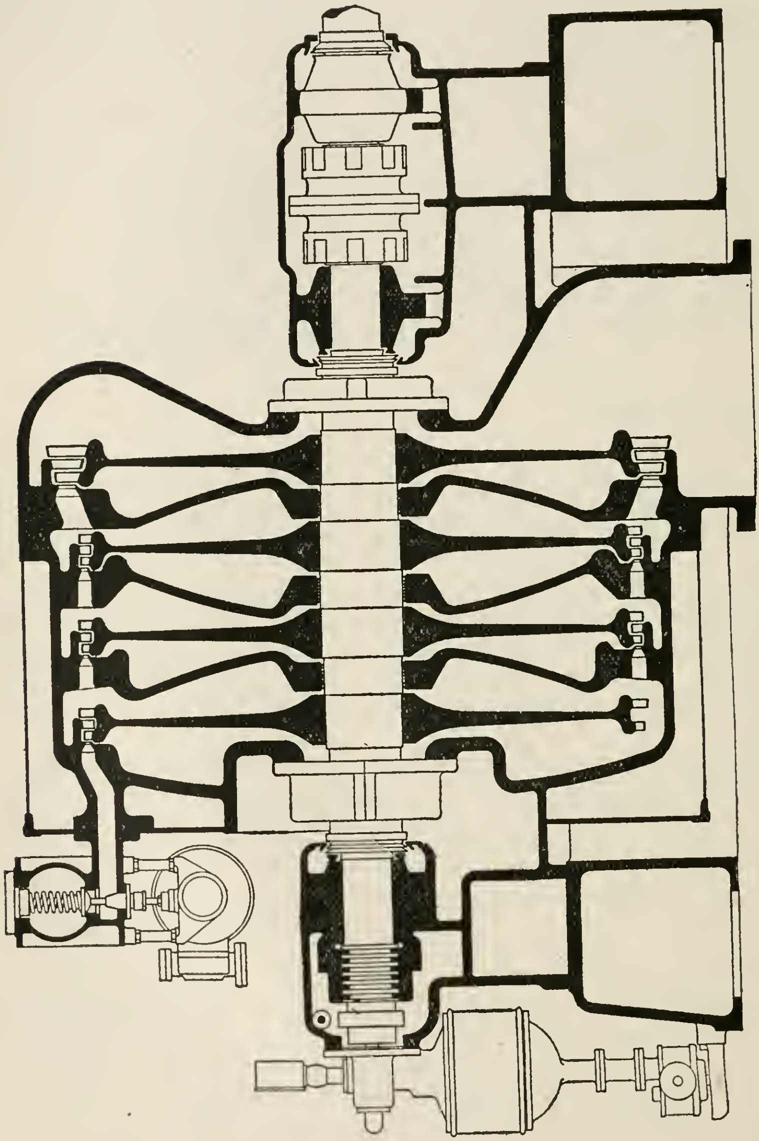
It is obvious from the description just given that any degree of expansion, with its resulting steam velocity, can be dealt with in one pressure stage by providing a sufficient number of rows of buckets or "velocity stages" to bring the speed of rotation down to a practicable limit.

This method, however, if pushed to an extreme, becomes inefficient, due to steam friction losses, and so another method of subdividing the steam energy is also utilised in the Curtis turbine. Instead of expanding the steam from boiler to exhaust pressure in one step, this operation is divided over two or more sections or "pressure stages."

The multi-stage machine can be best understood by imagining a complete turbine to consist of a number of smaller turbines placed in series. The distribution of steam pressure is regulated by the size of the exhaust opening in each section, which opening usually forms the nozzles for the succeeding section. No. 23 represents a four-stage turbine, and clearly shows how the machine is divided up into four sections or stages by means of steam-tight diaphragms, which also carry the nozzles.

Each set of nozzles is designed to utilise one quarter of the total energy in the steam during expansion from the boiler down to the exhaust pressure. This being so, a relatively low velocity is imparted to the steam jet in each stage, which in turn permits of a comparatively low speed of rotation of the wheels.

The steam, after leaving the last row of moving buckets in the first stage, has had nearly all of its velocity extracted, but on passing through the nozzles in the diaphragm separating the first and second stages it is again expanded and the same velocity as before imparted to it. This occurs again in the third and fourth stages, the steam finally emerging from the latter at the pressure of the exhaust, having given up practically all the energy it previously possessed



No. 23.—Longitudinal Section of 4-Stage Curtis Compound Impulse Turbine.

when in the boiler. It will be noted that the nozzles and buckets of the successive stageshave to be made gradually larger to deal with the increase in volume of steam resulting from each expansion.

The simplest form of multi-stage machine is found in the simple Impulse Turbine, which consists of a number of pressure stages having but one row of moving buckets in each stage.

Curtis Compound Impulse and Combined Impulse Turbines.

To obtain the highest efficiency necessitates the proper blending of "velocity steps" (number of rows of buckets per stage) with "pressure steps" (number of stages.) These conditions have received the most careful consideration in the design of the Curtis machine, hence the adoption by the British Thomson-Houston Co. of the two main types of turbines, viz., the Compound Impulse and the Combined Impulse machines. The choice of type employed in any instance is determined by the conditions to be met.

The difference between the two lies in the fact that in the Compound Impulse turbine there are two or more rows of moving buckets in each stage of the machine, while in the Combined Impulse turbine there are two rows of moving buckets in the first stage, but only one row in each of the following stages.

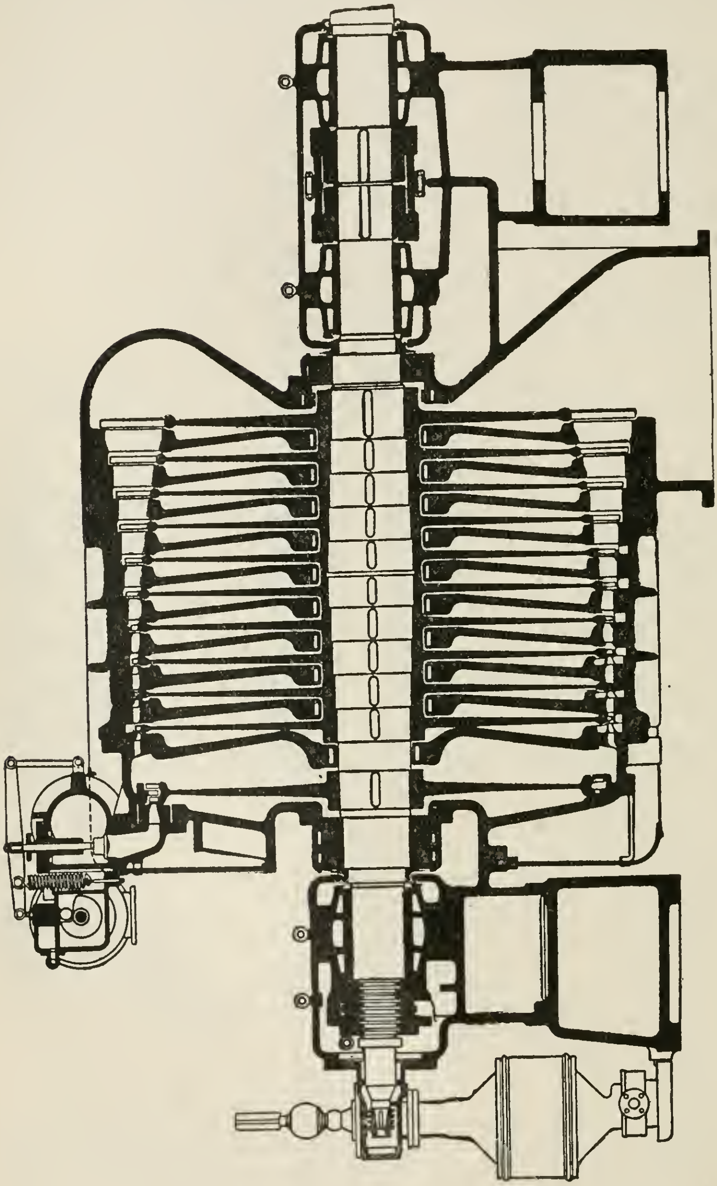
The Compound Impulse turbine, as a consequence, is a type possessing few stages and is very short in length. The Combined Impulse turbine, requiring as it does a greater number of stages than the Compound Impulse machine, is longer over all. In the larger sizes, under certain steam conditions, the Combined Impulse turbine offers better economy than the Compound Impulse machine.

In the Compound Impulse turbine in the first stage the pressure drop is much greater in proportion to the kinetic energy developed than in the second stage, whilst in the third and fourth stages thy pressure drop decreases, although producing the same kinetic energe as in the second stage. In the fifth stage, whilst the energy liberated is nearly as great as for the three previous stages, the pressure drop here is insignificant. This shows indirectly how an increase in vacuum (*i.e.*, a drop in back pressure) adds so rapidly to the amount of energy available per lb. of steam.

In the Combined Impulse turbine the pressure drop in the first stage is also greater in proportion to the kinetic energy liberated than in the succeeding stages; but the kinetic energy liberated in the first stage is very much greater than in the stages which follow. In fact this is the peculiarity of the Combined Impulse turbine, and under certain conditions it is conducive to economy.

Construction.

In No. 24 is shown a longitudinal section of a large high pressure 11-stage Curtis Combined Impulse turbine. The steam, passing through the valve chest shown at the top left-hand side, proceeds down the passage to the nozzle plate, and expanding through the nozzles enters the moving and fixed rows of buckets of the first stage. This stage comprises a single wheel having two rows of buckets with a row of fixed intermediate blades between. From the first stage the steam passes through the expanding nozzles in the



No. 24.—Longitudinal Section of 11-Stage Curtis High Pressure Combined Impulse Turbine.

dish-shaped diaphragm and enters the buckets of the second stage. From the second stage the steam passes to the nozzles and buckets of the third stage, and so on through the succeeding stages, finally exhausting through the opening at the bottom right-hand side of the casing. The second and successive stages all contain wheels with a single row of buckets.

A great advantage of the Curtis turbine lies in the fact that, after expansion through the nozzles at the high pressure or governing end of the turbine adjacent to the main bearing, the temperature and the pressure of the steam are very much reduced.

As a result the steam sealing glands, the end cover, and the main bearing at the high pressure end of the turbine are not subject to the excessive temperatures and pressures found in turbines of other types.

Some Advantages of B.T.H. Curtis Turbines.

A feature of the greatest importance in connection with the Curtis, as compared with the reaction turbine, is that the buckets at the inlet end are not subjected to the effects of steam at high temperature. In expanding through the first set of nozzles the steam temperature falls considerably, so that no risk of damage to the buckets arises, even when using steam superheated to the highest degree.

One of the most important advantages of the Curtis turbine is that in the event of a quantity of water coming over from the boilers with the steam, there is little danger of stripping the blades or other damage.

Curtis turbines can be started up promptly from a cold condition without danger from accumulation of condensed steam, or fear of distortion due to change of temperature of the casing.

As compared with the reaction turbine, the Curtis machine shows to great advantage in the matter of temperature fluctuation.

It is not always possible to keep the steam pressure and superheat constant, and when these fluctuations occur they are, in the case of reaction machines, transmitted direct to the turbine.

These changes, however, are of small importance in the Curtis turbine, owing to the expansion of the steam in the first set of nozzles, in which the fluctuations of pressure and temperature are damped down before the steam enters the turbine proper. Thus there is not the same tendency for mechanical distortion in the Curtis as is present in the reaction machine.

In the Combined Impulse turbine, the first wheel with a double-row of moving buckets takes the place of the first three wheels in a simple impulse turbine, and the advantages gained are as follows :—

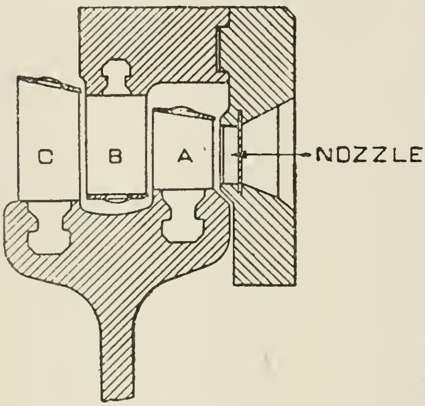
(1) The density of the steam surrounding the single compound wheel is about two-thirds of the mean density of the steam surrounding the three single-row wheels, causing the compound wheel to have only about one-third the rotation losses of the latter wheels.

(2) The lower stage pressure associated with the single compound wheel causes less leakage of steam through the diaphragm and end packing. This leakage is further reduced by the fact that, as compared with the simple impulse machine, the Curtis turbine requires fewer stages and so is shorter in length. This

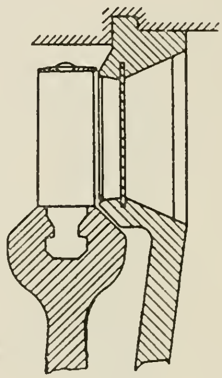
enables a smaller shaft diameter to be used than is possible with a longer machine, and so offers a considerably reduced area for steam to leak through the diaphragm packing.

(3) It is possible in the Combined Impulse turbine to obtain overloads without by-passing the first stage. In a Simple Impulse Turbine, unless the energy drop through the first wheel is very large, it is not possible to obtain overloads without by-passing, so that whether the method adopted is to allow an abnormal energy drop or to by-pass the steam, the result is loss of efficiency.

The design of the Curtis turbine allows ample room for substantial wheel bosses, as well as for efficient diaphragm packings. A sound



No. 25.



No. 26.

Relation of Fixed and Moving Buckets in Curtis Turbines.

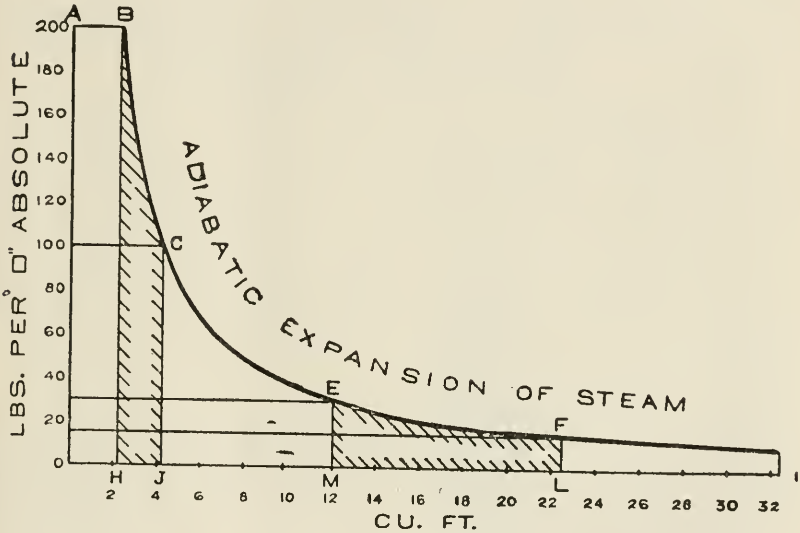
mechanical construction is therefore possible, (1) entirely obviating danger from the bucket wheels opening at the boss and becoming slack upon the shaft due to centrifugal force, and (2) effectually preventing leakage of steam from one stage to another.

Provision for Decreased Velocity of Steam.

It will be seen in No. 22 and No. 24 that the successive rows of buckets in any given stage differ in length one from another. No. 25 shows diagramatically and on a larger scale the relation of moving and stationary buckets in any one stage of a Curtis compound impulse turbine. This figure also represents the first stage of a combined impulse turbine, while the succeeding stages are as shown in No. 26. Referring to No. 25, A and C are the moving and B the stationary lines of buckets respectively. It might be assumed that the divergence of the bucket area as shown is to provide for expansion, and consequently increasing volume per unit weight of steam. This, however, is not exactly the case.

Steam enters A at a higher velocity than it leaves C. During its passage a continuous fall in velocity takes place, so that an increasingly wider passage and consequently longer buckets are necessary to accommodate the steam as it gradually moves more slowly.

This action may be compared with the flow of a non-expansive fluid such as water through a funnel-shaped tube. If the tube be filled with water moving from the smaller to the larger end, the



No. 27.—Diagram showing Adiabatic Expansion of Steam.

velocity will be high at the entrance, but relatively low at the discharge end. Thus the velocity of the liquid drops considerably, but without expansion or increase in volume.

Expansion of Steam.

It is a fact not generally realised that the energy liberated by the expansion of a given weight of dry steam depends upon the number of times that the original volume is increased, or upon the ratio of the pressure drop, rather than the actual pressure drop. For example, a pressure drop from 30 lbs. down to 15 lbs. absolute, liberates very nearly the same amount of energy as a pressure drop from 200 lbs. to 100 lbs. The pressure ratio is the same in both cases, but the pressure difference is very much greater at the higher pressures.

In No. 27 the curve represents diagrammatically the adiabatic expansion* of 1 lb. of dry steam from 200 lbs. down to 10 lbs. absolute,

* Expansion without loss or gain of heat to or from surrounding bodies.

and is limited only by lack of space from continuing the process further. The steam pressure is measured vertically, and the volume horizontally from the point O, so that the area of the diagram (which is the product of its length and average height) is proportional to the energy developed, this being the product of pressure by change of volume.

The volume occupied by one pound of dry steam at 200 lbs. absolute pressure is 2·288 cubic feet, which is represented by the position of the letter H in the diagram. If now the steam is expanded down to 100 lbs. (C) the volume will be increased to the point J, *i.e.*, by 84 per cent., and the amount of energy liberated, assuming the back pressure to be zero, will be denoted by the shaded area BCJH.

In the same way the area EFLM shows the work done by the expansion of the steam from 30 to 15 lbs. This area is less than the area BCJH, but is not nearly as much reduced as the pressure drop has been reduced. The fall of pressure in the first case is 100 lbs., whereas in the latter case it is only 15 lbs. The process of halving, the pressures, etc., can still be continued, and it can be shown that the adiabatic expansion of 1 lb. of steam from 2 lbs. to about 0·64 lb. absolute, *i.e.*, from a vacuum of approximately 26″ to one of 28·7″ of mercury (with barometer at 30″), liberates just as much energy as the expansion of the same weight of steam from 200 lbs. to 100 lbs. absolute.

As a matter of fact, in the steam turbine expansion takes place in accordance with a curve which lies slightly higher than the adiabatic curve, so that there is actually more energy liberated in the lower expansions than is shown in No. 27. The point at issue, however, will be clear, namely, that a very large amount of energy is available in steam at low pressures.

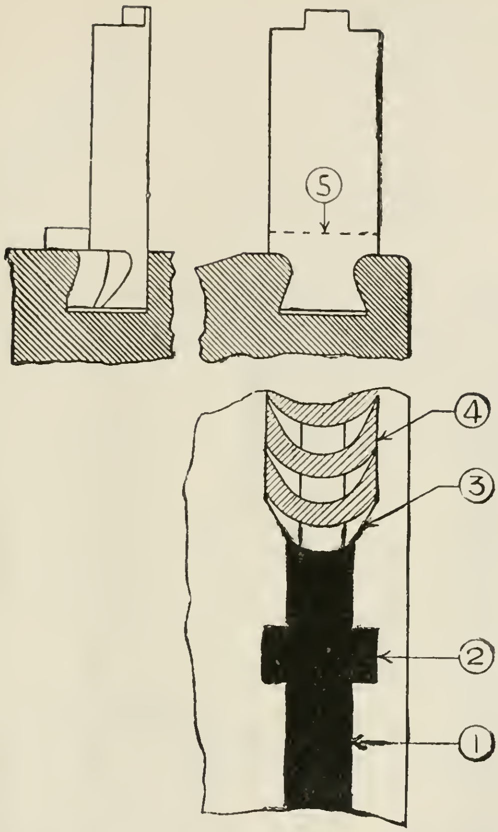
Unsuitability of the Reciprocating Engine for Low Pressures.

The volumes of steam corresponding to the low pressures just mentioned are very great, and for this reason reciprocating engines cannot be adapted to accommodate them. The mechanical difficulties presented by the undue size of the low pressure cylinders and valve gear necessary preclude any advantage being realised with a vacuum higher than 26″.

Suitability of the Turbine for Low Steam Pressures.

Turbines, however, due to their inherent design and the absence of slide valves and ports, together with the fact that in them the steam is constantly in motion, can be designed to accommodate very large volumes of steam at the lowest pressures, without entailing any great increase in cost.

For this reason turbines can be used to great advantage for running on the exhaust steam from reciprocating engines or other steam using devices, and extracting thereby a very considerable amount of the energy in the steam by expanding it down to the lowest practicable limit.



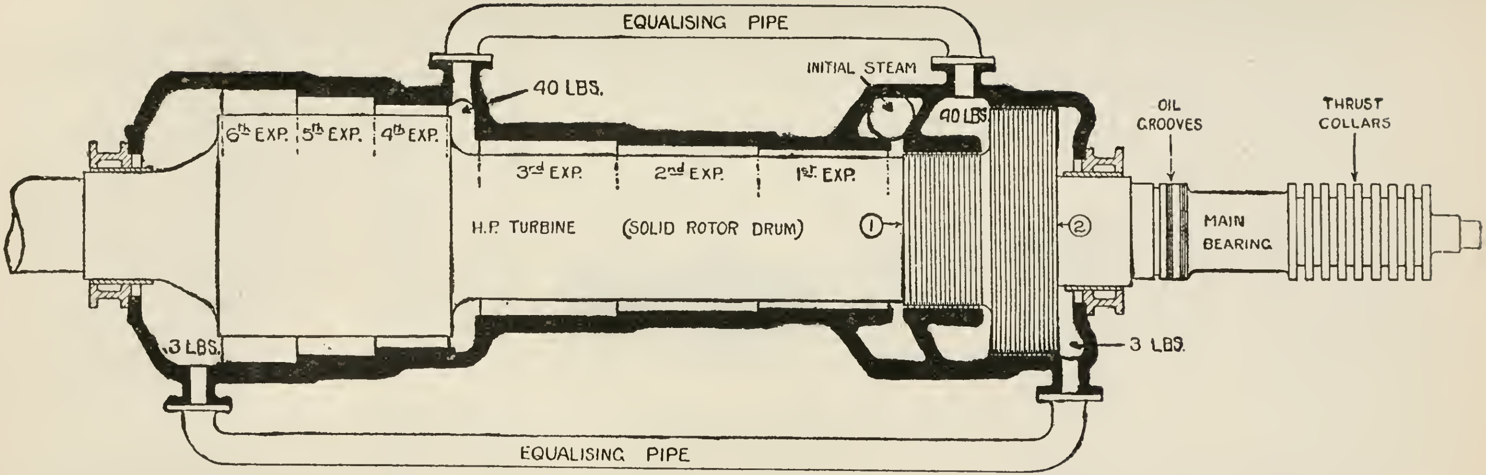
No. 28.—Blades and Caulking Groove.

(Curtis Turbine.)

1. Groove in rim of impulse wheel.
2. Opening for insertion of blade packing pieces.
3. Blade (with undercut root).
4. Packer.
5. Top edge of packer.

The types of blades shown are fixed by means of "circumferential" caulking in place of "radial" caulking, and the enlarged opening (2) is closed up after the blades and packers are all assembled by a closing key or caulking piece which is hammered down and locked in place.

Note.—The blades can be entered in the grooves by turning them sideways, but the packers can only be entered by means of the specially cut opening (2).

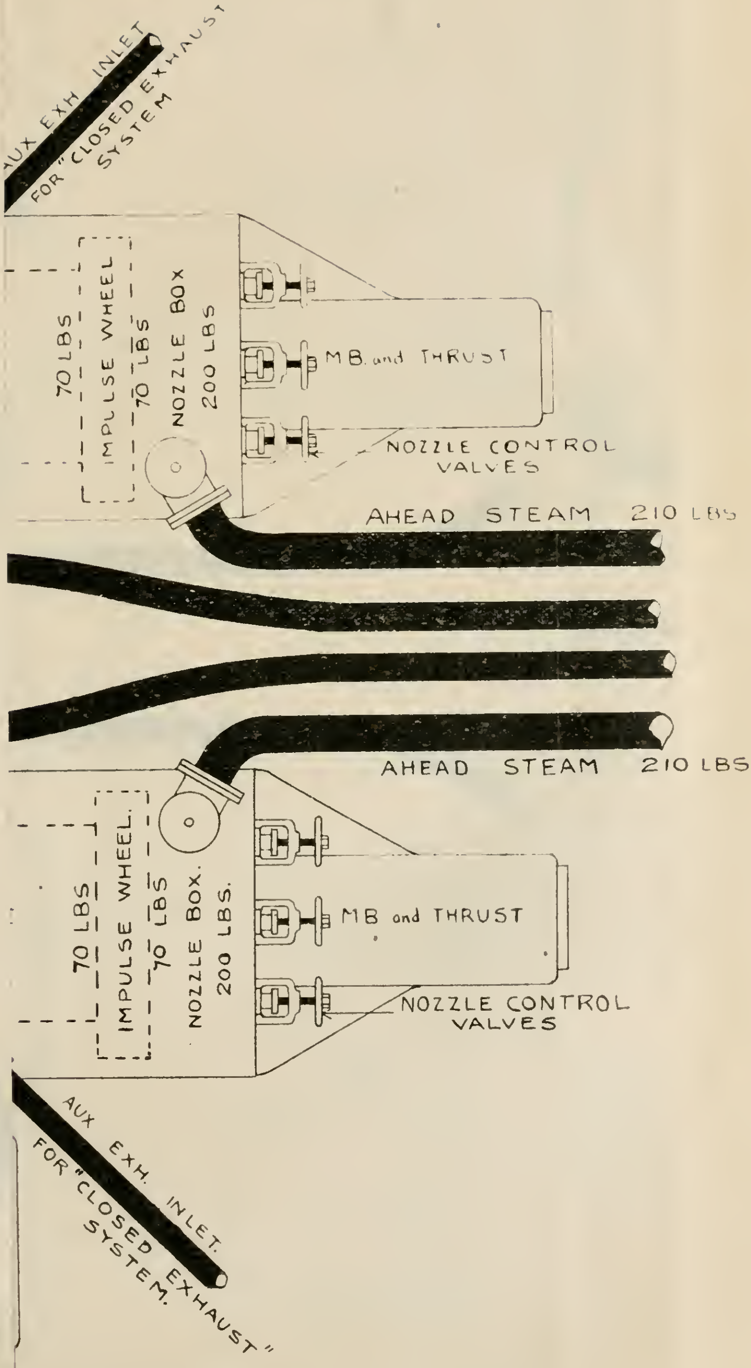


No. 29.—Balance Pistons and Equalising Pipes.

High Pressure Turbine of Geared-Down Set.

1. After Dummy. **2. Forward Dummy.**

The rotor is of solid construction, and is fitted with two balance dummies placed at the forward end. The steam thrust aft is counterbalanced by means of the two pressure equalising pipes, one of which balances the pressure on the first three expansions of the smaller diameter of the rotor, and the other the pressure on the last three expansions of the larger diameter of rotor.



Reaction Turbines.

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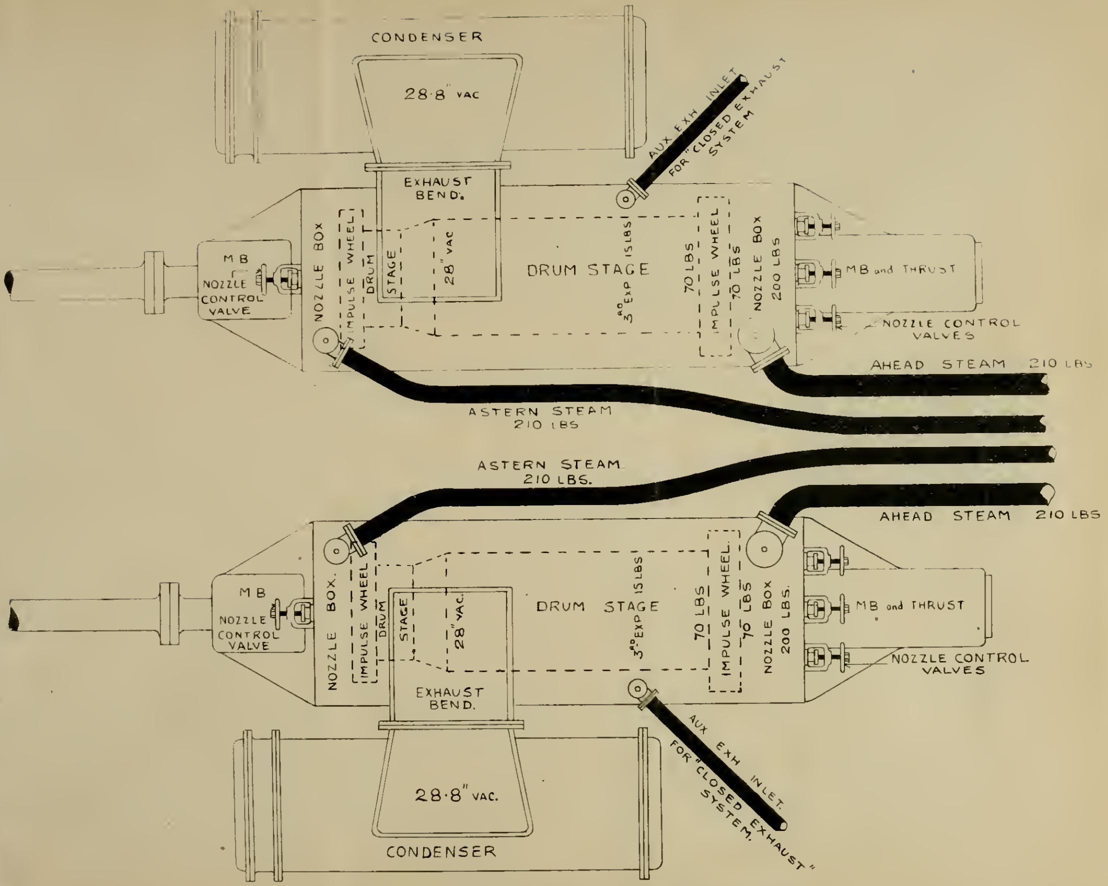

No. 30.—Parson's Type Combined Impulse and Reaction Turbines.
With Average Pressures from Practice.

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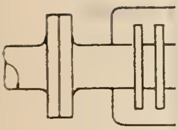

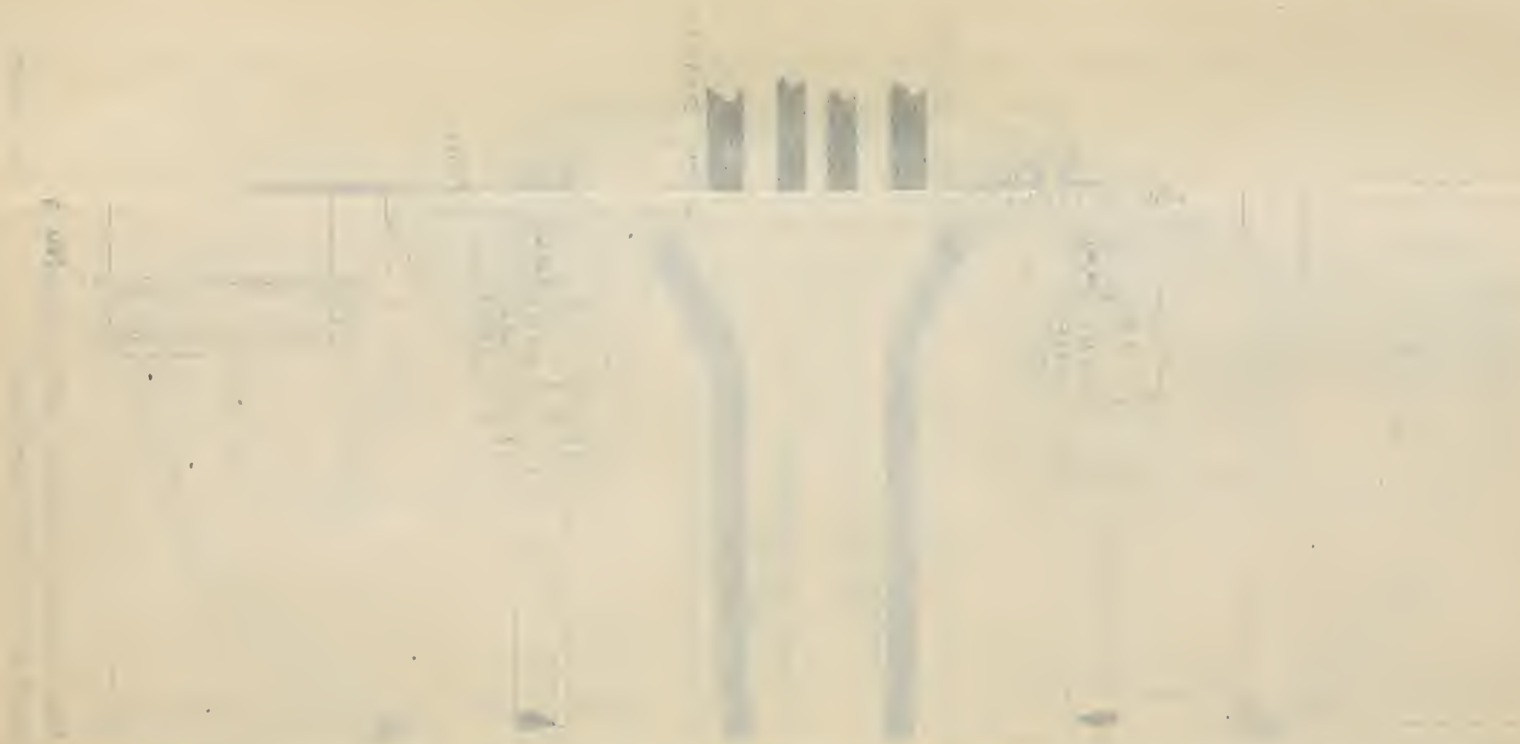

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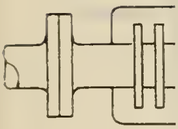

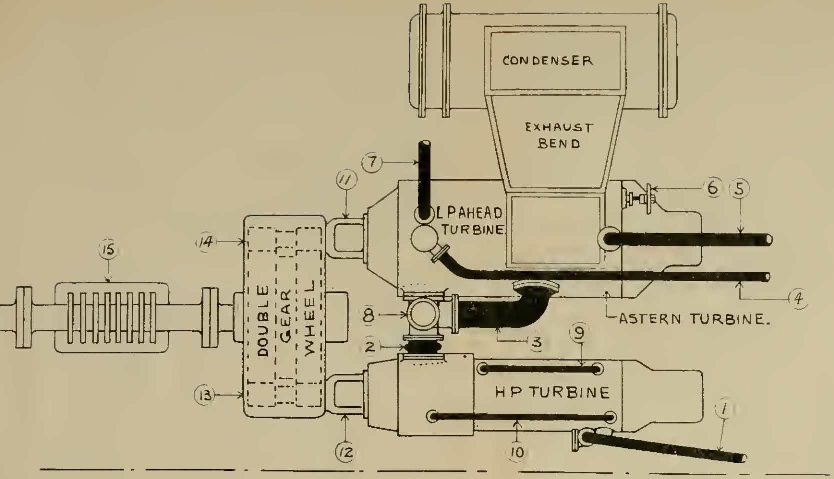

No. 31.—Port Turbines and Gear Wheels, T.S.S. "Tuscania."

1. H.P. Turbine steam.
2. H.P. exhaust to L.P. through strainer.
3. H.P. exhaust direct to condenser.
4. L.P. ahead steam.
5. L.P. astern steam.
6. Control valve to astern impulse nozzles.
7. Auxiliary exhaust steam.
8. Strainer.
9. Pressure equalisating pipe.
10. " " "
11. Flexible coupling.
12. " "
13. H.P. turbine pinion wheel.
14. L.P. " " "
15. Thrust block.

Note.—**Astern Turbine.**—The nozzle control valve shown is only intended to be used in cases of emergency, as the main manœuvring valves give steam direct to the stern turbine for about half the astern power available, but if extra power for astern running is required, the nozzle control valve can also be opened by hand, and this admits an additional head of steam to the astern blading for the maximum astern power available.

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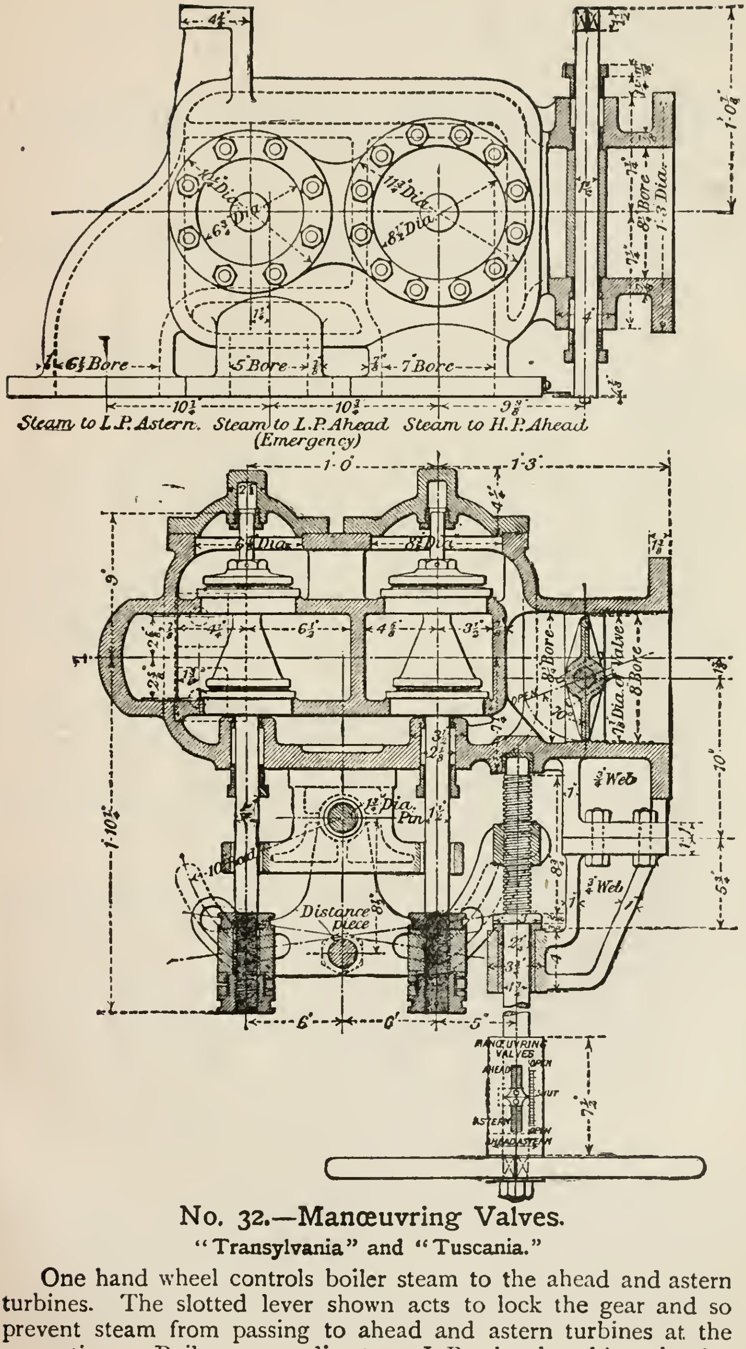
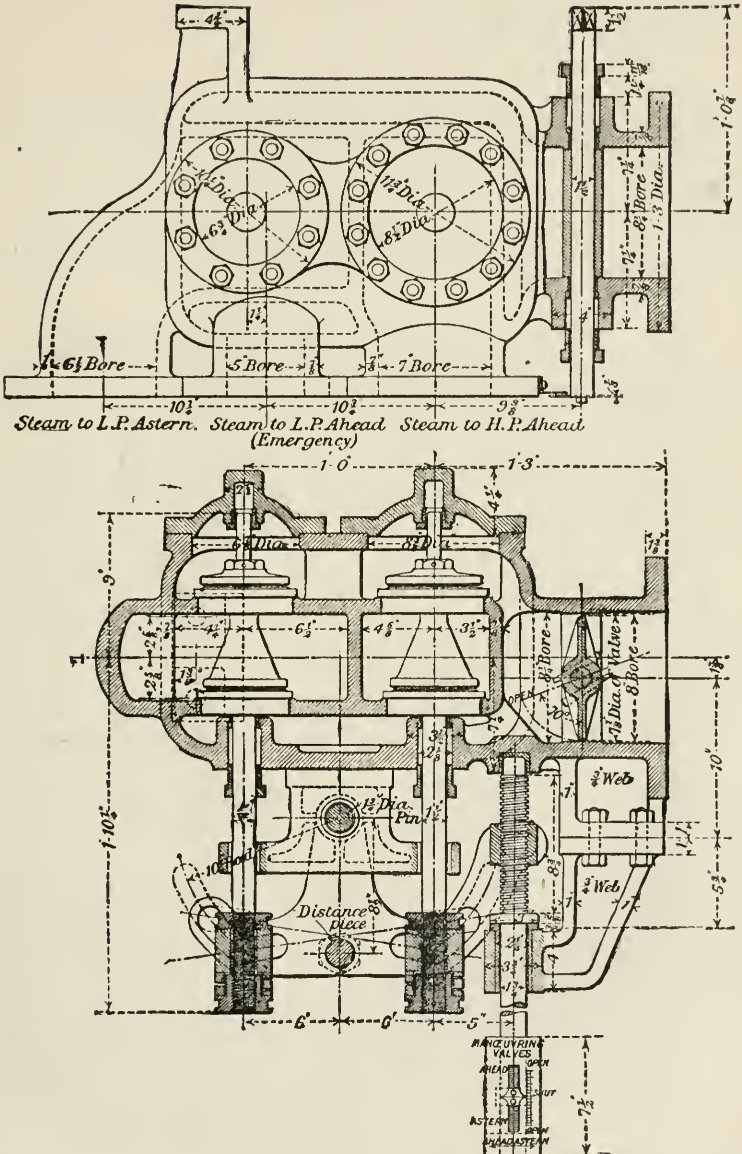


No. 32.—Manœuvring Valves.

"Transylvania" and "Tuscania."

One hand wheel controls boiler steam to the ahead and astern turbines. The slotted lever shown acts to lock the gear and so prevent steam from passing to ahead and astern turbines at the same time. Boiler steam direct to L.P. ahead turbines is also arranged for as shown. A throttle valve connected up to the turbine governors is fitted in the inlet branch of the chest.



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Practical Operation of Turbine Machinery

Heating up.

The matter of warming up the turbines previous to starting is of vital importance, and should receive special attention from the engineers in charge.

1. Start running of the circulators and air pumps, and admit heating steam at about 10 lbs. pressure right through the turbines to the condensers, after first getting rid of the heated air from the boilers, etc.

2. Uniform heating of all parts should be aimed at, and during this process (for which about six hours should be allowed) the rotor should be moved round slowly by the turning gear for quarter turn every thirty minutes.

3. During heating up the various clearances should be carefully read and noted down. The turbine drains should be opened to get rid of condensed water, which, if allowed to remain, might result in blade stripping when starting up.

4. The pipe lines ahead and astern should also be heated up gradually by opening up the steam admission valves of same.

5. The forced lubrication system should be tested by running the oil to flood up the bearings, and then trying the test cocks on the main bearings for oil flow through each.

6. *Note.*—In some steamers a rather ingenious tell-tale arrangement is fitted which indicates when the oil flow is checked in the turbine bearings. The action is as follows:—The oil flow from the turbine bearings passes through a small float tank in which a metal float moves vertically on a small spindle. When the oil flow falls below a certain level, the float in dropping makes contact with a metal stop below, and by means of a bell battery and connection starts the ringing of a small alarm bell, thus indicating to the engineer on watch that the oil service is at fault.

7. Steam should also be turned on to the steam glands until the required pressure shows on the gauges.

8. In testing the turbines give steam to ahead for a few revolutions, then shut off and give steam to astern for a few revolutions.

9. Previous to reporting "ready" take dummy clearance.

10. Whenever possible work up to full power by degrees only.

11. When running up to speed required set gland steam and drains, the latter at minimum opening, as any excess opening means leakage.

12. Previous to getting under way start up oil pumps and obtain required pressure of oil (this varies from 15 lbs. to 30 lbs.). Examine sight glass for oil flow through bearings. (The sight glasses mentioned, unfortunately, soon become dirty and unreliable.)

13. When running astern be careful to take dummy clearance and make record of same. The dummy clearance is affected by the following :—

(*a*) Unequal expansion of rotor and casing.

(*b*) Amount of rotor "float" allowed for oil film at thrust.

(*c*) Effect of thrust pressure forward and steam pressure aft, and which varies under different speed or power conditions.

14. When running at any required speed or revolutions, it is best to open the minimum number of nozzle valves which will give sufficient steam for the purpose at the maximum boiler pressure.

It is bad practice to open more valves than actually necessary, and then reduce the pressure by throttling to keep down the speed or power to the required figure.

15. In changing speed or reversing it is advisable to work up or work down to the power and speed required gradually, so that the risk of accident due to sudden expansion or to sudden contraction of the rotor or casing may be eliminated.

16. For quick emergency stopping of the turbines when running ahead, steam can be admitted to the reverse end.

17. For any given number of nozzle valves open, the main steam, vacuum, and gland pressures should be noted. This will afford a guide to the operation of the turbines.

18. To allow for quick emergency changes from ahead to astern the dummy clearances should not be kept too fine.

19. All impulse nozzles in use are best to be kept full open, and all not in use tight shut.

20. A record should be kept of the revolution speed obtained by various nozzle openings. This is important.

21. With Curtis turbines the gland steam seals are worked at a pressure of about 3 lbs. Take accurate records of the oil temperatures when entering and leaving the main bearings, the maximum of which should not exceed about 130° Fahr.

22. The main bearing at initial end of turbine requires most attention, as the highest temperature exists there.

23. The importance of the gland steam seal on the L.P. and astern turbines should not be overlooked.

24. The efficiency of turbine machinery depends principally on the following :—

(*a*) High pressure at initial end.

(*b*) Vacuum in condenser end, and in idle turbines.

(*c*) Fine blade tip clearance.

(*d*) Tightness of glands against steam leakage out, and against air leakage in.

(*e*) Closed exhaust system.

It should be noted that leaky L.P. glands allow the admission of air, and therefore produce a reduced vacuum in the condenser.

25. When standing by after stopping, stop oil pumps, slow down circulating pump, and open heating up steam to turbines, to prevent cooling down and contraction of the working parts ; also open drains from turbines.

26. Running ahead under closed exhaust system the auxiliary exhausts are usually led to the third expansion of the turbine, but when running astern or manœuvring the auxiliary exhausts require to be led to the condenser.

27. Turbine oil only is suitable for forced lubrication, as under pressure ordinary engine oil tends to form an emulsion.

28. The steam strainers require to be examined and cleaned at regular intervals.

29. The oil cooling service pipes, strainers, and other connections require regular attention.

30. After lifting turbine top half casings for examination, cleaning, or repair, before replacing same make careful examination for any articles left behind, such as chisels, hammers, loose blades, or packers, also turn rotor round slowly and examine closely for loose binding wires on the blading. The neglect of this has in many cases led to damage due to blade stripping, etc., when under way again, as in this respect turbines are by no means "fool-proof."

31. A record should be kept of the blade tip clearances, the blade side clearances, and, most necessary of all, of the dummy clearance, taken "cold," "heating up," and when "running" ahead and astern

32. Blade stripping is often caused by water: it is therefore important that any tendency towards boiler priming should be carefully guarded against. Defective drainage will also act in the same way.

The Weir "Dual" Air Pump.

THE duty of an air pump is to take from the condenser a mixture of air, water, and vapour; and to obtain economical working, this should be done at the highest possible temperature. If a single pump is used to handle this mixture, the hotwell temperature for a given vacuum is dependent on the amount of air leakage and the air pump capacity. Consequently, with a single air pump the temperature of the mixture must be a considerable degree colder than the theoretical temperature due to the vacuum.

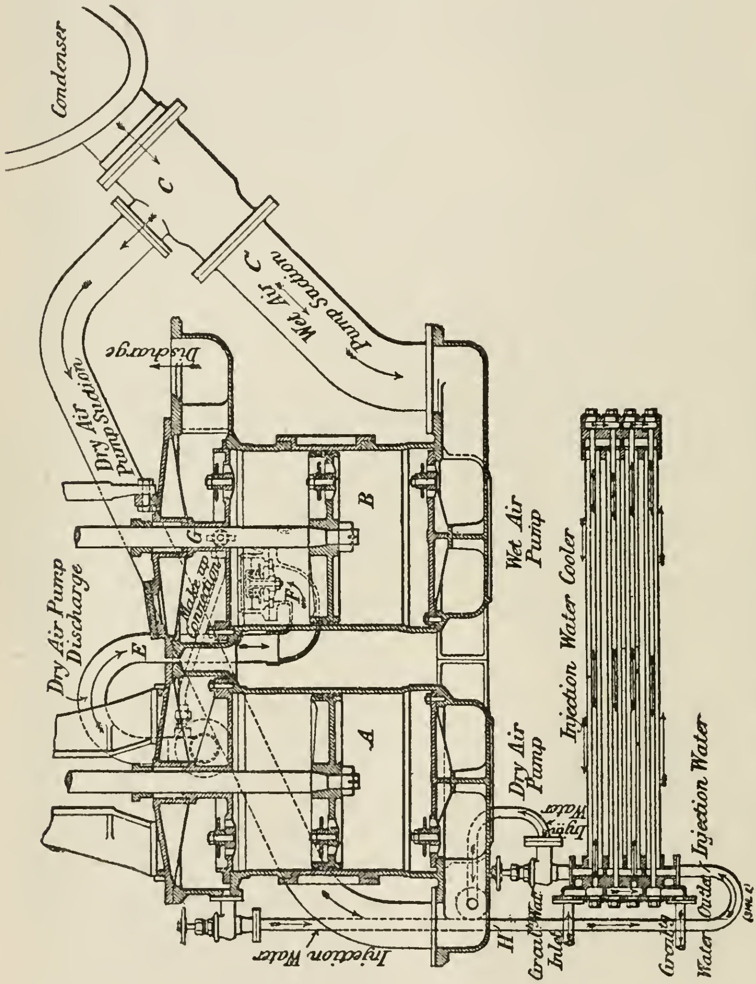
By the use of separate pumps handling the air and water, the dry pump having a cold injection non-returnable to the feed, the above conditions are changed, and higher thermal efficiency is rendered possible.

This method enables the wet (or water) pump to handle water at approximately the steam temperature, while at the same time the dry pump will deal with the air and vapour at the volume and temperature conditions imposed by the temperature of the injection water.

With the "Dual" pump the above advantages are obtained, together with increased efficiency of the dry pump, due to its contents being densified or decreased in volume through cooling by the injection water, which circulates continuously, never leaves the system, and is never subjected to atmospheric pressure with consequent aeration. The dry air pump further works only at less than half the pressure range, as it discharges below the head valves of the wet pump. The apparatus, moreover, is compact, self-contained, and of good mechanical design.

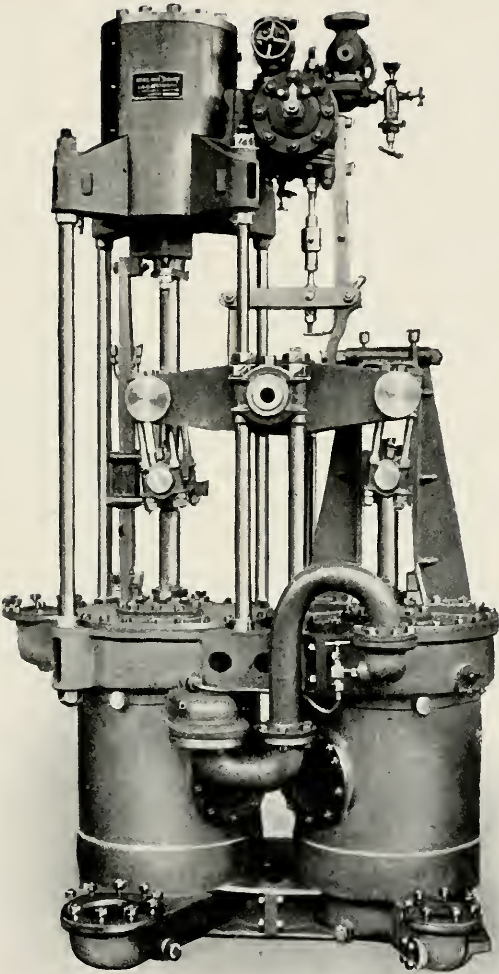
Description.—No. 1 shows in a diagrammatic form the arrangement of Surface Condenser, "Dual" air pump, and injection water cooler. In all cases the pump A or wet pump is situated below the steam cylinder, as this pump is the only one which works under any considerable load; the dry pump B is driven by the beam and links in the usual manner. One connection C is made to the condenser, but a branch pipe is led to the dry pump, the connection being made in such a manner that the water will all pass by C′ to the wet pump. Both pumps are generally of the three-valve marine type, but in certain cases the dry pump may be of the suction valveless type.

The first and most important difference from an ordinary twin pump consists in the dry pump discharging through the return pipe E, through a spring-loaded valve F, into the wet pump at a point below its head valves. The next point concerns the supply of water to the dry pump for water sealing, clearance filling, cooling, and vapour condensing. When starting the pump the filling valve G must be opened for a minute or so to enable the vacuum to draw in a supply from the hotwell of the wet pump. The valve is then closed, and the water passes from the hotwell of the dry pump by the pipe H to the annular cooler, and after being cooled passes into the suction of the dry pump, then, passing through the pump, it becomes heated and again passes to the cooler, and so on in a continuous



No 1.—Diagrammatic Section of Weir "Dual" Air Pump, Cooler and Connections.

closed circuit, any excess passing over the pipe E to the wet pump. The spring-loaded valve F is adjusted to maintain about 20 in. vacuum in the dry pump hotwell when the condenser is working at 28 in. vacuum, and this 8 in. difference of pressure is sufficient to cause the water to overcome the cooler friction and pass into the

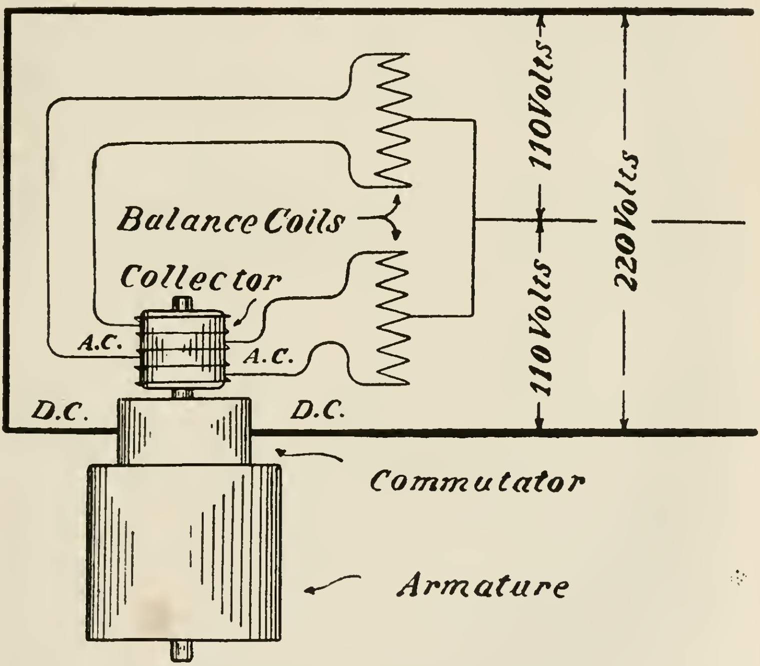


THE WEIR "DUAL" AIR PUMP.

suction, and at the same time never allow any direct air connection between the dry suction and discharge.

Operation of Three-Wire System.

The generators of this system consist of an ordinary direct current machine with two or more slip rings on the armature shaft extending as shown in the sketch. These slip rings are connected to the armature winding in such a manner as to give single phase in the case of two slip rings, three phase in the case of three slip rings, and two phase in the case of four slip rings.



No. 3.—Three-Wire System.
(With Single Dynamo.)

The slip rings are also connected through the brushes to one or more auto-transformers or balance coils, the middle or neutral point of the balance coils constituting the mid point of the direct current circuit.

The sketch shows diagrammatically a two-pole direct current armature. The direct brushes are shown as bearing on the commutator, and the four slip rings are connected to points on the armature diametrically opposite. An alternating voltage appears at the slip rings, having a frequency equal to the number of poles multiplied by the speed. Thus a four-pole generator running at 3000 revolutions per minute would have 12,000 alternations per minute = 100 periods per second.

The balance coil is simply an alternating current transformer with a single winding, and a tap from the middle point, and since it is connected to an alternating current circuit, an alternating magnetising current will flow through its windings; but as this magnetising current is ordinarily very small, it may be neglected when considering the flow of the direct current from the neutral wire to the armature windings.

Thus, under ordinary conditions, the only current in the coil which need be considered is direct current.

The object of the balance coil is to give a point midway between the direct current brushes to which the neutral wire of the system may be connected, and to afford a path by which the current flowing in the neutral wire may pass through the armature to one of the main brushes.

It will be seen from the diagram that this system is very valuable where it becomes necessary to have electric lighting on a low voltage, and heavy driving power from the outers or high voltage.

An out of balance load up to 15 per cent. may safely be carried between one outer wire and the neutral wire.

Knocking in Engines.

Knocking may be caused as follows:—

1. Want of clearance in cylinder, top or bottom; if on bottom, due perhaps to wear down.
2. Slack guide shoes.
3. High compression forcing slide valve off face at end of stroke.
4. Loose piston rod nut.
5. Loose crosshead nut.
6. Worn top end brass or bottom end brass.
7. Water in cylinder due to priming.
8. Want of compression (remedy—shut in link).
9. Junk ring pins slackening back, or working out altogether.
10. Worn valve gear brasses, &c.

Diagram cards, if taken off at the time, would indicate causes 1, 3, 7, and 8; for the others endeavour to locate position of knock, whether internal or external.

If the knocking is due to priming, this will further show by slowing down of the engines as the steam supply is reduced by the water passing over with it from the boilers.

Engine Data.—The following data referring to the machinery of a modern twin screw meat-carrying steamer may be studied with advantage by the student, as a good idea of the relative proportions of the various working parts can be obtained by reference to the data supplied.

Twin Screw Steamer, I.H.P. (Combined) 5000.

Cylinders - - - - -	25, 42, 69 in. (2 off).
Stroke - - - - - -	4 ft.
Crank shaft - - - -	$13\frac{1}{2}$ in. diameter.
„ pin - - - -	$13\frac{1}{2}$ „
Length of crank pin - - -	14 in.
Condensing surface - - -	6000 sq. ft. (?).
Thrust surface - - - -	1324 (2).
Piston rods - - - -	$6\frac{3}{4}$ in. diameter.
Connecting rod - - - -	$6\frac{1}{4}$ and $7\frac{1}{4}$ in. diameter.
„ „ centres - - -	8 ft. 6 in.
Valve rod, diameter in gland - -	$3\frac{3}{4}$ in.
Main steam pipe - - -	8 in. diameter.
Propellers (2) - - - -	16 ft. $7\frac{1}{2}$ in. diameter. 20 ft. pitch.
Expanded area - - -	68·8 sq. ft.
Projected „ - - - -	53·5 sq. ft.
Blades - - - - -	4.
„ thickness at root - -	$6\frac{3}{8}$ in.
„ „ top - -	$\frac{1}{2}$ in.
Boss - - - - -	Cast iron.
Blades - - - - -	Manganese bronze.
Studs - - - - -	Mild steel.
Nuts - - - - -	Gun metal.
Shaft nut - - - - -	Mild steel.
Boilers - - - - -	5 single-ended, 15 ft. 6 in. by 11 ft. 6 in.
Pressure - - - - -	180 lbs.
Tube surface - - - -	11,508 sq. ft.
Heating surface - - - -	13,560 „
Grate area - - - -	295 „
Tubes - - - - -	8 ft. 2 in. and 8 ft. $1\frac{3}{16}$ in.
„ diameter - - - -	$2\frac{1}{2}$ in.
Length of fire bars - - -	5 ft. 6 in.
Furnaces - - - - -	3 ft. 7 in. diameter.

Stop Valves.

Boiler Stop Valve, $5\frac{1}{4}$ in. Diameter.

Inches Open.	Number of Turns of Hand Wheel.	Steam Speed. (Feet per Second.)
$\frac{3}{4}$ in.	3	154
$\frac{5}{8}$ "	$2\frac{1}{2}$	185
$\frac{1}{2}$ "	2	231
Engine-room Stop Valve (balanced), 8 in. Diameter.		
$2\frac{1}{4}$ in.	$18\frac{1}{2}$	90·6
2 "	$16\frac{1}{4}$	95·2
$1\frac{3}{4}$ "	$14\frac{1}{2}$	100·2
$1\frac{1}{2}$ "	$12\frac{1}{2}$	117·2
$1\frac{1}{4}$ "	$10\frac{1}{2}$	140·5

Pressures on Bearing Surfaces.

M. Bearings.	Crank Pin.	C. Rod. Top Ends.	Slipper Guide.	Thrust Surface.	Stern Bearing.
$13\frac{1}{2} \times 14$ in.	$13\frac{1}{2} \times 13\frac{7}{8}$ in.	$7\frac{1}{2} \times 8$ in.	26×21 in.	44·2 lbs. per sq. in.	28·8 lbs. per sq. in.
234 lbs. per sq. in.	472 lbs. per sq. in.	737 lbs. per sq. in.	38·1 lbs. per sq. in.		

Steam Speeds.

Cylinders.	Revolutions.	Piston Speed.	Port Area.	Port Opening.	Exhaust Speed.	Steam Speed.	Valve Travel.	Cut-off.	Speed in Pipes.	Exhaust to Condenser.
			Sq. In.	Sq. In.			In.			
H.P.	...	...	67	53·8	71·4	88·8	$5\frac{1}{2}$	·7	8 in. dia., 95 F.P.S.	
I.P.	73	584	112	93·8	120·5	143·8	$5\frac{1}{2}$	·7	...	
L.P.	...	...	$246\frac{3}{4}$	229	147·3	159	$5\frac{1}{2}$	·7	...	110 F.P.S.

Economy of Feed Water Heating.

(Weir System.)

Data.—

Boiler Steam, 185 lbs. gauge.
L.P. „ 10 „
Hot well Temperature, 110°.
„ feed „ 220°.
Heating steam drawn from L.P. chest.

Then, $185 + 15 = 200$ lbs. absolute boiler steam.

200 lbs. absolute $= 381{\cdot}7$ temperature (from Table, page 709).

Total heat $= 1115 + {\cdot}3 \times T°$.

„ $= 1115 + {\cdot}3 \times 381{\cdot}7 = 1229{\cdot}5$ B.T.U.

Again, $10 + 15 = 25$ lbs. absolute L.P. steam.

25 lbs. absolute $= 240°$ temperature (from Table, page 705).

Total Heat $= 1115 + {\cdot}3 \times 240 = 1187$ B.T.U.

Heat available for feed heating per lb. L.P. steam $= 1187 - 110° = 1077$ B.T.U.

Increase of feed temperature required $= 220° - 110° = 110°$.

Percentage of L.P. steam required for feed heating. $= 1077 : 110 :: 100$ per cent. $= 10{\cdot}21$ per cent.

Heat required per lb. water for evaporation in boiler without feed heating. $= 1229{\cdot}5 - 110 = 1119{\cdot}5$ B.T.U.

Heat required per lb. water for evaporation in boiler with feed heating $= 1229{\cdot}5 - 220° = 1009{\cdot}5$ B.T.U.

Extra water evaporated due to feed heating $=$ As, $1119{\cdot}5 : (1119{\cdot}5 - 1009{\cdot}5) :: 100$ per cent. $= 9{\cdot}8$ per cent.

Assuming about equal horse-power in each of the three cylinders:—

Loss of power due to withdrawal of L.P. steam $= 10{\cdot}21$ per cent. $\div 3 = 3{\cdot}4$ per cent.

Then, Clear gain by feed heating $= 9{\cdot}8$ per cent. $- 3{\cdot}4$ per cent. $= 6{\cdot}4$ per cent.

From the foregoing it will be evident that the heat units given up by condensation in the heater by each pound of L.P. steam is sufficient (neglecting transmission losses, etc.) to heat up 9·6 lbs. of feed water from 110° to 220° temperature, as $1077 \div 110° = 9{\cdot}6$ lbs. water heated.

Boiler Efficiency.

The efficiency of a boiler is the ratio of actual to theoretical evaporation of 1 lb. coal for water from and at a temperature of 212°, which latter is expressed as the "equivalent evaporation," and is the standard taken for test purposes.

To obtain "Equivalent Evaporation."

Example. Pressure, 165 lbs. gauge; temperature, 372·8°; feed temperature, 160°; actual evaporation (measured), 8·5 lbs. water per pound coal. Find equivalent evaporation.

Then, Equivalent evaporation $= \dfrac{(1115 + \cdot 3T^\circ - t^\circ) \times \text{lbs. water}}{966}$.

Where, T° = Steam temperature, t° = Feed temperature.

So that, $\dfrac{(1115 + \cdot 3 \times 372{\cdot}8^\circ - 160^\circ) \times 8{\cdot}5}{966} = 9{\cdot}38$ lbs. (from and at 212).

Observe that 966 B.T.U. $= 1115 + \cdot 3 \times 212^\circ - 212^\circ$.

Calorific (Heat) Value of Coal.—The following table gives the average composition and heat unit value of Welsh, Newcastle, and Scotch coal, also of petroleum.

COMPOSITION AND HEAT VALUE OF FUEL.

Name.	Carbon.	Hydrogen.	Oxygen.	Sulphur.	Ash, Nitrogen, etc.	Total Heat Units per Pound.
Welsh coal -	84	4·8	4	1·5	5·7	14700 B.T.U.
Newcastle coal	82	5·3	0	1·2	5·5	14600 ,,
Scotch coal -	78	6	9	1	6	14000 ,,
Petroleum -	86·5	12·3	1·2	...	...	20000 ,,

The heat value of the fuel must be known to obtain anything like accurate results, and usually a sample of the coal is specially analysed to obtain the average calorific value per pound.

Then,

Boiler efficiency (per cent.) $= \dfrac{\text{Equivalent evaporation} \times 966 \times 100 \text{ per cent.}}{\text{Calorific value}}$.

Example 1. Steam pressure (gauge), 170 lbs.: feed temperature, 180°; actual evaporation per pound coal (measured), 8·6 lbs. water; calorific value of fuel (Scotch coal), 14000 B.T.U. Find boiler efficiency.

Then, $170 + 15 = 185$ lbs. absolute pressure.

And, 185 lbs. $= 375^\circ$.

Also, $1115 + \cdot 3 \times 375^\circ - 180^\circ = 1047{\cdot}5$ B.T.U. required for evaporation.

Equivalent evaporation $= \dfrac{1047{\cdot}5 \times 8{\cdot}6}{966} = 9{\cdot}32$ lbs. water from and at 212°.

Boiler efficiency $= \dfrac{9{\cdot}32 \times 966 \times 100 \text{ per cent.}}{14000} = 64{\cdot}3$ per cent.

Example 2. Pressure, 200 lbs.; feed temperature, 170°; actual evaporation (measured), 9 lbs. water per pound coal; calorific value of coal, 12500 B.T.U. Find boiler efficiency.

Then, $200 + 15 = 215$ lbs. absolute.

And, 215 lbs. $= 388^\circ$ temperature (from Table, page 709).

Equivalent evaporation $= \dfrac{(1115 + \cdot 3 \times 388^\circ - 170^\circ) \times 9}{966} = 9{\cdot}88$ lbs. from and at 212°.

Boiler efficiency $= \dfrac{9{\cdot}88 \times 966 \times 100 \text{ per cent.}}{12500} = 76{\cdot}3$ per cent.

The efficiency ranges from about 65 to 80 per cent. in ordinary cases, and varies with type of boiler. The efficiency is increased when the steam is superheated.

Coal and Water Consumption.—In testing a boiler for efficiency the coal is supplied in measured quantities, and the feed water is passed through measuring tanks or barrels, so that accurate records may be obtained of the actual coal burnt and water evaporated in a given time. From this the actual evaporation per pound coal is obtained as follows:—

$$\frac{\text{Total feed water evaporated, say, per hour (as measured)}}{\text{Total coal consumed per hour (as measured)}} = \text{Actual evaporation.}$$

Example.—In testing a boiler plant for the efficiency, the coal fired in twenty-four hours (test period) measured 54620 lbs., and the water evaporated in the same time as measured with the tanks was 478500 lbs. Find the actual evaporation per pound.

Then, Actual evaporation $= 478500 \div 54620 = 8{\cdot}76$ lbs. Answer.

Exercise 1. Find boiler efficiency, given the following:—Pressure, 180 lbs. gauge; feed water temperature, 162°; actual evaporation, 8·8 lbs. water; and heat value of fuel, 13000 B.T.U. per pound. Answer, 72 per cent.

Exercise 2. Find boiler efficiency, given the following:—Pressure, 215 lbs. gauge; feed water, 190° temperature; total coal burnt per hour, 2500 lbs.; and total water evaporated per hour, 21500 lbs. Heat value of coal, 13500 B.T.U. per pound. Answer, 66·4.

Superheated Steam.—If the steam is superheated the number of heat units required to produce the rise of superheat temperature must be added to the total heat of the steam. To obtain the number of heat units, the specific heat constant of superheated steam at constant pressure has to be used, and this is usually taken at ·48, which means that to raise one pound of steam 1° Fahr. requires only ·48 of the heat required to raise one pound of water 1° Fahr.

Therefore, Heat units $= {\cdot}48 \times$ Degrees superheat.

Example. Find number of heat units added to steam to produce a superheat of 100° Fahr. (from 366° to 466° Fahr.).

Then, Heat units $= {\cdot}48 \times 100^\circ = 48$ B.T.U. Answer.

Equivalent Evaporation (with superheated steam).

$$\frac{[(1115 + {\cdot}3 \times T^\circ - t^\circ) + {\cdot}48 \times (T_1^\circ - T^\circ)] \times \text{Pounds water}}{966}$$

Where, T° = Boiler steam temperature.

" t° = Feed temperature.

" T_1° = Temperature of superheated steam.

" 966 = Latent heat of evaporation from and at 212°.

Example. Boiler pressure, 200 lbs. gauge, surperheated to 500° Fahr. Feed temperature, 190°; actual evaporation (measured), 9·3 lbs. water per pound coal; and calorific value of coal, 13000 B.T.U. per pound. Find boiler efficiency.

Then, $200 + 15 = 215$ lbs. absolute.

Temperature $= 388°$ (from Table, page 709).

Therefore,

$$\left.\begin{array}{r}\text{Equivalent}\\ \text{evaporation}\end{array}\right\} = \frac{[(1115 + \cdot 3 \times 388° - 190° + \cdot 48 \times (500° - 388°)] \times 9\cdot 3}{966}$$

$$= \frac{(1115 + 116\cdot 4 - 190 + 53\cdot 76) \times 9\cdot 3}{966}$$

$$= \frac{(1095\cdot 16 \times 9\cdot 3}{966} = 10\cdot 45 \text{ lbs. water from and at } 212°.$$

$$\text{Boiler efficiency} = \frac{10\cdot 43 \times 966 \times 100 \text{ per cent.}}{13000} = 77\cdot 6 \text{ per cent. Answer.}$$

Exercise. Boiler pressure, 185 lbs. gauge superheated up to 510° Fahr.; feed temperature, 165°; actual evaporation, 9 lbs. water per pound coal; and heat value of coal, 13500 B.T.U. per pound. Find boiler efficiency. Answer, 74·9 per cent.

Engine and Boiler Data.

The following data constitutes a useful and valuable reference to the various proportions of parts, as usually designed, for boilers and engines of a given power, and the reader should carefully study and compare the dimensions, proportions, and results as tabulated.

Vessel.	Length	425 ft. B.P.
	Breadth	54 ft.
	Depth	31 ft. 6 in.
	Mean draught	27 ft. 6 in.
	Gross tonnage	7300 tons.
	Dead weight	9250 ,,

Boilers. Four single-ended, fitted with Schmidt type superheater tubes, $\frac{3}{4}$ in. diameter.	Diameter	16 ft. 2 in.
	Length	12 ft.
	Four furnaces	Deighton type.
	Diameter at mouth	3 ft. $4\frac{1}{2}$ in.
	,, inside corrugations	2 ft. $11\frac{3}{4}$ in.
	Length of bars	5 ft.
	G.S.	65 □ per boiler.
	No. of plain tubes per boiler	206.
	,, stay ,,	12 of $\frac{7}{16}$ in. thick.
	,, ,, ,,	60 ,, $\frac{3}{8}$,,
	,, ,, ,,	98 ,, $\frac{5}{16}$,,
	Length of tubes	7 ft. 6 in.
	Diameter of all tubes	3 in.
	Funnel diameter	8 ft.
	,, height	44 ft. 6 in. above casing.
	Boiler pressure	220 lbs. gauge.

Engines.
Quadruple expansion balanced on Yarrow, Schlick, and Tweedy system, total I.H.P., 4300.

Diameter of cylinders -	H.P., 27·5 in.; 1st I.P., 39 in.; 2nd I.P., 56 in.; L.P., 81·5 in.
Stroke - - -	54 in.
Length of connecting rods	9 ft. 6 in.
Diameter of piston rods -	8 in.
Depth of stuffing box -	10 in.
Diameter of stuffing box -	$10\frac{3}{4}$ in.
,, valve spindles	$4\frac{3}{4}$,,
Depth of stuffing box -	8 ,,
Diameter of stuffing box -	7 ,,
,, crank shaft -	16 ,,
,, ,, pins -	$16\frac{1}{4}$,,
,, tunnel shaft -	$15\frac{1}{4}$,,
,, propeller shaft	$16\frac{7}{8}$,,
Diameter of propeller shaft over liner - -	$18\frac{3}{8}$,,

Propeller.
Four-bladed R.H.

Diameter - - -	18 ft. 9 in.
Pitch - - -	18 ft. $7\frac{1}{2}$ in.
Surface - - -	120 □'
Boss of cast steel.	
Blades of Bull's metal.	
Diameter of studs -	$3\frac{1}{4}$ in.

Condenser.
Weir's Uniflux type.

Cooling surface - -	3500 □'
Number of tubes - -	1620
Diameter of tubes (ext.) -	$\frac{3}{4}$ in.
Thickness ,, - -	18 B.W.G.
Length between tube plates	11 ft.

NOTE.—I.H.P. = 4300.

Then, $\frac{3500}{4300} = \cdot 8$ □' cooling surface per I.H.P.

Piston Clearances.

	Top.	Bottom.
H.P. - - - -	$\frac{5}{16}$ in.	$1\frac{1}{8}$ in.
1st I.P. - - - -	$\frac{7}{16}$,,	$1\frac{1}{16}$,,
2nd I.P. - - - -	$\frac{9}{16}$,,	$1\frac{1}{8}$,,
L.P. - - - -	$\frac{3}{4}$,,	$1\frac{1}{16}$,,

Pump Clearances.

	Top.	Bottom.
Air pump - - -	$\frac{7}{16}$ in.	$\frac{15}{16}$ in.
Bilge pump - - -	$\frac{1}{2}$,,	$\frac{3}{4}$,,

VALVE SETTINGS.

Travel of all Valves, 8 in.	H.P.		1st I.P.		2nd I.P.		L.P.	
	Top.	Bottom.	Top.	Bottom.	Top.	Bottom.	Top.	Bottom.
	In.	In.	In.	In.	In.	In.	In.	In.
Lead	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{5}{16}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{11}{16}$
Steam lap	$1\frac{15}{16}$	$1\frac{13}{16}$	$2\frac{1}{16}$	$1\frac{15}{16}$	2	$1\frac{7}{8}$	$1\frac{7}{8}$	$1\frac{3}{4}$
Port opening	$2\frac{1}{16}$	$2\frac{3}{16}$	$1\frac{15}{16}$	$2\frac{1}{16}$	2	$2\frac{1}{8}$	$2\frac{1}{8}$	$2\frac{1}{4}$
Exhaust lap	$+\frac{1}{16}$	$+\frac{11}{16}$	$-\frac{1}{8}$	$+\frac{5}{8}$	$-\frac{1}{16}$	$+\frac{11}{16}$	$+\frac{1}{16}$	$+\frac{13}{16}$
Cut-off	42	$37\frac{3}{4}$	$39\frac{3}{4}$	$35\frac{1}{2}$	40	$35\frac{1}{2}$	$40\frac{3}{4}$	$36\frac{1}{4}$
Compression (from end of stroke)	$5\frac{1}{2}$	6	$5\frac{3}{4}$	$6\frac{1}{4}$	$6\frac{1}{2}$	7	$7\frac{1}{8}$	$7\frac{5}{8}$
	Piston valve. Diam. at top, $15\frac{1}{4}$ in. Diam. at bot., 15 in.		Piston valve. Diam. at top, $22\frac{1}{4}$ in. Diam. at bot., 22 in.		Twin Piston Valve. Diam. at top, $18\frac{1}{4}$ in. Diam. at bot., 13 in.		Double Ported Slide Valve.	

Observe that the difference in steam lap and in port opening top and bottom is exactly equal to the difference in lead top and bottom ($\frac{1}{8}$ in.); also notice that, as link motion valve gear is fitted, the cut-off is later on the down stroke in each case. Again, the 1st I.P., 2nd I.P., and L.P. valves have all minus exhaust lap on top and plus exhaust lap on bottom: this is designed to reduce the top compression and to increase the bottom compression.

Pressures.

Boiler steam - - 220 lbs. gauge.
H.P. ,, - - 208 ,,
1st I.P. ,, - - 108 ,,
2nd I.P. ,, - - 47 ,,
L.P. ,, - - 11·5 ,,
M.E.P., H.P. - - 78·2 lbs. □ (mean cut-off = 78 per cent.).
,, 1st I.P. - - 39·4 lbs. □ (mean cut-off = 71·5 per cent.).

Revolutions, power, speed.

,, 2nd I.P. - - 20 lbs. □ (mean cut-off = 70·7 per cent.).
,, L.P. - - 12 lbs. □ (mean cut-off = 71·6 per cent.).
Revolutions - - 78.
I.H.P. of H.P. - - 988
,, 1st I.P. - - 1000
,, 2nd I.P. - 1047
,, L.P. - - 1331
Total I.H.P. 4366

Consumption and Slip.		
	Tons coal per day	60·5 tons.
	Speed	12·86 knots.
	Slip per cent.	7·6.
	Coal per I.H.P. hour	1·3 lbs.
	Sea	Moderate.

NOTE. $\frac{60{\cdot}5 \times 2240}{4366 \times 24} = 1{\cdot}3$ lbs. per I.H.P. hour.

STANDARD SPECIFICATION FOR CARGO-STEAMER ENGINES.

Guidance Specification for Reciprocating Triple Expansion Marine Engines for Cargo Boats.*

Compiled by the Council and Members of the North-East Coast Institution of Engineers and Shipbuilders, 1917.

Specification.

1. **Indicated Horse-Power.**—For calculation purposes in this specification and in average sea conditions the I.H.P. is to be found as follows :—

$$\text{I.H.P.} = \frac{D^2 S N}{700}$$

D=Diameter of L.P. cylinder in inches.
S=Stroke in feet.
N=Revolutions per minute. Found as per section 2.

The divisor is adjusted for a referred mean pressure of 30 lbs. per sq. in.

2. **Revolutions.**—

$$N = \frac{32\ (S + 4)}{S}$$

3. **Boiler Pressure.**—180 lbs. per sq. in. (gauge).

4. **Ratios of Cylinder Areas.**—Ratio for 180 lbs. pressure.

H.P.	M.P.	L.P.
1	About 2·74	About 7·5
	,, 1	,, 2·74

5. **Cuts off at Sea Power.**—About 57·5 per cent.; 57·5 per cent.; 55 per cent.

6. **Speeds of Steam.**—The mean steam speeds to be calculated as follows :—

$$\frac{\text{Area of cylinder in square inch} \times \text{piston speed in feet per second.}}{\text{Area of pipe, port or opening in square inch.}}$$
$$= \text{speed in feet per second.}$$

* Extract from paper read at the North-East Coast Institution of Engineers and Shipbuilders, 3rd February 1917.

Table of mean steam speeds in feet per second :—

	H.P.	M.P.	L.P.
Main steam pipe - -	110	—	—
Port opening - - -	110	150	240
Steam ports - - -	80	85	100
Exhaust passage or pipe -	60	65	75

Width of Steam Ports.—Width of ports to be about 0·8 of diameter of cylinder.

7. **Maximum Load.**—The maximum load on main working parts to be taken as the product of the area of H.P. cylinder in inches, and the boiler pressure in lbs. per square inch (gauge).

8. **Crankshaft.**—The diameter of crankshaft in body to be to nearest $\frac{1}{8}$ in. above Lloyd's Rule, and the proportions of the remaining parts to be not less than the following :—

1. Diameter of crank-pin to be equal to diameter of shaft.
2. Diameter of crankshaft in web to be equal to diameter of shaft, plus $\frac{1}{2}$ in.
3. Diameter of webs to be equal to diameter of crank-pin by 1·85.
4. Thickness of webs to be equal to diameter of shaft by 0·62.
5. Thickness of couplings to be equal to diameter of shaft by 0·25.
6. Six coupling bolts to be used for shafts up to and including 15 in. diameter. Nine coupling bolts to be used for shafts above 15 in. diameter.

*7. Diameter of pitch circle of coupling bolts to be 1·43 diameter of crankshaft.

*8. Diameter of coupling bolts to be equal to :—

$$0{\cdot}7 \times \sqrt{\frac{\text{diameter of shaft}^3 \text{ in inches.}}{\text{number of bolts} \times \text{diameter of pitch circle in inches.}}}$$

Bolts to be parallel.

9. **Length of Connecting Rod.**—Length of connecting rod between centres to be twice the stroke or four times the crank radius.

Diameter of Connecting Rod.—Connecting rods may be made parallel, same diameter as piston rod body.

Connecting Rod Top Ends.—Connecting rods to have single top-end gudgeons for all engines having H.P. cylinders of 25-in. diameter and under.

10. **Crosshead Guides.**—Main crosshead guides to be of the single type in all sizes of engine.

* These two rules may be varied, provided that equivalent strength is given.

Load on Main Crosshead Guides.—Maximum load in lbs. on crosshead guides to be taken as :—

$$\frac{\text{Area of H.P. cylinder in square inches} \times \text{boiler pressure in pounds per square inch (gauge)}}{4}$$

11. **Maximum Pressures on Principal Bearing Surfaces.**—

	Lbs. per sq. in.
Main bearings - - - -	250
Crank-pins - - - - -	500
Crosshead gudgeons - - -	1000
Guide shoes (ahead) - - -	55
„ „ (astern) - - -	110

Diameter by length to be taken as area of bearings.
Overall length by overall breadth as area of guide shoes.

12. **Maximum Stresses on Principal Working Parts.**—

	Lbs. per sq. in.
Ingot steel piston rod at screw - - -	6000
Piston rod body (after deducting $\frac{1}{4}$ in. from diameter to allow for returning) - - - -	3000
Piston and connecting rod bolts at screw - -	5500
Main bearing bolts - - - - -	4500
Main bearing keeps (if forged) - - -	6000
Connecting rod bottom end keep (if forged) -	7500
Piston rod keep (if forged) - - - -	7500

(The keeps are calculated as beams with distributed load and supported ends.)

13. **Valve Gear.**—The valve gear sizes to be determined from the load on the M.P. slide valve spindle, calculated as follows :—

$$\text{Load in lb.} = 0{\cdot}165 \left\{ 54\ (A - B) - 9\ C \right\}$$

Where A = Area of face of M.P. valve in square inch.
B = Combined area of steam ports in valve face in square inches.
C = Combined area of exhaust ports in valve face in square inches.

Valve Spindles.—Diameter of valve spindles at gland to be not less than :—

$$\frac{\text{Diameter of piston rod at gland}}{2} \times \tfrac{5}{8} \text{ in.}$$

Maximum Pressures on Bearing Surfaces of Valve Gears.—

	Lbs. per sq. in.
Link block gudgeon - - - -	500
„ „ slippers - - - -	300
Eccentric rod top end pins - - -	500
„ sheaves (ahead and astern) - -	85

14. **Thrust Block.**—When of horseshoe type the pressure on thrust collars not to exceed 70 lbs. per sq. in. when calculated from indicated thrust, which is determined as follows :—

$$\text{Lbs. Indicated thrust} = \frac{\text{I.H.P.}\quad 33{,}000}{\text{Pitch in feet} \times \text{revolutions per minute.}}$$

15. **Circulating Water.**—The amount of circulating water supplied to be forty times the feed, taking the latter at 15 lbs. of steam per I.H.P. per hour.

16. **Main Engine-Driven Reciprocating Circulating Pump (Double-acting).**—To be proportioned to deliver the above quantity of water at a displacement efficiency of 80 per cent.

17. **Maximum Speeds of Circulating Water.**—The speeds of circulating water are to be calculated as follows:—

$$\frac{\text{0·8 area of bucket in sq. in.} \times \text{bucket speed in ft. per sec.}}{\text{Area of passage in sq. in.}} = \text{speed of water.}$$

Approximate Speeds in Feet per Second.—

	Ft. per sec.
Main injection - - - - -	9·0
Passages in pump - - - - -	5·0
Valve grids - - - - - -	6·0
Past lift of valves - - - - -	9·5
Discharge pipe - - - - -	7·5

18. **Air Pump.**—Capacity of air pump not less than $\frac{1}{16}$ of the capacity of L.P. cylinder.

19. **Main Engine-Driven Feed Pumps.**—Capacity of *each* engine-driven feed pump $\frac{1}{700}$ of capacity of L.P. cylinder.

20. **Pump Gear.**—Load on pump gear to be calculated as follows: Load in lb. = 25 (area of air pump bucket + area of circulating pump bucket) + 15 (area of both feed pump rams + area of both bilge pump rams). All in square inches.

Maximum Pressures on Pump Gear Bearing Surfaces.—

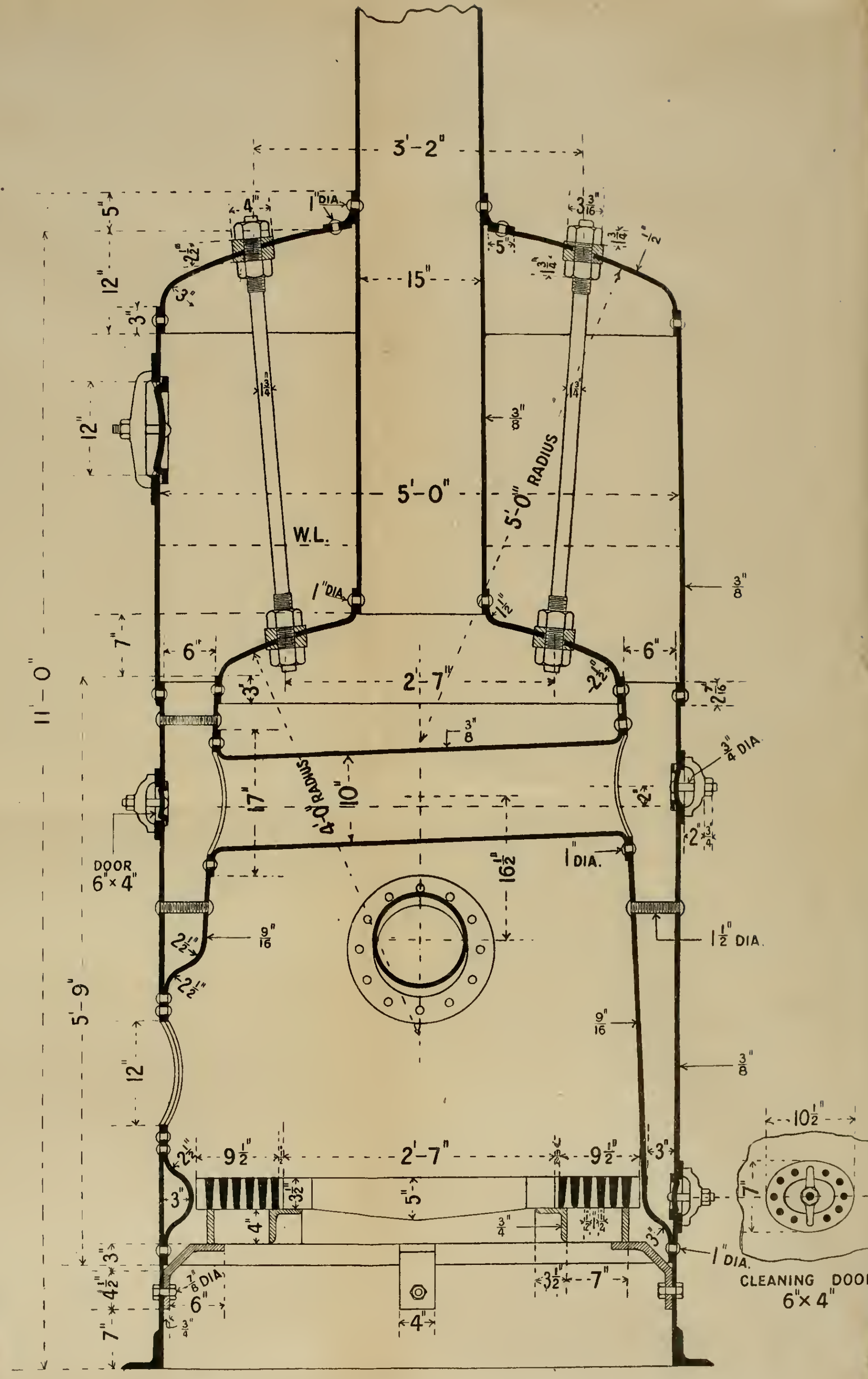
	Lbs. per sq. in.
Pump link pins - - - - -	400
Engine link pins - - - - -	300
Pump lever centre gudgeon bearings - -	250

For cargo vessels of large tonnage it is recommended that the circulating and feed pumps be independently driven pumps.

21. **Utilisation of Heat in Exhaust Steam from Auxiliary Engines.**—A source of very considerable economy in a marine installation being the complete absorption by the feed water of the heat in the exhaust steam from the various auxiliaries, including the steering engine, electric-light engine and evaporator, such a vacuum should be carried in the main condenser as will enable this to be effected in all seas in which the vessel trades. A vacuum of 27 in. maintained in the steam space of the condenser, the temperature of the sea being 70° Fahr. (barometer 30 in.) has been found to meet these requirements on an average cargo boat.

22. **Cooling Surface.**—In determining the amount of cooling surface per I.H.P. average at sea, provision should be made for the rapid initial degrading effect of oil and scale on the tube surfaces, and also for the permanent prejudicial effect on the condensing efficiency of the residual air in the condenser.

38. Pump Crosshead and Links.



No. 1.—Steel Vertical Type Donkey Boiler. Pressure, 80 lbs. (gauge) per square inch.

[To face page 701.

DESIGN DRAWINGS AND CALCULATIONS.

FIRST CLASS.

The following Set of Drawings and Design Calculations include, among others, those given at the Board of Trade Examinations to First-Class Engineer Candidates, and for practice these should be drawn out to the scales marked on each.

Note.—The calculations shown for the various proportions of parts represent average practice, but it should be noted that these vary to some extent in the designs of different engine builders.

Sheets 1 and 2 show proportions of Nuts, Bolts, and Screws.

Boilers—

1. Single-Ended Combustion Chamber.
2. Double-Ended Combustion Chamber.
3. Furnace and Fire Bars.
4. Water Gauge Column.
5. Vertical Donkey Boiler.
6. Fire Bars and Bearers for Vertical Boiler.

Valves—

7. Dead Weight Safety Valve.
8. Spring-Loaded Safety Valve.
9. Boiler Stop Valve.
10. Engine Room Stop Valve.
11. Feed Check Valve.
12. Bilge Suction Valve Chest.

12A. Bilge Injection Valve.

13. Side Discharge Valve.
14. Cylinder Relief Valve.
15. Slide Valve and Spindle.
16. Inside Steam Piston Valve.
17. Double Ported Slide Valve.

Pumps—

18. Air Pump.
19. Feed Pump Complete.

19A. Feed Relief Air Vessel and Pump Valves.

Pistons, etc.—

20. H.P. piston and Rod.
21. L.P. Piston and Rod.
22. L.P. Cylinder Cover.
23. Donkey Pump Cylinder and Valve.

Eccentric, etc.—

24. Eccentric and Rod Complete.
25. Quadrant Bars, etc.
26. Reversing Bell Crank.

Shafting, etc.—

27. Crank Shafting.
28. Thrust Shaft and Shoe.
29. Thrust Block.
30. Stern Tube and Shaft.
31. Propeller Boss.

Various—

32. Bottom Blow-Off Cock.
33. Three-Way Change Cock.
34. Main Bearing.

34A. Tunnel Bearing Block.

35. Steam Pipe Expansion Joint.
36. Pump Levers.
37. Connecting Rod.
38. Pump Crosshead and Links.

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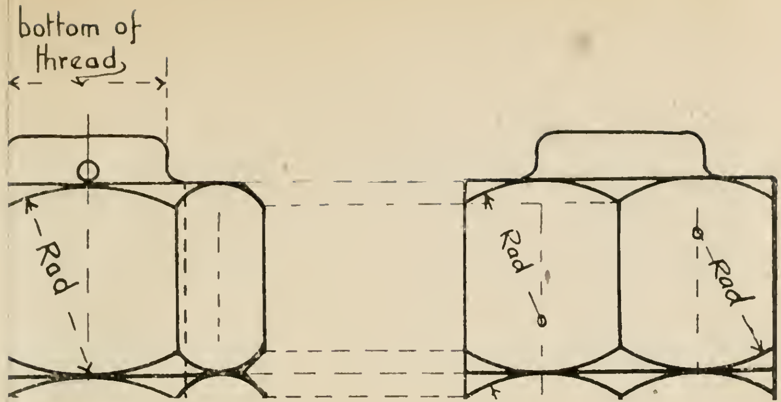
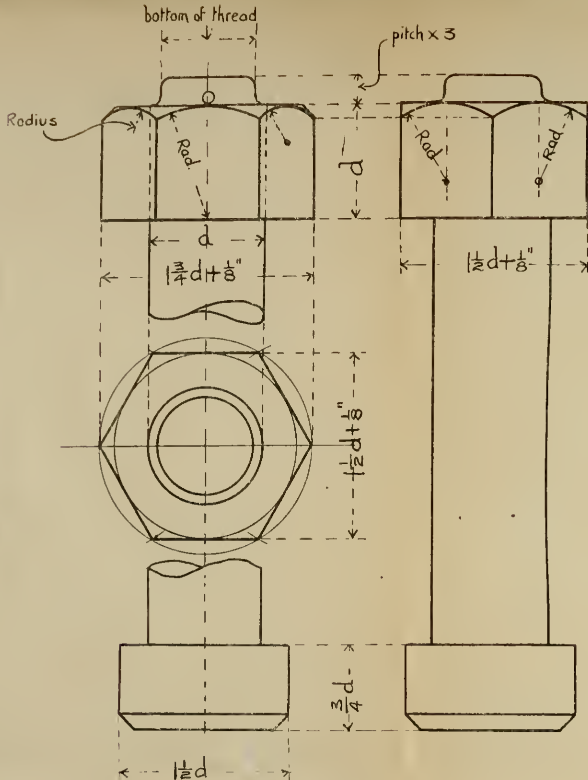
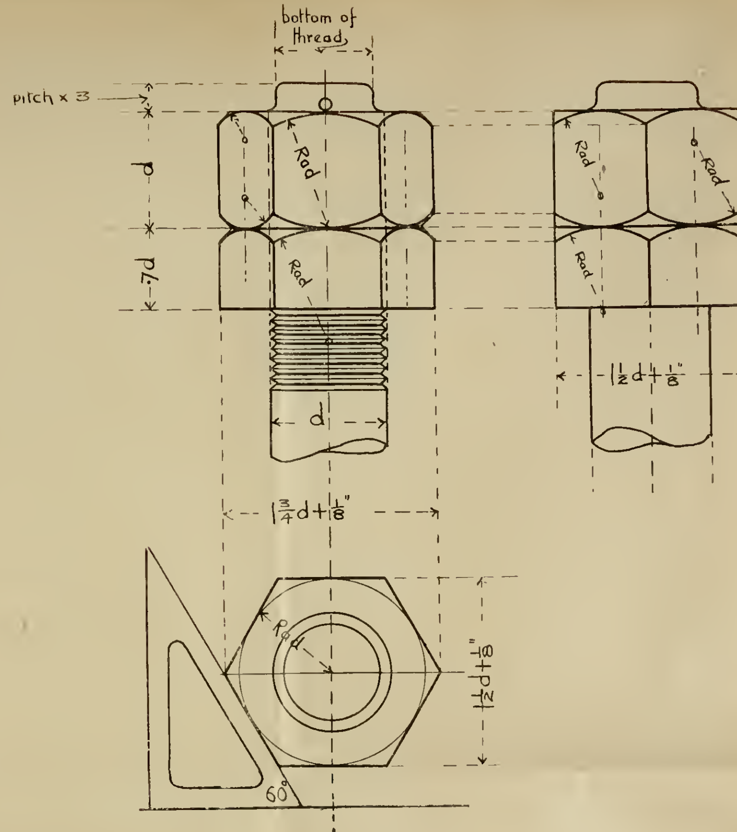

bottom of thread
Rad
Rad
Rad



Bolt and Nut Complete.

Notice that the radius of curvature is taken as equal to the depth of the nut (equal to d): this is not strictly correct, but is quite near enough for practical purposes. Observe the positions of radius for the nut curvature in the side view, which is about two-thirds distant from the top edge.

In the plan the construction circle (equal in diameter to the distance over angles) is divided up into six by the circle radius distance in the compasses.



Bolt and Lock Nuts.

The hexagon sides of the nut shown in the plan may be drawn in by applying the 60 degree set square tangentially to the circle over flats.

This circle $= 1{\cdot}5 \times d + \frac{1}{8}$ in.

The nuts are chamfered off to an angle of 45°.

EXAMPLE.—Calculate the required dimensions of nut and bolt head for a bolt 2 in. diameter, with $4\frac{1}{2}$ threads per inch.

Then,	Diameter over angles	$= 1{\cdot}75 \times 2 + {\cdot}125$ in.	$= 3{\cdot}625$ in.	$= 3\frac{5}{8}$ in.
	Diameter over flats	$= 1{\cdot}5 \times 2 + {\cdot}125$ in.	$= 3{\cdot}125$ in.	$= 3\frac{1}{8}$ in.
	Depth of nut	$= d$	$=$	2 in.
	Depth of lock nut (if fitted)	$= {\cdot}7 \times 2$ in.	$=$ 1·4, say	$1\frac{1}{2}$ in.
	Diameter of bolt head	$= 1{\cdot}5 \times 2$ in.	$=$	3 in.
	Depth of bolt head	$= {\cdot}75 \times 2$ in.	$=$	$1\frac{1}{2}$ in.

Diameter at bottom of thread $= 2 \text{ in.} - \left(\frac{1{\cdot}28}{4{\cdot}5}\right) = 1{\cdot}716$ in. say, $1\frac{23}{32}$ in. diameter.

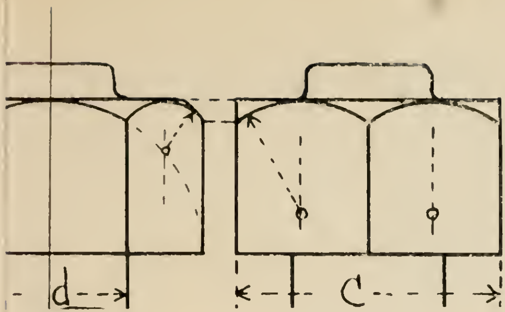
Pitch of thread $= 1 \div 4{\cdot}5 = {\cdot}22$ in.

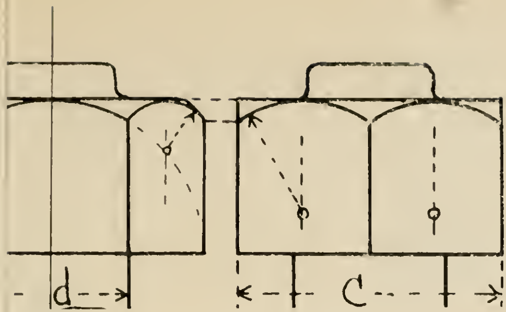
Nuts, Bolts, and Screws.

(Whitworth Standard.)

Let d = diameter of bolt.

Then.	Width across angles of nut	$= 1{\cdot}75 \times d + \frac{1}{8}$ in.
„	Width across flats of nut	$= 1{\cdot}5 \times d + \frac{1}{8}$ in.
„	Depth of nut	$= d$.
„	Diameter of round bolt head	$= 1{\cdot}5 \times d$.
„	Depth of round bolt head	$= {\cdot}75 \times d$.
„	Diameter of point of bolt	= diameter at bottom of thread
„	Depth of point of bolt	= pitch of thread × 3.
„	Depth of top lock nut	$= d$.
„	Depth of bottom lock nut	$= {\cdot}7 \times d$.
„	Diameter of washer	$= 2{\cdot}2 \times d$
„	Thickness of washer	$= {\cdot}2 \times d$


d
C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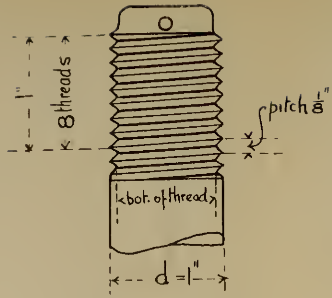


Screw for 1 in. diameter Bolt.

Whitworth Screws.

(Angular Threads.)

Angle of thread = 55 degrees.
Depth ,, = Pitch of thread × ·64.
Pitch .. = 1 in. ÷ threads per inch.
Diameter at bottom of thread = Bolt diameter $-\left(\dfrac{1\cdot28}{\text{Threads per inch}}\right)$.

Bolt Screws.

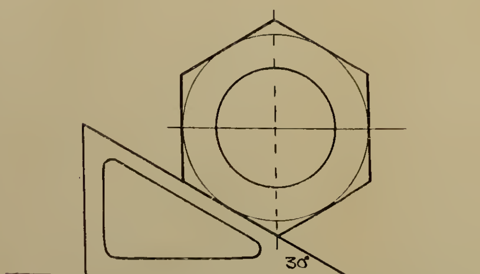
Diameter of Bolt or Stud.	Number of Threads per inch.	Diameter at Bottom of Thread (Effective Diameter).	Area at Bottom of Thread (Effective Area).
$\frac{1}{4}$ in.	20	·186	·027 □"
$\frac{1}{2}$,,	12	·393	·121 ,,
$\frac{3}{4}$,,	10	·622	·308 ,,
1 ,,	8	·840	·554 ,,
$1\frac{1}{2}$,,	6	1·287	1·30 ,,
2 ,,	$4\frac{1}{2}$	1·715	2·31 ,,
$2\frac{1}{2}$,,	4	2·180	3·73 ,,
3 ,,	$3\frac{1}{2}$	2·634	5·45 ,,
$3\frac{1}{2}$,,	$3\frac{1}{4}$	3·106	7·57 ,,
4 ,,	3	3·573	10·02 ,,

EXAMPLE —

Find the diameter and area at bottom of thread for a 2-in. bolt, having $4\frac{1}{2}$ threads per inch.

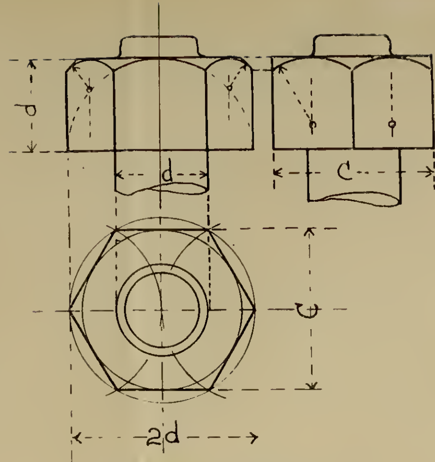
Then, $2-\left(\dfrac{1\cdot28}{4\cdot5}\right) = 1\cdot716$ in. diameter.

Area at bottom of thread $= 1\cdot716^2 \times \cdot7854 = 2\cdot31$ □".



Hexagon Sides of Nut.

In this method of construction the 30 degree angle set square is applied tangentially to the circle over flats ($1\cdot5 \times d + \frac{1}{8}$ in.), as shown above.

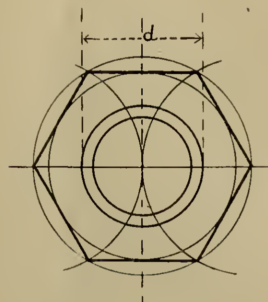


Rough Method of Drawing Nuts.

For small sizes of nuts, or for small scale drawings, it is sufficient to set off the nuts as shown above.

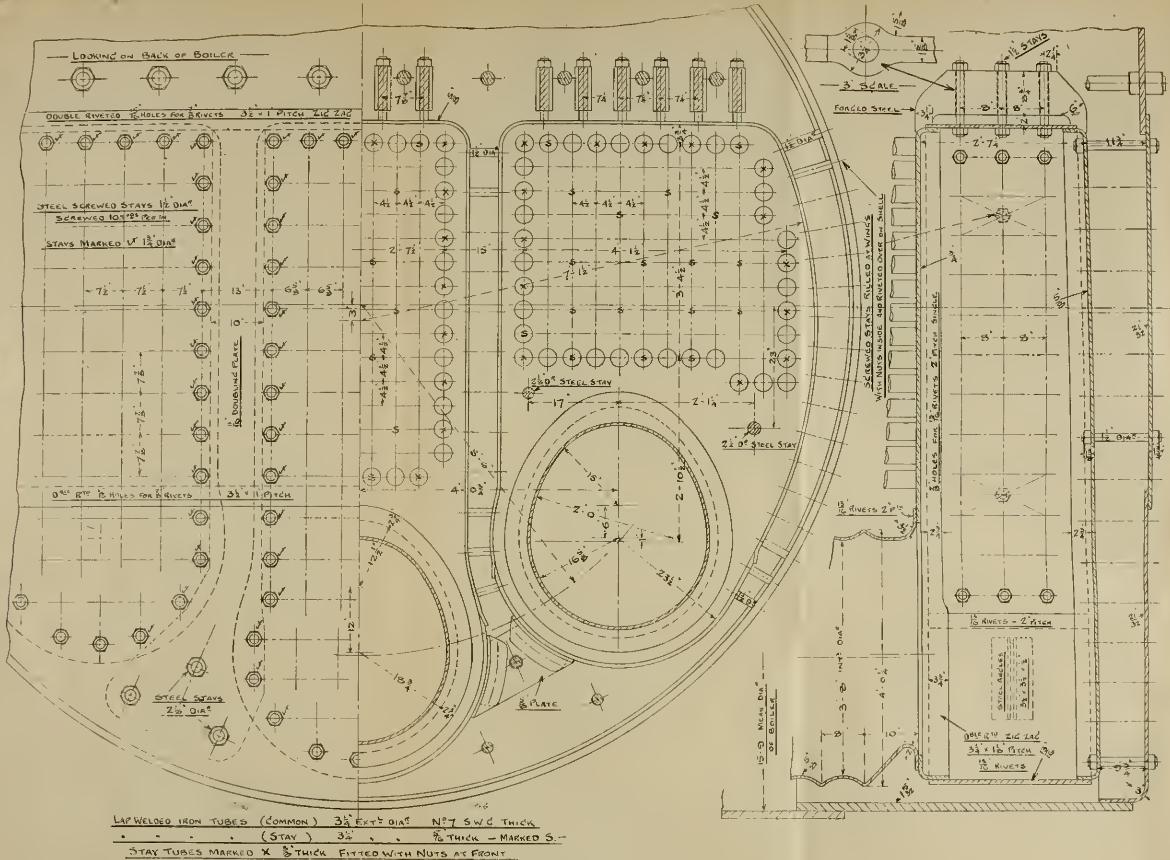
Observe that the distance over nut angles is taken as equal to $2 \times d$; also that the size over flats C is simply measured from the plan and transferred to the elevation view on right.

This method is, of course, approximate only, but is quite allowable under conditions of small sizes or small scales.



Hexagon Sides of Nut.

In this method of construction the circle over angles ($1\cdot75d + \frac{1}{8}$ in.) is divided up into six by means of the circle radius distance in the compasses, and the sides are then drawn in as shown.



No. 1. —Single-Ended Combustion Chamber.

(Pressure 180 lbs. natural draught.)

(Draw to a scale of 1 in = 1 ft.)

Girders.

To find required depth of girders, first fix on the thickness, in this case $1\frac{5}{8}$ in. (single plate and bossed out for the stays)

Rule, Depth of girders $= \sqrt{\frac{(W-P)\times d\times L\times \text{Press}}{T+C}}$.

Where, W = Width of combustion chamber.
P = Pitch of top bolts.
d = Distance between girders.
L = Length in feet.
T = Thickness
C = 825 for 3 bolts and 935 for 4 bolts.

Then, Depth $= \sqrt{\frac{(31{\cdot}25-8)\times 7{\cdot}25\times [illegible] \times 180}{1{\cdot}625\times 825}} = 8$ in. nearly.

To pass other surveys, such as Lloyds or British Corporation, the girders are often made deeper than that required by the B. of T. In the present case $8\frac{3}{4}$ in. is the size decided on, in place of barely 8 in.

NOTE.—Length of girder $= 31{\cdot}25 + \frac{5}{8} = 31{\cdot}875$ in., say 32 in.

Tube Plates

Rule Thickness $= \frac{\text{Pressure}\times W\times D}{(D-d)\times C}$.

Where, W = Width of combustion chamber.
D = Pitch of tubes.
d = Inside diameter of tubes
C = 28000 (compressive stress limit).

Then, Thickness $= \frac{180\times 31{\cdot}25\times 4{\cdot}5}{(4{\cdot}5-2{\cdot}89)\times 28000} = {\cdot}56$ in., say $\frac{9}{16}$ in.

NOTE.—Tube thickness = 7 S.W.G., which is equal to ·176 in.

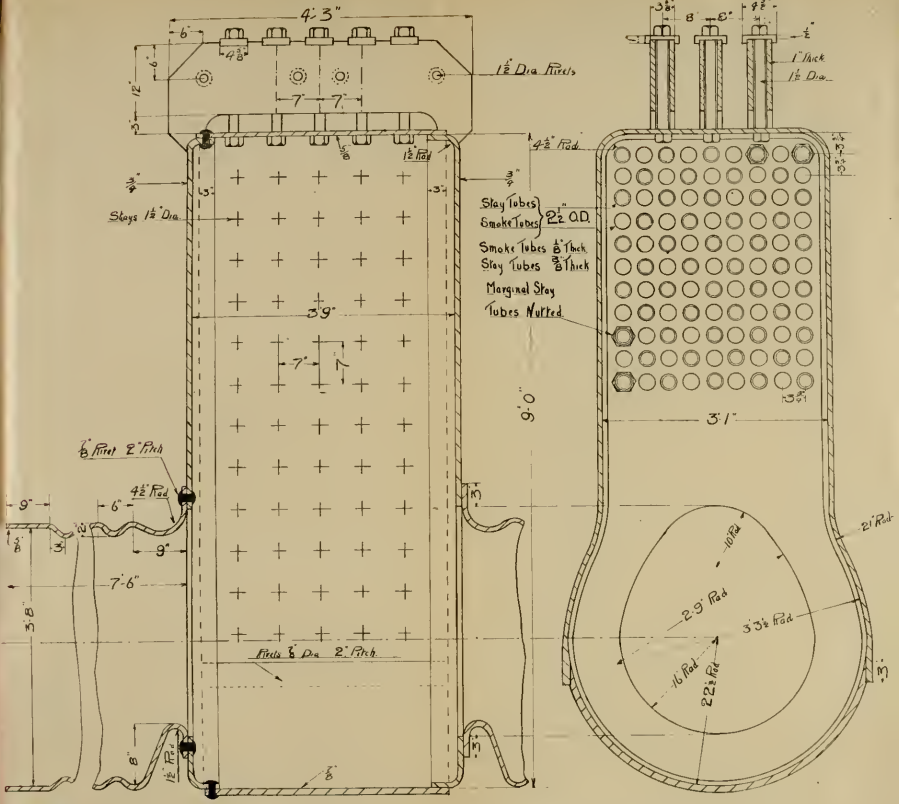
Therefore, $3{\cdot}25$ in. $-$ (·176 in. × 2) = 2·89 in. inside diameter of tubes

Diameter of Stays (Steel).

Rule, Stay area × 9000 = Surface × Pressure. As the pitch is unequal, being $7\frac{1}{4}$ in., by 8 in. the Surface $= \frac{7{\cdot}25''+8''}{2} = 58{\cdot}28$ sq. in

Then, Diameter of stays $= \sqrt{\frac{58{\cdot}28\times 180}{{\cdot}7854\times 9000}} = 1{\cdot}21$ in., or say $1\frac{1}{4}$ in. diameter.

To satisfy surveys other than the B. of T. the stays are, in the present example, increased to $1\frac{1}{2}$ in. diameter.

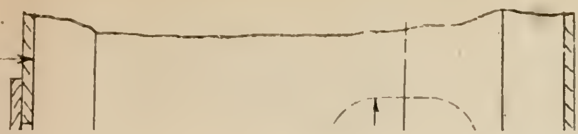


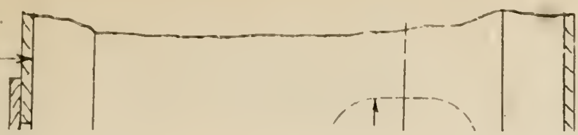
No. 2. Double-Ended Combustion Chamber of 8-Furnace Boiler.

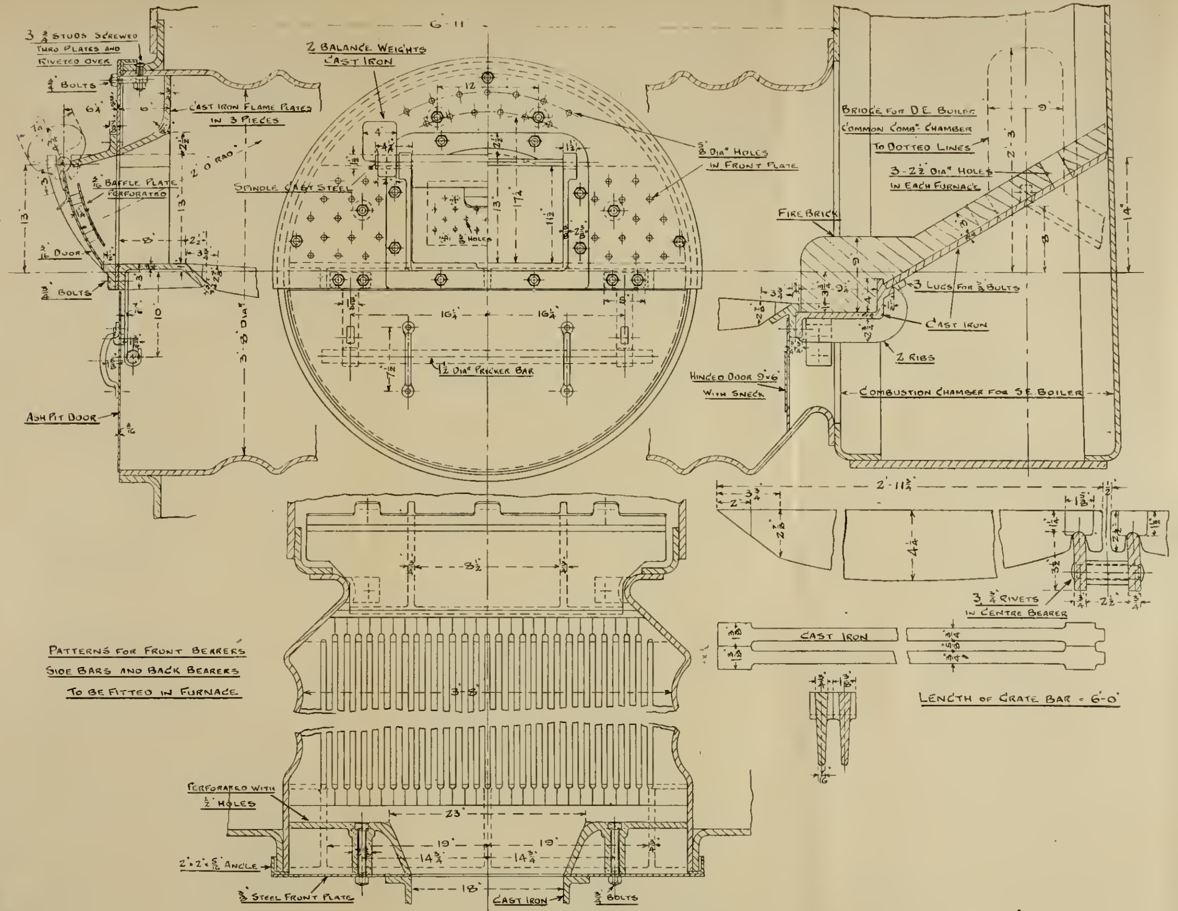
(Draw to a scale of 1 in. = 1 ft.)

(Pressure, 200 lbs.)

NOTE.—The calculations are similar to those of No. 1.

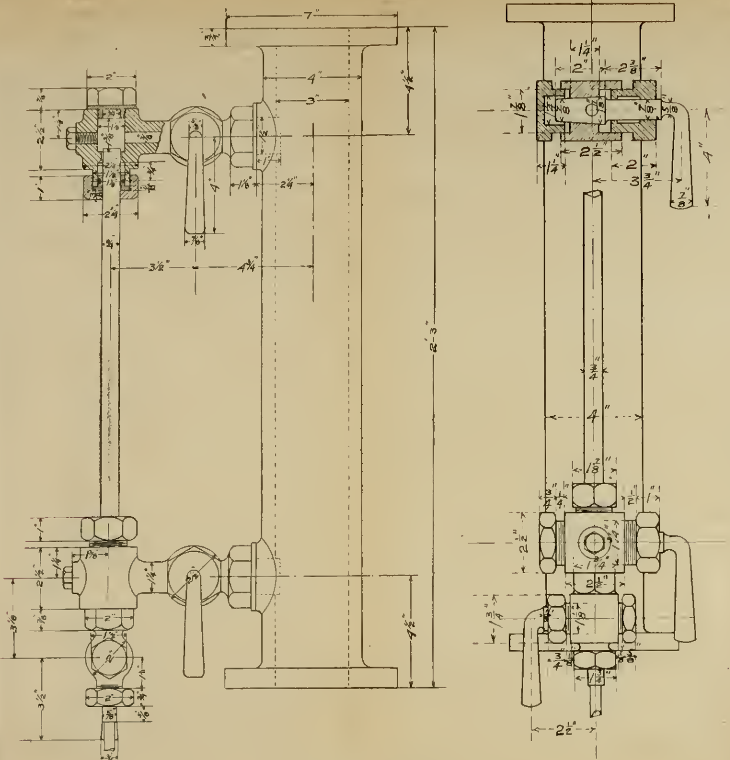






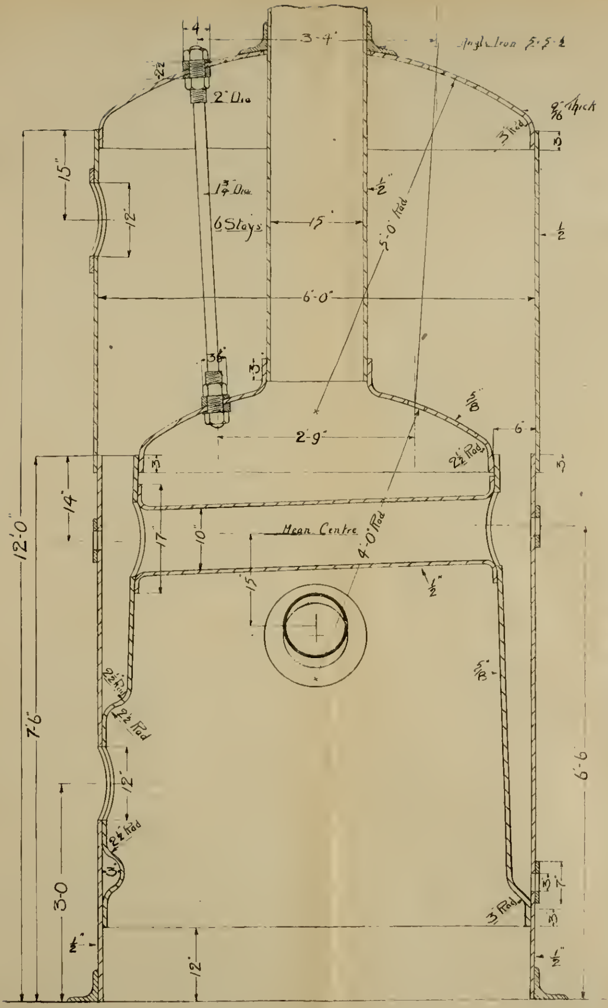
No. 3.—Furnace with Bars Complete.

(Draw to a scale of 1½ in. = 1 ft.)



No. 4.—Water Gauge Column.

(Draw to a scale of 6 in. = 1 ft.)



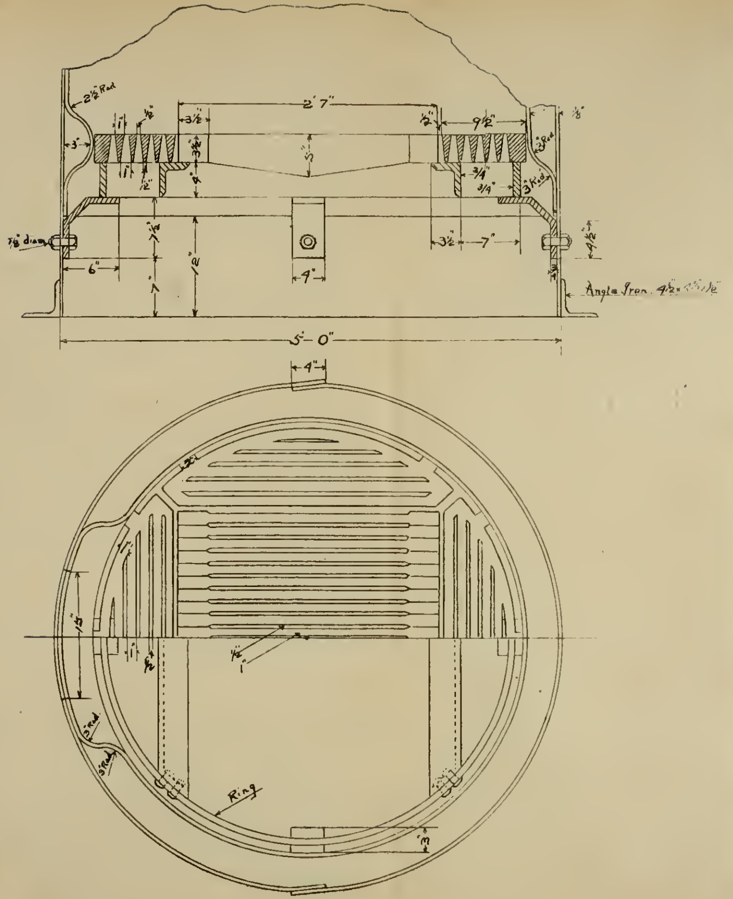
No. 5. Vertical Type Donkey Boiler.

(Draw to a scale of 1 in. = 1 ft.)

NOTE.—Vertical shell seams (double riveting)—Diameter of rivets, $\frac{7}{8}$ in.; Pitch, $2\frac{3}{4}$ in.
Circumferential shell seams (single riveting)—Diameter of rivets, $\frac{7}{8}$ in.; Pitch, 2 in.

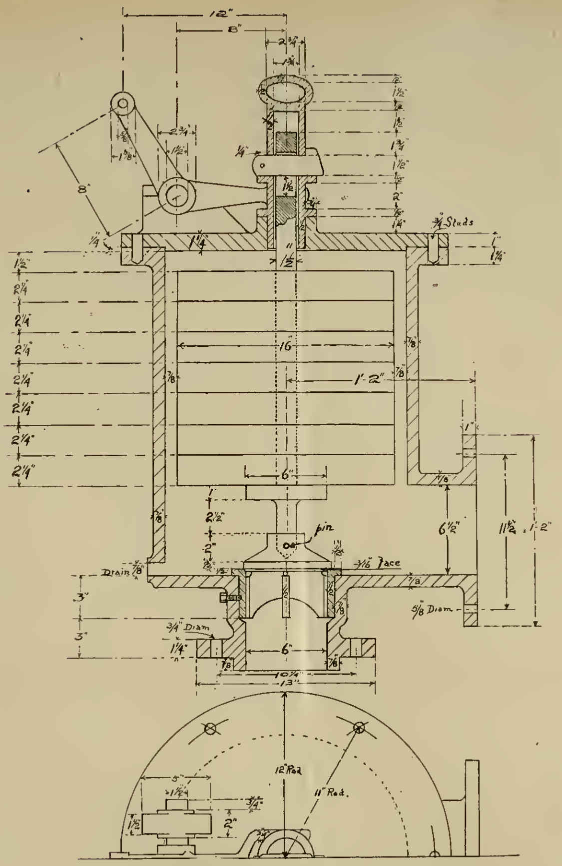
Pressure, 100 lbs. per sq. in.

7/8



No. 6.—Donkey Boiler Fire Bars.

(Draw to a scale of 1½ in. = 1 ft.)



No. 7.—Dead Weight Safety Valve.

(Scale, $1\frac{1}{2}$ in. = 1 ft.)

(Draw to a scale of 3 in. = 1 ft.)

Data.—Boiler grate surface consists of 2 furnaces, each 5 ft. by 3 ft. 4 in. Pressure, 30 lbs. (gauge).

Then, Total grate surface $= 2 \times 5 \times 3{\cdot}33 = 33{\cdot}\dot{3}$, say 34 sq. ft.

Rule, $37{\cdot}5 \div$ Absolute pressure = Valve area in sq. in. per sq. ft grate.

Therefore, $37{\cdot}5 \div (30 + 15) = {\cdot}833$ sq. in.,

and, Total valve area $= 34 \times {\cdot}833 = 28{\cdot}3$ sq. in.

Then, Diameter of valve $= \sqrt{\dfrac{28{\cdot}3}{{\cdot}7854}} = 6$ in. diameter.

and, Load on valve $= 6^2 \times {\cdot}7854 \times 30 = 848{\cdot}2$ lbs.

Assuming that weight of valve and spindle amount to, say, 20 lbs.,

Then, Actual load required $= 848{\cdot}2 - 20 = 828{\cdot}2$.

Again, Fix on diameter of weights as, say, 16 in.

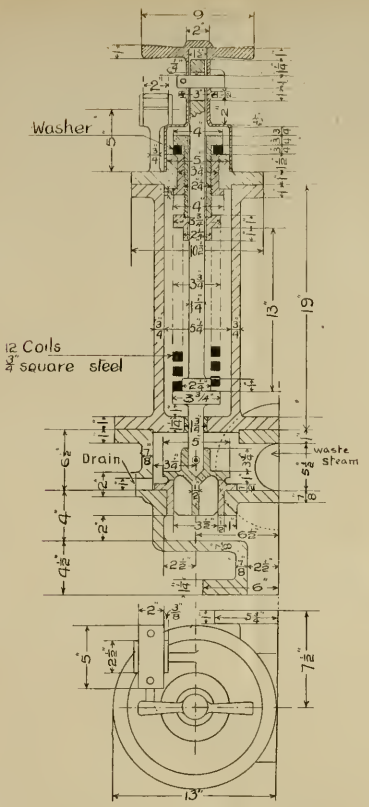
Then, Depth of weights $= \dfrac{828{\cdot}2}{{\cdot}26 \times 16^2 \times {\cdot}7854} = 15{\cdot}8$, say $15\frac{3}{4}$ in.

Allowing 7 separate weights.

Then, $\dfrac{15{\cdot}75 \text{ in.}}{7} = 2{\cdot}25$ in. $= 2\frac{1}{4}$ in. thickness of each weight.

Note.—If a pair of valves are fitted, each one only requires to be half the area of the single valve.

Or, Diameter of each valve $= \sqrt{\dfrac{6^2}{2}} = 4{\cdot}2$, say $4\frac{1}{4}$ in. diameter.



No. 8.—Pair of Spring-Loaded Safety Valves.

(Scale, $1\frac{1}{2}$ in. = 1 ft.)

(Draw to a scale of 3 in. = 1 ft.)

Data.—Pressure 160 lbs. gauge, 3 furnaces, each grate 5 ft. 6 in. by 3 ft. 6 in., forced draught, consumption 30 lbs. per. sq. ft. of grate per hour.

Then,

$\frac{37\cdot5}{160+15} = \cdot214$ sq. in valve area per sq. ft. of grate at 20 lbs. consumption per sq. ft. of grate.

$$\text{Total valve area} = \cdot214 \times 3 \times 5\cdot5 \times 3\cdot5 \times \frac{30}{20} = 18\cdot53 \text{ sq. in.}$$

$$\text{Diameter of each valve} = \sqrt{\frac{18\cdot4}{2\times\cdot7854}} = 3\cdot4 \text{ in., say } 3\tfrac{1}{2} \text{ in.}$$

Rule, $11000 \times d^3 = \text{Load on valve} \times \text{Mean diameter of coil.}$

Where, $11000 = $ Constant for square steel.

,, $d = $ Side of coil in inches.

Then, $11000 \times d^3 = 3\cdot5^2 \times \cdot7854 \times 160 \times 3$ in.

And, $d = \sqrt[3]{\frac{3\cdot5^2 \times \cdot7854 \times 160 \times 3}{11000}} = \cdot75$ in.

Notice that the *cube root* requires to be extracted after division.

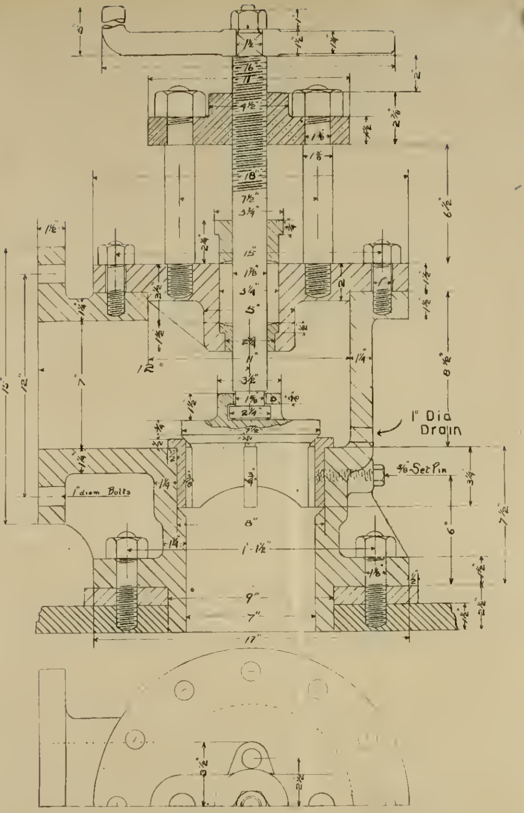
Diameter of boiler branch bore $= \sqrt{3\cdot5^2 \times 2} = 4\cdot9$ in., say 5 in. diameter.

Flange Studs.

Allow, say, 3000 lbs. per sq. in. stress on studs, and assume pressure to act out as far as the pitch centre line of the studs; also take pitch circle diameter as 10 in. or $10\frac{1}{2}$ in. and fix on number of studs as 6.

Then, $\text{Diameter of studs} = \sqrt{\frac{10\cdot5^2 \times 160}{3000 \times \cdot7854}} = \cdot9$, say 1 in. diameter.

Note.—Only one valve of the pair is drawn out.



No. 9 – Boiler Stop Valve.

(Draw to a scale of 2 in. = 1 ft.)

Data.—Pressure 160 lbs. (gauge). Heating surface (double ended boiler), 3200 sq. ft. Forced draught.

Rule, Diameter of valve $= 1{\cdot}6 \times \sqrt{\dfrac{\text{Heatiog surface}}{\text{Absolute pressure}}}$

Then, Diameter of valve $= 1{\cdot}6 \times \sqrt{\dfrac{3200}{160+15}} = 6{\cdot}83$ in., say, 7 in. diameter.

Rule, Diameter of spindle $= \dfrac{\text{Valve diameter}}{50} \times \sqrt{\text{Pressure}} + {\cdot}12.$

Then, Diameter of spindle $= \dfrac{7}{50} \times \sqrt{160} + {\cdot}12 = 1{\cdot}78$ in., say $1\frac{7}{8}$ in.

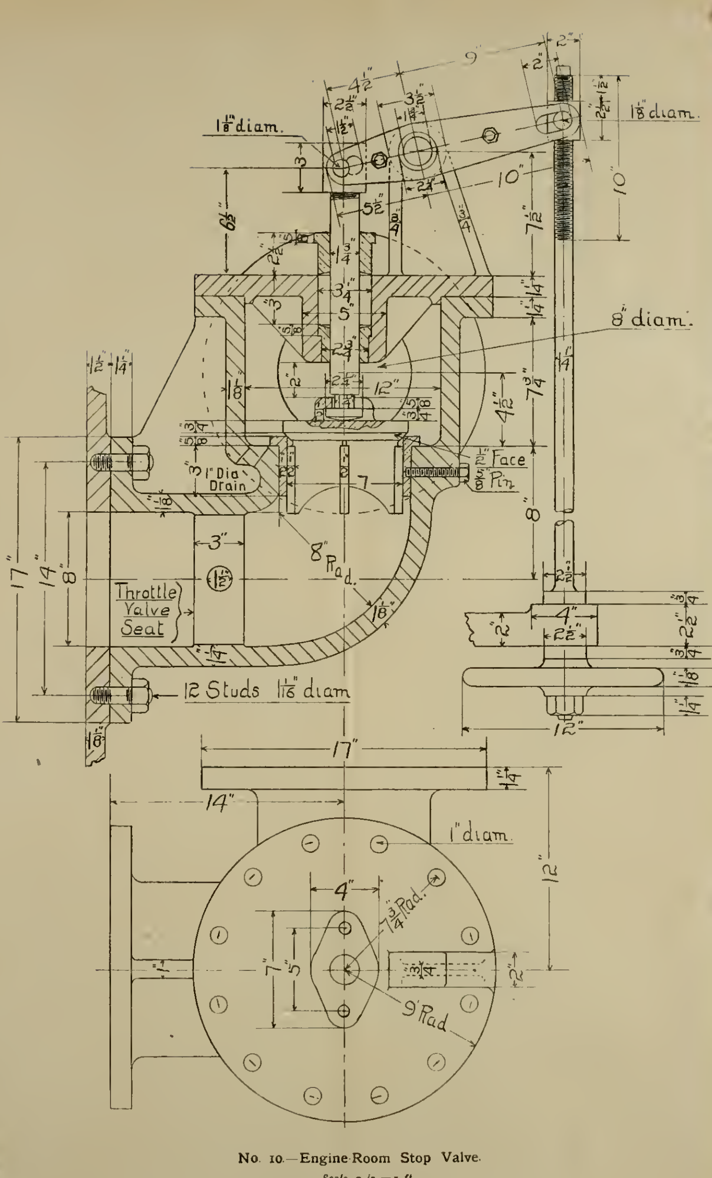
Cover Studs.

Assume that 1 in. diameter studs are decided on for the cover, and the stress on each is not to exceed, say, 2000 lbs. per sq. in.

Then, Number of studs $= \dfrac{11^2 \times 160}{1^2 \times 2000} = 9{\cdot}6$ studs.

To allow for a wide margin of safety it will be better to allow, say, 12 studs of 1 in. diameter.

NOTE.—Inside diameter of chest = 11 in.

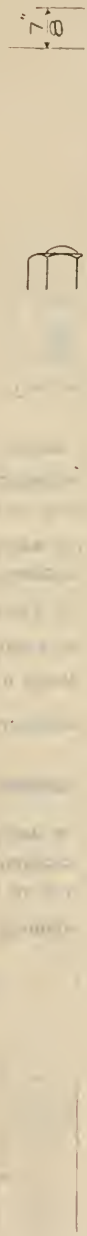


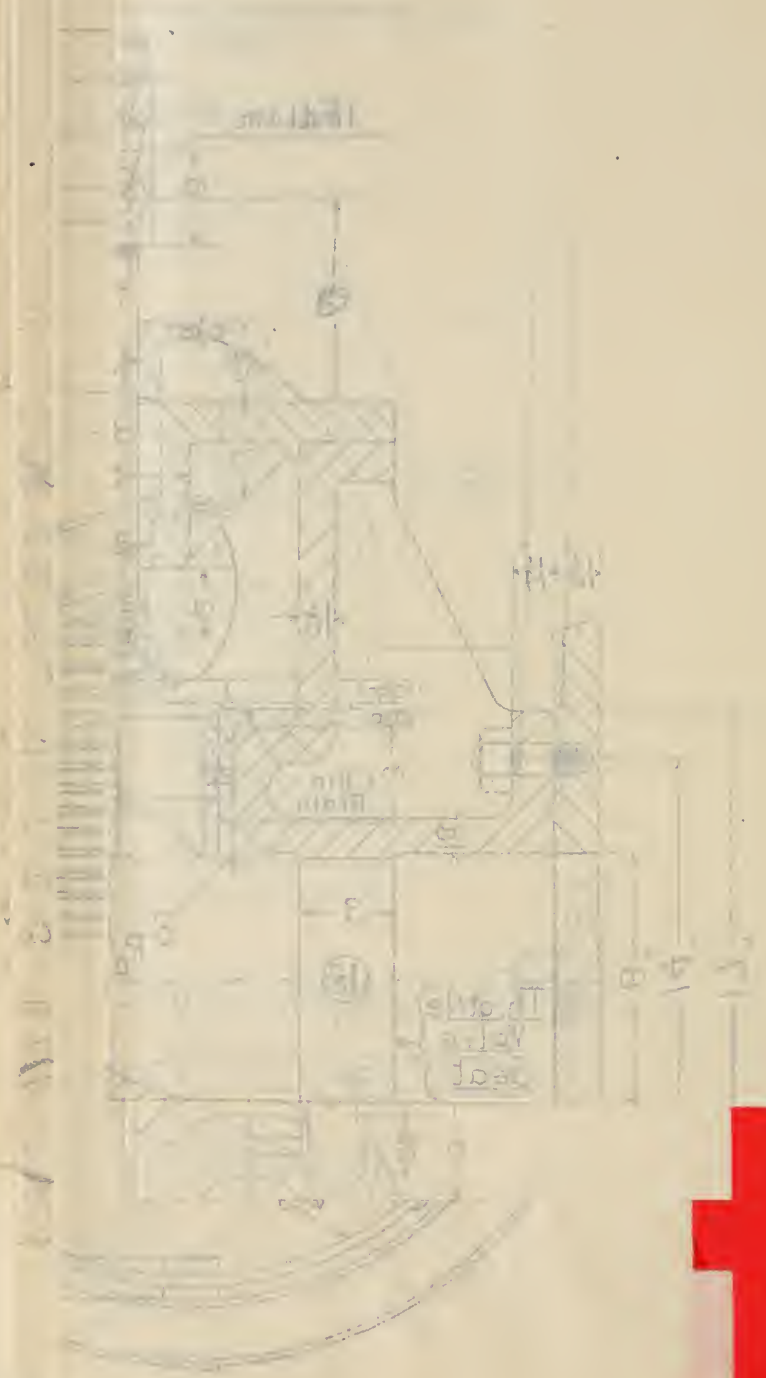
No. 10.—Engine-Room Stop Valve.

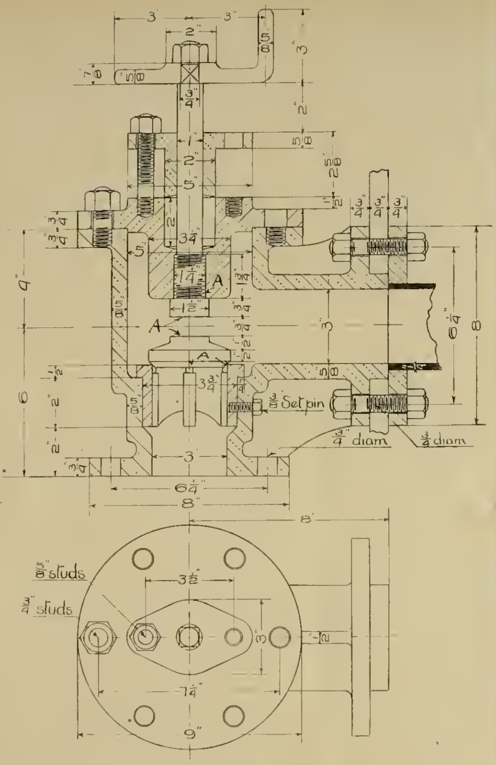
Scale, 2 in. = 1 ft.

(Draw to a scale of 3 in. = 1 ft.)

NOTE.—The calculations are similar to those of the boiler stop valve, No. 9.


7/8"


Throttle
Valve
Seat



No. 11.—Feed Check Valve.

(Scale, 3 in. = 1 ft.)

Data.—Pressure, 200 lbs. Valve, 3 in. diameter.

Allow diameter of chest, inside, to be not less than $1\frac{1}{2}$ times diameter of valve.

Then, 3 in. × 1·5 = 4·5 in.; say 5 in. diameter.
Diameter of spindle = diameter of valve ÷ 3 = 3 in. ÷ 3 = 1 in. diameter.
Allow a tensile stress of 3000 sq. in. on studs, and fixing on, say, 6 studs.

Then, $\text{Diameter of studs} = \sqrt{\dfrac{\text{diameter of cover (inside)}^2 \times \text{pressure}}{3000 \times \text{No. of studs}}}$

$\text{Diameter of studs} = \sqrt{\dfrac{5^2 \times 200}{2500 \times 6}} = \cdot 57 \text{ in.}$

To allow of a good safety margin fix on, say, $\frac{5}{8}$ in. diameter studs.
Allow width of flange for studs = stud diameter × 3,

Then, ·625 × 3 = 1·875 in., say 2 in. on each side.

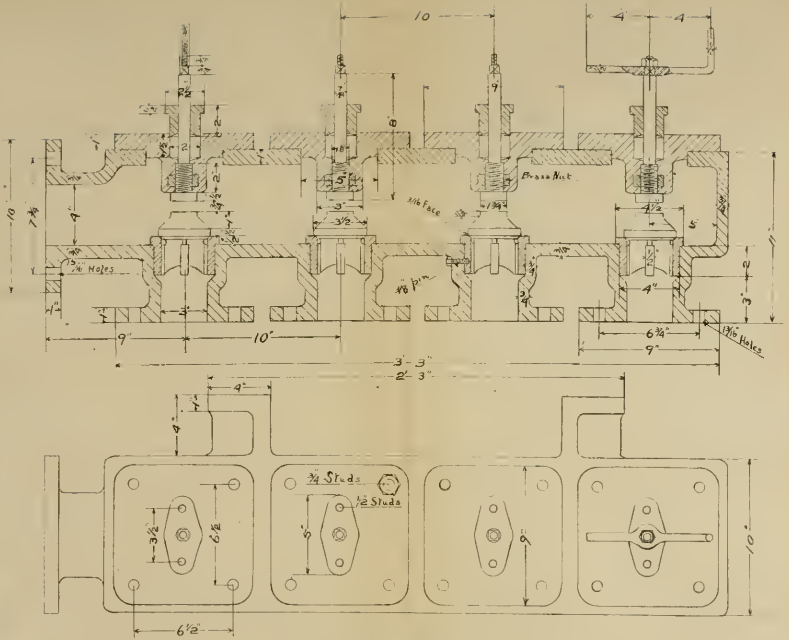
Then, Diameter of cover = 5 in. + 2 in. + 2 in. = 9 in.
Allow full lift clearance of valve equal to $\frac{1}{4}$ diameter.

Then, 3 in. ÷ 4 = ·75 in. lift clearance.

The thickness of the chest is taken as $\frac{5}{8}$ in.; for a larger valve allow $\frac{3}{4}$ in.

NOTE.—Allow the thickness of the flanges to always be in excess of the chest thickness

NOTE.—The wearing parts are marked A.



No. 12.—Bilge Suction Box.

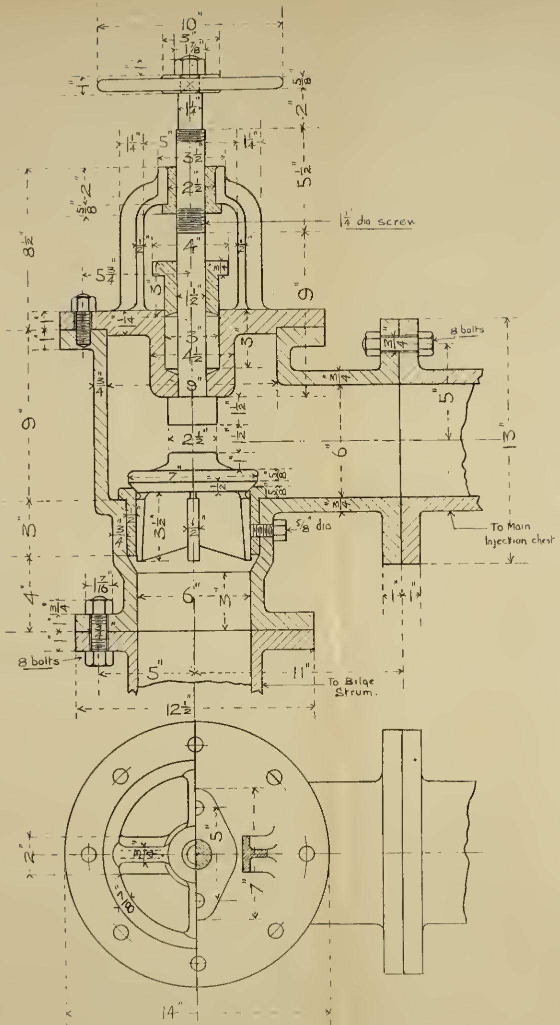
(Draw to a scale of 3 in. = 1 ft.)

then,
Allow di
Allow pi
Then,

Then,

then,
Allow di
Allow pi
Then,

Then,



No. 12a.—Bilge Injection Valve.

(Draw to a scale of 4 in. = 1 ft.)

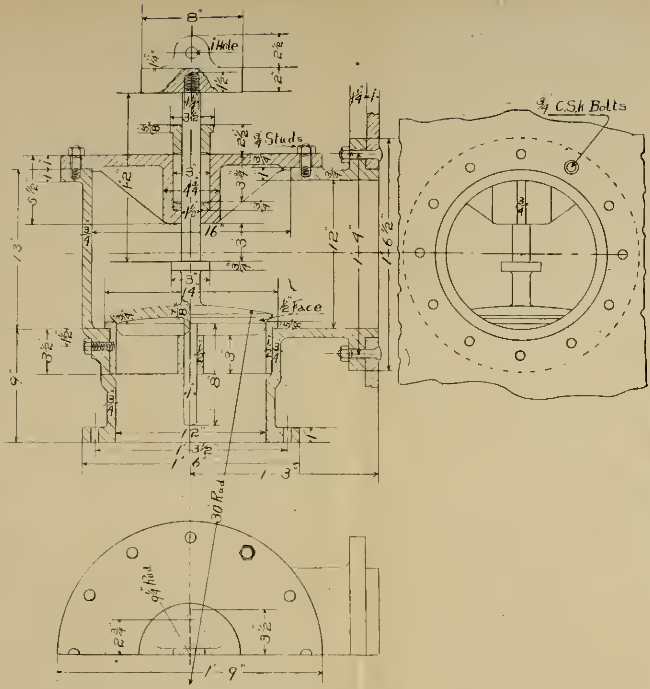
No special calculations are required for this drawing as the stresses on the working parts are low. Full lift clearance of valve = diameter ÷ 4 — 6 in. ÷ 4 = 1½ in.

then,
Allow di
Allow pi
Then,

Then,

then,
Allow di
Allow pi
Then,

Then,



No. 13.—Side Discharge Valve.

(Scale, 1½ in. = 1 ft.)

(Draw to a scale of 2 in. = 1 ft.)

As the pressure on this type of valve is low, it is sufficient if the various parts are of light construction as shown above. The inside diameter of the chest is equal to about 1·3 times the diameter of the valve.

Width of flange = stud diameter × 3 = ·75 × 3 = 2·25 in.

Then, Diameter of cover = 2·25 in. + 16 in. + 2·25 in. = 20½ in., or, say, 21 in.

Allow full lift equal to ¼ diameter of valve, or 3 in.

The studs in the flange on ship's side are screwed through the plate, countersunk, and rivetted over on the outside, with clearing holes in the chest flange inside.

1" I

2" Rad

then,
Allow di
Allow pit
Then,

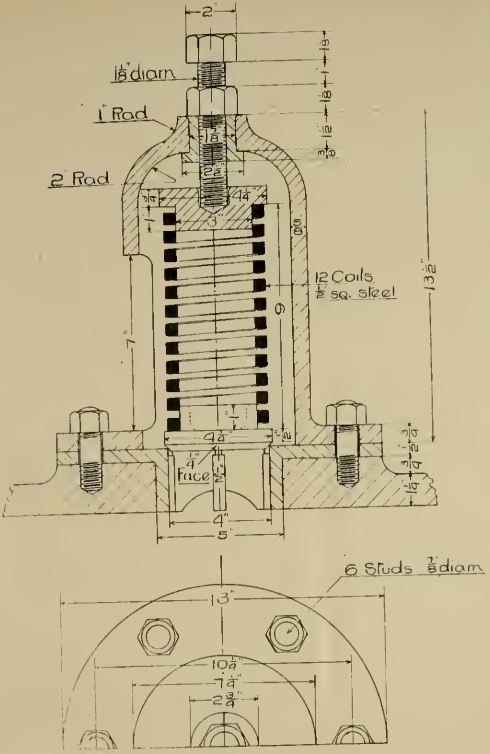
Then,

1" I

2" Rad

then,
Allow di
Allow pi
Then,

Then,



No. 14. Cylinder Relief Valve.
(*Scale*, 3 *in.* = 1 *ft.*)
(Draw to a scale of 4 in. = 1 ft.)

Data.—**Valve, 4 in. diameter. Pressure, 20 lbs.**

Fix on mean diameter of spring as being about equal to diameter of valve or, say, 3½ in

Then, Rule, $11000 \times T^3 =$ **Mean diameter of coil** $\times$ **Load on valve.**

„ $11000 \times T^3 = 3{\cdot}5 \times 4^2 \times {\cdot}7854 \times 20.$

Then, $$T^3 = \frac{3{\cdot}5 \times 4^2 \times {\cdot}7854 \times 20}{11000} = {\cdot}0799$$

$$T^2 = \sqrt{{\cdot}0799} = {\cdot}43 \text{ in., or, say, } \tfrac{1}{2} \text{ in. coil.}$$

Observe that *cube root* extraction is required.
Allow 6 studs of ample strength, say ¾ in. or ⅞ in. diameter.

Diameter of spindle = Valve diameter ÷ 3.

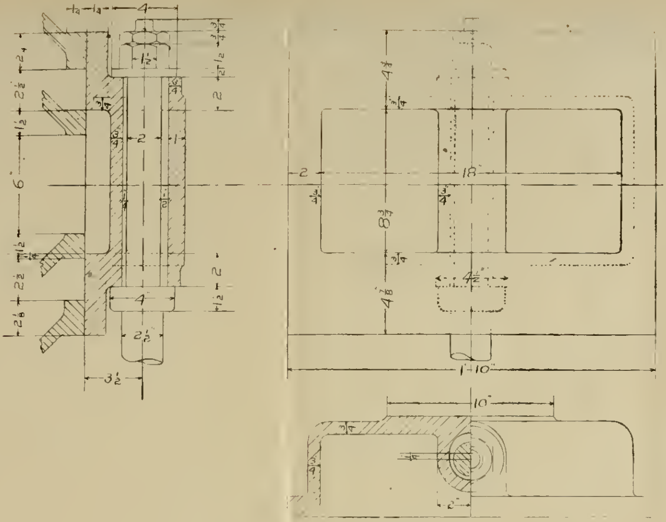
„ „ = 4 in. ÷ 3 = 1·11, say 1⅛ in. diameter.

then,
Allow di
Allow pi
Then,

Then,

then,
Allow di
Allow pi
Then,

Then,



No. 15.—Slide Valve and Spindle.

(Scale, 2 in. = 1 ft.).

(Draw to a scale of 3 in. = 1 ft.).

Data.—Cylinder ports = $2\frac{1}{2}$ io. steam, 6 in. exhaust.
Centre bars = $1\frac{1}{2}$ in.
Valve travel = 7 in.
Top lead = $\frac{1}{8}$ in.
Bottom lead = $\frac{1}{4}$ in
Top exhaust lap = 0 in.
Bottom exhaust lap = $+\frac{1}{4}$ in.
Pressure = 56 lbs.
Cylinder diameter = 24 in.
Stroke = 36 in.

Assume that, Top steam port opening = $1\frac{1}{4}$ in.
Then, Bottom steam port opening = $1\frac{3}{8}$ in.

As the bottom lead is $\frac{1}{8}$ in. more than top lead, the bottom port opening is also $\frac{1}{8}$ in. more than top port opening.

Rule, $\frac{1}{2}$ travel = Steam lap + Lead (either top or bottom).

Theo,

(Top), 7 in. ÷ 2 = 3·5 in., and 3·5 in. − 1·25 = 2·25 steam lap.

And,

(Bottom), 7 in. ÷ 2 = 3·5 in., aod 3·5 − 1·375 = 2·125 steam lap.

Length of ports = Diameter of cylinder × ·75.
,, ,, = 24 in. × ·75 = 18 in.

Allow bearing width at sides as, say, 2 in.

Then,

Depth of valve face = $2\frac{1}{4} + 2\frac{1}{2} + 1\frac{1}{2} + 6 + 1\frac{1}{2} + 2\frac{1}{2} + 2\frac{1}{8}$ = 19 in.

And, Width of valve face = 2 in. + 18 + 2 = 22 in.

Rule, Diameter at screw = $\sqrt{\dfrac{\text{Face area} \times \text{Pressure} \times \cdot 2}{3500 \times \cdot 7854}}$

Allow ·2 for friction, and 3500 lbs. tensile stress per sq. in.

Then,

Diam. at screw = $\sqrt{\dfrac{22 \times 19 \times 56 \times \cdot 2}{3500 \times \cdot 7854}}$ = 1·3 in., say $1\frac{1}{2}$ in. diam

Diameter of spindle below screw = 1·5 in. × 1·33 = 1·99 in., say 2 in. diameter.

Diameter of spindle in gland = 1·5 in. × 1·6 = 2·4 in., say $2\frac{1}{2}$ in. diameter.

NOTE.—Allow $\frac{1}{4}$ in. clear in front of spindle and $\frac{1}{2}$ in. clear at back of spindle to allow for wear in.

Bottom exhaust opening with crank on top centre = Top steam lap + Lead − Bottom exhaust lap.

Then, 2·25 + ·125 − ·25 = 2·125 opening to exhaust.

Areas of piston and steam port opening compared (for bottom).

Then, $\dfrac{24^2 \times \cdot 7854}{18 \times 1\cdot 375}$ = 18, or $\frac{1}{18}$.

Area of piston and exhaust port opening compared (crank on top).

Then, $\dfrac{24^2 \times \cdot 7854}{18 \times 2\cdot 125}$ = 11, or $\frac{1}{11}$.

Data

Valve Travel 7"	7
Lead	,

then,
Allow di
Allow pi
Then,

Then,

Data

Valve Travel 7"	7
Lead	.

then,

Allow di

Allow pi

Then,

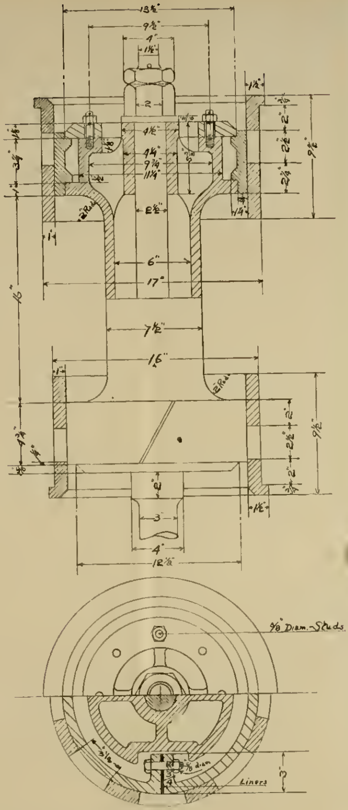
Then,

Data

Valve Travel 7"	Top	Btm.
Lead	1/8"	3/8"
Steam lap	2 1/4"	2"
Port opening	1 1/4"	1 1/2"
Exhaust lap	0"	1/4"

Diam of Cylinder is 25"

Stroke of piston is 36"



No 16—Inside Steam Piston Valve.

(Draw to a scale of 3 in. = 1 ft.)

Rule Diameter of valve rod at body = Valve diameter × ·15

Then, ,, ,, ,, = 15 in. × ·15 = 2·25 in., say 2¼ in.

Half valve travel = Steam lap + Port opening.

Then, ,, ,, = (Top) 2¼ in + 1¼ in. = 3½ in.
,, ,, = (Bot.) 2 in + 1½ in. = 3½ in. } 7 in. travel.

Observe that the difference in lead top and bottom is also equal to the difference in steam lap and port opening top and bottom (that is, ¼ in. difference). Again, notice that the sum of "steam lap + lead" is the *same* for top and bottom.

Thus, Top, 2¼ in. + ⅛ in. = 2⅜ in.

and Bottom, 2 in. + ⅜ in. = 2⅜ in.

then,
Allow di
Allow pi
Then,

Then,

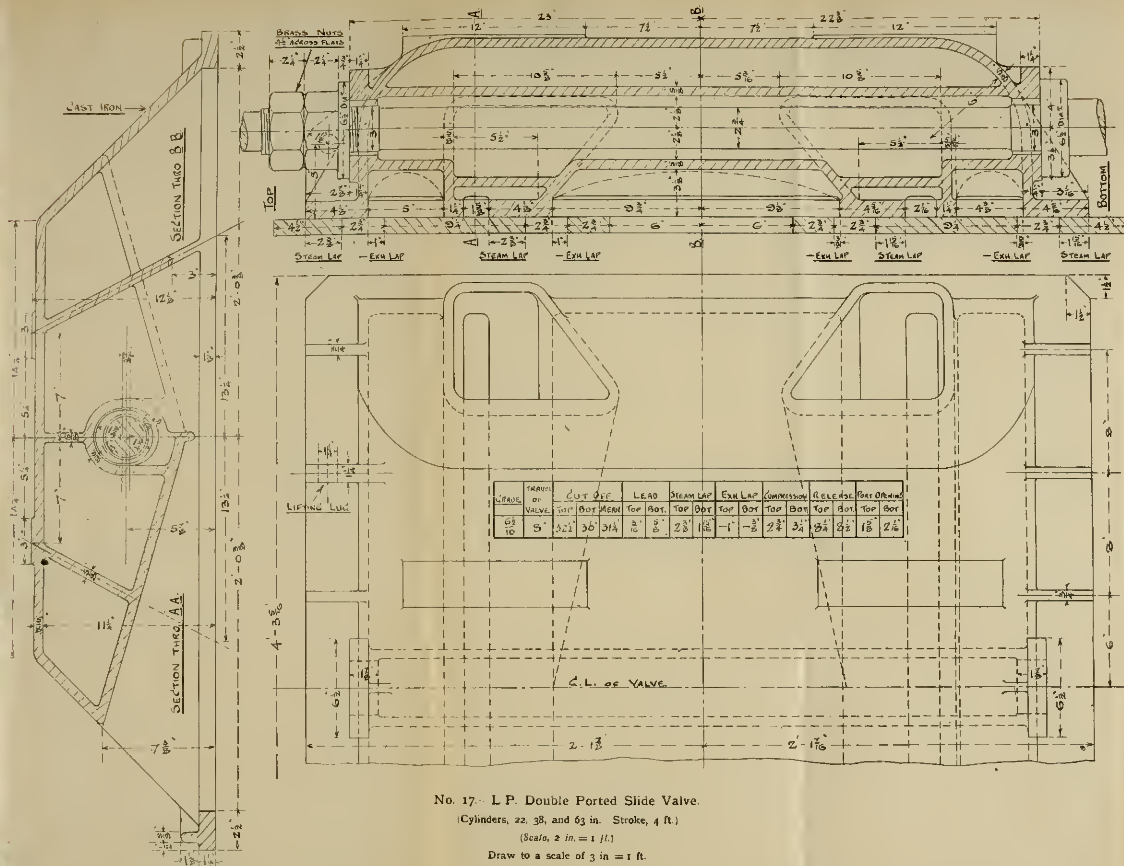
then,

Allow di

Allow pi

Then,

Then,



GRADE	TRAVEL OF VALVE	CUT OFF			LEAD		STEAM LAP		EXH LAP		COMPRESSION		RELEASE		PORT OPENINGS	
		TOP	BOT	MEAN	TOP	BOT.	TOP	BOT	TOP	BOT	TOP	BOT	TOP	BOT.	TOP	BOT
$\frac{6\frac{1}{2}}{10}$	8"	$32\frac{1}{2}$"	36"	$31\frac{1}{4}$"	$\frac{3}{16}$"	$\frac{5}{8}$"	$2\frac{3}{8}$"	$1\frac{15}{16}$"	-1"	$-\frac{3}{8}$"	$2\frac{3}{4}$"	$3\frac{1}{4}$"	$8\frac{1}{4}$"	$8\frac{1}{2}$"	$1\frac{5}{8}$"	$2\frac{1}{16}$"

No. 17.—L P. Double Ported Slide Valve.

(Cylinders, 22, 38, and 63 in. Stroke, 4 ft.)

(*Scale, 2 in. = 1 ft.*)

Draw to a scale of 3 in = 1 ft.

Data.—Cylinder, 63 in. ; Stroke, 4 ft. ; Revolutions, 82. Exhaust steam velocity to be 130 ft. per sec. Length of ports = ·77 of cylinder diameter.

Then, Port length = 63 · ·77 = 48·51 in., say $48\frac{3}{4}$ in

Rule

Piston area × twice stroke in ft × Revolutions = Exhaust steam speed per min × Port area

Then. $63^2 \times \cdot 7854 \times 4 \times 2 \times 82 = 130 \times 60 \times$ port area.

∴ $\text{Port area} = \frac{63^2 \times \cdot 7854 \times 4 \times 2 \times 82}{130 \times 60} = 262$ sq. in.

„ depth = 262 ÷ 48·75 = 5·37 in., say $5\frac{1}{2}$ in

As the valve is double-ported,

then, 5·5 ÷ 2 = 2·75 in., depth of each port.

NOTE.—It should be noted that steam ports only open part width for steam admission, but open full for exhaust ; it is therefore advisable to calculate the actual port sizes from the exhaust data

Half travel = Steam lap + Port opeoing

Then, Top 8 in ÷ 2 = 4 in., and 4 in. − $2\frac{3}{8}$ in. = $1\frac{5}{8}$ in. port opening for steam.

Again, Bottom 8 in. ÷ 2 = 4, and 4 − $1\frac{15}{16}$ in. = $2\frac{1}{16}$ in. port opening for steam

The difference in steam lap and port opening top and bottom is exactly equal to the lead difference top and bottom.

Thus, Lead Bottom = $\frac{5}{8}$ in

Top = $\frac{3}{16}$ in.

$\frac{7}{16}$ difference

So that the steam lap difference will be $\frac{7}{16}$ in., and the port opening difference also $\frac{7}{16}$ in., as shown io the valve data given.

It should be carefully noted that the inner ports of the valve only *admit steam* to the cylinder, and do not take exhaust steam ; it is therefore sufficient to make the depth of these ports equal to the port openings for steam.

For the top this is, as previously shown, $1\frac{5}{8}$ in., and for the bottom $2\frac{1}{16}$ in. which are, therefore, the sizes of the ports referred to.

Depth of Large Bars.

Depth = Steam lap + Port opening + Metal thickness + Half travel.

∴ $= 2\frac{3}{8} + 1\frac{5}{8} + 1\frac{1}{4} + 4$ in. $= 9\frac{1}{4}$ in.

The depth of the cylinder exhaust port is of no consequence, provided that it is at least equal to twice that of the steam ports.

Diameter of Valve Rod (Bottom of Thread).

Rule. $\sqrt{\frac{\text{Face Area} \times 30 \times \cdot 2}{2600 \times \cdot 7854}} = \text{Diameter.}$

NOTE.—Allow 30 lbs. as the pressure on L P. valves.

„ 2600 lbs. tensile stress per sq in.

„ ·2 for friction.

Then, $\text{Diameter} = \sqrt{\frac{52 \times 50 \times 30 \times \cdot 2}{2600 \times \cdot 7854}} = 2{\cdot}76$, say $2\frac{3}{4}$ in. diameter.

NOTE.—The valve face measures roughly 50 in. by 52 in Allow diameter of rod at gland = diameter at screw × 1·4

Therefore, $2{\cdot}75 \times 1{\cdot}4 = 3{\cdot}85$, say $3\frac{7}{8}$ in. diameter.

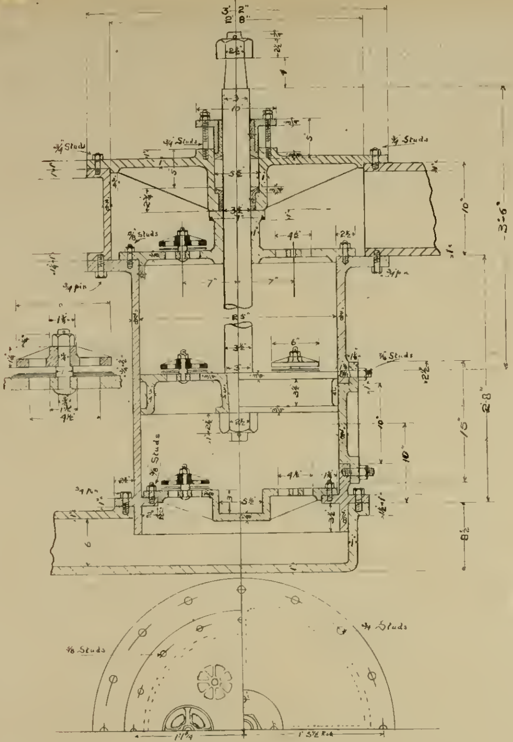
The rod is swelled out at top and bottom to 3 in diameter, which forms the outside diameter of the screw Allow spindle $\frac{1}{4}$ in. clear at front and $\frac{1}{2}$ io clear at back for wear in.

then,
Allow di
Allow pit
Then,

Then,

then,
Allow di
Allow pi
Then,

Then,



No. 18.—Air Pump Complete.

(Draw to a scale of 2 in. = 1 ft.)

Data.—L.P. cylinder, 74 in. diameter. Stroke, 42 in. Air pump stroke, 20 in. Then to find diameter of pump.

Rule. Air pump capacity $= \dfrac{\text{L.P. cylinder capacity}}{20}$.

Then, Air pump diameter $= \sqrt{\dfrac{74^2 \times 42 \text{ in.}}{20 \times 20 \text{ in.}}} = 24$, say 25 in.

NOTE.—·7854 cancels out top and bottom.

Rule, Diameter of pump rod $= \dfrac{\text{Diameter of pump}}{9} + {\cdot}6$.

Then, " " " $= \dfrac{25}{9} + {\cdot}6 = 3{\cdot}37$ in., say $3\frac{1}{2}$ in.

Rule, Thickness of barrel $= \dfrac{\text{Diameter of pump}}{60} + {\cdot}3$.

Then, " " " $= \dfrac{25}{60} + {\cdot}3 = {\cdot}716$ in., say $\frac{3}{4}$ in.

Length of barrel *Inside* = Stroke + Bucket + Valve depth + Clearance.

" " " = 20 + 5 in. + $3\frac{1}{2}$ + 2 in. = $30\frac{1}{2}$ in.

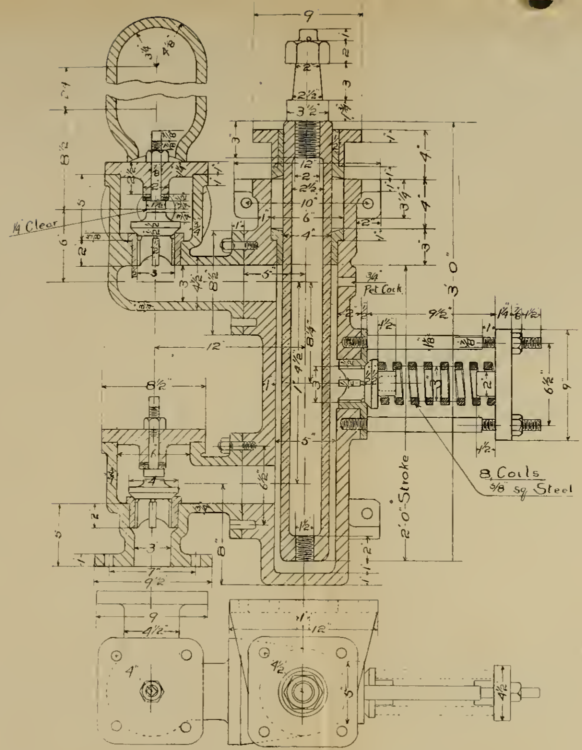
NOTE.—Allow, say, 1 in. clearance top and bottom.

then,
Allow di
Allow pi
Then,

Then,

then,
Allow di
Allow pi
Then,

Then,



No. 19.—Feed Pump Complete with Air Vessel.

(Draw to a scale of 3 in. = 1 ft.)

Data.—I. H. P., 1500. Engine revolutions or pump strokes, 65 per min. Stroke of pump, 24 in. Steam or feed water consumption taken as 15 lbs per I. H. P. hour. Assume pump efficiency as equal to 50 per cent., or ·50. Then to find diameter of each feed pump of a pair.

Pump Plunger.

Rule. I. H. P. × lbs. water × 27·66 = Diameter² × ·7854 × Stroke in. × Strokes per min. × 60 × Efficiency.

Therefore, $1500 \times 15 \times 27{\cdot}66 = \text{Diameter}^2 \times {\cdot}7854 \times 24 \text{ in.} \times 65 \times 60 \times {\cdot}50$,

so that $$\text{Diameter}^2 = \frac{1500 \times 15 \times 27{\cdot}66}{{\cdot}7854 \times 24 \times 65 \times 60 \times {\cdot}50} = 16{\cdot}9.$$

and $\text{Diameter} \times \sqrt{16{\cdot}9} = 4$ in. diameter of each feed pump plunger.

Note.—27·66 cub. in. of fresh water = 1 lb.

Relief Valve Spring.

Rule. $11000 \times d^3 =$ Load on valve × Mean diameter of spring. Assume a maximum loading pressure of, say, 125 lbs. per sq. in.

Then $$11000 \times d^3 = 3^2 \times {\cdot}7854 \times 125 \times 3,$$

so that $$d^3 = \frac{3^2 \times {\cdot}7854 \times 125 \times 3}{11000} = {\cdot}24,$$

and $d = \sqrt[3]{{\cdot}24} = {\cdot}62$ in., say, $\frac{5}{8}$ in. square steel.

Note.—11000 = constant for square steel.
,, d = side of square of steel spring.
,, Mean diameter of coil = 3 in.

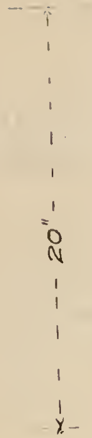
Air Vessel.

Allow capacity of air vessel to be equal to 1·7 times capacity of the pump chamber, and fix diameter at, say, $6\frac{1}{2}$ in.

Then, Capacity of air vessel (exclusive of curvature top and bottom) $= 5^2 \times {\cdot}7854 \times 24 \text{ in.} \times 1{\cdot}7 = 800$ cub. in.

Note.— 5 in. = inside diameter of pump barrel.
,, 24 in. = working stroke of pump.

Then, $$\text{Height of air vessel} = \frac{800}{6{\cdot}5^2 \times {\cdot}7854} = 24 \text{ in}$$

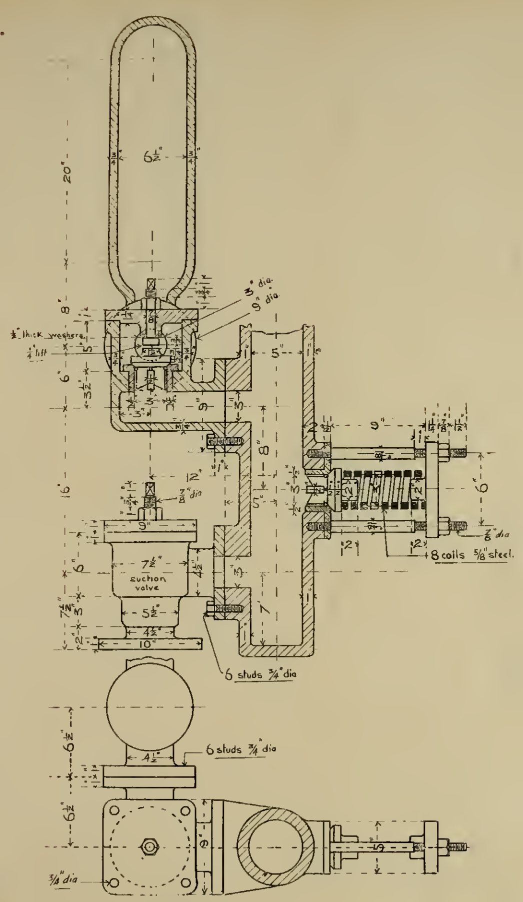


Then,
then,
then,
Allow di
Allow pi
Then,

Then,

Then,
then,
then,
Allow di
Allow pi
Then,

Then,



No. 19a. Feed Relief, Air Vessel, and Pump Valves.

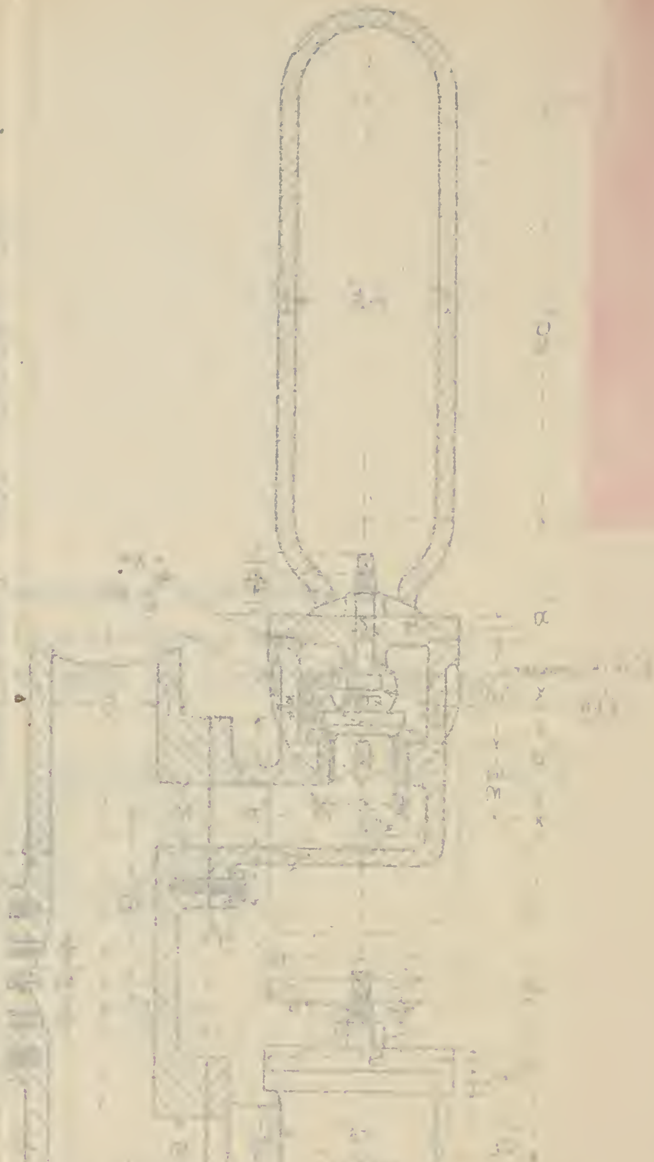
(Draw to a scale of 3 in. = 1 ft.)

See Drawing 19 No. for calculations and data.

Runsbottom
$\frac{3}{4}$" Square.

Then,
then,
then,
Allow di
Allow pi
Then,

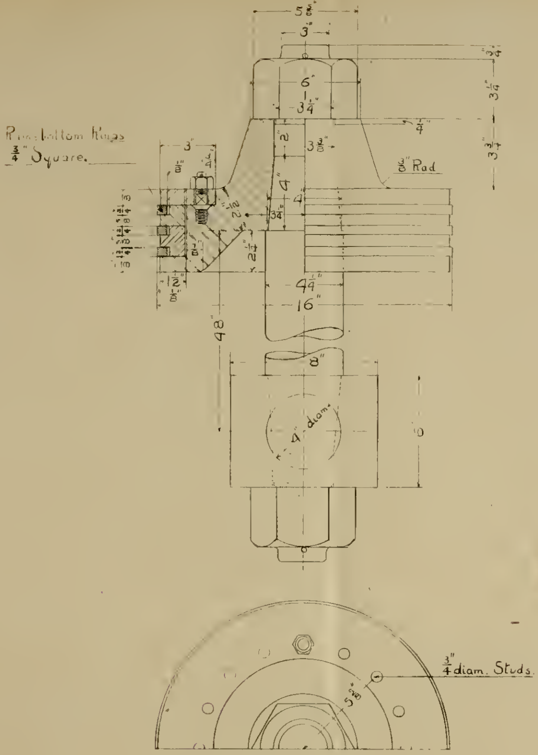
Then,



Runs bottom
$\frac{3}{4}$" Square.

Then,
then,
then,
Allow di
Allow pi
Then,

Then,



No. 20.—H.P. Piston and Rod (Cast Steel).

(*Scale, 2 in. = 1 ft.*).

(Draw to a scale of 2 in = 1 ft.).

Data.—Piston, 16 in. diameter.
Pressure, 180 lbs.

$$\text{Diameter of rod at screw (bottom of thread)} = \sqrt{\frac{\text{Piston area} \times \text{Pressure}}{5000 \times \cdot 7854}}.$$

$$\text{Diameter of rod at body} = \sqrt{\frac{\text{Piston area} \times \text{Pressure}}{3000 \times \cdot 7854}}.$$

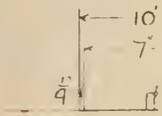
Allow, 5000 lbs. sq. in. for tensile stress limit.
,, 3000 lbs. sq. in. for compressive stress limit.

Then, $\dfrac{\text{Diameter of screw}}{\text{(bottom of thread)}} = \sqrt{\dfrac{16^2 \times 180}{5000}} = 3$ in. diameter.

,, Diameter of rod $= \sqrt{\dfrac{16^2 \times 180}{3000}} = 3\cdot 9$ in., or, say, $4\frac{1}{4}$ in., which allows of a good margin of safety.

Allow, Depth of piston at boss = Rod diameter × 1·5.
Then, Depth = 4·25 × 1·5 = 6·3 in., or, say, 6 in. only.
Allow ⅓ of this as taper, and make the remainder parallel.
Allow a taper of ⅝ in. for 4 in. of length as shown.

The number and size of studs allowed provide ample strength for the junk ring.
A ⅛ in. shoulder is allowed for the fitting of the piston on the rod.



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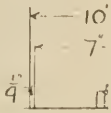

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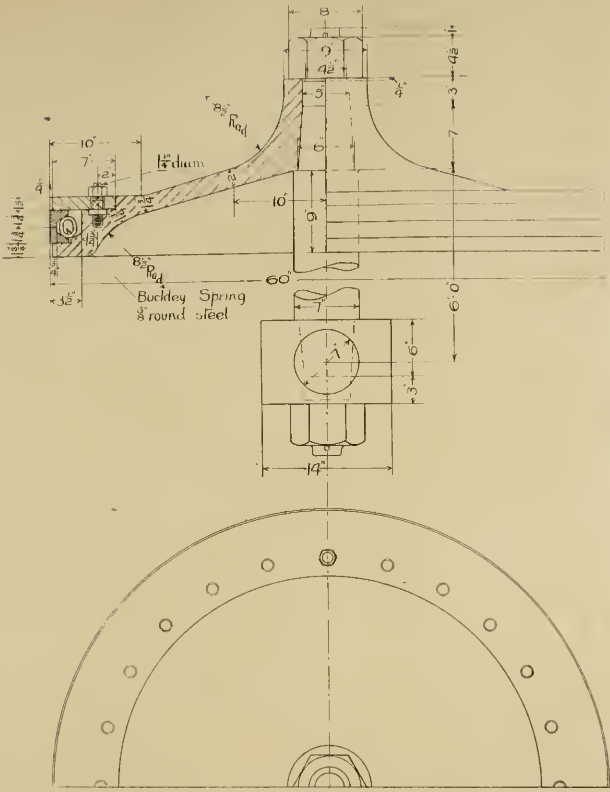


No. 21. L.P. Piston and Rod (Cast Steel).

(Scale, 1 in. = 1 ft.).

(Draw to a scale of 2 in. = 1 ft.).

Data.—Cylinder, 60 in. diameter. Pressure, 25 lbs.

Diameter of rod at screw (bottom of thread) $= \sqrt{\frac{60'' \times 25}{4400}} = 4{\cdot}5$ in. diameter.

Diameter of rod at body $= \sqrt{\frac{60'' \times 25}{2000}} = 6{\cdot}7$ in., or, say, 7 in diameter.

Allow 4400 lbs. per sq. in. for tensile stress limit (large piston rods).
,, 2000 ,, ,, compressive ,, (,, ,,)

Depth of piston at boss = Rod diameter × 1·5.
,, ,, = 7 × 1·5 = 10 in.

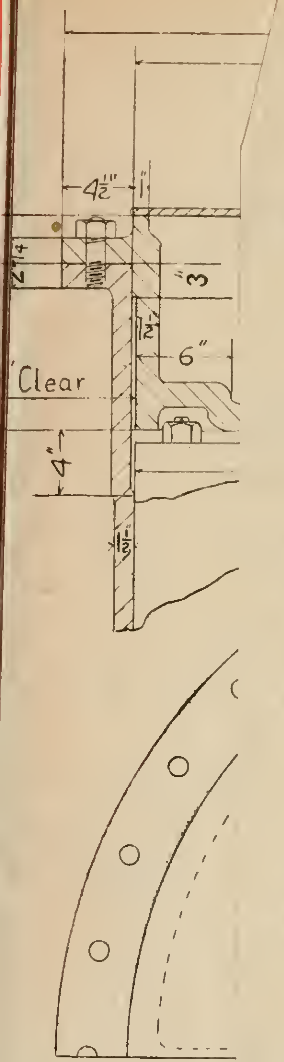
Allow, say, 7 in. for taper, and 3 in. for parallel portion of rod.

Thickness of piston at boss fillet = Rod diameter × ·288.
,, ,, ,, = 7 in. × ·288 = 2 in. thick.

Make thickness of piston near junk ring rather less than near boss, say 1¾ in.

Thickness of piston rings = ¾ in.

Diameter of spring coil = ⅜ in.



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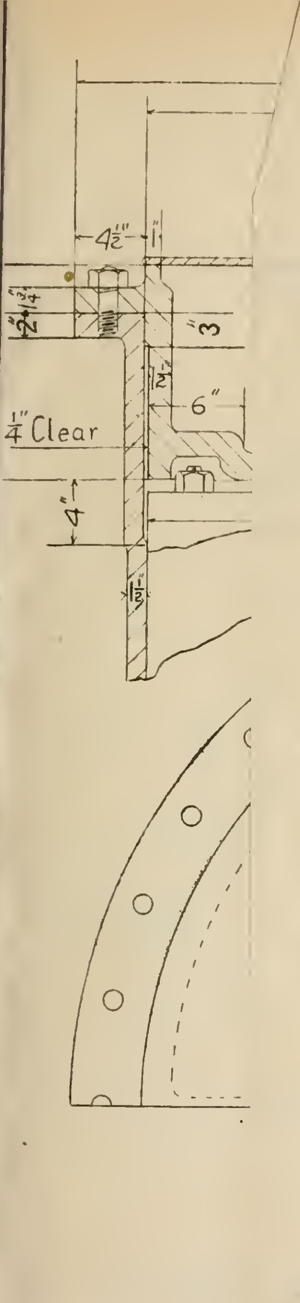

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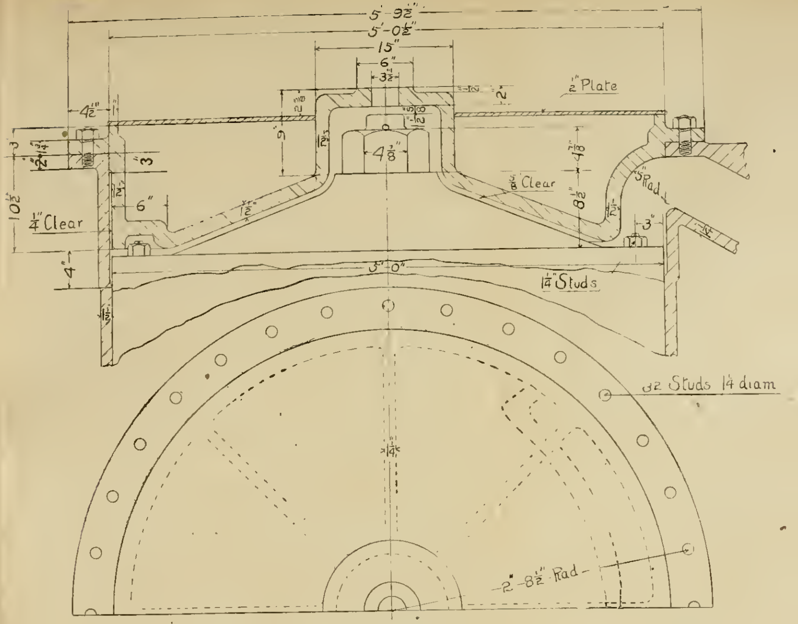


No. 22.—L.P. Cylinder Cover.

Scale, 1 in. = 1 ft.

(Draw to scale of 2 in. = 1 ft.)

Cylinder - - - - -	60 in. diameter.
Pressure - - - - -	20 lbs.

Allow thickness of cylinder walls to be, say, $1\frac{1}{4}$ in. or $1\frac{1}{2}$ in.

Then, Thickness of cover $= 1{\cdot}25$ in. $+ {\cdot}25 = 1{\cdot}5$ in.

then, Thickness of cover flange $= 1{\cdot}25$ in. $\times 1{\cdot}4 = 1{\cdot}75$ in.

then, Thickness of cylinder flange $= 1{\cdot}25$ in. $\times 1{\cdot}6 = 2$ in.

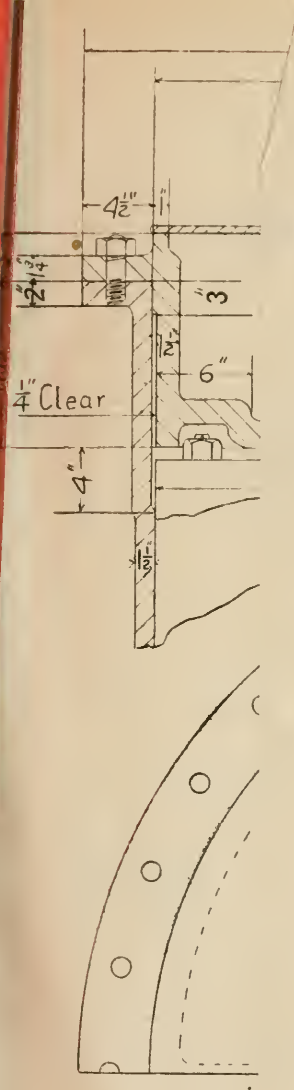
Allow diameter of flange studs to be equal to cylinder thickness, that is, $1\frac{1}{4}$ in. diameter.

Allow pitch of studs to equal 5 stud diameters.

Then, $5 \times 1{\cdot}25$ in. $= 6{\cdot}25$ in. pitch limit.

Take pitch circle as $64\frac{1}{2}$ in. diameter (centre to centre of flange).

Then, $64{\cdot}5$ in. $\times 3{\cdot}1416 \div 6{\cdot}25$ in. $= 32{\cdot}6$, or, say, 32 studs exactly.



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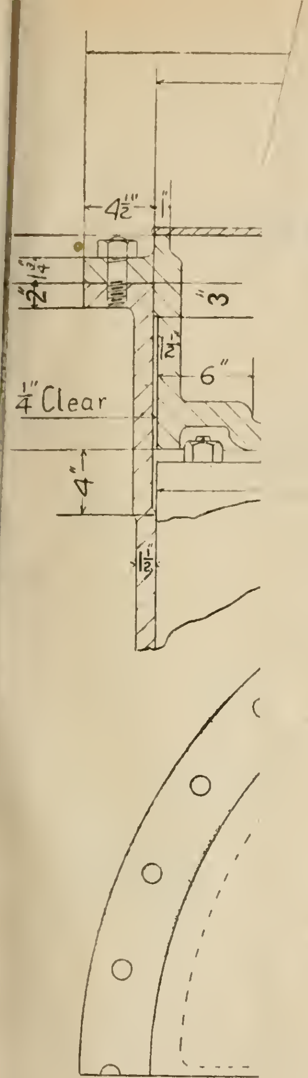

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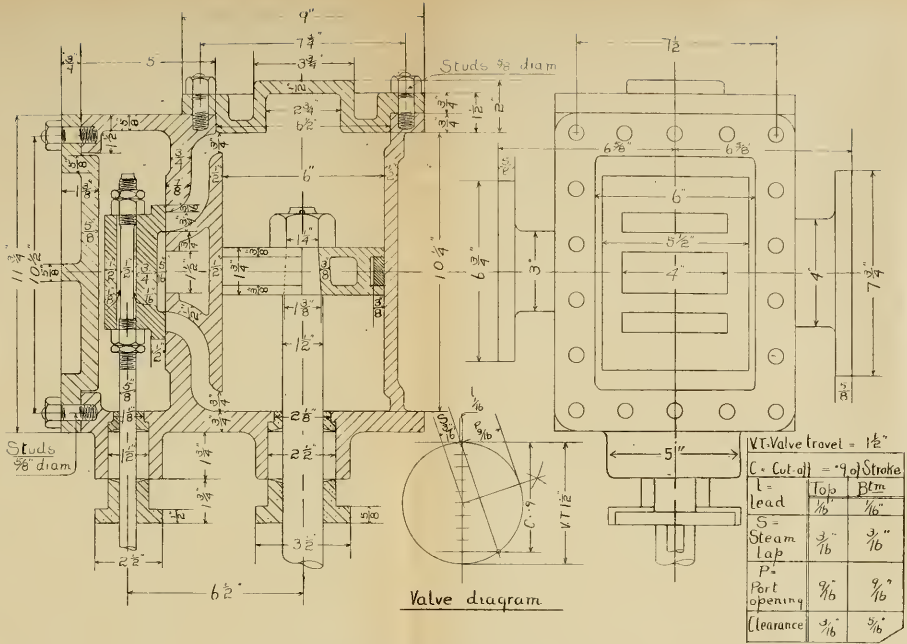

No. 23. Donkey Pump Cylinder and Valve.

(Draw to a scale of 6 in. = 1 foot.)

Valve travel = 1½ in. Cut off = ·9 stroke.

NOTE. As this engine is not fitted with a connecting rod, equal steam lap and lead is given at either end of the valve.

Data. Pressure, 180 lbs. Revolutions, 72 per min. Stroke, 8 in.

NOTE.—Stroke + Piston + Top and bottom clearance = Length of cylinder.

Then, $\text{Length} = 8 + 1\tfrac{3}{4} + \tfrac{3}{16} + \tfrac{5}{16} = 10\tfrac{1}{4}$ in.

Piston Rod Diameter.

Allow a compressive stress limit of, say, 3000 lbs. per sq. in.

Then, $\text{Diameter of rod (at body)} = \sqrt{\dfrac{6^2 \times \cdot 7854 \times 180}{3000 \times \cdot 7854}} = 1{\cdot}4$ in., say $1\tfrac{1}{2}$ in.

Thickness of Cylinder.

Rule. $\text{Thickness} = \dfrac{\text{Diameter} \times \text{Pressure}}{5000} + \cdot 25.$

Then, $\text{Thickness} = \dfrac{6 \text{ in.} \times 180}{5000} + \cdot 25 = \cdot 466$ in., say $\tfrac{1}{2}$ in.

Steam Port Areas.

Allow speed of exhaust steam to be, say, only 20 ft. per second

Rule, Piston area × stroke in ft. × 2 × Revolutions = 20 × 60 × Port area

Then, $6^2 \times \cdot 7854 \times \tfrac{8}{12} \times 2 \times 72 = 20 \times 60 \times \text{Port area}.$

So that, $\text{Port area} = \dfrac{6^2 \times \cdot 7854 \times \tfrac{8}{12} \times 2 \times 72}{20 \times 60} = 2{\cdot}26$ sq. in.

Depth of port $= 2{\cdot}26 \div 4$ in. $= \cdot 56$ in., say $\tfrac{3}{4}$ in. deep.

NOTE. Width of port = 4 in.

Allow depth of exhaust port to be equal to twice that of the steam ports, therefore $\tfrac{3}{4}$ in. $\times 2 = 1\tfrac{1}{2}$ in.

NOTE. For small sizes of auxiliary engine cylinders the steam speeds allowed per second are much less than for main engines.

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NOTE.—Depth of key =

Thickness ,,

Observe that "linked up
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NOTE.—Depth of key =

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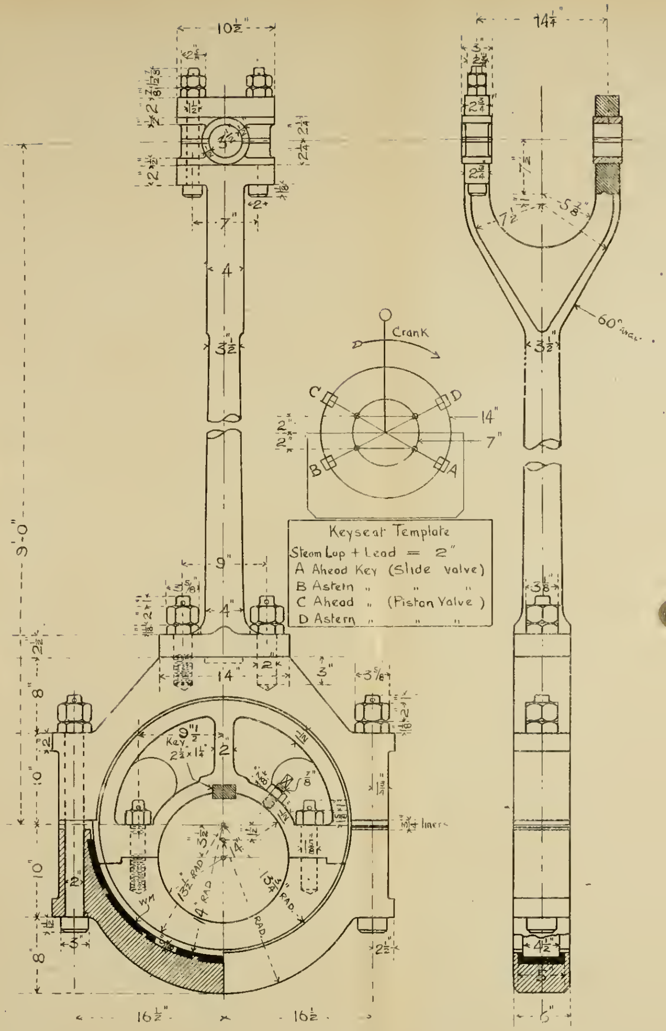

No. 24.—Eccentric and Rod Complete.

(Draw to a scale of 2 in. = 1 ft.)

Data.—Shaft 14 in. diameter. Valve travel 7 in. Diameter of valve spindle (d) 3½ in.

Rule. Diameter of pulley $= d \times 7{\cdot}8 = 3{\cdot}5 \times 7{\cdot}8 = 27{\cdot}3$ in., say 28 in.

Diameter of bolts and studs $= (d \times {\cdot}5) + {\cdot}2$.

Then, „ „ „ $= (3{\cdot}5 \times {\cdot}5) + {\cdot}2 = 1{\cdot}95$ in., say 2 in.

Diameter of eccentric rod pins $= d \times {\cdot}85 + {\cdot}5 = 3{\cdot}47$ in., say 3½ in.

Depth of eccentric key $= \dfrac{\text{Shaft diameter}}{6} = \dfrac{14 \text{ in.}}{6} = 2{\cdot}33$ in., say 2⅓ in.

Diameter of bolts in top end brass $= \sqrt{\dfrac{\text{diameter of one strap bolt}}{2}}$

„ „ „ „ $= \sqrt{\dfrac{2^2}{2}} = 1{\cdot}4$ in., say 1·5 in. diameter.

Thickness of cap and butt = diameter of bolts $\times 1{\cdot}33 = 1{\cdot}5$ in. $\times 1{\cdot}33 = 1{\cdot}99$ in., say 2 in.

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Note.—Depth of key =

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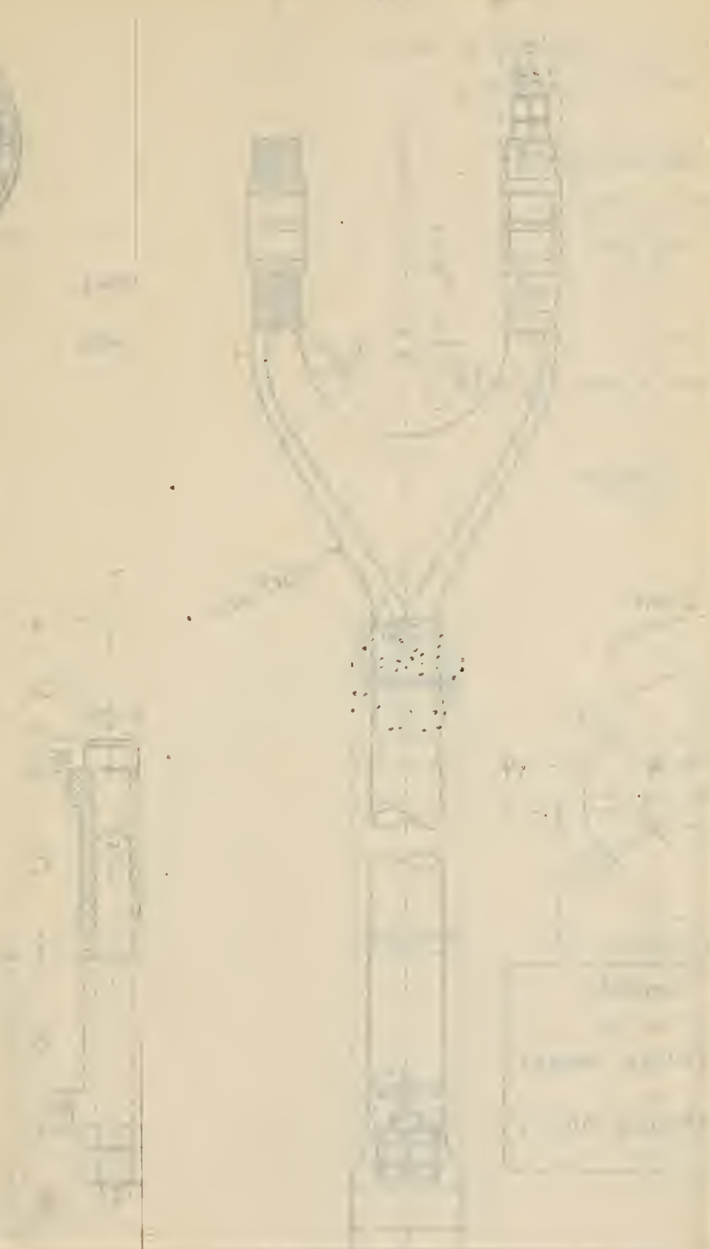

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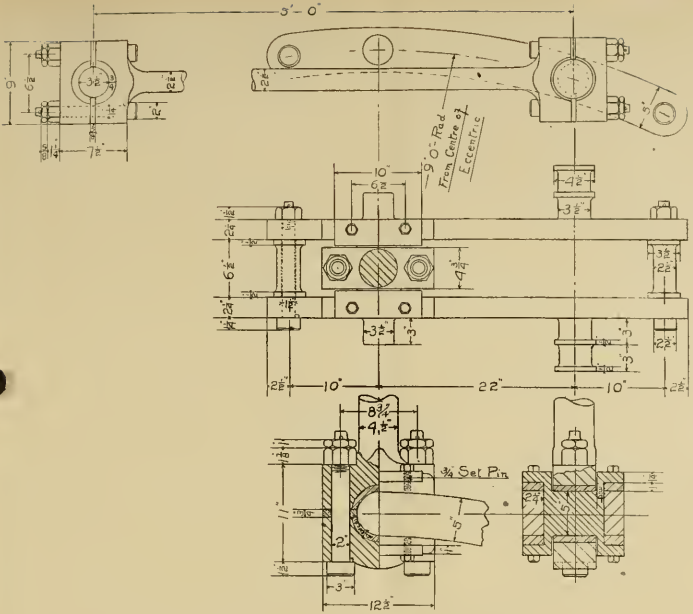

No. 25.—Quadrant Bars, Valve Eye Block, and Drag Links.

(Draw to a scale of 2 in. = 1 ft.)

Data.—Valve travel, 7 in. Diameter of valve spindle at body, $3\frac{1}{2}$ in. $= d$.

Rule, Distance between bars $= d \times 1{\cdot}75 + {\cdot}25$.

Then, ,, ,, ,, $= 3{\cdot}5 \times 1{\cdot}75 + {\cdot}25 = 6{\cdot}37$, say $6\frac{1}{2}$ in.

Diameter of valve rod eye (steel) $= 1 \times$ depth of bars.

Then, ,, ,, ,, ,, $= 1 \times 5$ in. $= 5$ in.

Allow eye block brass to be, say, $\frac{3}{4}$ in. thick.

Length of slipper $= d \times 3{\cdot}6$.

Then, ,, ,, $= 3{\cdot}5 \times 3{\cdot}6 = 12{\cdot}6$, say $12\frac{1}{2}$ in.

Allow thickness of slipper to be about 1 in. at centre.

Allow eye block thickness at sides to be about $\frac{3}{4}$ in.

Allow eye block thickness at top and bottom to be about $1\frac{1}{4}$ in.

Diameter of bolts in eye block $= d \times {\cdot}5 + {\cdot}2$.

,, ,, ,, $= 3{\cdot}5 \times {\cdot}5 + {\cdot}2 = 1{\cdot}95$, say 2 in.

Diameter of drag links (2) $= d \times {\cdot}72$.

,, ,, $= 3{\cdot}5 \times {\cdot}72 = 2{\cdot}52$, say $2\frac{1}{2}$ in.

NOTE. For other proportions of quadrant bars and eccentric rods, pins, etc., see drawing No. 24.

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NOTE.—Depth of key =

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Observe that "linked up

horizontal

Mean thickness of cylinder walls to be, say [illegible]

Thickness of [illegible] [illegible]

Thickness of [illegible] [illegible]

Thickness of [illegible] flange [illegible]

[illegible] flange studs [illegible]

[illegible]

[illegible]

Take pitch circle of [illegible]

[illegible]

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NOTE.—Depth of key =

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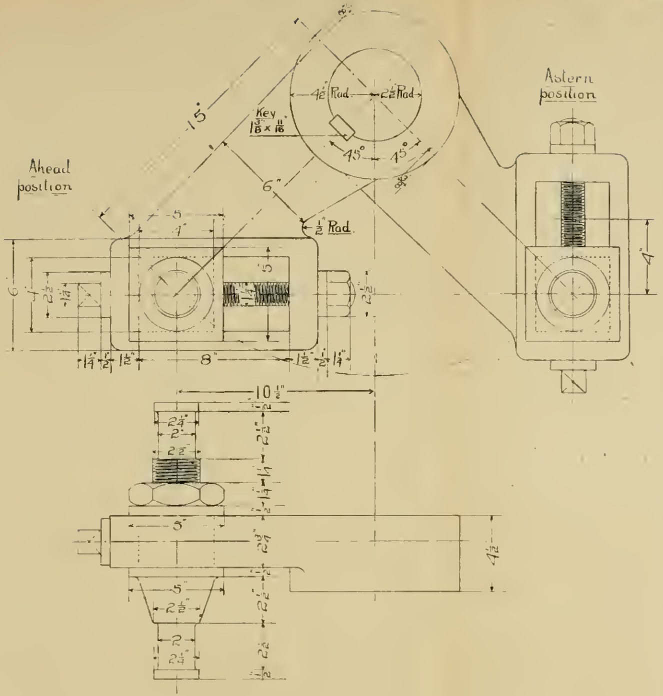


No. 26.—Reversing Gear Bell-Crank and Expansion Slot.

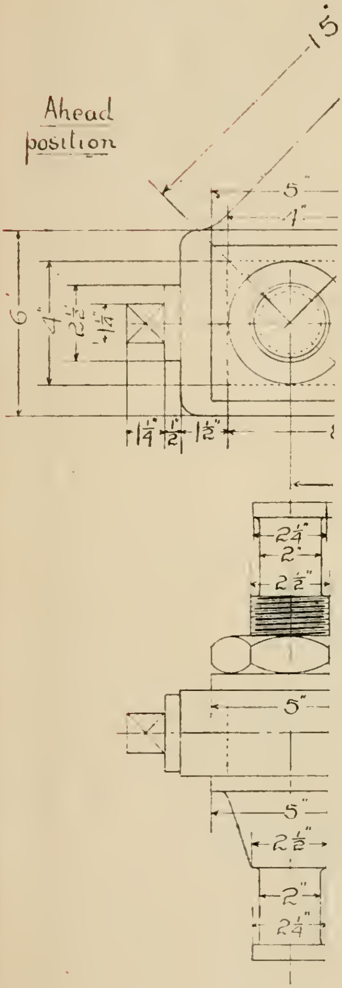
(Draw to a Scale of 3 in. = 1 ft.)

The expansion block has a range of 4 inches io the slot.

NOTE.—Depth of key $= \frac{\text{shaft diameter}}{4} + \cdot 125 = \frac{5 \text{ inches}}{4} + \cdot 125 = 1 \cdot 375$, or $1\frac{3}{8}$ inches.

Thickness ,, $=$ Depth $\times \cdot 5 = \cdot 6875$, or $\frac{11}{16}$ inch.

Observe that "linked up" ahead becomes "full gear" astern, owing to the change from the horizontal position to the vertical position of the expansion slot.



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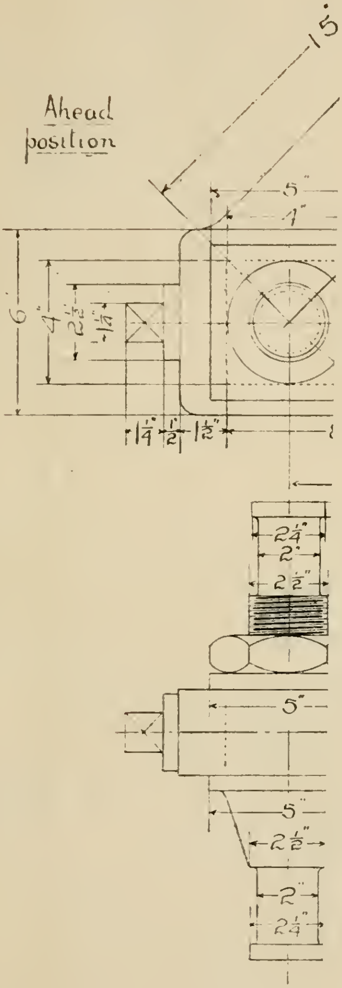

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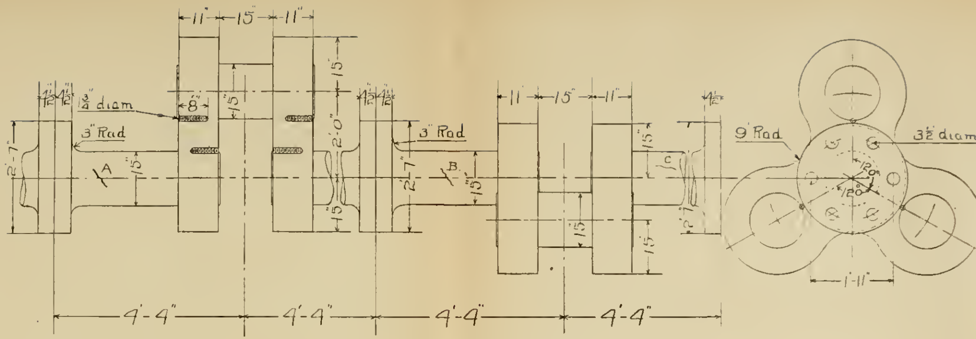


No. 27.—Crank Shafting.

(*Scale, $\frac{3}{8}$ in. = 1 ft.*)

(Draw to a scale of $\frac{3}{4}$ in. = 1 ft.)

Data.—Cylinders 30 in., 52 in., 80 in.
Stroke, 48 in.
Boiler steam, 185 lbs.

Rule $\qquad C \times P \times D^2 = S^3 \times \text{Constant} \times \left(2 + \frac{D^2}{d^2}\right).$

Where C = Length of crank.
,, P = Absolute boiler pressure.
,, D = L.P. cylinder.
,, d = H.P.
,, S = Shaft diameter.
Constant = 1110 for propeller shafts and crank shafting.
,, = 1295 ,, tunnel shafting.

Then, $$S^3 = \frac{C \times P \times D^2}{\text{Constant} \times \left(2 + \frac{D^2}{d^2}\right)}.$$

,, $$S^3 = \frac{24 \text{ in.} \times 200 \times 80^2}{1110 \times \left(2 + \frac{80^2}{30^2}\right)} = 3041 \cdot 2.$$

,, S (shaft diameter) = $\sqrt[3]{3041 \cdot 2}$ = 14·5 in., or, say, 15 in. diameter.

Note.—Observe that cube root extraction is required.
Note.—Stroke = 48 in. therefore crank (C) = 24 in.
,, 185 + 15 = 200 = P.

Diameter of couplings = Shaft diameter × 2 (at least).
,, ,, = 15 × 2 = 30, say 31 in. diameter.

Thickness of couplings = shaft × ·3.
,, ,, = 15 × 3 = 4·5 in.
Coupling fillet radius = shaft × ·2.
,, ,, = 15 × ·2 = 3 in.
Bolt pitch circle diameter = shaft × 1·5.
,, ,, = 15 × 1·5 = 22·5 in., say, 23 in.

$$\text{Bolt diamater} = \sqrt{\frac{\text{Shaft diameter}^2 \times \text{half shaft radius}}{\text{Bolt pitch radius} \times \text{number of bolts}}}.$$

$$\text{,, ,,} = \sqrt{\frac{15^2 \times (7 \cdot 5 \text{ in.} \div 2)}{11 \cdot 5 \times 6}} = 3 \cdot 5 \text{ (nearly) say, } 3 \cdot 5 \text{ in. diameter.}$$

Six bolts have been allowed.

Bolt radius = 23 in. ÷ 2 = 11·5.
Thickness of crank webs = shaft × ·7.
,, ,, = 15 × ·7 = 10·5, or, say, 11 in.
Diameter of web bosses = shaft × 2.
,, ,, = 15 × 2 = 30 in.

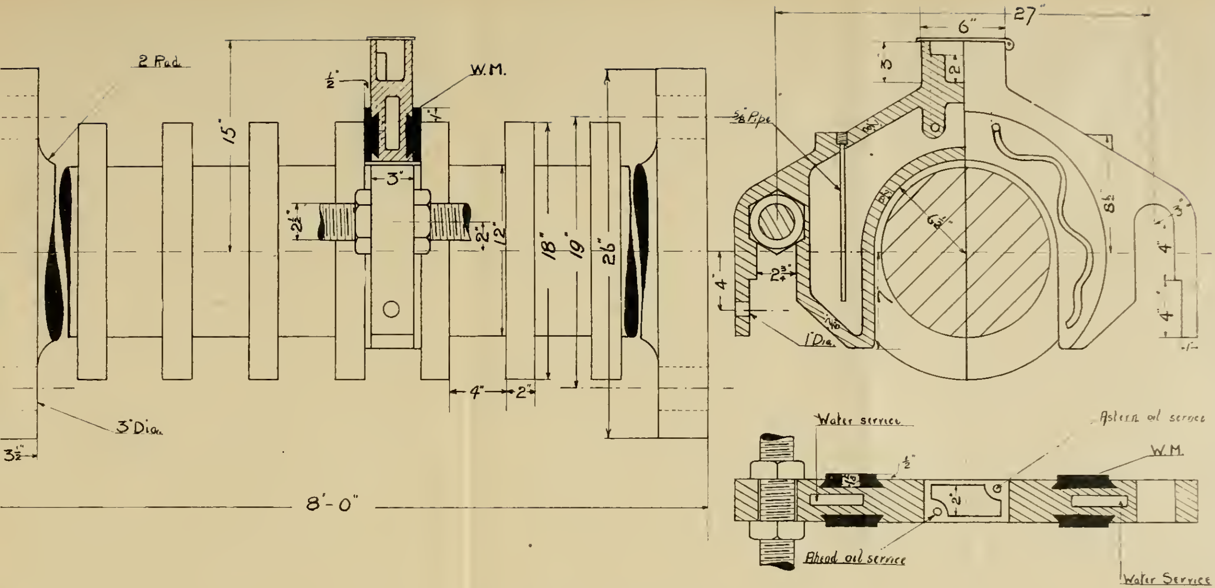
Allow length of crank pin = Diameter of crank pin = Diameter of shaft.
Therefore. Crank pin = 15 in. × 15 in.

Length from centre of coupling to centre of crank pin is equal to half length of pin + web thickness + clearance + twice pulley thickness + length of bearing + clearance and coupling radius + coupling thickness.

Allowing pulley thickness as $4\frac{1}{2}$ (each) and bearing 15 in. length.

Then,

7·5 + 11 + 1 in. + 9 in. + 15 in + 1 in. + 3 in. + 4·5 in. = 52 in. centres



No. 28.—Thrust Shaft and Shoe.

(Draw to a scale of $1\frac{1}{2}$ in. = 1 ft.)

For complete design calculations see Drawing No. 29.

Data.—I. H. P. 1500.
Speed, 10 knots.
Pressure on shoes - - - 60 lbs. per sq. in.
Ahead surface - - - - 567 sq. in.
Astern ,, - - - - 486 sq. in.

Notice that the actual bearing surface is equal to about $\frac{2}{3}$ only of the total surface of the shaft collars.

Then, Shaft + clearance $= 12$ in. $+ \frac{1}{2}$ in. $+ \frac{1}{2}$ in. $= 13$ in.

Annulus area of collar $= (18^2 - 13^2) \times \cdot 7854 = 121\cdot 7$ sq. in.

Actual bearing surface $= 121\cdot 7 \times \frac{2}{3} = 81\cdot 1$.

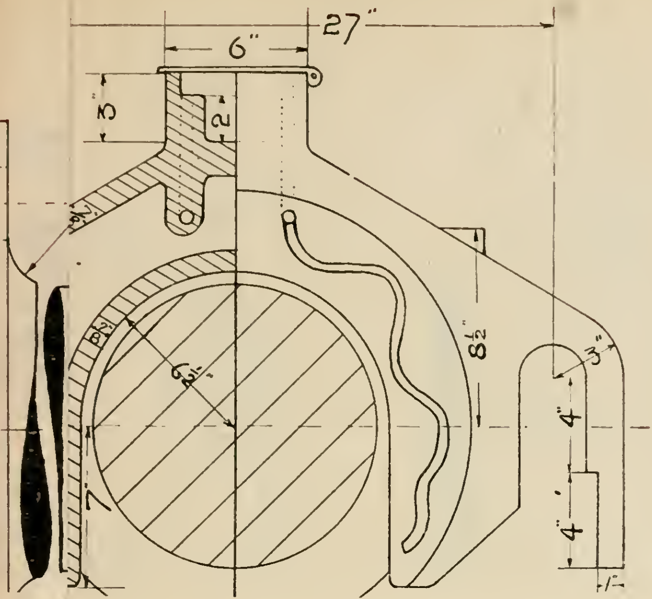
As there are 7 shoes for ahead, then, $81\cdot 1 \times 7 = 567\cdot 7$ sq. in.

,, ,, 6 ,, astero, ,, $81\cdot 1 \times 6 = 486\cdot 6$ sq. in.

$$\text{Total pressure on block} = \frac{1500 \times 33000 \times \cdot 68}{\dfrac{10 \times 6080}{60}} = 33218 \text{ lbs.}$$

$$\text{Pressure per square inch on each shoe} = \frac{33228}{567} = 58 \text{ lbs., or, say, } 60 \text{ lbs.}$$

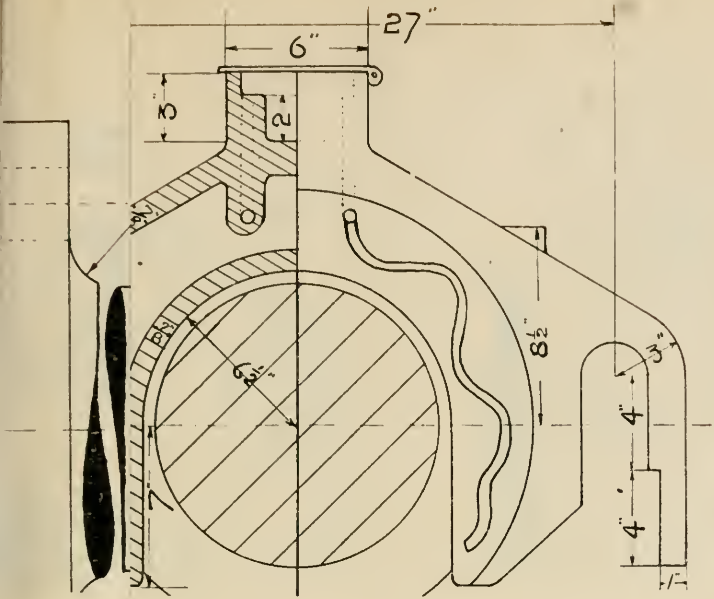
NOTE.—Allow ·68 of total I.H.P. as effective power on block.



So that, S (diameter of shaft) $= \sqrt[3]{1508} = 11{\cdot}5$, nearly, say, 12 in. shaft.

Observe that *cube root* extraction is required.

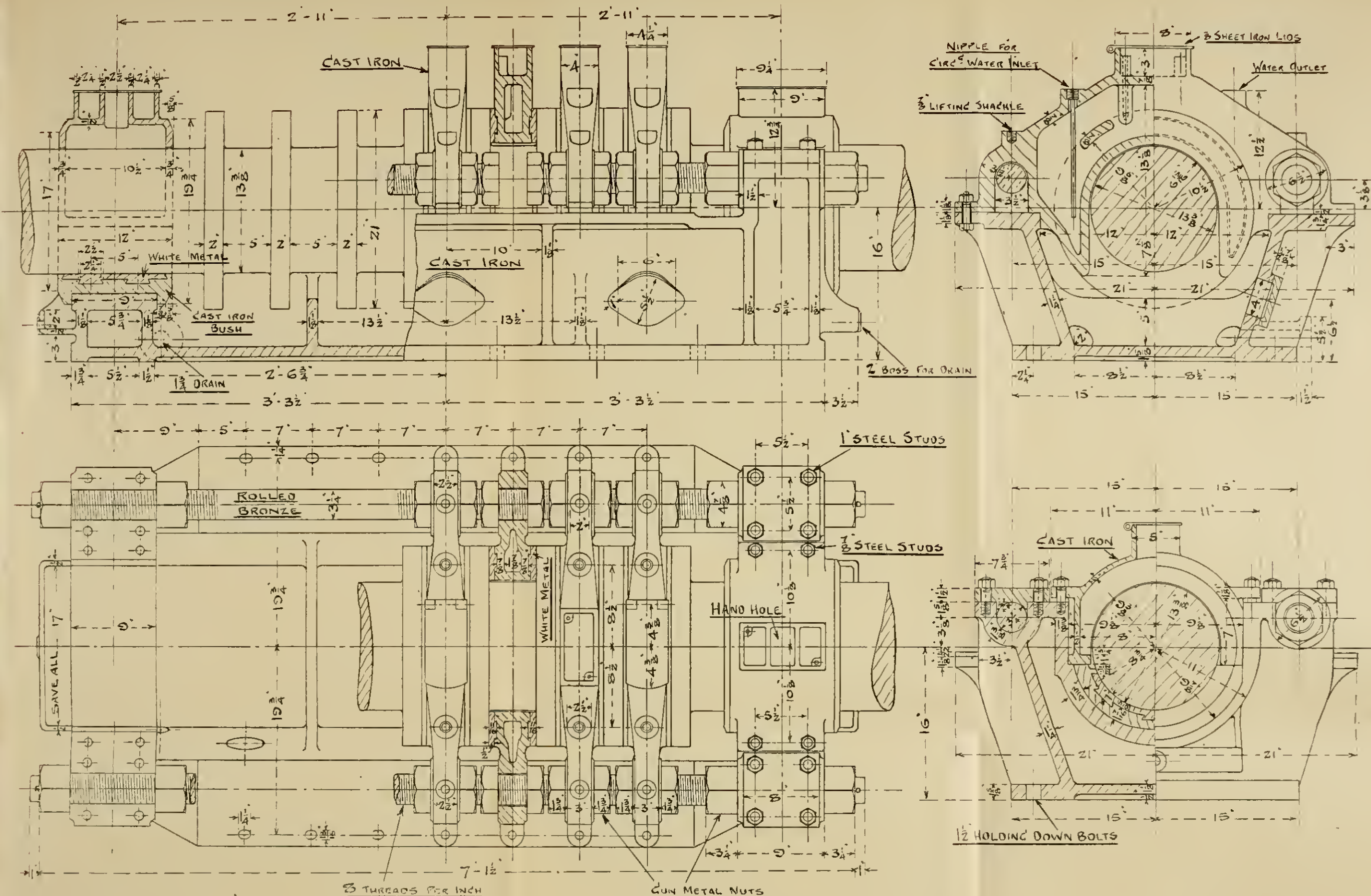
Stroke, 42 in. $\div 2 = 21$ in. length of crank. $180 + 15 = 195$ lbs. absolute.



So that, S (diameter of shaft) $= \sqrt[3]{1508} = 11{\cdot}5$, nearly, say, 12 in. shaft.

Observe that *cube root* extraction is required.

Stroke, 42 in. $\div 2 = 21$ in. length of crank. $180 + 15 = 195$ lbs. absolute.



No. 29. Thrust Block

(Draw to a scale of 1½ in. = 1 ft.)

Data.—I.H.P. 2000. Ship speed, 15 knots. To find required bearing surface of shoes allowing a maximum of 60 lbs. per sq. in., and 7 shoes to be fitted.

Then Total pounds pressure on block $= \dfrac{\text{I.H.P.} \times 33000 \times \cdot 68}{\text{Speed in feet per min.}}$

,, ,, ,, $= \dfrac{2000 \times 33000 \times \cdot 68}{1520} = 30000$ lbs. nearly.

NOTE. Speed in feet per min. $= \dfrac{15 \times 6080}{60} = 1520$ ft.

The effective I.H.P. on thrust block is generally assumed as equal to 68 per cent. of total I.H.P.

Pressure on each shoe $= \dfrac{30000}{7} = 4300$ lbs. nearly.

Bearing area of each shoe $= \dfrac{4300}{60} = 71 \cdot 6$ sq. in.

Diameter of collars = Shaft diameter $\times 1 \cdot 6$.

,, ,, $= 13 \cdot 375 \times 1 \cdot 6 = 21$ in.

Thickness of collars = Shaft diameter $\times \cdot 15$

,, ,, $= 13 \cdot 375 \times \cdot 15 = 2$ in.

Thickness of shoes = Collar thickness $\times 2$

,, ,, $= 2$ in. $\times 2 = 4$ in.

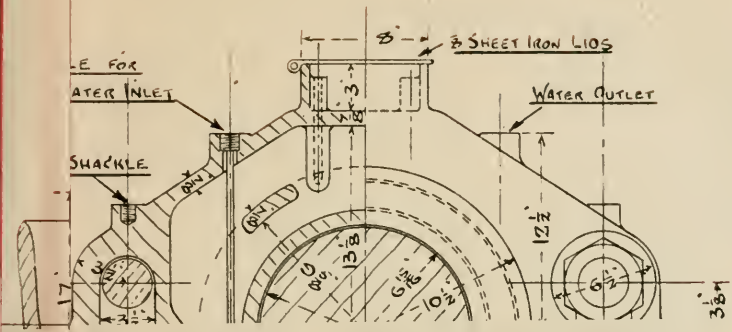
Thickness of white metal = (Collar thickness $\div 5$) $+ \cdot 08$.

,, ,, ,, $= 2 \div 5 + \cdot 08 = \cdot 48$ in.

Distance between collars = 5 in.

Allowing a stress of, say, 2200 lbs. on side stay bars, then,

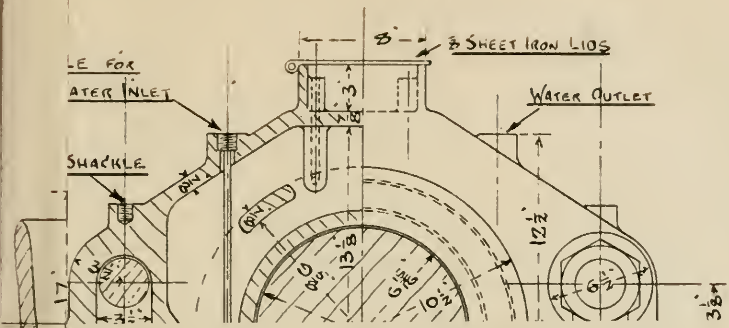
Diameter of bars $= \sqrt{\dfrac{30000}{2200 \times 2 \times \cdot 7857}} = 2 \cdot 9$ in., or say 3¼ in. diameter over thread.



So that, S (diameter of shaft) $= \sqrt[3]{1508} = 11{\cdot}5$, nearly, say, 12 in. shaft.

Observe that *cube root* extraction is required.

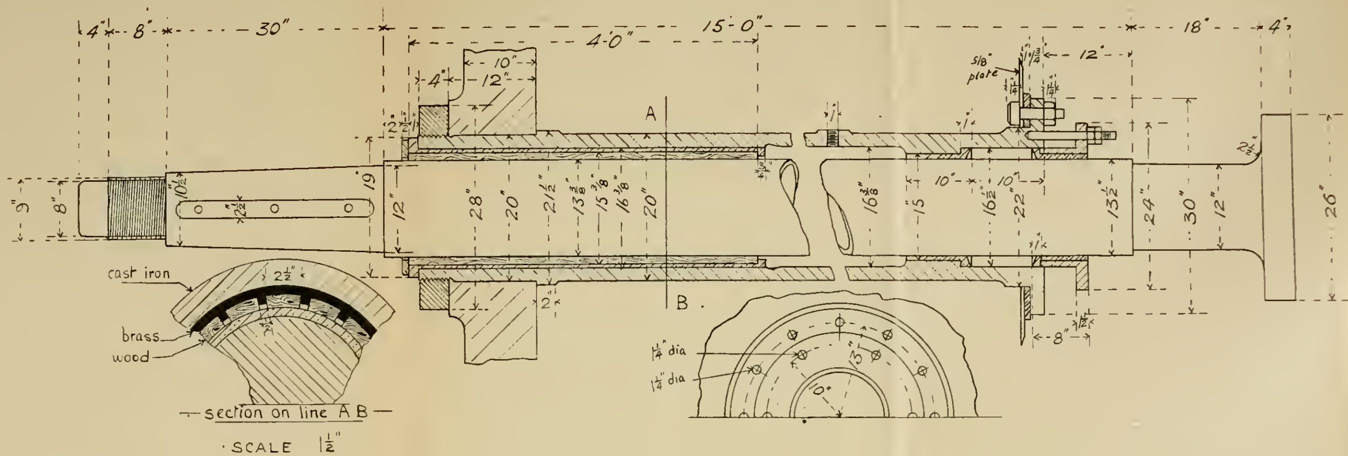
Stroke. 42 in. $\div 2 = 21$ in. length of crank. $180 + 15 = 195$ lbs. absolute.



So that, S (diameter of shaft) $= \sqrt[3]{1508} = 11{\cdot}5$, nearly, say, 12 in. shaft.

Observe that *cube root* extraction is required.

Stroke, 42 in. $\div 2 = 21$ in. length of crank. $180 + 15 = 195$ lbs. absolute.



No. 30.—Stern Tube and Shaft.

(*Scale, ⅝ in. = 1 ft.*)

(**Draw to a scale of 1½ in. = 1 ft.**)

Data.—Cylinders, 24 in. — 40 in., 66 in.
Stroke, 42 in.
Boiler pressure, 180 lbs. gauge.
Diameter of tunnel shafting, 11½ in.
Shaft, 12 in. diameter. Length of taper, 30 in.

Allow thickness of brass liner to be ¾ in. at forward end and $\frac{1}{16}$ less at after end to allow of easy withdrawal and insertion of the shaft.

For the same reason make diameter of tube shoulder at forward end (22 in.) ½ in. more than diameter of shoulder at after end (21½): *this is important.*

Allow a taper of about ⅝ in., or ¾ in. per foot, which reduces the shaft diameter from 12 in. to about 10½ in., as shown.

Allow lignum strips of 1 in. by 2½ in. or 3 in.

Allow length of after bearing (lignum vitæ) to be equal to four times diameter of shaft. Then, 12 in. × 4 in. = 48 in. length of wood in bearing.

Observe the lip (¾ in. thick) cast on the forward end of the brass bush to hold the wood in place, also the check ring fitted aft for the same purpose.

Note.—**The sectional view on line A B is drawn to double the scale of the other view.**

The diameter of the shaft is determined as follows:—

Rule, $C \times P \times D^2 = S^3 \times \text{Constant} \times \left(2 + \frac{D^2}{d^2}\right)$.

Where, C = Length of crank.
P = Absolute boiler pressure.
D = L.P. cylinder.
d = H.P. cylinder.
S = Shaft diameter.
Constant = 1110 for propeller shafts and crank shafting.
" = 1295 for tunnel shafting.

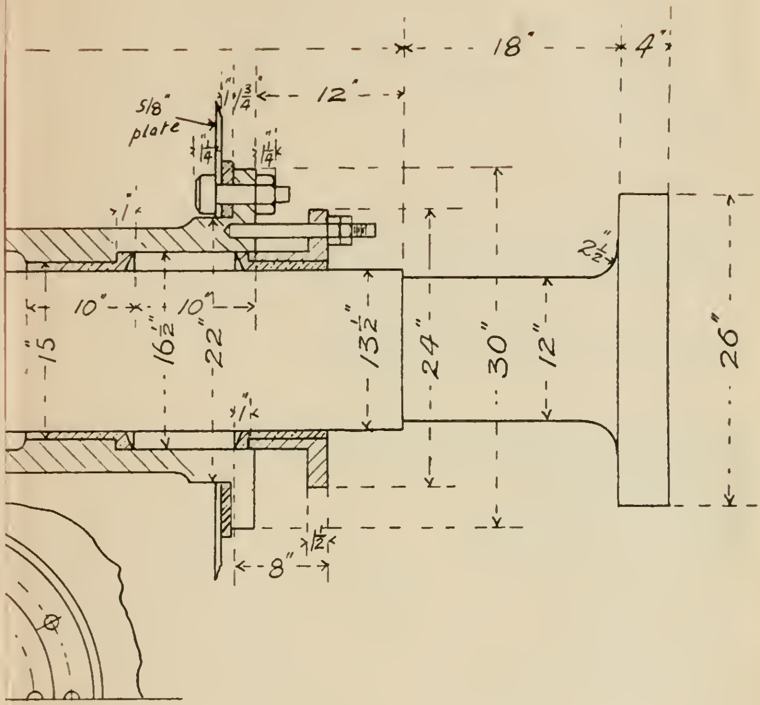
Then, $$S^3 = \frac{C \times P \times D^2}{\text{Constant} \times \left(2 + \frac{D^2}{d^2}\right)}.$$

" $$S^3 = \frac{21 \times 195 \times 66^2}{1110 \times \left(2 + \frac{66^2}{24^2}\right)} = 1508.$$

So that, S (diameter of shaft) $= \sqrt[3]{1508} = 11{\cdot}5$, nearly, say, 12 in. shaft.

Observe that *cube root* extraction is required.

Stroke, 42 in. ÷ 2 = 21 in. length of crank. 180 + 15 = 195 lbs absolute.



$d =$ H.P. cylinder.
S = Shaft diameter.
Constant = 1110 for propeller shafts and crank shafting.
,, = 1295 for tunnel shafting.

Then, $$S^3 = \frac{C \times P \times D^2}{\text{Constant} \times \left(2 + \frac{D^2}{d^2}\right)}.$$

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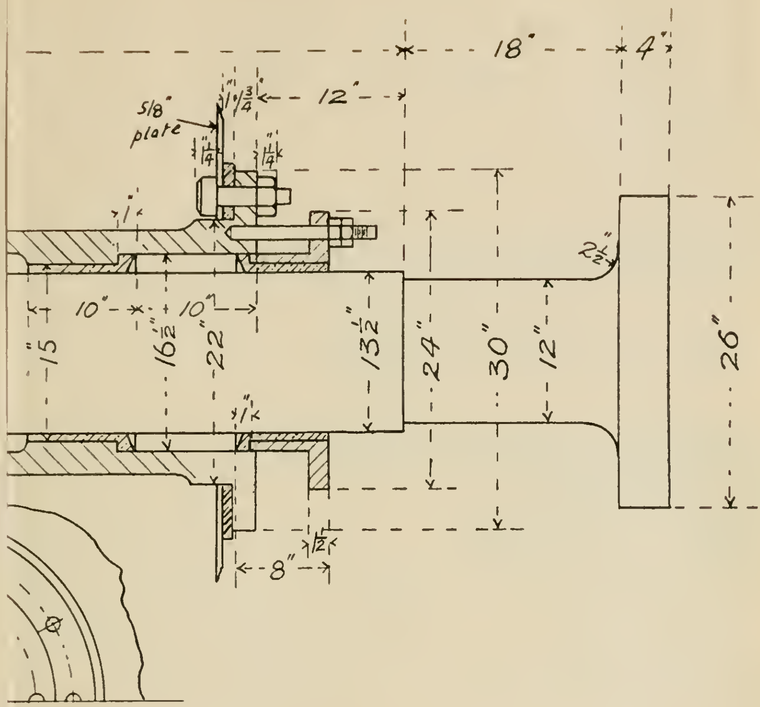

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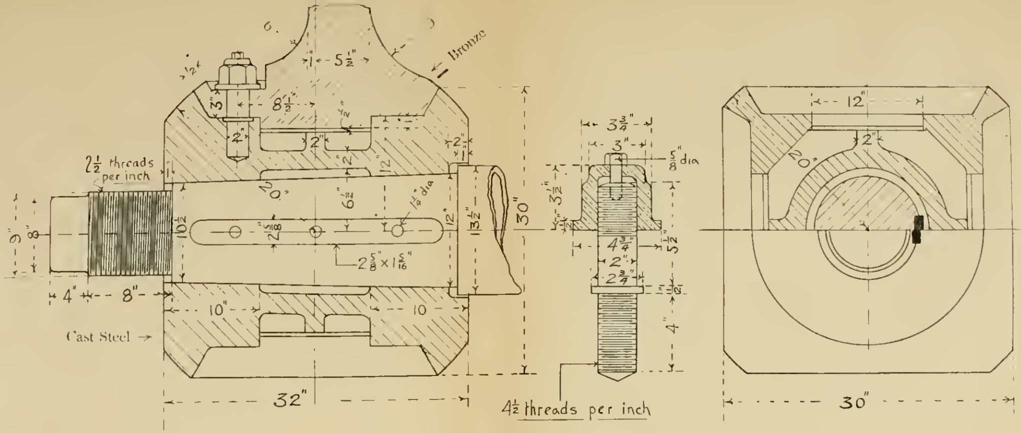


No. 31.—Propeller Boss.

(4 Bladed Propeller.)

(*Scale*, 1 *in.* = 1 *ft.*)

(Draw to a scale of 1½ in. = 1 ft.)

Data.—Cylinders	- - -	24 in., 40 in., 68 in.
I.H.P.	- - - -	1600.
Pitch of propeller	- -	16 ft.
Diameter of propeller	- -	14 ,,
Circumference of propeller	-	44 ,,
Slip per cent.	- - -	10 per cent.
Tunnel shaft	- -	11½ in. diameter.
Propeller ,,	- - -	12 in. ,,

Length of boss = Tail shaft diameter $\times$ 2·6 ft.

,, ,, = 12 in. $\times$ 2·6 = 31·2 in., or, say, 32 in.

Diameter of boss = Shaft diameter $\times$ 2·5 = 30 in.

,, of blade flange = Shaft diameter $\times$ 2·1.

,, ,, ,, = 12 in. $\times$ 2·1 = 25·2, or, say, 25¼ in.

Thickness of blade flange = Shaft diameter $\times$ ·25.

,, ,, = 12 in. $\times$ ·25 = 3 in. (bronze).

Combined area of studs for each blade } = Shaft area $\times$ ·22.

Combined area of studs for each blade } = $12^2 \times ·7854 \times ·22 = 24·8$ sq. in.

Allowing 8 studs per blade.

Then, $24·8 \div 8 = 3·1$ sq. in. area of each stud.

So that, Diameter of studs $= \sqrt{\dfrac{3·1}{·7854}} = 1·9$ in., or, say, 2 in. diameter.

Breadth of key $= \dfrac{\text{Shaft diameter}}{6} + ·6.$

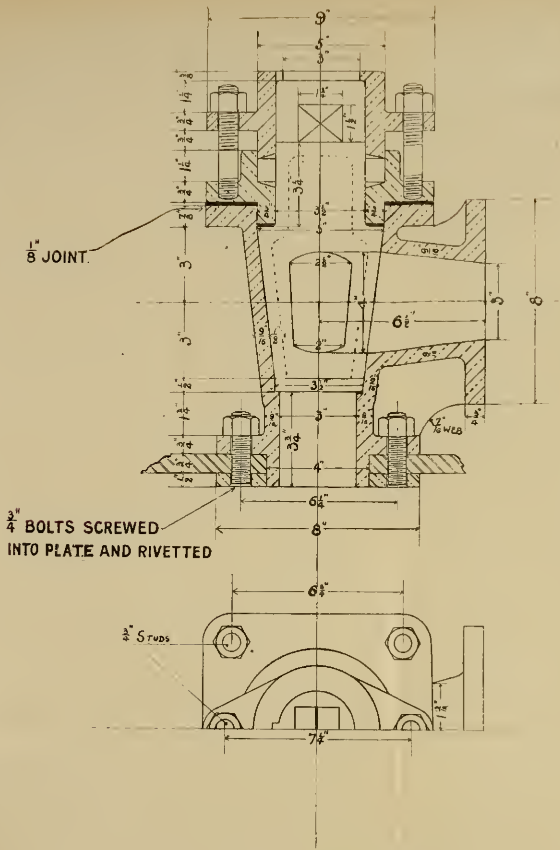
,, ,, $= \dfrac{12}{6} + ·6 = 2·6$, or $2\frac{5}{8}$ in.

Thickness of key = Breadth of key $\times$ ·5.

,, ,, = 2·625 in. $\times$ ·5 = 1·3125, or $1\frac{5}{16}$ in.

The boss is recessed out to fit the blade flanges at an angle of 60 degrees.

Note.—Blades of bronze. Boss of cast steel.



No. 32.—Bottom Blow-Down Cock.

(Ship's Side.)

(Draw to a scale of 6 in. = 1 ft.)

Boiler pressure, 180 lbs. gauge per sq. in.

Least area of port = Area of pipe. Taper of plug = 3 in. per ft.

Mean width of port $= \frac{2{\cdot}5+2}{2} = 2{\cdot}25$

Area of port $= 4 \times 2{\cdot}25 = 9$ sq. in. (approximately).

" pipe $= 3^2 \times {\cdot}7854 = 7$ sq. in.

Allow a stress of 3000 lbs. per sq. in. on the four studs.

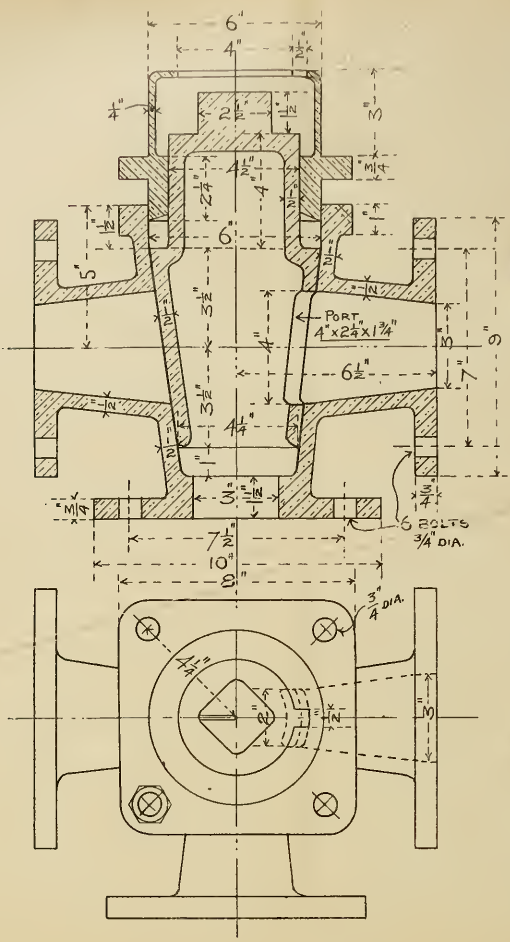
Then, Diameter of studs $= \sqrt{\frac{5^2 \times {\cdot}7854 \times 180}{4 \times {\cdot}7854 \times 3000}} = {\cdot}6$ in, say $\frac{3}{4}$ in diameter.

Note. Studs of $\frac{5}{8}$ in. diameter would be sufficient, but to allow for a margin of safety say $\frac{3}{4}$ in. diameter.

Rule, Cover or lap when closed $= \left(\frac{\text{Mean circumference of plug}}{2}\right) - (\text{Mean width of port}).$

Then, " " " $\left(\frac{4{\cdot}25 \times 3{\cdot}1416}{2}\right) - (2{\cdot}25) = 4{\cdot}42$ in.

Note $\frac{5 \text{ in.} + 3{\cdot}5 \text{ in.}}{2} = 4{\cdot}25$ in. mean diameter of plug



No. 33.—Three-Way Change Cock.

(Draw to a scale of 6 in. = 1 ft.)

Allow taper of plug to be about 3 in. per ft. Port area (at least) = Pipe area. Port depth taken as 4 in.

Then, $\quad 4 \text{ in.} \times \text{Mean width} = 3^2 \times \cdot 7854$ in.

,, $\quad 4 \text{ in.} \times \text{Mean width} = 7$ sq. in.

,, $\quad \text{Mean width} = 7 \div 4 = 1{\cdot}75$ in. (say 2 in.).

If, therefore, ¼ in. is taken off the bottom (to suit the plug taper) it will add ¼ in. to the top, thus giving the port sizes as 4 in. deep by 2¼ in. by 1¾ in. in width.

Diameter of Gland Studs.

The stress allowed on the gland studs may be taken as 3000 lbs per □″, assuming a water pressure of say 50 lbs. per □″.

Then,

$$\text{Diameter} = \sqrt{\frac{6^2 \times 50}{4 \times 3000}} = \cdot 4 \text{ in. (nearly).}$$

To allow for a wide margin of safety, say 4 studs of ¾ in. diameter.

NOTE.—The pressure has been taken as acting on an area equal to a 6 in. diameter circle (top of plug), and the size of studs calculated from this.

To Find Cover or Lap when Cock is Closed.

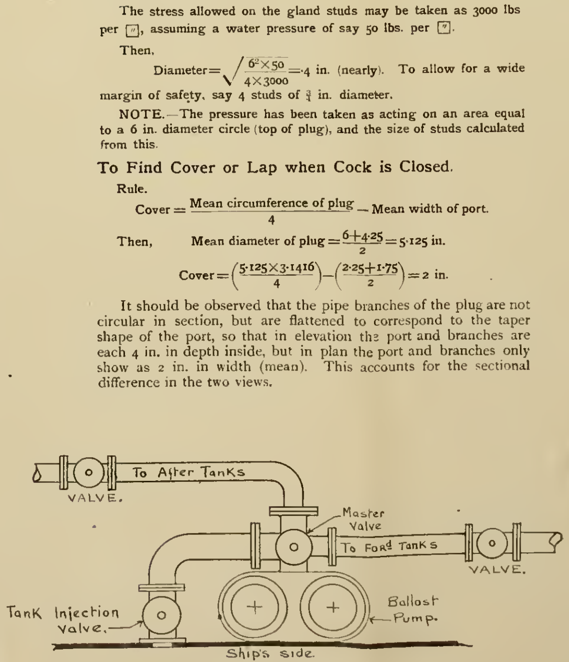
Rule.

$$\text{Cover} = \frac{\text{Mean circumference of plug}}{4} - \text{Mean width of port.}$$

Then, $\quad \text{Mean diameter of plug} = \frac{6 + 4{\cdot}25}{2} = 5{\cdot}125$ in.

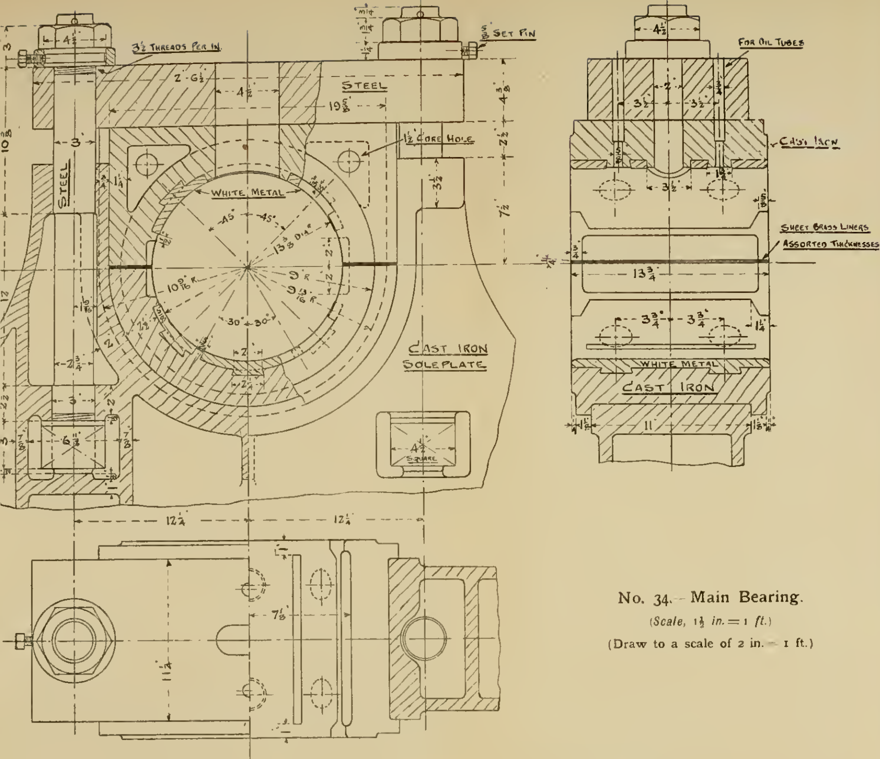
$$\text{Cover} = \left(\frac{5{\cdot}125 \times 3{\cdot}1416}{4}\right) - \left(\frac{2{\cdot}25 + 1{\cdot}75}{2}\right) = 2 \text{ in.}$$

It should be observed that the pipe branches of the plug are not circular in section, but are flattened to correspond to the taper shape of the port, so that in elevation the port and branches are each 4 in. in depth inside, but in plan the port and branches only show as 2 in. in width (mean). This accounts for the sectional difference in the two views.



No. 33a.—Sketch showing Tank-filling Arrangements.

NOTE.—The ballast donkey can pump out the tanks, or can pump them up if required.



No. 34.—Main Bearing.

(Scale, 1½ in. = 1 ft.)

(Draw to a scale of 2 in. = 1 ft.)

Data.—Cylinders 24, 38, 64 in. H.P. steam, 180 lbs. Allowing a tensile stress limit of 3300 lbs. per sq. in., then the diameter of the bolts can be found as follows:

Rule, Holding down bolt diameter $= \sqrt{\dfrac{\text{H.P. area} \times \text{Pressure}}{4 \times 3300 \times \cdot 7854}}$.

Then, Holding down bolt diameter $= \sqrt{\dfrac{24^2 \times 180}{4 \times 3300}} = 2{\cdot}6$, say $2\frac{3}{4}$ in. diameter (bottom of thread)

NOTE.—Allow a tensile stress of 3300 lbs. per sq. in.

NOTE.—·7854 cancels out top and bottom.

NOTE.—The total load on each crank is taken up by a *pair* of main bearings, hence 4 bolts.

Thickness of white metal $= \cdot 04 \times$ Diameter of shaft $+ \cdot 1$ in.

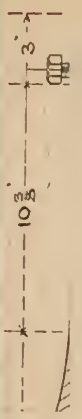
" " $= \cdot 04 \times 13{\cdot}375 + \cdot 1 = \cdot 635$ in., say $\frac{5}{8}$ in.

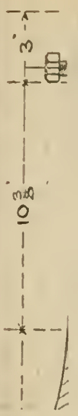
Thickness of metal (W.M. included) = Diameter of shaft $\times \cdot 144 + \cdot 3$.

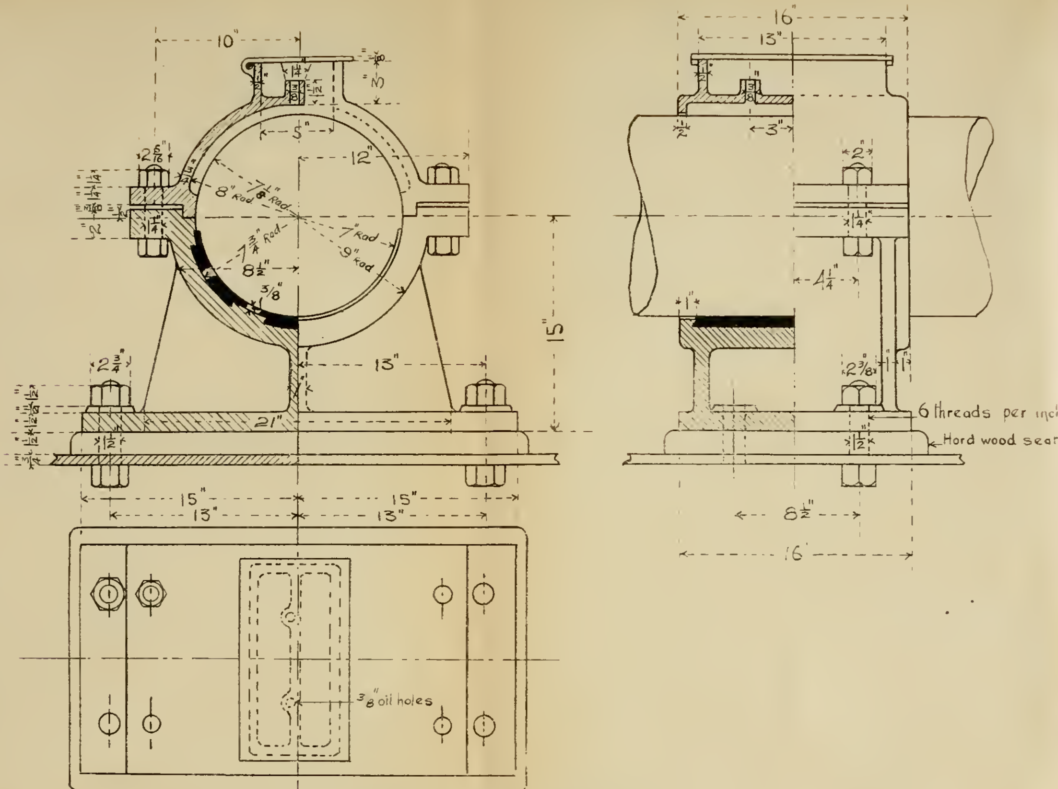
" " " $= 13{\cdot}375 \times \cdot 144 + \cdot 3 = 2{\cdot}22$, say $2\frac{1}{4}$ in.

Thickness of caps = Bolt diameter $\times \cdot 7 +$ Bolt pitch $\times \cdot 1$.

" " $= 2{\cdot}75 \times \cdot 7 + 24{\cdot}5 \times \cdot 1 = 4{\cdot}37$ in., say $4\frac{3}{8}$ in.


3'
10 3/8


3"
10 3/8"

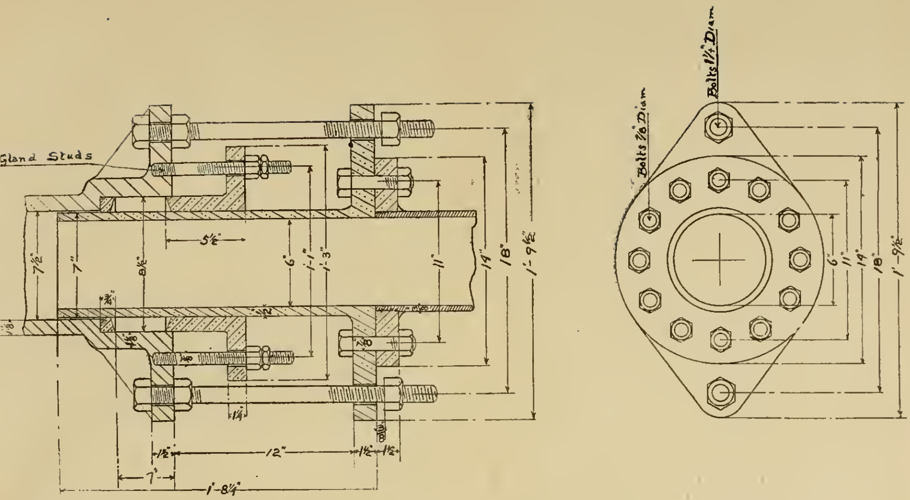


No. 34a—Tunnel Bearing Block

Scale = $1\frac{1}{2}$ in. per ft.

(Draw to a scale of 2 in. = 1 ft.)

The size of the holding down bolts depends on the shearing stresses set up by the rolling or pitching of the vessel.
White metal at thickest part = shaft diameter × ·05 = 14 × ·05 = ·7 in., say $\frac{3}{4}$ in.



No. 35.—Steam Pipe Expansion Joint.

(Draw to a scale of 3 in. = 1 ft.)

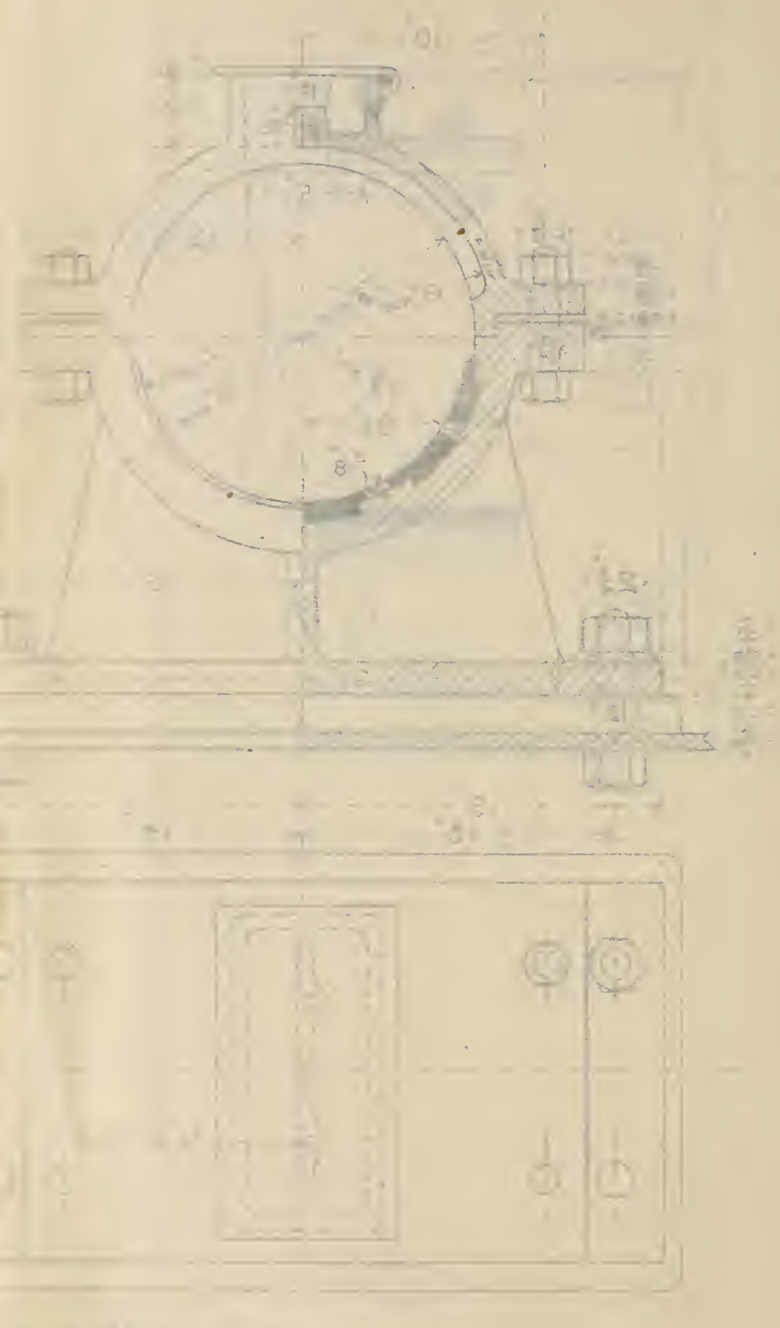
(*Pressure*, 180 *lbs. per sq. in.*)

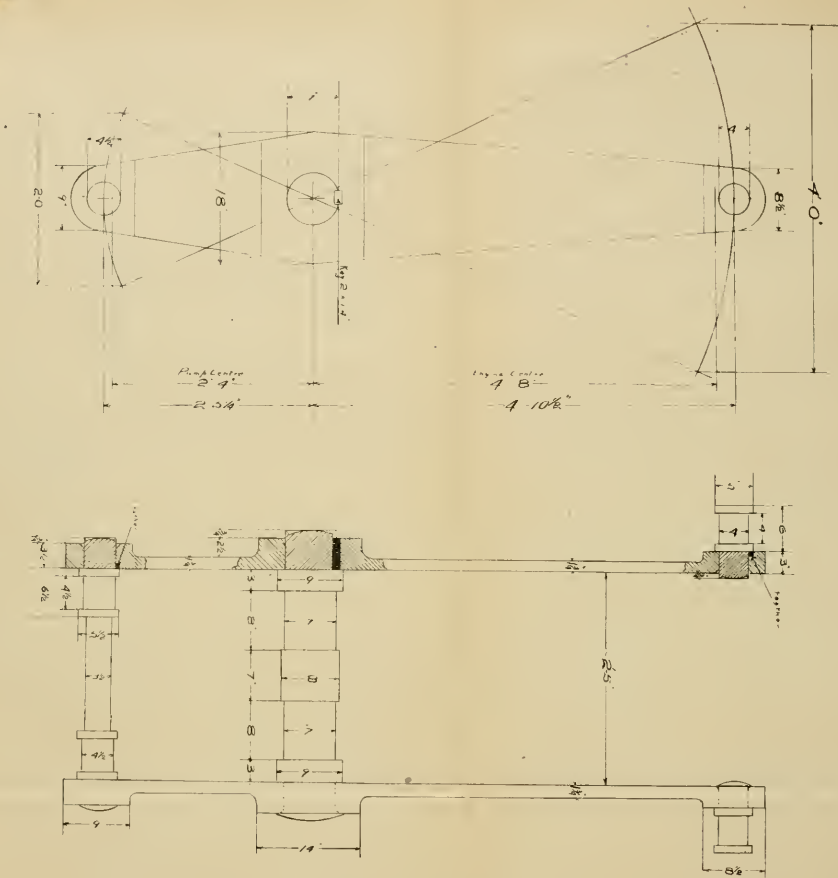
Diameter of Tie Bolts.

Allow a tensile stress of 3700 lbs. per sq. in. on the bolts.

Then, Diameter of each tie bolt $= \sqrt{\dfrac{8^2 \times 180}{2 \times 3700}} = 1{\cdot}24$ in., say $1\frac{1}{4}$ in. diameter.

Note.—The pressure is taken as acting on an 8 in. diameter circle, in place of that of the pipe bore.





No. 36.—Pump Levers.

(Scale, $\frac{3}{4}$ in. = 1 ft.)

(Draw to a scale of $1\frac{1}{2}$ in. = 1 ft.)

Data { Engine stroke	48 in.	Cylinder	66 in. diameter.
{ Pump stroke	24 in.	Air pump	24 in. diameter.

Assume a distance between centre of cylinder and centre of pump, say 7 ft., and set off the respective strokes at either end. If now a line is drawn diagonally from one end of the pump stroke to the opposite end of the engine stroke, it will cut the horizontal centre line at the required centre of the main gudgeon. After finding the rocking shaft centre in this way, arrange (by trial) that the centre of the pin in pump end of lever is exactly as far short of the pump centre line when on top or bottom position as it is *beyond* the pump centre line when at half stroke. Repeat this for the engine end and it will be found on scaling that the distance from pin centre at pump end to rocking shaft centre is exactly 2 ft. $5\frac{1}{4}$ in., and from pin centre at engine end to rocking shaft centre is exactly 5 ft. $10\frac{1}{2}$ in.

The correct position of these centres is all-important.

We have then the following dimensions determined as described:—

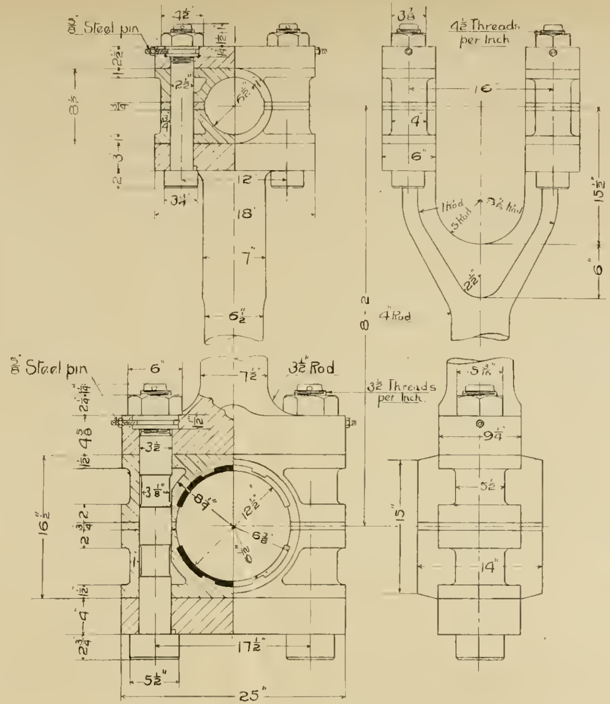
Centre of pump to centre of gudgeon	2 ft. 4 in.	As the strokes.
,, engine ,, ,, ,,	4 ft. 8 in.	As 2 : 1.
Length of pump end of lever, centre to centre	2 ft. $5\frac{1}{4}$ in.	The other sizes must also
,, cylinder ,, ,, ,, ,,	4 ft. $10\frac{1}{2}$ in.	be as 2 : 1.

Square inches of effective bearing surface of studs (pins) per square inch of bucket area $= \frac{4{\cdot}5 \times 4{\cdot}5 \times 2}{24^2 \times {\cdot}7854} = {\cdot}08$ sq. in.

Square inches of stud (pin) area per square inch of bucket $= \frac{4{\cdot}5^2 \times {\cdot}7854 \times 2}{24^2 \times {\cdot}7854} = {\cdot}07$ sq. in.

Note.—The bearing surface of a pin, stud, or journal is equal to Length × Diameter (not circumference).

In the foregoing calculations the two pins of the pump end are taken.



No. 37.—Connecting Rod.

(*Scale* 1 *in.* = 1 *ft.*)

(Draw to a Scale of $1\frac{1}{2}$ in. = 1 ft.)

Data.—Cylinders, 24 in., 40 in., 66 in.
Stroke, 42 in.
Boiler steam, 170 lbs
Crank pin, $12\frac{1}{2}$ in. diameter by $14\frac{1}{2}$ in.
Crosshead pin, $6\frac{1}{2}$ in. diameter by $6\frac{3}{4}$ in.
Piston rod, $6\frac{1}{2}$ in. diameter.

Allow length of rod, centre to centre, to be equal to stroke × 2·3.

Then, 42 in. $\times$ 2·3 = 96·6, or, say, 98 in.

Make diameter of rod at small end equal to diameter of piston rod, and allow 1 in. taper in length, which gives $6\frac{1}{2}$ in. and $7\frac{1}{2}$ in. as the two diameters.

Make thickness of jaw = rod diameter at top $\times$ ·55.
Then, 6·5 in. $\times$ ·55 = 3·57 in., or, say, $3\frac{5}{8}$ in. thick.
Inner radius of jaw, 5 in., and outer radius = 5 in. + $3\frac{5}{8}$ in. = $8\frac{5}{8}$ in.

After striking off the jaw curve at $8\frac{5}{8}$ in. radius, continue the line of jaw into rod with a 60° angle set square, as shown.

Width of jaw = diameter of rod (small end) $\times$ 1·1.
Then, 6·5 in. $\times$ 1·1 = 7·1 in., say 7 in. width.

BOLTS

Diameter of bottom end bolts (2 bolts) $= \sqrt{\dfrac{\text{Load on piston}}{2 \times \text{stress} \times \cdot 7854}}$.

Diameter of top end bolts (4 bolts) $= \sqrt{\dfrac{\text{Load on piston}}{4 \times \text{stress} \times \cdot 7854}}$.

Allow a tensile stress of 4000 lbs. per sq. in. on bottom end bolts.

Then,

Diameter $= \sqrt{\dfrac{24^2 \times 170}{2 \times 4000}} = 3{\cdot}49$, **or, say, 3·5 in. diameter.**

Allow a tensile stress of 4000 lbs. per sq. in. on top end bolts.

Then,

Diameter $= \sqrt{\dfrac{24^2 \times 170}{4 \times 4000}} = 2{\cdot}47$, **or, say, 2·5 in. diameter.**

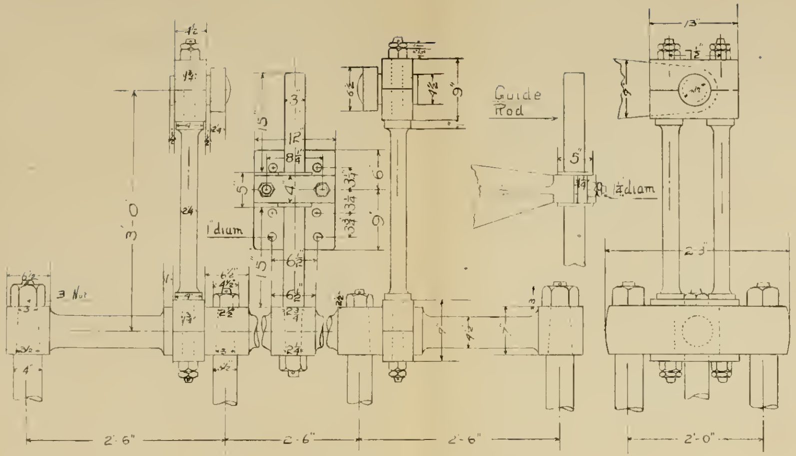
Thickness of connecting rod butt = bolt diameter $\times$ 1·32.
Then, 3·5 in. $\times$ 1·32 = 4·62 in., or, say, $4\frac{5}{8}$ in. thick
Thickness of butt at jaws = bolt diameter $\times$ 1·2.
Then, 2·5 $\times$ 1·2 = 3 in. thick.

The thickness of the caps, top and bottom, is slightly less than that of the butts, as will be observed.

Total thickness of metal round crank pin and crosshead pin = pin diameter $\times$ ·16.
Then, 12·5 $\times$ ·16 = 2 in. (bottom end),
and, 6·5 $\times$ ·16 = 1·04 in. (top end).

Allow thickness of white metal $\frac{5}{8}$ in. to $\frac{3}{4}$ in. reduced to $\frac{1}{2}$ in. between recessed dovetails.

Depth of nuts + collars = diameter of bolts (at least).
Width of bottom end butt and cap = bolt diameter $\times$ 2·6.
Then, 3·5 in. $\times$ 2·6 = 9·10 in., or, say, $9\frac{1}{4}$ in.
Width of top end butt and caps = length of crosshead pin, less clearance at sides.



No. 38. Pump Crosshead and Links.

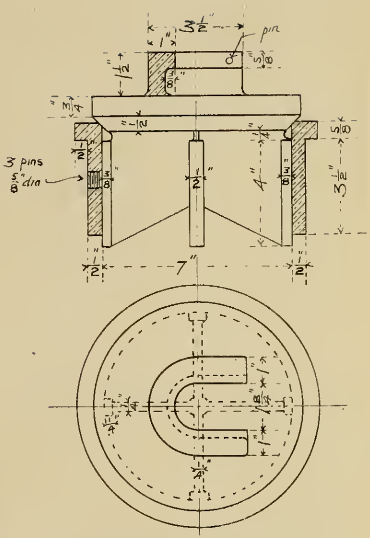
(*Scale*, $\frac{3}{4}$ *in.* = 1 *ft.*)

(Draw to a scale of $1\frac{1}{2}$ in. = 1 ft.)

Data { Air pump, 24 in. diameter.
Circulating pump, 18 in. diameter.

Feed pumps (2), 4 in. diameter.
Bilge ,, (2), 4 ,,

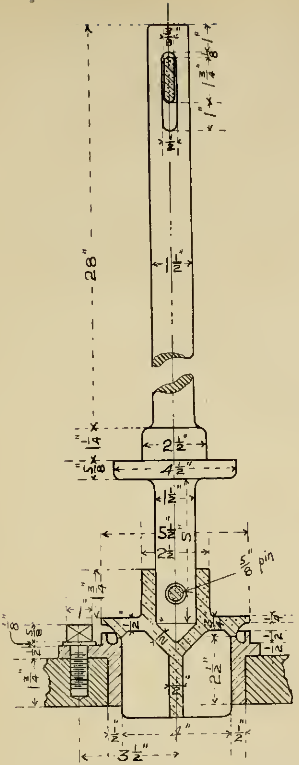
For pumps of the sizes given, the above are the average dimensions and proportions adopted in practice.



No 7.—Steam Stop Valve and Seat.

(Draw to a 4" scale.)

The valve seat is made of brass.
The valve is made of brass.
The seat is secured by three set pins $\frac{5}{8}$" diameter.



No. 8.—Valve and Seat for Spring Safety Valve.

(Draw to a 4" scale.)

The valve is made of brass.
The spindle ,, ,,
A pair of valves as shown are sufficient for 100 square feet of grate.

SECOND CLASS DRAWINGS.

(Given at Board of Trade Examinations.)

UNDER the revised Regulations for Second Class Certificates (1917), candidates are now required to make a sketch or, if preferred, a scale drawing of some simple part of the machinery or boilers.

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LIST OF DRAWINGS.

(SECOND CLASS.)

NOTE.—It is important that all sizes be marked on the drawings.

1. Main Bearing Bolt, complete.
2. Bottom End Bolt, complete.
3. Tapered Coupling Bolt, complete.
4. Distance Piece for Bottom End Brass.
5. Main Bearing "Brass."
6. Bottom End Brass.
7. Steam Stop Valve and Seat.
8. Safety Valve, Seat, and Spindle.
9. Winch Slide Valve.
10. Slide Valve Spindle.
11. Eccentric Pulley.
12. Eccentric Strap.
13. Crank Shaft and Webs.
14. Thrust Shaft.
15. Tail End Shaft.
16. Piston Rod and Crosshead.
17. Guide Shoe.
18. Gland and Stuffing Box.
19. Winch Cylinder Piston.
20. Air Pump Valve.
21. Feed Pump Plunger.
22. Tee Piece for Steam Pipe.
23. Link and Lever for Indicator Gear.
24. Box Spanner.
25. Spanner.
26. Boiler Manhole.
27. Fire Bar.
28. Fire Bar Bearer.
29. Stay Tube and Common Tube.
30. Tube Stopper.
31. Combustion Chamber Girder.
32. Double Butt Strap Joint.
33. Main Stay.
34. Combustion Chamber Stay.
35. Double Riveted Lap Joint.
36. Crank Pin Disc for Winch.
37. Check Valve Chest Cover and Spindle.
38. Hand Wheel for Steam Stop Valve.
39. Air Vessel.
40. Water Gauge Column Bracket.

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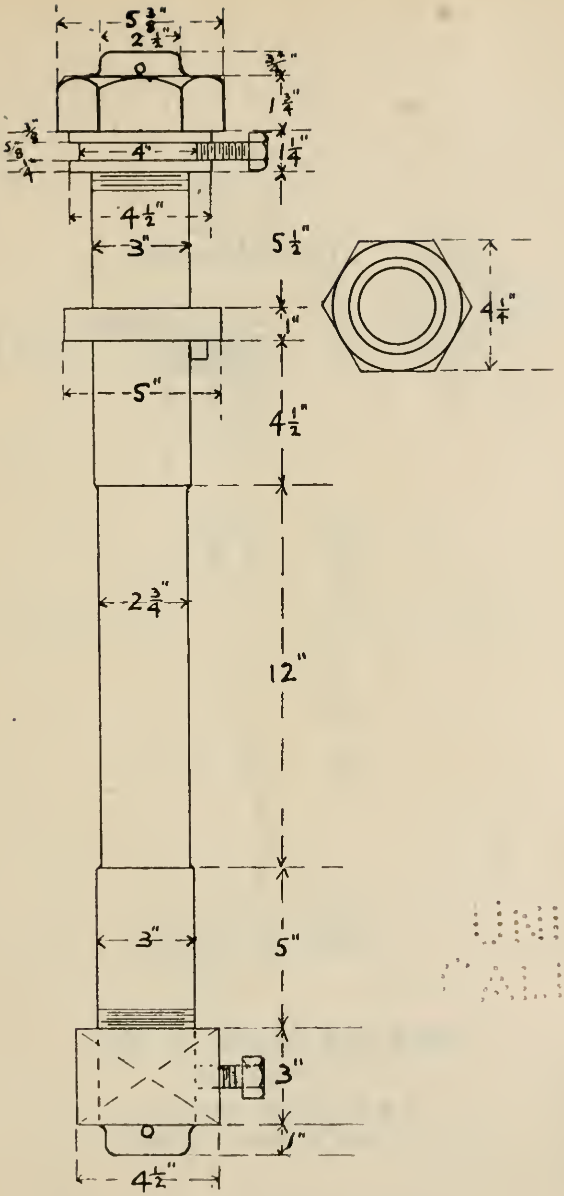
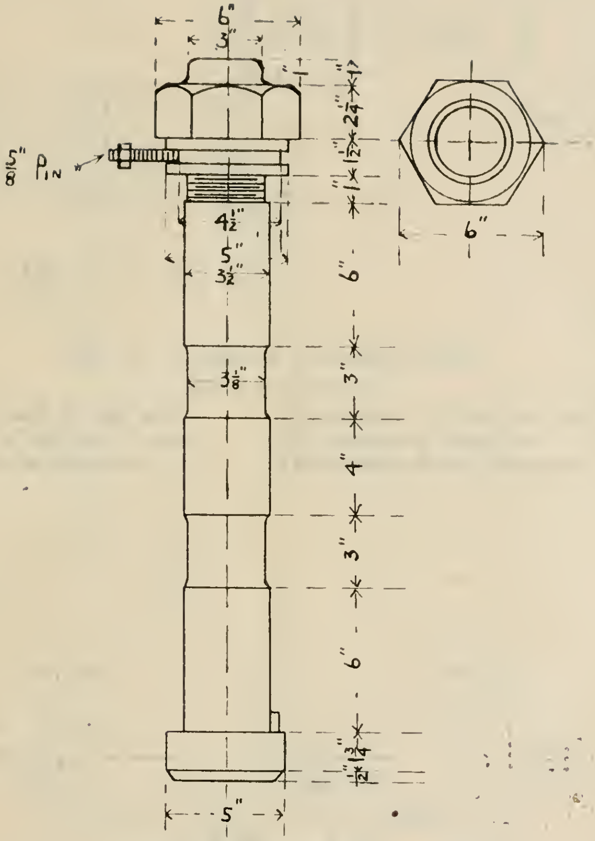


No. 1.—Main Bearing Bolt.

(Draw to a 3" scale.)

The total stress on bolt = Area of bolt × 3000 lbs. $= 2{\cdot}75^2 \times \frac{11}{14} \times 3000 = 17820$ lbs.
The bolt is made of mild steel, $3\frac{1}{2}$ threads per inch.
The nuts are made of iron.

NOTE.—Allow 3000 lbs. tensile stress per $\boxed{''}$.

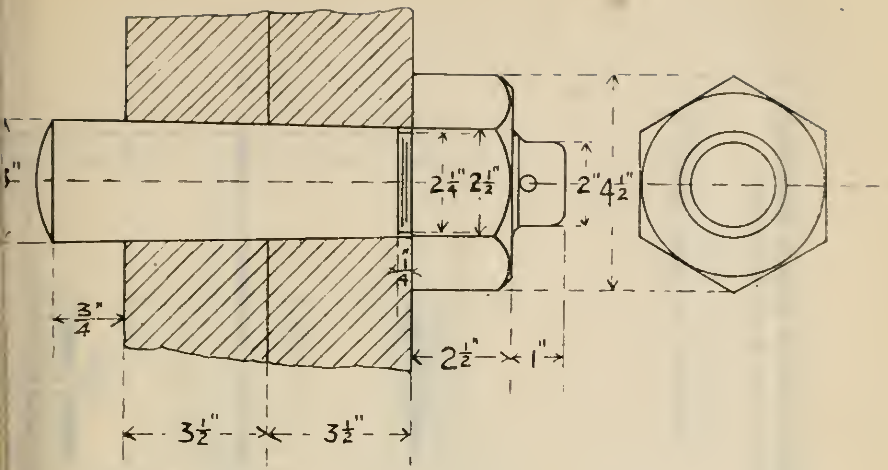


No. 2.—Bottom End Bolt.

(Draw to a 3″ scale.)

The diameter of the rod is 6″.

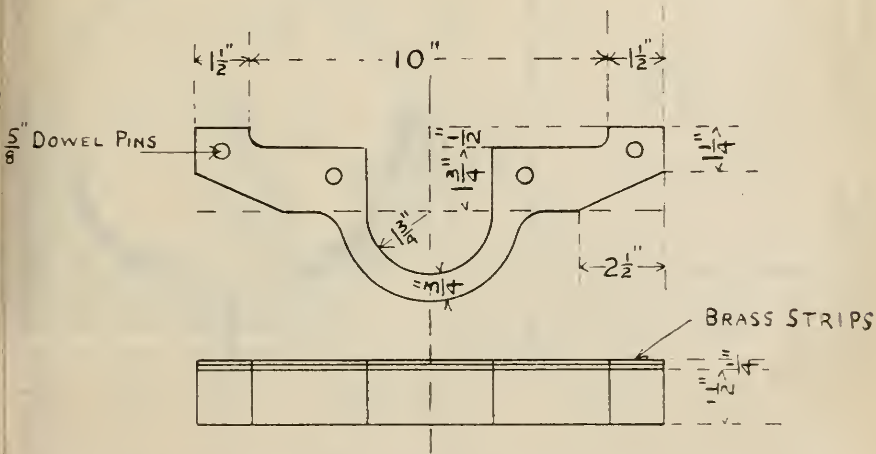
The bolt is made of steel.



No. 3.—Tapered Coupling Bolt.

(Draw to a 4″ scale.)

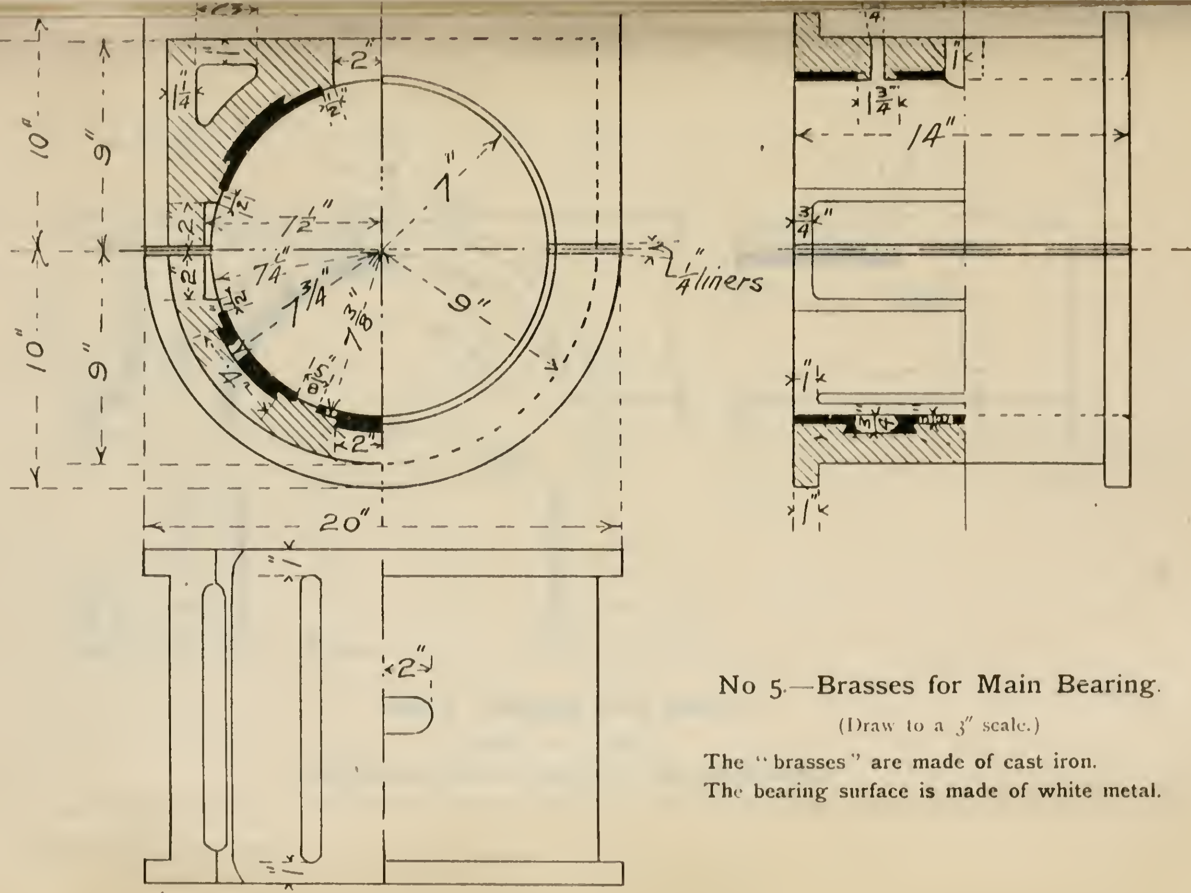
The bolt is made of mild steel.
The I.H.P. of engines is 1200.
The diameter of shaft is 12″.
The diameter of the pitch circle is 18″.
The number of bolts is 6.
The revolutions per minute are 65.



No. 4.—Distance Piece for a Bottom End Brass.

(Draw to a 4″ scale.)

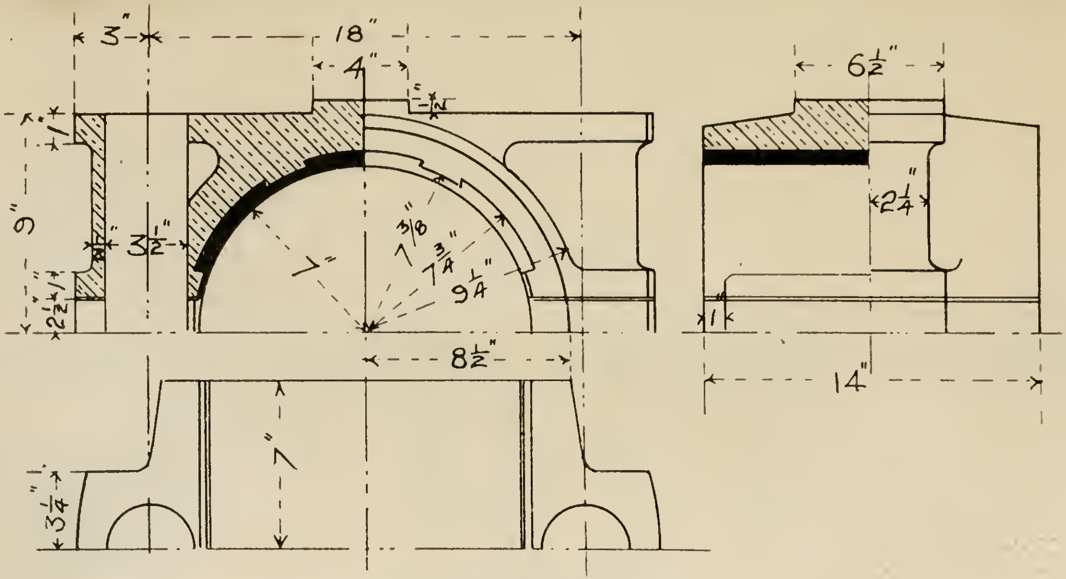
The diameter of the bolts is $3\frac{1}{2}$″.
,, ,, connecting rod is 7″.



No 5.—Brasses for Main Bearing.

(Draw to a 3" scale.)

The "brasses" are made of cast iron.
The bearing surface is made of white metal.

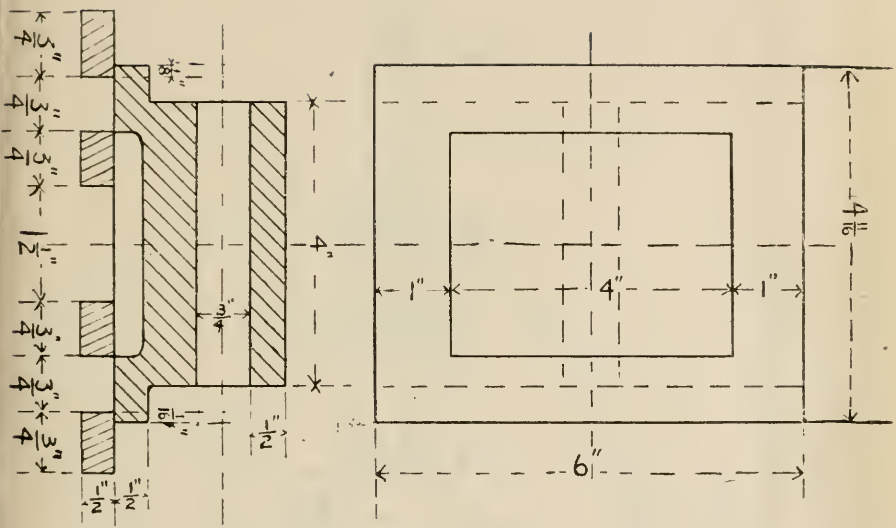


No. 6.—Bottom End Brass.

(Draw to a 3″ scale.)

The brass is made of brass lined with white metal.

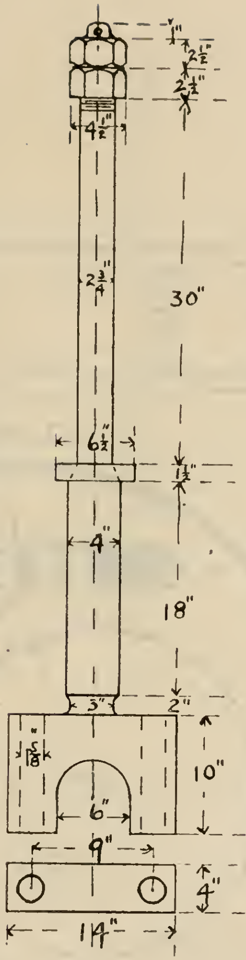
$4\frac{11}{16}$



No. 9.—Winch Slide Valve.

(Draw to a 6" scale.)

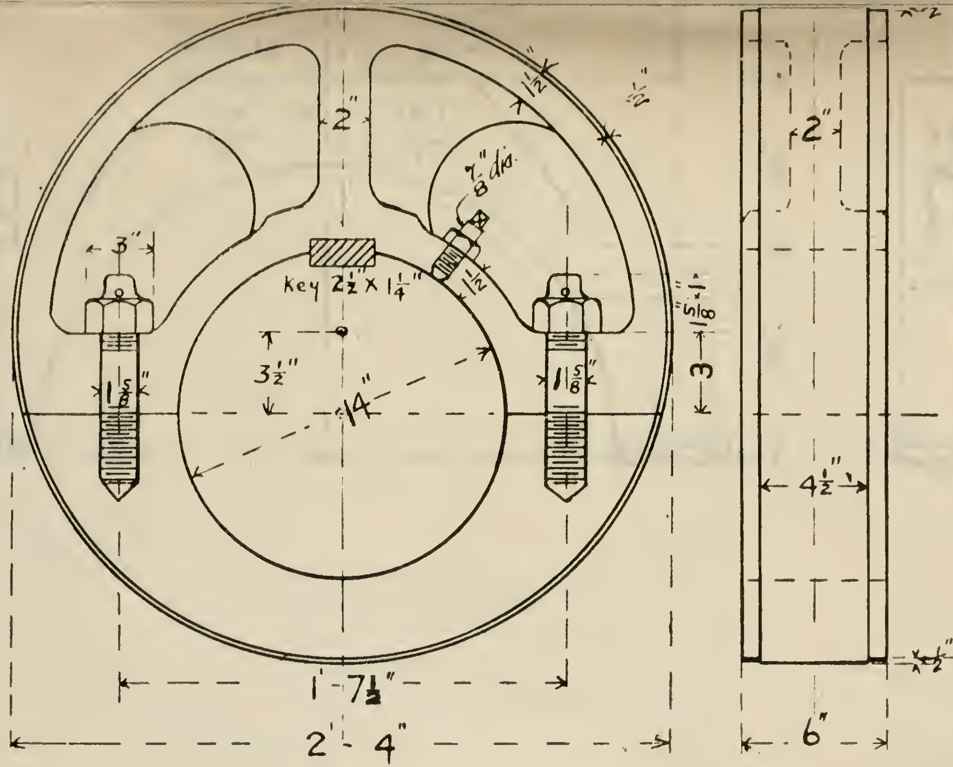
The travel of the valve is $1\frac{1}{2}$".
The lead ,, ,, $\frac{1}{16}$" top and $\frac{1}{8}$" bottom.
The lap ,, ,, $1\frac{1}{8}$" ,, ,, $\frac{1}{16}$" ,,



No. 10.—Slide Valve Spindle.

(Draw to a 2″ scale.)

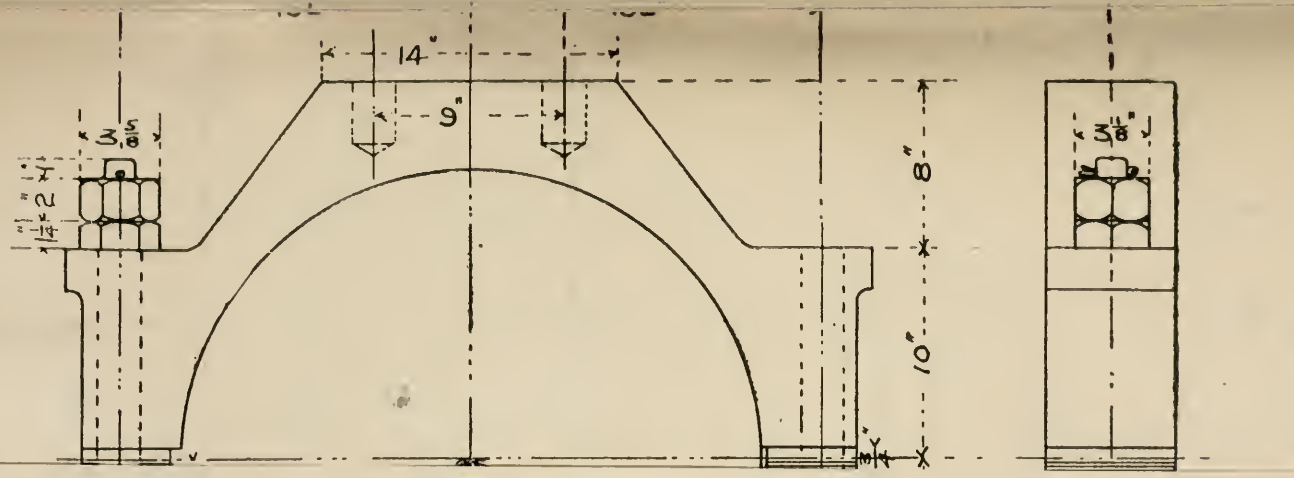
The valve spindle is made of mild steel.
The nuts are made of brass.
The boiler pressure is 100 lbs. per square inch.

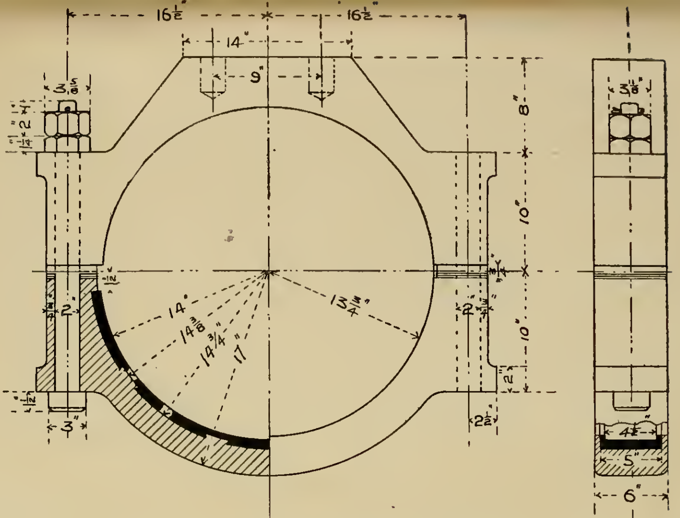


No. 11.—Eccentric Sheave.

(Draw to a 2″ scale.)

The sheave is made of cast iron.
The travel of the valve is 7 in.

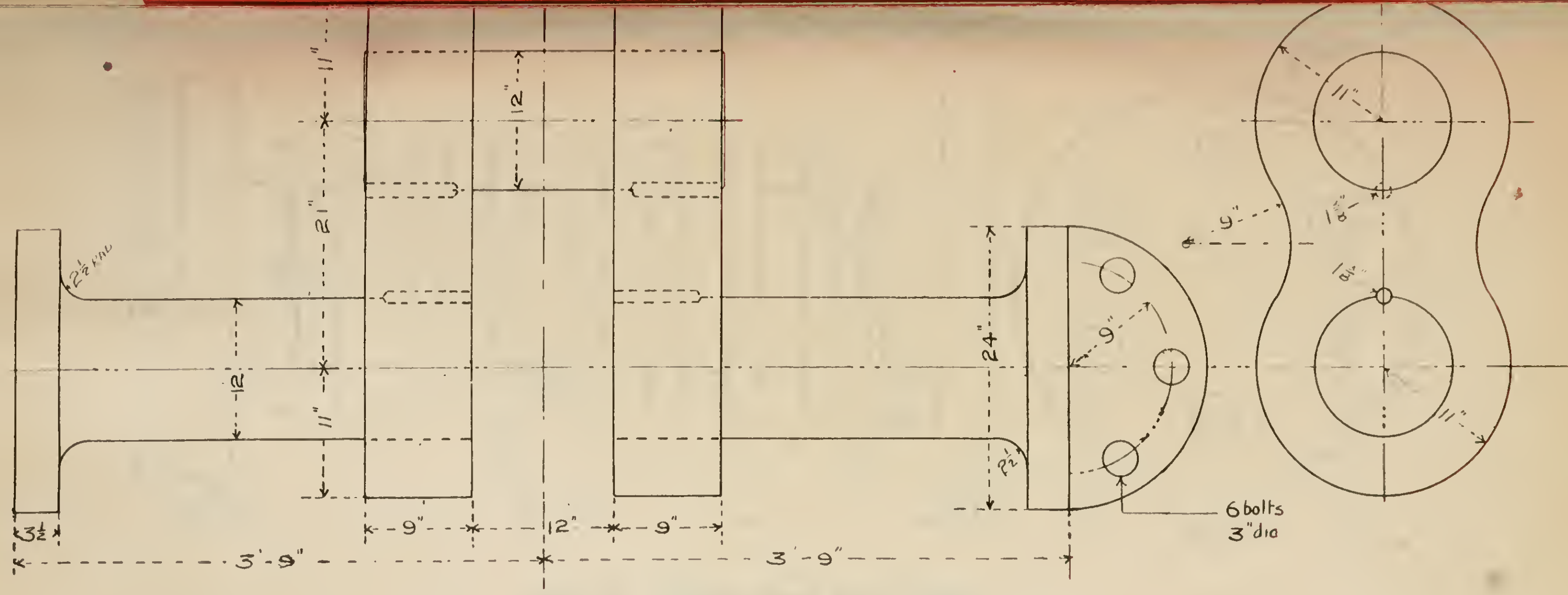

14"
9"
8"
10"



No. 12.—Strap for an Eccentric Sheave.

(Draw to a 2" scale.)

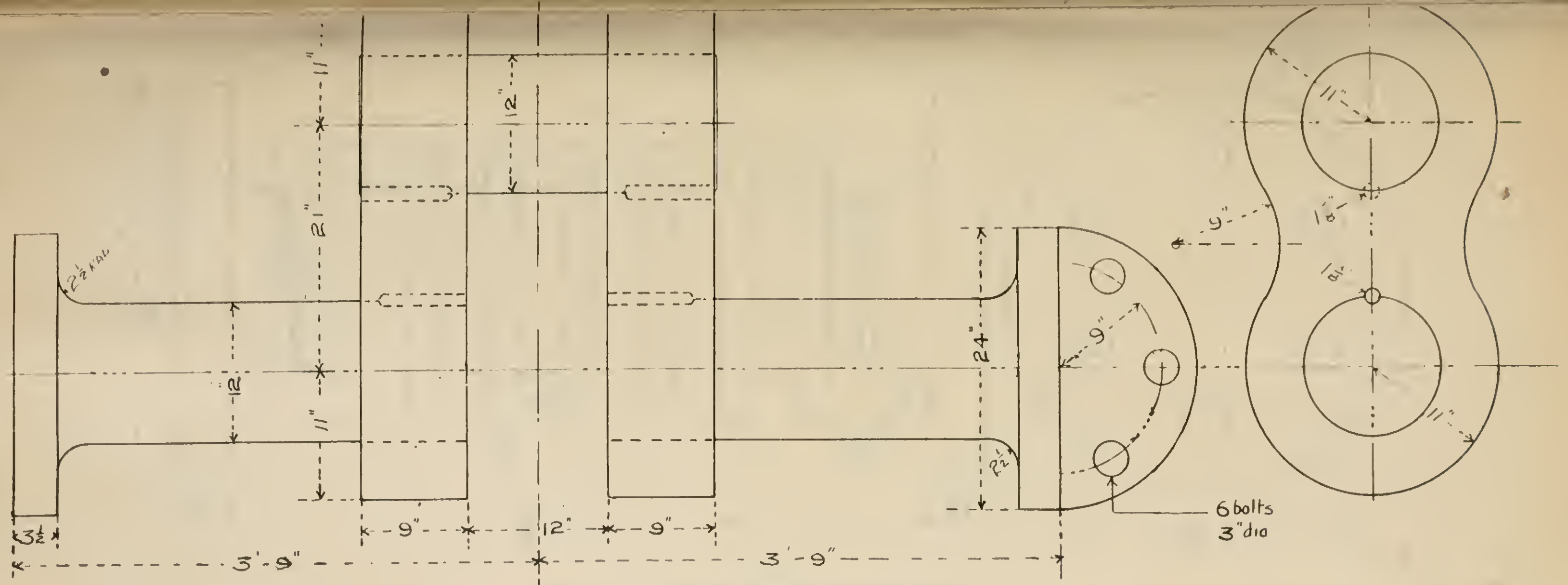
The strap is made of cast iron.
The bearing surface of white metal.
The diameter of shaft is 12".



No. 13.—Crank Shaft and Web.

(Draw to a 1" scale.)

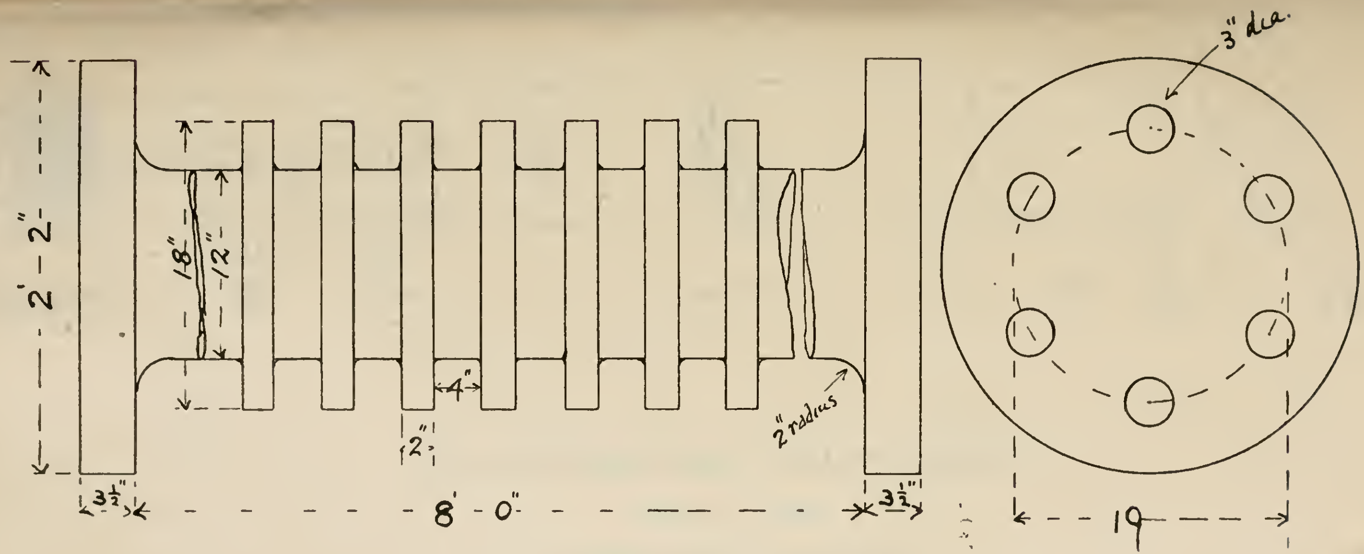
The boiler pressure is 180 lbs. per square inch.
1 H.P. = 1200.
Shaft of mild steel.



No. 13.—Crank Shaft and Web.

(Draw to a 1" scale.)

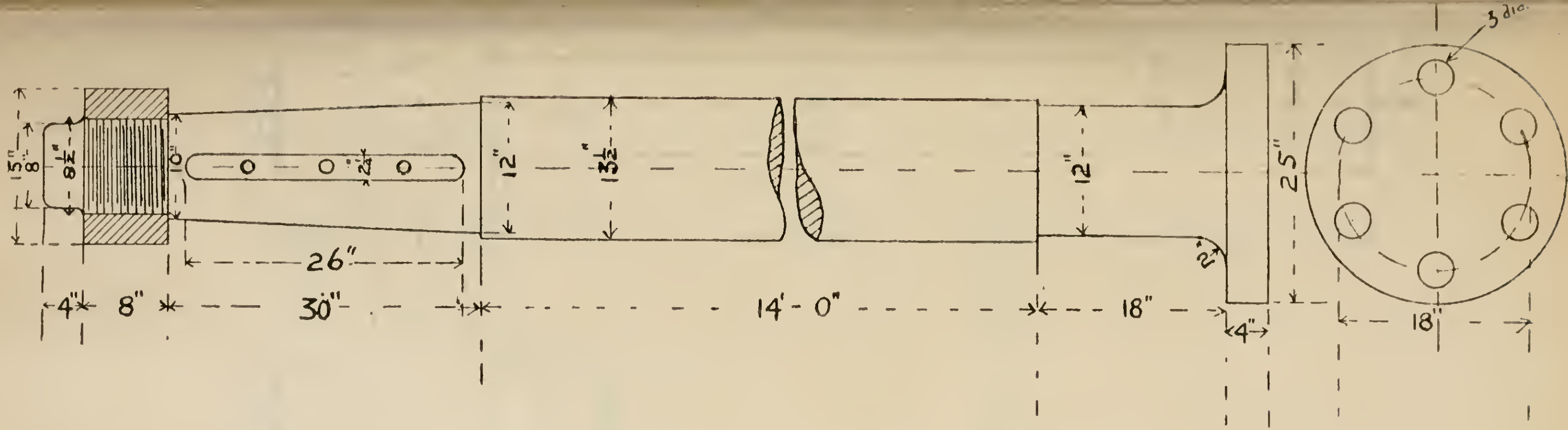
The boiler pressure is 180 lbs. per square inch.
1 H.P. = 1200.
Shaft of mild steel.



No. 14.—Thrust Shaft and Collars.

(Draw to a 1" scale.)

The shaft is made of forged steel.
The I.H.P. of engines is 1200.
Pressure on shoes about 50 lbs. per □"

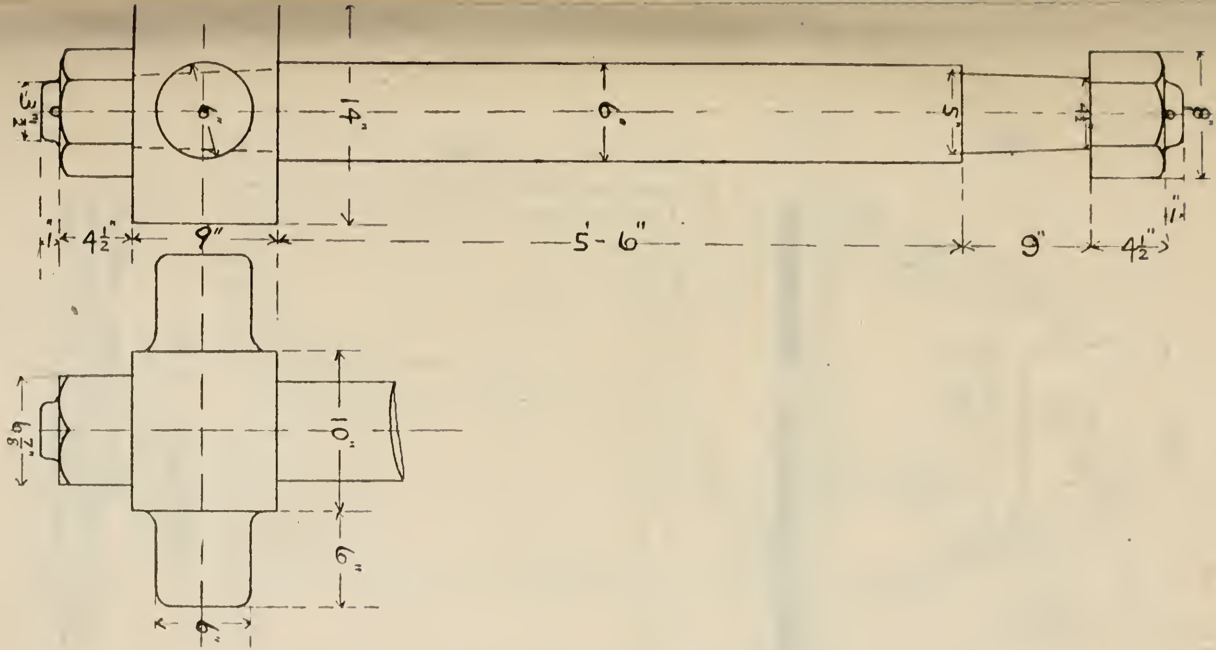


No. 15.—Propeller Shaft with Liner and Nut.

(Draw to a 1" scale.)

The stresses exerted on this shaft are:—

Torsion.	Compression (going ahead).
Bending.	Tensile (going astern).
Shaft of wrought iron.	Liner of brass.
Nut of iron. I.H.P. about 1200.	Revs. 65.



No. 16.–Piston Rod, Crosshead, and Nut.

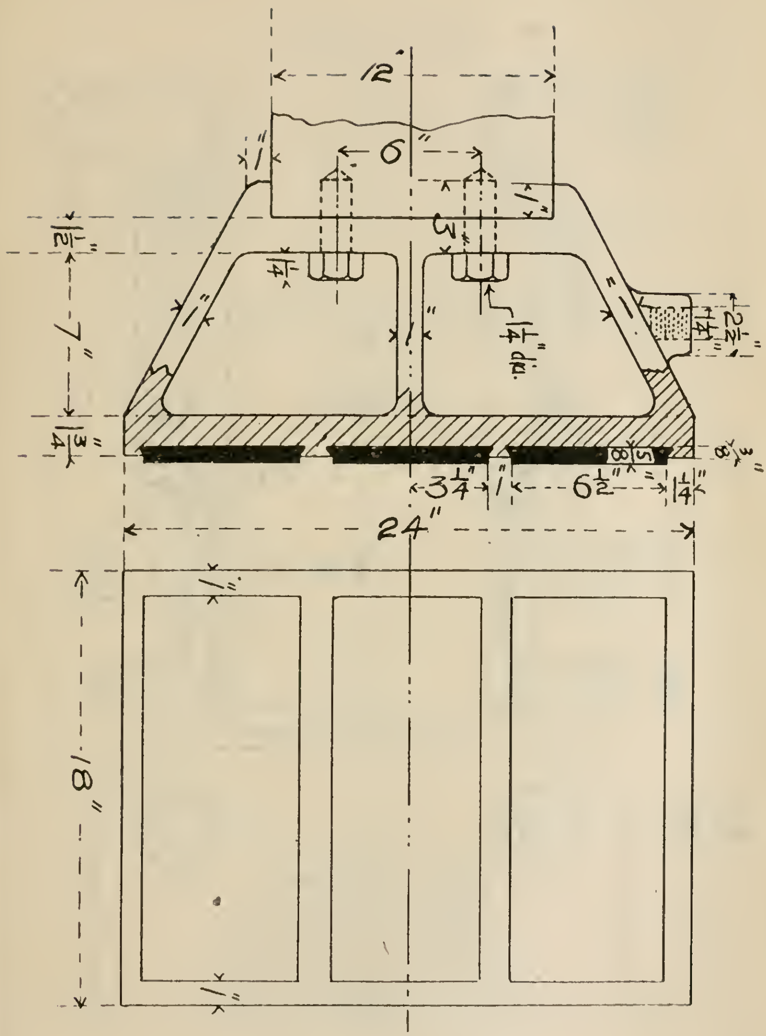
(Draw to a 1½" scale.)

The rod is made of mild steel.

The stroke of the engine is 42".

The boiler pressure is 180 lbs. per $\boxed{''}$.

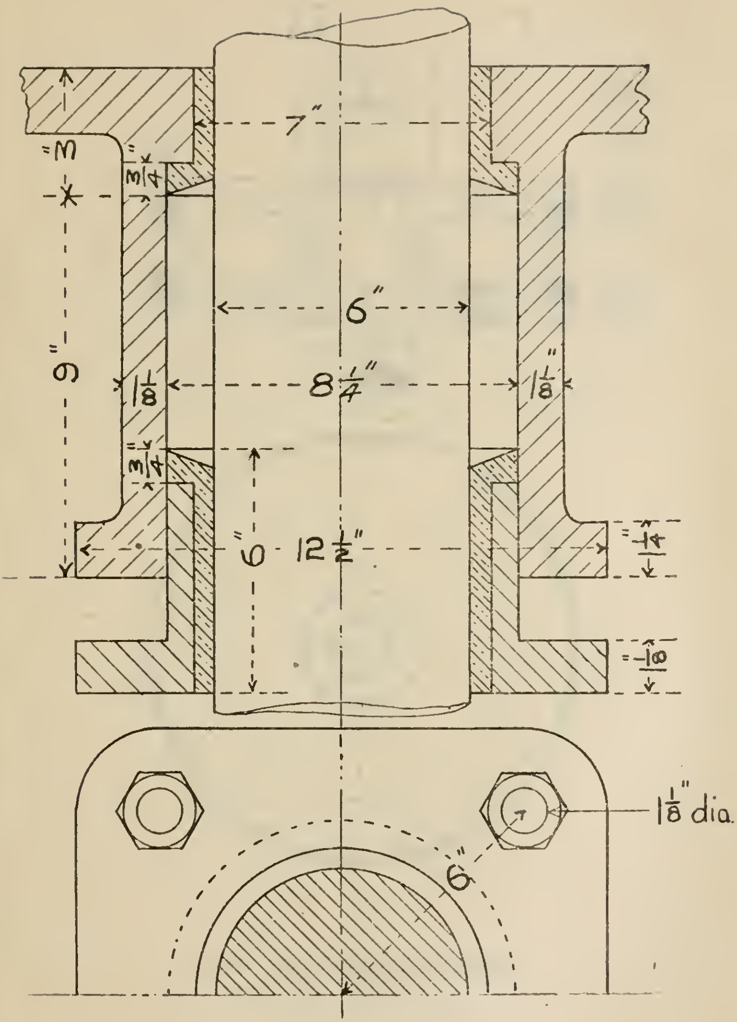
The diameter of the H.P. piston is 24"



No. 17.—Guide Shoe.

(Draw to a 2″ scale.)

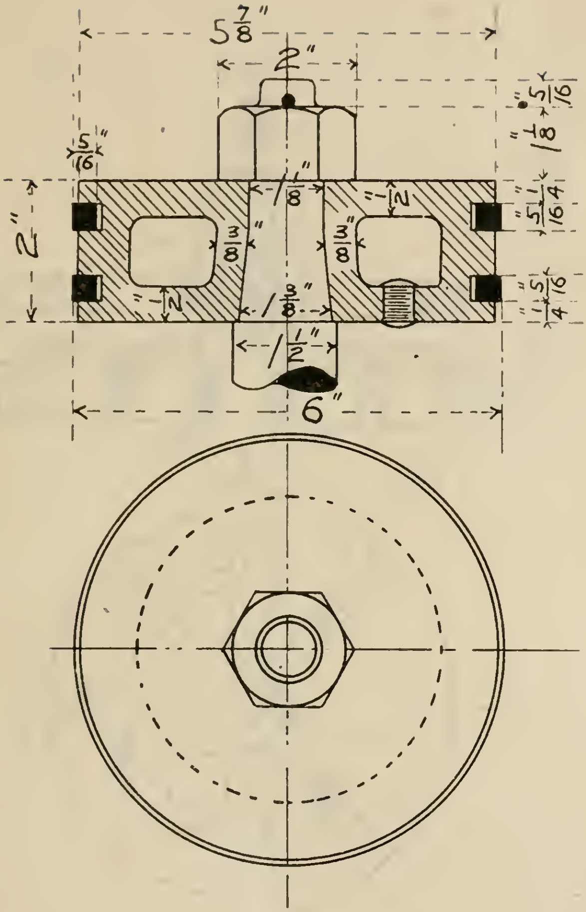
The shoe is made of cast iron.
The rubbing surface is made of white metal.
The pressure on the guide is about 40 lbs. per $\boxed{''}$.



No. 18.—Gland and Stuffing Box for L.P. Cylinder.

(Draw to a 3" scale.)

The gland is made of cast iron.
The gland bush is made of brass.



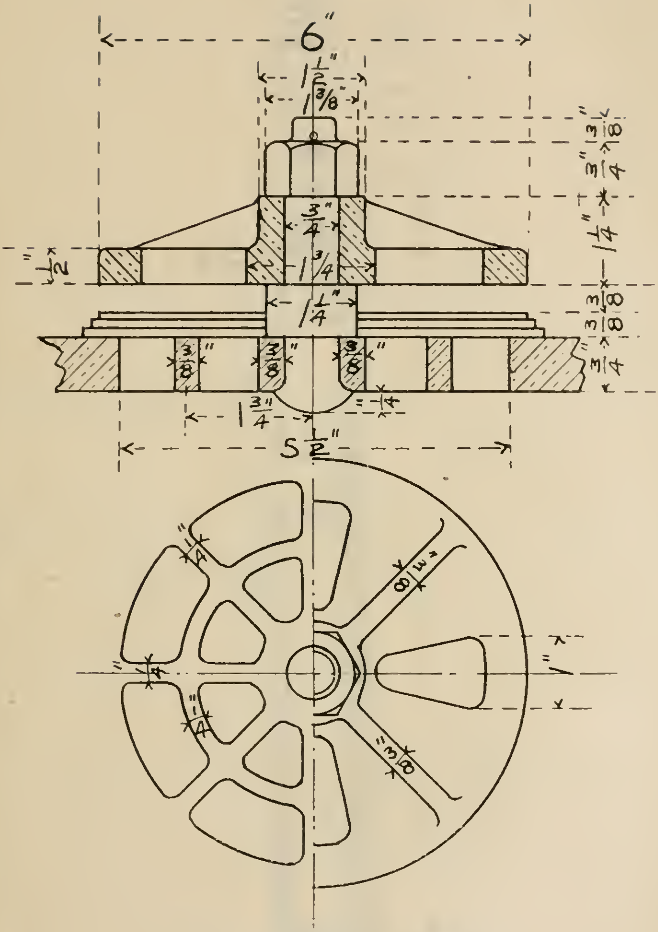
No. 19.—Winch Piston.

(Draw to a 6″ scale.)

The piston is made of cast iron.

The packing rings are made of cast iron.

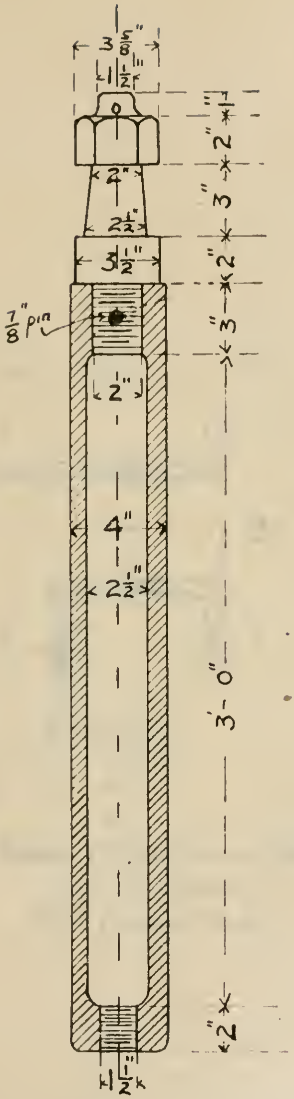
The difference in diameter allowed before cutting ring is $\frac{3}{16}''$.



No 20.—Air Pump Valve.

(Draw to a 6″ scale.)

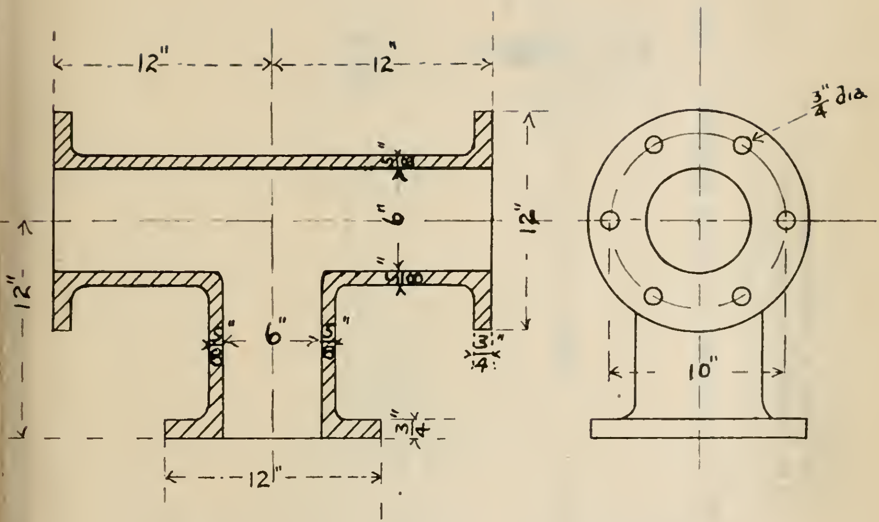
The guard is made of brass.
The valve is formed of thin brass plates.



No. 21.—Plunger for Feed Pump.

(Draw to a 2" scale.)

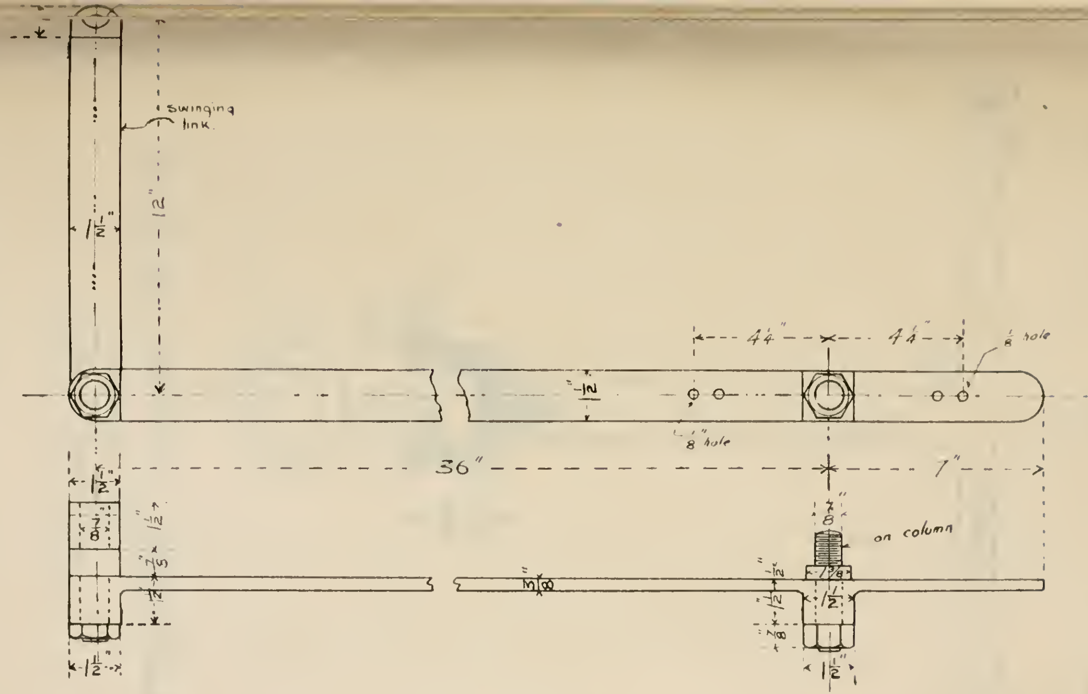
The plunger is made of brass.
The stud at top is of wrought iron.



No. 22.—Brass T-Piece for a Steam Pipe.

(Draw to a 2" scale.)

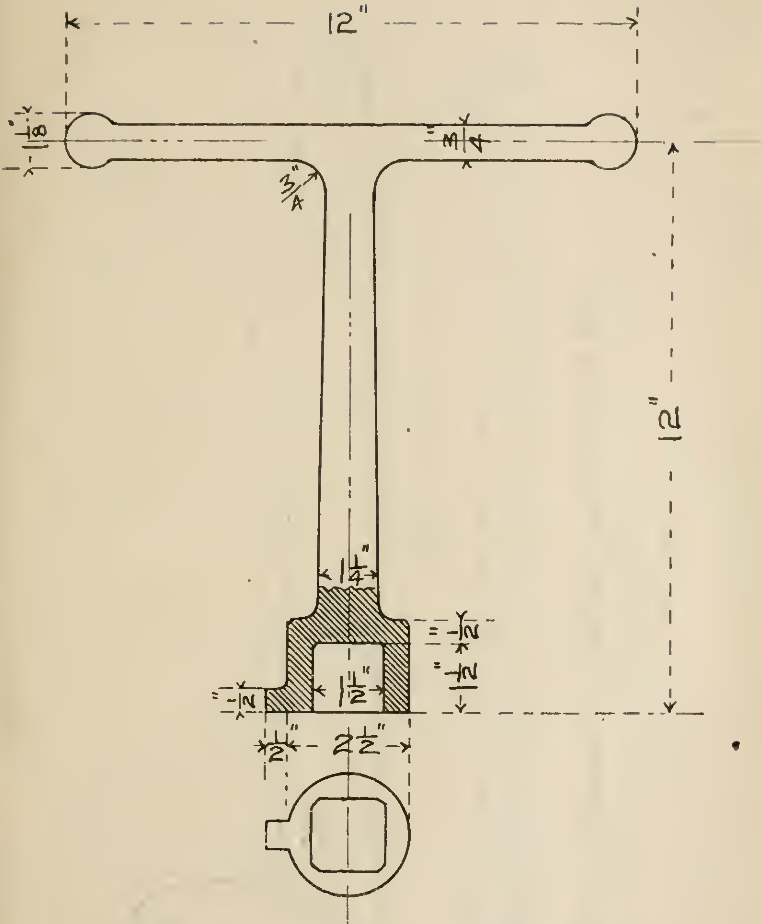
Boiler pressure, 150 lbs.



No. 23.—Link, Lever, and Pins for Indicator Gear.

(Draw to a 3″ scale.)

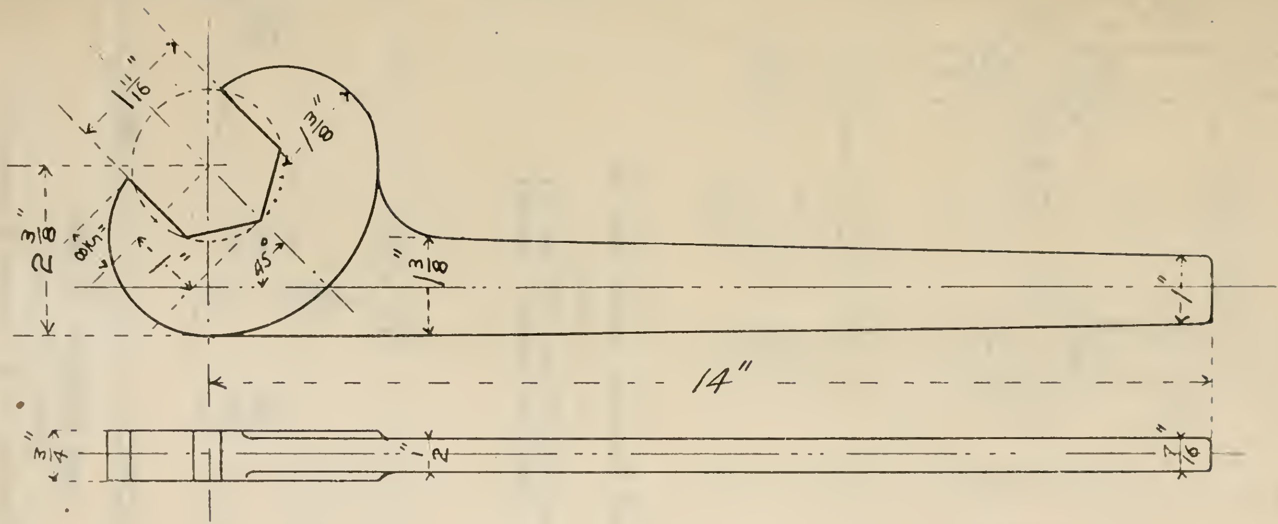
The stroke of the engine is 42″.
The length of diagram card is 5″.



No. 24.—Box Spanner for a Blow-off Cock.

(Draw to a 4" scale.)

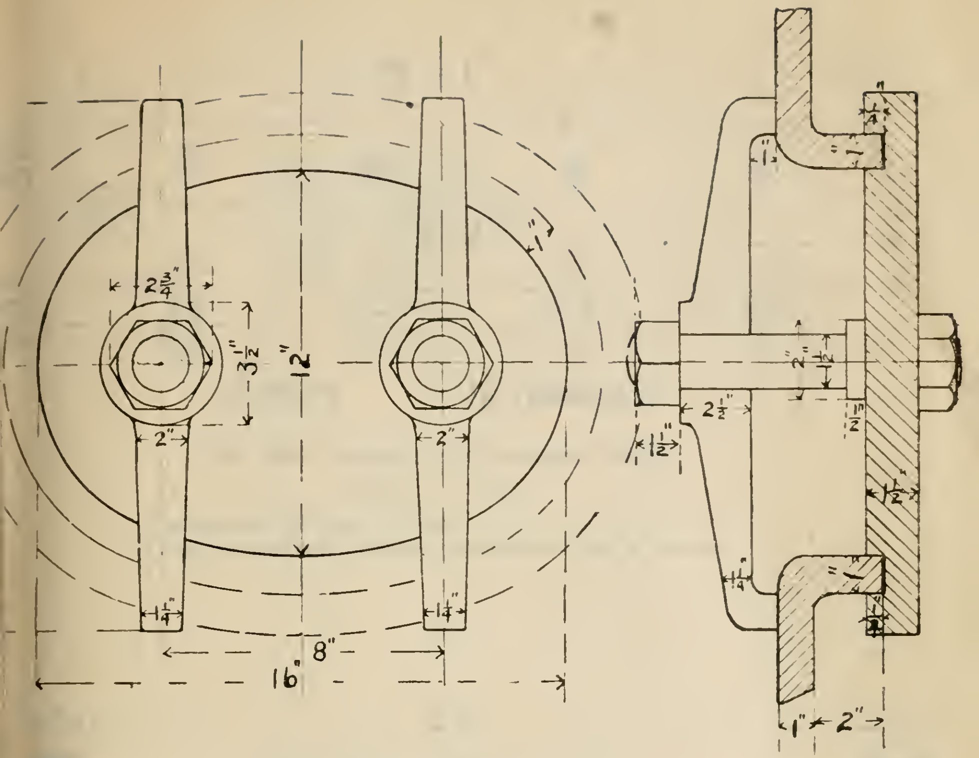
The spanner is made of iron.



No. 25.—Spanner for Nut for 1-inch Diameter Bolt.

(Draw to a 6″ scale.)

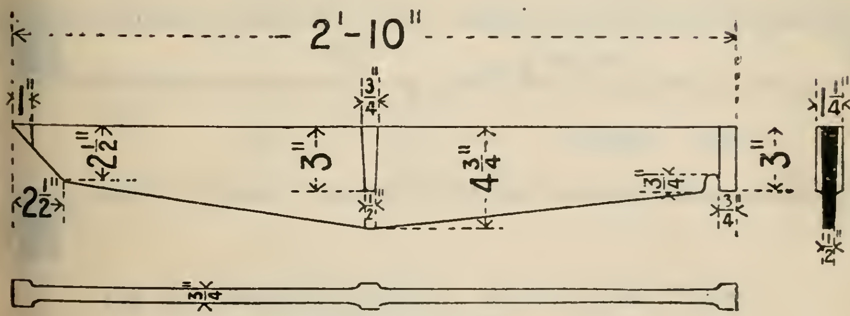
The angle of jaw with shank is 45°.



No. 26.—Boiler Manhole Door.

(Draw to a 2″ scale.)

The size of the manhole is 16″ × 12″.
The door is made of mild steel.

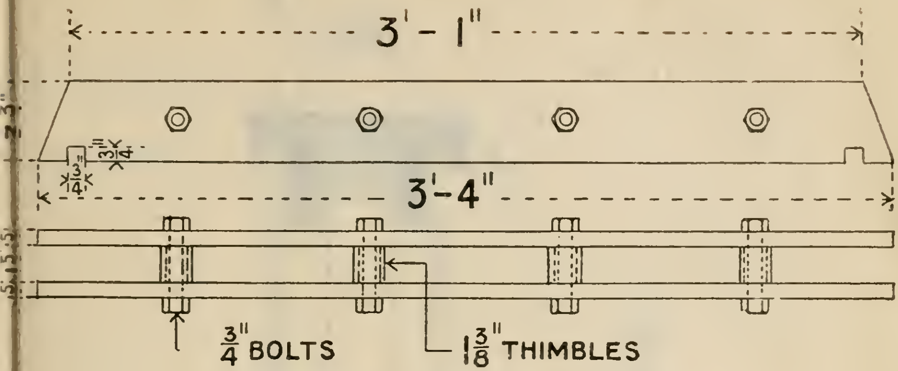


No. 27—Furnace Bar.

(Draw to a 2″ scale.)

The amount of expansion of bar averages about ¼″.*

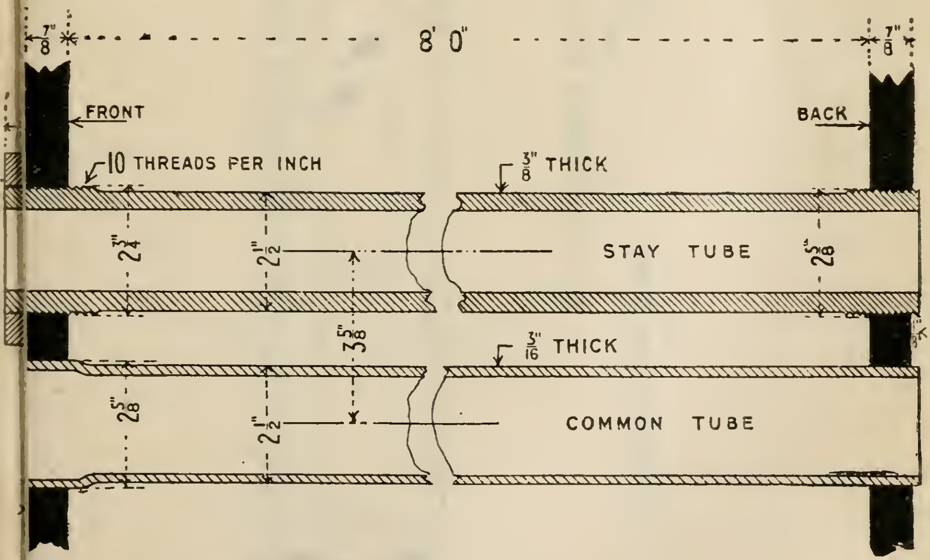
r.1" Notes and Sketches.



No. 28.—Bearer for Furnace Bars.

(Draw to a 2" scale.)

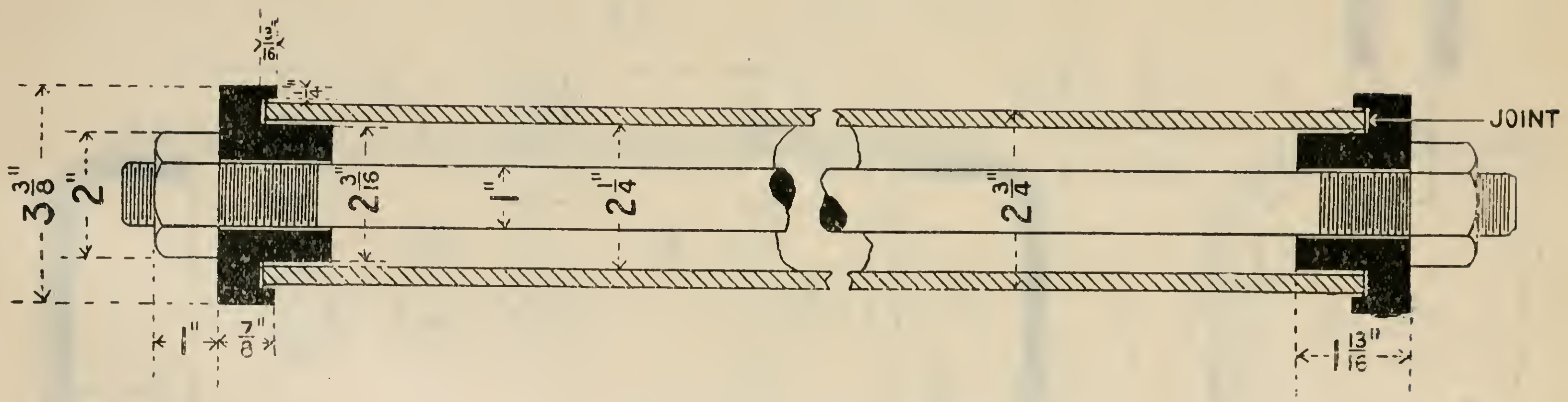
The bearers are made of iron.
They are supported at ends by snugs on side of furnace.



No. 29.—Boiler Stay Tube; Boiler Smoke Tube.

(Draw to a 4" scale.)

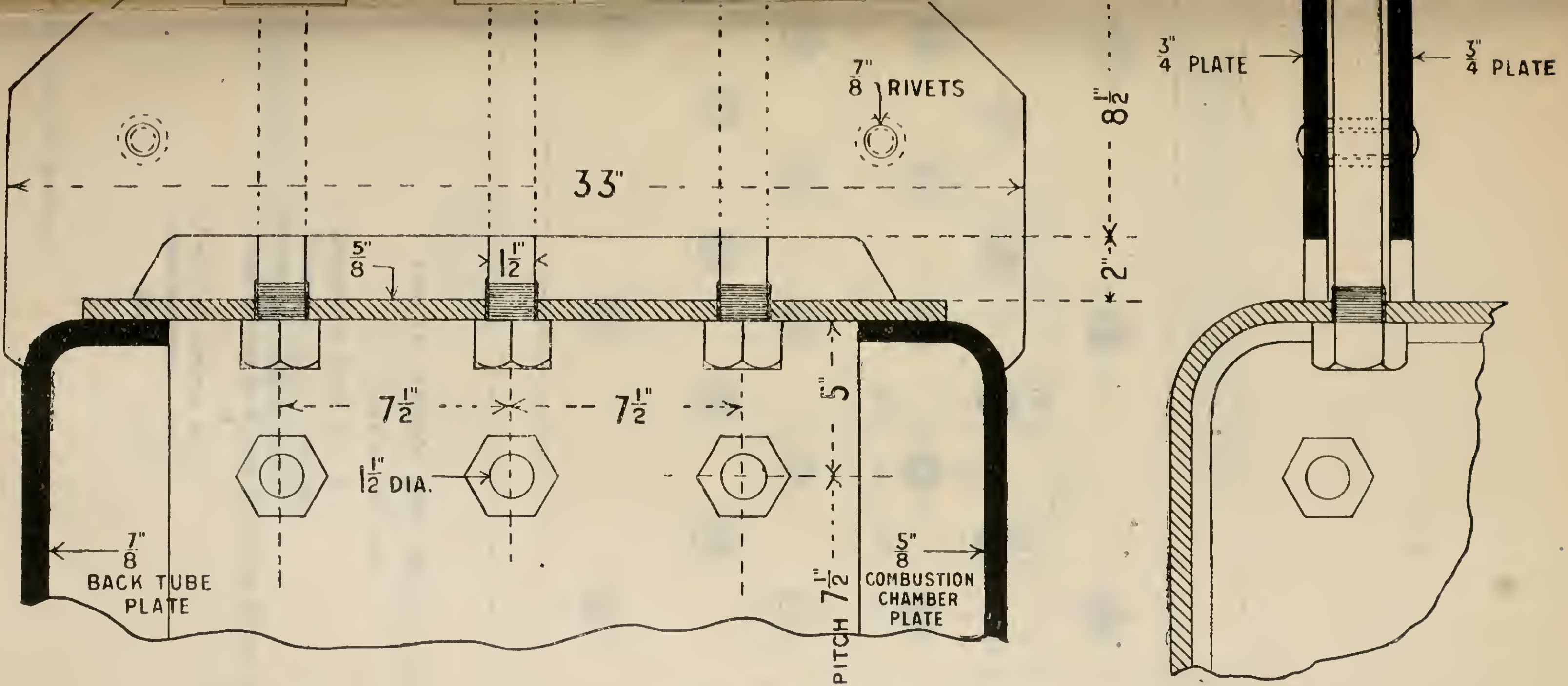
These tubes are made of iron.
The number of stay tubes fitted is about ⅓ of total tubes.



No. 30.—Common Tube Stopper.

(Draw to a 4" scale.)

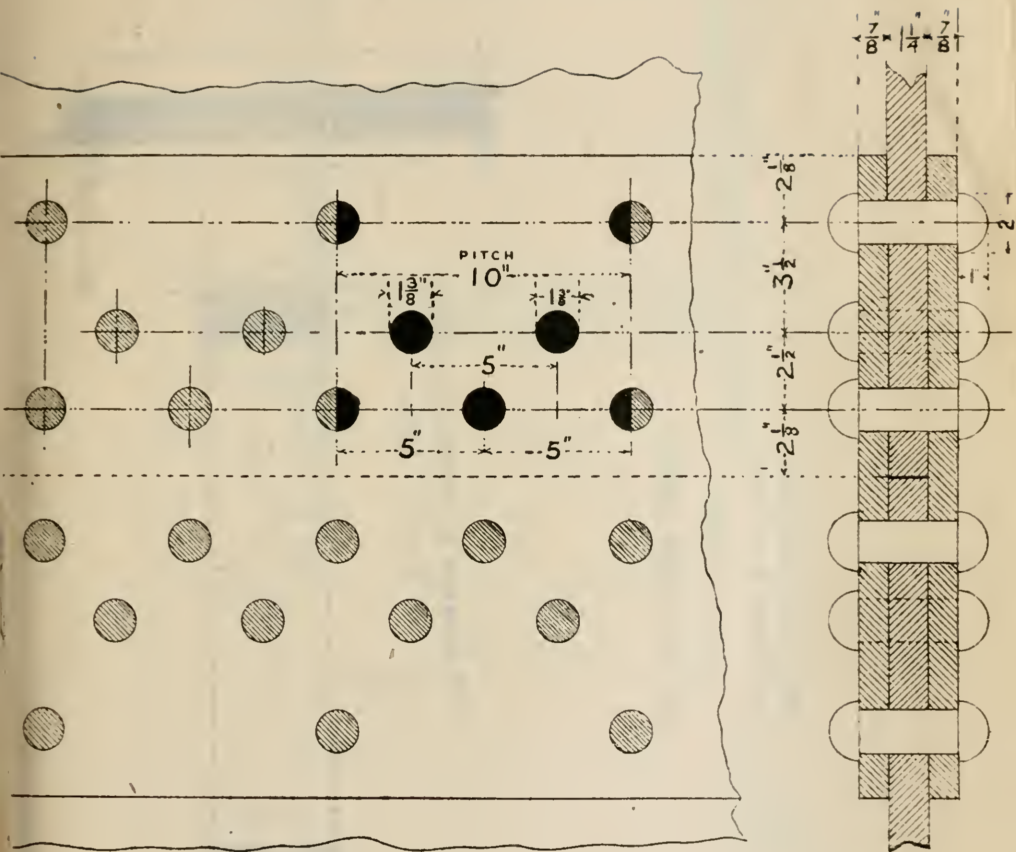
The stopper is made of iron. The joint is made of asbestos tape.



No. 31.—Combustion Chamber Girder.

(Draw to a 2″ scale.)

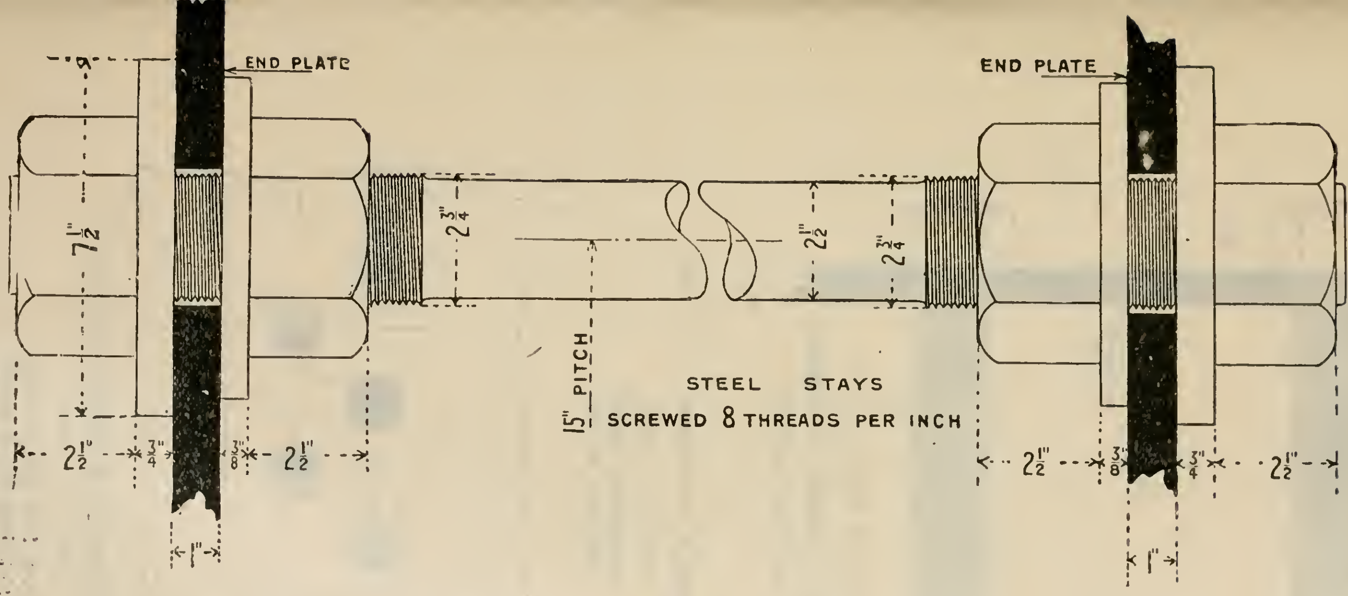
The stays are 1½″ in diameter.
,, ,, 7½ ,, pitch.
The girders are placed 8½″ apart.
,, ,, ,, of mild steel.
,, stays ,, of mild steel.



No. 32.—Double Butt Strap Joint.

(Draw to a 2″ scale.)

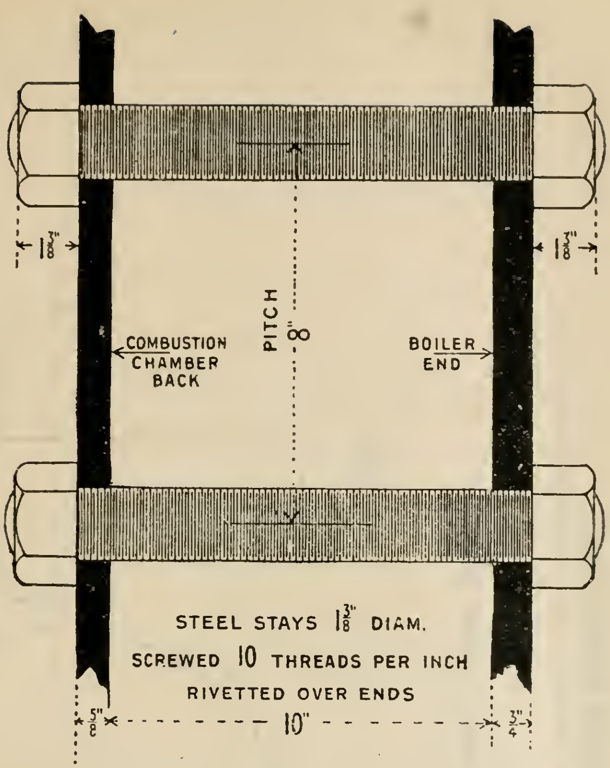
Diameter of boiler is 14′-6″.
Pressure ,, 180 lbs.
Rivets are made of steel.
Stress allowed is 23 tons per square inch on rivets.
,, ,, 28 ,, ,, plate.
Factor of safety is 4·5.
Rivet diameter ,, $1\frac{3}{8}$″.
,, pitch ,, 10″.



No. 33.—Main Stay.

(Draw to a 3″ scale.)

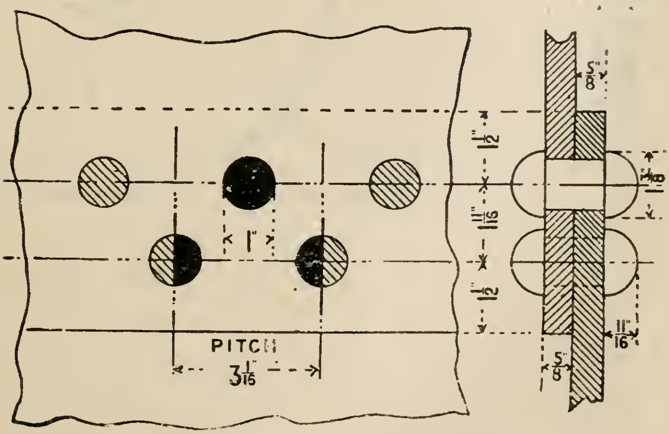
Stays made of mild steel.
Nuts made of mild steel.



No. 34.—Combustion Chamber Stay.

(Draw to a 4″ scale.)

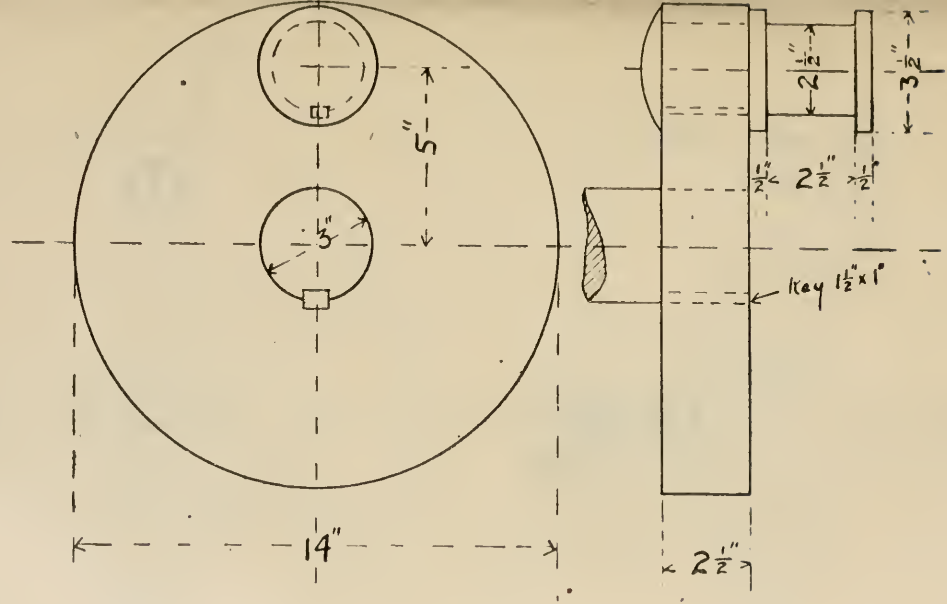
Stays of mild steel.
Nuts of mild steel.



No. 35.—Double Rivetted Lap Joint.

(Draw to a 4″ scale.)

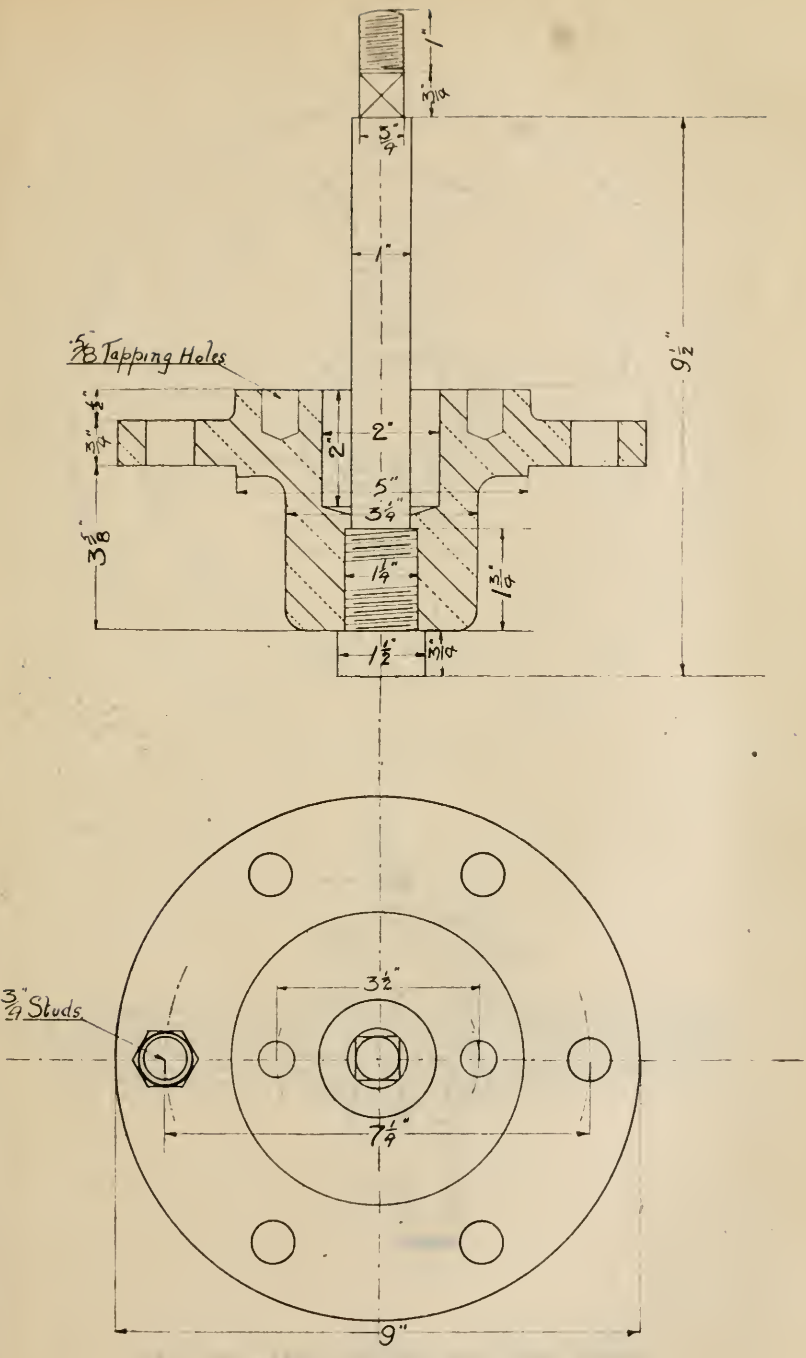
Rivets are of mild steel.
Plates are of mild steel.
Joint strength, 67 %.



No. 36.—Winch Crank Disc.

(Draw to a 3″ scale.)

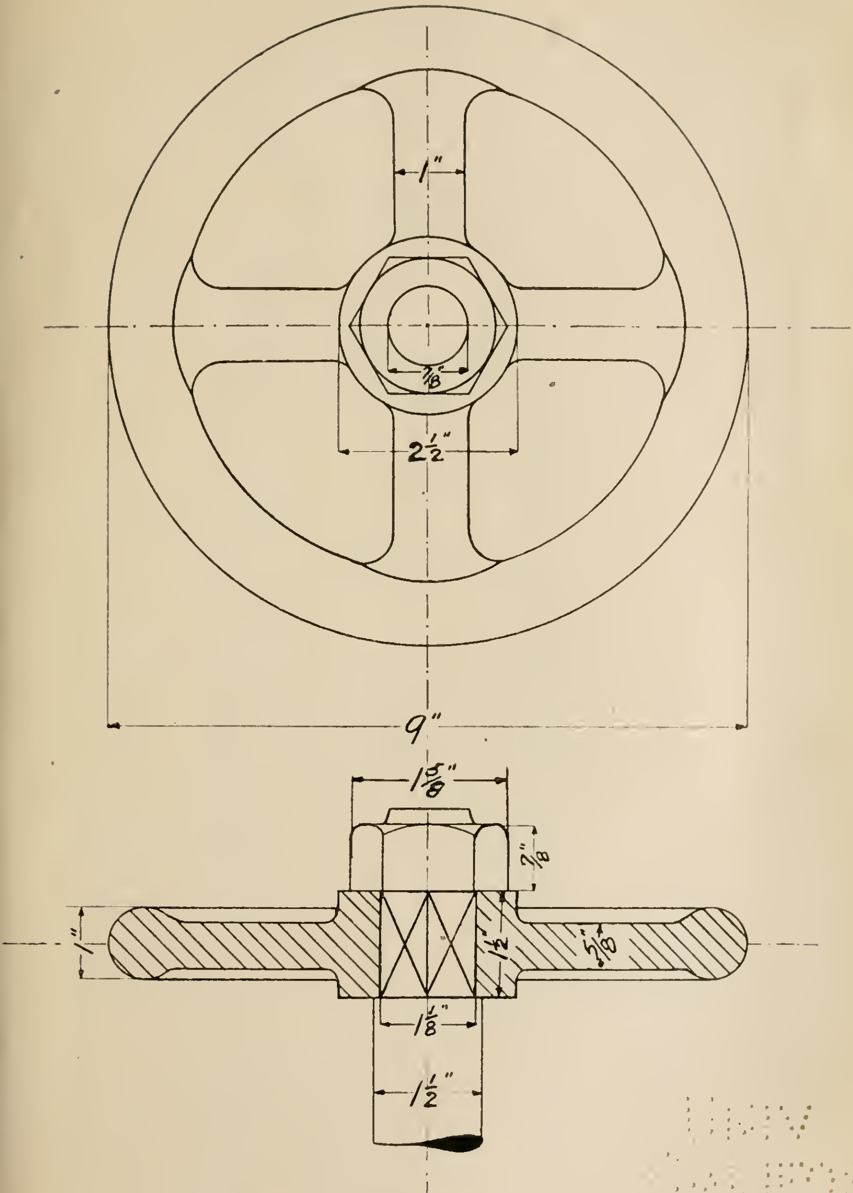
The winch is fitted with two cylinders, each 6″ diameter and stroke 10″. The disc is made of cast iron and the pin of steel.



No. 37.—Check Valve Cover.

(Draw to a 4″ scale.)

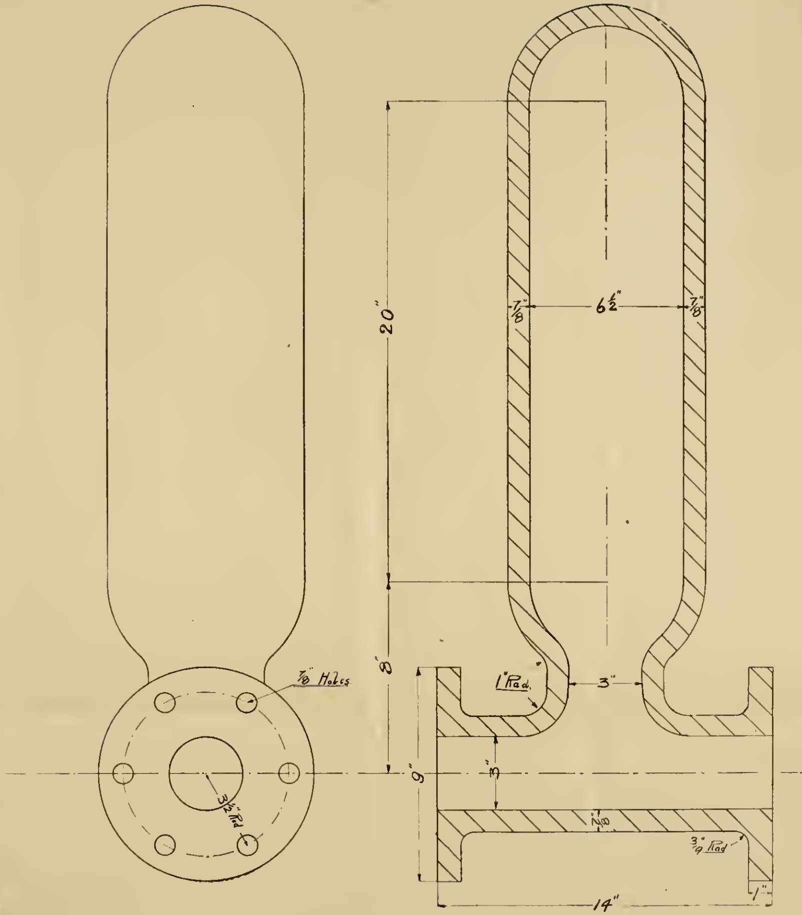
Cover of brass.
Spindle of mild steel.



No. 38.—Hand Wheel for Stop Valve.

(Draw to a 6" scale.)

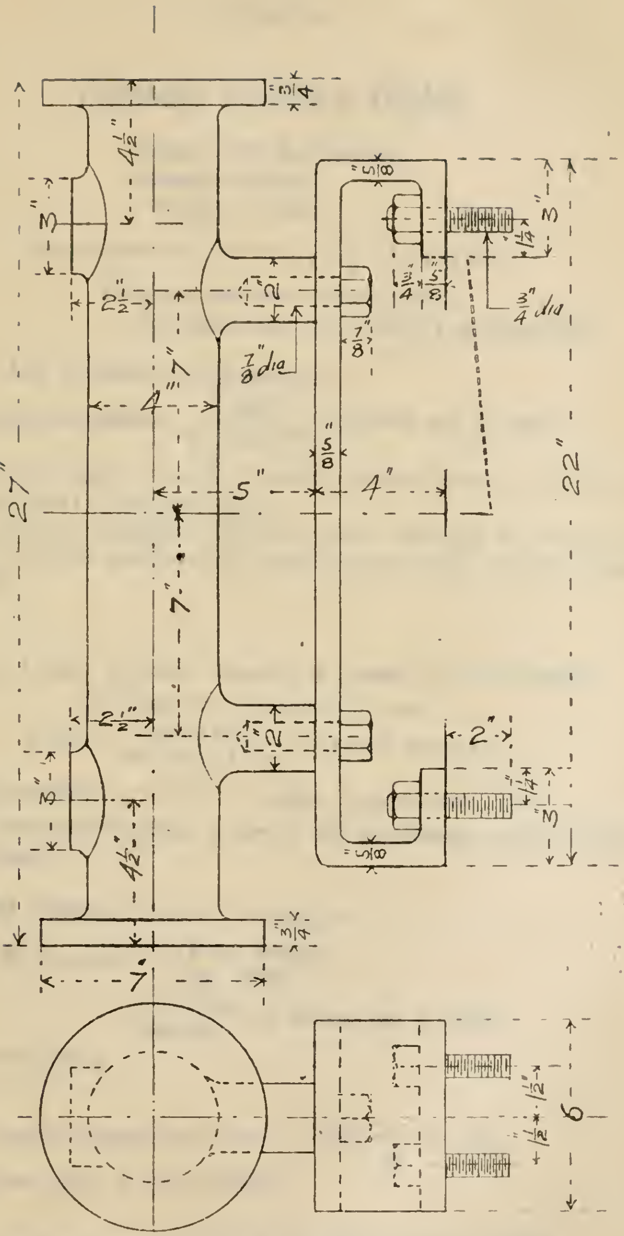
The wheel is made of cast iron.



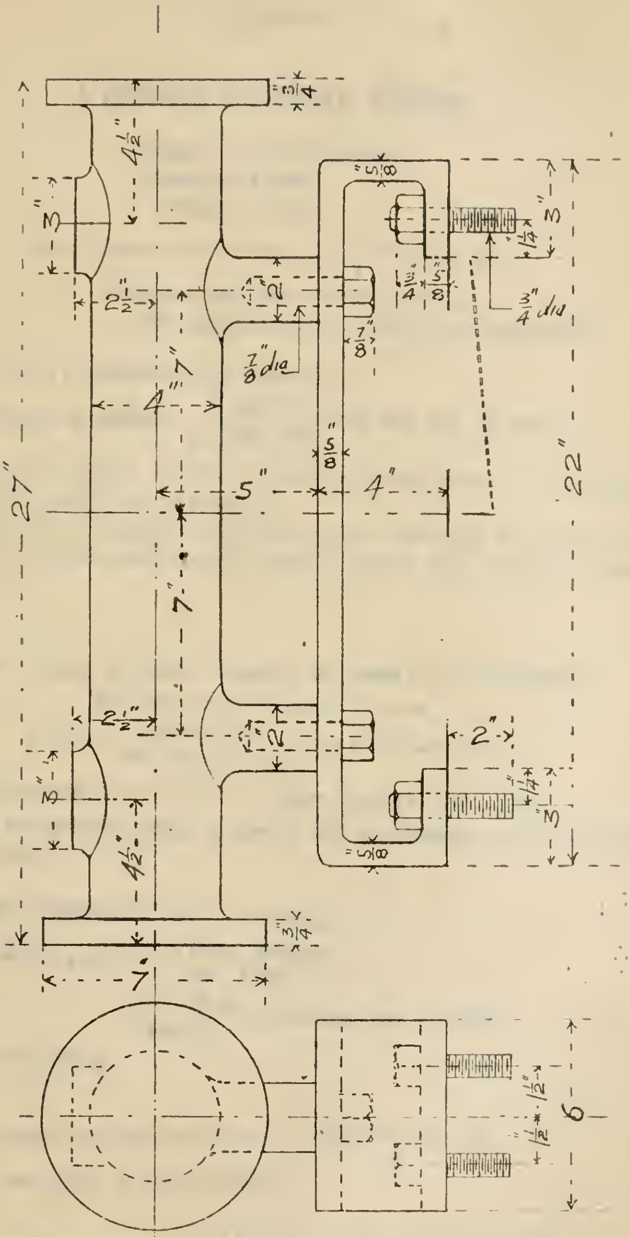
No. 39 Air Vessel

(Draw to a 2" scale.)

Air vessel of cast iron feed pump plunger 3" diameter.
Pressure in pipe, 30 lbs. (heater).



No. 40.—Water Gauge Column Bracket.



No. 40.—Water Gauge Column Bracket.

Vertical Donkey Boiler

Data.—

Pressure = 80 lbs. (gauge).
Diameter = 5 feet.
Height = 11 feet.

$$\text{Mean diameter of fire box} = \left(\frac{4\cdot5+4}{2}\right) = 4\cdot25 \text{ feet.}$$

Height of fire box = 5 feet 9 inches.

Fire grate area = $4\cdot5^2 \times \cdot7854 = 15\cdot9$ square feet.

RULE.—

Area of uptake = Grate area $\times \frac{1}{12}$.

Then, $$\text{Diameter of uptake} = \sqrt{\frac{15\cdot9\times1}{\cdot7854\times12}} = 1\cdot29 \text{ feet, say 15 inches.}$$

Vertical shell seams double riveted, circumferential shell seams and all internal parts, single riveted.

Allow a factor of safety of 6, and a joint strength of about 70 per cent. for the vertical seams and about 54 per cent. for the circumferential seams.

Shell.—

$28 \times 2240 \times \text{T inch} \times 2 \times \text{Joint} = \text{Diameter in inches} \times \text{Factor} \times \text{Pressure}.$

So that, $28 \times 2240 \times \text{T}'' \times 2 \times \cdot70 = 60'' \times 6 \times 80.$

Therefore, $$\text{T inch} = \frac{60\times6\times80}{28\times2240\times2\times\cdot70} = \cdot32 \text{ inch, say } \tfrac{3}{8} \text{ inch.}$$

Rivet diameter $= 1\cdot2\sqrt{\text{T}} = 1\cdot2 \times \sqrt{\cdot375} = \cdot734$ inch, say $\frac{3}{4}$ inch.

NOTE.—As the plate thickness is low, it will be advisable to allow rivets of, say, $\frac{13}{16}$ inch diameter.

Vertical Shell Seams (Double riveting).—

$$\text{Rivet pitch} = \frac{100 \times \text{Rivet diameter}}{100 - \text{Joint}}.$$

Then, $$\text{,,} \quad \text{,,} \quad = \frac{100\times\cdot8125}{100-70} = 2\cdot7 \text{ inches, say } 2\tfrac{3}{4} \text{ inches.}$$

NOTE.—$\frac{13}{16}$ inch $= \cdot8125$.

RULE.—

$$\text{Distance between rivet rows} = \frac{\sqrt{(11p+4d)\times(p+4d)}}{10}.$$

NOTE.—p = rivet pitch, d = rivet diameter.

Then,

$$\text{Distance between rows} = \frac{\sqrt{(11\times2\cdot75+4\times\cdot8125)\times(2\cdot75+4\times\cdot8125)}}{10}$$

$= 1\cdot41$ inches, say $1\frac{1}{2}$ inches.

Distance between edge of plate and centre of rivet $= 1\cdot5 \times d = 1\cdot5 \times \cdot8125$
$= 1\cdot21$ inches, say $1\frac{1}{4}$ inches.

Width of lap $= 1\frac{1}{4} + 1\frac{1}{2} + 1\frac{1}{4}$ inches $= 4$ inches.

Circumferential Shell Seams (Single riveting).—

Rivet diameter same as before $= \frac{13}{16} = \cdot 8125$.

$$\text{Rivet pitch} = \frac{100 \times \cdot 8125}{100 - 54} = 1 \cdot 7 \text{ inches, say } 1\tfrac{3}{4} \text{ inches.}$$

Width of lap (3 rivet diameters) $= \frac{13}{16} + \frac{13}{16} + \frac{13}{16}$ inch $= 2\frac{7}{16}$ inches.

Fire Box (Welded).—

RULE.—

$$90000 \times T^2 = (\text{Height in feet} + 1) \times \text{Diameter in inches} \times \text{Pressure.}$$

Therefore, $$T = \sqrt{\frac{(\text{Height in feet} + 1) \times \text{Diam. in inches} \times \text{Pressure}}{90000}}.$$

,, $$T = \sqrt{\frac{(5 \cdot 75 + 1) \times 51 \text{ inch} \times 80}{90000}} = \cdot 55 \text{ inch, say } \tfrac{9}{16} \text{ inch.}$$

Allow, say, six stays round uptake, each with a working stress of 9000 lbs. per square inch.

Then, Area of surface to be supported $= (54^2 - 30^2) \times \cdot 7854 = 1583 \cdot 3$ square inches.

NOTE.—As the fillets of the shell and fire box act to stay the top end of the boiler, the unstayed area may be taken to be equal to the difference between a 54-inch circle and a 30-inch circle, which will sufficiently allow for the strengthening effect of the uptake ring.

$$\text{Diameter of stays} = \sqrt{\frac{1583 \cdot 3 \times 80}{6 \times \cdot 7854 \times 9000}} = 1 \cdot 7 \text{ inches, say } 1\tfrac{3}{4} \text{ inches diameter.}$$

Fire Box Riveting.—

Plate $\frac{9}{16}$ inch thick. Rivets 1 inch diameter.
Joint strength = 53 per cent.

$$\text{Pitch} = \frac{100 \times 1 \text{ inch}}{100 - 53} = 2 \cdot 12 \text{ inches, say } 2\tfrac{1}{8} \text{ inches.}$$

$$\text{Seam strength} = \frac{2 \cdot 125 - 1}{2 \cdot 125} \times 100 = 53 \text{ per cent. of solid plate (nearly).}$$

$$\text{Rivet strength} = \frac{1^2 \cdot 7854 \times 1 \times 23 \times 100}{2 \cdot 125 \times \cdot 5625 \times 28} = 54 \text{ per cent. of solid plate (nearly).}$$

As the smaller result = joint, then 53 per cent. is the joint strength. This riveting also holds good for the crown plate of the shell, the cross water-tube flanges, and the solid ring joining the uptake and the crown plate.

To Verify Joint Strength.

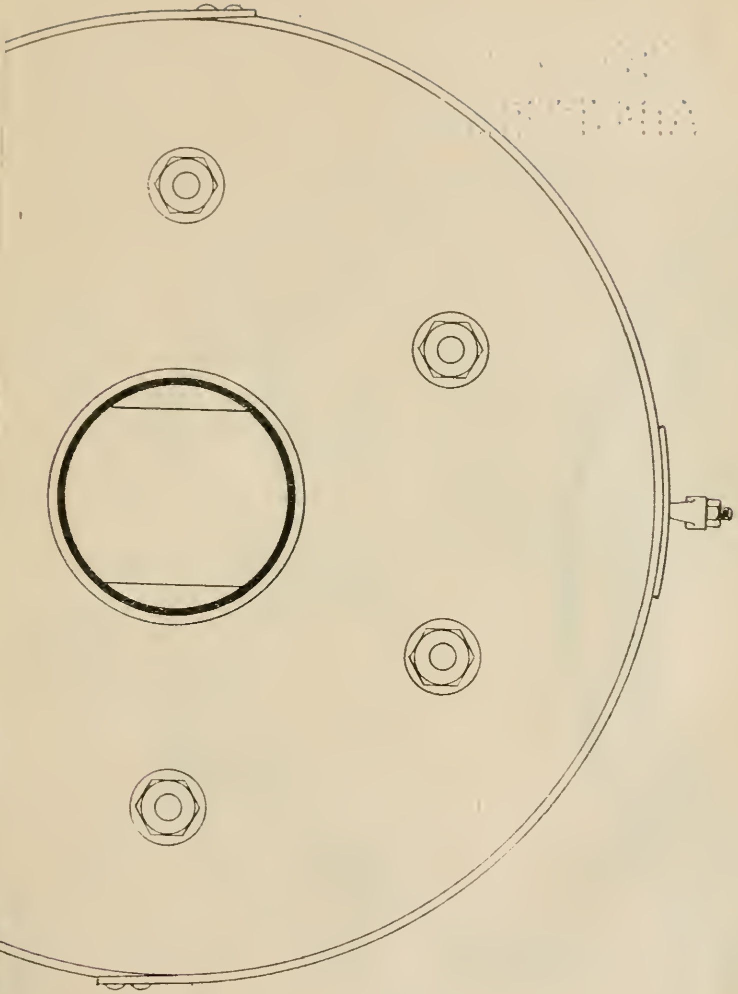
Vertical Shell Seams.—

RULE.—

$$\text{Seam section strength} = \frac{\text{Pitch} - \text{Diameter}}{\text{Pitch}} \times 100.$$

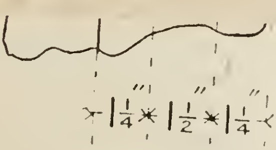
$$\text{Rivet ,, ,,} = \frac{\text{Rivet area} \times \text{No. in a pitch} \times 23 \times 100}{\text{Pitch} \times \text{Thickness} \times 28}.$$

NOTE.—Shear strength of steel rivets = 23 tons per square inch.
Tensile ,, ,, plates = 28 ,, ,, ,,



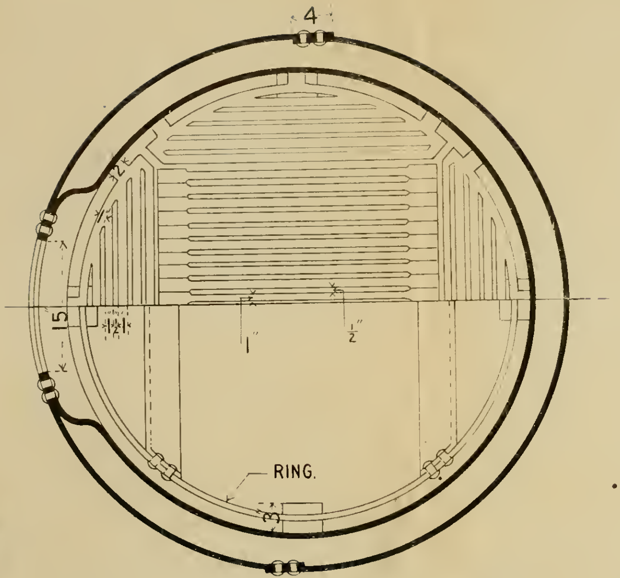
No. 3.—Plan of Boiler.

[To face page 702.

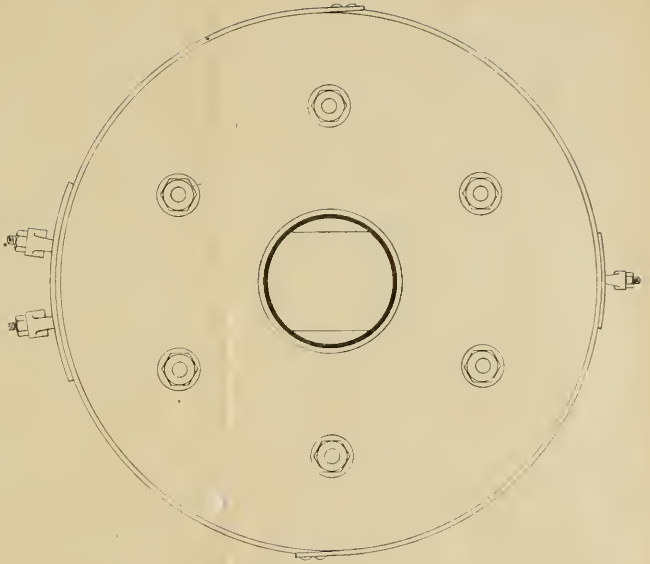


No. 5.—Longitudinal (Vertical) Shell Riveting.

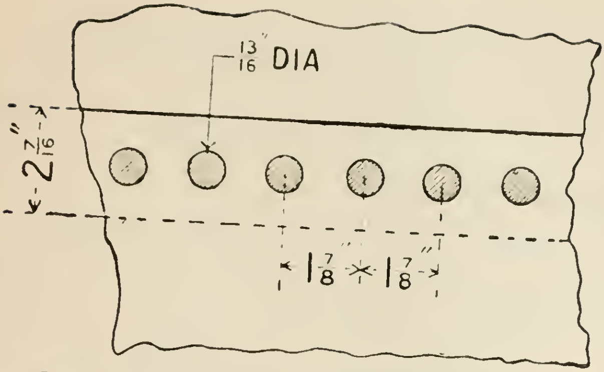
Plates $\frac{3}{8}$ inch thick.



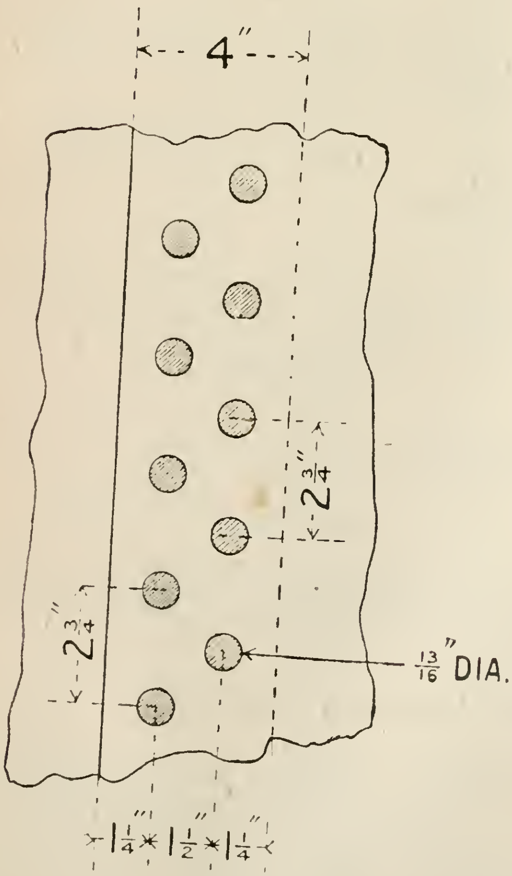
No. 2.—Sectional Plan showing Fire Bars and Bearer Ring.



No. 3.—Plan of Boiler.



No. 4.—Circumferential Shell Riveting.
Plates $\frac{3}{8}$ inch thick.

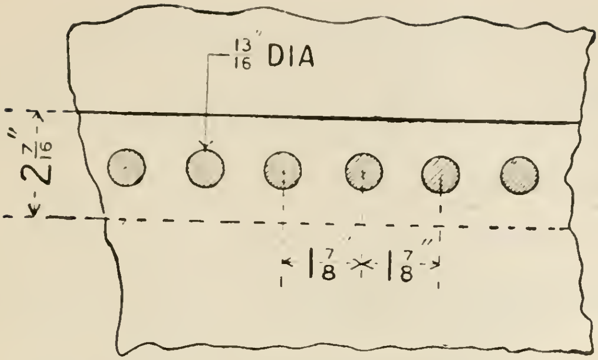


No. 5.—Longitudinal (Vertical) Shell Riveting.
Plates $\frac{3}{8}$ inch thick.

$$\text{Seam section strength} = \frac{[\text{illegible}]}{\text{Pitch}} \times 100.$$

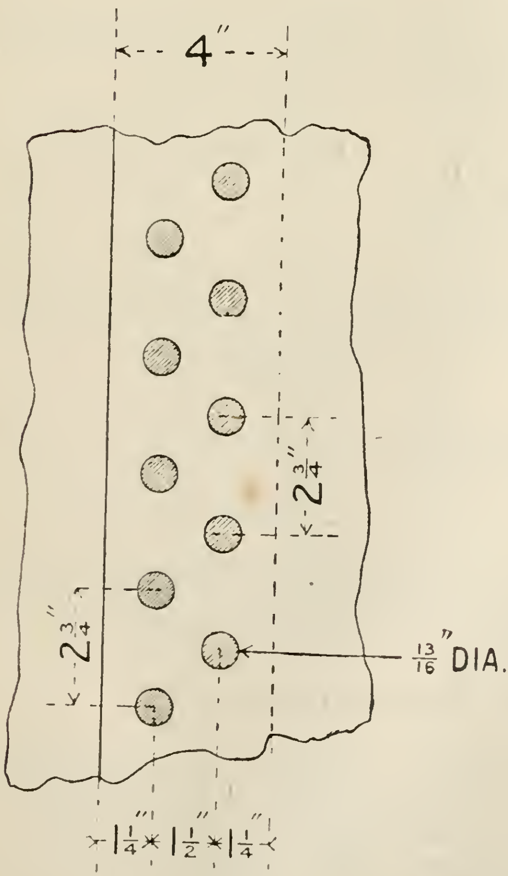
$$\text{Rivet} \quad ,, \quad ,, \quad = \frac{\text{Rivet area} \times \text{No. in a pitch} \times 23 \times 100}{\text{Pitch} \times \text{Thickness} \times 28}.$$

NOTE.—Shear strength of steel rivets $= 23$ tons per square inch.
Tensile ,, ,, plates $= 28$,, ,, ,,



No. 4.—Circumferential Shell Riveting.

Plates $\frac{3}{8}$ inch thick.



No. 5.—Longitudinal (Vertical) Shell Riveting.

Plates $\frac{3}{8}$ inch thick.

Then,

$$\text{Seam strength} = \frac{2{\cdot}75 - {\cdot}8125}{2{\cdot}75} \times 100 = 70{\cdot}4 \text{ per cent. of solid plate.}$$

$$\text{and, Rivet strength} = \frac{{\cdot}8125^2 \times {\cdot}7854 \times 2 \times 23 \times 100}{2{\cdot}75 \times {\cdot}375 \times 28} = 82{\cdot}5 \text{ per cent. of solid plate.}$$

Circumferential Shell Seams.—

$$\text{Seam strength} = \frac{1{\cdot}875 - {\cdot}8125}{1{\cdot}875} \times 100 = 56 \text{ per cent. of solid plate.}$$

$$\text{Rivet strength} = \frac{{\cdot}8125^2 \times {\cdot}7854 \times 1 \times 23 \times 100}{1{\cdot}875 \times {\cdot}375 \times 28} = 60{\cdot}5 \text{ per cent. of solid plate.}$$

Stresses on Shell Seams.

Vertical (longitudinal) Seams.—

RULE.—

$$\text{Diameter in inches} \times \text{Pressure} = T'' \times 2 \times \text{Stress per square inch.}$$

$$\text{Therefore, Stress square inch} = \frac{\text{Diameter}'' \times \text{Pressure}}{T'' \times 2}.$$

$$\text{,, ,, ,,} = \frac{60'' \times 80}{{\cdot}375 \times 2} = 6400 \text{ lbs. square inch.}$$

Circumferential Seams.—

RULE.—

$$\text{Diameter} \times \text{Diameter} \times {\cdot}7854 \times \text{Pressure} = D'' \times 3{\cdot}1416 \times T'' \times \text{Stress per sq. in.}$$

$$\text{or,} \quad \text{Diameter}'' \times \text{Pressure} = T'' \times 4 \times \text{Stress per square inch.}$$

NOTE.— $$\frac{\cancel{D''} \times D'' \times \cancel{{\cdot}7854}}{\cancel{D''} \times 3{\cdot}\cancel{1416}} = \frac{D''}{4}.$$

$$\text{Therefore, Stress square inch} = \frac{\text{Diameter}'' \times \text{Pressure}}{T'' \times 4}.$$

$$\text{,, ,, ,,} = \frac{60'' \times 80}{{\cdot}375 \times 4} = 3200 \text{ lbs. square inch.}$$

As will be seen from the foregoing, the stress longitudinally is exactly equal to twice the stress circumferentially in all cases, hence the necessity for the stronger type of joints required for the longitudinal shell seams.

Manhole Door Compensating Ring.—

RULE.—

Breadth of ring (if same thickness as shell) = Small diameter of door × ·5.

Then, Breadth of ring = 12 inches × ·5 = 6 inches.

Therefore, Outside sizes of compensating plate ring =

$$\left.\begin{array}{l} 6 \text{ inches} + 16 \text{ inches} + 6 \text{ inches} = 28 \text{ inches} \\ 6 \text{ inches} + 12 \text{ inches} + 6 \text{ inches} = 24 \text{ inches} \end{array}\right\} \text{or 28 inches by 24 inches.}$$

Properties of Saturated Steam

Of from 0·5 lb. to 250 lbs. Absolute Pressure per Square Inch.

Absolute Pressure per Square Inch.	Temperatures.	Total Heat of 1 lb. of Steam from Water supplied at 32° Fahr.	Total Latent Heat of Steam.	Density or Weight of 1 Cubic Foot of Steam.	Volume of 1 lb. of Steam.
Lbs.	Deg. Fahr.	Units.	Units.	Lbs.	Cubic Feet.
0·5	80·2	1105·5	1058·4	·001376	726·608
1	102·1	1112·5	1042·9	·003027	330·360
1·5	115·9	1116·7	1033·2	·004433	225·580
2	126·3	1119·9	1025·8	·005811	172·080
2·5	134·6	1122·5	1019·9	·007169	139·488
3	141·6	1124·6	1015·0	·008511	117·500
3·5	147·7	1126·4	1010·6	·009839	101·632
4	153·1	1128·1	1006·8	·01116	89·632
4·5	157·9	1129·6	1003·4	·01246	80·231
5	162·3	1130·9	1000·3	·01370	72·991
5·5	166·4	1132·1	997·4	·01505	66·428
6	170·2	1133·3	994·7	·01634	61·201
6·5	173·6	1134·3	992·3	·01762	56·761
7	176·9	1135·3	990·0	·01889	52·936
7·5	180·0	1136·3	987·8	·02016	49·610
8	182·9	1137·2	985·7	·02142	46·686
8·5	185·7	1138·0	983·8	·02268	44·097
9	188·3	1138·8	981·9	·02394	41·777
9·5	190·8	1139·5	980·1	·02547	39·261
10	193·3	1140·3	978·4	·02642	37·845
10·5	195·6	1141·0	976·7	·02767	36·145
11	197·8	1141·7	975·2	·02890	34·599
11·5	200·1	1142·4	973·6	·03026	33·045
12	202·0	1143·0	972·2	·03137	31·879
12·5	204·0	1143·6	970·8	·03260	30·678
13	205·9	1144·2	969·4	·03382	29·573
13·5	207·8	1144·8	968·1	·03504	28·536
14	209·6	1145·3	966·8	·03627	27·573
14·7	212·0	1146·1	965·2	·03797	26·360
15	213·1	1146·4	964·3	·03870	25·843
16	216·3	1147·4	962·1	·04112	24·320
17	219·6	1148·3	959·8	·04253	23·513
18	222·4	1149·2	957·7	·04594	21·766
19	225·3	1150·1	955·7	·04834	20·687
20	228·0	1150·9	953·8	·05074	19·710
21	230·6	1151·7	951·9	·05311	18·828
22	233·1	1152·5	950·2	·05549	18·022
23	235·5	1153·2	948·5	·05786	17·282
24	237·8	1153·9	946·9	·06023	16·603
25	240·1	1154·6	945·3	·06259	15·977
26	242·3	1155·3	943·7	·06495	15·401

Properties of Saturated Steam—*continued.*

Absolute Pressure per Square Inch.	Temperatures.	Total Heat of 1 lb. of Steam from Water supplied at 32° Fahr.	Total Latent Heat of Steam.	Density or Weight of 1 Cubic Foot of Steam.	Volume of 1 lb. of Steam.
Lbs.	Deg. Fahr.	Units.	Units.	Lbs.	Cubic Feet.
27	244·4	1155·8	942·2	·06728	14·863
28	246·4	1156·4	940·8	·05971	14·345
29	248·4	1157·1	939·4	·07196	13·896
30	250·4	1157·8	937·9	·07430	13·459
31	252·2	1158·4	936·7	·07663	13·050
32	254·1	1158·9	935·3	·07894	12·666
33	255·9	1159·5	934·0	·08128	12·300
34	257·6	1160·0	932·8	·08358	11·964
35	259·3	1160·5	931·6	·08590	11·640
36	260·9	1161·0	930·5	·08821	11·337
37	262·6	1161·5	929·3	·09050	11·050
38	264·2	1162·0	928·2	·09282	10·773
39	265·8	1162·5	927·1	·09510	10·515
40	267·3	1162·9	926·0	·09740	10·267
41	268·7	1163·4	924·9	·09946	10·054
42	270·2	1163·8	923·9	·1020	9·806
43	271·6	1164·2	922·9	·1042	9·592
44	273·0	1164·6	921·9	·1065	9·386
45	274·4	1165·1	920·9	·1088	9·191
46	275·8	1165·5	919·9	·1111	9·003
47	277·1	1165·9	919·0	·1134	8·821
48	278·4	1166·3	918·1	·1156	8·650
49	279·7	1166·7	917·2	·1179	8·482
50	281·0	1167·1	916·3	·1202	8·322
51	282·3	1167·5	915·4	·1224	8·170
52	283·5	1167·9	914·5	·1247	8·021
53	284·7	1168·3	913·6	·1269	7·880
54	285·9	1168·6	912·8	·1292	7·741
55	287·1	1169·0	912·0	·1314	7·610
56	288·2	1169·3	911·2	·1337	7·482
57	289·3	1169·7	910·4	·1357	7·370
58	290·4	1170·0	909·6	·1382	7·238
59	291·6	1170·4	908·8	·1404	7·123
60	292·7	1170·7	908·0	·1426	7·011
61	293·8	1171·1	907·2	·1449	6·902
62	294·8	1171·4	906·4	·1471	6·798
63	295·9	1171·7	905·6	·1493	6·696
64	296·9	1172·0	904·9	·1516	6·596
65	298·0	1172·3	904·2	·1538	6·502
66	299·0	1172·6	903·5	·1560	6·410
67	300·0	1172·9	902·8	·1583	6·318
68	300·9	1173·2	902·1	·1604	6·233
69	301·9	1173·5	901·4	·1627	6·147

Properties of Saturated Steam—*continued.*

Absolute Pressure per Square Inch.	Temperatures.	Total Heat of 1 lb. of Steam from Water supplied at 32° Fahr.	Total Latent Heat of Steam.	Density or Weight of 1 Cubic Foot of Steam.	Volume of 1 lb. of Steam.
Lbs.	Deg. Fahr.	Units.	Units.	Lbs.	Cubic Feet.
70	302·9	1173·8	900·8	·1650	6·059
71	303·9	1174·1	900·3	·1671	5·984
72	304·8	1174·3	899·6	·1693	5·905
73	305·7	1174·6	898·9	·1716	5·829
74	306·6	1174·9	898·2	·1738	5·764
75	307·5	1175·2	897·5	·1760	5·683
76	308·4	1175·4	896·8	·1782	5·610
77	309·3	1175·7	896·1	·1803	5·544
78	310·2	1176·0	895·5	·1826	5·476
79	311·1	1176·3	894·9	·1848	5·411
80	312·0	1176·5	894·3	·1870	5·348
81	312·8	1176·8	893·7	·1892	5·286
82	313·6	1177·1	893·1	·1912	5·230
83	314·5	1177·4	892·5	·1936	5·167
84	315·3	1177·6	892·0	·1957	5·109
85	316·1	1177·9	891·4	·1980	5·052
86	316·9	1178·1	890·8	·2001	4·996
87	317·8	1178·4	890·2	·2023	4·942
88	318·6	1178·6	889·6	·2046	4·889
89	319·4	1178·9	889·0	·2067	4·837
90	320·2	1179·1	888·5	·2088	4·790
91	321·0	1179·3	887·9	·2111	4·737
92	321·7	1179·5	887·3	·2133	4·688
93	322·5	1179·8	886·8	·2154	4·642
94	323·3	1180·0	886·3	·2176	4·595
95	324·1	1180·3	885·8	·2198	4·549
96	324·8	1180·5	885·2	·2220	4·505
97	325·6	1180·8	884·6	·2241	4·462
98	326·3	1181·0	884·1	·2263	4·419
99	327·1	1181·2	883·6	·2286	4·375
100	327·9	1181·4	883·1	·2307	4·335
101	328·5	1181·6	882·6	·2329	4·305
102	329·1	1181·8	882·1	·2350	4·256
103	329·9	1182·0	881·6	·2372	4·216
104	330·6	1182·2	881·1	·2393	4·178
105	331·3	1182·4	880·7	·2415	4·140
106	331·9	1182·6	880·2	·2437	4·104
107	332·6	1182·8	879·7	·2458	4·068
108	333·3	1183·0	879·2	·2480	4·033
109	334·0	1183·3	878·7	·2502	3·998
110	334·6	1183·5	878·3	·2523	3·963
111	335·3	1183·7	877·8	·2545	3·930
112	336·0	1183·9	877·3	·2566	3·897

Properties of Saturated Steam—*continued.*

Absolute Pressure per Square Inch.	Temperatures.	Total Heat of 1 lb. of Steam from Water supplied at 32° Fahr.	Total Latent Heat of Steam.	Density or Weight of 1 Cubic Foot of Steam.	Volume of 1 lb. of Steam.
Lbs.	Deg. Fahr.	Units.	Units.	Lbs.	Cubic Feet.
113	336·7	1184·1	876·8	·2588	3·865
114	337·4	1184·3	876·3	·2610	3·832
115	338·0	1184·5	875·9	·2631	3·801
116	338·6	1184·7	875·5	·2653	3·770
117	339·3	1184·9	875·0	·2674	3·740
118	339·9	1185·1	874·5	·2696	3·710
119	340·5	1185·3	874·1	·2717	3·681
120	341·1	1185·4	873·7	·2738	3·652
121	341·8	1185·6	873·2	·2760	3·623
122	342·4	1185·8	872·8	·2781	3·595
123	343·0	1186·0	872·3	·2803	3·567
124	343·6	1186·2	871·9	·2824	3·541
125	344·2	1186·4	871·5	·2846	3·514
126	344·8	1186·6	871·1	·2867	3·488
127	345·4	1186·8	870·7	·2889	3·462
128	346·0	1186·9	870·2	·2910	3·436
129	346·6	1187·1	869·8	·2931	3·411
130	347·2	1187·3	869·4	·2951	3·388
131	347·8	1187·5	869·0	·2974	3·362
132	348·3	1187·6	868·6	·2996	3·338
133	348·9	1187·8	868·2	·3017	3·315
134	349·5	1188·0	867·8	·3038	3·291
135	350·1	1188·2	867·4	·3060	3·268
136	350·6	1188·3	867·0	·3080	3·246
137	351·2	1188·5	866·6	·3102	3·224
138	351·8	1188·7	866·2	·3123	3·201
139	352·4	1188·9	865·8	·3145	3·180
140	352·9	1189·0	865·4	·3166	3·159
141	353·5	1189·2	865·0	·3187	3·138
142	354·0	1189·4	864·6	·3209	3·117
143	354·5	1189·6	864·2	·3230	3·096
144	355·0	1189·7	863·9	·3251	3·076
145	355·6	1189·9	863·5	·3272	3·056
146	356·1	1190·0	863·1	·3293	3·037
147	356·7	1190·2	862·7	·3315	3·017
148	357·2	1190·3	862·3	·3336	2·998
149	357·8	1190·5	861·9	·3357	2·979
150	358·3	1190·7	861·5	·3378	2·960
151	359·0	1190·9	861·1	·3400	2·941
152	359·5	1191·0	860·7	·3421	2·923
153	360·0	1191·2	860·4	·3442	2·905
154	360·5	1191·4	860·0	·3463	2·887
155	361·1	1191·5	859·6	·3484	2·870

Properties of Saturated Steam—*continued.*

Absolute Pressure per Square Inch.	Temperatures.	Total Heat of 1 lb. of Steam from Water supplied at 32° Fahr.	Total Latent Heat of Steam.	Density or Weight of 1 Cubic Foot of Steam.	Volume of 1 lb. of Steam.
Lbs.	Deg. Fahr.	Units	Units.	Lbs.	Cubic Feet.
156	361·6	1191·7	859·2	·3505	2·853
157	362·1	1191·8	858·9	·3527	2·836
158	362·6	1192·0	858·5	·3548	2·818
159	363·1	1192·1	858·1	·3569	2·802
160	363·6	1192·3	857·8	·3590	2·785
165	366·0	1192·9	856·2	·3696	2·706
170	368·2	1193·7	854·5	·3801	2·631
175	370·8	1194·4	852·9	·3905	2·559
180	372·9	1195·1	851·3	·4011	2·493
185	375·3	1195·8	849·6	·4115	2·430
190	377·5	1196·5	848·0	·4220	2·370
195	379·7	1197·2	846·5	·4324	2·313
200	381·7	1197·8	845·0	·4419	2·263
210	386·0	1199·1	841·9	·463	2·157
220	389·9	1200·3	839·2	·484	2·065
230	394·0	1201·0	836·0	·505	1·98
240	397·0	1202·0	833·0	·525	1·90
250	401·0	1203·0	831·0	·546	1·83

Table of Circumferences and Areas of Circles.

Diam.	Circum.	Area.
Inches.	Inches.	Sq. In.
·03125	·0981	·00077
·0625	·1963	·00307
·125	·3926	·01227
·25	·7853	·04909
·375	1·1781	·11045
·5	1.5708	·19635
·625	1·9635	·30680
·75	2·3561	·4417
·875	2·748	·601
1	3·1416	·7854
1·125	3·5342	0·9940
1·25	3·9269	1·2272
1·375	4·3196	1·4849
1·5	4·7129	1·7671
1·625	5·1050	2·0739
1·75	5·4977	2·4053
1·875	5·8904	2·7612
2	6·2831	3·1416
2·125	6·6758	3·5466
2·25	7·0865	3·9761
2·375	7·4612	4·4301
2·5	7·8539	4·9087
2·625	8·2466	5·4119
2·75	8·6393	5·9396
2·875	9·0320	6·4918
3	9·4247	7·0686
3·125	9·8174	7·6699
3·25	10·2102	8·2958
3·375	10·6029	8·9462
3·5	10·9956	9·6211
3·625	11·3883	10·321
3·75	11·7810	11·045
3·875	12·1737	11·793
4	12·5664	12·566
4·125	12·9591	13·364
4·25	13·3518	14·186
4·375	13·7445	15·033
4·5	14·1372	15·904
4·625	14·5299	16·800
4·75	14·9226	17·721
4·875	15·3153	18·665
5	15·7080	19·635
5·125	16·1007	20·629
5·25	16·4934	21·648
5·375	16·8861	22·691
5·5	17·2788	23·758
5·625	17·6715	24·850
5·75	18·0642	25·967
5·875	18·4569	27·109

Diam.	Circum.	Area.
Inches.	Inches.	Sq. In.
6	18·8496	28·274
6·125	19·2423	29·465
6·25	19·6350	30·680
6·375	20·0277	31·919
6·5	20·4204	33·183
6·625	20·8131	34·472
6·75	21·2058	35·785
6·875	21·5984	37·122
7	21·9911	38·485
7·125	22·3838	39·871
7·25	22·7765	41·282
7·375	23·1692	42·718
7·5	23·5619	44·179
7·625	23·9546	45·664
7·75	24·3473	47·173
7·875	24·7400	48·707
8	25·1327	50·265
8·125	25·5224	51·849
8·25	25·9181	53·456
8·375	26·3108	55·088
8·5	26·7035	56·745
8·625	27·0962	58·426
8·75	27·4889	60·132
8·875	27·8816	61·862
9	28·2743	63·617
9·125	28·6670	65·397
9·25	29·0597	67·201
9·375	29·4524	69·029
9·5	29·8451	70·882
9·625	30·2378	72·760
9·75	30·6305	74·662
9·875	31·0232	76·589
10	31·4159	78·540
10·125	31·8086	80·516
10·25	32·2013	82·516
10·375	32·5940	84·541
10·5	32·9867	86·590
10·625	33·3794	88·664
10·75	33·7721	90·763
10·875	34·1648	92·886
11	34·5575	95·033
11·125	34·9502	97·205
11·25	35·3429	99·402
11·375	35·7356	101·62
11·5	36·1283	103·87
11·625	36·5210	106·14
11·75	35·9137	108·43
11·875	37·3064	110·75

Diam.	Circum.	Area.
Inches.	Inches.	Sq. In.
12	37·6991	113·10
12·125	38·0918	115·47
12·25	38·4845	117·86
12·375	38·8772	120·28
12·5	39·2699	122·72
12·625	39·6626	125·19
12·75	40·0553	127·68
12·875	40·4480	130·19
13	40·8407	132·73
13·125	41·2334	135·30
13·25	41·6261	137·89
13·375	42·0188	140·50
13·5	42·4115	143·14
13·625	42·8042	145·80
13·75	43·1969	148·49
13·875	43·5896	151·20
14	43·9823	153·94
14·125	44·3750	156·70
14·25	44·7677	159·48
14·375	45·1604	162·30
14·5	45·5531	165·13
14·625	45·9458	167·99
14·75	46·3385	170·87
14·875	46·7312	173·78
15	47·1239	176·71
15·125	47·5166	179·67
15·25	47·9093	182·65
15·375	48·3020	185·66
15·5	48·6947	188·69
15·625	49·0874	191·75
15·75	49·4801	194·83
15·875	49·8728	197·93
16	50·2655	201·06
16·125	50·6582	204·22
16·25	51·0509	207·39
16·375	51·4436	210·60
16·5	51·8363	213·82
16·625	52·2290	217·08
16·75	52·6217	220·35
16·875	53·0144	223·65
17	53·4071	226·98
17·125	53·7998	230·33
17·25	54·1925	233·71
17·375	54·5852	237·10
17·5	54·9779	240·53
17·625	55·3706	243·98
17·75	55·7633	247·45
17·875	56·1560	250·95

Table of Circumferences and Areas of Circles—*continued.*

Diam.	Circum.	Area.	Diam.	Circum.	Area.	Diam.	Circum.	Area.
Inches.	Inches.	Sq. In.	Inches.	Inches.	Sq. In.	Inches.	Inches.	Sq. In.
18	56·5487	254·47	24	75·3982	452·39	30	94·2478	706·86
18·125	56·9414	258·02	24·125	75·7909	457·11	30·125	94·6405	712·76
18·25	57·3341	261·59	24·25	76·1836	461·86	30·25	95·0332	718·69
18·375	57·7268	265·18	24·375	76·5763	466·64	30·375	95·4259	724·64
18·5	58·1195	268·80	24·5	76·9690	471·44	30·5	95·8186	730·62
18·625	58·5122	272·45	24·625	77·3617	476·26	30·625	96·2113	736·62
18·75	58·9049	276·12	24·75	77·7544	481·11	30·75	96·6040	742·64
18·875	59·2976	279·81	24·875	78·1471	485·98	30·875	96·9967	748·69
19	59·6903	283·53	25	78·5398	490·87	31	97·3894	754·77
19·125	60·0830	287·27	25·125	78·9325	495·79	31·125	97·7821	760·87
19·25	60·4757	291·04	25·25	79·3252	500·74	31·25	98·1748	766·99
19·375	60·8684	294·83	25·375	79·7179	505·71	31·375	98·5675	773·14
19·5	61·2611	298·65	25·5	80·1106	510·71	31·5	98·9602	779·31
19·625	61·6538	302·49	25·625	80·5033	515·72	31·625	99·3529	785·51
19·75	62·0465	306·35	25·75	80·8960	520·77	31·75	99·7456	791·73
19·875	62·4392	310·24	25·875	81·2887	525·84	31·875	100·138	797·98
20	62·8319	314·16	26	81·6814	530·93	32	100·531	804·25
20·125	63·2246	318·10	26·125	82·0741	536·05	32·125	100·924	810·54
20·25	63·6173	322·06	26·25	82·4668	541·19	32·25	101·316	816·86
20·375	64·0100	326·05	26·375	82·8595	546·35	32·375	101·709	823·21
20·5	64·4026	330·06	26·5	83·2522	551·55	32·5	102·102	829·58
20·625	64·7953	334·10	26·625	83·6449	556·76	32·625	102·494	835·97
20·75	65·1880	338·16	26·75	84·0376	562·00	32·75	102·887	842·39
20·875	65·5807	342·25	26·875	84·4303	567·27	32·875	103·280	848·83
21	65·9734	346·36	27	84·8230	572·56	33	103·673	855·30
21·125	66·3661	350·50	27·125	85·2157	577·87	33·125	104·065	861·79
21·25	66·7588	354·66	27·25	85·6084	583·21	33·25	104·458	868·31
21·375	67·1515	358·84	27·375	86·0011	588·57	33·375	104·851	874·85
21·5	67·5442	363·06	27·5	86·3938	593·96	33·5	105·243	881·41
21·625	67·9369	367·28	27·625	86·7865	599·37	33·625	105·636	888·00
21·75	68·3296	371·54	27·75	87·1792	604·81	33·75	106·029	894·62
21·875	68·7223	375·83	27·875	87·5719	610·27	33·875	106·421	901·26
22	69·1150	380·13	28	87·9646	615·75	34	106·814	907·92
22·125	69·5077	384·46	28·125	88·3573	621·26	34·125	107·207	914·61
22·25	69·9004	388·82	28·25	88·7500	626·80	34·25	107·600	921·32
22·375	70·2931	393·20	28·375	89·1427	632·36	34·375	107·992	928·06
22·5	70·6858	397·61	28·5	89·5354	637·94	34·5	108·385	934·82
22·625	71·0785	402·04	28·625	89·9281	643·55	34·625	108·788	941·61
22·75	71·4712	406·49	28·75	90·3208	649·18	34·75	109·170	948·42
22·875	71·8639	410·97	28·875	90·7135	654·84	34·875	109·563	955·25
23	72·2566	415·48	29	91·1062	660·52	35	109·956	962·11
23·125	72·6493	420·00	29·125	91·4989	666·23	35·125	110·348	969·00
23·25	73·0420	424·56	29·25	91·8916	671·96	35·25	110·741	975·91
23·375	73·4347	429·13	29·375	92·2843	677·71	35·375	111·134	982·84
23·5	73·8274	433·74	29·5	92·6770	683·49	35·5	111·527	989·80
23·625	74·2201	438·36	29·625	93·0697	689·30	35·625	111·919	996·78
23·75	74·6128	443·01	29·75	93·4624	695·13	35·75	112·312	1003·8
23·875	75·0055	447·69	29·875	93·8551	700·98	35·875	112·705	1010·8

Table of Circumferences and Areas of Circles—*continued.*

Diam.	Circum.	Area.	Diam.	Circum.	Area.	Diam.	Circum.	Area.
Inches.	Inches.	Sq. In.	Inches.	Inches.	Sq. In.	Inches.	Inches.	Sq. In.
36	113·097	1017·9	**42**	131·947	1385·4	**48**	150·796	1809·6
36·125	113·490	1025·0	42·125	132·340	1393·7	48·125	151·189	1819·0
36·25	113·883	1032·1	42·25	132·732	1402·0	48·25	151·582	1828·5
36·375	114·275	1039·2	42·375	133·125	1410·3	48·375	151·975	1837·9
36·5	114·668	1046·3	42·5	133·518	1418·6	48·5	152·367	1847·5
36·625	115·061	1053·5	42·625	133·910	1427·0	48·625	152·760	1857·0
36·75	115·454	1060·7	42·75	134·303	1435·4	48·75	153·153	1866·5
36·875	115·846	1068·0	42·875	134·696	1443·8	48·875	153·544	1876·1
37	116·239	1075·2	**43**	135·088	1452·2	**49**	153·938	1885·7
37·125	116·632	1082·5	43·125	135·481	1460·7	49·125	154·331	1895·4
37·25	117·024	1089·8	43·25	135·874	1469·1	49·25	154·723	1905·0
37·375	117·417	1097·1	43·375	136·267	1477·6	49·375	155·116	1914·7
37·5	117·810	1104·5	43·5	136·659	1486·2	49·5	155·509	1924·2
37.625	118·202	1111·8	43·625	137·052	1494·7	49·625	155·904	1934·2
37·75	118·596	1119·2	43·75	137·445	1503·3	49·75	156·294	1943·9
37·875	118·988	1126·7	43·875	137·837	1511·9	49·875	156·687	1953·7
38	119·381	1134·1	**44**	138·230	1520·5	**50**	157·080	1963·5
38·125	119·773	1141·6	44·125	138·623	1529·2	50·125	157·472	1973·3
38·25	120·166	1149·1	44·25	139·015	1537·9	50·25	157·865	1983·2
38·375	120·559	1156·6	44·375	139·408	1546·6	50·375	158·258	1993·1
38·5	120·951	1164·2	44·5	139·801	1555·3	50·5	158·650	2003·0
38·625	121·344	1171·7	44·625	140·194	1564·0	50·625	159·043	2012·9
38·75	121·737	1179·3	44·75	140·586	1572·8	50·75	159·436	2022·8
38·875	122·129	1186·9	44·875	140·979	1581·6	50·875	159·829	2032·8
39	122·522	1194·6	**45**	141·372	1590·4	**51**	160·221	2042·8
39·125	122·915	1202·3	45·125	141·764	1599·3	51·125	160·614	2052·8
39·25	123·308	1210·0	45·25	142·157	1608·2	51·25	161·007	2062·9
39·375	123·700	1217·7	45·375	142·550	1617·0	51·375	161·399	2073·0
39·5	124·093	1225·4	45·5	142·942	1626·0	51·5	161·792	2083·1
39·625	124·486	1233·2	45·625	143·335	1634·9	51·625	162·185	2093·2
39·75	124·878	1241·0	45·75	143·728	1643·9	51·75	162·577	2103·3
39·875	125·271	1248·8	45·875	144·121	1652·9	51·875	162·970	2113·5
40	125·664	1256·6	**46**	144·513	1661·9	**52**	163·363	2123·7
40·125	126·056	1264·5	46·125	144·906	1670·9	52·125	163·756	2133·9
40·25	126·449	1272·4	46·25	145·299	1680·0	52·25	164·148	2144·2
40·375	126·842	1280·3	46·375	145·691	1689·1	52·375	164·541	2154·5
40·5	127·235	1288·2	46·5	146·084	1698·2	52·5	164·934	2164·8
40·625	127·627	1296·2	46·625	146·477	1707·4	52·625	165·326	2175·1
40·75	128·020	1304·2	46·75	146·869	1716·5	52·75	165·719	2185·4
40·875	128·413	1312·2	46·875	147·262	1725·7	52·875	166·112	2195·8
41	128·805	1320·3	**47**	147·655	1734·9	**53**	166·504	2206·2
41·125	129·198	1328·3	47·125	148·048	1744·2	53·125	166·897	2216·6
41·25	129·591	1336·4	47·25	148·440	1753·5	53·25	167·290	2227·0
41·375	129·993	1344·5	47·375	148·833	1762·7	53·375	167·683	2237·5
41·5	130·376	1352·7	47·5	149·226	1772·1	53·5	168·075	2248·0
41·625	130·769	1360·8	47·625	149·618	1781·4	53·625	168·468	2258·5
41·75	131·161	1369·0	47·75	150·011	1790·8	53·75	168·861	2269·1
41·875	131·554	1377·2	47·875	150·404	1800·1	53·875	169·253	2279·6

Table of Circumferences and Areas of Circles—*continued.*

Diam.	Circum.	Area.	Diam.	Circum.	Area.	Diam.	Circum.	Area.
Inches.	Inches.	Sq. In.	Inches.	Inches.	Sq. In.	Inches.	Inches.	Sq. In.
54	169·646	2290·2	60	188·496	2827·4	66	207·345	3421·2
54·125	170·039	2300·8	60·125	188·888	2839·2	66·125	207·738	3434·3
54·25	170·431	2311·5	60·25	189·281	2851·0	66·25	208·131	3447·2
54·375	170·824	2322·1	60·375	189·674	2862·9	66·375	208·523	3460·2
54·5	171·217	2332·8	60·5	190·066	2874·8	66·5	208·916	3473·2
54·625	171·609	2343·5	60·625	190·459	2886·6	66·625	209·309	3486·3
54·75	172·002	2354·3	60·75	190·852	2898·6	66·75	209·701	3499·4
54·875	172·395	2365·0	60·875	191·244	2910·5	66·875	210·094	3512·5
55	172·788	2375·8	61	191·637	2922·5	67	210·487	3525·7
55·125	173·180	2386·6	61·125	192·030	2934·5	67·125	210·879	3538·8
55·25	173·573	2397·5	61·25	192·423	2946·5	67·25	211·272	3552·0
55·375	173·966	2408·3	61·375	192·815	2958·5	67·375	211·665	3565·2
55·5	174·358	2419·2	61·5	193·208	2970·6	67·5	212·058	3578·5
55·625	174·751	2430·1	61·625	193·601	2982·7	67·625	212·450	3591·7
55·75	175·144	2441·1	61·75	193·993	2994·8	67·75	212·843	3605·0
55·875	175·536	2452·0	61·875	194·386	3006·9	67·875	213·236	3618·3
56	175·929	2463·0	62	194·779	3019·1	68	213·628	3631·7
56·125	176·322	2474·0	62·125	195·171	3031·3	68·125	214·021	3645·0
56·25	176·715	2485·0	62·25	195·564	3043·5	68·25	214·414	3658·4
56·375	177·107	2496·1	62·375	195·957	3055·7	68·375	214·806	3671·8
56·5	177·500	2507·2	62·5	196·350	3068·0	68·5	215·199	3685·3
56·625	177·893	2518·3	62·625	196·742	3080·3	68·625	215·592	3698·7
56·75	178·285	2529·4	62·75	197·135	3092·6	68·75	215·984	3712·2
56·875	178·678	2540·6	62·875	197·528	3104·9	68·875	216·337	3725·7
57	179·071	2551·8	63	197·920	3117·2	69	216·770	3739·3
57·125	179·463	2563·0	63·125	198·313	3129·6	69·125	217·163	3752·8
57·25	179·856	2574·2	63·25	198·706	3142·0	69·25	217·555	3766·4
57·375	180·249	2585·4	63·375	199·098	3154·5	69·375	217·948	3780·0
57·5	180·642	2596·7	63·5	199·491	3166·9	69·5	218·341	3793·7
57·625	181·034	2608·0	63·625	199·884	3179·4	69·625	218·733	3807·3
57·75	181·427	2619·4	63·75	200·277	3191·9	69·75	219·126	3821·0
57·875	181·820	2630·7	63·875	200·669	3204·4	69·875	219·519	3834·7
58	182·212	2642·1	64	201·062	3217·0	70	219·911	3848·5
58·125	182·605	2653·5	64·125	201·455	3229·6	70·125	220·304	3862·2
58·25	182·998	2664·9	64·25	201·847	3242·2	70·25	220·697	3876·0
58·375	183·390	2676·4	64·375	202·240	3254·8	70·375	221·090	3889·8
58·5	183·783	2687·8	64·5	202·633	3267·5	70·5	221·482	3903·6
58·625	184·176	2699·3	64·625	203·025	3280·1	70·625	221·875	3917·5
58·75	184·569	2710·9	64·75	203·418	3292·8	70·75	222·268	3931·4
58·875	184·961	2722·4	64·875	203·811	3305·6	70·875	222·660	3945·3
59	185·354	2734·0	65	204·204	3318·3	71	223·053	3959·2
59·125	185·747	2745·6	65·125	204·596	3331·1	71·125	223·446	3973·1
59·25	186·139	2757·2	65·25	204·989	3343·9	71·25	223·838	3987·1
59·375	186·532	2768·8	65·375	205·382	3356·7	71·375	224·231	4001·1
59·5	186·925	2780·5	65·5	205·774	3369·6	71·5	224·624	4015·2
59·625	187·317	2792·2	65·625	206·167	3382·4	71·625	225·017	4029·2
59·75	187·710	2803·9	65·75	206·560	3395·3	71·75	225·409	4043·3
59·875	188·103	2815·7	65·875	206·952	3408·2	71·875	225·802	4057·4

Table of Circumferences and Areas of Circles—*continued.*

Diam.	Circum.	Area.	Diam.	Circum.	Area.	Diam.	Circum.	Area.
Inches.	Inches.	Sq. In.	Inches.	Inches.	Sq. In.	Inches.	Inches.	Sq. In.
72	226·195	4071·5	78	245·044	4778·4	84	263·894	5541·8
72·125	226·587	4085·7	78·125	245·437	4793·7	84·125	264·286	5558·3
72·25	226·980	4099·8	78·25	245·830	4809·0	84·25	264·679	5574·8
72·375	227·373	4114·0	78·375	246·222	4824·4	84·375	265·072	5591·4
72·5	227·765	4128·2	78·5	246·615	4839·8	84·5	265·465	5607·9
72·625	228·158	4142·5	78·625	247·008	4855·2	84·625	265·857	5624·5
72·75	228·551	4156·8	78·75	247·400	4870·7	84·75	266·250	5641·2
72·875	228·944	4171·1	78·875	247·793	4886·2	84·875	266·643	5657·8
73	229·336	4185·4	79	248·186	4901·7	85	267·035	5674·5
73·125	229·729	4199·7	79·125	248·579	4917·2	85·125	267·428	5691·2
73·25	230·122	4214·1	79·25	248·971	4932·7	85·25	267·821	5707·9
73·375	230·514	4228·5	79·375	249·364	4948·3	85·375	268·213	5724·7
73·5	230·907	4242·9	79·5	249·757	4963·9	85·5	268·606	5741·5
73·625	231·300	4257·4	79·625	250·149	4979·5	85·625	268·999	5758·3
73·75	231·692	4271·8	79·75	250·542	4995·2	85·75	269·392	5775·1
73·875	232·085	4286·3	79·875	250·935	5010·9	85·875	269·784	5791·9
74	232·478	4300·8	80	251·327	5026·5	86	270·177	5808·8
74·125	232·871	4315·4	80·125	251·720	5042·3	86·125	270·570	5825·7
74·25	233·263	4329·9	80·25	252·113	5058·0	86·25	270·962	5842·6
74·375	233·656	4344·5	80·375	252·506	5073·8	86·375	271·355	5859·6
74·5	234·049	4359·2	80·5	252·898	5089·6	86·5	271·748	5876·5
74·625	234·441	4373·8	80·625	253·291	5105·4	86·625	272·140	5893·5
74·75	234·834	4388·5	80·75	253·684	5121·2	86·75	272·533	5910·6
74·875	235·227	4403·1	80·875	254·076	5137·1	86·875	272·926	5927·6
75	235·619	4417·9	81	254·469	5153·0	87	273·319	5944·7
75·125	236·012	4432·6	81·125	254·862	5168·9	87·125	273·711	5961·8
75·25	236·405	4447·4	81·25	255·254	5184·9	87·25	274·104	5978·9
75·375	236·798	4462·2	81·375	255·647	5200·8	87·375	274·497	5996·0
75·5	237·190	4477·0	81·5	256·040	5216·8	87·5	274·889	6013·2
75·625	237·583	4491·8	81·625	256·433	5232·8	87·625	275·282	6030·4
75·75	237·976	4506·7	81·75	256·825	5248·9	87·75	275·695	6047·6
75·875	238·368	4521·5	81·875	257·218	5264·9	87·875	276·067	6064·9
76	238·761	4536·5	82	257·611	5281·0	88	276·460	6082·1
76·125	239·154	4551·4	82·125	258·003	5297·1	88·125	276·853	6099·4
76·25	239·546	4566·4	82·25	258·396	5313·3	88·25	277·246	6256·7
76·375	239·939	4581·3	82·375	258·789	5329·4	88·375	277·638	6134·1
76·5	240·332	4596·3	82·5	259·181	5345·6	88·5	278·031	6151·4
76·625	240·725	4611·4	82·625	259·574	5361·8	88·625	278·424	6168·8
76·75	241·117	4626·4	82·75	259·967	5378·1	88·75	278·816	6186·2
76·875	241·510	4641·5	82·875	260·359	5394·3	88·875	279·209	6203·7
77	241·903	4656·6	83	260·752	5410·6	89	279·602	6221·1
77·125	242·295	4671·8	83·125	261·145	5426·9	89·125	279·994	6238·6
77·25	242·688	4686·9	83·25	261·538	5443·3	89·25	280·387	6256·1
77·375	243·081	4702·1	83·375	261·930	5459·6	89·375	280·780	6273·7
77·5	243·473	4717·3	83·5	262·323	5476·0	89·5	281·173	6291·2
77·625	243·866	4732·5	83·625	262·716	5492·4	89·625	281·565	6308·8
77·75	244·259	4747·8	83·75	263·108	5508·8	89·75	281·958	6326·4
77·875	244·652	4763·1	83·875	263·501	5525·3	89·875	282·351	6344·1

Table of Circumferences and Areas of Circles—*continued.*

Diam.	Circum.	Area.	Diam.	Circum.	Area.	Diam.	Circum.	Area.
Inches.	Inches.	Sq. In.	Inches.	Inches.	Sq. In.	Inches.	Inches.	Sq. In.
90	282·743	6361·7	94	295·310	6939·8	98	307·876	7543·0
90·125	283·136	6379·4	94·125	295·702	6958·2	98·125	308·269	7562·2
90·25	283·529	6397·1	94·25	296·095	6976·7	98·25	308·661	7581·5
90·375	283·921	6414·9	94·375	296·488	6995·3	98·375	309·054	7600·8
90·5	284·314	6432·6	94·5	296·881	7013·8	98·5	309·447	7620·1
90·625	284·707	6450·4	94·625	297·273	7032·4	98·625	309·840	7639·5
90·75	285·100	6468·2	94·75	297·666	7051·0	98·75	310·232	7658·9
90·875	285·492	6486·0	94·875	298·059	7069·6	98·875	310·625	7678·3
91	285·885	6503·9	95	298·451	7088·2	99	311·018	7697·7
91·125	286·278	6521·8	95·125	298·844	7106·9	99·125	311·410	7717·1
91·25	286·670	6539·7	95·25	299·237	7125·6	99·25	311·803	7736·6
91·375	287·063	6557·6	95·375	299·629	7144·3	99·375	312·196	7756·1
91·5	287·456	6575·5	95.5	300·022	7163·0	99·5	312·588	7775·6
91·625	287·848	6593·5	95.625	300·415	7181·8	99·625	312·981	7795·2
91·75	288·241	6611·5	95·75	300·807	7200·6	99·75	313·374	7814·8
91·875	288·634	6629·6	95·875	301·200	7219·4	99·875	313·767	7834·4
92	289·027	6647·6	96	301·593	7238·2	100	314·159	7854·0
92·125	289·419	6665·7	96·125	301·986	7257·1			
92·25	289·812	6683·8	96·25	302·378	7276·0			
92·375	290·205	6701·9	96·375	302·771	7294·9			
92·5	290·597	6720·1	96·5	303·164	7313·8			
92·625	290·990	6738·2	96·625	303·556	7332·8			
92·75	291·383	6756·4	96·75	303·949	7351·8			
92·875	291·775	6774·7	96·875	304·342	7370·8			
93	292·168	6792·9	97	304·734	7389·8			
93·125	292·561	6811·2	97·125	305·127	7408·9			
93·25	292·954	6829·5	97·25	305·520	7428·0			
93·375	293·346	6847·8	97·375	305·913	7447·1			
93·5	293·739	6866·1	97·5	306·305	7466·2			
93·625	294·132	6884·5	97·625	306·698	7485·3			
93·75	294·524	6902·9	97·75	307·091	7504·5			
93·875	294·917	6921·3	97·875	307·483	7523·7			

Hyperbolic Logarithms.

No. of Expansions R.	Hyp. log.	No. of Expansions R.	Hyp. log.	No. of Expansions R.	Hyp. log.	No. of Expansions R.	Hyp. log.
1·1	0·0953	3·6	1·2809	6·1	1·8083	8·6	2·1518
1·2	0·1823	3·7	1·3083	6·2	1·8245	8·7	2·1633
1·3	0·2624	3·8	1·3350	6·3	1·8405	8·8	2·1748
1·4	0·3365	3·9	1·3610	6·4	1·8563	8·9	2·1861
1·5	0·4055	4·0	1·3863	6·5	1·8718	9·0	2·1972
1·6	0·4700	4·1	1·4110	6·6	1·8871	9·1	2·2083
1·7	0·5306	4·2	1·4351	6·7	1·9021	9·2	2·2192
1·8	0·5878	4·3	1·4586	6·8	1·9169	9·3	2·2300
1·9	0·6419	4·4	1·4816	6·9	1·9315	9·4	2·2407
2·0	0·6931	4·5	1·5041	7·0	1·9459	9·5	2·2513
2·1	0·7419	4·6	1·5261	7·1	1·9601	9·6	2·2618
2·2	0·7885	4·7	1·5476	7·2	1·9741	9·7	2·2721
2·3	0·8329	4·8	1·5686	7·3	1·9879	9·8	2·2824
2·4	0·8755	4·9	1·5896	7·4	2·0015	9·9	2·2925
2·5	0·9163	5·0	1·6094	7·5	2·0149	10·0	2·3026
2·6	0·9555	5·1	1·6292	7·6	2·0281	10·5	2·3513
2·7	0·9933	5·2	1·6487	7·7	2·0412	11·0	2·3979
2·8	1·0296	5·3	1·6677	7·8	2·0541	11·5	2·4430
2·9	1·0647	5·4	1·6864	7·9	2·0669	12·0	2·4849
3·0	1·0986	5·5	1·7047	8·0	2·0794	12·5	2·5262
3·1	1·1314	5·6	1·7228	8·1	2·0919	13·0	2·5649
3·2	1·1632	5·7	1·7405	8·2	2·1041	13·5	2·6027
3·3	1·1939	5·8	1·7579	8·3	2·1163	14·0	2·6391
3·4	1·2238	5·9	1·7750	8·4	2·1282	15·0	2·7081
3·5	1·2528	6·0	1·7918	8·5	2·1401	16·0	2·7726

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