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*The Running and Maintenance of Marine Steam Turbines

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It will be understood that the title of this paper does not call for a detailed investigation into the theoretical aspect of the design of turbine machinery, but it is thought that brief references to this side of the subject are necessary in order that the paper may be made comprehensive.

In modern marine turbine practice, high-pressure superheated steam is in general use. Many marine installations to-day are using steam at 400 to 450lb. pressure and 750° F. total temperature, and the prospect is that pressures and temperatures will be still further increased with a view to greater fuel economy.

Turbines may be divided into four distinct types and a brief description of each will now be given. The types are:—

(1) Impulse turbines.
(2) Axial flow reaction turbines (Parsons).
(3) Axial flow impulse reaction.
(4) Radial flow reaction (Ljungstrom).

* This paper is included in the second edition of "The Running and Maintenance of Marine Machinery" Handbook, published by The Institute at 7s. 6d. plus 7d. postage. Other subjects included in the Handbook are steam reciprocating engines, boilers, Diesel engines, electrical machinery, refrigerating machinery, pumping arrangements, and steering gears.

(1) Impulse Turbines.

In this type the pressure drop takes place in the nozzles, the pressure remaining constant in the moving blades. The greater the pressure difference between the steam entering the nozzles and leaving them the higher the velocity of the issuing steam. Impulse turbines are of the following types:—

(*a*) *Simple Impulse.*

In the simple impulse turbine there is only one set of nozzles and one row of moving blades fixed to a wheel mounted on a shaft, the total pressure drop from the initial steam pressure to the exhaust pressure taking place in the nozzles. This results in a very high issuing steam speed from the nozzles and consequently very high revolutions of the rotor, *i.e.* a high blade speed. The de Laval turbine is of this type.

(*b*) *Pressure-compounded Impulse Turbines.*

In turbines of this type, the total pressure drop between the initial pressure and the exhaust pressure is divided up into several stages each stage consisting of a diaphragm containing the nozzles followed by a wheel with a single row of blades fixed to its periphery. The blades and nozzles increase in size at each stage to accom-

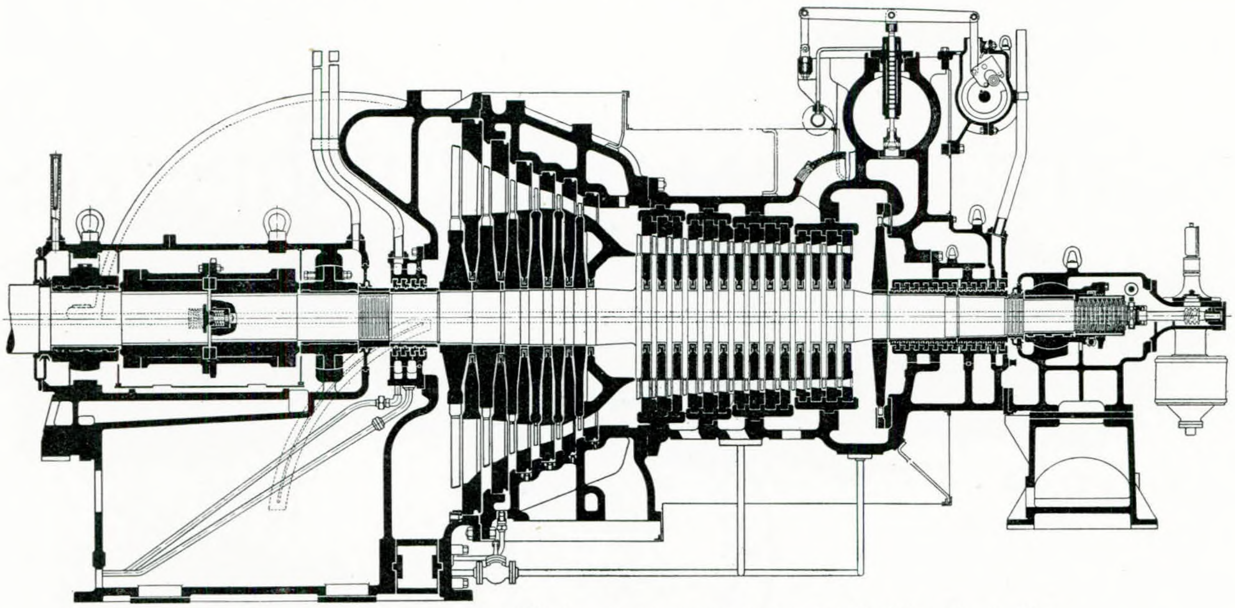


FIG. 1.—*Sectional arrangement of high-pressure, pressure-compounded impulse turbine.*

modate the greater volume of the steam as the pressure falls.

Turbines of the pressure-compounded type are used for driving the turbo alternators in land stations and it is this form of turbine which is installed in vessels propelled by the turbo-electric drive.

Fig. 1 shows a pressure-compounded impulse turbine as installed for the turbo-electric drive.

(*c*) *Pressure-velocity Compounded Impulse Turbines.*

In this type of turbine, originally designed and produced by Mr. Curtiss, an American engineer, the pressure drop between the initial and exhaust pressures is divided into a smaller number of stages than in the pressure-compounded machine. Each stage consists of a diaphragm containing the nozzles and a wheel on which are fixed two or more rows of moving blades. Between each row of moving blades there is a row of fixed blades in the casing to guide the steam to the next row of moving blades.

In modern designs with high pressure and superheat this type of turbine has been superseded by the pressure-compounded machine described under (*b*) above. It is, however, normal practice in pressure-compounded machines using high-pressure superheated steam to make the first stage a two-velocity stage, that is the first wheel has two rows of moving blades.

(2) Axial Flow Reaction Turbines (Parsons).

The Parsons turbine consists primarily of a casing in which rows of blades are fitted, and a rotor on which are fixed a corresponding number of rows of moving blades.

In the impulse turbine, as stated above, the pressure drop takes place in the fixed nozzles only and is constant in the moving blades, whereas in the Parsons turbine the pressure falls continuously in both the fixed and moving blades. It will therefore be seen that, for the same initial and final steam conditions, the Parsons machine has a far greater number of stages than the impulse machine. Since the steam speed is dependent upon the pressure drop or more correctly speaking the heat drop in each stage, it follows that the steam speeds in a reaction turbine are much lower than in an impulse machine.

The writer is familiar with two propelling installations, one a pressure-compounded impulse turbine for turbo-electric drive, and the other a reaction turbine driving through single-reduction gearing, both using steam at 425lb. pressure (725° F.) and exhausting at 28½in. vacuum, in which there are 21 impulse stages in the former and 105 double stages in the latter. A double stage consists of one fixed blade and one moving blade.

Detailed particulars of a geared turbine installation will be given later.

(3) Impulse Reaction Turbines.

In modern high-pressure geared turbine installations it is usual practice to fit a two-velocity stage wheel as the initial stage of the high-pressure turbine followed by the reaction blading. This has the advantage of reducing the length of the H.P. turbine and the pressure and temperature of the steam entering the first reaction stage. Such a turbine is shown in Fig. 3.

(4) Radial Flow Reaction Turbines (Ljungstrom).

This turbine, which was developed and perfected by Mr. Berger Ljungstrom of Sweden, consists primarily of two rotating discs to which the blades are fixed. One disc rotates in the clockwise direction and the other anti-clockwise. The steam enters at the centre of the turbine and flows radially outwards and finally to the condenser.

The criterion of efficiency of any type of turbine is the ratio of the blade speed to the steam speed. In the pressure-compounded impulse machine described under (*b*) above, the highest efficiency is obtained when the blade speed is 47 per cent. of the steam speed. In the two-velocity stage machine this ratio becomes approximately $23\frac{1}{2}$ per cent. and where three rows of moving blades are fitted $11\frac{1}{2}$ per cent. In the reaction turbine maximum efficiency is obtained when the ratio of blade speed to steam speed is 94 per cent. In the axial flow reaction machine this ratio is not approached in the H.P. and I.P. turbines in spite of the large number of stages fitted, but it will be seen that in the Ljungstrom turbine the blade speed is relatively doubled due to the contrary rotations of the blades, and hence the efficiency of such a turbine is higher than the axial flow type. Moreover the number of stages is considerably less and the axial length of the turbine much smaller than in the axial flow machine.

Turbines of Ljungstrom design have been fitted in power stations, there being one alternator at each side of the turbine, one rotating clockwise and the other anti-clockwise. The Ljungstrom turbine has been put forward for the turbo-electric drive of ships, but so far, apart from a vessel fitted some years ago, no progress has been made in this direction.

General Description of a Modern Geared Turbine Installation.

Fig. 2 shows the general lay-out of a modern triple-cylinder turbine installation. The distribution of power in this case is about 30 per cent. in the H.P., 30 per cent. in the I.P., and 40 per cent. in the L.P. The H.P. and I.P. turbines run at 2,350 r.p.m. and the L.P. at 1,700.

From Fig. 3, which shows a sectional drawing of the H.P. impulse reaction turbine, it will be seen that the

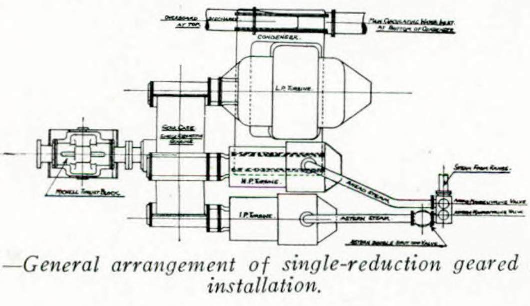


FIG. 2.—*General arrangement of single-reduction geared turbine installation.*

steam is admitted to the turbine by a triple valve box; each valve controls the number of nozzles necessary for the power required. *A* is the nozzle plate which contains in this instance 33 nozzles. The nozzle plate for high-pressure superheated steam is in this case made of phosphor bronze and the vanes comprising the nozzles are of Hecla steel cast in. The valve box mentioned enables various combinations of nozzles from a minimum of three to a maximum of 33 to be obtained. The steam after passing through the nozzles issues at a velocity of about 1,400ft. per second and impinges on the first row of blades of the impulse wheel, then through the fixed guide blade which deflects it on to the second row of moving blades at the correct angle. The steam then enters the first expansion of the reaction blading, then the second and finally the third whence it exhausts to the I.P. ahead turbine, Fig. 4.

It will be seen from Fig. 3 that the reaction part of the turbine consists of a casing in which a number of rows of blades are fixed, increasing in height in each expansion to suit the increasing volume of steam as the pressure falls, and a rotor on which a similar number of rows of corresponding blades are fitted.

The H.P. turbine casing is made of cast steel and since the superheat is carried throughout the I.P. turbine the casing of this turbine, Fig. 4, is also made of cast steel. The rotor of both the H.P. and I.P. turbine is of forged steel and should preferably be solid. If these rotors are hollow there is a danger of the rotor blading binding, that is the shrouding contacting with the casing blades owing to the more rapid cooling down of the rotor as compared with the thick casing, which is efficiently lagged, when the turbines are stopped for any length of time when manœuvring.

The reaction blading used in modern H.P. and I.P. turbines is of the end-tightened type as shown in Fig. 5. This type of blading is essential for obtaining and maintaining high efficiency for, since the higher the steam pressure the smaller is the volume per pound of steam, the clearance losses with the old type of blading due to steam passing over the tops of the blades would be considerable. The running clearance of the installation under

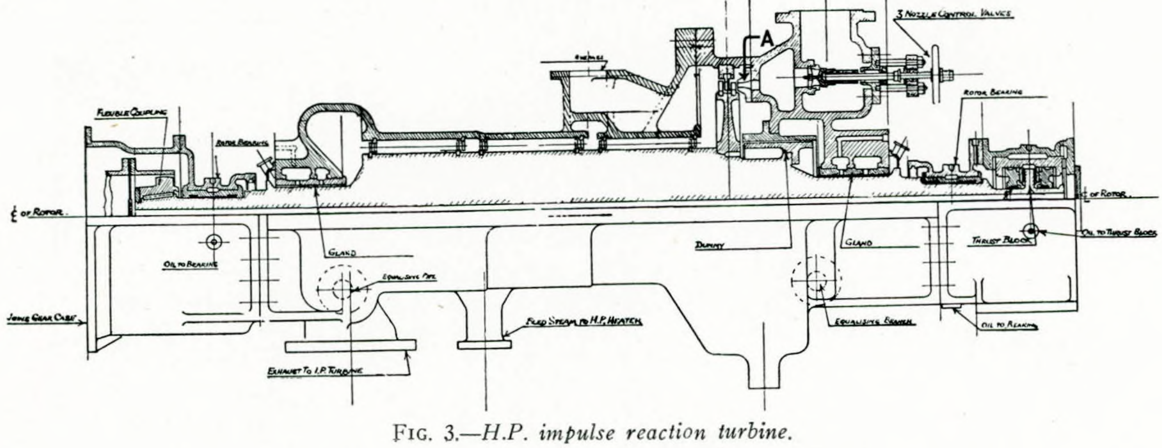


FIG. 3.—*H.P. impulse reaction turbine.*

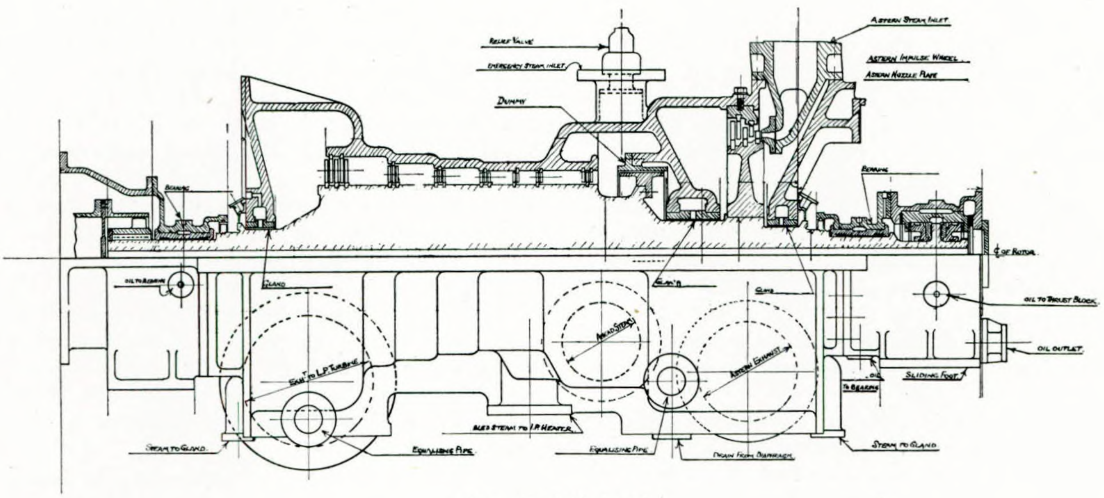


Fig. 4.—*I.P. turbine.*

review as shown in Fig. 5 is $\frac{20}{1000}$ths inch, but blading of this type can quite safely be run at $\frac{5}{1000}$ths inch. In turbines with end-tightened blading, it is a distinct advantage to have a means of moving the rotor bodily aft to

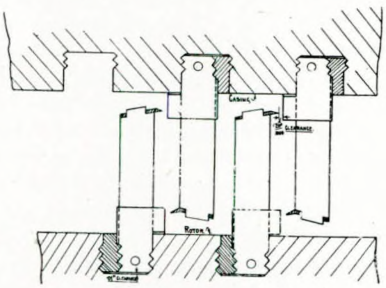


Fig. 5.—*End-tightened blading.*

increase the axial clearance at stand-by and when manœuvring to take care of any relative expansion there may be between the rotor and casing, and so prevent any possibility of binding. The gear for doing this is shown in Fig. 6.

With regard to the material of the blading, the impulse blading of the first-stage wheel should be preferably of Hecla A.T.V. steel, and the reaction blading of the H.P. and I.P. turbines of Monel metal or stainless iron.

Referring to Fig. 3, it will be observed that at the

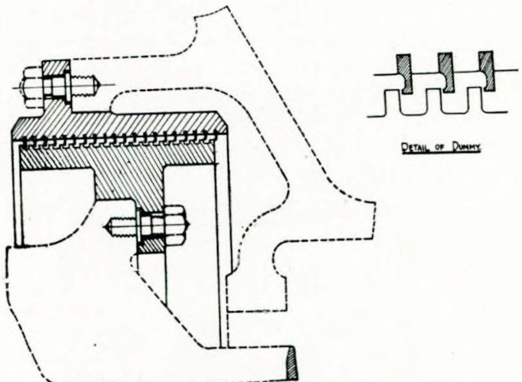


Fig. 7.—*Axial type turbine dummy.*

steam inlet end there is a dummy. As will be readily understood, due to the difference in pressure in each expansion of the turbine and the increasing height of the

Fig. 6.—*Adjusting gear for turbines with end-tightened blading.*

Fig. 8.

blades, there is a steam thrust acting on the rotor in the direction of steam flow which has to be balanced. This is the function of the dummy. Dummies are of two types. In the high-pressure and intermediate-pressure turbines they are of the "contact" type and in the low-pressure turbine of the radial type similar in form to the labyrinth gland (Figs. 9 and 10). The two types are

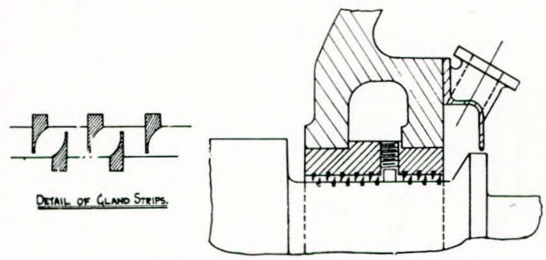


FIG. 9.—*Single-pocket turbine gland.*

shown in Figs. 7 and 8. In the contact dummy the clearance between the casing fins and the collars on the rotor is the same as the running clearance of the end-

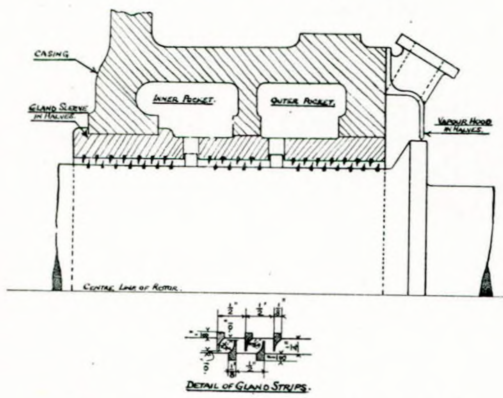


FIG. 10.—*Two-pocket labyrinth gland.*

tightened blading, and care should be taken to see that this clearance is maintained.

Some builders do not fit a dummy in the L.P. turbine, but in the writer's opinion a dummy should be fitted as there is a possibility in quick astern moves following ahead running for steam to get trapped and "whipped" up to a high temperature, producing rapid local expansion which may cause damage.

An equalising pipe is fitted (Fig. 3) between the exit end of the dummy and the exhaust belt of the turbine to balance the pressure at each end of the turbine.

At each end of the turbine there is a labyrinth gland which in the H.P. and I.P. turbines reduces the steam leakage into the engine room to a minimum. In the L.P. turbine the glands prevent air leakage into the turbine which would destroy the vacuum. The gland consists of a number of brass fins on the rotor shaft, and a corresponding number in the turbine casing, the radial clearance between the rotor fins and the casing and the casing fins and rotor being usually $\frac{10}{1000}$ths inch. The number of fins required is determined by the difference in pressure between the atmosphere and the turbine side of the gland.

The glands may have one, two or three pockets. The latter number are sometimes fitted to the H.P. turbine, but more often a two-pocket gland is fitted. In the I.P. and L.P. turbines a single-pocket gland is normal practice. Figs. 9 and 10 show single- and two-pocket glands. The glands are steam packed and in recent installations the supply of steam to the glands is arranged as shown in Fig. 11. It will be seen from the gland steam diagram (Fig. 11) that there is a collector which is kept at a pressure of from 3 to 4lb. gauge. Steam is supplied from the collector to the outer pockets of the glands and, when running ahead, steam is supplied to the collector from the inner pockets through valve *D*, the desired pressure to maintain a feather of steam at the gland mouths being obtained by regulation of valve *C*, which is connected to the low-pressure turbine steam belt. At stand by, valves *C* and *D* are closed and the pressure in the collector is maintained by steam from the auxiliary steam line by regulating the valves *A* and *B*, the latter valve being connected to the condenser. Hence with this system the steam to glands is easily regulated by the operation of two valves instead of the multiplicity of valves necessary in older installations.

In a very recent marine installation the vapour outlet from the glands was passed into a small condenser, one to each gland, the cooling water being the condensate. The condensed vapour was led into the distilled feed-water tanks or alternatively into the hard feed-water tanks. When it is remembered that in all high-pressure installations water-tube boilers are a *sine qua non,* and consequently only distilled fresh water may be used, any saving of make-up feed water is to be welcomed. The writer can say that the installation of these vapour condensers was highly successful, and the saving in make-up feed water in comparison with other installations of similar power output was very satisfactory.

Bearings are fitted at each end of the rotor. The top and bottom halves are made of gunmetal lined with white-metal. The centre line of the bore of the white-metal is half the total oil clearance above the centre line of the shaft. The outer ends of the bearing sleeves, which form safety strips should the white-metal be run owing to loss of oil, are bored $\frac{30}{1000}$ths in. larger in diameter than the white-metal and concentric with it. The total oil clearance is usually not less than $\frac{15}{1000}$ths in., increasing according to the shaft diameter. As a guide the total oil clearance for rotor and pinion bearings may be taken as $\frac{15}{1000}$ths in. for 6-in. to 7-in. diameter shafts, $\frac{16}{1000}$ths in. for 8-in. to 9-in. diameter, $\frac{17}{1000}$ths in. for 9-in. to 10-in. diameter and $\frac{18}{1000}$ths in. for $10\frac{1}{2}$-in. to 11-in. diameter. In the event of the white-metal being run the shaft would come down on the safety strips and prevent the rotor blading being stripped by fouling the casing. There are no oil gutters or grooves in the white-metal, the lubricating oil being admitted through large channels on the horizontal centre line at each side of the bearing. Care should be taken when remetalling to see that the white-metal is carefully chamfered in way of the channels to retain their contour and facilitate the flow of oil round the journal. This feature is shown in the drawing of a rotor bearing, Fig. 12.

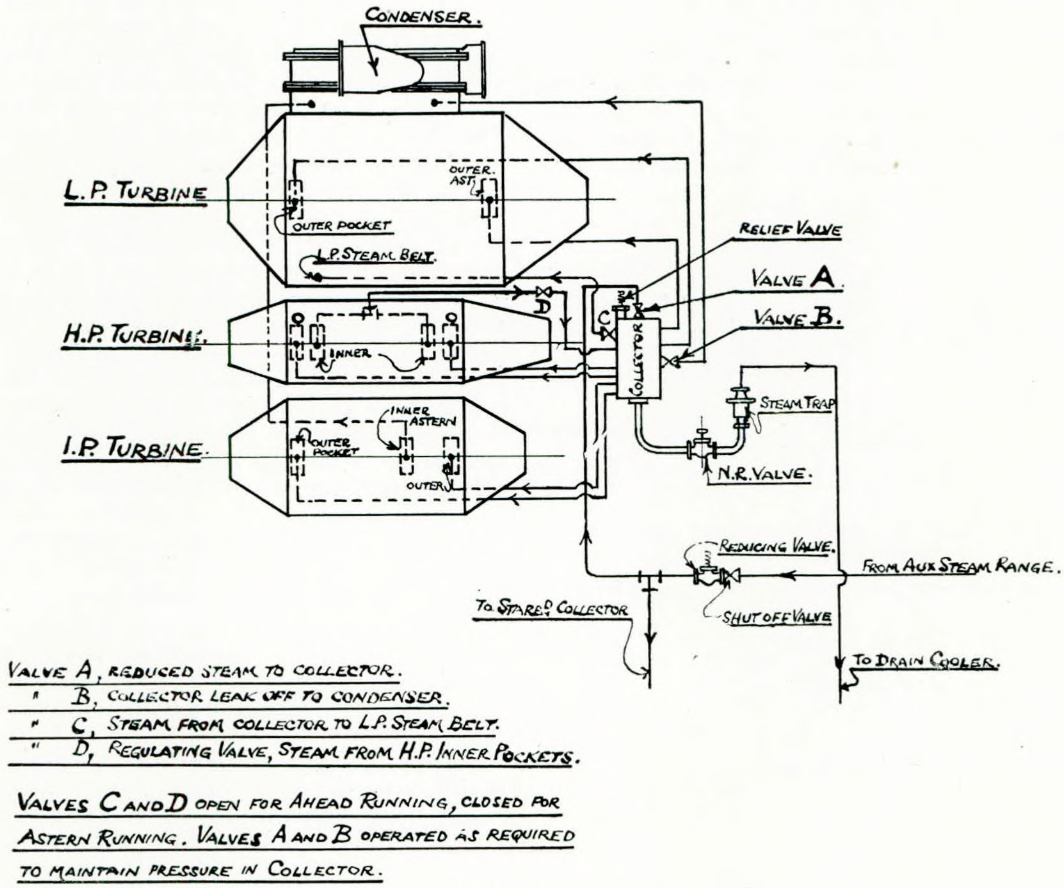


FIG. 11.—*Gland steam diagram.*

At the forward end of the turbine a thrust block of the Michell type is fitted. This block takes any residual steam thrust on the rotor blading not taken up by the dummy, but its principal function is the correct axial adjustment of the rotor relative to the casing so as to maintain the designed dummy and blading clearances.

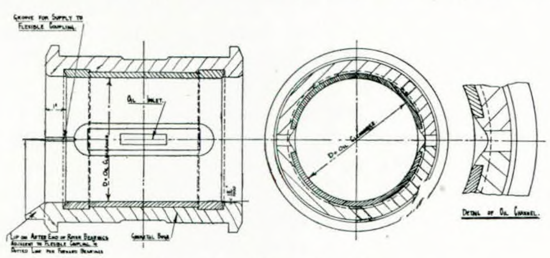


FIG. 12.—*Rotor bearing.*

As mentioned above, with end-tightened blading a means of varying the axial blading clearance is advantageous. The thrust block is therefore constructed so that it can slide inside its housing between fixed liners to permit of the rotor being moved bodily between the prescribed limits of running and stand-by clearances. Fig. 13 shows such a block. When the block is hard against the forward liner the minimum running clearance is obtained,

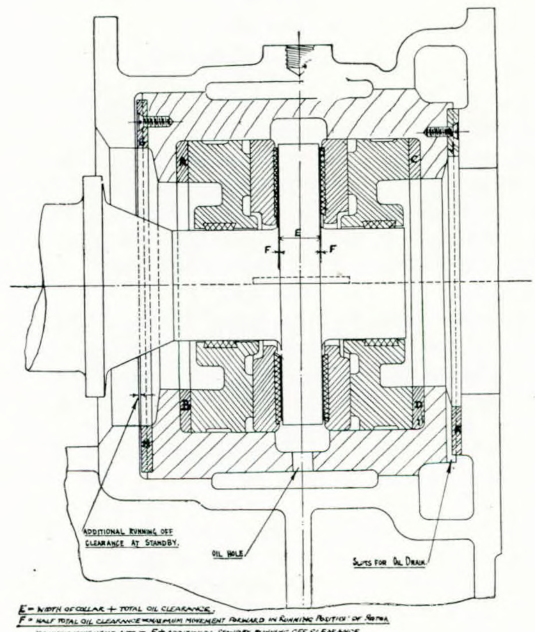


FIG. 13.—*Turbine adjusting block for end-tightened blading.*

and in this particular design the rotor can be moved aft to give an additional $\frac{30}{1000}$ths in. at stand-by. For turbines not fitted with end-tightened blading the adjusting block is of the type shown in Fig. 14.

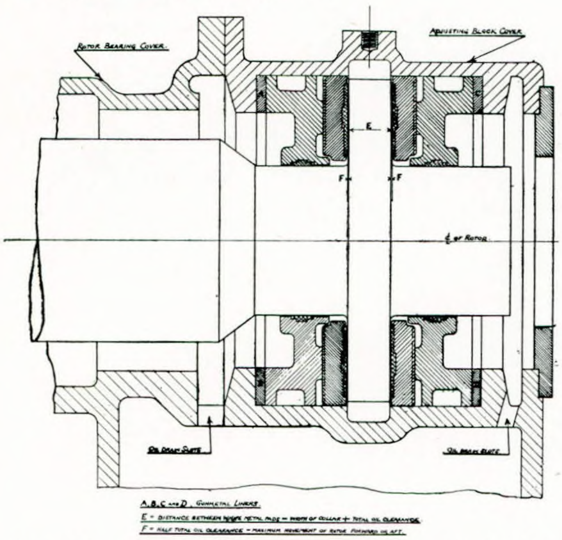


FIG. 14.—*Adjusting block for turbines not fitted with end-tightened blading.*

At the after end the connection between the turbine and the pinion is by means of a flexible coupling, Fig. 15. The function of the flexible coupling is to permit the pinion freedom to take up its correct alignment with the gear wheel, and to allow for any difference in expansion between the rotor and casing. It will be seen that the coupling consists of a forged-steel claw piece keyed to the conical end of the rotor shaft, there being a retaining nut which has a locking pin to prevent its slacking back.

A similar claw piece is fitted to the conical end of the pinion shaft. The claw teeth gear with similar teeth formed on the inside of the outer forged-steel sleeve. A steel plate is fitted between the forward and after sections of the sleeve; the clearance between the end of the rotor shaft and the plate is $\frac{3}{16}$in. and also between the end of the pinion shaft and the plate, making a total relative axial movement between rotor and pinion of $\frac{3}{8}$in. Oil grooves are cut in the ahead and astern faces of each claw tooth. Oil is supplied to the coupling from oil grooves cut in the horizontal joint of the rotor and pinion bearings, the grooves connecting with the oil channels previously mentioned, Fig. 12. A lip is formed on the after end bottom half of the rotor bearing, and on the forward end bottom half of the pinion bearing to guide the flow of oil into similar lips on the coupling whence it passes through the oil holes to the teeth. These oil holes are sometimes led to the ahead faces of the teeth only or they may be disposed centrally as shown in Fig. 15. Two $\frac{5}{16}$in. diameter drain holes are arranged in the forward half of the outer sleeve, and two in the after half to ensure an ample flow of oil through the coupling. The lubrication of the coupling should be carefully watched, and the coupling should be opened up from time to time for cleaning and removing any sediment and also to verify freedom of axial movement. Witness marks are typed on the ends of two of the claw teeth, and on the intervening sleeve tooth, to ensure that the coupling after overhaul is assembled with the teeth in their original positions. The coupling is enclosed in a casing which is attached to the main gear case, the oil from the coupling draining into the gear-case sump.

The explanations given of the dummy, glands, bearings, thrust or adjusting block, and the flexible coupling for the H.P. turbine apply also to the I.P. and L.P. turbines.

In geared turbine installations comprising three turbines as shown in Fig. 2, astern turbines are incorporated in the I.P. and L.P. turbine casings. The astern power of these turbines should be at least 70 per cent. of the full ahead power.

Fig. 4 shows a typical arrangement of the I.P. turbine of Fig. 2. In this turbine the astern portion is a three-velocity stage impulse wheel, the ahead portion being all reaction with end-tightened blading. The gland between the ahead and astern turbine is preferably connected to the condenser as shown in gland steam diagram, Fig. 11.

Fig. 16 shows the L.P. turbine. Here the astern turbine is of the impulse-reaction type, the impulse wheel being a two-velocity stage followed by reaction blading. The ahead portion of the turbine is all reaction. End-tightened blading is not necessary in L.P. turbines as the

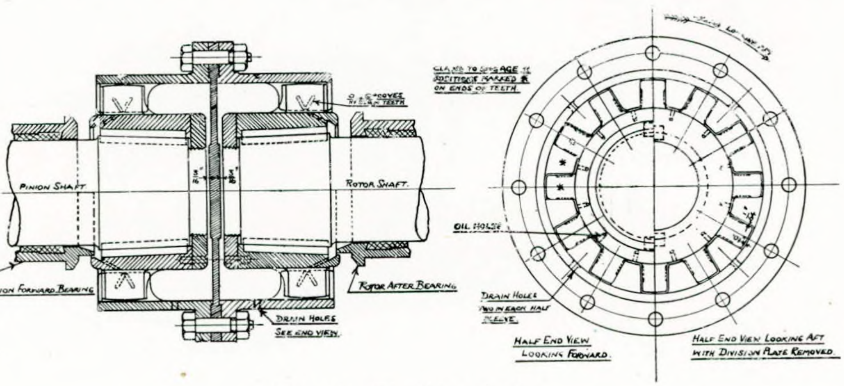


FIG. 15.—*Flexible coupling.*

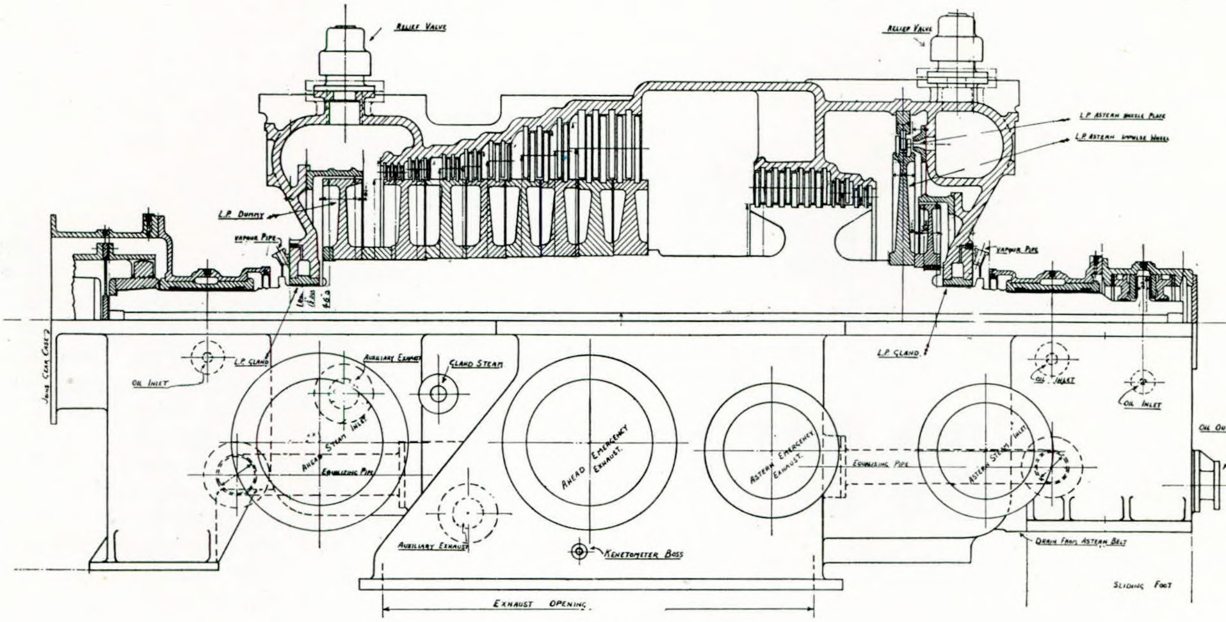


Fig. 16.—*L.P. turbine.*

volume per pound of steam is high, hence the clearance losses with ordinary blading are comparatively small.

GEARING.

In modern vessels single-reduction double-helical gearing is standard practice, there being a forward helix cut to one hand and an after helix to the opposite hand, thus balancing the thrust. The forward helix takes half the load and the after helix the other half. Fig. 17 shows a general arrangement of a single-reduction gear. The contour of the tooth in earlier gears was of involute form, but this has now been superseded by the "all-addendum" tooth. An illustration of this form of tooth is shown in Fig. 18.

The main gear wheel in Merchant Service vessels has a cast-iron spider keyed to a slightly tapered cone on the shaft, with a forged-steel rim shrunk on to its periphery on which the teeth are cut. The pinions are of solid-forged nicked steel. Extreme accuracy in cutting the gears is absolutely essential for quiet running, efficiency and durability.

The life of the gear will materially depend upon its

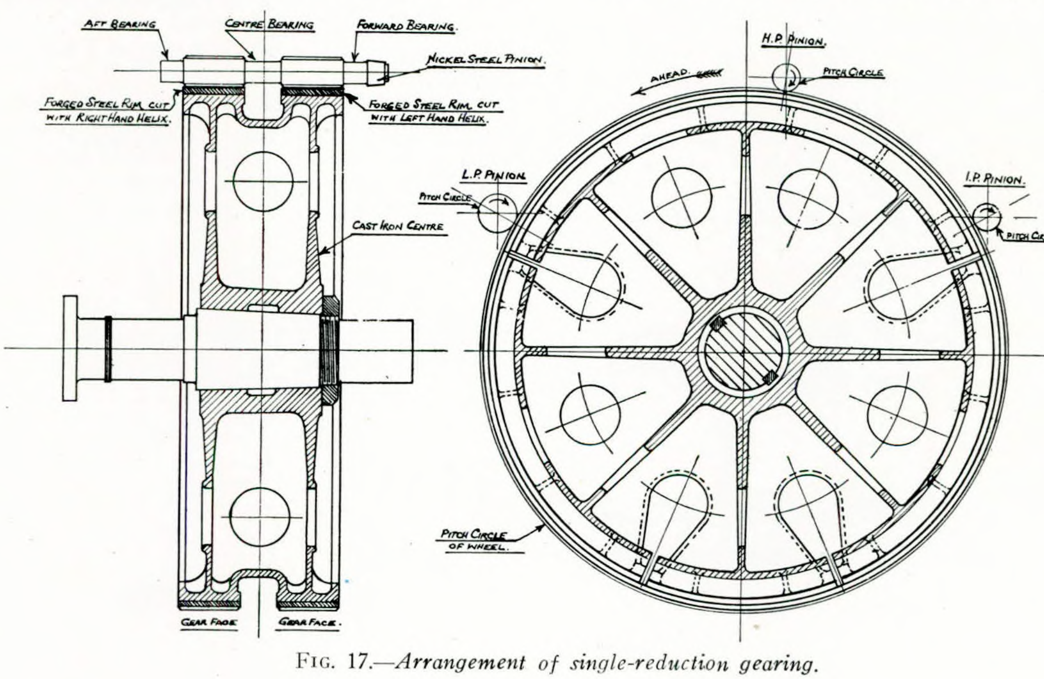


Fig. 17.—*Arrangement of single-reduction gearing.*

treatment in the first year of its running and during this period it is not advisable to load the gearing for any length of time beyond 70 per cent. of the full power for which it was designed. In this way the gears will be carefully "run-in". In gears of the all-addendum type there

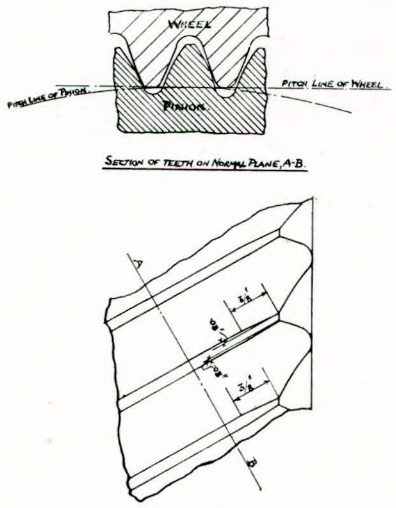


FIG. 18.—*All-addendum gearing.*

may be a certain amount of pitting during the early period of its running, and care should be taken at terminal ports to hone the hard spots on the teeth to improve the bearing surface, and it is essential to see that the ends of the teeth are chamfered away for at least ⅜in. at each end of each helix to relieve the pressure. The action of the teeth is, first, sliding contact, followed by rolling contact; this is mentioned because there is another form of tooth called the "enveloping tooth" in which the action is principally rolling. Enveloping tooth gears have been fitted in certain vessels, and a sketch of the tooth form is shown in Fig. 19.

With regard to the number of pinion bearings,

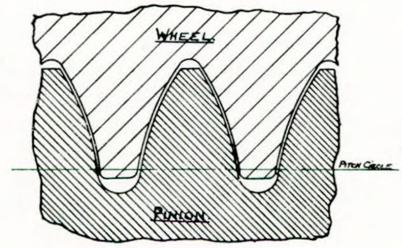


FIG. 19.—*Enveloping gear teeth.*

normal practice in merchant liners is to fit three bearings, one at each end, and one in the centre. As a general rule if the width of the gear face, *i.e.* the two helixes, exceeds four times the pitch circle diameter of the pinion, three bearings must be fitted, and this is normally the case in merchant vessels.

As will be appreciated, ample lubrication of the gearing is essential. For this purpose sprayers are fitted to the gear case, spaced about five to six inches apart, which project the oil along the tangent plane of the pinion and wheel. Each sprayer has a visible sight glass usually of the spinner type through which the oil passes before entering the sprayer, thus ensuring that oil is passing to the gearing. Any stoppage of flow must be immediately investigated. Magnetic strainers are fitted on the lubrication system, and it is advisable that they should be examined and cleaned at least every two weeks as the particles adhering to the magnets give an indication of gear wear. The forced-lubrication system is discussed in a later section.

For taking the thrust of the propeller, a Michell thrust block (Fig. 20) is now standard practice. The

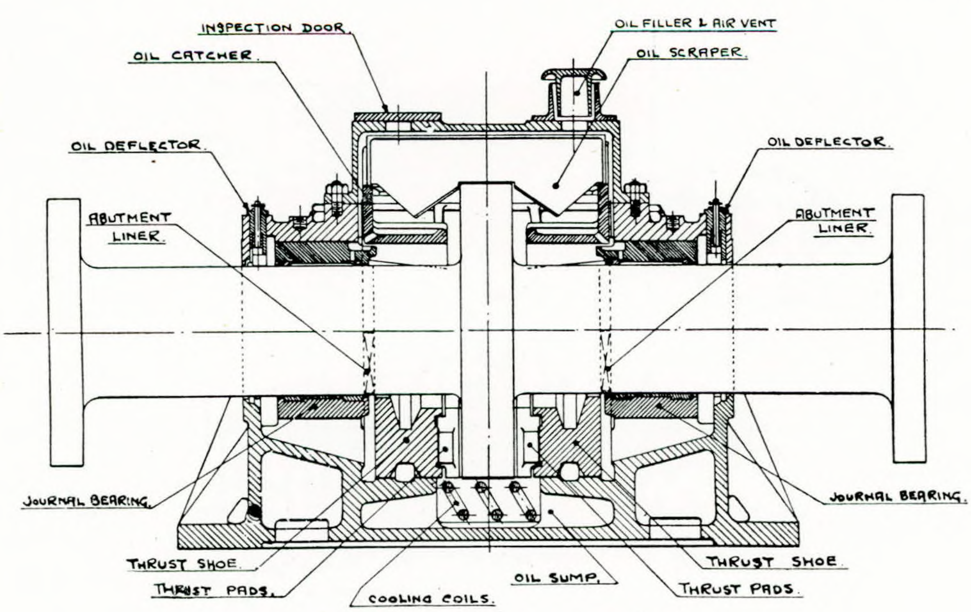


FIG. 20.—*Standard marine thrust block.*

Michell thrust block is so well known that a detailed description is not called for. The usual form has its own oil sump, but in some installations the block is supplied with oil from the forced-lubrication system.

The Forced-Lubrication System.

In modern twin-screw liners the forced-lubrication system usually consists of two electrically-driven either reciprocating or rotary pumps, one pump working and one stand-by. Each pump has a capacity of at least 1 gallon of oil per hour per shaft horse-power. That is, in a 20,000 s.h.p. installation each pump should have a capacity of at least 20,000 gallons per hour. The pumps draw from a drain tank through the magnetic strainers and discharge through filters of the Auto-Klean, Lolos or other fine type to the lubricating oil coolers, and thence to the gravity tank which is placed at a height in the engine room casing to give a pressure of about 10lb. per sq. inch at the turbine bearings and gearing oil sprayers. A continuous overflow from the gravity tank is led to the drain tank, a sight glass visible from the starting platform being fitted in this pipe line. The cool oil from the gravity tank is led to all the bearings, turbine thrust blocks and gearing sprayers. The oil from each turbine bearing is led by a separate pipe to a sight glass, where the flow may be sighted and the temperature determined, and thence back to the drain tank. The drain from each gear-case sump is led separately to the drain tank. In connection with the lubricating system it is normal practice to install two centrifugal lubricating oil purifiers, the pumps of which draw a portion of the oil from the bottom of the drain tank and continuously purify it, the oil having been previously heated in a steam or electric heater to about 160° F., and return it to the drain tank. It is an added advantage if hot fresh water in the form of a spray is admitted into the top of the purifier bowl. The designed purifier revolutions should always be fully maintained and from time to time should be checked by a tachometer. In addition to the gravity and drain tanks there is a settling tank and a reserve lubricating oil tank, each having about half the capacity of the gravity tank which should hold about 10 minutes' supply of oil. Con-

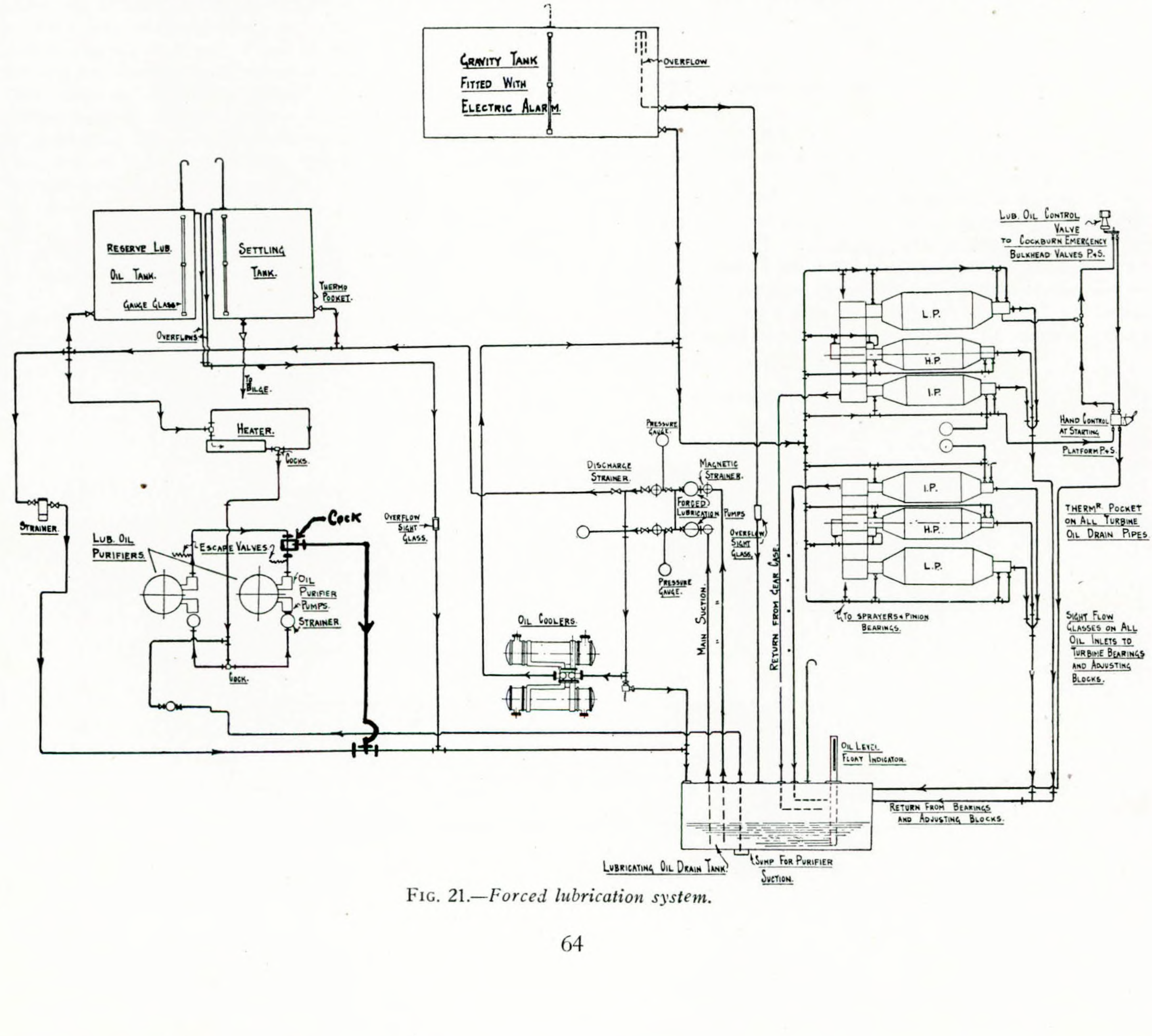


FIG. 21.—*Forced lubrication system.*

nections from the forced-lubrication system are also led to the Aspinall governor, Cockburn emergency bulkhead stop valves and, in vessels having the closed-feed system, to the L.P. turbine excess steam pressure valve, descriptions of which items will be given later. In running the system it is essential to see that the overflow from the gravity tank mentioned above is continuous, that all gearing oil sprayers are working, and that ample oil is flowing to the flexible couplings and leaving the turbine bearings and thrust blocks. An electric alarm is fitted to the gravity tank to warn the engineers on watch should the level fall below a predetermined level, and either an audible warning or a warning light at the starting platform to inform the engineers should the lubricating oil pump stop.

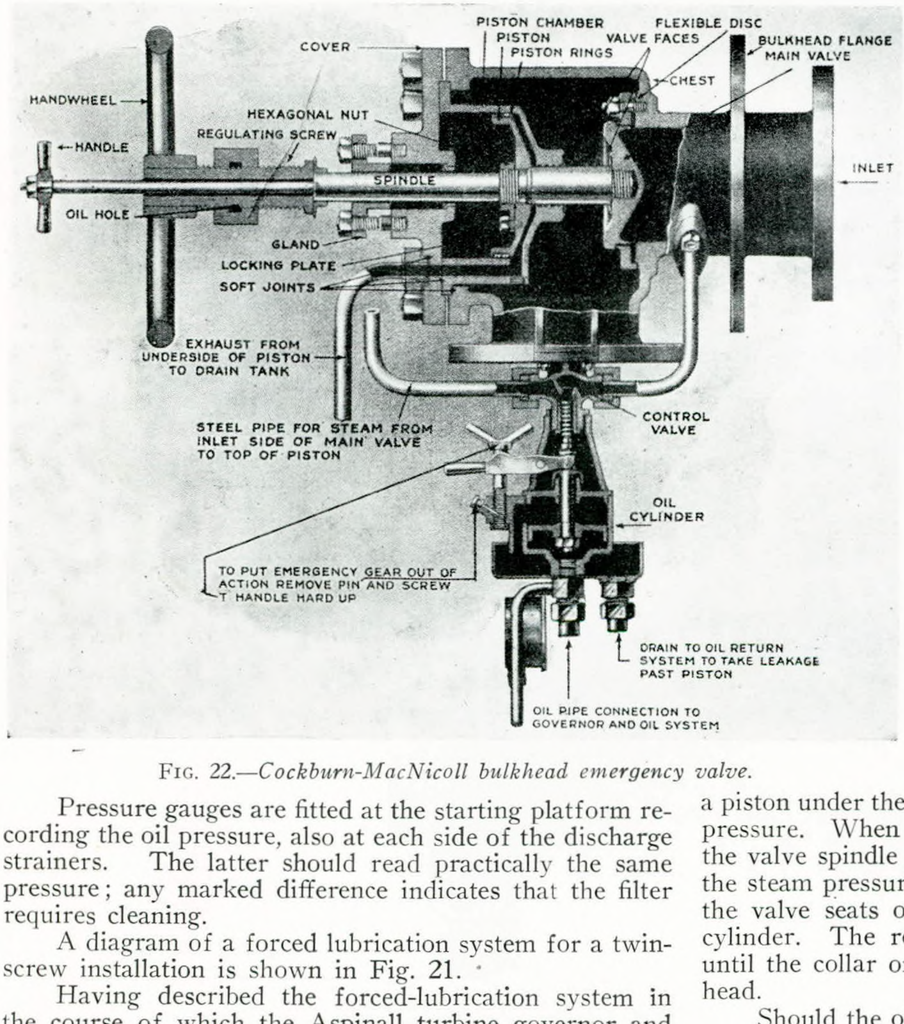


FIG. 22.—*Cockburn-MacNicoll bulkhead emergency valve.*

Pressure gauges are fitted at the starting platform recording the oil pressure, also at each side of the discharge strainers. The latter should read practically the same pressure; any marked difference indicates that the filter requires cleaning.

A diagram of a forced lubrication system for a twin-screw installation is shown in Fig. 21.

Having described the forced-lubrication system in the course of which the Aspinall turbine governor and Cockburn-MacNicoll emergency bulkhead shut-off steam valves have been mentioned, a description of these items will now be given.

In recent vessels a governor is fitted to one turbine only of each set of turbines. The importance of the governor is apparent, for if through any cause the propeller should be lost the turbines would run away, shed their blades, and might burst with disastrous results. The governor works in conjunction with the emergency bulkhead stop valves through the medium of the forced-lubrication system, so that these valves are under the dual control of the governor and the forced-lubrication oil pressure. In modern high-pressure geared-turbine-driven vessels in which the closed-feed system is essential, the bulkhead emergency valves are also under the control of any excessive steam pressure in the L.P. turbine which may take place owing to the stripping of the H.P. or I.P. turbine blading. This control is necessary as there is a closed circuit from the boiler through the condenser and back to the boiler, and in the event of an accident to the turbines mentioned, the relief valves on the L.P. turbine could not get rid of the steam in time to prevent the pressure rising to a point which might burst the L.P. casing or the condenser. Arrangements are made in twin-screw vessels in which the port bulkhead emergency valve is connected to the port governor and excess pressure L.P. turbine valve, so that in the event of the port turbine running away due to losing the port propeller or an excess of pressure in the port L.P. turbine, only the port set of turbines is shut down leaving the starboard set running. The same conditions apply to the starboard emergency valve.

In the event of losing the forced-lubrication pressure, both sets of machinery are closed down by the automatic closing of the port and starboard emergency valves.

The Cockburn-MacNicoll Emergency Valve is shown in Fig. 22. As will be seen there is a piston attached to the valve spindle working in a cylinder incorporated in the valve chest, the piston being of larger diameter than the valve. A pipe led from the boiler side of the valve to the top side of the piston is controlled by a small valve which is operated by a piston under the direct influence of the forced-lubrication pressure. When the regulating screw which slides over the valve spindle is moved back, the valve follows due to the steam pressure on its under side until the top side of the valve seats on the valve face on the bottom of the cylinder. The regulating screw should be moved back until the collar on its inner end contacts with the cross-head.

Should the oil pressure under the piston of the small control valve fall below about 3lb. per sq. inch, the control valve opens and passes steam to the top side of the piston which, being of larger area closes the stop valve. On the oil pressure being restored the control valve shuts and the main valve opens again, the steam in the cylinder having leaked past the piston rings and drained away to the drain tank.

A hand control valve is fitted to each oil-operated control valve to enable the respective emergency bulkhead valves to be closed from the starting platform. It is also normal practice to have a similarly operated control at the entrance to the engine room to enable the whole turbine installation to be shut down in case of an accident of such a nature as to prevent the engineers getting to the hand controls at the starting platform. The hand controls at the starting platform should always be tried at each port when "Finished with Engines" is rung on the telegraph.

In all twin-screw vessels there are two lines of main steam pipes in the boiler room leading to the bulkhead emergency valves with a cross connecting valve between. If the latter valve is on the boiler side of the emergency valves, it is an ordinary screw-lift valve, but if it is on the engine room side it must be a Cockburn emergency type valve.

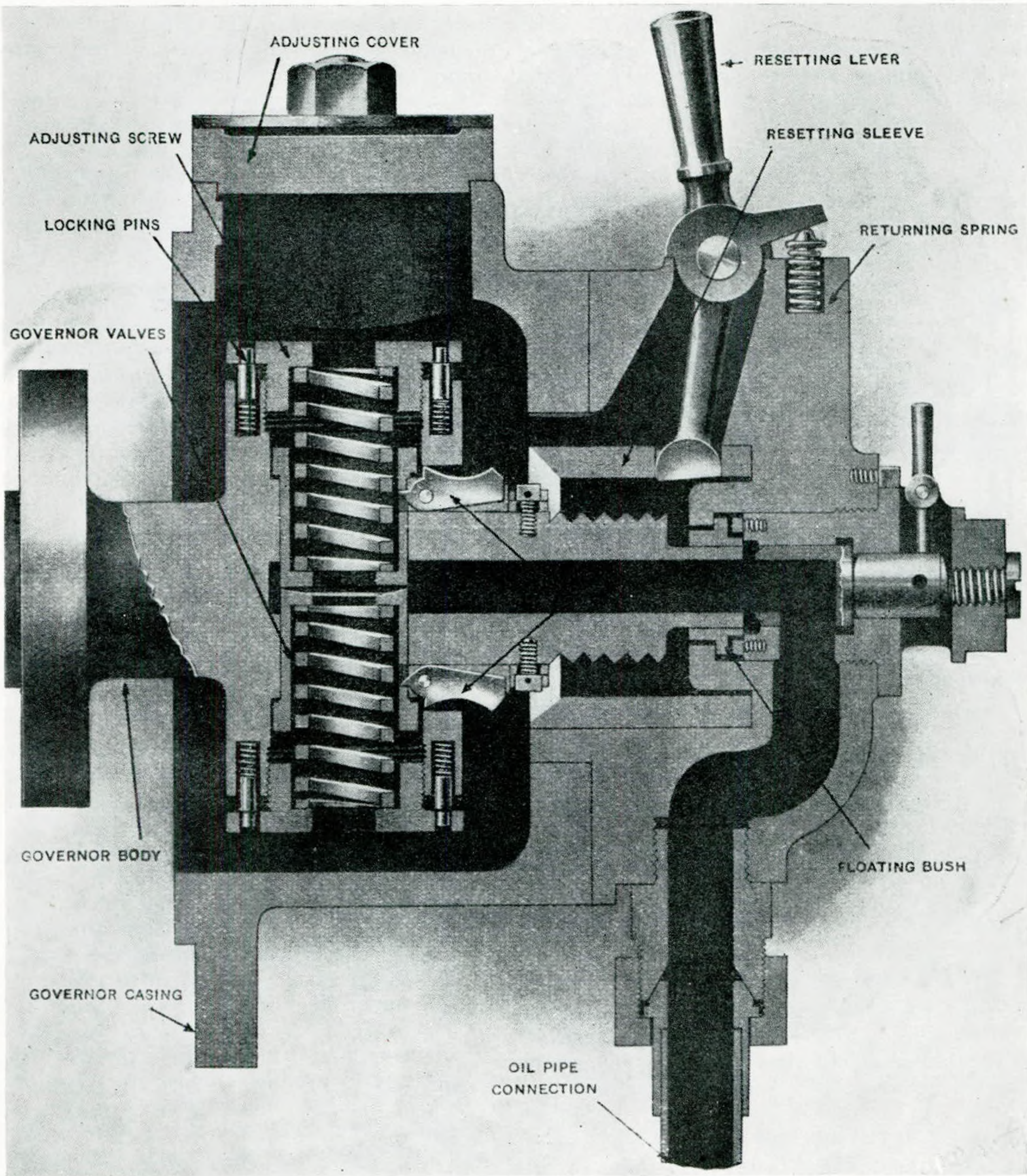


Fig. 23.—*Aspinalls patent governor.*

The Aspinall governor which works in conjunction with the emergency bulkhead valve is shown in Fig. 23. The governor body which is connected direct to the turbine rotor shaft has a cross cylinder containing two piston valves held closed by the compression of two springs, the compression determining the speed at which the valves open by centrifugal force and drop the oil pressure, thus closing the emergency bulkhead valve. The oil to the governor is supplied by the forced-lubrication system to a lantern sleeve having a small radial clearance permitting of its lubrication and allowing for movement of the turbine rotor. The illustration shows a combined lantern sleeve and poker gauge for obtaining the axial position of the turbine rotor. The cut-out speed is generally 15 per cent. above the normal full power revolutions of the turbine. When this speed is reached, the two pawls shown lock the governor valves open, thus allowing the oil pressure to drop and close the bulkhead valve. The governor should not be reset until the bulkhead valve regulating screw is run up into the closed position. The resetting lever may then be pulled forward pushing the resetting sleeve aft, the bevelled end of which places the pawls in their normal position and allows the governor valves to close.

Ahead and Astern Manœuvring Valves.

The steam from the bulkhead emergency valves is led to the ahead and astern manœuvring valves which control the steam to the turbines. The type of valve used is of the balanced double-beat type usually of Cockburn design. It will be seen from Fig. 24 that the valves are arranged in one chest, side by side, and in the illustration the valves take steam on the outside. The top beat valve is larger than the bottom one, and to balance the difference in pressure a balance piston is fitted as shown. The top valve has a loose washer and a flexible disc. The object of the disc is to allow for the difference in expansion between the valve faces and the seat faces, so that the top and bottom valves may be kept tight under all conditions.

In adjusting the valves after overhaul, the complete valve should be carefully lowered until the disc is just resting on the top seat. The bottom valve should then be just off its seat by about $\frac{1}{1000}$ths inch. This ensures that when the bottom valve is tightened down on its seat the disc is flexed and thus keeps the top valve tight.

A double shut-off screw-lift valve should always be fitted adjacent to the astern valve, worked from the starting platform similarly to the manœuvring valves, so that should the astern valve develop a leak the steam will not pass to the astern turbines and cause an excessive temperature in the L.P. astern turbine casing. The writer has experience of two vessels in which the astern casing adjacent to the condenser was cracked due to this cause. A long-distance thermometer with a dial located at the starting platform should be fitted to the L.P. astern casing, and in addition a pressure gauge should be fitted between the astern manœuvring valve and the double shut-off valve to inform the engineers if any pressure is building up between the valves, in which case they will know that the astern valve is leaking. In a high-pressure superheated job, the temperature of the L.P. astern casing should not exceed 270° F. when running ahead. Also it will be obvious that any leakage of steam to the astern turbines when running ahead will have a retarding effect.

On the ahead and astern steam branches, steam strainers are fitted to receive any foreign matter which may come through with the steam and to prevent its entering the turbine and damaging the blading. The steam enters the inside of the strainer, the cage of which can be withdrawn and cleaned without breaking any pipe joints. These strainers should be taken out and cleaned at each terminal port.

Having dealt with the steam supply to the turbines, the condenser will now be described.

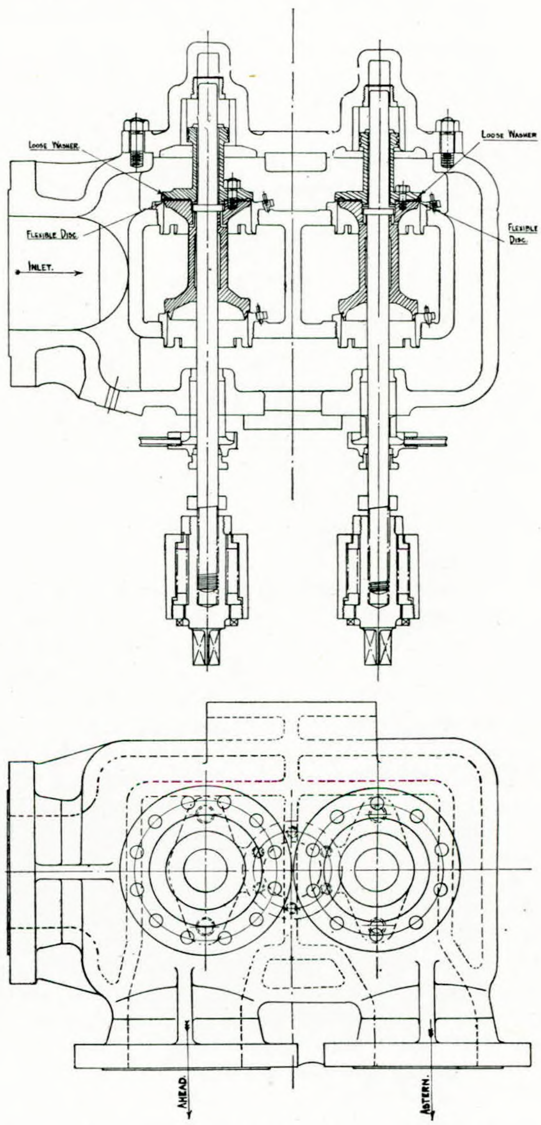


FIG. 24.—*Ahead and astern manœuvring valves.*

The Main Condenser.

The condensers adopted in all modern turbine installations are of regenerative type. With the older type of condenser the temperature drop between the exhaust steam and the condensate was of the order of 10° F. to 15° F. This temperature drop is equivalent to approximately 1 to $1\frac{1}{2}$ per cent. increase in fuel consumption. The underlying principle of the regenerative condenser is to pass a certain proportion of the exhaust steam direct to the bottom of the condenser to heat up the condensate to approximately the steam temperature. In practice the difference in the steam and condensate temperature with regenerative condensers is between 1° F. and 2° F. An air cooler formed by baffling off a certain proportion of the tubes is an essential feature in the design of all high vacua condensers. For efficiency in steam consumption it is of paramount importance in turbine installations to maintain the highest possible vacuum commensurate with the sea-water temperature. The following table may be taken as a guide to the best vacuum obtainable under practical running conditions with a two-pass regenerative condenser; the temperature of the condensate allowing 2° F. drop is also given.

Sea-water temperature. ° F.	*Vacuum in inches on 30" barometer.*	*Condensate temperature.* ° F.
50	29·25	68·0
55	29·15	72·0
60	29·0	77·0
65	28·8	82·5
70	28·6	87·4
75	28·35	92·7
80	28·10	97·3
85	27·80	102·2
90	27·45	107·3
95	27·05	112·3

A typical drawing of a regenerative condenser is shown in Fig. 25.

The majority of merchant ship condensers have ferrules at each end of the tube, the packing of which should preferably be of the metallic type. Three-quarter inch outside diameter tubes 16 to 18 gauge thick are standard practice, and tubes and ferrules should be made of either cupro-nickel 70/30 mixture or aluminium brass, both of which materials have given satisfactory results in practice. Leakage of sea water must be avoided at all costs. Care should be taken when condensers are opened up for examination to see that all stay nuts are tight, as they are often the cause of slight leakage into the condensate especially when manœuvring. A soft copper varnished washer between the nut and the tube plate has been found often to effect a cure.

At the bottom of the condenser a well is formed having a water capacity according to the size of the installation in which a float works operating the supplementary feed valve which maintains the level which is shown by a gauge glass adjacent to the condenser. The make-up supplementary feed should enter the condenser through a sprayer as high up on the condenser as possible to ensure that the water is properly de-aerated before passing into the feed system. De-aeration of the feed water is essential in water-tube boiler installations and in the closed-feed system, which will be explained later; all feed-water passing to the boilers is led to the condenser for de-aeration.

The condenser doors should preferably be hinged and large manholes should be fitted in each door to enable tubes to be stopped or packed without having to open or remove the complete door. Air pipes should be fitted to the top of each waterbox to prevent air locks and remove air from the circulating system.

For the measurement of the vacuum in high vacua condensers the vacuum gauge is not reliable and a special instrument known as a Kenotometer is fitted for this purpose, the vacuum gauges only acting as a guide to the engineers.

In high-pressure installations in which clean steam is a *sine qua non,* it will be found that the steam side of the condenser tubes keeps exceptionally clean but it is necessary to brush through the water side of the tubes on the completion of each voyage or at least every three to four months. The non-corrosion plates fitted in the water boxes should be maintained in an efficient condition as experience has proved they are a decided advantage in the protection of the cast iron.

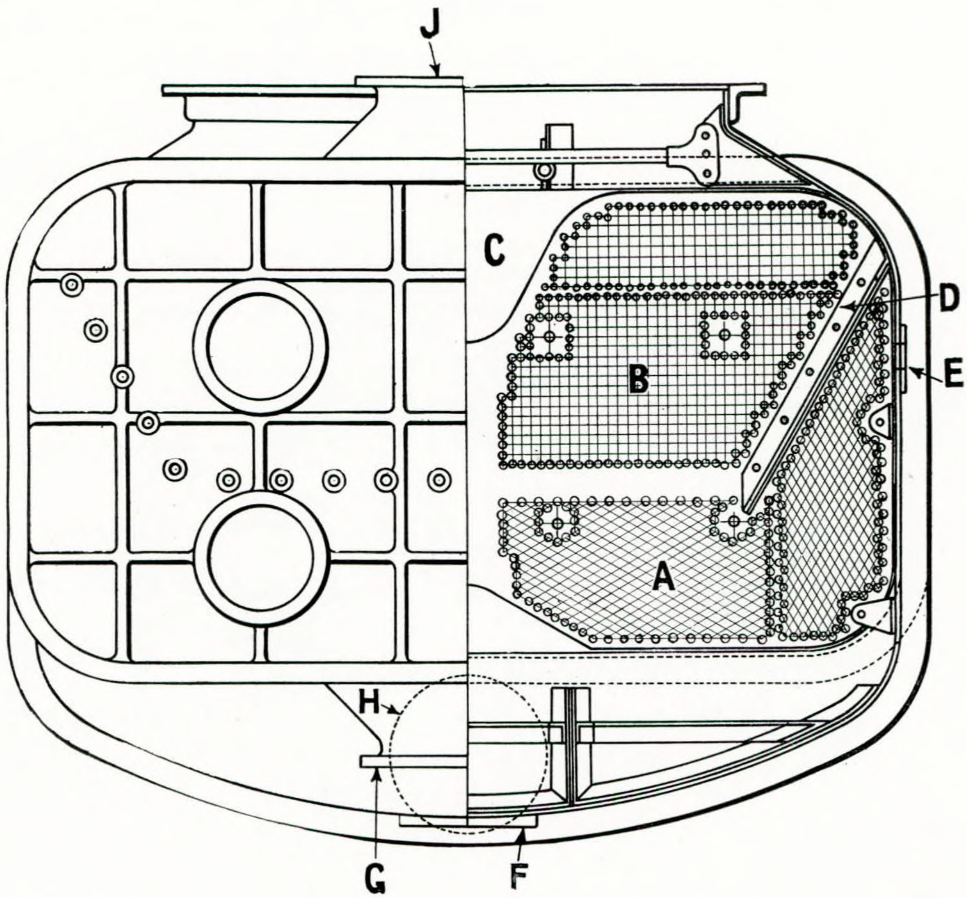
The Closed-feed System.

The closed-feed system is essential in all high-pressure water-tube boiler installations. As its name implies there is a closed circuit from the boilers to the condenser and back to the boilers. The oxygen content of the feed water leaving a regenerative condenser is extremely small and due to the closed feed circuit the possibility of absorption of oxygen by the feed water between the condenser and boilers is eliminated. Oxygen-free feed water for high-pressure water-tube boilers is essential in the prevention of boiler tube corrosion. Fig. 26 shows a typical arrangement of a closed-feed system for a modern high-pressure turbine installation with three-stage feed heating and incorporating a low-pressure evaporator. The latter is not an essential feature of the closed-feed system and so far has not been fitted in many vessels, but in ships so fitted it has proved itself to be a very efficient unit. Referring to the diagram, it will be seen that the condensate is drawn from the bottom of the condenser by a rotary water extraction pump, the head above the pump suction being not less than three feet, and is pumped at about 20 to 25lb. pressure through the air ejectors, then through the drain cooler to the suction of the turbo feed pump. Sometimes, according to the feed temperature conditions adopted in the design, the L.P. heater is placed between the water extraction pump discharge and the turbo feed pump suction, but in the installation under review all the heaters were placed on the discharge side of the turbo feed pump. The feed water passes from the latter pump through the L.P., I.P. and H.P. feed heaters and feed filters to the boilers. Steam for feed heating is bled from the turbines at three points, one to each heater. The drain from the H.P. heater goes into the I.P., the drain from this heater passing to the L.P. heater, the combined drain from the latter being led to the drain cooler and finally to the condenser.

Since the condensate is used to condense the steam supplied to the air ejector, it is necessary at stand-bys and when stopped to have a circulating connection through the air ejector and back to the condenser as shown in the illustration, Fig. 26. A three-stage air ejector is shown in Fig. 27 which is self-explanatory.

A float-operated control valve is fitted to the well formed at the bottom of regenerative condensers and should the level of water in the well rise above the normal working level the control valve allows the surplus water to return to the distilled feed-water tanks. On the other hand, if the level falls the control valve allows water to be drawn from the distilled feed-water tanks into the condenser. Fig. 28 shows the type of control valve installed.

In starting up the closed-feed system, the main circulating pumps having been previously started, see that the gauge glass adjacent to the condenser shows that there is ample water in the condenser well before starting the water extraction pump. The circulating valve permitting a flow of water through the ejectors should then be opened, also the ejector drain valves before admitting steam to the air ejector nozzles. The air cocks fitted to the ejector coolers should be opened and having verified that the coolers are free of air they should then be closed. When starting up the main turbines, close the circulating valve; the system then functions automatically. To shut down the system, shut off steam to the ejectors and stop the water extraction pump.



"A", first flow.
"B", second flow.
"C", clear centre passage.
"D", baffle plates.
"E", one of air suction branches.
"F", condensate suction branch.
"G", circulating water inlet.
"H", facing for the float-controlled supplementary feed valve.
"J", circulating water discharge.

FIG. 25.—*Part section of typical Weir regenerative condenser.*

Care and Maintenance.

Having dealt with the construction and the functions of the various parts and units of a modern geared turbine installation, a careful study of which is essential, the subject of "care and maintenance" will now be considered.

Warming up.

The important feature of this operation is that it must be done slowly to prevent distortion and to ensure uniform expansion. The period of time necessary is dependent upon the size of the installation but from four to five hours is a normal time.

Before commencing to "warm up", all drains from steam pipes, steam valves, turbines and air ejectors must be opened and the manœuvring valves and all steam valves on the turbines must be closed.

Verify that the rotors of turbines fitted with axial adjusting gear are run aft to give maximum axial clearance. When warming-up open steam to glands to take chill off turbines and turn rotors at intervals of about 20 minutes for about two hours by the turning gear. The turning gear should then be pinned "out of gear". The main circulating pump should then be started up slowly, then open slightly the circulating valve on the air ejector circulating connection, start up the water extraction pump and open steam to air ejectors to maintain a vacuum of about five inches in the condenser. The sliding feet at the forward end of each turbine should then be examined to see that they are free to expand; it is advisable to give these a few drops of oil daily when at sea. The oil level in gravity and drain tanks should then be checked and both tanks should be tested for water and having verified that the lubricating system is in order, start up the forced lubrication oil pump. See that oil is flowing freely to all bearings, gearing sprayers and turbine adjusting blocks and that the sight-glass on the overflow pipe shows that the gravity tank is overflowing; now ease the main bulkhead stop valve and the ahead manœuvring valve to admit steam to the turbines but not to rotate them. These

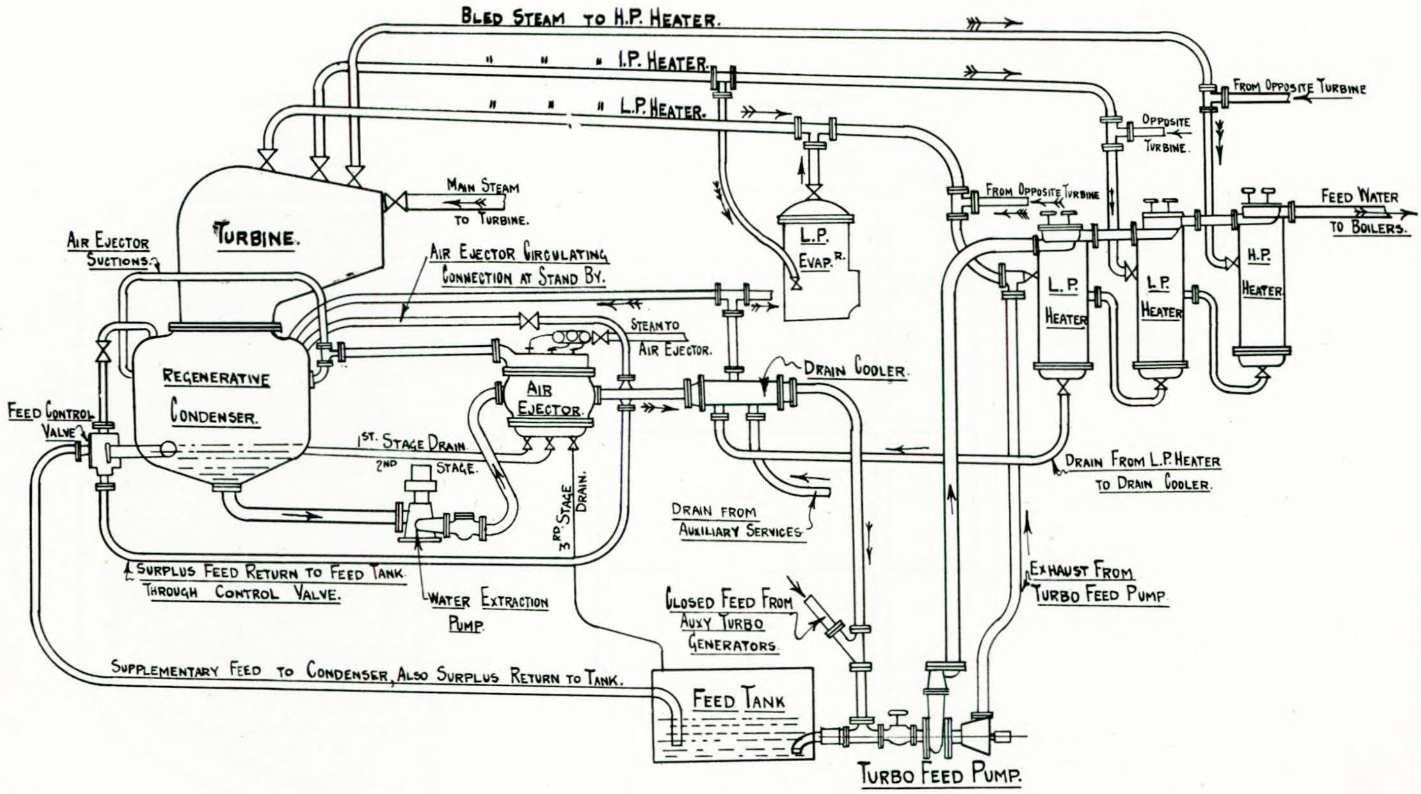


FIG. 26.—*Closed-feed diagram.*

conditions should be maintained for about one hour after which the bulkhead valve and manœuvring valve should be closed and the turbines allowed to "soak" for another one to two hours to permit the rotor and casing to become of uniform temperature throughout. The turbines will now be thoroughly warmed through and everything is ready to take a turn out of the engines. But before doing so ascertain from the bridge that all is clear aft. Having obtained verification that all is clear, increase the vacuum to 25 inches and take a few turns ahead, but before opening the ahead manœuvring valve make sure that the astern valve and double-shut-off valve are closed, otherwise an undue load is put on the astern blading which has been known to cause stripping. Similar precautions must be taken before taking a few turns astern to see that the ahead valve is closed before opening the astern double shut-off valve and astern manœuvring valve.

The ahead and astern manœuvring valves and astern double shut-off valve should now be closed and on receipt of the order to "Stand-by" again check that the drains from regulating valves and on turbines and air ejectors are open and that main circulating pumps, forced-lubrication pumps, oil coolers, water extraction pumps, air ejectors, turbo feed pumps and closed-feed system are working as required.

When underway, after "Full Ahead" has been rung on the telegraph, allow about four to five minutes to elapse and then close turbine drain valves. During the watch examine the level in oil gravity and drain tanks, also see that oil is flowing freely to all gearing sprayers and from the outlets of all bearings and adjusting blocks. Should there be any stoppage of oil flow at any of these sources the turbines must be stopped immediately and the cause ascertained. The magnetic and suction and discharge strainers must be examined from time to time to see that they are in good order. The magnetic strainers will give an indication if any wear of the gearing teeth is taking place. The lubricating oil pressure before and after the discharge strainers will indicate whether the strainer requires cleaning and should the forced-lubrication pump be running faster than usual with a lower discharge pressure the suction strainer should be examined.

In the event of the forced-lubrication pump stopping, the indicating light at the starting platform will warn the engineer of the watch and there is usually time to start the pump, or the stand-by pump, before the emergency bulkhead valves (Fig. 22) close due to the oil pressure being insufficient to support the relay valve. Should the indicator light be unobserved or fail to function, the electric alarm in the gravity tank will warn the engineer that the pump has stopped. However, in modern vessels the forced-lubrication pumps are usually electrically driven and should there be a "black out" owing to the turbo-generator sets coming off the switchboard the lubricating oil pump would stop. In these circumstances the engineer of the watch would immediately shut off steam to turbines; the electric alarm in the gravity tank would not function and in due course the emergency bulkhead valves would close. It will be realised, however, that although steam has been shut off to the turbines they

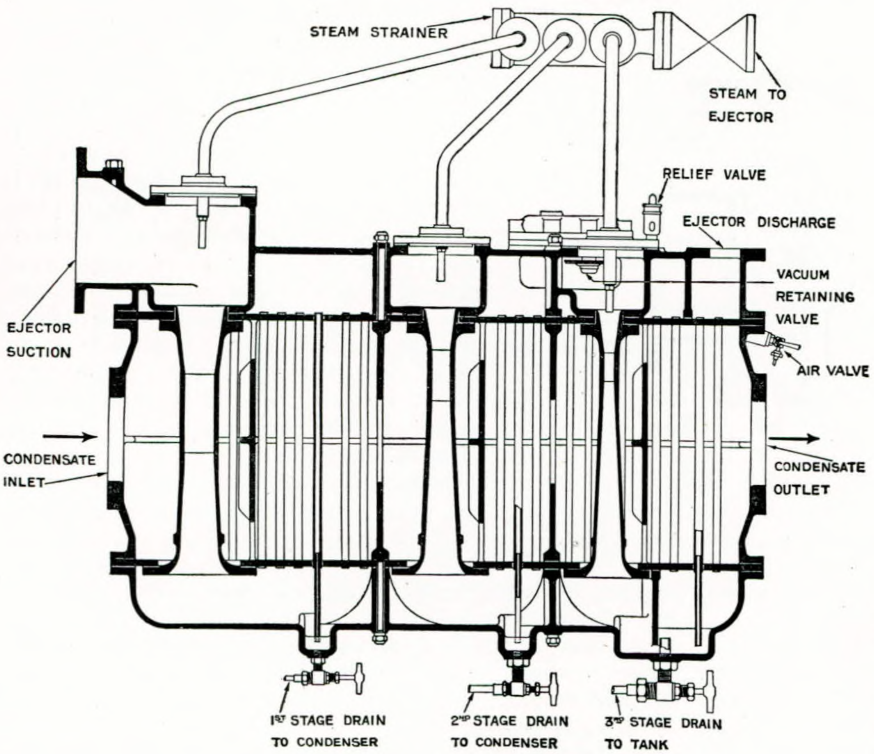


FIG. 27.—*Three-stage air ejector.*

will continue to revolve for some appreciable time due to the momentum of the ship. It is quite possible therefore that during part of this time the gearing and bearings may be running without sufficient oil supply, with consequent damage. To obviate this possibility the astern manœuvring valve and double shut-off should be eased after the ahead valve has been closed, in order to assist in bringing the turbines and gearing to rest. To prevent the generators coming off the board due to a sudden overload, in most modern vessels preferential trips are fitted at the switchboard which cut out the non-essential services and leave emergency lighting, navigation lights, steering gear, electric telegraphs, forced draught fans and all propelling and boiler auxiliaries (where electrically driven) in operation. Nevertheless, "black-outs" have been known to occur and the engineer should be prepared to deal with this emergency.

The fore and aft position of the rotor relative to the casing should be taken by the poker gauge every watch to ascertain if any wear has taken place in the adjusting block.

The following data should be taken each watch and recorded in the engineers' log book :—

Boiler pressure and temperature.

Steam pressures and temperatures at each turbine.

Number of nozzles open.

Throttle valve opening.

First stage pressure.

Bled steam pressures at each bled steam stage.

Kenotometer vacuum reading, and temperature of exhaust to condenser.

Temperature of sea, overboard discharge, extraction pump discharge, feed temperature to each heater, feed temperature to boilers, oil temperatures entering and leaving oil coolers, oil temperature from gear case and leaving each bearing and oil temperature in main thrust block.

Revolution counter and revolutions per minute.

Torsion meter reading giving s.h.p.

Readings of hard fresh-water tanks for boiler feed also distilled boiler feed tanks and make-up feed water used.

Should vessel be fitted with steam or feed recorders, the records must be taken and logged daily.

The recording of the log data mentioned above, enables any differences under similar conditions of running to be immediately detected and remedied.

When " Finished with Engines " is rung on the telegraph, verify that manœuvring valves and the astern double-shut-off valves are closed and close emergency bulkhead valves by means of the hand gear control. Shut turbine nozzle valves, bled steam valves and steam to glands. Shut off steam to air ejectors, stop water extraction pump, then main circulating pump, close main injection and overboard discharge valves, stop forced-lubrication oil pump and shut off oil coolers. Open all turbine drains and drains from manœuvring valves and connections.

In Port.

While in port the oil at the bottom of the drain and gravity tanks should be examined and any sludge or water removed by the means provided. The oil purifier should be run to remove any impurities and to cleanse the oil. Too much stress cannot be laid on the importance of keeping the lubricating oil in good condition as the life of the gearing and bearings depends upon it. A chemical analysis of the lubricating oil should be obtained at the termination of each voyage. It is recommended that the drain tank be emptied and thoroughly cleaned out every six months.

The gear-wheel and pinion teeth should be examined through the access doors on the sides of the gear case and it is recommended that the bottom of the gear case should be cleaned of any sludge or sediment every six to eight weeks. Care must be taken when opening up any part of a turbine installation to see that no foreign material is allowed to fall into the turbine casings or gearing. At terminal ports the steam and oil strainers should be cleaned, the wear down of turbine and gearing bearings recorded by the bridge gauges supplied and the oil clearance of the adjusting blocks checked. On the ter-

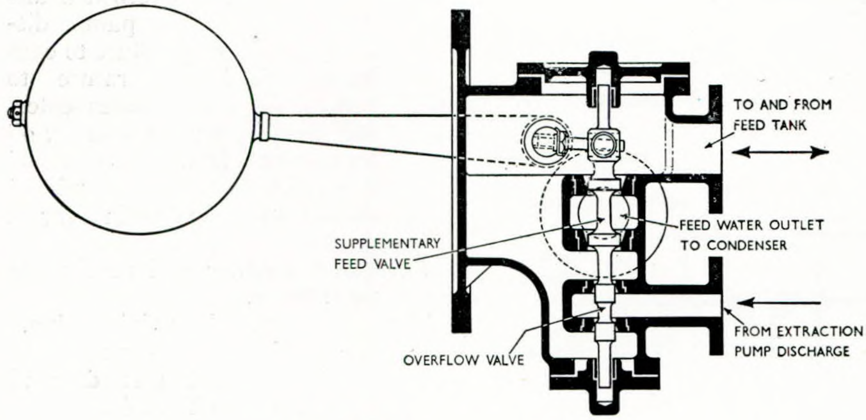


FIG. 28.—*Feed control valve.*

mination of each voyage the fore and aft "slog" of the gears should be taken by means of a clock gauge and carefully recorded and compared with previous readings. Such records are important in giving information regarding the wear that is taken place in the gearing.

When opening up the turbines for internal examination, readings of the wear down of all turbine and gearing bearings, also the dummy clearance and adjusting block oil clearance must be taken and recorded. Verification of these readings must also be recorded after re-assembly.

The wear down of bearings is taken by means of the bridge gauges supplied by the builders. A typical sketch of a bridge gauge is shown in Fig. 29.

The dummy and oil clearances are obtained by the finger piece which is fitted at the steam end of each turbine (see Fig. 29). A dimension called the "finger piece" reading is supplied by the builders, the reading being the distance between the after face of the finger piece and the forward face of a collar formed on the rotor shaft when the rotor is in its correct axial position and the dummy and oil clearances are correct.

In the case of contact dummies as fitted in H.P. and I.P. turbines having end-tightened blading and drawing-up gear (Fig. 6), the dummy clearance is obtained by pulling the rotor forward so that the adjusting block is hard against the forward liners shown in Fig. 13. Gauge and note the finger-piece reading. This should be makers' finger-piece reading minus half the total oil clearance. If reading is the same as given by the builders the dummy clearance is correct. If less or more note the difference, push rotor aft, remove adjusting block keep and fit new forward liners as required to restore designed dummy clearance.

In turbines fitted with adjusting blocks of the fixed type shown in Fig. 14, fit the adjusting gear supplied by the makers and proceed as stated above. If the turbine has radial dummies the readings and adjustments are necessary to ensure that the rotor blading is in its correct axial position in relation to the casing blades.

To obtain the oil clearance for adjusting blocks as fitted to turbines without the drawing-up gear, Fig. 14, fit the adjusting gear supplied by the builders and pull the rotor as far forward as it will come. Gauge and note the finger-piece reading. Next push rotor hard aft and again gauge and note finger-piece reading. The difference is the total oil clearance which is equally divided between the ahead and astern faces of the block. The first reading should be the makers' finger-piece reading minus half the total oil clearance and the second the makers' finger-piece reading plus half the total oil clearance. The makers give the finger-piece reading and the total oil clearance on their drawings, with the rotor in the running position. If the readings obtained above differ from the makers' figure, note the difference, pull rotor forward, remove adjusting block keep and fit after liners as may be required, after which verify that oil clearance is correct.

For adjusting blocks of turbines having end-tightened blading and drawing up gear, Figs. 6 and 13, the oil clearance is obtained by pushing the rotor aft to bring block hard against after liners, noting the finger-piece reading. Since the total oil clearance is equally divided between the ahead and astern faces of the block, in this position the reading should be :—

Makers' finger-piece reading plus designed running-off clearance plus half total designed oil clearance.

Next pull rotor forward to bring block hard against forward liners, gauge and note finger-piece reading, which should be :—

Makers' finger-piece reading minus half total designed oil clearance.

The difference between the first and second readings

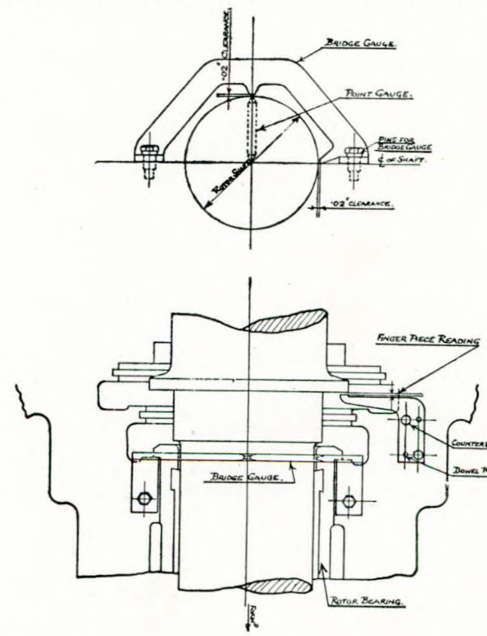


FIG. 29.—*Arrangement of bridge gauge and finger piece.*

equals the running-off clearance plus the total oil clearance. Deducting the running-off clearance which the makers give on the adjusting block drawing, the total oil clearance is immediately determined. If the oil clearance obtained differs from the makers' figure, remove adjusting block keep and fit liners *A* and *B*, Fig. 13, as may be required, after which verify that oil clearance is correct.

Opening up turbines for survey.

When lifting the turbine casings and rotors for survey, first fit the four graduated casing guide columns, two at each end. In turbines fitted with end-tightened blading and drawing-up gear, Fig. 6, push rotor hard aft to give maximum blading and dummy clearance. In turbines not fitted with this gear the adjusting gear supplied by the makers must be fitted, the adjusting block keep removed and the after liners taken out to enable rotor to be pushed aft. Having removed all horizontal joint nuts, fit the starting screws and raise cover off the joint face; then, by means of the lifting blocks, lift cover evenly and when high enough insert the supporting columns.

Before lifting the rotor, remove bearing and adjusting block keeps and fit the rotor guide columns at each end. Lift the rotor evenly with the lifting blocks until it is high enough to be lowered on to the cottars inserted through the guide columns.

It is now possible to examine all casing and rotor blading. See that all binding wires, and where fitted, junction wires, are intact, also blade shrouding. Examine all blading for corrosion and erosion; look for both these effects on the inlet edges of blades, particularly the impulse blading and the later stages of the L.P. turbine blading. Thoroughly clean all blading with a wire brush, after which the casing should be blown out by means of the air compressor plant which in modern vessels also has a vacuum attachment. The latter attachment should then be used to remove all powder or grit from the blading, leaving it in a perfectly clean condition. The blading and casings should then be given a coating of protective compound. Examine the dummy and gland fins. Remove bottom halves of rotor and adjusting block bearings and clean oil wells and passages. Replace bottom half bearings and carefully lower rotor into place. Adjust fore and aft position of rotor with the adjusting gear so that dummy fins are correctly positioned in relation to the grooves, remove cover supporting columns and carefully and evenly lower the cover. Replace nuts and gradually harden them up evenly all round.

Errata—"The Computation of the Stresses in a Propeller Blade Section."

The stresses shown in the curves of Fig. 7 in Mr. S. A. Smith's paper "The Computation of the Stresses in a Propeller Blade Section" published in the April, 1941 issue of the TRANSACTIONS, page 47, should have read as follows:—

Tensile stress at	*A*	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*J*
lb./sq. in.	3,796	5,309	5,864	5,722	5,580	5,439	5,298	4,460	2,706
Compressive stress at	*M*	*N*	*O*	*P*	*Q*	*R*	*S*		
lb./sq. in.	992	4,554	7,502	9,562	9,985	8,461	3,608		

Naval Architecture and Ship Construction (Chapter IX).

The publication of Chapter IX is unavoidably deferred until the next issue of the TRANSACTIONS.

ADDITIONS TO THE LIBRARY.

Presented by the Publishers.

Mechanical Vibrations. By J. P. Den Hartog. McGraw-Hill Publishing Co., Ltd., 2nd edn., 448 pp., 282 illus., 35s. net.

In reviewing the first edition of this book in 1934 we mentioned that while most authors assumed that their readers were already conversant with the fundamental principles of the subject and treated special problems in great mathematical detail, with the result that very little literature on the subject of vibration suitable for students existed, Professor Den Hartog's book was intended for students and was so excellently written that it adequately achieved its purpose.

The book deals with the general principles of vibration, first with simple and then with more complex systems. Following the general treatment of the subject, the problems of multi-cylinder engines and of rotating machinery (turbines, etc.) are dealt with. There are also chapters on self-excited vibration and on systems with non-linear characteristics.

The second edition differs from the first in that a number of errors have been rectified and it has been brought up to date by the inclusion of discussions on the more important advances made in the subject during the last six years. The new matter is concerned primarily with electrical measuring instruments, centrifugal pendulum dampers, aircraft and ship propellers, and automatic balancing machines. The number of problems has been increased by almost fifty per cent., and at the suggestion of many readers a second appendix has been added listing in a comprehensive manner the formulæ that are likely to occur in practice.

It will be seen therefore that most vibration problems which the engineer is likely to meet in practice are treated in this volume, and the student is led to the more difficult material in a simple and straightforward manner. The theories in the book are illustrated throughout by examples of their use in practice. Each chapter contains a large number of questions for the student to work out, the answers to these being given at the end of the volume. A list of symbols—a valuable feature often omitted from technical books—is also included. Vibration is a problem of increasing importance to engineers, and for those who wish to make themselves thoroughly familiar with the subject the author has produced an excellent book.

Works Boiler Plant. By F. J. Matthews, B.Sc. Hutchinson's Scientific & Technical Publications, 184 pp., 88 illus., 10s. 6d. net.

Mr. F. J. Matthews has succeeded in his object of producing a book which can be used as a basis of instructions for foremen; it can also usefully be employed as a refresher by the works engineer. It contains much useful information on the all-important subject of steam raising and many useful hints on the economical working of boilers, a matter of vital importance in these days of war. Very useful and clear information is given on calorific values, clinker troubles, superheating, heat losses and insulating materials. Chapter VII, dealing with feed water treatment, should be found very useful to those who do not enjoy the advantages of boiler feeding with condensate.

The reviewer does not agree that it is good practice to ease safety valves at least once daily, as in his opinion this leads to "feathering" and, ultimately, serious leakage; easing of safety valves once every two weeks should be sufficient, and this should only be done by a skilled operator.

The author should have used stronger expressions in his notes on reducing valves. He states that "a relief valve is desirable between reducing valve and L.P. equipment", "a further safeguard is the provision of a pressure gauge on the L.P. side of the reducing valve", and "an isolating valve between the reducing valve and L.P. equipment is useful".

These three fittings are compulsory under the 1937 Factories Act, see Part II, Para. 30 (a) (b) (c) and (d).

In addition to the safety notes on boiler blow down cocks mention should have been made of the requirements of the above Act in this respect. Para. 29, Sub. Para. (b) stipulates that where a battery of boilers blow down to a common blow-off pipe, the blow-down cocks should be so constructed as to permit the withdrawal of the key only when the cock is in the closed position and that only one key be available for the whole battery.

The book is well produced, has a good index, and the illustrations are clear. Notes and calculations are separate from the text but are on the page opposite to their relevant matter—a very convenient arrangement.

The Boilermaker's Assistant. By J. Courtney and G. C. Malden. The Technical Press Ltd., 108 pp., 102 illus., 4s. 6d. net.

The centre and major portion of this book is given over to setting out and development or "templating", and is well worth reading by any apprentice boilermaker. The opening chapters, however, are devoted to matters which appear to be rather elementary, although it is possible that the formulæ and tables included have proved of value in the earlier editions. The latter portion of the book, dealing with the actual construction of boilers, is disappointing. It could be extended considerably while still remaining a manual for a boilermaker's assistant. The student reader would certainly require to cover much more ground on the subject of boilers and their construction than is given in this book, before he had been more than a few months at his job.

Practical Mechanics and Strength of Materials. By C. W. Leigh, B.S., and J. F. Mangold, B.E., C.E. McGraw-Hill Publishing Co., Ltd., 3rd edn., 498 pp., 441 illus., 21s. net.

Mechanics is the basis of all sections of physics and an attempt to develop its principles in association with one section must interfere with its broad treatment.

The present volume excludes kinetics and deals only with statics. All sections needing Newton's Law and the energy concept are also necessarily excluded, so that the instruction in both mechanics and materials is of limited value to mechanical and marine engineers. The book was written for use in night school classes at the Armour Institute attended by men engaged in offices, foundries, shops and construction companies, and structural engineers would find pleasing features in its pages.

The section dealing with beam deflections omits the usual formal differential equations and uses the more easily understood direct reasoning from the machinery of flexure. The authors state on pages 76 and 77 that owing to transverse deformations due to longitudinal forces, *stresses* will be in the same ratio. This needs explanation.

The book is excellently set out with simple but good illustrations. Practical applications use American standards and empirical formulæ, but their value to British readers is not seriously affected.

Abstracts of the Technical Press

New System of Rustproofing.

A new system for rustproofing metal during production is announced by an American firm. Known as the "corrosion solvent" process, its fundamental purpose is to provide a passive, resistant, iron-phosphate surface over which to apply finish coats at a minimum cost. The process consists of the application of "Corrosol" inhibitor wash followed by a "Metalbond" dip bath to give a heavier iron-phosphate base. The inhibitor wash itself, however, is sufficient for most rustproofing needs. It cleans and conditions in a single operation, removing all foreign matter from the metal and neutralising all rust stimulating agents. At the same time, it converts a microscopic thickness of the metal itself into iron phosphate. If a denser iron-phosphate base is desired, the "Metalbond" treatment is applied. This builds up on the metal a substantial deposit of basic phosphates of iron which will meet any specification calling for an iron-phosphate coating on metal, it is claimed.—*"Iron and Coal Trades Review", Vol. CXLII, No. 3,813, 28th March, 1941, p. 373.*

Swedish Auxiliary Schooner with Reversible-blade Propeller.

The auxiliary motorship "Silva", recently completed by the Karlstads Mek. Verkstad A/B, is a typical example of the class of small cargo carrier now being constructed in Sweden in considerable numbers. She is a steel-built three-masted schooner without topsails, having a length b.p. of about 100ft., a moulded breadth of about 24ft., and a depth of approximately 11ft., with a gross tonnage of 188 tons and a d.w. capacity of 300 tons. The ship has large deep tanks fore and aft, and a continuous double bottom. The ballast tanks hold some 50 tons, while the after D.B. tank holds nine tons of fuel oil. Accommodation for the officers is aft under the poop, while the four members of the crew are berthed forward in a sunken forecastle. The hull is strengthened forward for service in ice by means of closely-spaced frames, and the two cargo holds are served by two large hatches, each provided with a Diesel-driven winch of 1½ tons capacity. The schooner has a sail area of 2,444ft.^{2} and is also equipped with a twin-cylinder 150/160-h.p. Bolinder heavy-oil engine driving a reversible blade propeller at 300 r.p.m. and giving the vessel a service speed of about 9 knots. The engine is started by compressed air without pre-heating, the heat required for starting being supplied by electrically-heated coils in the combustion chamber, with current taken from a 12-volt battery. The reversible-blade propeller is very popular in Sweden as its employment makes it possible to use a simple non-reversing engine without the addition of heavy and complicated reverse gear, these advantages being claimed to outweigh the presence of a complicated mechanism within the stern tube. The propeller blades are adjusted by hydraulic gear, a pump fitted near the engine flywheel compressing oil which acts on a vertical piston, the motion being transmitted to the reversing rods of the propeller blades by a system of links. Manual control of the piston position is provided by a fully-balanced slide and is independent of the speed and power developed by the engine. The slide can be moved into the ahead or astern positions either from the engine room or from the wheelhouse without de-clutching the propeller. The ship's dynamo is driven by an 8-h.p. Bolinder engine which is also connected to a ballast pump. The "Silva's" electrical equipment includes W/T direction-finding apparatus.—*"Lloyd's List and Shipping Gazette", No. 39,406, 26th February, 1941, p. 8.*

New Oil-engine Crosshead Design.

An improved method of assembling and detaching the crossheads of Diesel engines has been evolved and patented by three distinguished engineers associated with Harland and Wolff, Belfast. Referring to the accompanying diagrams, Fig. 1 shows the engine frame and guide dismantled. On the engine frame (A) there is a raised face (B) and a slot (C) which takes the spigot (E) of the slipper guide (D). The spigot (E) is held in the slot by a plate (F, Fig. 2) carried on a stud which is screwed into the frame at the top of the slot. Assuming that there is a cylinder on each side of the frame, two guides (D) are required and these are shown in the cross-sectional view (Fig. 2). Bolt holes (G) are drilled through the frame, and the guides and parts are secured by bolts (U), one plate (F) being sufficient for a pair of guides. Fig. 3 shows the design applied to an opposed-piston engine in which the side rods (K) attached to the upper piston (J) are connected to eccentric rods (L) the two eccentrics being formed on the main shaft immediately alongside the crank webs for the lower piston (H). The ends of the eccentric rods (L) are pivoted to the crosshead shoes (N), which are screwed on the side rods (K) and reciprocate in the guides (D). The clip-plates and assembling bolts are not shown in the diagram. The crosshead is dismantled by first unscrewing the units (O) of the side rods and then removing the plates and bolts which

FIG. 1.

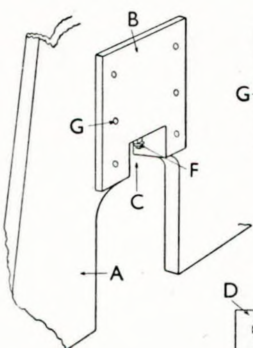
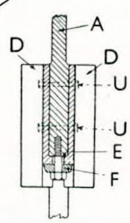


FIG. 2 (*right*).



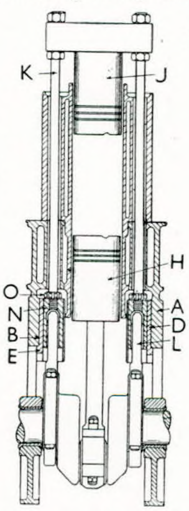
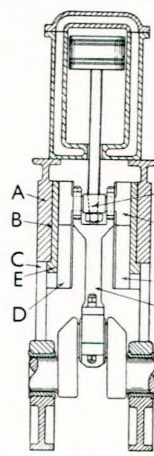


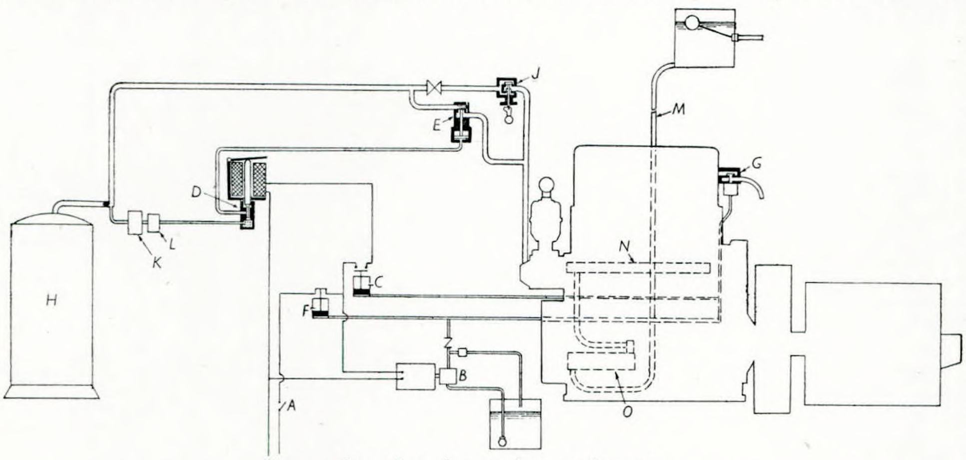
FIG. 3 (*left*).
FIG. 4 (*below*).



hold the crosshead together. The guides (D) complete with the crosshead shoes (N) can then be swung out on the eccentric rods (L) either to the front or the back of the engine, and withdrawn through the usual inspection door. In the case of an ordinary single- or double-acting engine as shown in Fig. 4, the crosshead (S) is detachable. When it has been removed, both guides (D) can be taken away from the frames (A) in the manner already described, and the guides, together with the crosshead block (S) and the shoes (N) can then be swung out either way on the connecting rod (T). The entire crosshead assembly is very compact and the arrangement is applicable to engines with horizontal or inclined cylinders as well as to those of the vertical type.—*"The Motor Ship", Vol. XXI, No. 253, February, 1941, p. 369.*

Remotely Controlled Air-starting Equipment.

The provision of remotely-controlled starting equipment for small Diesel generators presents no difficulties in cases where the energy needed to set the engine in motion can be supplied by an electric motor type of starter, but where the unit has a rated output of over 50 b.h.p. per cylinder, a greater effort is required and the advantages of air starting are obvious. The accompanying diagram shows the arrangement adopted for an emergency generating set consisting of a three-cylinder 165-b.h.p. Diesel engine running at 600 r.p.m., directly coupled to a 100-kW. 230-volt d.c. dynamo, both the engine and generator being made by the English Electric Co., Ltd. The switchboard attendant, when wishing to start up the set, pushes button (A) and keeps it depressed until a light on the switchboard tells him that the



Connections for air-starting equipment.

engine is running. The first event is that the lubricating-oil pump (B), which is electrically driven by a battery-fed motor, is set in motion, and raises the lubricating-oil pressure in the engine system. This pump is parallel with, but quite independent from, the engine-driven unit. When the oil pressure in the system reaches 5lb./in.^{2}, an oil-operated switch (C) closes, connecting the coils of the electro-pneumatic valve (D) to the mains. Air passing through this valve from the receiver (H) is brought down from the 300lb./in.^{2} pressure to 50lb./in.^{2} for this pilot circuit, the reduction being effected by the valve (K), after which, as a safety measure, there is a blow-off valve (L) set to 100lb./in.^{2} From the electro-pneumatic valve the low-pressure air is taken to the automatic starting valve (E), which is opened, thus permitting air at full receiver pressure to pass through to the air-admission valves of the standard type of engine. The unit thereupon starts up and as the generator voltage rises, the circuit-breaker automatic-closing sequence is initiated. The pressure in the lubricating-oil system increases and at 10lb./in.^{2} another oil-operated switch (F) opens, thus cutting off the current passing to the electrically-driven oil pump (B) and the electro-pneumatic valve (D). Thereafter the lubricating-oil pressure is maintained by the engine-driven pump. The discharge valve (G) of the cooling-water system is likewise oil-pressure controlled, and when the set is under way, the valve opens and circulating water flows through it. When the engine is stopped, falling oil pressure gradually allows the water discharge valve to be shut off by spring energy, and the water ceases to flow, but the jackets are kept full in readiness for the next start. The circulating-water system includes a gravity tank feeding through the pipe (M) and the oil cooler (O), to the water inlet manifold of the engine (N). A feature of the starting of the set is the gradual acceleration of the engine effected by a duplex-piston device mounted close to the governor casing. Under the influence of rising lubricating-oil pressure, first one piston and then the other rises, moving a pivoted arm, which is connected by linkage to the speed-control rod of the engine. This action gives increased rack-rod opening with consequent larger fuel injection quantities until the centrifugal governor takes control and stabilizes the 600 r.p.m. rate. Normal hand control of the air-starting system is in no way affected by the addition of this automatic equipment, as all that has to be done is to open a stop-cock between the receiver and the manual-starting valve (J), which is moved to the "start" position, after the hand control for fuel quantity has been appropriately set. In service, the engine is stopped by manual control and barred round to the starting position in readiness for the next start.—*"The Oil Engine", Vol. VIII, No. 94, February, 1941, pp. 276-277.*

Recent Motor Ship and Oil Engine Patents.

MOND NICKEL TANKER CONSTRUCTION

In Fig. 1 is illustrated a system of tanker construction embodying composite metals. The oil cargo is carried in a hold (1). Next to a longitudinal bulkhead (3) there is an expansion trunk (2). At the side of this trunk is a bunker (4) and in the upper deck there is an oil hatch (5). The shell of the hold is formed from composite plates, each consisting of two layers (6, 7). The layer 7 is directly exposed to the

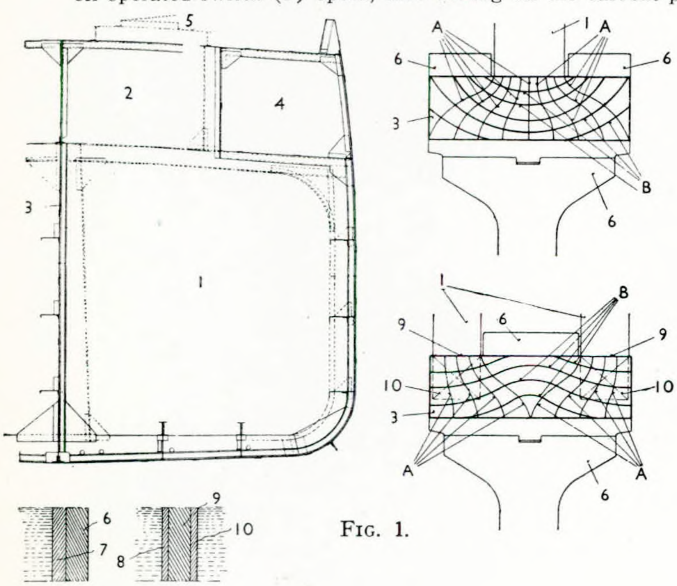


FIG. 1.

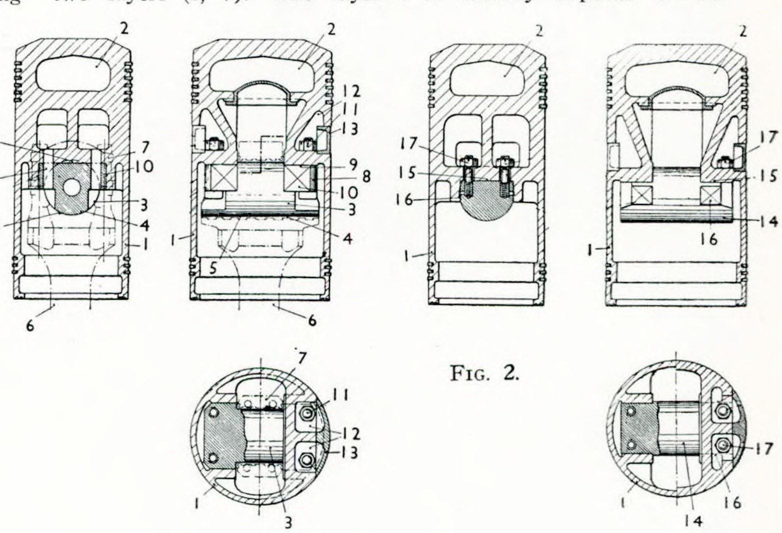


FIG. 2.

corrosive action of the oil cargo and is made of steel more anodic than that of the layer 6, which may be about $\frac{7}{16}$ in. thick, and made from an alloy steel containing about 1 per cent. copper and 2 per cent. nickel, while the layer 7 may be about $\frac{1}{4}$ in. thick and made from plain carbon steel. Another material from which the layer 6 may be made is an alloy steel containing about 5 per cent. nickel. By reversing the potentials of the exposed and foundation layers and by making both of substantial thickness, the life of a tanker may, it is understood, be considerably increased. Such parts as the longitudinal bulkhead (3) are subjected to non-uniform corrosive attack on both sides. Accordingly, they may be made from composite plates consisting of three layers (8, 9, 10), the layers 8 and 10 being made of plain carbon steel and the middle layer 9 of steel, cathodic with respect to both the outer layers. Thus, the middle layer may be about $\frac{7}{16}$ in. thick and of nickel steel, and the outer layers may be about $\frac{1}{8}$ in. thick and of carbon steel. In electro-chemical potential tests in which plain carbon steel was coupled with 1 per cent. copper—2 per cent. nickel steel, the potential difference was about 0.08 volt in a 3 per cent. sodium-chloride solution. Correspondingly, the potential difference when the 5 per cent. nickel steel was coupled with plain carbon steel of the same analysis in the same solution amounted to about 0.11 volt. In general, it is stated to be preferable to maintain a difference of at least about 0.07 volt, and it is unnecessary for practical purposes to exceed about 0.2 volt. The composite plates or other products may be made, for example, by forming a composite ingot and then rolling it into shape.

SULZER PISTON AND CONNECTING ROD ASSEMBLY.

Various diagrams shown in Fig. 2 refer to methods of assembling Sulzer engine pistons and connecting rods In the usual arrangement the centre of the gudgeon pin is secured to the piston (1), while the connecting rod (6) is bifurcated at its end; the ends of the gudgeon pin become more highly heated than the centre, and the heat-flow lines (A) converge towards the centre part where the pin is secured to the piston. The isothermic lines (B) are strongly curved; the curvature of these lines indicates the extent to which the gudgeon pin bearing surfaces will become correspondingly curved. According to the lower heat-flow diagram, the ends of the gudgeon pin (3) are secured to the piston (1), and the connecting rod is provided with a central part which encircles the centre of the pin; the flow of heat, indicated by the lines A, is more favourable, since the heat is transmitted from the surfaces of the gudgeon pin to the piston by a comparative short path. Further, the isothermic lines (B) now extend, mainly, axially of the gudgeon pin. The pin, even when considerable heat is generated, will thus be practically unaffected by curvature. In one of the arrangements the piston is provided with a cooling chamber (2) and a gudgeon pin (3); the surface (4) (remote from the head) is engaged by a bearing surface (5) in the connecting rod (6), which also surrounds the centre part of the gudgeon pin by a bearing (7). The ends of the pin are flange projections (8), which are secured to the body of the piston by bolts (11) and have flat locating surfaces (9, 10). The surfaces (10) which are in good metallic contact with the piston (by reason of accurate fitting of the gudgeon pin) transmit lateral forces between the piston and gudgeon pin, while the locating surfaces (9) transmit the axial forces between the piston and the pin. Owing to the manner of securing the projections (8) to the piston, a relatively large surface area for the discharge of heat from the gudgeon pin to the piston is stated to be obtained. The nuts for the bolts (11) are located in recesses (12) closed by cover plates (13). In the alternative construction illustrated, the piston (1) with its cooling chamber (2) has a gudgeon pin (14) provided at its ends with locating surfaces (15, 16) produced by forming flats on an initially cylindrical pin, so that these surfaces lie nearer the axis of the pin than the cylindrical bearing surface. The locating surfaces co-operate with corresponding parts on the piston, and the ends of the gudgeon pin are secured by studs and nuts (17).

AN EVAPORATOR WITH CENTRIFUGAL SEPARATOR DISCS.

Fig. 3 represents an evaporator with a rotating element. (A) is the housing of the evaporator and (B) is the heating coil. The heating steam enters the coil (B) at (C) and leaves it as condensate at (D). At (E) sea water containing impurities, in

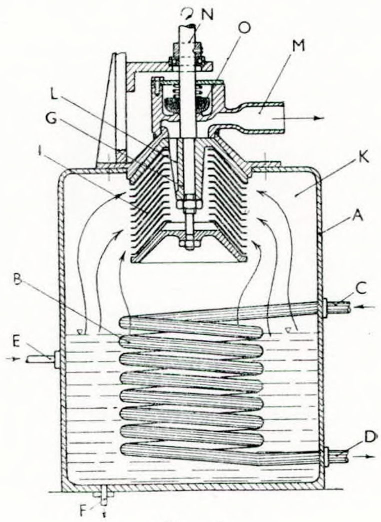


FIG. 3

solution, is fed into the evaporator. The sediment is discharged through the opening (F). A centrifugal separator (G) having a set of conical discs (I) operating at high speed is built in the dome (K) or upper part of the evaporator. The separator is driven by a shaft (N) having a stuffing box (O), and is fixed to the housing (A) of the evaporator in such a manner that the conical discs (I) extend down into the space (K). The vapour being produced moves upwards in the dome and at the circumference it enters the discs of the separator. The water drops, containing salt and dirt particles and entering the discs together with the vapour, are separated by the centrifugal force of the rotating discs, so that only dry pure vapour is discharged through the channel (L) in the interior of the bowl. The vapour then leaves the machine through the nozzle (M) and flows into a condenser.—*"The Motor Ship," Vol. XXI., No. 253, February, 1941, p. 386.*

Ships Cargo-handling Gear.

The paper explains the reasons why cargo-handling gear should receive due consideration in addition to the propelling machinery, when designing a cargo winch to enable the most economic arrangement to be provided for any given service, from the aspects of first cost, running expenses and cost of upkeep. The available types of motive power for cargo handling are briefly described, with their advantages and disadvantages, which result in the choice being in nearly every case narrowed down to that between steam and electric winches. The relative merits of these two types in general are discussed in detail. The various forms of steam and electric winches available are briefly mentioned, together with their outstanding features and their effect on the economy of a ship. Derrick arrangements in general, and heavy-lift derricks in particular, are also referred to, but equipment for handling bulk cargo, such as grain and oil, is not dealt with by the author.—*Paper by L. T. Morton, B.Sc., read at a general meeting of the N.-E. Coast Institution of Engineers and Shipbuilders on the 14th February, 1941.*

Shipbuilding Conditions in Japan.

In view of the shortage of steel and fuel in Japan, Osaka shipbuilders are reverting to primitive conditions by the construction of wooden ships for short-sea trading. It is reported that 70 shipwrights are now employed at the Hamada shipyard on the building of five 380-ton wooden vessels equipped with 180-h.p. Diesel auxiliary engines. Two of these ships are approaching completion and the remaining three should be ready for delivery about March or April. Compared with similar steel-built vessels, the building cost is reduced by half. In order to enforce closer co-operation among shipbuilders, the Japanese Government have established a rigid control of the industry and will henceforth prepare designs each year for the construction of all ships ot over 50 metres (152ft. 6in.) in length and fix the total tonnage to be built during the year. Ships will be classified according to employment and type. The construction of other than standard ships will not be permitted except in special cases, and shipping control bodies will decide upon allotments to shipbuilders and then submit their plans for Government approval. Shipbuilders must decide upon the types of engines to be installed. — *"The Syren", Vol. CLXXVIII, No. 2,321, 19th February, 1941, p. 314.*

Self-servicing Floating Dock for U.S. Navy.

The Todd Shipyards Corporation are to construct a new type of floating dock for the U.S. Navy Department. The dock, which is to cost $2,254,342 and will be completed in about nine months, is intended for the drydocking of destroyers and light cruisers stationed in the Gulf of Mexico and will be moored at Galveston, Texas. The construction will be such as make the dock self-servicing, with two end sections and a centre section. The latter will be capable of drydocking the end sections and these, in their turn, will be able to drydock the centre section. The dock will weigh 4,800 tons and its lifting power will exceed 18,000 tons. The overall length will be 614ft., although this will not limit the length of ships to be drydocked to that figure. The length between the outriggers will be 544ft. and that of the centre section 384ft. The overall width will be 116ft. and the width between the wings 90ft. Besides the self-servicing feature, a novel item in the design calls for a circular bottom which, it is stated, will provide greater buoyancy. When in operation, it is expected that the dock will be able to lift some 30 per cent. more in relation to its own weight than has been possible with older types of floating docks. The dock will be of all-welded construction. — *"The Journal of Commerce", (Shipbuilding and Engineering Edition), No. 35,271, 20th February, 1941, p. 7.*

Pulverised Coal Engines.

The annual report of the Fuel Research Board, which has recently been issued, contains an account of a series of experiments carried out on one of the standard designs of a well-known British firm of Diesel engine manufacturers, the cylinder of the engine being 12in. in diameter, the piston stroke 18¾in. and the designed speed 230 r.p.m. Conversion was effected by the substitution of a coal-dust injector valve and a pre-combustion chamber for the original oil-fuel valve, and the provision of a new valve lever, cam nose-piece and a coal-dust hopper, the original cylinder cover being retained without alteration. By using certain types of chromium-plated liner in conjunction with ordinary cast-iron pistons and rings, the rate of liner wear was reduced to about one-seventieth of that experienced with the ordinary cast-iron liner.—*"Shipbuilding and Shipping Record", Vol. LVII, No. 8, 20th February, 1941, p. 171.*

Powdered Steel.

A powdered metal, "Sinterloy", which can be pressed into practically any desired form and then sintered to produce a dense, homogeneous steel, is being marketed in the United States. Suitable for the production of gears, cams, pump rotors, washers, pins, rivets and splined shafts, it is available in three compositions, having 0·15, 0·40 and 0·80 per cent. carbon, each containing carefully determined amounts of chromium; 1·5 to 3 per cent. nickel may be added where toughness is desired. The first two compositions are suitable for case-hardening by the pack-hardening method. The second composition can be heat-treated to a hardness of Rockwell C 40, while the third composition can be hardened to Rockwell C 50. Tensile strengths of from 80,000 to 120,000lb./in.^{2} have been obtained. — *"Iron & Coal Trades Review", Vol. CXLII, No. 3,808, 21st February, 1941, p. 241.*

A Mercury Propelled Cargo Ship.

The peculiar fitness of the mercury-vapour process for ship propulsion is largely due to the extreme simplicity of generating steam by the condensation of mercury vapour. The author presents a design for a 9,000-h.p. cargo vessel of the type adopted by the U.S. Maritime Commission having a single propeller driven through reduction gears by two turbines, one driven by mercury vapour and the other by steam, with an astern turbine incorporated in the L.P. end of the steam turbine. In addition to the mercury and steam turbines, the proposed design provides for two oil-fired mercury boilers, a mercury condenser, a steam condenser, a superheater and an air preheater. The weight of the mercury propelling plant is estimated at about 216 tons, *i.e.*, about 3½ per cent. less than that of a normal steam-turbine installation of equivalent power. The fuel consumption for all purposes is estimated at 0·443lb./s.h.p.-hr., and the overall thermal efficiency at 31·1 per cent.—*Paper by W. L. E. Emmett, presented at the 1940 meeting of the Society of Naval Architects and Marine Engineers, New York, and reproduced in "Transactions of the Institute of Marine Engineers", Vol. LIII, No. 1, February, 1941, pp. 6-12.*

Preliminary Calculations in Ship Design.

The paper deals with the preliminary calculations appertaining to certain data which should be collected as routine and describes the methods of application of such information to derive the various quantitative and qualitative properties of a speculative ship design. These include dimensions and form, lightweight and deadweight, speed and power, capacity, tonnage, trim, stability, subdivision, strength and vibration. An arrangement of cross-connecting pipes between the P. & S. wing fuel tanks fitted in a recent large liner for the purpose of providing automatic levelling in the event of one or more wing compartments being laid open to the sea is illustrated. Alternative arrangements which would permit cross-flooding at an earlier stage and thus reduce the initial heel after damage are shown in another diagram. The paper concludes with a bibliography and a quantity of tabulated data.—*Paper by E. E. Bustard, read at a general meeting of the N.E. Coast Institution of Engineers and Shipbuilders, on the 28th February, 1941.*

Research at the William Froude Laboratory.

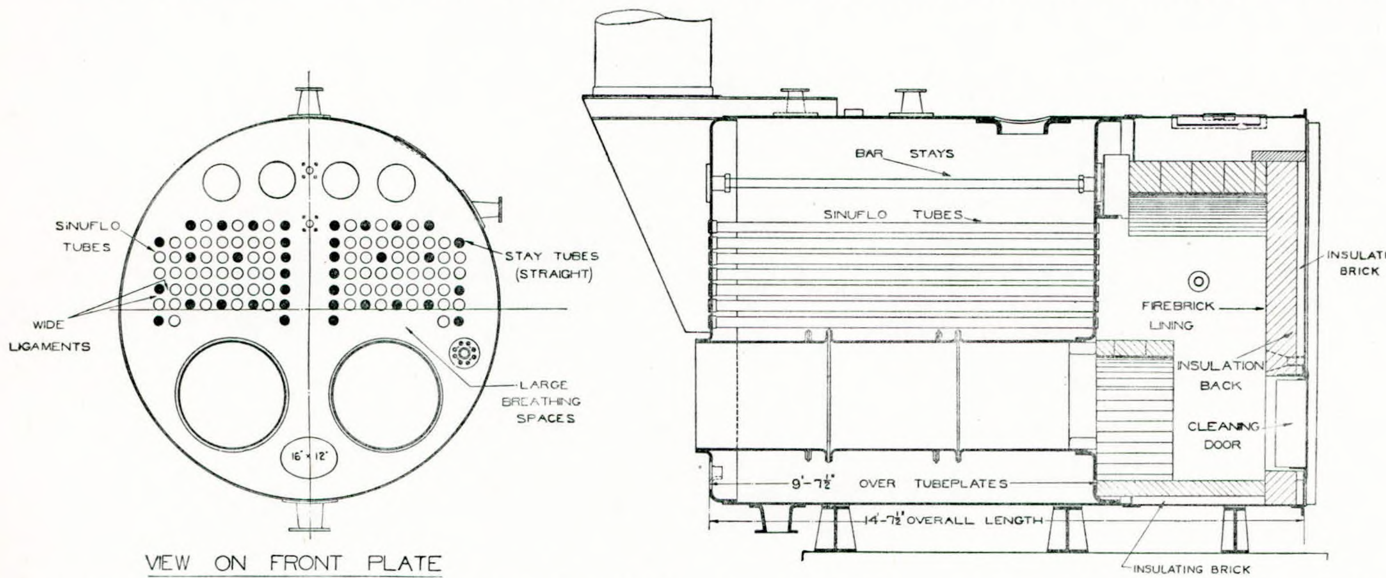
The author gives a brief account of the research work carried out in the Yarrow Tank at the William Froude Laboratory of the N.P.L. during the thirty years of its existence, in connection with the hull form of merchant ships; air resistance; the propulsion of merchant ships; steering; and vibration. The results of these experiments have been published in the shape of reports and papers read before various British and foreign technical societies, and a comprehensive list of these publications, under seven separate headings, is contained in an appendix to the paper.—*Paper by G. S. Baker, O.B.E., D.Sc., "Transactions of the Institute of Marine Engineers", Vol. LIII, No. 1, February, 1941, pp.1-6.*

Pumping Hot Water.

When a reciprocating pump is handling hot water a partial vacuum is formed by the suction and the hot water may vaporise in this partial vacuum, since in a vacuum the boiling point is lowered. Under these conditions vapour will form in the pump which will then operate with a jerky motion and fail to discharge water. This condition is sometimes called "steam bound" and may be remedied by cooling the water going to the pump. Playing cold water from a hose on the pump barrel and suction pipe may also help. Such a condition is unlikely to occur where the pump is under a head of several feet, as in such circumstances the vacuum created by the suction is quickly followed by the entering water and there is no vapour space formed where boiling may take place. When the water level is below the pump a vapour space or pocket can be formed even in the suction pipe. A pump can lift water as long as there is a definite supply of solid water to fill the suction, but when the water is very hot it may vaporise in the pump under the influence of the vacuum created by the suction. When vapour pockets form there is no longer a solid flow of water, and the vapour will get into the pump and cause the latter to lose its suction. This does not happen, however, unless the temperature of the water at the pump suction is somewhere near the vaporisation point, say at about 190° F.—*"Marine Engineering and Shipping Review", Vol. XLVI, No. 2, February, 1941, p. 94.*

"Sinuflo" Economic Boiler for Natural or Medium Draught.

The makers of the Cochran induced-draguht type of "Sinuflo" economic boiler have now introduced two other models intended for operation with natural and medium draught respectively. Both these new types embody, as far as possible, the features of the standard induced-draught type, and are fitted with "Sinuflo" fire tubes, which, owing to their increased heat-transmission efficiency, have proved so successful in operation. Both boilers are extremely accessible for the purpose of cleaning, scaling, and general examination of the water side, and each includes a very ample refractory-lined and insulated combustion chamber to ensure that any unburnt gases reaching it are completely consumed. When fitted with suitable stokers these boilers are stated to be practically smokeless. Ample breathing spaces have been provided in order to relieve the boiler shell of any undue stresses. The fire tubes are widely pitched to assist water circulation, and are considerably fewer in number than in most other boilers of the economic type. As the boiler is of "single-pass" design tube sweeping is stated to be only about half that involved in the usual types of similar boilers. Neither of these new type boilers is intended to displace the well-known Cochran type of vertical boiler, but it is recognised that there are many instances in which, for one reason or another, the latter may not be the most suitable for the site or other circumstances, and it is claimed that in such cases the natural—or medium—draught "Sinuflo" boiler should prove a satisfactory substitue.—*"Boiler House Review," Vol. 54, No. 8, February, 1941, p. 271.*



Heat Losses in Diesel Engines.

Although the percentage of heat lost varies in different types of Diesel engines, the following amounts represent what can be expected between one-half and full power:—

Useful Work.—32 to 35 per cent. This represents engine efficiency and does not vary to any extent over a considerable variation in engine power.

Cooling Water Losses.—27 to 30 per cent. This is about 90 per cent. of the useful work.

Exhaust and Radiation Losses.—26 to 27 per cent.

Mechanical Losses.—8·5 to 14 per cent. This is represented mainly as heat lost due to friction and is largely indicated by the amount of heat imparted to the lubricating oil. It can be reduced by the employment of ball or roller bearings where practicable, as well as by the cooling of the hot lubricating oil by some agent requiring heating.

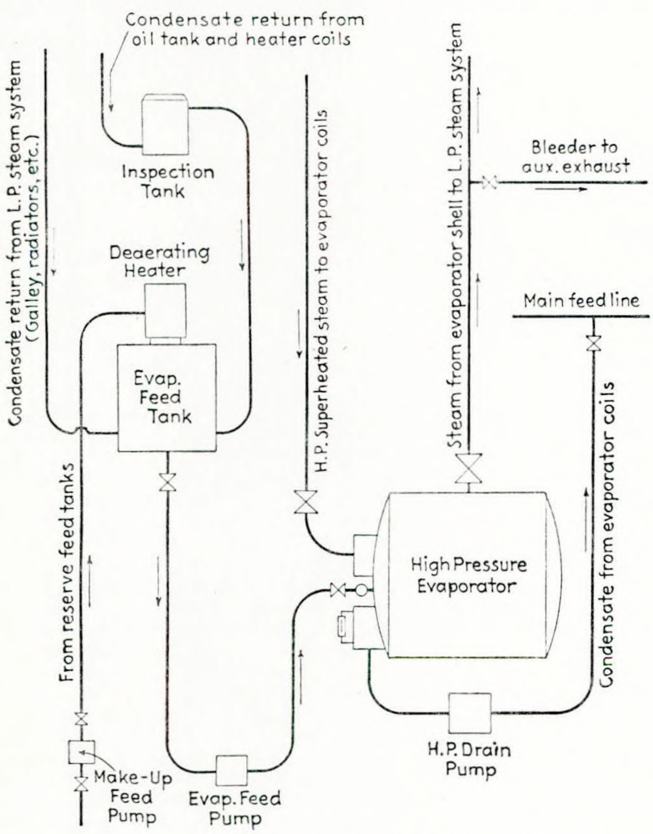
The loss of heat in the cooling water presents some possibility of recovery if it can be utilised for preheating something else usefully, but although it is sometimes applied to heating systems or hot water requirements in industrial installations, the possibilities on board ship are more restricted. Exhaust losses can to a considerable extent be recovered in waste-heat boilers, and as such losses are relatively high, even a 50 per cent. recovery is equivalent to a saving of 12 per cent. The B.Th.U. per b.h.p.-hr. for Diesel engines is from 7,400 to 8,200. Although the Diesel engine uses only about 35 per cent. of the heat of the fuel, it is the most efficient heat engine known at the present time. The binary mercury-steam system gives a fuel economy intermediate between that of the Diesel engine and the best steam installations.—*"Marine Engineering and Shipping Review", Vol. XLVI, No. 2, February, 1941, pp. 94 and 96.*

Couplings for Emergency Pipe Connections.

The U.S. firm manufacturing the Rolagrip pipe connections are now marketing them for marine engineering purposes. These couplings afford a simple and reliable means for joining plain or bevelled-end pipes without any preparation of the pipe ends. A Rolagrip coupling consists of two housing halves, two belts and an oil- and flame-resisting gasket. The housings contain ribbed, cadmium-plated, case-hardened steel rollers, and are made of corrosion-resisting copper-bearing malleable cast iron. The plain pipe end is firmly gripped by the ribbed rollers which imbed themselves in the circumference of the pipe and the recesses of the coupling housing. These recesses are so designed and proportioned as to give play to the normal forces of expansion and contraction, but when the desirable limits of pipe movement are reached the gripping rollers are, by greater constriction of the recesses, caused to oppose separation of the pipe ends. The advantages claimed for the Rolagrip coupling are that valuable time and expense can be saved by the use of a pipe connection which can be quickly applied without any previous pipe-end preparation and that it thereby eliminates the necessity of protecting such pipe ends from damage. It also eliminates the need for frequent trips to the machine shop to correct prefabrication errors, because the lengths of pipe and bends can be cut on the job. In cases where it is feasible to use thin, plain-end piping, substantial savings in cost and transport charges can be effected. The radial deflection of 5° to 7° available with Rolagrip couplings saves an immense amount of trouble and expense in the preparation of templates and offsets in piping layout. This ability to provide axial and radial deflection compensates for pipe movement due to "working" of the ship and pipe expansion and contraction, without the expense, in many cases, of installing special fittings. Bulkhead stress can be relieved if the pipes are connected by Rolagrip couplings, as any axial movement is absorbed by the latter, each coupling in the line taking up its share of the movement, thus preventing the pipe movement from being concentrated at one point. An important use of Rolagrip couplings is in connection with emergency pipe repairs on board ship, as a damaged section of piping can be cut out and a temporary length quickly substituted with the aid of Rolagrip couplings.—*"Marine Engineering and Shipping Review", Vol. XLVI, No. 2, February, 1941, p. 84.*

Contaminated Steam Systems.

Contaminated steam systems are frequently employed in connection with H.P. steam installations in which distilled water is used for make-up feed purposes, and generally serve a dual purpose: that of providing distilled water for the high-pressure system and that of providing steam at a lower pressure for the "contaminated steam system", or low-pressure system as it is more usually termed. The evaporator coils are supplied with steam from the main boilers, but in some installations a small oil-fired Scotch boiler is used as an evaporator. The distilled water is obtained by bleeding some of the steam from the evaporator shell into the auxiliary exhaust system from which it eventually passes into the main feed tank. The water level in the latter is controlled by the bleeder valve; any excess water in the feed tank may be pumped to the distilled-water storage tanks. The L.P. system provides steam for the galley, radiators, oil-fuel heaters, oil-fuel tank heating coils, salt-water evaporator coils and other purposes where there is a possibility of contamination by oil, salt water or other impurities leaking into the system. Returns from the oil-fuel tank heating system are trapped and then led through an inspection tank before being returned to the evaporator feed tank. The other returns are trapped, led through a drain cooler and then to the evaporator feed tank. The cooler



Diagrammatic sketch of contaminated steam system.

is usually necessary, as the temperature of the returning condensate is often high enough to make the feed tank boil. A feed pump is provided for feeding the evaporator with water from the feed tank. To compensate for the steam used in making distilled water and for the losses in the L.P. system a make-up feed tank is provided, which is supplied with water from the reserve feed tanks, discharging it into the evaporator feed tank after it has passed through a deaerating heater. When H.P. steam is used in the evaporator coils the condensate is removed by a high-pressure drain pump which discharges directly into the main feed system. The general arrangement of a typical contaminated steam system is shown in the accompanying sketch. —*R. H. Jacobs, "Marine Engineering and Shipping Review", Vol. XLVI, No. 2, February, 1941, p. 88.*

American Motorship "Julius H. Barnes".

The twin-screw, oil-engined cargo vessel "Julius H. Barnes", recently built for the Erie and St. Lawrence Corporation, N.Y., by the Charleston Shipbuilding and Dry Dock Company, Charleston, S.C., is the largest ship ever constructed by that firm and also the first vessel ever built for the Great Lakes and coastwise shipping trade under the auspices of the U.S. Maritime Commission. She is intended for service on the Great Lakes and the Erie Canal during the summer months and will be used for the carriage of bulk cargoes, without trans-shipment, from Lake ports as far as the Carribean ports of S. America at other times. The ship is of all-welded construction and resembles a tanker in appearance, the propelling machinery being aft. The main dimensions are an overall length of 290ft., a moulded beam of 42ft. and a moulded depth of 18ft., with a register of 1,680 tons. The d.w. capacity is 3,250 tons and the displacement 4,096 tons at a maximum draught of 13½ft. There are five cargo holds with a total bale capacity of 111,000 cu. ft. They are entirely free of stanchions and are primarily designed for the carriage of sugar, sulphur, phosphate, soda, ore, iron, steel and similar bulk cargoes. No cargo-handling gear or winches are provided, loading and discharging being carried out by dock facilities. Quarters for the engineer officers and crew are arranged aft on a deck above the engine room, while those of the captain and deck officers are on a sunken deck below the wheelhouse and bridge amidships. Accommodation is provided for a crew of 25, and an unusual feature of the ship is that both groups of quarters are completely air-conditioned. On the port side of the midship living spaces is a compartment containing the CO_2 fire-extinguishing system, while a corresponding compartment on the starboard side houses the heating and sanitary equipment. The propelling machinery consists of two sets of 5-cylinder Fairbanks-Morse Diesel engines, each rated at 450 h.p. at 360 r.p.m., driving twin propellers through direct-drive shafts. The engines are designed for fresh-water cooling and drive their own air compressors and oil and water pumps. The stern-tube bearings are of a novel pattern, being non-metallic and made of cotton duck impregnated under pressure with a phenolic resin, the bearing being encased in a bronze shell. It is claimed that these bearings will outwear metallic bearings, require no lubrication and are impervious to electrolytic action. Fuel oil to the amount of 128 tons is carried in tanks located in the E.R. wings. The whole of the auxiliary machinery is electrically operated, the deck equipment consisting of an anchor windlass, capstan and mooring winches, with Sperry electric steering gear. Electric current is furnished by two F.M. Diesel-generator sets, with a 60-cell, 114-amp.-hr. battery floated on the line. One of these sets is a 60-kW. unit driven by a 6-cylr., 4-stroke engine and the other is a 40-kW. unit driven by a similar 4-cylr. engine.—*"Marine Engineering and Shipping Review", Vol. XLVI, No. 2, February, 1941, pp. 54-56 and 58.*

Maiden Voyage of American Motorship "Sea Witch".

The American Pioneer Line's 6,240-ton C-2 type motorship "Sea Witch" recently returned to New York from her five months' maiden voyage to Far Eastern and Australian waters, in the course of which she covered 31,180 sea miles at an average speed of 15·007 knots. The vessel's propelling machinery, rated at 6,000 s.h.p., consists of two sets of 9-cylr. 2-stroke Nordberg

Diesel engines driving a single propeller through hydraulic couplings and reduction gears, while auxiliary power is furnished by two 300-kW. Diesel generator sets. Although the ship called at 23 different ports during the voyage, the total time spent in harbour was remarkably small, the longest stop in port being five days at Sydney. The performance of the ship's propelling machinery is considered to be highly satisfactory and no serious defects whatever developed during the voyage. On one occasion it was found necessary to stop one engine at a time for a total of 11 hours to clean the nozzles of the injection valves, but this trouble was subsequently eliminated by advancing the fuel-valve cams to give approximately 2° earlier opening of the valves. The stoppage of the engines had little effect on the vessel's average speed owing to the flexibility of the two-engine reduction-gear, hydraulic-coupling drive. Either engine could be cut out at any time by emptying its coupling and proceeding with the other engine. The only difficulty experienced with the main engines was the breaking of piston rings which occurred during the return trip, and steps have been taken by the engine builders to remedy this. Both the main and auxiliary engines were run on low-grade fuel oil ("Bunker C") throughout the voyage, the average daily consumption for all purposes, while at sea, being approximately 20·4 tons (or 0·42 barrels per mile). A matter to be considered in this connection is that the mean effective pressures and engine revolutions indicated that the propeller was not entirely satisfactory and it is probable that the fitting of a more suitably dimensioned propeller will improve both the speed and fuel consumption of the ship, although they are very good now. After arrival at New York the main and auxiliary machinery was opened up for examination by the engine builders and everything was found to be in good order with the exception of some carbon in the piston ring slots of the generator engine cylinders. Some rings were worn enough to warrant renewal. The desirability of minor changes in some of the engine equipment and ship's fittings has been pointed out by the vessel's engineers; the capacity of the auxiliary condenser is insufficient, the pneumercator pipes lack protection where they pass through the decks, there is inadequate protection of valves and pipes against corrosion, and similar items characteristic of a new ship, but nothing that interferes seriously with the operation of the vessel. The "Sea Witch" only spent 11 days at New York before leaving on her second voyage.—*"Motorship and Diesel Boating", Vol. XXVI, No. 2, February, 1941, pp. 116-118.*

Safety Rules in Shipyards.

The *Report of the Chief Inspector of Factories for 1939*, includes a chapter on safety which states that although yards in which shipbuilding and repairs are carried on were working to full capacity during the year, and, since September, under lighting conditions of undoubted difficulty, the regulations were in general well observed. An explosion which occurred during the use of an oxy-acetylene burner on the outside of a shell plate close to an untested oil fuel tank, caused one death and much material damage. This directs attention to the importance of strict observance of the required preliminary test; such tanks may contain petroleum oil with a lower flash point than is expected. An interesting device employed in one shipyard where tankers are repaired, is a fitting, similar to road crossing lights, situated at the ship's gangway. If the ship is "gas-free" a green light is shown; if she is entirely unfit for the use of naked lights, fire or hot rivets, a red light is shown; when oil vapour is encountered in some of the tanks only, a yellow light is displayed together with the injunction "Please see the certificate". Experience has shown the danger of using compound ropes of fibre and wire for the suspension of stages. Corrosion of the wires is particularly likely to occur when such ropes are exposed to the action of sea water and it is difficult to detect because of the outer covering of fibre.—*"The Journal of Commerce" (Shipbuilding and Engineering Edition), No. 35,277, 27th February, 1941, p. 2.*

Extension Spray Guns.

The use of the conventional type of spray gun is more or less limited to convenient situations, and if awkward corners of walls or bulkheads have to be covered this usually calls for the erection of staging. Similarly, if small interiors such as tubes or casings have to be treated, difficulty is experienced in covering the surface to a distance of more than a few inches. To overcome these difficulties, a series of extension spray guns is now being produced by a firm in this country. It includes a long-reach gun available in lengths of 3ft., 5ft., and 6ft. 6in., weighing 3¼, 3¾, and 4lb. respectively. The gun is of robust construction and easy to handle, the air-control valve being in the handle and the material supply being regulated by a needle valve in the spray head. Both actions are controlled by a single hand-grip lever. Another type in the series is intended for coating the interior of pipes with bitumastic solution or for painting pipework on electrical transformers, tubular boilers, radiators, etc. This gun is designed for use with a pressure paint container.—*"Shipbuilding and Shipping Record", Vol. LVII, No. 9, 27th February, 1941, p. 195.*

Detachable-blade Screws.

Propellers with bronze blades bolted to a separate boss have lately fallen out of favour, as it is claimed that integral cast or solid screws, having somewhat smaller bosses, are more efficient. Some of the early bosses of built-up propellers were undoubtedly rather clumsy, but modern design should enable a compact and well-streamlined boss to be produced, and it is probable that the shape is of greater importance than the actual diameter within certain limits. This may account for specific statements of the advantage to be gained by deliberately adopting a large boss, alone or in conjunction with other stern features. The built-up propeller has one advantage for high-speed work in that it lends itself to dynamic balancing, which is often regarded as impracticable in the case of marine screws, provided each blade is balanced separately against a master before assembly. Greater manufacturing tolerances on pitch are feasible in view of the possibility of adjustment in place, and a single blade design may be standardised for a given diameter over a wide range of pitch. Wartime advantages lie in the wider manufacturing facilities and labour available for making blades alone, and the fact that the most readily available material may be chosen for the boss, which need not be of bronze. — *"Shipbuilding and Shipping Record", Vol. LVII, No. 9, 27th February, 1941, p. 195.*

The "Arc Torch".

A new electric welding device, known as the "Arc Torch", is described in a recent issue of *"Steel"*. The apparatus, it is said, greatly enlarges the scope of electric arc-welding equipment, the arc torch giving an extremely hot, easily controlled flame whose chief characteristic is lack of pressure. Thus molten metal is not forced away, but can be flowed where wanted. It is particularly valuable for welding aluminium and its alloys, for welding and brazing cast iron, and for brazing all types of steel.—*"Iron and Coal Trades Review", Vol. CXLII, No. 3,812, 21st March, 1941, p. 348.*

A Possible Danger with Blank Flanges.

Blank flanges are much used to seal the ends of steam pipes. Often, the sealing is done as a temporary measure whilst certain alterations to the plant are carried out, and there is a possible danger here in the fact that any convenient piece of flat plate may be used for making the flange, without due consideration being given to the important question of whether the plate is quite strong enough to resist the pressure. It should be realised that in the case of a large pipe under considerable pressure, the force tending to burst the flange may be several tons. Thus, a pipe 12in. diameter has a sectional area of 113in.², and if the steam pressure is 200lb./in.², the total pressure acting on the flange will be 22,600lb., or approximately 10 tons. It should also be realised that the flat form of plate is the weakest of all forms for resisting pressure, the hemispherical being the strongest. Hence, there may be real danger of explosion if a flat plate is chosen at random to serve as a flange for sealing the end of a large pipe. In an actual case, where a 10in. pipe carrying superheated steam at a pressure of 250lb/in.² had to be blanked whilst additional plant was being installed, a piece of ¾in. hard brass plate was taken from the stores, and made into the required flange. The flange, which should have been approximately twice the thickness it was, failed, and as the consequence, three men lost their lives, and one was injured. In their book entitled *"Steam Boiler Construction"*, the National Boiler & General Insurance Co. Ltd. give the following rule for determining the working pressure which may be safely imposed on the flat ends of cylindrical vessels up to 20in. diameter and 1in. thick, for any given thickness within the limit stated :—

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

where P = working pressure, in lb./in.^{2},
C = a constant, 300,
t = thickness of plate, in sixteenths of an inch,
A = area of flat plate subject to pressure, in sq. ins.

The rule applies to mild steel having a tensile strength of from 24 to 28 tons/in.^{2}. When the pressure is known, the required thickness of plate can be determined by the formula,

$$t = \sqrt{\frac{P \times A}{C}}.$$

which is easily derived from the original. This useful rule is given because many engineers have difficulty in dealing with the strength of flat plates under pressure, the subject being complex if dealt with from first principles. The flange bolts, as well as the flange itself, must, of course, be of ample strength. — *E. Ingham, "Boiler House Review", Vol. 54, No. 9, March, 1941, pp. 292 and 294.*

Flexible Couplings for Internal-combustion Engines.

It is not always sufficiently understood that most "flexible" couplings have two distinct functions, which may in certain instances seriously interfere with each other's efficiency. This is especially true of the flexible couplings applied to a great number of internal-combustion engines. The two functions of a "flexible" coupling are linear flexibility and torsional resilience, and it would be a good thing to adopt a name like "flexible-resilient" coupling in place of "flexible" coupling when referring to those couplings whose principal function is not to make allowance for deviation from perfect alignment in the shafts coupled. There are several kinds of "flexible" coupling in quite common use, the best known being the belt, pin and buffer, fibre-disc, and spring-loaded types. As the resilience of the first three types is negligible they might be considered as flexible couplings proper, whilst the spring-loaded type might well be distinguished under the name "flexible-resilient coupling. This is the type of coupling most frequently used with internal-combustion engines in the vast majority of directly-coupled sets where the engine drives a generator or compressor. As is known, both are mounted on a single bedplate, the coupling being installed on account of its torsional resilience and not because of its linear flexibility. The spring-loaded type of coupling is illustrated diagrammatically in Fig. 1. There are many individual designs, but they all depend upon the same fundamental principles.

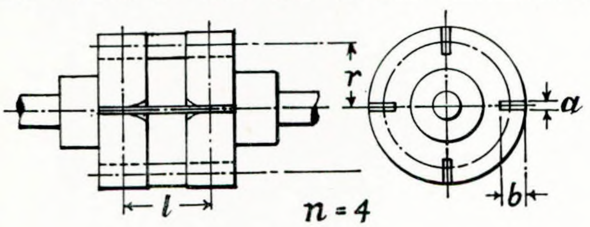


FIG. 1.

When a torque is applied to one half the springs are deflected until they exert an effort sufficient to overcome the load resistance, and the two bosses of the coupling move relatively to each other to the extent θ (radians) as shown in Fig. 2. The stiffness or torsional rigidity of the coupling is then :—

$$\frac{T}{\theta} = \frac{12\,E\,I\,r^2\,n}{l^3} \qquad \ldots \ldots \ldots \ldots \ldots (1)$$

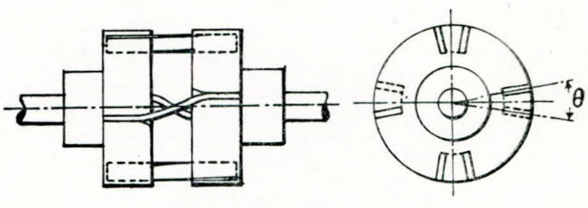


FIG. 2.

where T is the torque, E is the modulus of elasticity of the springs, and I is the moment of area of the spring section in inches^4 units. When the drive is running, the bosses maintain the same relative displacement so long as the torque is constant, and θ then represents a phase difference or lag between the two shafts. But when either the driving resistance varies then the deflection of the springs varies and θ changes accordingly. This must happen continually with an internal-combustion engine drive on account of the explosion impulses. The effect of a flexible-resilient coupling in these circumstances is similar to that of a flywheel in reducing the amplitude of the cyclical torque and speed variations. More exactly, the effect is the same as that of a length of steel shafting, and in fact the equivalent length of shafting for a coupling can be found *for any given value of* θ by equating their torsional rigidities to get the expression

$$L = \frac{\pi}{384}\,\frac{d^4\,l^3\,G}{E\,I\,r^2\,n} \qquad \ldots \ldots \ldots \ldots \ldots (2)$$

when L is the equivalent length of shaft, d is the diameter of equivalent shaft, and G is the modulus of rigidity for shaft steel. This effect of the coupling is very valuable in sets where a particularly steady drive is needed, as in the case of d.c. lighting sets. This same resilience property, of course, acts in response to load fluctuations and reduces the stresses they set up in the engine. But a spring-loaded "flexible-resilient" coupling does more than this.

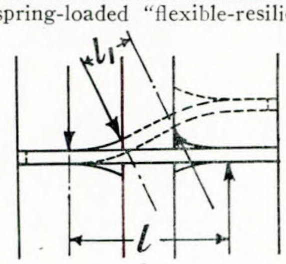


FIG. 3.

In Fig. 3 the springs and their slots in the coupling bosses are shown to a larger scale. The spring slots are usually flared out to a special shape for about half their inner lengths. The effect of this is to cause a reduction in the free length of the springs as the torque increases, down to a minimum length of l_1, which is only slightly greater than the clearance between the bosses. The maximum or skin stress in the springs is given by :—

$$f = 3\,\frac{T\,l}{a^2\,b\,r\,n} \qquad \ldots \ldots \ldots \ldots \ldots (3)$$

and therefore this shortening of the free length protects the springs from overstressing when high torques are transmitted. By designing the curve of the slot suitably, the torsional rigidity of the coupling can be made to increase as a known function of the increase in torque. The slot curves are usually based on the deflection curvature of the springs. So far the use of a flexible-resilient coupling as a "de-tuner" for critical speeds has not been mentioned. All modern internal-combustion engines are, of course, carefully designed so that their normal range of speed will be well below a major critical speed, and this is nearly always also true of the driven machine. Coupling together the two machines, however, often affects the elastic-inertia factors of the system as a whole in such a way that the critical speeds for the set may be considerably reduced; and the danger may arise of running through criticals. If it is found that a major critical falls close to the normal working speed then, in the writer's opinion, the design of the set should be altered, if at all possible, in some effective manner to avoid this, as no "flexible-resilient" coupling or damper should be called upon to act on a permanent critical speed unless absolutely unavoidable. The way in which the coupling acts is to break up possible resonances between the impulse period of the engine and the natural periods of the set on either side of the coupling.

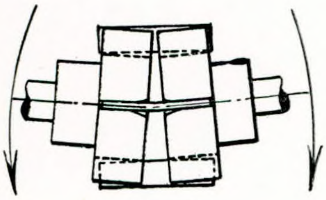


FIG. 4.

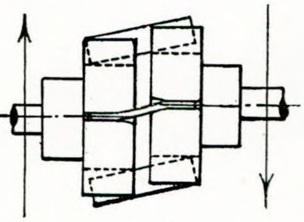


FIG. 5.

If a critical speed has to be run through, then as soon as the revolution speed enters the danger region elastic vibration stresses commence to develop, but the effect of the resilience of the coupling is to check the amplitude of these stresses transmitted across the coupling, and the alteration in the torsional rigidity breaks up the resonant tendency. This de-tuning effect can be understood quickly if it is remembered that the coupling is equivalent to a piece of steel shafting the effective length of which varies in a pre-determined manner with the instantaneous torque transmitted. As soon as an increase of torque is produced by incipient resonant vibration then the natural period of the system is altered by a change in the stiffness of the coupling. If the coupling is used to allow for shaft mis-alignments then the exaggerated effect is shown diagrammatically in Figs. 4 and 5 which separate angular from radial mis-alignment, although, of course, the two kinds are often combined. It is clear that before any torque is applied the springs must be distorted, and that when the coupling rotates an alternating spring deflection from zero to maximum and back occurs twice per revolution. This initial deflection means also that the special shape of the slot flarings is wasted, making the behaviour of the coupling as a vibration damper or de-tuner uncertain and less efficient. The pre-deflection of the springs produces an initial skin stress of considerable magnitude, and results either in danger of over-stressing the springs at the maximum deflection (with increased rate of fatigue), or else in a serious reduction in the rating of the coupling. When "flexible-resilient" couplings of this type run out of correct alignment, therefore, it is necessary to compensate for the consequent pre-stressing of the springs by using larger sizes than would otherwise be necessary. The continuous sliding movement of the springs in their slots not only causes more rapid wear, with the probability of "backlash" developing and frequent spring replacements, but it also causes an entirely wasteful loss of energy and a rise of temperature which may cause lubrication trouble. When this type of "flexible-resilient" coupling runs out of alignment there must be reactions to the deflecting forces or moments, as indicated by the arrows in Figs. 4 and 5. These reactions set up additional rhythmical stresses and pressures in the adjacent shafts and bearings; and besides increasing the dangers of fatigue, excessive wear, and lubrication interference, they may result in the introduction of harmonic torque variations by the very coupling whose function should be to damp them. A troublesome end-thrust also may be set up. All these factors should be considered when a "flexible-resilient" coupling is used under conditions of mis-alignment when its principal function is required to be that of preventing dangerous or destructive stresses from critical-speed torsional vibrations. Some designs of coupling aim at meeting the difficulties mentioned, usually by special spring design, but sometimes by articulating the coupling and taking all mis-alignment in a part of the coupling separate from the spring-carriers. Whenever unavoidable mis-alignment is anticipated in the conditions of a de-tuning coupling one of these special couplings should be used, or else provision must be made by selecting a coupling sufficiently larger than the normal size for the duty. Two important conclusions should be emphasized bearing upon the use of "flexible-resilient" couplings as vibration dampers: firstly, shaft alignments should be as accurate as possible, and, secondly, full elastic-inertia data should be supplied to the coupling designer in addition to the usual information about rating and speed.—*J. Leigh, B.Sc., "Gas and Oil Power," Vol. XXXVI., No. 426, March, 1941, pp. 59-60.*

Shallow-draught Suction Dredgers for U.S. War Department.

The Pusey and Jones Corporation, Wilmington, Del. have received a $3,000,000 order for the construction of two twin-screw Diesel-electric hopper dredgers for the U.S. War Department. The dredgers are to be of a new, fast, shallow-draught type, 216ft. in length and capable of carrying 700 cu. yds. of material without drawing more than 12 feet, at a speed of 13 knots. In general appearance the vessels will resemble well-proportioned tankers, with four hoppers amidships and accommodation for 12 officers and 40 men in deckhouses forward and aft. A single 16-in. electrically-driven pump will be installed forward, connected to drag pipes at each side of the ship, while the engine and motor room will be located aft. The main machinery installation will comprise two 650-b.h.p. Diesel-driven generators so connected that the power output may either be divided between the 350-h.p. dredge pump motor and the geared 475-s.h.p. direct-current propulsion motors, or applied entirely to the latter, as dredging routine requires. The total output of the main generators will be 1,000 kW. and will suffice to give the ship a speed of 16 knots in light condition. The propulsion motors will be controlled directly from the bridge. Auxiliary power will be furnished by a 50-kW. generator and an additional generator with an output of 10 kW. will be provided for emergency use. The two 40-h.p. drag-pipe winch motors will be controlled by regulating the fields of 35-kW. generators driven by a common 75-h.p. motor. There will be eight hopper doors in the ship's bottom, opened and closed by eight hydraulic rams at the weather deck level, operated by a pressure system constantly maintained at 300lb./in.². The first dredger is due for delivery in December, 1941 and the second one six months later. — *"Marine Engineering and Shipping Review", Vol. XLVII, No. 2, February, 1941, p. 71.*

Malmö-Copenhagen Train Ferry.

The Swedish State Railways are reported to be contemplating the construction of a new train ferry for service between Malmö and Copenhagen, to replace the present vessel which was built over 40 years ago. The new ship will have five decks and an overall length of 308ft. 8in., a breadth of 52ft. 6in. and a depth (to boat deck) of 45ft. 3in., the maximum draught being 13ft. 5in. and the d.w. capacity 595 tons. The vessel will be able to carry about 1,300 first- and third-class passengers in addition to railway carriages, or about 1,800 passengers without railway carriages. The train deck will have two tracks of a total length of over 534ft., the rails being let into the deck, which will thus be free of obstructions to enable motor vehicles to be carried. The ship will have a raking icebreaker-type stem, a single funnel, and a short signal mast on the wheelhouse. The propelling machinery is to consist of two Diesel engines developing a total of about 4,500 s.h.p. for a service speed of 15 knots.—*"The Syren", Vol. CLXXVIII, No. 2,323, 5th March, 1941, p. 386.*

Rules for the Measurement of Register Tonnages.

The principle underlying the regulations for the measurement of ships is that the *net register tonnage* of the ship shall represent the tonnage of her *freight* earning spaces. With this purpose in view all *enclosed* spaces situated above the top of the floors or ceiling if fitted thereon, and within the limits of the inside surfaces of the frames or the inside surfaces of the sparring if fitted to the frames, are measured at various transverse sections of the ship. The position of these sections being determined by the length of the ship, measured on the *tonnage deck,* which is the second deck in the case of two, three or more decked ships and the upper deck in a single-deck ship. The tonnage *below* the tonnage deck is first measured, and is usually referred to as the under-deck tonnage. Then all

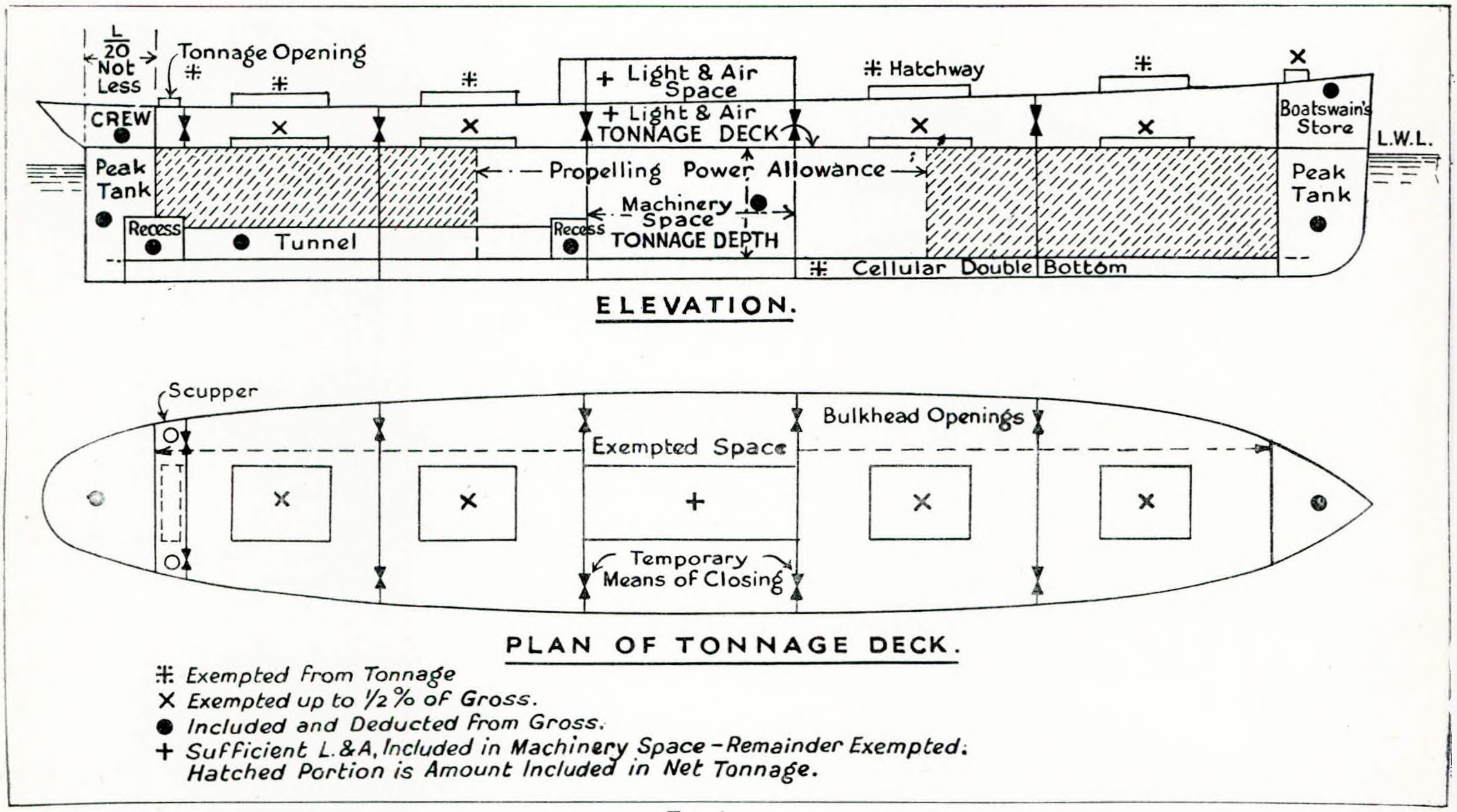


FIG. 1.

enclosed spaces situated *above* the tonnage deck are measured and added to the tonnage below the tonnage deck, the total being the registered gross tonnage. To obtain the net registered tonnage certain spaces are measured and deducted from the gross tonnage. Before proceeding it is well to understand the difference between *deducted* and *exempted* spaces. The former must first be measured and included in the gross tonnage before they can be deducted for net tonnage. The latter are not included in the gross tonnage, but they are measured and particulars inserted in the Certificate of Register, which are helpful to the Dock Authorities if cargo is found to be carried in an exempted space. In this respect it is sometimes confusing to the Authorities, when assessing the net tonnage upon which dues are based, to find that cargo is carried in double-bottom spaces situated below the top of the floors of the ship, particulars of which do not appear on the ship's register. The Authorities are quite at liberty to add such spaces to the net register tonnage for the assessment of their dues. The important spaces which are recognized by the regulations for deduction from the gross tonnage of screw steamers are :—

(*a*) Machinery spaces. A deduction of 32 per cent. of the gross tonnage is permitted, called the propelling power allowance, if the tonnage of these spaces as measured is not more than 13 and is less than 20 per cent. of the gross tonnage. If the tonnage of the measured spaces is outside this limit the allowance is the tonnage of the spaces as measured with an addition of 75 per cent., subject to certain restrictions.

The additional allowance in excess of the actual measurement is assumed to compensate for the fuel spaces which are not measured.

(*b*) Master's and crew spaces.
(*c*) Water ballast spaces and cofferdams.
(*d*) Boatswain's stores.
(*e*) Donkey engine and boiler spaces.
(*f*) Chart room.
(*g*) Spaces used exclusively for the working of the helm, capstan and anchor gear, if situated below the upper deck.
(*h*) Wireless installation.
(*i*) Pump rooms (if avaliable for pumping out bilges).
(*j*) Lamp rooms.

The difference between deducted and exempted spaces has already been explained. Superstructures situated on or above the upper deck, such as a poop, bridge or forecastle, or a combination of such erections, if they have one or more openings in their sides or ends, and these openings are not fitted with *permanent means of closing*, are not included in the gross tonnage. Such openings are usually fitted with weather boards in riveted channels or portable steel plates with hook bolts which do not pass through the bulkhead plating. When the upper deck is covered completely by a superstructure and there is an opening in the superstructure deck of certain limiting size and position without permanent means of closing, and the openings in the bulkheads between the upper and superstructure decks are also fitted with temporary means of closing; subject also to certain methods of draining the space under the tonnage opening being observed, the 'tween deck space is not included in the gross tonnage. Other spaces exempted from gross tonnage are :—

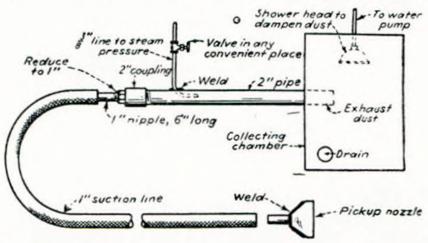
(*a*) Approved spaces erected on the upper deck for the shelter of deck passengers.
(*b*) Certain light and air spaces situated over the machinery space.
(*c*) The donkey boiler if situated above the upper deck and not connected with the main propelling machinery.
(*d*) Hatchways up to one-half per cent. of the gross tonnage.
(*e*) Spaces used exclusively for the working of the helm, capstan and anchor gear, if situated above the upper deck.

(*f*) Companionways, skylights and domes.
(*g*) Galleys, condenser space and bakeries, if situated above the upper deck.
(*h*) Water closets and bath rooms for the use of officers and crew, if situated above the upper deck.
(*i*) Double bottom spaces, if used exclusively for water ballast.

When the gross tonnage has been measured and the deductions allowed the result is the *net registered tonnage*. Fig. 1 shows the spaces which are exempted, deducted and included in the gross tonnage of a ship fitted with a complete superstructure having a tonnage opening in the superstructure deck.—*E. W. Blocksidge, " The Dock and Harbour Authority," Vol. XXI, No. 245, March, 1941, pp. 98-101.*

Steam-operated Vacuum Cleaners.

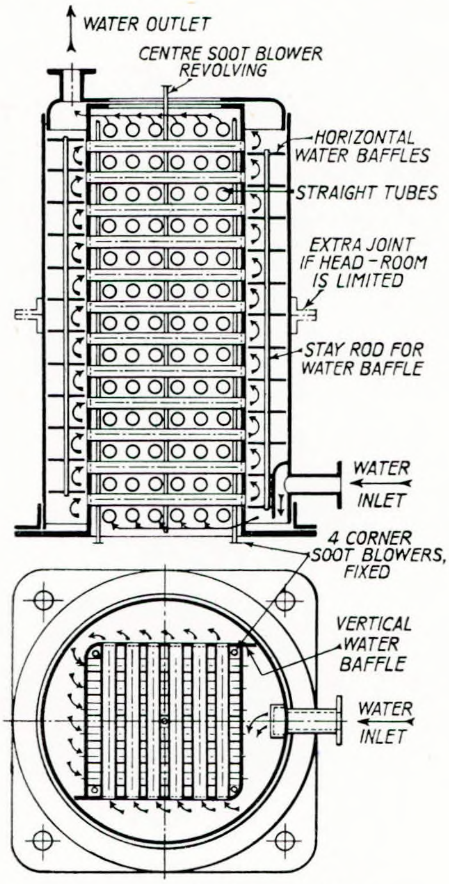
The accompanying illustration shows one of several arrangements suggested in *Power* for removing soot, dust and dirt from the tops of boilers. Suction is provided by steam flow through a $\frac{3}{8}$-in. pipe, which discharges the collected dust into any convenient box or chamber, where it is precipitated by a water spray and thus washed down the drain. A reducing coupling at the other end of the 2-in. pipe carries a nipple for the 1-in. suction hose which terminates in a nozzle forged from flattened pipe. This apparatus can be made largely if not entirely from scrap material. An alternative arrangement uses a commercial steam ejector, drawing dirt through a perforated



stub-pipe in a receiver which traps coarse material brought in by the vacuum hose; the ejector discharge, carrying the fine dust, goes to the boiler blowdown tank. Yet another modification of the same principle uses seamless electric conduit and fittings for the vacuum line from the ejector to conveniently situated cocks for the air hose. A cap closing the end of the fixed vacuum pipe is reduced to $\frac{1}{32}$ in. at the centre to form a safety disc guarding against the remote possibility of steam pressure building up in the conduit. The ejector nozzle is fitted on a threaded pipe to permit of adjustment for maximum suction, and the discharge is into the breeching or ashpit, according to preference.—*" The Power Works Engineer," Vol. XXXVI., No. 417, March, 1941, p. 69.*

Criss-Cross Exhaust Boiler.

The accompanying diagrammatic drawings show the arrangement of a novel form of silencer-heater fitted to the exhaust of a Diesel engine in a large commercial establishment in London. The construction employs only straight water tubes, and is claimed to possess three excellent features, *viz.* :—a very compact arrangement of the heating surface, a counterflow relation of the paths of the exhaust gas and the water, and an impingement at right angles of the gas flow upon the water tubes throughout. Furthermore, by the adoption of the same methods as in the well-known thimble-tube boilers, the water sides of the tubes and baffles are made readily accessible. The exhaust gases pass downwards through the spaces between the tubes, while the water to be heated enters at the bottom, first passing through one set of tubes in parallel and then around the outside of the tube stack, so that the water passes into the next highest row of tubes at right angles. This flow proceeds up the entire boiler, and thus the sets of tubes are in series, producing the same effect as long tubes. Long tubes, however, could not be arranged so that the gas would impinge upon them at all points at right angles, and this layout is obviously far more compact, in addition to which it secures the best conditions for heat transfer. Access to the water passages and tube interiors is obtained by lifting the outer casing, which may be



Arrangement of criss-cross boiler.

in two parts if the headroom is limited. Cleaning on the gas side is provided for by five soot blowers, four of these being stationary and arranged one at each corner for the cleaning of the long portions of the several clearances between adjacent rows of tubes, with a central blower, which can be rotated, for cleaning the central portion. The construction of the tube stack, water baffles and external casing is very simple, and the tubes are expanded into the tube plates at both ends. As the expanded ends are water-cooled, no trouble should be experienced from expansion phenomena. Welding is largely employed in the fabrication of the heater, including the baffle assembly.—*" The Power and Works Engineer," Vol. XXXVI., No. 417, March, 1941, p. 67.*

Propeller Nomenclature.

Some of the terms used for describing screw geometry and performance are apt to be confusing, partly from the wealth of synonyms and in other cases from the nicety of the distinctions implied. Sweepback, skewback, throwround and (in aeronautical parlance) lag are all more or less equivalent, as are rake, tilt and (aeronautically again) dihedral. The matter is complicated by the fact that the terms are not even geometrically exclusive, since a blade having a linear throwback is in practice barely distinguishable from one having rake only, except in the manner of drawing. Slip terminology is notoriously obscure, and even an expert might hesitate to distinguish between some of the more closely-related terms such as true and real slip, while the apparent slip of the engineer's log-book is not altogether what it seems. Delivered and developed horse-power have the doubtful advantage of being summarised as direct horse-power, coupled with the fact that some naval architects use the one term where others would use the other. Some still speak of lift and drift, though the majority regard the latter as long obsolete in favour of drag. Even the most recent terms are not free from ambiguity, since humming and singing must be regarded as one and the same thing.—*"Shipbuilding and Shipping Record", Vol. LVII, No. 10, 6th March, 1941, p. 219.*

Breakage of Flanges and Bolts.

The flanges of cast-iron pipes are brittle, and may break suddenly if subjected to severe bending stresses, which can be imposed in a number of ways. Thus, if the bolts at one part of the flange are tightened up whilst those at the other side are left slack for the moment, the flange will be slightly tilted, and will therefore be subjected to a heavy bending moment when the other bolts are tightened. All the nuts should, of course, be screwed up a little at a time until the flange faces are brought to bear more or less uniformly and hard together; but because thoughtless or unskilled men screw up the nuts carelessly, flanges are frequently strained and often broken. Another way in which flanges may be subjected to severe bending stresses is by driving wedges between the flange faces for the purpose of removing an old, or fitting a new joint ring. In addition to the bending stresses there are shock stresses arising from the driving in of the wedges, so that the risk of breakage is considerable, and in any case the flange faces may be damaged. If wedges must be used they should be of slow taper, and sufficiently thin at the entering end to prevent burring the edges of the flanges when driven in. They should be spaced evenly round the flanges and driven in gently, first one a little, then another, and so on. A chisel is a convenient and often ready-at-hand form of wedge, and chisels are therefore commonly used for parting flange faces. But their use for this purpose is to be condemned, because the taper is too keen, and they are a clumsy form of wedge. Fig. 1 shows

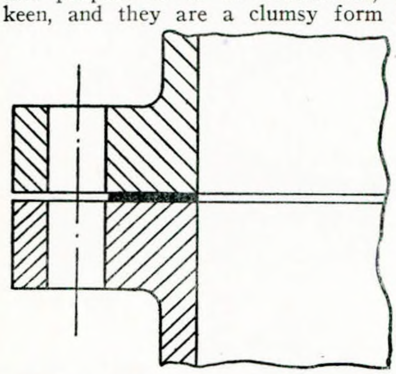


Fig. 1. Section of flanged pipe joint fitted with a flat joint ring.

a flanged pipe joint fitted with a flat joint ring which covers the flange faces only from the internal bore of the pipe to the nearer edges of the bolt holes, instead of extending for the full width of the flange faces; in other words, the rings are cut and arranged in such a way that the outer portions of the flanges are not supported by the packing; hence, these portions are exposed to a heavy bending action when the bolts are tightened up. If the flanges are of wrought steel, and of ample strength, this bending action may not be seriously objectionable, but otherwise it may lead to failure. The following remarks bearing on this question are taken from a Board of Trade Report dealing with the explosion of a cast-iron stop-valve chest. "The practice of fitting jointing material only within the bolt circle in the case of steam-pipe joints is not good practice, as it is liable to result in the adjacent necks of valve chests and pipe ends being stressed more severely than is desirable, especially where cast iron is the material used for these parts." Rings which lie wholly within the bolt circle are commonly used in preference to rings of the full width of the flange faces because they offer certain advantages over the full width rings. Thus, they are more simple to make, as they require no bolt holes, they are more readily removed and fitted than are the full-width rings, and they may be more effective in making a tight joint when the bolts are tightened up because they are subjected to greater pressure per unit area than are the wide rings. Flanges may be exposed to a certain amount of bending if the pipes are out of alignment, or if they sag much owing to inadequate support. Wrought steel flanges are secured to the pipes in various ways, *i.e.,* by expanding, screwing, welding, and riveting. With these flanges inferior workmanship may account for leakage troubles, and incur considerable risk of the flanges breaking away from the pipes, which means a more or less serious explosion. This has happened in a large number of instances, often with fatal consequences. Since the possibility of bad work cannot be ignored, it behoves steam users to maintain pipe mains in good alignment and support, and to take care that the pipes are not exposed to unwonted stresses from expansive or vibratory movements. Flange bolts may be broken by excessive tightening up of the nuts, a practice which is all too commonly resorted to when joints leak. In most cases of leakage the proper remedy is to re-make the joints, and certainly no attempt should be made to stop the leakage by tightening the nuts whilst the pipes are under pressure. It cannot be too often stressed that tampering with steam joints under pressure is dangerous, and has been responsible for many fatal accidents, for which reason it should be most strictly prohibited. With small bolts, considerable care and judgment are necessary to avoid over-stressing the bolts by tightening the nuts, or even breaking them; if the bolts are stressed beyond the elastic limit of the material they will gradually stretch, so that they cannot be expected to keep the joint tight; and in all probability they will fail sooner or later. A possible danger with flange bolts is that of the section being seriously reduced by the erosive action of leaking steam. In one case bolts $\frac{7}{8}$ in. diameter were reduced by this action to less than $\frac{5}{16}$ in., and they failed whilst the fireman was in the act of tightening the nuts to prevent the leakage, with the result that the flange gave way, and the fireman was immediately killed. Evidently, occasional inspection of the bolts is desirable, and any which are found reduced, or badly worn about the screw threads, should be replaced. Sometimes the bolts are too short, so that the nuts have only a partial holding on the threads. In one instanec where a boiler

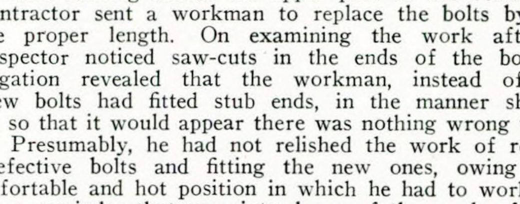


Fig. 2. Arrangement of bolt fitted with stub end.

inspector discovered a number of such bolts in the flanges of a steam main running across the upper part of the boiler house, the contractor sent a workman to replace the bolts by others of the proper length. On examining the work afterwards the inspector noticed saw-cuts in the ends of the bolts, and investigation revealed that the workman, instead of fitting the new bolts had fitted stub ends, in the manner shown in Fig. 2, so that it would appear there was nothing wrong with the bolts. Presumably, he had not relished the work of removing the defective bolts and fitting the new ones, owing to the uncomfortable and hot position in which he had to work. This case is a reminder that superintendence of the work of making flanged pipe joints may be well worth while.—*E. Ingham, "Boiler House Review," Vol. 54, No. 9, March, 1941, pp. 292 and 294.*

New Oil-retrieving Device.

A simple and effective device for the recovery of lubricating oil from ships' bilge water has been invented by two Swedish marine engineers. Tests extending over a period of four months were carried out with the new device in three Swedish tugs and are reported to have resulted in a saving of lubricating oil of no less than 75 per cent. The Swedish authorities are recommending the employment of this waste-oil separator, which is now being manufactured on a commercial scale..—*"The Journal of Commerce" (Shipbuilding and Engineering Edition), No. 35,283, 6th March, 1941, p. 7.*

A New Steam Engine and Boiler.

The Knox engine in its present state is non-condensing, but is readily adaptable for use with a condenser, thereby utilising to good advantage the additional energy obtainable from the exhaust. The engine is a high-speed compound uniflow unit of the steam reverse type, designed to run at 1,000 r.p.m. with steam at a pressure of 700lb./in.^{2} and 750° F. temperature. It is double expanding, with one H.P. and two L.P. cylinders. The inlet valves are of the piston type, actuated by an auxiliary crankshaft, which is gear-driven from the main crankshaft, while exhaust from all cylinders takes place through uniflow ports, with auxiliary exhaust provided by the piston valves. The outstanding feature of the engine is the valve gear, by which a number of useful functions are performed without introducing more moving parts than the minimum number required for a simple non-reversing engine with the same number of cylinders. The most important of these functions are the provision of expansion and the automatic increase of torque at starting and low speeds, without additional mechanism. These special features give the engine a steam consumption of 14·6lb./b.h.p.-hr. (non-condensing), and an engine efficiency of 53·3 per cent., obtained under test when the engine was loaded to 70 h.p., or about 75 per cent. of its capacity. The general construction of the engine is shown in Fig. 2, but the steam passages from the valves to the cylinders are omitted. The H.P. cylinder, 3¾ in. by 4½ in., is located between the two L.P. cylinders, 4½in. by 4½in., as shown in the cross section of the cylinder block (Fig. 3). Steam enters through a control valve, which serves both as a throttle and as a distributing valve, the direction of rotation of the engine being changed from forward to reverse by simply pushing the control lever to its forward or reverse position. When running forward the steam passes from the control valve (marked V in Figs. 2 and 3) to the inner edges of the H.P. valve *a* (Fig. 3). This valve, as well as the other two valves *b* and *c*, is a piston valve, but all three differ from the usual type of piston valve in two important respects: first, the valve has a rolling motion as well as a reciprocating motion; thus the path of any point on the valve is an ellipse. The second special feature is the serrated edges, which can be seen in Fig. 5. The combination of this unusual motion with the shape of the valve edges gives rise to the characteristics explained below. The H.P. cylinder discharges into a receiver space formed by the unused volume of the cylinder block casting. The discharge takes place through exhaust ports in the middle of the cylinder walls, which are uncovered by the piston in the conventional uniflow manner. Auxiliary exhaust, which is necessary to keep compression down to a reasonable amount, is provided by the piston valves mentioned previously. Each of these valves is connected to two cylinders, the outer edges to one cylinder, the inner edges to another. If the direction of rotation of the engine is such that the outer edges of a valve admit steam to a cylinder the inner edges of the same valve will open during the exhaust stroke of the other cylinder, thus providing auxiliary exhaust for the second cylinder without any additional parts being required. Running forward, steam from the receiver is directed by the outer edges of the valve *b* into the L.P. cylinder A (Fig. 3) and by the inner edges of the valve *c* into the L.P. cylinder B. These two cylinders exhaust through central ports similar to those in the H.P. cylinder, the steam being carried through a manifold to the exhaust line. Auxiliary exhaust is provided for cylinder B by the inner edges of valve *b* and for cylinder C by the outer edges of valve *c*. A unique feature of this engine, as compared with other steam reverse engines, lies in the fact that each valve supplies a particular cylinder when the engine is running forward, but a different cylinder when the engine is reversed. When the control valve is put to the reverse position steam is no longer admitted to the inner edges of valve *a*, but goes to the outer edges of valve *c*. In the same way steam passses from the receiver to the outer edges of valve *a* and the inner edges of valve *b*, whence it passes to cylinders A and B respectively. The opposite edges of the valves also act as auxiliary exhaust valves.

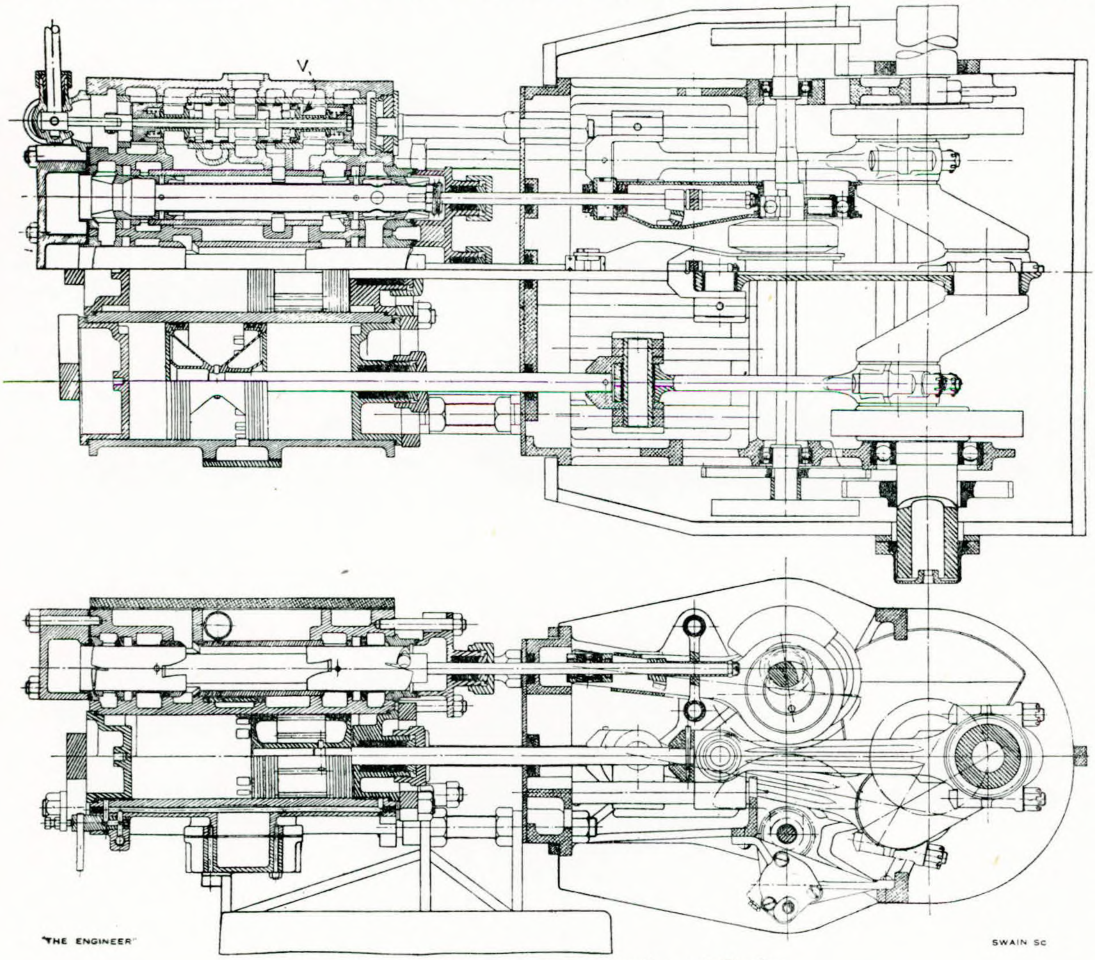


FIG. 2. *Plan and Cross Section of Engine.*

It is by this means that a cut-off of 30 to 40 per cent. (depending upon the slope of the serrated edges) can be obtained in all the cylinders of this engine. Earlier steam reverse engines necessarily had 100 per cent. cut-off, and consequently their steam rate was several times as high as that of the Knox engine. The usefulness of the serrated valve edges, in combination with the rolling motion of the valves, lies in the fact that the effective cut-off can be lengthened, thus increasing the torque of the engine. The rolling motion is imparted by a simple arrangement which combines a segmental bevel pinion on the valve stem with a segment of bevel gear on the eccentric strap. The vertical component of the motion of the strap caused by the rotation of the eccentric causes the valve stem to turn first in one direction and then in the other. Both stem and strap have the same reciprocating motion, and so the gear segments are always in mesh. The serrated forward admission edges of the valves shown in Fig. 5 exemplify one of the advantages which may be obtained by this combined reciprocating and oscillating movement of the valves. In an engine where high power-to-weight ratio is desired, as in the present engine, this construction allows the admission period to be lengthened for a given angle of

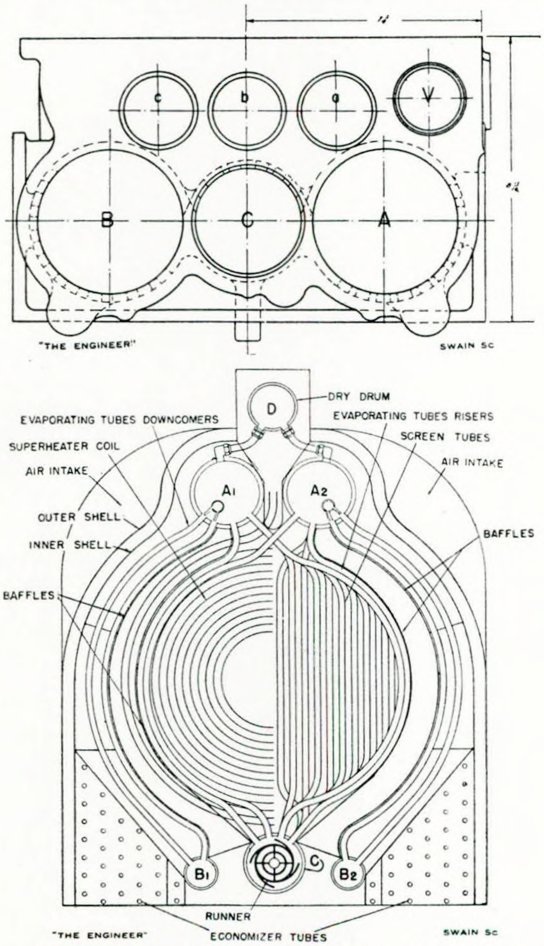


FIG. 3 & 4. *Cross Sections of Cylinder Block and Boiler.*

advance of the eccentrics. This angle is limited to a somewhat narrow range to obtain reversibility without serious reduction of efficiency. Because of the serrations the valve, rolling over as it reaches its dead centre, remains open for a longer period than would otherwise be possible with the proper angle of lead. In engines where the power-to-weight ratio is not a prime consideration, these serrations can be omitted and the admission edges provided with piston rings, thus giving a shorter effective cut-off, with resultant increases in total expansion ratio. At S, Fig. 5, is shown a small slot or notch running longitudinally. The purpose of this slot is to lengthen the cut-off beyond that which would be possible with only the reciprocating motion of the valve, when the eccentrics are set

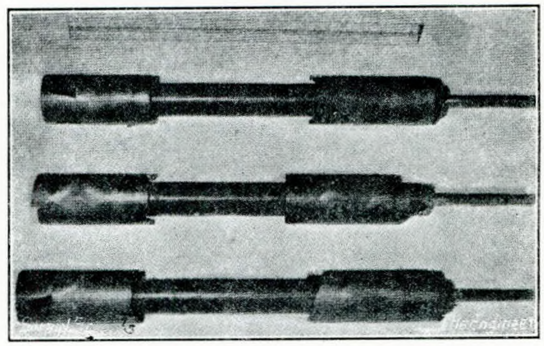


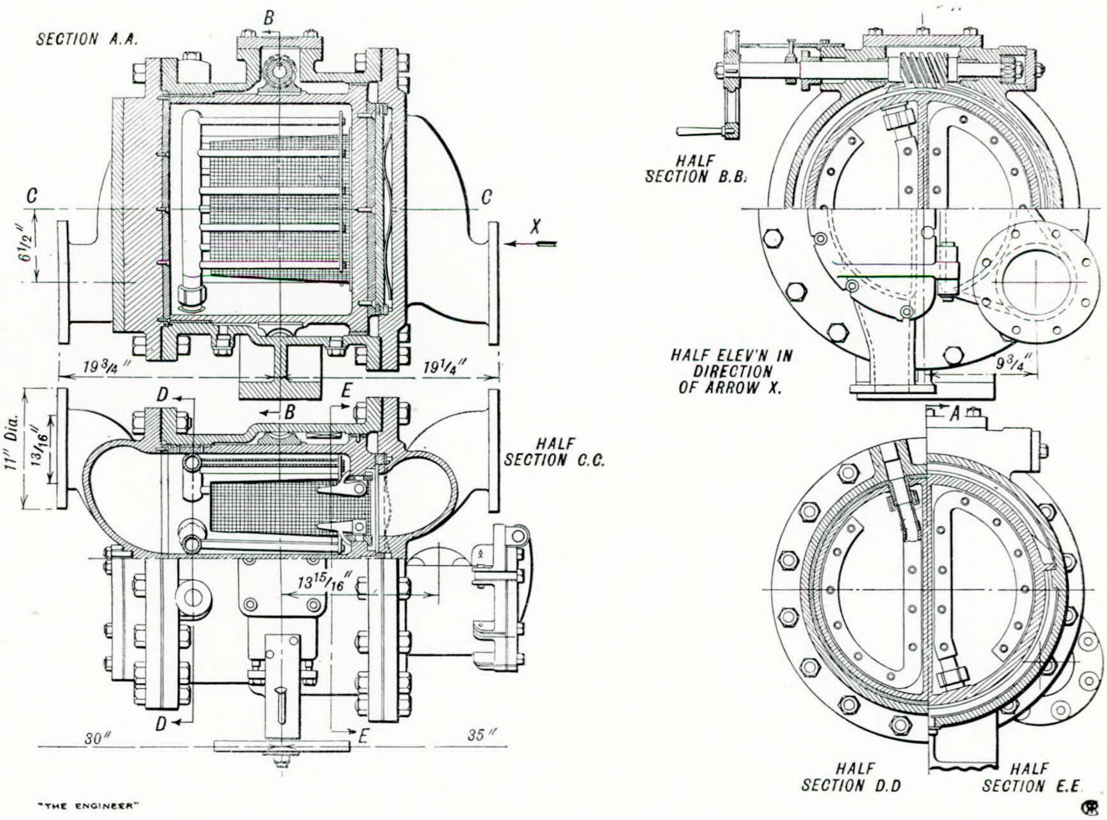
FIG. 5.

at an angle of advance which gives an economical point of cut-off. If this slot were exposed to steam when the valve approaches the point of admission, steam would be admitted to the cylinder so far in advance of dead centre that the engine would not operate. However, due to the motion of the valve, the slot is covered by a bridge as it approaches the point of admission, and is rolled out into the steam space only after dead centre has been passed. It remains exposed to steam until nearly the end of the stroke, however. Its width ordinarily would be from 2 to 3 per cent. of the circumference of the valve. Thus, when the engine is starting or running at slow speed, it is supplied with steam at approximately full pressure during practically the entire stroke. As it speeds up the amount of steam which gets through per stroke becomes less important, and at maximum speed the effect is negligible. Thus it is possible to obtain the effect of a long cut-off, and the resulting increased torque at starting and low speeds, with a fixed economic eccentric position which would normally give the same short cut-off at all speeds, except for this slot and the rolling motion of the valve. Engine balance is stated to be so effective that there is virtually no vibration when the engine is running either forward or in reverse. This is accomplished by two sets of counterbalances, one set being mounted on the eccentric, which rotates in one direction in a plane above the cylinders, while the other set is mounted on the oil-pump shaft which rotates in the opposite direction in a plane below the cylinders. Splash lubrication is employed for the bearings, cranks and eccentrics, in addition to forced lubrication from a small vane pump, through holes drilled in the crankshaft, to the surface of each crankpin. Lubrication of the pistons and valves is provided by a multi-plunger lubricator, which sends cylinder oil into the main line and to each end of the three valves. The total weight of the engine is 450lb. and the maximum power 90 h.p., giving a weight of 5lb./b.h.p.

The Knox boiler shown in Fig. 4 is compact, relatively light, and able to produce 1,500lb./hr. of steam at 700lb./in.^{2} and 750° F., with smokeless combustion and high efficiency. Forced circulation is employed, a simple impeller in the centre lower drum causing the water to flow up through the riser tubes which surround the combustion chamber. The feed water is pumped into the two steam drums A_1 and A_2, after passing through economisers. The downcomers from these drums are located in the third pass of the boiler, and serve to connect them to the smaller lower drums B_1 and B_2. The centre drum C at the bottom of the boiler is connected to the outer drums by headers at both ends, and contains a hollow impeller driven by a shaft, which emerges from the header through a simple stuffing-box. When the impeller is rotated the pumping action creates a pressure which forces the water in drum C to pass upward through the risers which form the walls of the combustion chamber. The suction caused by the displacement of this water causes the water in the drums B_1 and B_2 to flow into the centre drum, and thus a vigorous forced circulation is established, which is quite uniform along the length of the drum. The risers consist of four rows of stainless steel tubes, of $\frac{1}{2}$ in. outside and 0.43 in. inside diameter, approximately 3 ft. long. The tubes are staggered, the distance between centre lines being $1\frac{3}{16}$ in. Although the combustion space is thus waterwalled, the first few rows of tubes near the burner are covered with refractory to provide radiant heat for vapourising the oil fuel, so ensuring smokeless combustion and higher efficiency. The heat transmission in the downcomer tubes in the third pass of the boiler is much less intense, as they are not exposed to the radiation from the flame. Dry steam for the superheater is provided by a dry drum D above A_1 and A_2, to which it is connected by a number of small tubes. This dry drum is only necessary because of the small diameter of the upper drums in this particular boiler, and could be dispensed with in a larger one. The superheater consists of a spiral coil of tubing, and is located at the back of the combustion space, facing the burner, but separated from the combustion space by a screen of riser tubes. The temperature of the superheated steam is regulated by the injection of feed water into the superheater inlet. Three economisers are used, the first two being located in the last pass on either side of the boiler, while the third, which consists of coiled tubing, is at the base of the uptake. The temperature of the flue gases leaving this last economiser is always below 400° F.—*Paper by S. L. G. Knox and J. I. Yellott, presented at the December, 1940, meeting of the American Society of Mechanical Engineers and summarised in "The Engineer," Vol. CLXXI., No. 4,445, 21st March, 1941, pp. 197-198.*

Self-cleaning Strainer.

The accompanying drawings show the construction of a new and improved form of self-cleaning strainer primarily designed for use in power stations, but equally suitable for other applications where water is likely to contain *débris*, small stones, etc., and where the flow must be maintained uninterrupted whilst the strainer is being cleaned. In this device the operating mechanism is sealed off from the fluid flowing through the strainer. An internal drum, which can be rotated through 180° by an external handwheel and worm and wheel gear, is divided into two compartments, the water flowing through the strainer in one of them, whilst the strainer in the other compartment is being cleaned. The ends of the drum are sealed against leakage by metal-to-metal faces, and the whole is enclosed in a cylindrical casing. Spray pipes are fitted inside the drum for cleansing purposes. When the strainer in way of the flow is to be cleaned, the drum is turned through 180° by the handwheel until stops prevent further motion, the strainer in the other compartment being by this means brought in way of the flow. At the same time the cleaning water supply connected to the cleansing compartment is automatically brought into line with the spray-pipe inlet. By opening a small valve which controls the mains supply water under pressure is admitted to the spray-pipe and washes away the *débris* from the strainer, after which a drain valve is opened to allow the washing water to flow away. Normally not more than ten minutes' flow of the washing water is necessary to clean the strainer thoroughly. Access is provided at each end of the cleaning compartment for the renewal of strainer and spray-pipe units, work which can de done without interfering with the operation of the strainer. The inlet and outlet flanges of the casing are so arranged that the strainer can be fitted into any straight run of pipe. A foot is also provided to enable the strainer to be mounted on a floor pad.—*"The Engineer," Vol. CLXXI., No. 4,445, 21st March, 1941, p. 198.*



Arrangement of self-cleaning strainer.

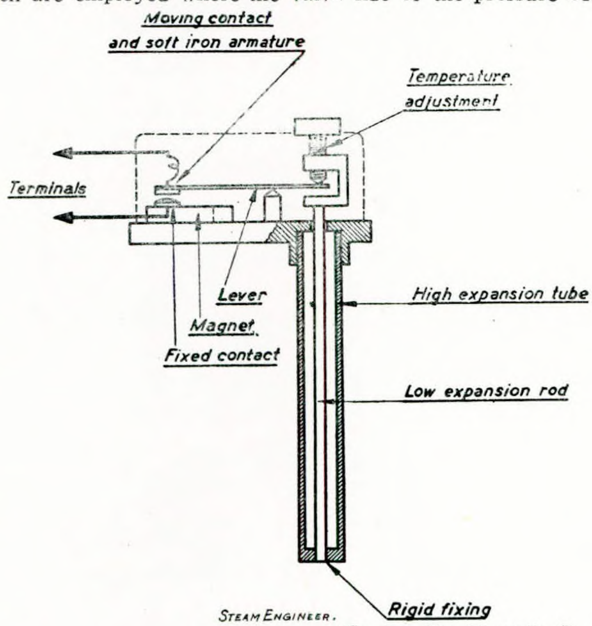
Indirectly-driven Fans.

The general adoption of superheaters and air heaters for marine boilers is making the employment of some form of assisted draught essential, this being effected by the use of either induced- or forced-draught fans or possibly both. The form of drive to be employed for these fans requires careful consideration, particularly in the case of induced-draught fans where the steam engine or electric motor used for the purpose may be in an inaccessible position and subjected to the effects of the high-temperature flue gases, for which reason it may not receive the attention necessary to maintain it in an efficient condition. For maximum fan efficiency it is desirable that the speed of rotation should be kept low, and certain firms specialising in the design and manufacture of these fans suggest that an indirect drive is to be preferred, using either vee belts or roller chains.—*"Ship-*

building and Shipping Record", Vol. LVII, No. 10, 6th March, 1941, p. 219.

Automatic Temperature Control.

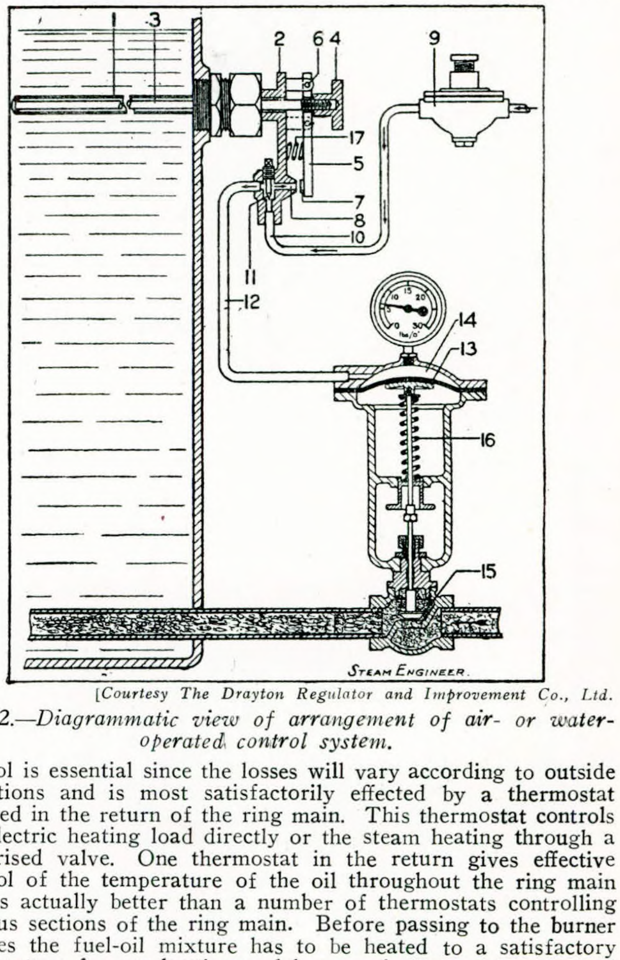
In the course of a paper entitled "Principles and Applications of Automatic Temperature Control" recently read by M. J. Gardside, B.Sc.(Eng.), at a meeting of the Manchester Association of Engineers, the author dealt with six different types of controls. These comprised:—(1) Bimetallic-operated controls with short-break switches, largely used for the control of electric heating, particularly with a.c. circuits. Owing to their inherent simplicity and robustness, thermostats of the type shown in Fig. 1 and others of similar design, are widely used for all general purposes, *e.g.*, boiler and calorifier control, operation of motorised valves, etc. (2) Bimetallic instruments incorporating mercury tubes, frequently used for the control of d.c. electric water-heaters. The reliability is not so great as that obtained with a bimetallic thermostat of the microgap switch type employed with a relay. (3) Liquid expansion or vapour pressure instruments incorporating various forms of electric switch, which are useful where the space available for the sensitive element is small, or where it is impossible to accommodate the switch portion near the sensitive element. (4) Thermostatic valves of the direct-operation type, such as are generally employed for small sizes of tanks and calorifiers; smaller types of directly-operated valves in which the expansion of a bimetallic tube operates a valve directly, are used in domestic gas appliances such as cookers and water-heaters. (5) Electrical thermostats operating motorised valves, which are employed where the valve size or the pressure of the



[*Courtesy The Rheostatic Co., Ltd.*

FIG. 1.—*Diagrammatic view of bimetal micro-gap switch thermostat.*

steam, hot water, oil or gas constituting the heating medium, requires the application of more power than is available from direct operation. When used in this way a sensitive thermostat can command an electric motor capable of developing a high torque and operating valves or dampers of large sizes. Such a system is also invaluable when the sensitive element is located at a considerable distance from the valve to be controlled. Motorised valve operation, which can be either of the on-off, floating or modulating type, may be considered the general-purpose method of providing temperature control of steam or hot-water heating systems. (6) Air- or water-pressure-operated valves of the type illustrated in Fig. 2 have the same field of application as modulating electric controls and are advantageous where it is necessary to be independent of electric supply, or where air or water pressure is available. Temperature control plays an important part in the handling and burning of special liquid fuels which may become too viscous to handle below a certain temperature, while at slightly higher temperatures sludge is deposited which clogs filters and again prevents satisfactory handling. Delivered warm, the fuels (*e.g.*, mixtures of creosote and pitch) are stored in electrically—or steam-heated tanks with straightforward temperature control. From the storage tank the fuel is then pumped around a distributing ring main to the various burners, heat loss during the passage round the ring main being offset by running a steampipe or electric heating cable alongside it. Temperature



[*Courtesy The Drayton Regulator and Improvement Co., Ltd.*

FIG. 2.—*Diagrammatic view of arrangement of air- or water-operated control system.*

control is essential since the losses will vary according to outside conditions and is most satisfactorily effected by a thermostat inserted in the return of the ring main. This thermostat controls the electric heating load directly or the steam heating through a motorised valve. One thermostat in the return gives effective control of the temperature of the oil throughout the ring main and is actually better than a number of thermostats controlling various sections of the ring main. Before passing to the burner nozzles the fuel-oil mixture has to be heated to a satisfactory temperature for combustion and here again temperature control plays an important part in ensuring efficiency. — *"The Steam Engineer", Vol. X, No. 114, March, 1941, pp. 154-156.*

New Swedish Pilot Cutter.

The pilot cutter "Gävle" recently launched at the Helsingborg Shipyard is being built to the order of the Swedish Government for service in northern waters. The hull of the vessel is strengthened for navigation in ice and has an overall length of 121ft., a beam of 28ft. 10in. and a depth of 18ft. The stem is of the round-nose type and a Simplex streamline rudder is fitted. There are two masts carrying 10-ton derricks. The propelling machinery is located amidships and consists of a four-cylinder Diesel engine developing 500 h.p. at 220 r.p.m. The propeller is of Ka-Me-Wa construction, with adjustable stainless steel blades. —*Lloyd's List and Shipping Gazette", No. 39,418, 12th March, 1941, p. 8.*