The

INSTITUTE of MARINE ENGINEERS

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SESSION 1938

Vol. L. Part 1.

President: STEPHEN J. PIGOTT, ESQ., D.Sc.

Modern Marine Steam Turbine Design.

READ

By S. S. COOK, F.R.S. (Member).

On Tuesday, January 11th, 1938, at 6 p.m.

CHAIRMAN: MR. R. RAINIE, M.C. (Chairman of Council).

Synopsis

INTRODUCTORY. The turbine as a hydrodynamic machine. General remarks on differences between land and marine turbines.

The leading features of a turbine are described with reference to a concrete example of a modern marine installation with three turbines per shaft and single reduction. Commencing with questions to be considered in the preparation of the blading schedule and the attachment of the blades to the structures which carry them, matters of interest in the latest practice are discussed in relation to the main structural parts, balance pistons, glands, bearings, turbine adjusting blocks, main thrust block, and finally mechanical gearing, with illustrations from the example chosen. The paper concludes with a few remarks on the economy of an installation as a whole.

Many treatises have been written and many papers read before this and other institutions on the theory, construction and operation of steam turbines both land and marine. When therefore the author had the honour of being requested by your Council to present a paper on "Modern Marine Steam Turbine Design" he realised that it would be difficult to avoid repetition of many things already thoroughly well known. He felt moreover that what was required was an account not so much of the latest new thing in marine turbine design but of such construction as had been well tried in practice, believing that information which had a proved basis of reliability would be the more welcome to this Institute. The marine steam turbine having during the last few years developed in certain definite directions in the interest of attaining the highest standard of reliability and economy, the author proposes to present a description of it in relation to one of its latest examples, embodying the essential principles which still have to be observed and with constructive features which have been well proved in actual service.

The steam turbine is one of the simplest of mechanisms. It is characterised by an almost complete absence of contact between fixed and moving elements. In the working elements in fact, where high temperatures, high pressures and high speeds vessel to be quite independently driven. The use of reduction gearing allows a wide choice of the revolutions of the turbines comprising a single unit, which choice is then governed by the requirements of the turbine design, except in the case of units of the largest power. In the latter it is frequently found that the design of the gearing itself imposes practical limits upon the revolutions of the turbine, a position however which is obviously greatly eased by the distribution of the total power over several points of engagement on the main gear wheel.

These questions which are fundamental to the design leave considerable scope for the exercise of judgment and experience on the part of the designer (1) in making the most suitable subdivision of the turbine, (2) in choice of revolutions. In this he is generally disposed to repeat as closely as possible the design of a previous installation with such modifications as may be required to meet the new conditions.

A more difficult question because of the variety of views is the choice of initial pressure and temperature. By suitable design of the turbines, when using high-pressure steam all stresses can be kept within safe values, and there is from this point of view no reason to abstain from adoption of the highest pressure which offers a prospect of economy. For the highest economy high pressures and high temperatures are required and for vessels of large power theoretical investigations have shown it to be advantageous to adopt pressures as high as 1,000lb. per sq. inch and temperatures as high as the material of the superheater tubes will permit. Still higher pressures show a gain in economy if reheating is adopted. A paper was read here last April by Mr. McEwen describing the application of this device to the turbines of the "Conte Rosso", in which steam was provided for additional high-pressure turbines at a pressure of 1,850lb. per square inch and after expansion in these turbines was resuperheated before passing at a pressure of 200lb. to the lower pressure turbines originally installed in the vessel.

Up to the present, however, for marine work in general conservative views prevail, and initial pressures have not exceeded the 500 to 550lb. of the river steamer "King George V" and of H.M.S. "Acheron". High initial pressure means an increased percentage of water of condensation at the exhaust end of the turbine unless reheating is resorted to. Boilermakers and turbine builders would not hesitate to adopt 750lb., but above that pressure it would be essential to resort to intermediate reheating; with this a prospect of substantial gain is immediately apparent up to a pressure of say 1,500lb. per square inch.

The temperature limit is more clearly defined. Above 450° C. the resistance of materials falls off rapidly with increasing temperature. The temperature of the superheater tubes must of necessity be higher than that of the steam since they are transmitting heat to it, and it is necessary to choose a steam temperature such as will still leave a safe margin for the superheater tubes. A liberal rating of superheater surface and high velocity of steam through the tubes will reduce the margin that is necessary. Some margin is also necessary for possible fluctuation in temperature. Generally speaking, from these considerations it is not advisable to exceed about 750° F. for the temperature of the steam at the turbines, unless special material is used for the last passes of the superheater.

For powers therefore of, say, 6,000 s.h.p. per shaft upwards, one may say that a pressure of 500lb. per sq. inch and a temperature of 750° F. represents sound and reliable practice, already covered in fact by actual examples.

The first step in the design of a turbine is the preparation of the blade scheme, which consists in the determination of the number of turbine elements required to suit the steam conditions of each turbine, their mean diameters, their heights and the type and sizes of blades to be employed. With reaction turbines it is usual to employ a stepped system, a group of rows of constant height followed by a second group of increased height and so on. The blade heights from group to group are graded to correspond with the increasing volume of the steam, with the steam velocity increasing with the mean diameter so as to preserve a practically uniform velocity ratio. It is quite unnecessary to enter into any explanation of velocity diagrams either for reaction or for impulse blades, the principles of which are now well known and can be found in any text book on the subject. The values to be adopted for the efficiency and the discharge coefficient rest in the last resort upon experimental determinations. Confining attention to blades of the reaction type these are usually of similar form throughout the greater part of the turbine and one curve of efficiency and one value of the discharge coefficient suffices for the purpose of design. As far as the blade itself is concerned, the efficiency is a function of its velocity ratio, which for satisfactory adaptation of the results of experiment to the requirements of design is usually taken as the ratio of the mean blade speed to the steam velocity theoretically due to the pressure drop. For normal turbines at full power the velocity ratio is usually from ·8 to ·85, giving a basic blade efficiency of about 90 per cent. Corrections have, however, to be applied to this figure to take account of leakage and other losses.

Two main requirements have to be met, efficiency and capacity. The turbine must be capable of passing a certain quantity of steam with a given initial steam pressure and condition, sufficient that is to say, when the overall efficiency has been determined, to give the required power. For a given steam pressure and condition the capacity depends upon the steam velocity and the blade area. Fundamentally this can only be arrived at by a step-by-

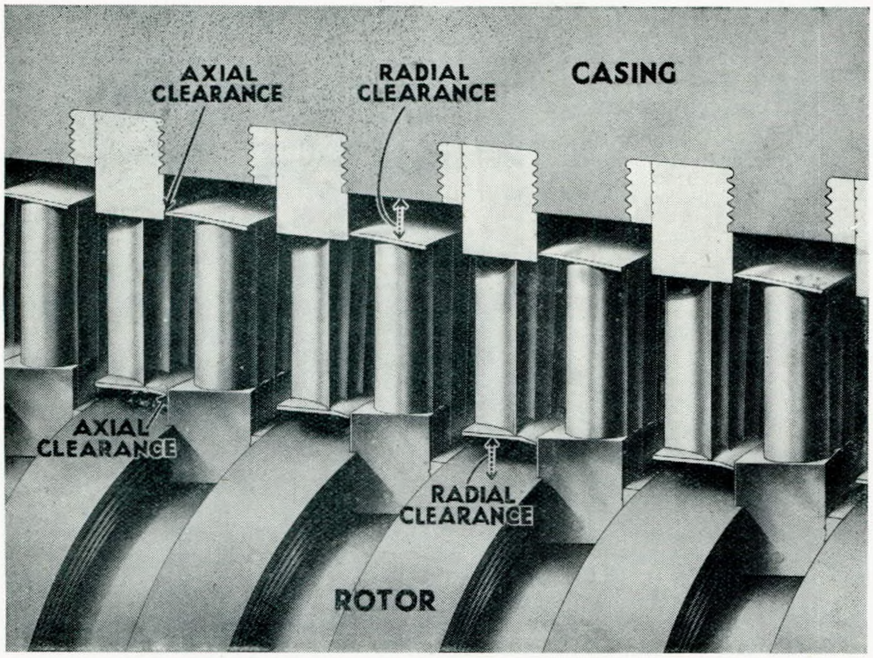


FIG. 2.—End-tightened blading.

step calculation from the exhaust end of the turbine where the pressure is known, being either the pressure corresponding to the vacuum of the condenser or the initial pressure of the turbine immediately following that under consideration. Such a calculation, however, is somewhat tedious and manufacturers have various short-cut methods to obtain closely approximate values without the labour of a full calculation. It is an interesting point that the quantity of steam which can be passed through a turbine with a given initial pressure is almost independent of the speed of the rotor. To be strictly accurate, with normal reaction blading this quantity is only a few per cent. more when the turbine is standing than when it is running at full speed. This practical constancy of capacity is very useful, because it means that for a given turbine over a large range of conditions the pressure is a direct indication of the amount of steam passing; it is also very convenient for the designer, and a design is usually approached from that point of view. Generally speaking the quantity of steam of a given temperature that can be passed through a given turbine is proportional to the pressure of admission. This implies that for the earlier portion of the blading at any rate the steam velocity is constant, and a function of the blading itself, not of the steam pressure, which can be proved to be the case for a considerable degree of throttling or reduction of the admission pressure.

In arranging the blade scheme, it is frequently necessary to make some special provision for the attainment of economy at reduced powers. If the steam is merely throttled, there is loss due to reduction in the pressure of admission as well as loss due to reduced blade velocity and consequent reduction of velocity ratio. One method of improving consumptions at low powers is to use an impulse wheel for the first stage and maintain the initial pressure by closing a portion of its nozzles; for small reductions of power better results are obtained by fitting an additional stage with shorter blades which is by-passed for full power, while for large reductions of power a separate cruising turbine is employed which is either disconnected at full power or allowed to run in a vacuum.

For a mercantile vessel a fairly flat curve of consumption rate can be obtained by the second of these methods. An example of such a curve is given in Fig. 20.

The high relative speed of fixed and moving parts makes it necessary that there should be some clearance between them, that is to say between the tips of the rotor blades and the inner surface of the casing, and between the tips of the fixed blades and the surface of the rotor. Where the blades are not provided with shrouding at the tips, it is usual to sharpen the tips so as to present as narrow an edge as possible to the opposing surface and thus minimise the abrasion and the heat generated by it in the event of an accidental contact. This is in accordance with the general maxim first propounded by Sir Charles Parsons that of two bodies running in close proximity at a high relative speed,

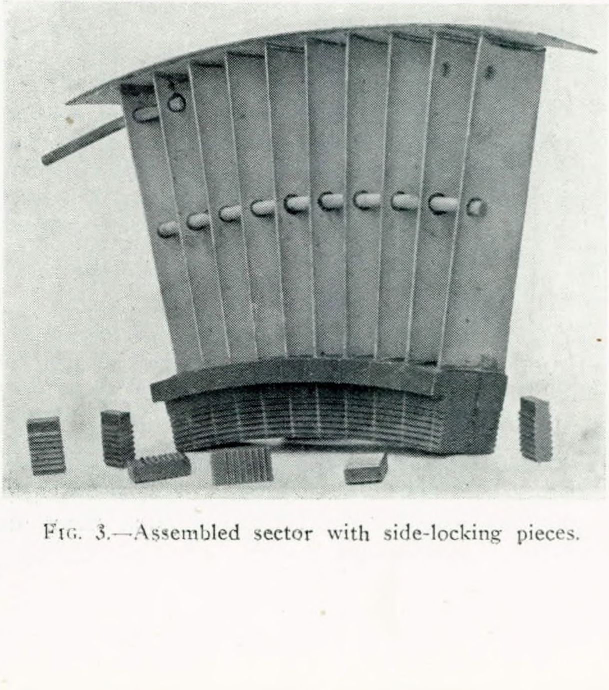
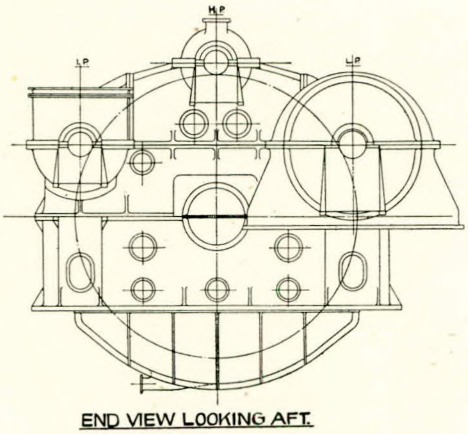
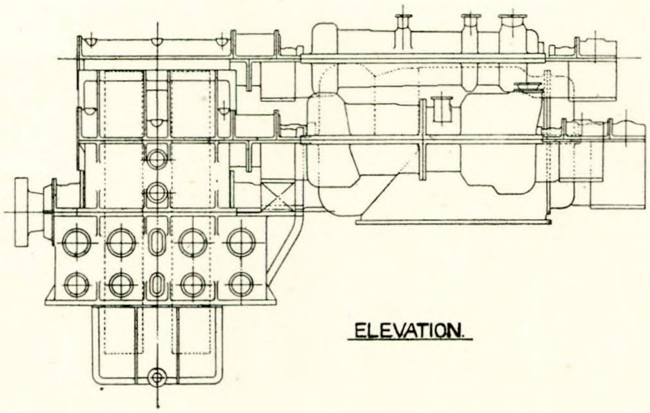


FIG. 3.—Assembled sector with side-locking pieces.





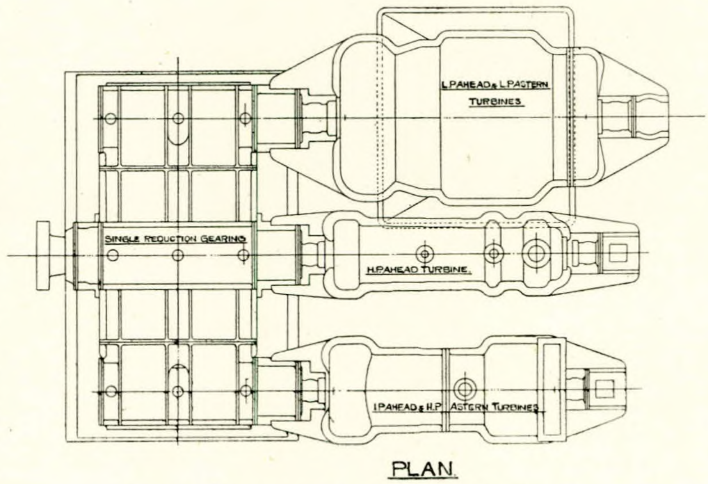


FIG. 4.—General layout of an installation.

one of them must be provided with sharp edges.

In modern practice, however, for blades of a length below about 4 inches, it is customary to adopt an improved system of providing this clearance. The blades are fitted with a thin shrouding usually of the same metal as the blades covering the end of the blade and projecting a short distance axially, with a sharpened edge which runs in close proximity to a surface at the root of the adjoining blade (Fig. 2). This system has several advantages. The clearance so provided is adjustable by end movement of the rotor, it can be ground in light contact and then set back to a definite uniform value at all such edges throughout the turbine, it is in a direction in which it is less liable to be affected by distortion of the casing or vibration of the rotor, and it is on this account safe to adopt a smaller clearance in this direction than at the blade tips, with consequent reduction of the leakage losses, a matter which is of considerable importance in high-pressure turbines and in turbines of small power. This system of blading is known as the end-tightened type and has been extensively used for both land turbines and marine turbines for the last twenty years.

In the early turbines for the avoidance of corrosion the blades were made of brass, 70 per cent. copper, 30 per cent. zinc, and secured between spacing pieces caulked into grooves in rotor and casing. The sides of the grooves were serrated and the roots of the blades indented in order to increase the grip of the spacing pieces. This is still good practice where the conditions permit.

Brass, however, is unsuitable for high temperatures or for high stresses. It is also subject to erosion at the exhaust end of the low-pressure turbine through the rotor blades striking against drops of water of condensation. With the employment of higher temperatures and higher speeds other materials are required. A suitable material for the requirements of modern turbines is found either in Monel metal or in what is commonly called stainless iron. The latter is a stainless steel with so low a carbon content that it does not harden appreciably under the heat treatment received during the processes of brazing or soldering. The most satisfactory method that has been evolved both for convenience of manufacture and reliability of attachment is the assembled sector method. For the manufacture of an assembled sector, blades and packing pieces both cut from rolled strip of the correct profile are assembled together in a suitable frame to make up a sector consisting of about 12 blades, brazed or soldered at the roots and fitted with binding wire or shrouding. The root of the sector so formed is serrated at the sides, and fitted in the rotor or casing in grooves the sides of which are also serrated. These grooves are slightly wider than the root and the difference is made up with side locking pieces driven up endwise along the groove (Fig. 3).

From this point the author's remarks will be appended to a concrete example of a modern installation of geared turbines, with occasional digressions. Fig. 4 shows the general lay-out of a twin-screw installation of 24,000 s.h.p., each shaft

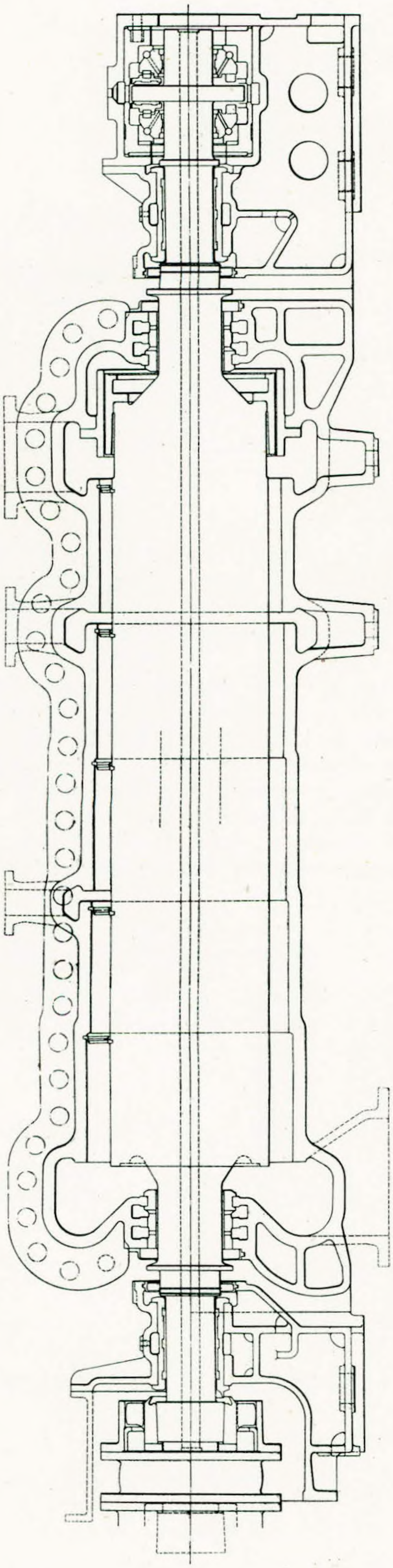


Fig. 5.—H.P. turbine for installation shown in Fig. 4.

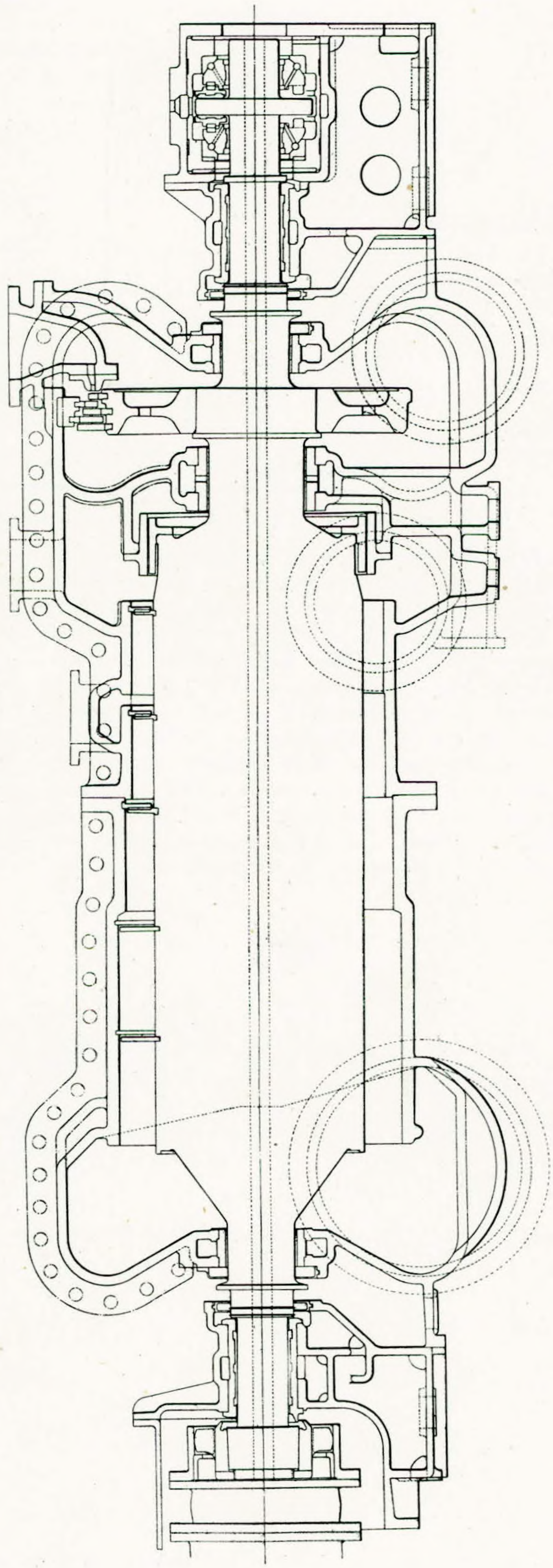


Fig. 6.—I.P. turbine for installation shown in Fig. 4.

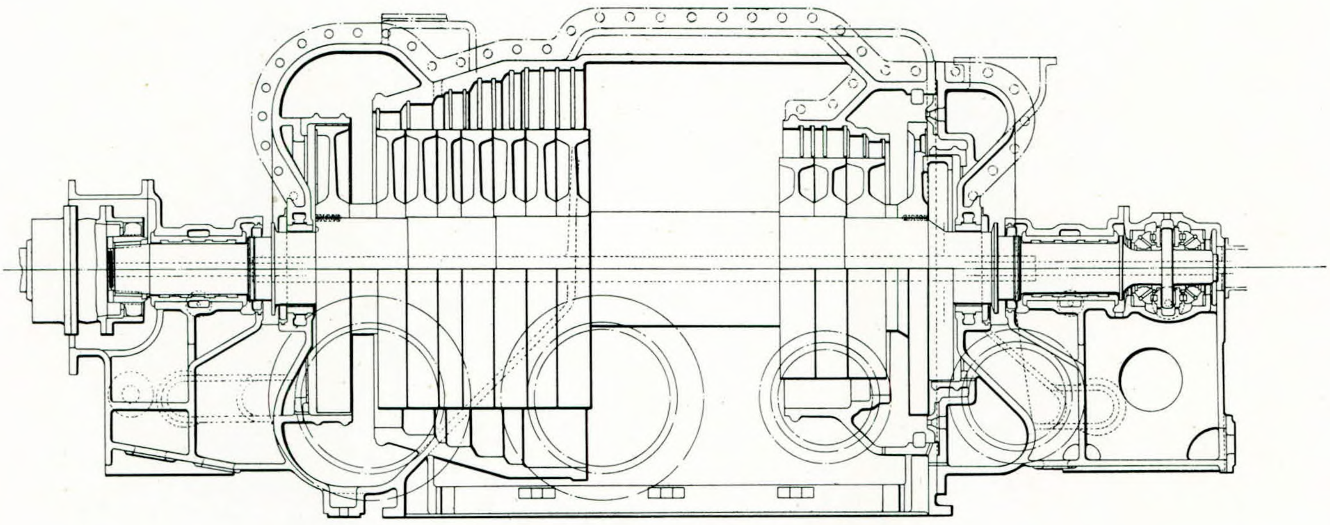


FIG. 7.—L.P. turbine for installation shown in Fig. 4.

of which is driven by a set of three turbines in series with single-reducton gearing to the propeller shaft, which latter runs at 150 r.p.m. The initial steam pressure is 400lb. per sq. in. absolute and temperature 750° F. The high-pressure turbine is shown in Fig. 5; it runs at 1,700 r.p.m. and develops one third of the total power of each set. Fig. 6 is the intermediate-pressure turbine which receives steam from the high-pressure turbine at 102lb. per sq. in. abs., runs also at 1,700 r.p.m. and develops one third of the power, and Fig. 7 is the low-pressure turbine, receiving steam at 14lb. per sq. in. abs. and exhausting to the condenser. This turbine runs at 1,500 r.p.m. and develops the remaining third of the power. Fig. 8 shows the gearing with three double-helical pinions engaging at different parts of the periphery of the main gear wheel.

Returning to Fig. 5 showing the high-pressure turbine, it will be observed at once how small a portion of the structure is occupied by the blades, the working organs of the turbine. They have of necessity to be accompanied by a mechanical structure to carry them, to hold the fixed blades, to allow the moving blades to move at the speed which ensures the conversion of the pressure energy into

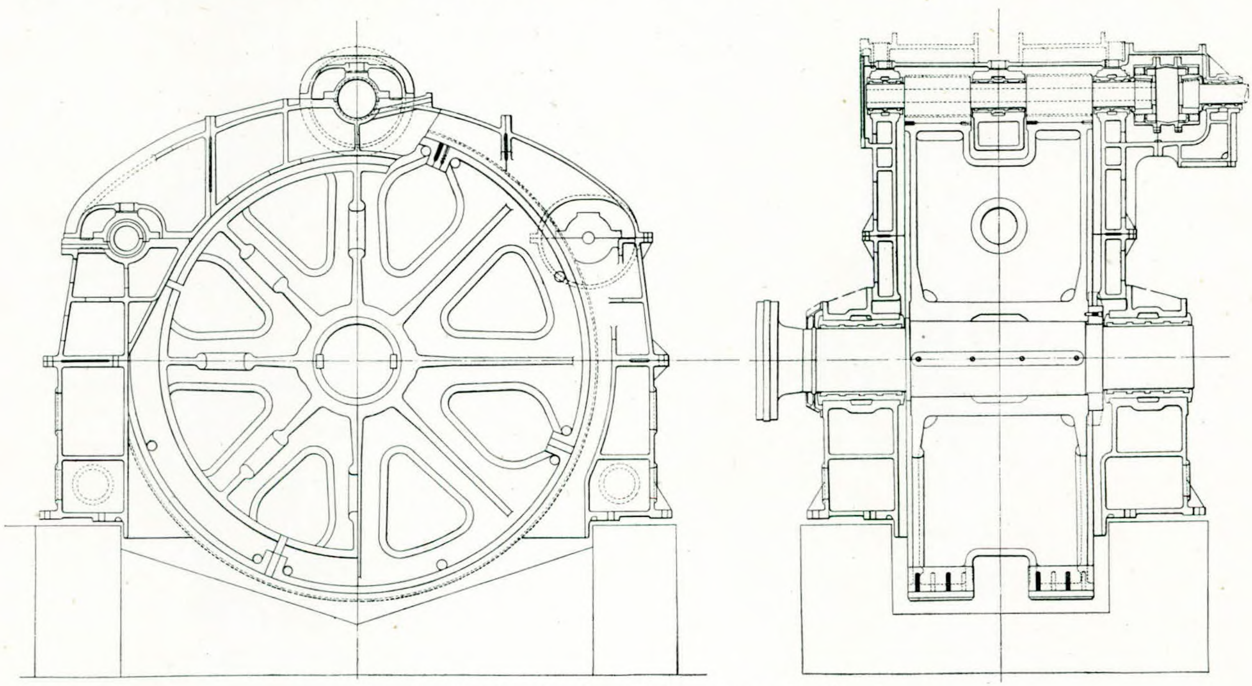


FIG. 8.—Gearing for installation shown in Fig. 4.

work at the required efficiency, and to conduct the steam to and from the blades. This necessitates a structure of rotor and casing such as is shown in Fig. 5 with suitable parts to fulfil these functions.

Confining attention at present to a brief enumeration of the essential parts making up the complete turbine, it will be seen clearly from the illustration how steam is led into and exhausted from the annular space occupied by the blading. Forward of the inlet space is a "dummy" or balance piston. The space behind this piston is connected by a pipe of large area to the space at the exhaust end of the turbine. The dummy is fitted with labyrinth packing, but its effect is precisely the same as that of a piston of the same diameter as the mean diameter of the packing, so that it balances more or less the end thrust due to the steam pressure on the blades. It is usual to arrange matters so that under all conditions there is a small residual thrust in a definite direction to be carried by the thrust block. At either end of the turbine casing is a gland usually also of labyrinth packing, where the rotor ends pass out of the casing. The object of the gland is to prevent steam issuing from the casing or on the other hand to prevent air entering into the casing, according as the space within the gland is at super-atmospheric or sub-atmospheric pressure. Next to the glands are the bearings which support the rotor with means for the supply and drainage of lubricating oil and baffles to prevent leakage of oil along the shaft. A collar on the shaft between gland and bearing gives further security against mixture of steam with oil. These bearings are carried on pedestals which form part of the bottom half of the casing, and the forward pedestal also carries a thrust block or turbine adjusting block for locating the axial position of the rotor. This thrust block has to carry the residual end thrust due to the steam pressure on the rotor blades and on the balance piston.

To prevent overspeed and also to shut down the turbine in the event of failure of oil supply, a shaft governor with oil relay to operate the emergency valve is fitted at the forward end of the rotor.

Beyond the after bearing is a coupling for transmission of the torque developed on the rotor shaft to the pinion of the gearing, whilst allowing axial freedom and a small amount of lateral freedom in the case of any misalignment.

The same features will be found in the other turbines. In the intermediate-pressure turbine (Fig. 6), however, there is in addition an h.p. astern turbine consisting of a three-row impulse wheel at the forward end of the turbine rotor, with a diaphragm gland separating it from the space behind the dummy. Location of the astern element at this end of the turbine ensures its being kept hot during ahead running and therefore ready for use on receipt of sudden order to go astern. It also makes the steam end thrust in the same direction as when going ahead. This steam thrust arises from the steam pressure on the annular area between the diameter of the external gland and that of the diaphragm gland, the latter being necessarily larger in order to secure stiffness of the shaft. In the latest constructions a pocket is provided in the diaphragm gland, which is put in connection with the condenser so that steam leaking across the gland when running ahead will not pass into the astern turbine, nor into the ahead turbine when running astern.

The low-pressure turbine (Fig. 7) has the same principal parts. For convenience this turbine is so placed that the ahead steam enters at the after end. It also contains an astern element, the low-pressure astern turbine, consisting of an impulse wheel followed by a few rows of reaction blading, and having a common exhaust with the ahead turbine. The low-pressure astern turbine operates in series with the h.p. astern wheel just referred to. The diameter of the low-pressure turbine is larger than that of the others, and the blade heights greater as will be seen, all in correspondence with the greater area required due to the increased volume of the steam. At the exhaust end of the ahead turbine, since these resources are still not sufficient to cope with the largely increased volume of the steam, the last few rows of blades are made of increased angle and discharge area.

The overall efficiency of a turbine plant is very sensitive to variation in vacuum. An increase of vacuum from 28″ to 29″ gives an increase of about $7\frac{1}{2}$ per cent. in the available energy of the steam. To utilise this to the full requires an increase of the exhaust area of the low-pressure turbine. The velocity energy of the steam as it leaves the turbine,

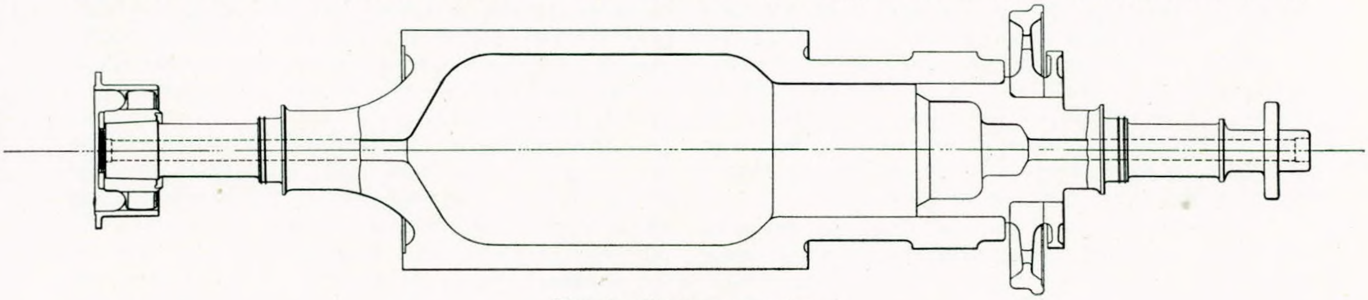


FIG. 9.—Drum type rotor.

practically that which corresponds to the axial component of the steam's velocity in the last annulus, is irrecoverable. This "leaving loss" as it is usually designated can only be kept low by providing a large annular area for the blading at the exhaust end, viz. in proportion to the volume of the steam. Thus the design of the exhaust end of a turbine is a special problem apart from the design of the turbine as a whole. Generally speaking it results in large diameters and low revolutions being required for the low-pressure turbine.

The common exhaust of the l.p. ahead and the l.p. astern turbine is connected with the condenser which is attached below the turbine, the steam exhausting through an exhaust branch of ample area and avoiding any loss of pressure between the turbine exhaust and condenser. This disposition of the condenser also provides complete drainage of water of condensation from the turbine, and eliminates any strains between the turbine casing and condenser, or any necessity to compensate for expansion. The condenser is sometimes supported on springs, set to a compression approximately equal to its weight.

We may now return to a more detailed consideration of the various parts of the turbine.

The high-pressure rotor as shown in Fig. 5 is a solid forging of carbon steel of about 38 tons per sq. inch ultimate. A hole of 2½in. diameter is bored along the axis throughout its length for examination to detect segregation cavities. The i.p. rotor, except for the astern impulse wheel which is shrunk on to an enlarged portion of the rotor end, is of the same construction. The impulse wheel is secured by end pegs which are not shown.

For large diameters it is usual to make the rotor either of disc or of drum construction. The rotor of Fig. 7 is an example of the former method. The blade carrying surface is made up of the rims of a number of discs shrunk on to a central spindle. These discs are designed on the same principle as discs of uniform stress, being reduced in thickness between the boss and the rim so as to allow the latter to take up its share of the stress. The ahead dummy and the astern impulse wheel are also discs shrunk on to the spindle. Where minimum weight is of importance a drum construction is adopted. Fig. 9 shows an example of a rotor of this type. The drum portion is forged integral with the spindle at one end, with an open mouth at the other end into which the other spindle is shrunk and pegged. A drum of carbon steel of the quality mentioned, viz. 34 to 38 tons ultimate stress, can be employed for surface speeds up to 400 ft. per second. Above this speed a disc construction is preferable in order to give a better distribution of stress in the material.

In consequence of the fact that the turbines during the handling of the vessel may have at some time to run at any speed below the full, the rotors are so designed that their first critical speed of whirling is well above the full power speed. This critical speed may be calculated in various ways, but the most practical way is to make a determination by graphical methods of its deflection under its own weight, from the value of which deflection a simple formula enables the critical speed to be found to a close approximation.

The high-pressure casing is a steel casting, the main part of it cylindrical and of suitable thickness to withstand the internal pressure. It is in halves separated at the horizontal plane and connected together by bolted flanges. About one-fifth of the bolts are fitted bolts to secure the two parts in constant relation. For turbines of moderate size as shown in Fig. 5, the bottom portion of the casing is in one piece with the end pedestals which support the rotor journals. In larger turbines these pedestals are separate castings and may then be of cast iron, but the main body of the casing is made of cast steel for all cases where the temperature exceeds 500° F. For lower temperatures cast iron is adopted. The bottom portion of the casing must be of sufficiently stiff construction to carry its weight suspended between the pedestal feet without appreciable deflection. There is no great difficulty in providing adequate stiffness. In fact in low-pressure turbines the casing is usually sufficiently stiff to carry not only its own weight but that of the condenser in addition.

The feet of the bearing pedestal at one end of the turbine casing, usually at the end remote from the gearing, are not attached rigidly to the seating stool, but fitted in guides which allow longitudinal freedom, thus permitting a small amount of expansion of the turbine casing when heated without any stress being thrown on to the casing or the stool.

Suitable branches are provided on the casing for the admission and exhaust of steam. The intermediate- and low-pressure casings have in addition branches for direct steam admission for

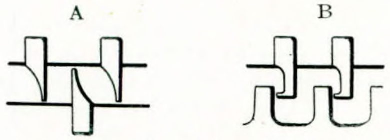


FIG. 10.—Labyrinth packing (A) radial clearance (B) axial clearance.

use under emergency conditions. Emergency connections are also provided for exhausting the high-pressure and intermediate-pressure turbines to condenser. By this means the subdivision of the whole turbine into separate turbines in series contributes greatly to reliability.

The purpose of the "dummy" or balance piston is, as has already been said, to balance the end thrust due to the pressure of the steam on the rotor blades. Without a dummy there would be not only an end thrust on the blades, but also on the annular area at the end of the rotor from the gland diameter up to the rotor surface. The dummy

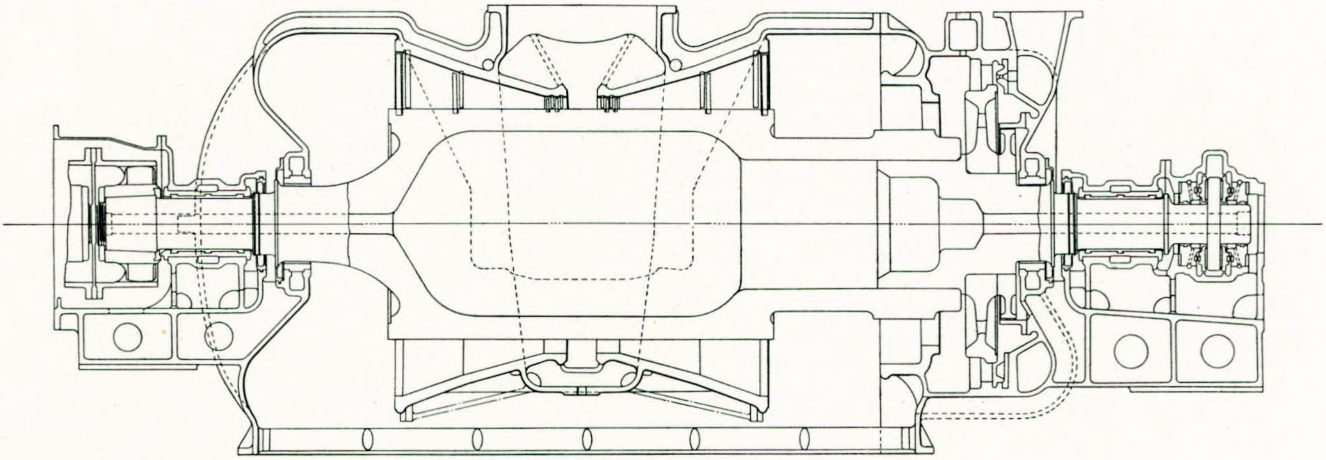


Fig. 11.—Double flow l.p. turbine.

has labyrinth packing either as shown in Fig. 10 as at (A) with radial clearance or as at (B) with axial clearance. The latter is usually employed in conjunction with blading of the end-tightened type. In either type it will be seen that it presents to the steam a series of small openings, whereby a difference of pressure is maintained at the expense of a small leakage, thus providing packing without contact.

It is not essential to have a dummy. The whole of the end thrust may be allowed to be taken by the thrust block. There is then no equalising pipe or connection to the exhaust end of the turbine, and no leakage loss at the dummy. The full pressure, however, then comes on to the gland, which it is necessary to lengthen. In fact it amounts to a replacement of the dummy by an additional section of the gland. The leakage of this section is less than that of the dummy, so that there is on the balance a slight increase of efficiency. The greatly increased load on the thrust block however, other things being equal, necessitates a larger thrust collar, with higher rubbing speed as well as increased load. This type of design has been adopted in many cases. On the whole, however, it is considered preferable to retain the dummy, the reduced pressure at the gland making it a simpler matter to keep the gland joint tight.

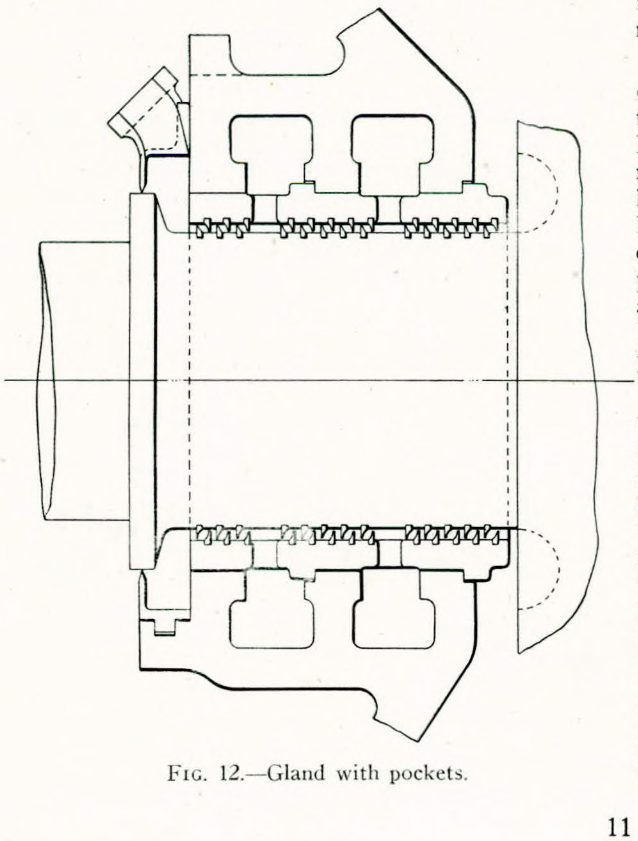


Fig. 12.—Gland with pockets.

In some of the earliest land turbines steam entered at the middle of the turbine and flowed through blading to right and left of the admission. In this case the thrust was balanced and no dummy necessary, but the blades were of course of only half the height, and there was greater loss by leakage. Such a double-flow system is now frequently adopted in low-pressure turbines of large power in order to avoid excessive height of blades, and in such cases of course no dummy is required. Fig. 11 is an illustration of a double-flow low-pressure turbine.

The glands, like the dummy, are usually of the labyrinth type, and are generally what is known as "steam sealed". They are provided with one or more pockets, or intermediate spaces between groups of labyrinth packing, and steam is supplied to or withdrawn from the outermost of these pockets in order to maintain that space at a pressure very slightly above atmospheric. This prevents leakage of air into the system which would otherwise spoil the vacuum of the condenser; on the other hand the pressure is so little above atmospheric that there is no excessive leakage of steam into the engine room. With a properly regulated gland there should be just a thin wisp of steam emerging from the gland, either into the engine room or into a vapour hood fixed over the gland mouth. Where more than one gland pocket is provided to a gland, these other pockets are connected to suitable parts of the turbine system to assist in the regulation of the gland, and to reduce the leakage losses (Fig. 12).

It will be understood that while some glands of some of the turbines have to withstand an internal pressure, the glands of other turbines have to withstand an external pressure if the pressure within the turbine casing is less than atmospheric. In these circumstances some of the glands of the system require to have their outer pockets supplied with steam, whilst the outer pockets of the others have steam withdrawn from them, a state of things only satisfactorily met by separate regulation of the glands of each turbine. There are, however, two systems of gland connections now in use which avoid this necessity. In one system the glands of all the turbines are made sub-atmospheric, the inner pockets of those which are exposed to pressure being connected to parts of the turbine casings at which the pressure is below atmospheric. All the outer pockets have then to be supplied with steam, and this can be done by connecting them all to a common receiver kept at a pressure slightly above atmospheric, the regulation of this pressure being the only control required.

An improved method of sealing, especially for an installation of turbines which comprises a cruising turbine, is one that for want of a better name is called the "cover-pocket" method. In this each gland has at least two pockets. The outer pockets are all connected by pipes of suitable diameter to one another and to a common receiver. The inner pockets are also connected to one another and to the same receiver. This leaves only a small pressure drop across the section immediately preceding the outer pocket, the steam in which is therefore in a quiescent state and is easily kept at the desired pressure slightly above atmospheric, the flow to or from each outer pocket being so small that it is virtually at the same pressure as the

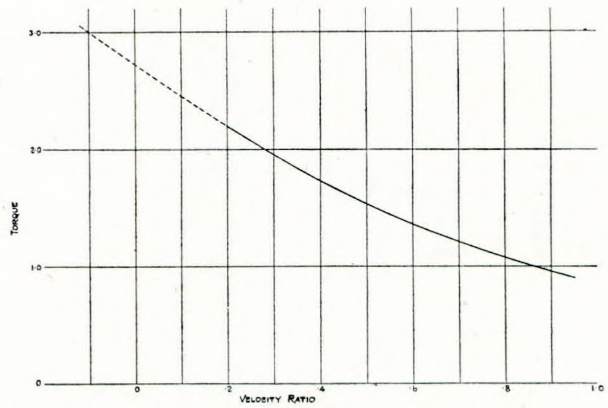


FIG. 14.—Curve of torque variation with speed.

receiver, and control of the receiver pressure is sufficient for the control of all the glands of the system. Also by way of the inner pockets there is interchange of steam between the glands which are super-atmospheric and those which are sub-atmospheric, so that the leakage of the former is used to pack the latter. The receiver is kept at its required pressure by a supply of steam from a suitable part of the turbine system, or at low powers by a supply of live steam through a reducing valve, with a valve controlled drain to condenser; in other words, it is supplied with steam or has steam taken from it according to the requirements of the case, in order to maintain it at the sealing pressure.

The bearings usually consist of two half-sleeves of gunmetal faced with white-metal. The white-metal surface is turned cylindrical of a bore from ·01″ to ·025″ larger than the journal diameter, according to the size. The entrainment of the oil by the motion of the journal surface into the wedge-shaped space which this clearance provides produces sufficient pressure in the oil film to support the journal. The oil film is fed with a plentiful supply of fresh oil led in centrally and along channels at parts of the bearing surface remote from the crown of the bearing and is drained freely

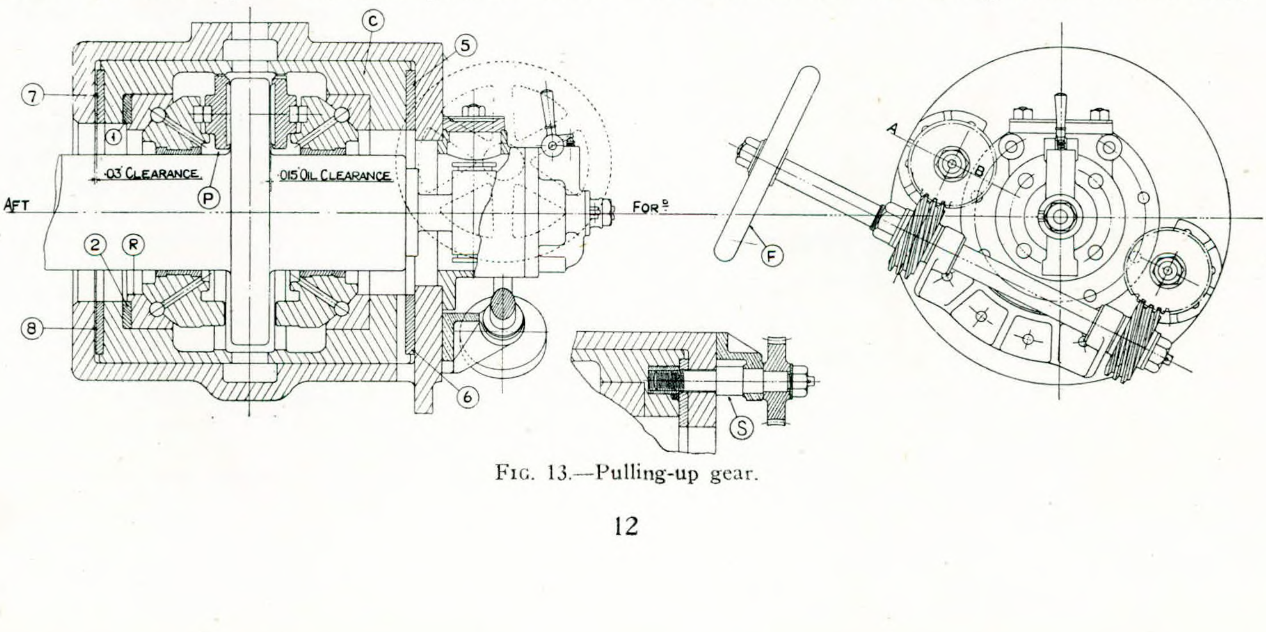


FIG. 13.—Pulling-up gear.

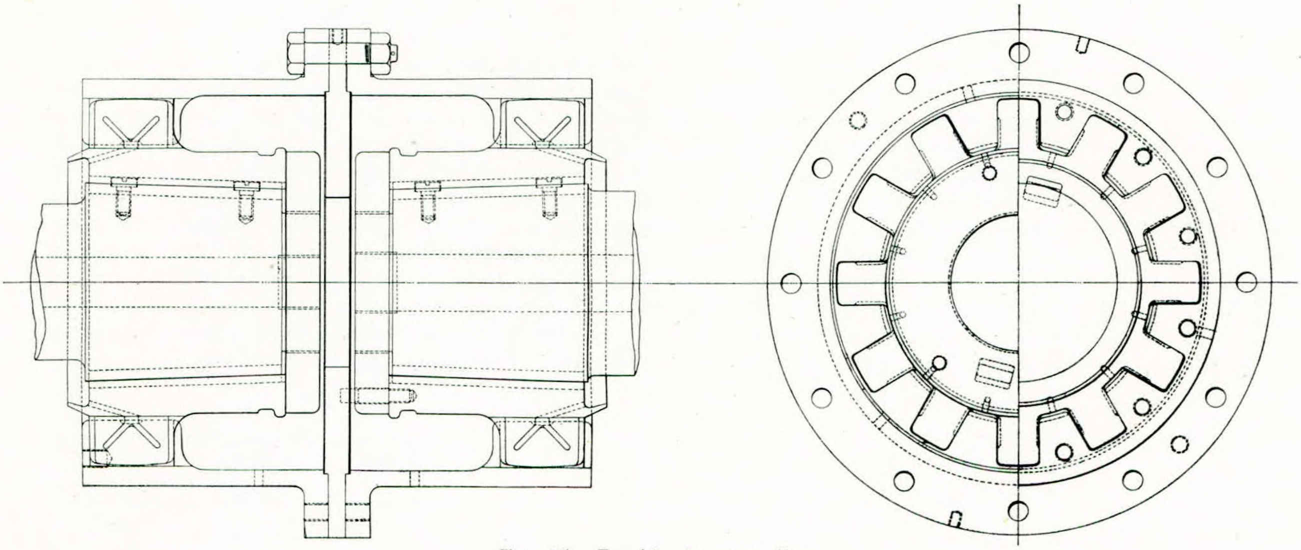


FIG. 15.—Double claw coupling.

away from the ends of the bearing. The turbine bearing sleeve extends a short distance beyond the white-metal at either end, and since the white-metal stands out from its surface ·015″ these ends provide safety strips which in the event of the white metal running out give temporary support to the journal and prevent contact at the tip of the turbine blades, a duty it is hoped they will never be called upon to perform. The oil is supplied to the bearings at a pressure of about 5 to 10lb. per sq. in., but ample quantity of supply is the important thing not the pressure, the latter being only an index that the supply is ample.

The same principle of producing pressure in the oil film by the viscous drag of the running surface of the collar is employed in the thrust block. The Michell thrust block is so well known at the present time that it needs no description here, except that to show the connection between its principle and that of the journal bearing it may be mentioned that whilst in the latter there is a naturally shaped wedge provided by the clearance space between two nearly equal cylindrical surfaces, in the Michell thrust block this wedge is ingeniously provided by pivoting the stationary member in such a way that it can tilt forward and allow

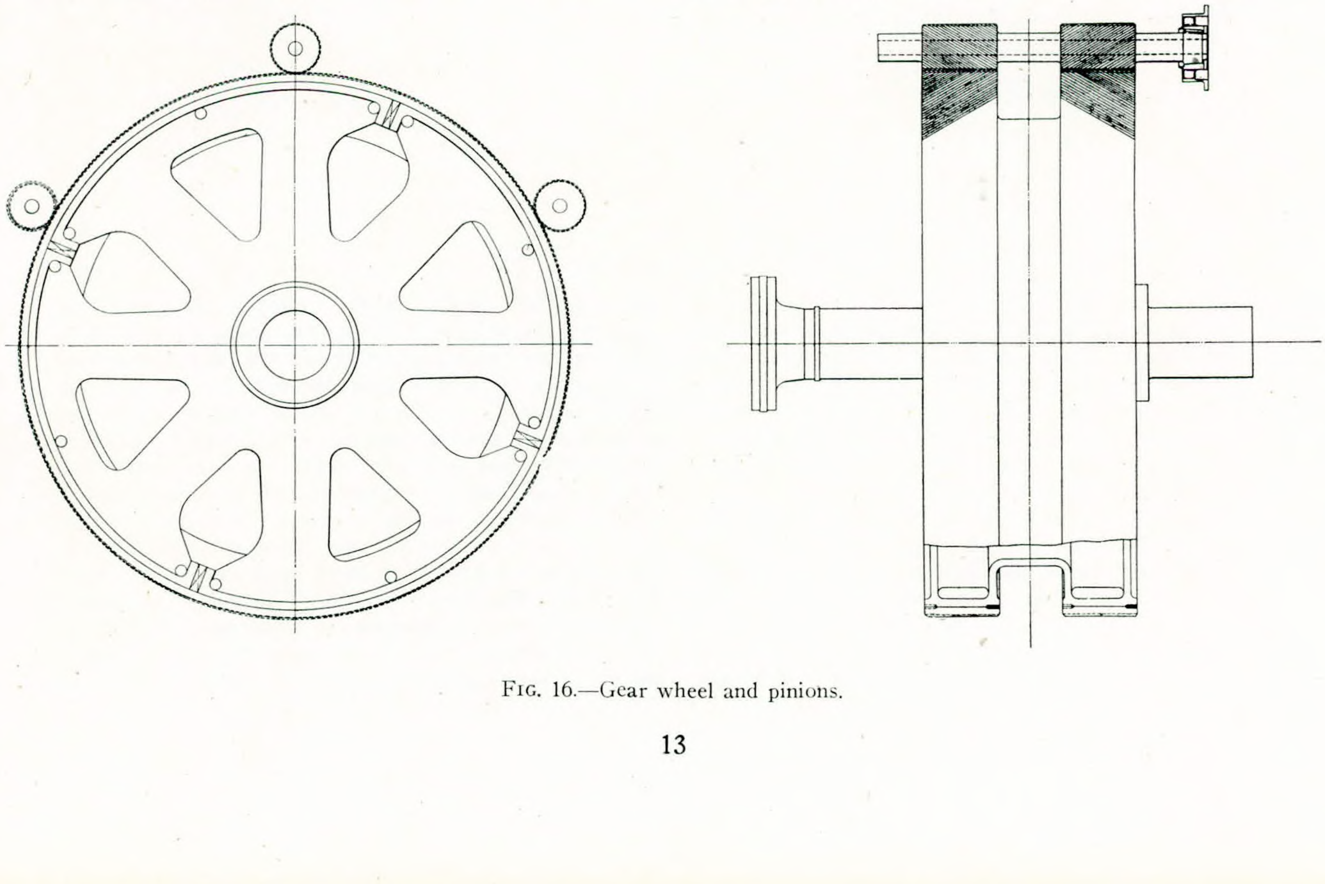


FIG. 16.—Gear wheel and pinions.

the entry of the oil. In the Michell thrust block (and in the Kingsbury thrust block which was introduced at about the same time in America) the stationary pads are pivoted in accordance with theoretical requirements about a line in advance of their centre of figure. It has been found, however, that such blocks function satisfactorily, apparently contrary to theory, if pivoted about a central point, and this symmetrical arrangement makes the block suitable for running in either direction, a condition which has to be met in marine turbines.

In turbines which have end-tightened blading it is useful to have means for controlling the clearance when running. For this purpose the whole adjusting block is made capable of sliding in an outer housing between certain limits controlled by fixed liners. This enables the clearance to be increased before shutting down or when standing by for manœuvring, under which conditions some variation in temperature and relative expansion of rotor and casing may occur. This "pulling-up" gear as it is called is illustrated in Fig. 13.

Main thrust blocks are of the same type as the turbine thrust blocks, but of larger dimensions. In these, however, since the astern pads are on the other side of the collar from the ahead pads, any particular pad has only to function for one direction of rotation, and the theoretical location of the pivoting point is satisfactory. Generally, therefore, it would seem sound practice to adopt a central position for the pivot in turbine adjusting blocks, and the advanced position in main thrust blocks, which is the case in the turbines which have been illustrated. It is usual to adopt a load of about 250lb. per sq. in. on main thrust blocks and with turbines provided with dummies about 100lb. per sq. in. on the adjusting blocks. These are, however, very conservative figures in comparison with the loads allowed for land turbines, and in comparison also with the loads that have been demonstrated safe in experimental work.

Reference has already been made to astern turbines. Provision has to be made for driving the propeller shaft in the reverse direction for stopping the ship and for actual going astern. For this purpose there are turbines with blades disposed to function in a direction opposite to that of the ahead turbines. In the i.p. ahead turbine of Fig. 6 is shown an astern impulse wheel which operates as a high-pressure astern turbine in series with the low-pressure astern turbine shown in Fig. 7 at the exhaust end of the l.p. turbine. Except for the handing of the blades, and the fact that for the sake of economy of weight and space a lower degree of efficiency is accepted, which is of course justified by the comparatively small part that their efficiency plays in the overall economy of a voyage, astern turbines are in no wise different from ahead

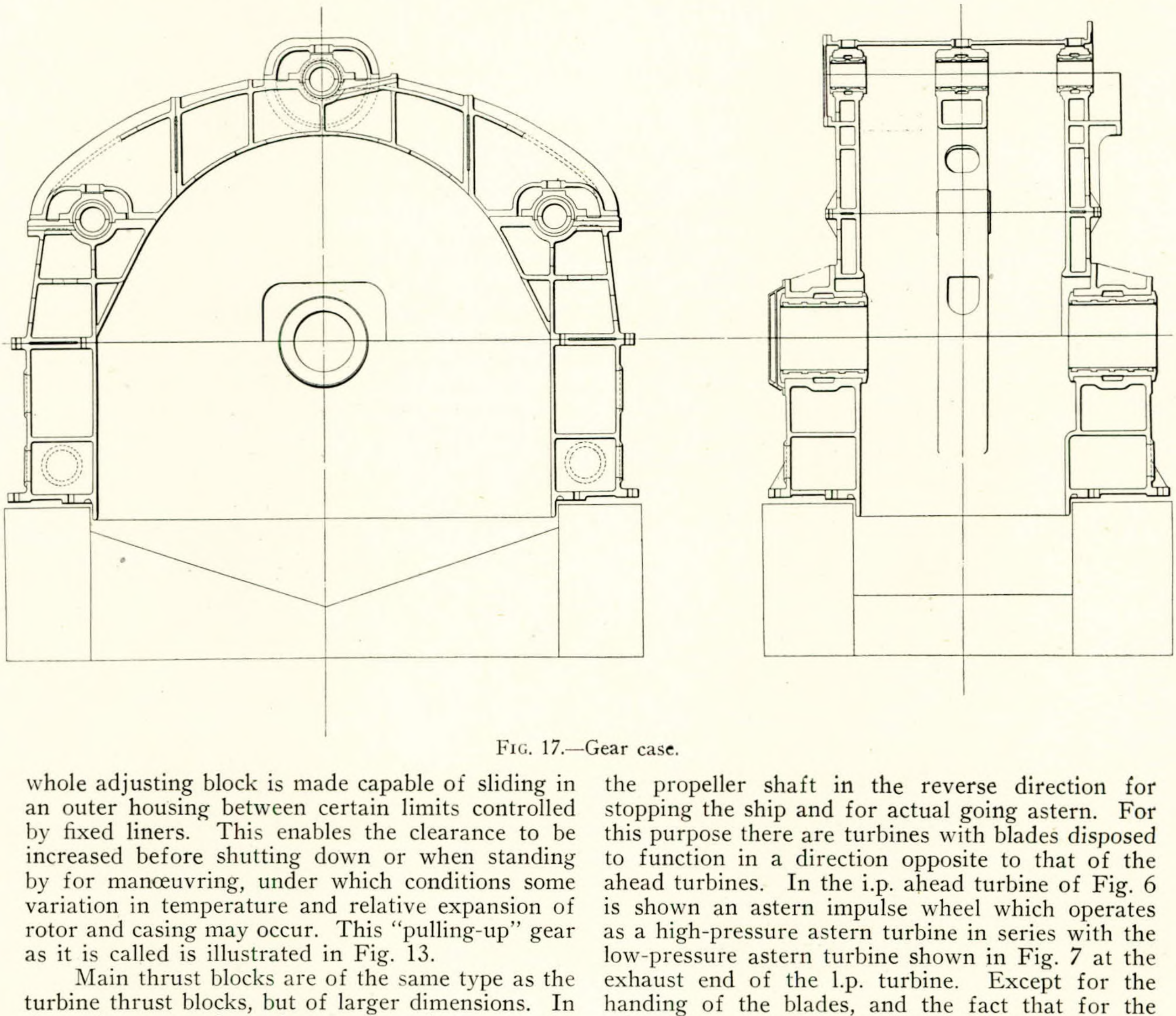
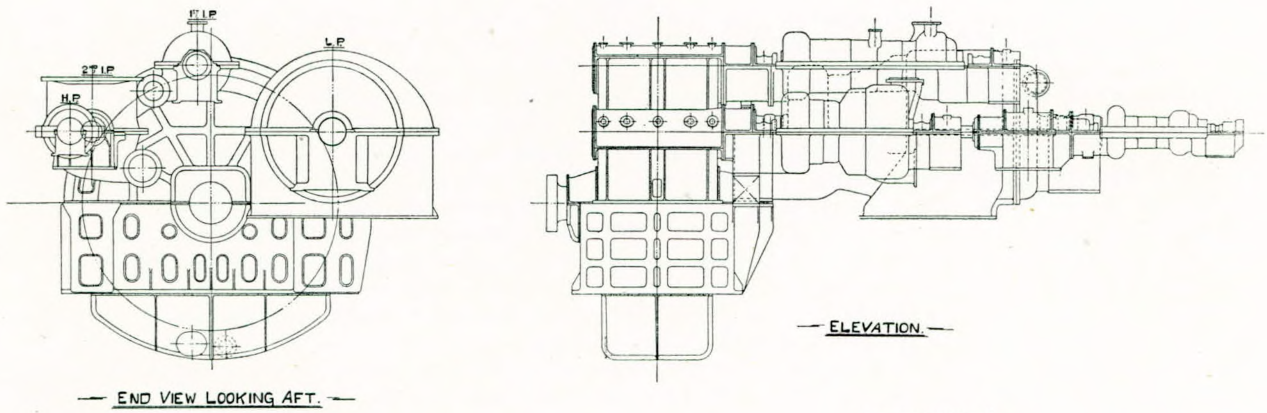


FIG. 17.—Gear case.



turbines. The steam is supplied to them by separate valves and pipes, and exhausts into the same condenser. The operation of reversing is merely that of closing the ahead valve and opening out the astern valve.

For installations of the class being described it is usual to provide for an astern power 70 to 75 per cent. of the full ahead power. This would give about 90 per cent. of the ahead revolutions, and an astern torque about 81 per cent. of the ahead torque. In a turbine, however, the capacity for steam, that is to say the amount of steam that can be passed for a given initial pressure, is practically independent of the revolutions, and during the first moments of reversal the full torque is immediately developed and is in fact considerably greater than the figure mentioned above owing to the fact that the force on the blades of a turbine increases as the blade speed is reduced. Fig. 14 is a typical curve of this variation of torque with blade speed, from which it will be seen that the value of the torque when standing is 2·8 times the value it has at a velocity ratio of ·85, and about 1·8 times the value it has at a velocity ratio of ·35, which is the velocity ratio corresponding to the above conditions for the astern turbine. So that if full steam is admitted to a standing astern turbine, the torque developed is $\cdot 81 \times 1\cdot 8 =$ say 1·46 times that of full speed ahead. If at the moment of admission of astern steam the turbine is actually running in the ahead direction the torque is considerably greater.

It will be seen that under these circumstances the astern turbine provides very effective stopping power. In stopping a ship the energy of the ship's motion and of the motion of the turbines has in some way to be absorbed. An astern turbine acts as a powerful brake, the energy is converted to heat in the steam which is absorbed in the condenser, which with its circulating water is capable of absorbing heat if necessary to the extent of ten times the rate of heat conversion representing the power of the machinery.

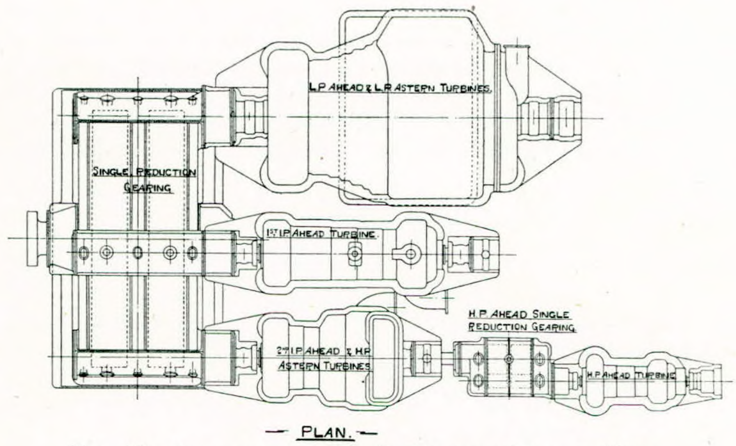


FIG. 18.—Arrangement with double-geared h.p. turbine.

Fig. 15 shows the double-claw coupling which connects the turbine rotor to the pinion of the gearing. Claws are formed on the rotor ends, or in a forging attached to a cone on the rotor end and similar claws on the end of the pinion shaft. These are connected by a sleeve having inwardly projecting claws. The outer ends of the rotor claws have a spherical surface on to which the inner surface of the sleeve bears with an easy fit. With a clearance at the claws of about $\frac{1}{64}$in. such a coupling accommodates a small amount of misalignment between the two rotors, leaves the pinion free to take up its natural position in its own bearings and also free to assume a balanced longitudinal position on the oppositely-handed helices of the gear teeth without transmitting any end thrust to the turbine rotor.

To provide this freedom for longitudinal motion is the primary function of the coupling. There is no premeditated misalignment. On the contrary the turbine and pinion bearings are carefully set in alignment. With changing temperatures and conditions of loading, however, it is desirable to have a small amount of lateral freedom such as the coupling provides.

The chief problem with the coupling is its lubrication. Oil is supplied to it from the general forced lubrication system, either separately or from the supply channels of the bearings on either side of it. When this type of coupling was first used it was found that it acted as a centrifugal separator and in the course of time the interior of the sleeve became filled with sediment accumulated from the oil, the remedy for which was found in providing small vent holes in the sleeve to promote a continuous circulation.

The gearing is shown in Fig. 16. It consists of three double-helical pinions gearing with a main wheel mounted on the propeller shaft. The pinions are supported between the oppositely-handed helices by central bearings. They are usually solid forgings of nickel steel, about 0·3 per cent. carbon with 3 to 5 per cent. nickel. They are sometimes oil quenched, but in the practice of the company with which the author is associated they are preferably only normalised, so as to avoid initial strains due to oil-quenching. It will be appreciated that with the surface cut up into separate elements to form the teeth, homogeneity is of supreme importance; also ductility is considered of more value than a high ultimate strength. The wheels are made with central parts of cast iron or cast steel and forged carbon steel rims shrunk on and secured by screwed pegs. The boss of the wheel centre is forced on to its shaft with a slight taper. For light construction the wheel is built up of steel end plates bolted to flanges on the shaft and to inwardly projecting flanges on the forged steel rim.

The outer surface of the pinion or of the gear wheel rim is turned concentric with its journals to the diameter of the tips of the teeth, which surface then becomes a datum surface for setting on the gear-cutting machine. The teeth are cut by the hobbing process in two portions of opposite hand at a helical angle of 30 degrees. The tooth surfaces are of the involute form with a pressure angle of $14\frac{1}{2}°$ and equal addendum and dedendum, that is to say the pitch line is at half the depth of the teeth.

Considerable advance has been made in gearing during recent years both as regards accuracy of cutting and quality of finish as a result of careful rectification of the gear-cutting apparatus. With the tooth pressures per unit length which are now normally adopted and given care in alignment to obtain a reasonably good distribution of bearing surface, present-day turbine gears can be relied upon for a long period of continuous service.

In many high-speed gears another form of tooth is adopted in which the pitch line is at the root of the pinion tooth, so that the pinion tooth surfaces are all-addendum and the wheel tooth surfaces are all-dedendum, which secures that the sliding of one surface over the other is always in one direction. This form has a pressure angle of $22\frac{1}{2}°$. It has, however, been found not to be suitable for low tooth speeds and is now only employed for speeds above 80 feet per second. With high tooth speeds teeth of this form are found to run successfully with higher tooth pressures than the ordinary type and have enabled reductions to be made in the length of face, in some instances to such an extent that it has been possible to dispense with a central bearing. The involute form is preferred to the epicycloid, because it has the advantages that correct engagement can be maintained with varying centre distance and, from a manufacturing point of view, that it can be cut with straight-sided hob teeth, the same hobs being suitable for both wheel and pinion and for any sizes of the latter.

For the assembly of the gearing, pedestals are required to carry the pinions and the wheels with their bearings truly parallel. They must therefore be rigidly connected to sustain not only the weight of the gear but the turning forces, provided with suitable feet to transmit these forces to the structure of the ship and completely enclosed to retain the oil. These requirements give rise to the struc-

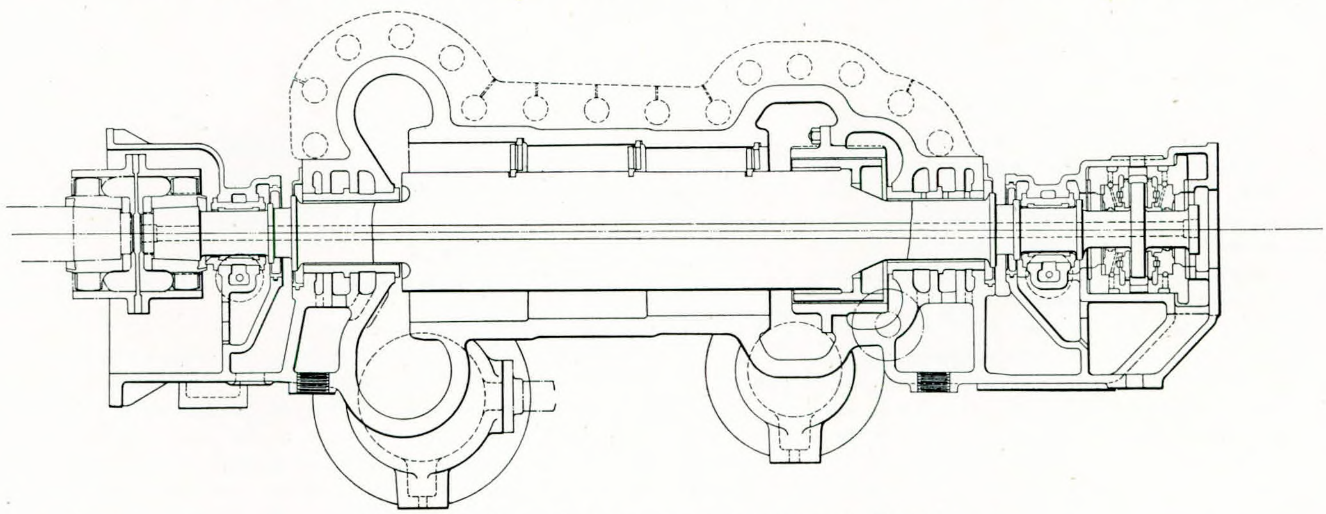


FIG. 19.—H.P. turbine of arrangement shown in Fig. 18.

ture known as the gear case, shown in Fig. 17. It is usually provided with a trough at the bottom to allow free drainage of oil from the teeth.

Oil is supplied to the teeth by sprayers connected up to the general pressure-lubrication system of the turbines, and consisting of a number of jets at intervals of about 3in., directed to the line of engagement, and lying within the common tangent planes to gear wheel and pinion.

The lubrication of the pinion and wheel bearings follows the same principles as that of the turbine bearings. In the pinion bearings however the resultant pressure is not always vertical but varies in position with different directions of running; the supply channels are therefore located to suit, so as to be remote from the point of pressure under all conditions of running.

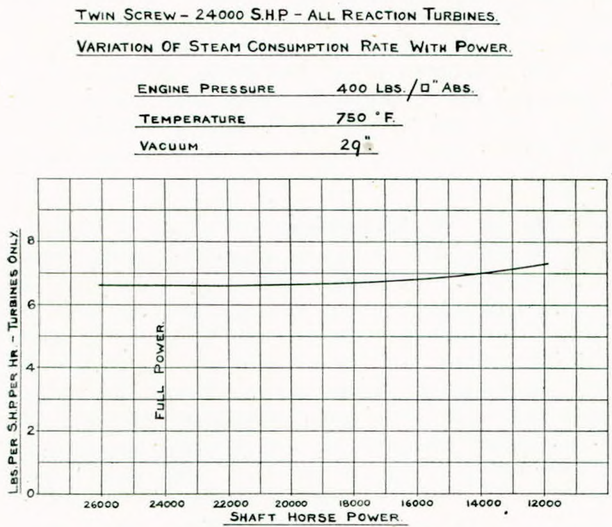


FIG. 20.—Curve of steam consumption rate.

Reference has already been made to the main thrust block. The thrust housing is usually separate from the gear case attached to its own seating but is sometimes incorporated with the gear case, in which latter event the thrust collar is formed on the gear wheel shaft instead of on a separate thrust shaft attached thereto.

The three-turbine type of arrangement illustrated has been widely adopted in many important vessels of various powers with steam pressures up to 400lb. per sq. in. For higher pressures it is considered desirable to have the high-pressure portion of the expansion carried out in a small additional turbine.

In the first high-pressure installation it was chiefly as a measure of precaution that the experimental part of the new installation was confined to an additional turbine which could be removed if found necessary and still leave the remainder of the installation suitable for the propulsion of the ship, but it is now recognised that such an arrangement with the h.p. turbine at high revolutions and of small diameter has distinct advantages. Confining the high temperature to a rigid turbine of small dimensions avoids distortion troubles and makes an easier problem of the design of the horizontal joint flange and bolts.

Fig. 18 is an example of a set designed for a pressure of 500lb. per sq. in. at 750° F. In the case illustrated the high-pressure turbine rotor is connected by gearing to the rotor shaft of the 2nd i.p. turbine. It is of $12\frac{1}{2}''$ diameter and runs at 3,600 r.p.m. The power is so distributed among the turbines as to give equal power to each of the three pinions gearing with the main wheel. Fig. 19 shows the high-pressure turbine. The bolts in the horizontal joint flange are of $2\frac{1}{2}''$ diameter, and their mean stress under working pressure is about 7,000lb. per sq. inch.

A turbine system such as is illustrated in Fig. 4 with a steam pressure of 400lb. per sq. in. abs., temperature of 750° F. and vacuum of 29″ Hg. has a steam consumption of 6·6lb. per s.h.p. turbines only at full power. Fig. 20 is a curve of the steam consumption rate at varying powers from full down to half power. This flat curve of consumption rate is obtained by the device referred to on page 5 in which the initial stage is bypassed for full power. The consumption at full power, viz. 6·6lb., corresponds, on the basis of a boiler efficiency of 87 per cent. and fuel of a calorific value of 19,000 B.Th.U. per lb., to a fuel consumption of ·54lb. of oil per s.h.p. With bled steam feed heating this can be reduced to ·51lb. The amount to be added for auxiliary purposes depends upon the system of auxiliary driving adopted, but assuming an efficient system of steam-driven auxiliaries having economical utilization of the auxiliary exhaust steam, the propulsive power can be produced on a total consumption of ·58lb. per shaft horse power.

Discussion.

Engineer Vice-Admiral Sir George Preece, K.C.B. (Member), opening the discussion, congratulated the author on the lucidity and value of his paper and said he would confine his remarks to examples of difficulties experienced in practice. These difficulties were no reflection on the designers, because they were definitely such as could not reasonably be expected to be foreseen.

In the first case some difficulty was experienced with vibration in H.M.S. "Acheron". The first theory put forward to account for it was that there was some shifting of the joint at the forward end of the rotor. Fig. 9 illustrated the point. In this illustration it would be seen that with the drum type of rotor the forward end carrying the impulse wheel was a separate forging. It was suspected that that was shifting and putting the rotor out of balance. Differences of opinion existed as to whether it should be at the forward or the after end. If it was put at the after end then it had to transmit the torque and that put load on the pegs and on the shrink. If it was put at the front it was in the place of high temperature, but there was the advantage that the somewhat complicated forging was a separate piece. They eventually decided that it was probably not due to a shift in the joint, but to the fact that when the turbine got hot the front end of the feet curled up. He thought there was little doubt that when it did this it brought up the oil-retaining strips of the adjusting block, so that these just touch the shaft, and that this was the cause of the vibration experienced. It was not the sort of thing one would expect, but they had ample evidence that that was the trouble.

In the earlier post-War destroyers trouble was experienced with stripping of the cruising stages, i.e. those stages which were by-passed at full power. He had been sent to investigate the trouble and when the turbine was first opened up for inspection the one thing of which he was certain was that the stage had been extremely hot. Fig. 5 illustrated a turbine with two by-pass stages. Great care must be taken—it could be guarded against in the design—not to open these by-passes too much, as it was thus possible to level the pressures and by-pass the first stage completely. It was just the churning of the steam in that space which produced these extremely high temperatures.

Turning now to gearing, he would say on behalf of the Admiralty, which was often subjected to criticism, that it was largely due to the pressure brought to bear by the Admiralty on gear manufacturers that the present admirable accuracy had been achieved. Admiralty requirements in this respect were not popular in the early stages, but it had been generally admitted that they were well meant and the result had been good. He did not think it followed that the more accurate a gear was the more silent it would be. If every tooth gave precisely the same note it might result in a noise which was distressing. It was an interesting point that gears might become supersonic if speeds increased.

The author had called their attention to the excellent drainage provided in the l.p. turbine, and here again he (the speaker) recalled a somewhat mysterious thing that occurred in one or two ships. Reference to Fig. 7 would show the l.p. astern turbine facing the end of the l.p. ahead turbine. He was not disclosing any secrets in saying that the Admiralty did not permit such a magnificent space between the turbines. There was little doubt that the damage which he had in mind was caused by the fact that when going astern water from the astern turbine was shot over and became entrained in the moving blades at the end of the ahead and pumped up into the ahead turbine. Of course one would expect the last rows of blades of the ahead to go, as being the first to meet the blow, but in their experience it was not so. Failure actually occurred further up in the turbine where the blades were possibly somewhat weaker. Another point was that the turbine illustrated was perfectly horizontal, whereas in a ship this was not so. Drainage holes where the stages changed diameter provided effective drainage into the condenser.

They were now approaching the stage when it might be necessary to change from two to three turbines in the interests of economy, and he would be glad if the author would care to express an opinion regarding the pressure and temperature at which it was desirable to change from two to three turbines, and also the increase in weight involved in the change. This might be expressed as a percentage of the two-turbine installation.

He was very interested in the author's statement that the all-addendum gear was not so good below 80ft. per second as it was at speeds above that figure. Had the author any theory which would account for that?

Mr. A. W. Richardson (Member of Council) said that he was specially interested in the paper in its relation to boilers and superheaters. The author had stated that it was advisable to limit the temperature of the steam to the turbines to about 750° F. unless special material was used for the last passes of the superheater. Would the author state the maximum temperature that standard turbine blading was suitable for, because it was quite a common practice for ordinary mild steel superheater tubes of, say, 0·10 to 0·15 per cent. carbon content to be used for steam temperatures up to 850° F., such steel having a tensile strength of 28-30 tons per sq. in.?

On a modern high-pressure marine installation a drop in steam temperature between the superheater outlet and the turbine stop valve would be in the neighbourhood of 12° F., but if it be assumed that the temperature drop was, say, 15° F. with ordinary mild steel superheater tubing, it would be possible to work up to a temperature of 835° F. at the turbine, which was a considerable increase on the temperature referred to by the author. Possibly the author had in mind the question of fluctuation in steam temperature. Even so, in limiting mild steel superheater tubing of the quality mentioned above to a steam temperature of 850° F., there was still a fair margin in hand for any emergency fluctuation in steam temperature. It was admitted, of course, that the temperature of the tube was very much higher than the temperature of the steam. He knew of an instance where the steam temperature in a superheater fitted with mild steel tubes had been in the neighbourhood of 900° F. No trouble had been experienced and it was still in use, though of course such practice could not be recommended.

With regard to the author's reference to special materials being used in the last passes of the superheaters, this was common practice. In fact, superheaters for land work were very often built up of tubing of various compositions, the sections being butt welded together. In some instances superheaters were made in sections and expanded into drums. Where molybdenum and chromium steel was used, the superheater tube end usually was butt welded to a mild steel tube to enable expanding to be easily effected in the superheater box. For temperatures above 850° F. it was usual to use alloy steel tubing of the 0·5 per cent. molybdenum and 4 to 6 per cent. chrome variety.

Mr. S. A. Smith, M.Sc. (Member) said he had thought that the author would give some data based on his unique experience to enable them more fully to understand how to design a set of turbines. Probably the author had left the members to ask him the questions necessary to draw this data from him.

In connection with the reaction turbine, the λ value which depended on the number of rows of moving blades, the mean blade ring diameter and the revolutions per minute, was an important factor in design. The λ value was an expression of the ratio of blade speed to steam speed and by increasing the number of rows of blades, the steam speed, for a given heat drop, was reduced, thereby increasing the velocity ratio and raising the efficiency. It would therefore appear that to obtain a flat curve of efficiency at reduced powers λ must remain sensibly constant in order to maintain a constant velocity ratio. To obtain this a greater number of blades had to be in use at reduced powers, a portion of which were by-passed at full power. Were his (the speaker's) deductions in this matter correct? The efficiency increased rapidly at small values of λ and more slowly at values above 140,000. The higher this value the higher the cost of the turbines. What did the author consider to be the economical value to adopt in normal practice?

The author stated that there was a prospect of substantial gains with reheating at pressures from 750 to 1,500lb. Re-superheating would add to the cost, complications and space occupied by the plant. What were the percentage gains for 750 and 1,000lb. pressure and what was the period of time required for the additional cost to be paid off for, say, a 24,000 s.h.p. installation.

A few years ago he looked into the problem of reheating for pressures up to 1,500lb. His investigations, with three-stage feed heating, indicated a small gain in thermal efficiency, the principal advantage being the higher dryness fraction of the steam in the l.p. turbine with consequent increased life for the blading.

In the light of present experience, he agreed with the author that it was not advisable at present to exceed a steam temperature of 750° F. to ensure reliability and long life for the boiler superheater tubes under continuous running conditions for passenger and cargo liners. Did the author consider fluctuations of pressure and temperature in a superheater tube had any effect on the life of the tube?

The author stated that the values to be adopted for the efficiency and the discharge coefficient rested in the last resort upon experimental determinations. It would be of considerable assistance if the author could give data from which these important factors in design could be determined.

In connection with blading material, the practice appeared to have changed in the past five years. To quote two vessels both having a steam pressure of 425lb., 725° F. at the h.p. turbines, the materials in 1932 and 1937 were as follows :—

	1932.	1937.
H.p. impulse ...	—	Hecla A.T.V.
H.p. ahead reaction	Monel	Monel
I.p. ahead reaction	Manganese copper	Stainless iron
I.p. ahead last stage	70% copper 30% zinc	Stainless iron
L.p. ahead first 6 stages	70% copper 30% zinc	70% copper 30% zinc
L.p. ahead last 5 stages	70% copper 30% zinc	Stainless iron
H.p. astern impulse	Monel	Hecla A.T.V.
L.p. astern impulse	Monel	Hecla A.T.V.
L.p. astern reaction	Stainless iron	Stainless iron

He thought it would be agreed that the 1937 practice was superior to that of 1932, but did the author not think it would be preferable to adopt say stainless iron or Monel metal throughout for the reaction blading and Hecla A.T.V. for the impulse blading, and that the additional cost involved would be more than repaid by the reduction in upkeep costs?

With regard to end-tightened blading, what would the author consider the safe limit of running clearance, and how did the efficiency fall off as this clearance was increased?

In connection with survey work with the present design of marine geared turbines, the nozzle box had a large casing and pipe joint and certain large steam pipes had to be removed before the h.p. turbine cylinder cover could be lifted. Was it possible to design the turbines so that the nozzle chest and all pipes between turbines were below the horizontal joint, thereby giving a clear lift to each turbine cylinder cover which would considerably facilitate survey work, especially when it was remembered that many vessels only had 7 to 14 days in port for this work to be carried out?

He would be glad if the author from his wide experience would state what he considered was the safe limit to adopt for the "k" value when designing gearing for continuous full power running, where

$$k = \frac{\text{load per lineal inch in pounds}}{\sqrt{\text{pitch circle diameter of pinion in inches}}}.$$

Engineer Rear-Admiral W. R. Parnall (Member) said that in Fig. 20 it would be observed that the curve of consumption showed the best figure at full power. He had been led to believe that some owners desired to operate their ships at the highest possible efficiency when at half power or in that region, the additional power being available when some change in conditions demanded increased speed. Would the author give his views on this aspect of the marine steam turbine, and say whether a practicable design could be developed on these lines?

Dr. S. F. Dorey (Vice-President) said that the paper was excellent in that it dealt with the subject in a most readable form which was suitable for marine engineers and particularly those at sea. While, however, it contained no mathematics and did not go deeply into the underlying principles of design, it referred to the many problems which had in recent years been met and, in most cases, solved.

The author had referred to the special materials known as Monel metal and stainless iron, and he thought it would be well to qualify those remarks with some particulars of the physical properties of the materials. A case had recently been brought to his notice where a vessel came in with several broken turbine blades and doubt was expressed of the condition of the unbroken blades. The blading was of the stainless iron type and it was found on testing the blades that whereas the hardness of the good blades fell between 265-274 Vickers Hardness Number, that of the broken blades fell between 193-196 V.H.N. Microscopic examination of the blades showed that a number had been incorrectly heat treated. The employment of special alloy steels necessitated care in heat treatment and in this respect the impact test would be found of value for indicating the correctness or otherwise of the heat treatment.

Another point in connection with design was whether at the high pressure end it was usual to design the blade so that the centre of gravity of the section coincided with the axis drawn through the centre of the root. In a recent case of failure brought to his notice this had not been done, and as a result additional stresses had been set up of a bending nature and failure had taken place at the root of the blades due to stress concentrations arising from machining, and unsatisfactory finish.

Could the author give any information in regard to the recent development of utilizing hollow blades in order to obtain strength combined with lightness and to what extent they had been used?

Mention had been made by a previous speaker (Mr. A. W. Richardson) of the temperature of 750° F. being the appropriate maximum temperature for marine work. Would the author say whether this temperature was mainly a boiler problem and not a turbine one? With regard to superheater tubes being heated to 900° F. he did not think this need cause anxiety if the period was short. In fact it might do the tubes good. The main point was the time they were subjected to this high temperature.

He thought the methods employed for the testing of the materials for rotors was worthy of inclusion in the paper, particularly in the case of those which rotated at high speed. The author had made a statement in regard to the type suitable for surface speeds up to 400ft. per second. In changing from the drum type of rotor to the disc type there was a stage in which certain manufacturers utilized plain carbon steel rolled plate. This plate was probably 3in. or more in thickness and the discs were cut from this material. The rolled plate or slab would naturally have different properties in different directions according to the amount of work put into the plate during cross rolling, but the rotor in service was stressed equally circumferentially. He thought that a rotor of this type could be used for speeds up to 600ft. per second, but he would like to have the author's confirmation of this view.

The question of blade material, particularly in regard to high temperature and erosion, had been mentioned. One thing that had to be considered was the erosion of the casing, and that brought in the question of cast iron. It was to be noted that use of cast iron was advocated for temperatures up to 500° F. and he would like to know if the cast iron for these casings was of any special composition? Lloyd's Register had a rule that where a temperature above 425° F. was employed, cast iron must not be used. Now this specified temperature had been in existence a considerable time and it was naturally hoped that someone would come along and ask for an increase. Now here was a temperature of 500° F. specified and information of satisfactory service at this temperature should be the means of raising the limit above the 425° F. previously mentioned. This should be possible with the improved qualities of cast iron now available.

He would like further information of the author's experience in regard to the tightness of joints at the high-pressure end. Had any special difficulties been experienced with the casing joints and the steam pipe joints at the turbine. Were metal-to-metal joints employed for high pressure and temperature work and had it been found necessary to adopt any special form of joint?

Gearing was naturally linked up with modern turbine design and amongst still recurrent troubles flaking of teeth might be mentioned. Perhaps the author would give us his views on remedies for this. One regretted that a little more space had not been devoted in the paper to tooth form—in particular the enveloping tooth type, with some results of actual running experience. Another point was the efficiency of different types of gearing in service. One of the claims of the manufacturers of turbo-electric propulsion was that the efficiency of this type of drive was something of the order of 92 to 93 per cent., and that as time passed this efficiency would be maintained. They also claimed that in time the efficiency with machined gears would be below the figure which they claimed they could maintain, and it would be interesting if the author had figures which would help in deciding whether this was the case. In that respect particulars of the efficiency for different types of gearing in service would be of special use.

With regard to the astern portion of the turbine, they had been given figures showing the large torque brought on the turbine to bring it up. Of course it took considerable time to switch from full ahead to full astern. Would the author give information in regard to the time taken from full speed ahead to full speed astern with different powered installations and various speeds?

Fig. 18 seemed to be rather an interesting development, and possibly this was the best arrangement for double-reduction geared turbines which would prevent a recurrence of the many troubles which had been experienced with double-reduction gears some 15 years ago.

In conclusion he would say that the paper did show what sound and yet not conventional design had done in recent years and thanks were due to the author for an interesting and instructive paper.

Engineer Rear-Admiral W. M. Whayman, C.B., C.B.E. (Vice-President) said that two years ago he read a paper before The Institution of Naval Architects in which he had expressed the view that at that time the boiler part of a marine machinery installation was a little ahead of the turbine section in its ability to make use of higher temperatures. He had been extremely interested to hear the point raised in the discussion and the author's reply thereto. The balance or equal capability of the boiler and turbine to use both higher pressures and higher temperatures was very important and should exercise the ingenuity of every engineer interested in marine propulsion by means of the water-tube boiler and turbine machinery, and continued co-operation between the designers, manufacturers and users of turbines and boilers would result in further advancement.

He then proposed a hearty vote of thanks to the author, not only for the paper which was a record of the progress of marine turbine machinery but for what he hoped might be a continued series of reports of steady progress. This was accorded with enthusiasm.

By Correspondence.

Mr. J. Hamilton Gibson, O.B.E., M.Eng. (Vice-President) wrote that he was particularly interested in the author's reference to cast steel high-pressure casings and end-tightened blading. In the early days, when the first 33-knot Tribal destroyers were building, he remembered attending a conference of contractors, with Sir Charles Parsons presiding, to discuss the pros and cons of cast steel versus cast iron for the h.p. turbines of these vessels. It was hoped to effect a slight saving of weight by using cast steel. At that time blade tip clearances were kept down to a minimum; moreover it was before thinning of the tips became a practice. Sir Charles dealt with the proposal in his usual intuitive manner. He pointed out that cast steel would distort more than cast iron under heat and that therefore it would be necessary to provide for an extra, say, 5 per cent. tip clearance. That would mean 5 per cent. more boiler capacity to make up for the extra tip leakage, which would more than swallow up the hoped-for saving in weight, and on balance it was decided to adhere to cast iron. Had end-tightened blading been then available, the question of distortion as affecting tip clearances would not have arisen and cast steel casings would have gone in as a matter of course.

Mention of distortion reminded one of another difficulty which presented itself, viz. "growth" of cast iron casings under the action of alternate heating and cooling. If both top and bottom halves of the casing grew equally, well and good; but sometimes there was a marked difference and new fitted bolts of larger size were required after the original bolt holes had been faired and reamered out. It would be interesting to know whether this difficulty had been entirely overcome, and how.

The writer noticed that Fig. 13 showed a type of adjusting block that was now practically obsolete. Ball and socket abutments which at first appeared ideal turned out to be a delusion and a snare. The pad carrying rings slid down in their sockets and consequently the bottom pads became overloaded. Originally it was expected that the axial thrust would push the carrier ring back into position, when the load would be equalised between top and bottom pads, but in actual fact this did not occur, and the ball and socket device was abandoned in favour of flat abutments as indicated in Figs. 11 and 19. The same remarks applied to main thrust blocks, best modern practice comprising square abutments and line-pivoted instead of point-pivoted pads.

As the author pointed out, centre-pivoted pads were the simplest for turbine adjusting blocks where the loads were comparatively small, and offset pads the best for marine thrusts loaded to 250lb. per sq. in. and upwards. In the latter case an exception occurred when trailing, the collar then revolving *ahead* on the *astern* pads. The load, however, was less than half the normal and the pads functioned quite satisfactorily, tapered films being ensured by side leakage of oil and decreased viscosity of the lubricant as it traversed the pad from leading to trailing edge.

Mr. W. W. Marriner, B.Sc. (Member) wrote that there was one important difference between a marine and a land turbine; in the latter the designer had only himself to blame for any vibration, while the designer of a marine turbine might have to allow for vibratory forces arising from causes apart from the turbine itself. The effect of these impressed vibrations on the rapidly moving blading gave rise to large periodic forces and if they should happen to synchronise with any part of the turbine such as the rotor or the blading the results might be serious. The velocity of a turbine blade was so high that it resisted with considerable force any attempt to make it move in a sinuous path ever so little out of the true circle. The excellent homogeneity of material of to-day and the accuracy of machining actually made structures more sensitive to periodic impulses.

With regard to the temperature limit for the superheated steam, Mr. Cook drew attention to the effect which variations in the feed temperature had on the superheat. It was also found that the superheat was affected by the percentage of excess air in the furnace, so that at sea it was advisable to have a larger margin of safety than on land, as if any of the joints exposed to superheated steam either in the boiler, the piping or the turbines were simply face-to-face joints (i.e., the joint being made by expanding or my means of bolts drawing two faces together) then care must be taken that at no time should the temperature of these expanded joints or bolts rise above the annealing temperature of the material. Ordinary mild steel was said to have as high an annealing temperature as any alloy steel. Annealing relieved the internal strain in the bolts or tubes and consequently the joint was no longer tight. This pointed to a more extended use of welded joints if it was hoped to work at higher degrees of steam temperature.

Considering the heat engine (so ably defined by the author as the whole combination) which was in use at so many of the large land power stations, it was remarkable for its high thermodynamic efficiency and it rather made one wonder if the line of development of marine propulsion was not in the direction of extending the system of putting on board ships an electric energy producing plant similar to what was used in a power station and driving the propellers by means of electric motors. In such an installation the turbines would be always running, which would avoid that dangerous period which occurred when turbines were stopped during manœuvres and unequal heating was liable to take place between the top and the bottom of the turbine or rotor. Also the turbines themselves would be definitely separated from any disturbing vibrations which might be transmitted through the propeller shafting.

The Author's Reply to the Discussion.

The author, in reply, said he was interested to hear from Admiral Sir George Preece the conclusions which were reached as to the real cause of the vibration troubles in H.M.S. "Acheron", and that this was not as at first suspected due to relaxation of the shrinkage fit of the rotor drum. In reference to Fig. 9, on the question whether the separate end shrunk into the drum should be at the forward or at the after end there was still some divergence of opinion. The heating-up of by-passed stages when running idle was avoided by providing for a small circulation of steam through them under this condition to absorb the heat generated. This heat was very small in proportion to the total power being developed, but if not absorbed or dissipated would naturally cause the blading to reach a high temperature. In some at least of the vessels to which Sir George Preece had referred, the author believed that steam was led to the full power inlet direct from the main steam pipe instead of by a by-pass branch from the cruising stage inlet, thus leaving the cruising stage in an atmosphere of stagnant steam. An adequately restricted by-pass would ensure a drop in pressure across the blading of the cruising stage, and a circulation of steam through it. The author freely agreed that the specifications of the Admiralty had set gear-cutters a high standard of work which had been most beneficial to the production of accurate gears. He did not believe that extreme accuracy would introduce notes from the teeth themselves. The frequency of tooth contacts was of the order of 3,000 per second, but the oblique disposition of the teeth and the chamfering of the ends to avoid shock at the beginning and end of engagement prevented any note arising from this source. As regards the point at which it was advantageous to change from two to three turbines the author considered this point had already been reached. For pressures above 250lb. per sq. in. there was a distinct gain both in weight and economy with three turbines. Unfortunately, in vessels of large power and restricted dimensions it was difficult to accommodate three turbines side by side forward of the gearing so that the three turbine design involved a small increase in overall length. The only explanation he could offer for the difference in behaviour of the all-addendum gear at high and low speeds respectively was that a certain speed of rolling was necessary for an adequate entrainment of the oil towards the point of contact of the teeth.

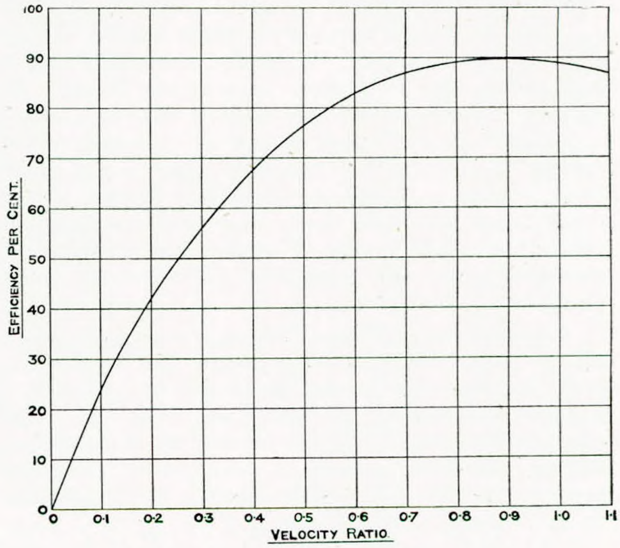
In reply to Mr. Richardson, whilst it was inevitable that the temperature of the superheater tubes should be considerably in excess of the temperature of the steam to which they were transmitting heat, in the turbine the converse was the case; all parts of the turbine would be at a temperature lower than that of the steam, also the highest temperature in the turbine would be confined to a small portion of it and in the case where there was an impulse wheel it would be confined to the nozzle box. So that there might be a difference of at least 150° F. between the temperatures which had to be considered for superheater and turbine respectively. The blades of the turbine were already made of a material superior to mild steel as regards its creep limits, also the stresses in the blades at the high-pressure end of a turbine were extremely low, and there was no difficulty in keeping the stresses in the turbine casing low by suitable design. In the author's opinion the question of the temperature limit was definitely at present not one of turbine design but of superheater design. It was necessary to provide a margin for fluctuation of steam temperature as well as for the excess of the tube temperature above that of the steam. One important cause of such fluctuation arose from the use of bled steam for feed heating, which increased the quantity of steam being generated and passed through the superheater. When the supply of steam for feed heating was stopped, as when changing over to astern, the quantity of steam passing through the superheater tubes was reduced and the steam temperature immediately rose. It became a question whether in marine work it would not be more profitable to abandon the practice of bleeding steam for feed heating, adopting instead a higher degree of superheat in the first place, since less margin was then required, and using an economiser to heat the feed water. It was a matter worth careful investigation.

Mr. Smith referred to what he called the λ value of a turbine. This was presumably the quantity better known to turbine designers as the K value, which was the sum of the products $D^2R^2/10^9$ for all the pairs of rows which constituted the turbine. It was not, however, a co-efficient, but must be considered in relation to the available B.Th.U. per lb. of steam. The velocity ratio, if constant throughout the turbine, would be equal to $\cdot 875\sqrt{\frac{K}{\text{B.Th.U}}}$, so that this last might be taken as a convenient expression for the average velocity ratio of the whole turbine. The use of K as a co-efficient in the sense referred to by Mr. Smith was therefore limited to a certain range of conditions. The proper criterion of efficiency was the velocity ratio, which for a high standard of efficiency was as stated in the paper of the order of ·8 to ·85. Nor was it possible to state for general application what was the most economical value of K. In direct turbines it was about 100 or even less. With geared turbines with freer choice of revolutions it was frequently as much as 450. In Mr. Smith's notation this would correspond to a λ value of 450,000.

As regards the economy of reheating with 1,500lb. initial pressure and reheat to 780° F. after expansion to 300lb. allowing for a drop of 30lb. in the reheater and pipes, there should be a gain of about 10 per cent. in comparison with an initial pressure of 400lb. without reheat. This gain was not entirely due to the reheat but partly to the higher initial pressure which reheating made it possible to use. For a vessel of 24,000 s.h.p. this represented a saving of 15 tons of fuel per day, say 3,000 tons per annum or about £4,000 per annum, which should give quite a good return on the first cost of reheater and fittings.

Small fluctuations of pressure and temperature of only a temporary nature would probably not have any serious effect on the life of superheater tubes. For a temporary rise in temperature the factor of safety might be considered in relation to the reduced ultimate strength at the temperature reached rather than in relation to the creep limits at that temperature.

Curves of efficiency experimentally determined for reaction blading had already been published. Fig. 21 showed the relation between efficiency and



EFFICIENCY OF STANDARD REACTION BLADING.

FIG. 21.

velocity ratio for standard blading. The discharge co-efficient for a series of rows of blading was about 1·03. This took into account increase of discharge by carry-over on the one hand and decrease of discharge on the other hand by friction.

Mr. Smith referred to a change in practice as regards blading material. Increased experience with stainless iron had led to its more general adoption. It was usually fitted in the assembled sector method, because this enabled annealing to be carried out after all the manufacturing processes were completed. Even with low carbon content a slight degree of hardening took place in stainless iron with processes involving application of heat such as brazing. Monel metal was used for the high temperature blading of the river steamer "King George V", but the merits of stainless iron were not at that time known.

As stated in the paper, brass blading was quite satisfactory when the conditions as regards temperature and stress permitted of its use. There was no application of heat during the building up of impulse blading and this type of blading was therefore most satisfactorily made of stainless steel or Hecla A.T.V.

As regards the minimum clearance of end-tightened blading, this blading could be safely run almost in contact, and frequently was in land turbines. Where, however, as was usually the case in marine work, the end thrust was in the direction of steam flow, the oil clearance of the thrust block made a difference between the maximum clearance with the rotor hard aft and the minimum with the rotor hard forward, and in order to avoid contact in the latter position, the running clearance which was that of the former position was one or two thousandths of an inch in excess of the thrust block clearance.

For an all-reaction turbine there was no serious difficulty in arranging the steam connection to the h.p. turbine on the bottom half of the casing. To arrange nozzles on the bottom half, however, made a somewhat complicated design of that end of the turbine which for high-pressure work it was desirable to keep as simple as possible. In any case it could be so arranged that it was only necessary to remove a small length of steam pipe. All intermediate pipes and emergency connections were made on the bottom casing.

In reply to Admiral Parnall it was quite possible to extend the by-pass principle and further reduce the consumption at half-power if required at a slight sacrifice of efficiency at full power. The consumption at half power shown by the curve in Fig. 20 could be reduced to about 6·9lb. per s.h.p. hour by this means, possibly with a slight addition to the weight of the turbines.

In reply to Mr. Gibson the use of spherical abutments for thrust blocks and adjusting blocks had now been almost generally abandoned. They were now made as shown in Figs. 11 and 19 rather than as they appeared in Figs. 5 and 13. As regards the phenomenon of "growth" of cast iron, it was now the usual practice to avoid the use of cast iron in turbine construction in all parts where there was likely to be a temperature in excess of 450° F., and with this precaution no trouble from this source should be encountered.

The author appreciated Mr. Gibson's agreement with the practice of central pivotting for the pads of turbine adjusting blocks and off-set pivotting for those of main thrust blocks. The fact that the latter worked satisfactorily when trailing, under which circumstances the off-set was on the wrong side of the centre, was confirmation if any confirmation were needed of the soundness of the practice of central pivotting for the pads of turbine adjusting blocks.

The author also appreciated Mr. Marriner's remarks regarding the effects of fluctuation of temperature upon the superheater. Mr. Marriner advanced further reasons for anticipating more fluctuation in temperature in marine work than in land work, emphasising therefore the necessity for a greater margin in the working temperature, and his remarks as to the possibility of the annealing effect of a temporarily high temperature relaxing the initial tension of expanded or bolted joints appeared to the author to deserve the most serious attention. It was hardly possible within the limits of a reply to a discussion to deal with the merits and demerits of electrical propulsion. There was no reason why high thermodynamic efficiency should not be realized in marine turbines without the losses of electric transmission.

In reply to Dr. Dorey the physical qualities of stainless iron and Monel metal were controlled by the following specification for composition and for mechanical tests applied to the raw material in the annealed condition :—

Stainless Iron.—Carbon ·07 to ·12, chromium 12½ to 14, manganese 0·3 maximum, nickel 0·5 maximum, silicon 0·25 maximum, ultimate 30/40 tons, yield 60 per cent. of ultimate. Elongation not less than 25 per cent. in 2″; reduction of area 50 per cent.

Monel Metal.—Nickel 68 per cent., copper 29 per cent., remainder chiefly manganese. Ultimate 30/40 tons, yield not less than 17½ tons, elongation not less than 25 per cent. in 2″; reduction of area 40 per cent.

Impact tests were not usually specified for these materials, but it should be observed that in the case of stainless iron any irregularity due to heat treatment during processes of building up the blading was removed by final annealing of the sectors at a temperature of 750° C.

Dr. Dorey had referred to an instance of blade breakage attributed to the material having a low hardness number. One would however expect the contrary to be the case, there being a greater possibility of brittleness with higher hardness number. Whilst it was easy to understand Dr. Dorey's point that failure at the root of a blade might be due to stress concentration arising from poor machining and unsatisfactory finish, it was difficult to see how this was related in any way to the position of the centre of gravity of the root. Where the blade was integral with its root it was best for the root to be attached to the convex side of the blade.

For the speeds and stresses hitherto adopted in marine work, hollow blades were not required.

On the question of temperature the author would refer Dr. Dorey to his replies to Mr. Richardson, Mr. Smith and Mr. Marriner in the present discussion.

He agreed with Dr. Dorey that turbine discs should be made of forged material rather than rolled slabs, in order that they might have the same elastic properties in all radial directions. For speeds as high as 600ft. per second a plain disc was hardly suitable; it was necessary to shape the disc in such a way as to obtain a more uniform distribution of the stress. The temperature at which the phenomenon known as growth in cast iron occurred was about 500° F. The adoption of a lower limiting value than this for the use of cast iron was, the author believed, merely due to the desire to have a margin on this figure. As regards the effect of the use of special grades of cast iron, the author understood that the phenomenon occurred at about the same temperature, only varying in rate of growth.

For high-pressure work metal-to-metal joints were preferred. For flanges of steam pipes the bolts should be made with a plus thread. This question was thoroughly explored in connection with the high pressure installation of the river steamer "King George V" and no difficulty found in keeping the joints tight.

The occasional occurrence of flaking of gear teeth to which Dr. Dorey referred was probably due to local concentration of load occurring before the tooth surfaces had become thoroughly bedded. As regards the efficiency of gears and the claim which Dr. Dorey stated was made by manufacturers of turbo-electric propulsion plant that mechanical gears fell off in efficiency as time passed, no doubt the authors of this statement would like it to be true. Unfortunately for them it was not. Numerous geared turbine installations could be mentioned which had retained their full efficiency after many years of service. The transmission loss in mechanical gearing was of so small an order that it was rarely considered necessary to measure it. In experimental gears it had been found to be less than 1 per cent.; this was true of all the types of teeth normally employed.

The time required to change-over from ahead to astern would vary somewhat with the type of valve control employed; the actual manipulation of the valves need not occupy more than a few seconds. According to measurements which had been made in various classes of vessel, reversal of the shaft from full speed could be made in from 25 to 30 seconds.

INSTITUTE NOTES.

CORRESPONDENCE.

Ignition Delay in Diesel Engines.

On page 185 of the October, 1937 Transactions a letter from Mr. A. F. Evans (Member) was published commenting on the article by Dr. Ing. W. Lindner entitled "Experiments on the Behaviour of Fuel in Diesel Engines" reproduced in the August, 1937 Transactions, from "The Motor Ship". The following reply to Mr. Evans' remarks has now been received from Dr. Lindner:—

"I fully acknowledge the often considerable influence which the course of the injection process exerts on the ignition delay and on the rise of the combustion pressure, and I can confirm that in the research laboratory of the M.A.N. we have carried out numerous investigations in this very direction. Some particulars regarding this point are already contained in the full reprint of my paper (Cf. "Brennstoff-und Wärmewirtschaft, Vol. 19, 1937, pp. 123-144).

I am in agreement with Mr. Evans that in regard to the determination of the ignition characteristics of a fuel, the aim should be to discover a method which is not affected by the characteristics of the engine (shape of the combustion space, injection function, etc.). At the same time, the results obtained must be applicable to the engine conditions, and it is to be regretted that in the chemico-technical testing procedure as known to-day, this is not the case to a sufficient extent. Nor am I fully satisfied with Mr. Evans' proposal to determine the ignition delay when injecting the fuel into an open flame in a state of extremely fine pulverisation, as the reaction processes which take place within the flame, e.g. owing to the appearance of activated molecules, may give rise to additional influences on the ignition delay as compared with injection into hot air of the same temperature. This is a factor which is not present in the compression-ignition which takes place in the engine.

Injection in a state of extremely fine pulverisation into a space of as high a temperature as that corresponding with the Bunsen flame results in ignition delays which are very small and can hardly be distinguished from each other, as above 700° C. (1,300° F.) the ignition delay-temperature curves of all known Diesel fuels approach a limiting value (Cf. F. A. Foord, Journ. Inst. Petr. Techn., 18, 1932, p. 545; and R. Muller, Kraftfahrtechn. Forschungshefte, 1936, No. 3, V.D.I.-Verlag, Berlin). In order to discover the factors which determine the behaviour with respect to ignition in the engine, it

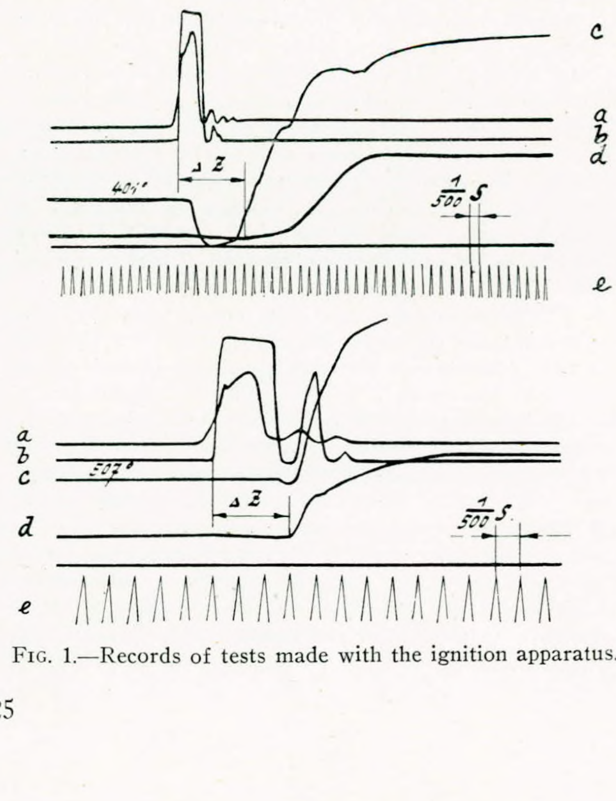


FIG. 1.—Records of tests made with the ignition apparatus.

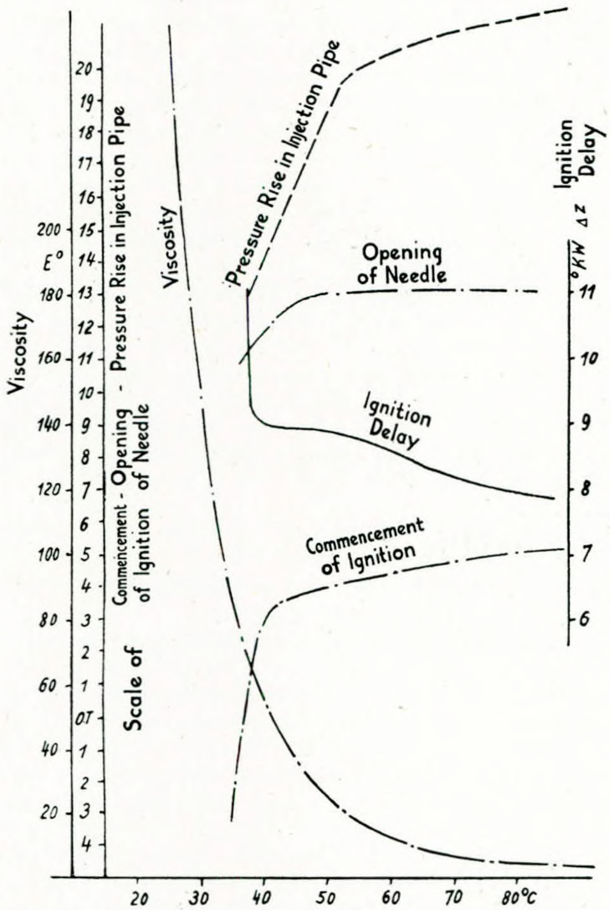


FIG. 2.—Influence of viscosity.

is therefore necessary to investigate the fuels at temperatures between about 400° C. and 600° C. (750° F. and 1,100° F.), i.e. the temperatures which can be realised in practice at the end of compression in the engine. Investigation in the engine therefore remains the best approach to the determination of the behaviour of fuels with respect to ignition. In order that reliable results may be obtained all measurements must be taken in identical working conditions, which also include the course of injection.

Here, as proposed by Mr. Evans, the recording of the impact pressure variations $\frac{1}{2}$ mv^2 in the free fuel jet furnishes a valuable means of observing the injection function. It is difficult, however, to obtain such records in the combustion space of the engine, and this method breaks down when the fuel jet is ignited. In our investigations we have therefore chosen the method of recording the stroke of the needle and the injection pressure for the purpose of supervising the injection process. Taken together these two records permit a satisfactory picture of the injection to be obtained, if the pressure is measured immediately in front of the injection valve. Fig. 1 shows the records obtained in two ignition tests on gas oil made with the apparatus described. The curves denote (a) the pressure in the injection pipe, (b) the stroke of the needle, (c) the temperature variation in the explosion vessel, which was measured in order to determine with accuracy the temperature at which ignition is initiated, (d) the pressure variation, and (e) the timing marks corresponding with intervals of ·002 sec. Apart from the investigation of the ignition delay of different fuels, the apparatus is, in particular, employed in the determination of the influences which the injection function, the design of the nozzle, the state of the air, and the shape of the vessel exert on the course of the ignition and combustion processes.

In order to illustrate the importance which attaches to a check on the ignition process, the results obtained in tests on a pre-combustion chamber engine are reproduced in Fig. 2. In all these tests, the working conditions in the engine were identical throughout, the viscosity of the fuel alone being varied by pre-heating it before it entered the fuel pump. In Fig. 2 the following characteristics are exhibited to a base of fuel temperature (viscosity): the commencement of the pressure rise in the injection pipe, the commencement of the stroke of the needle and of the ignition in degree of crank angle. The ignition delay results as the difference of the last two values. In the case of the injection pump used, which was fitted with a suction valve, increasing viscosity caused a retardation of the pressure rise in the injection line and at the same time, as the corresponding oscillograph records of Fig. 3

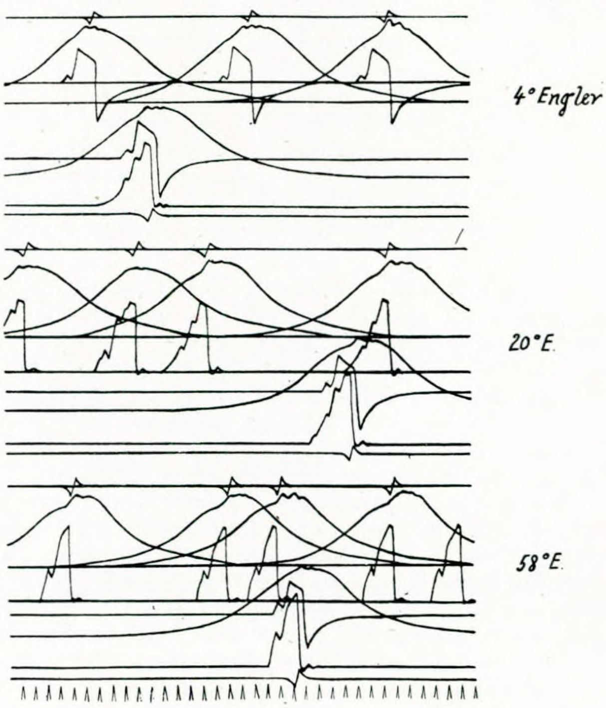


FIG. 3.—Records taken on a pre-combustion chamber engine for the investigation of the viscosity influence.

indicate, a steeper rise of the injection pressure, so that in the first instance, the commencement of the stroke of the needle remains unchanged at 13 degrees of crank angle before top dead centre. In spite of the steeper rise of the injection pressure the ignition delay here increases by nearly 1 degree of crank angle. The cause of this must be sought in the deterioration of the pulverisation of the fuel jet with increasing viscosity. The example given indicates that in this case the determination of the commencement of the stroke of the needle is in itself not sufficient to guarantee identity of conditions for the purpose of testing the fuels in the engine, so that the simultaneous recording of the variation of the injection pressure is required to obtain a clear idea of the inter-relation of the phenomena together with reliable results".

Mr. Evans has submitted the following comments on Dr. Lindner's reply :—

"I appreciate the scholarly and courteous reply of Dr. Lindner to the remarks that I made in connection with his paper which was entitled 'Ignition Delay in Diesel Engines'.

The objective of my experiments and of the remarks contained in my contribution to this discussion was to establish the fact that any combustible will ignite in zero time when exposed to a high temperature, providing it is fully pulverised, and conversely that no ordinary combustible will ignite with rapidity unless it is fully pulverised.

If we consider this in connection with an oil engine, while we must recognise that various practical considerations determine the maximum temperature of the combustion chamber at the moment of injection, there are no such limitations in connection with pulverisation.

Because of the fact that we are limited to about 600° C. for our combustion chamber we must take care that the first portion of the fuel charge to enter the combustion chamber does so in a state of fine pulverisation. If this is accomplished there will be ignition of this initial part of the charge within such a small period of time that detonation will not be recorded and the engine will be quiet. On the other hand, if the first portion of the fuel charge is in a coarse state there will be even greater delay than is the case at the high temperatures; therefore the comparatively low temperature of the combustion chamber makes it essential that fine pulverisation be provided.

I note that Dr. Lindner agrees that there is but a small variant in the ignition time at high temperature and full pulverisation; this I consider the main point of my contention, as it indicates that a search for a means of eliminating detonation by the selection of fuels is fundamentally wrong.

I regret that I cannot quite agree with Dr. Lindner in the analysis of the actual jet by inference. He suggests that the method I proposed of recording jet flow by $\frac{1}{2}$ mv^2 cannot be correlated to the combustion chamber, as the method breaks down when the jet is ignited. This arrangement records the actual flow of oil from a jet, if the recorder is removed and the jet offered to the combustion chamber; providing the pressure into which the jet flows is the same in each case we can assume an equal rate of discharge.

On the other hand the recording of fuel line pressure and the movement of the needle valve does not, in my opinion, represent the flow of fuel through the jet. The actual movement of the plunger will cause violent surges in the fuel line, of which we must have a full record before we can begin to plot the actual flow. It would certainly be interesting to obtain two records with a differential injector, one with the needle movement superimposed upon a $\frac{1}{2}$ mv^2 card and the other plotted from pressures and valve lift.

The last paragraph of Dr. Lindner's contribution is of exceptional interest. He shows that an increase of viscosity increases delay in ignition, which he explains is caused by deterioration of the pulverisation, which in turn causes retardation. While I am in entire agreement with this I must express some doubt as to the accuracy of the recording of flow by needle movement and pressure. The circumstances really call for a minute 'flow meter' that can be used with the engine".

ELECTION OF MEMBERS

List of those elected at Council Meeting held on Monday, 7th February, 1938.

Members.

Thomas Peter Bateman Archibald, 1, St. Nicholas Road, Wallasey, Ches.

Daniel Mackay Bain, 42, Haynes Road, Hornchurch, Essex.

Alexander Reid Bruce, Kamunting, Perak, F.M.S.

James Munro Burnside, 10, Lapstone Gardens, Kenton, Harrow.

James Douglas Cooper, Alma, Stanley, Perthshire.

Archibald Campbell Downie, 126, Novar Drive, Hyndland, Glasgow, W.2.

David McCracken Gillies, 71, Clanville Road, Roseville, Sydney.

Thomas Glyndon Jones, 85, Brooklands Crescent, Fulwood, Sheffield, 10.

William Alfred Langley, Cardiff House, Cleveland Terrace, Marton Road, Middlesbrough.

Sidney Arthur Lillywhite, 13, Park Mansions, Colehill Lane, Fulham, S.W.6.

Charles Stewart McCaskie, B.I.S.N.Co., Bombay.

Edward Upton Parry, 53, Granville Park, Lewisham, S.E.13.

William Roy, c/o Mrs. Kerr, 94, University Avenue, Belfast.

William Purdie Scott, 488, Finchley Road, N.W.11.

Norman Turnbull, Holt's Wharf, Pootung, Shanghai.

Herbert John Wood, 37, Ethelbert Gardens, Ilford, Essex.

Associates.

William Garland Long, Mundaring, Western Australia.

Douglas Arnold Hansard, Mayflower, Taplow, Bucks.

Edward Newton Hutchinson, 39, Percy Terrace, Sunderland.

Robert Woodward, c/o G.P.O., London.

Transfer from Associate to Member.

John William Lyons, 35, Kitchener Terrace, North Shields.

Re-instatement and Transfer from Student to Associate Member.

James Henry Rea, 283, Knowsley Road, Bootle, Liverpool.

Transfer from Student to Associate.

Robert McDonald, Glencoe, High Heworth Lane, Felling-on-Tyne, Co. Durham.

ADDITIONS TO THE LIBRARY.

Purchased.

Russian Technical Dictionary (Steam Boilers, Engines and Turbines). Constable & Co., Ltd., 24s. 6d. net.

Russian Technical Dictionary (Internal Combustion Engines). Constable & Co., Ltd., 10s. 6d. net.

The Official Year-Book of the Scientific and Learned Societies of Great Britain and Ireland. Charles Griffin & Co., Ltd., 172 pp., 8s. 6d. net.

Kempe's Engineer's Year-Book for 1938. Morgan Brothers (Publishers) Ltd., 31s. 6d. net.

Presented by the Publishers.

Aluminium Alloys. Extract from the book of Dr. Ing. Alfred von Zeerleder. Published for private circulation by Messrs. High Duty Alloys, Ltd.

Proceedings of the London Congress April 19th-24th, 1937, of the International Association for Testing Materials. Published by the International Association for Testing Materials, 28, Victoria Street, London, S.W.1.

Combustion-efficiencies of Gas and Oil Engines. By W. A. Tookey.

The First Diesel Engine in the World Developed during the Years 1893-1897 in the Shops of the Maschinenfabrik Augsburg. M.A.N., Augsburg.

Rudolf Diesel and Burmeister & Wain. Burmeister & Wain, Copenhagen.

G.E.C. Technical Descriptions, and G.E.C. Electrical Plant Installations. The General Electric Co., Ltd.

The following British Standard Specifications:—

No. 761, 1937. Cylindrical Vertical Multitubular Boilers.

No. 763, 1937. Sampling of Coal. (Report by E. S. Grumell, D.Sc. with special reference to the size-weight-ratio theory, and with notes on sampling and analysis for ash content by A. Crawford, M.Sc., Ph.D. and W. Reed.

No. 764, 1937. Automatic Change-over Switches and Contactors for Emergency Lighting Systems.

No. 765, 1938. Internal-Combustion Engines, Carburettor Type, excluding Aero Engines.

No. 768, 1938. Grub Screws.

No. 778, 1938. Steel Flanged Joints for Hydraulic Pipe Lines for Pressures up to 4,500lb. per sq. inch.

Transactions of the Junior Institution of Engineers, Vol. XLVII, 1936-37, containing the following papers:—

"Discovery and Evolution of Oxygen", by Lingwood.

"The Steel Contractor in Building Construction", by Newman.

"The Microscope in Engineering and Industry", by Bingham.

"The Mechanisation of Industrial Furnaces", by Matthews.

"Monolithic Furnace Construction", by Duguid.

"Metallurgical Developments and Engineering Progress", by Larke.

"Drainage of Birmingham", by Manzoni.

"Elements of Pumping Machinery", by Lymer.

"Geodetic Surveying for Map Production", by Lockett.

"Gas Engineering as Applied to Water Heating for Domestic Use", by Friedman.

"Deterioration in Paper Insulated Cables", by Kapp.

"The Modern Steam Car and Steam Wagon", by Lewis.

"Principles of the Modern Television Transmitter and Receiver", by Stockton.

"Development of Technical Apparatus", by Bancroft.

"The Standard Sunbury Cathode Ray Engine Indicator in Theory and Practice", by McGillewie and Withers.

"Utilization of Underground Water Resources", by Judd.

"Laying and Maintenance of Gas Mains in Congested Areas", by Rodgers and Gray.

"Automatic Fire Alarm Signalling", by Calvo.

"Elements of Efficient Power Transmission", by Baird.

"Modern Steam Locomotive", by Taylor.

"Notes on Dust Control", by Faris.

"Historical Review of Mine Lighting", by Garrett.

Proceedings of The Institution of Mechanical Engineers, Vol. 136, containing the following papers:—

"Journal and Thrust Bearings", by Swift.

"Engine Lubrication (Internal Combustion Engines)", by Ricardo.

"Engine Lubrication (Reciprocating Steam Engines)," by Stanier.

"Industrial Applications", by Auld and Evans.

"Properties and Testing", by Gough.

(The foregoing papers are reports for the various groups in the General Discussion on Lubrication and Lubricants).

"Shoe Machinery", by Cooper.

"Torpedo Boats", by Thornycroft.

"The Gas Engine and After", by Lanchester.

"Water Hammer in Pipes, including those supplied by Centrifugal Pumps; Graphical Treatment", by Angus.

"The Application of the Locomotive to Traffic Working", by Fairless.

"Modern Rolling Mill Design", by Poole.

The Journal of Commerce Annual Review. Charles Birchall & Sons, Ltd., 284 pp., illus., 2s. net.

As an informative and authoritative record of shipping, shipbuilding and marine engineering development the "Journal of Commerce Annual Review of Shipping, Shipbuilding, Marine Engineering and Allied Industries" has maintained a high standard for many years. This issue is no exception and can be confidently recommended to marine engineers as a publication of exceptional value at its modest price. Besides many of general interest, the review contains the following articles which will have special appeal to members of The Institute: "Ship Construction and Propulsion", by W. D. Kirkpatrick; "Welding as Applied to Shipbuilding"; "Hull Forms and Appendages", by W. A. Robb; "Marine Steam Engineering Progress", by J. Hamilton Gibson; "Electric Motors for Ship Propulsion"; "Motorship Tonnage Increasing", by A. C. Hardy; and "The Atlantic Ferry".

Heat and Thermodynamics. By M. W. Zemansky, Ph.D. McGraw-Hill Publishing Co., Ltd., 388 pp., illus., 24s. net.

This well-produced book is "designed as an intermediate textbook to supply the needs of students who are in preparation for a career in theoretical physics, theoretical chemistry, and engineering"; and the author strives (to quote again from the preface) "at a compromise between rigor and simplicity". The average engineering student in this country will certainly not doubt the rigor. Thermodynamics is a difficult subject and necessarily involves much use of mathematical methods, and the student will not expect to be excused a good knowledge of the differential and integral calculus. He may, however, be somewhat intimidated by the exactness of the terminology employed; as, for example, when he reads the suction at constant pressure in the Otto cycle described as "a quasistatic isobaric intake", and the drop of pressure at constant volume as "a quasistatic isovolumic drop of pressure".

The earlier chapters deal with the fundamentals of the subject—temperature and its measurement, heat, work, reversibility, the Carnot cycle, and entropy. The author then goes on to discuss the application of thermodynamic principles to physics, chemistry and engineering. A chapter of sixteen pages is devoted to the steam engine and refrigerator. A large number of problems for solution is provided, but the value of the book would be enhanced if answers to these were given.

While this volume will not be of great use to students preparing for examinations in heat engines similar in standard to that for a pass degree, to those who intend to specialize in the more theoretical aspects of the subject it will undoubtedly prove of particular value.

Dynamical Similarity in Fluid Flow. By A. C. Livingston, A.R.T.C. The Draughtsman Publishing Co., Ltd., 96, St. George's Square, London, S.W.1, 46 pp., 22 illus., 2s. net.

In all hydraulic laboratories scale model work is an outstanding feature and of paramount aid to industry. If full benefit is to be derived from these model experiments, then the fundamental principles connecting prototype and model must be thoroughly understood, otherwise imperfect results are obtained which may be the cause of complete failure of the finished product.

The questions at issue in practical problems are rarely so clear and well-defined as to suggest how the experiments should be conducted, or whether they will be successful. Generally each problem demands special experimental equipment and the least important factors of the problems may call for costly and intricate model work. Therefore, all questions which can be answered sufficiently well through mathematical analysis should be so eliminated from model experimental work, together with unimportant questions.

The author has made a contribution to the subject which should prove of value to those concerned with hydraulic work in particular, and which will not be without interest to those outside that field.

Protective Films on Metals. By Ernest S. Hedges, M.Sc., Ph.D., D.Sc., A.I.C. Chapman & Hall, Ltd., 2nd edn., 397 pp., 53 illus., 21st. net.

One of the most interesting chapters of modern science is that describing the work which has been done on the production and identity of surface films, and in this connection the name of Ulick Evans comes first to the mind. Dr. Hedges is also a well-known worker on these problems and is particularly fitted to deal with his subject. In the earlier part of his monograph he has given a detailed survey of the work of Evans and his collaborators and many other investigators of this and other countries and may fairly be said to have covered the whole field of existing knowledge.

In his preface the author claims to be writing for both student and practical man. The latter will be most interested in the second portion of the book where the various ways in which protective films can be established are described and the limitations on their performance outlined. Here will be found an account of the aluminium-brass condenser tubes developed by the British Non-Ferrous Metals Research Association and it will be understood why this brass withstands the high demands of modern marine engineering while the 70 : 30 brass which did excellent service for so many years fails rapidly.

Each new phase of engineering brings its own outstanding problem and the engineer who is concerned with Diesel motors finds the explanation of cylinder liner wear difficult. He will be interested in the sections dealing with the films formed on the "stainless" materials and by the various cementation processes and will hope that, from them and the newer theories of variation in liner wear, solid progress will eventuate.

The practical man no less than the scientific investigator will be intrigued by the account given of instances of the cyclic formation and dissipation of films described by Dr. Hedges, who has himself carried out a considerable amount of original work in this connection. The analogy between these changes and those operating in biological conditions is striking. The author deals with the more or less established methods by which it is claimed that films or coverings proof against corrosive attack can be produced and maintained and indicates their shortcomings.

The book contains a large number of references interpolated in the text. We note that Kenzel (in 1928) is given credit for the development of case-hardening by the use of nitrogen. The reviewer's own recollection is that Fry published a paper in February, 1926, in the Kruppsche Monatshafte on his work on this subject, and that he was largely responsible for the success of the process.

The book can be recommended to those who seek a better acquaintance with the present position of the maintenance engineer's chief concern, that is the struggle against the disruptive attack of air, water and other corrosive influences on metals in various states of stress.

JUNIOR SECTION.

Supercharging of Diesel Engines

An outstanding lecture was delivered under the above title by Dipl. Ing. Alfred Büchi (Member) at a joint meeting of the Junior Section and Students of the South East Essex Technical College, Dagenham, on Friday, January 7th, 1938. Mr. P. J. Haler, M.B.E., M.Sc. (Member and Principal of the College) occupied the Chair.

As might be expected from such a leading authority on supercharging as Mr. Büchi, the lecture, which was illustrated by lantern slides, was of an exceptionally valuable character, and it is hoped to publish a full report in a later issue of the Transactions.

On the proposal of Mr. Saunders of the College Engineering Society, a hearty vote of thanks was warmly accorded to Mr. Büchi.

ABSTRACTS OF THE TECHNICAL PRESS.

Ship Stresses in Rough Water in the Light of Investigations Made Upon the Motorship "San Francisco".

With the object of investigating stresses in ships in rough water a test cruise on the Hamburg-American motorship "San Francisco" was undertaken in the year 1934 under the direction of the author with the support of German societies. Some *results of these investigations have already been published by the author and his colleagues. The ship's normal service was from Hamburg to the Panama Canal and Vancouver and back. In this paper new investigations are published on the correlation of wave height and influence on the ship. For this purpose, the shape and dimensions of the waves were measured, the forces acting on the ship's bottom and the strains and deflections of the ship's girder. Finally, the accelerations by heaving and pitching angles were determined. The instruments used for measuring the waves, water pressure, strains and deflections, are described. Acceleration, heaving and pitching are found in good agreement with the measurement. Moments and deflections are in correlation with the forces acting on the bottom of the ship. Considering the maximum stresses of the ship, the measurements show that the stresses in sagging condition, as influenced by the sea-way, are greater than the stresses in hogging condition. The differences are caused by dynamical forces such as oscillations and impacts. The greatest stresses occur if the ship's length is nearly equal to the wave length; but there are also considerable strains if the wave length is greater than the ship's length. Stresses are also influenced to a certain degree by the ends of light superstructure and by end laps. The measured deflections correlate with the measured strains.—*Schnadel, Trans. of North-East Coast Institution of Engineers and Shipbuilders, January, 1938.*

Practical Experiment on the Greasing of Launching Ways.

The author states at the commencement of the article that from laboratory experiments it is possible to arrive at a decision as regards the comparison between the quality of the various kinds of greasing compositions, but on account of the smallness of the surface of the model, the non-uniformity of the grease, pressure, etc., it is not possible to make use of all the data supplied by the laboratory in the case of an actual launch. To obtain practical results, one of the shipbuilding yards in the U.S.S.R. carried out an experiment corresponding as nearly as possible to actual launching conditions. A sliding way six metres long by 0·8 metres by 0·5 metres, having a total working surface of 4·55 sq. metres, was placed on an experimental sliding way with an inclination of 0·056 metres. The standing way for a distance of 31 metres was greased with a composition determined by experiments carried out at the laboratory. The sliding way was then loaded with 38 cast iron blocks having a total weight of 88·5 tons, for the uniform distribution of the load. Ten steel girders were laid on top of the sliding way; the total weight of the blocks, girders, way and and fastenings was 91 tons (Fig. 1, not reproduced).

A wedge-shaped piece was cut out from the rear end of the sliding way, and a wedge 2·520 metres in length was fitted into the space; this wedge remained in position for a distance of 14 metres after the sliding way commenced to move. It was held in place at the thin end by a trigger which was released by a rope 13·7 metres in length, attached to the ground, another rope 14 metres in length being connected to the rear end of the wedge and also fastened to the ground; the trigger was therefore released 300mm. before this latter rope came into operation. The wedge remained stationary while the remaining part of the sliding way continued on its course.

At the time the sliding way commenced to move the specific pressure was :—

$$\frac{91}{4{\cdot}55}=20 \text{ tons per sq. metre.}$$

When the wedge-shaped portion of the sliding way was withdrawn, a redistribution of the reaction on the ways took place, with an increase of pressure at the edge "K" of the sliding way, now in contact with the standing way; this pressure might be designated the "forefoot" pressure. The curve of the pressures as a first approximation may be represented by a straight line, forming a triangle, the centre of gravity of which coincides with the centre of gravity of the load, and is therefore 0·48 metres from the edge "K" of the sliding way; the length of the base of the triangle "A" $=3\times0{\cdot}48=1{\cdot}44$ metres.

* Schnadel, Horn, Weinblum and Weiss. "Hochseemessfahrt": *Jahrbuch der Schiffbautechnischen Gessellschaft, 1936.* Lempf. Trans. N.E.C.I., 1937, p. D15.

The specific pressure P_0 may be determined from the equation :—

$$\frac{P_0 \times a}{2} \times b = P, \text{ where } b = 0{\cdot}8 \text{ (the breadth of the way)}$$

$$\therefore P_0 = \frac{2P}{a \times b} = \frac{2{\cdot}91}{1{\cdot}44 \times 0{\cdot}8} = 158 \text{ t/m}^2$$

In this manner we have the speed of the sliding way at normal pressure, and afterwards also the speed at the actual specific pressure at the "forefoot". It is interesting to note that the standing way was coated with grease down to the water's edge only; the way beyond this at the time of the experiment was floating on the surface. The floating way, 47 metres in length, was not greased or even cleaned from dirt. Wedges were placed on this part of the way to retard the movement of the sliding way. When the dog-shore was knocked out, the sliding way did not move, but with a slight thrust from the hydraulic ram it commenced to move evenly down the standing way. At a distance of 13·7 metres from the starting point the trigger was released, and at 14 metres the wedge-shaped portion of the sliding way remained stationary, and the sliding way with increased "forefoot" pressure moved for a distance of 17 metres down the greased portion of the way; it knocked away the wedges, ran the full length of the floating way (sinking this) then dropped over the edge and sank. The speed of the sliding way was registered by a special instrument.

A diagram plotted from data obtained from the instrument is given in the article, and shows that the speed of the way at the moment of the increased pressure at the "forefoot" was 2·55 m/sec; at the end of the greased part of the way the speed was 3·8 m/sec., and at the moment it dropped off the end of the way, the speed was reduced to 2·0 m/sec. The coefficient of friction at the commencement of the moving of the sliding way was 0·043, afterwards decreasing to 0·030; from the commencement of the increased "forefoot" pressure it began to rise; at the end of the greased portion of the standing way the coefficient was 0·036, and for a distance of 46 metres the coefficient equalled 0·060. At the moment the "forefoot" pressure commenced, the tallow was forced out; at the same time no heating of the ways was observed. The results of the experiment carried out under launching conditions were considered satisfactory.—*"Soudostroienie", No. 9, 1937.*

Launching Particulars of the Icebreaker "Lazar Kaganovitch".

This icebreaker was launched in April, 1937, the launching particulars being as follows :—

Weight of hull and outfit at time of launch	3,377 tons.
Weight of launching arrangement ...	185 tons.
Inclination of launching ways ...	0·056 metres.
Maximum camber of ways	0·200 metres.
Number of ways	2
Breadth of the two sliding ways ...	2·26 metres.
Total working area of sliding ways	160 sq. metres.
Specific pressure on ways	23·3t/m².
Calculated coefficient of friction at the commencement of moving and	0·04.
when moving	0·03.

The vessel was launched in the usual manner, stern first; the keel was 1,130mm. above the launching ways and parallel to same.

The greasing of the ways was as follows: The ways were at first coated with a layer of mutton tallow 4mm. thick, then a layer of beef tallow with a mixture of 7 per cent. paraffin 9mm. thick at after end and 6mm. at fore end, then a layer of beef tallow with 5 per cent. paraffin 3mm. thick at after end and 4mm. at the fore end, then a layer of beef tallow 2mm. over the full length, followed by a coating of beef tallow with 50 per cent. linseed oil, and at the finish the whole of the ways was covered with soft soap.

The under-water portion of the standing ways, after being dried, was coated with a layer of pure mutton tallow 2mm. thick, then a layer of mutton tallow with 4 per cent. linseed oil 3mm. thick, then a 6 to 7mm. layer of 50 per cent. soft mutton tallow and 50 per cent. hard mutton tallow plus 7 per cent. linseed oil; this was followed by a coating of beef tallow 2mm. in thickness, and finally a layer consisting of 50 per cent. beef tallow and 50 per cent. linseed oil. The total weight of tallow was 5½ tons.

Previous to deciding on the composition of the grease, preliminary experiments were carried out at the laboratory with various mixtures, and afterwards a practical experiment was made with one section of a sliding way, loaded to correspond to the calculated pressure anticipated at the actual launch (see previous abstract "Practical Experiment on the Greasing of Launching Ways", for a description of this experiment).

The length of the sliding ways was 79·1 metres, and the width of each way 1,130mm.; each side consisted of 7 sections.

The holding arrangements were as follows :— 4 dog-shores 300 × 300mm. and 2 triggers. The calculated pressure on the dog-shores was about 40 tons. Each of the holding devices was capable of resisting this pressure. As the triggers were the last to be released, they were calculated to resist the moving force, plus an 80 tons pressure from the hydraulic rams. The maximum pressure which could be exerted by 4 hydraulic rams was 200 tons. The fore poppets were constructed of 16 vertical pine beams with a length of about 5 metres. The pressure on the fore poppets when the stern began to lift was about 26kg/cm². The poppets were securely connected together by 1in. bolts; six poppets were fitted at the fore part on each side and lashed together by wooden beams (Fig. 1, not reproduced), each pair being bound together with four turns of steel wire rope 63mm. circumference.

The lower end of each poppet rested direct on the sliding ways; their upper ends rested on pine packing, fitted under the apron knee plates. The aprons of the extreme forward poppets consisted of plates 22mm. thick, and 1,600mm. wide; the maximum fore poppet pressure which came on them, was about 1,000 kg/cm^2. Knee plates 16mm. thick were riveted to the aprons to take the thrust of the fore poppets.

The launching of this icebreaker was somewhat different from the launching of an ordinary merchant vessel owing to the short length, special form, the stern being well cut away, and in proportion a large launching weight. Further the keel was laid 300mm. higher than at first intended, i.e. 1,130mm. above the standing ways.

All these conditions required very careful consideration at the launching of the vessel. At the time of the year the vessel was launched, the depth of water over the way ends varies from 2·1 to 2·4 metres. According to careful calculations the vessel could be safely launched with a depth of water over the way ends of not less than 2·3 metres. The actual depth of water at the time of launching was 2·32 metres. To guard against the vessel tipping over the way ends, two steel caissons with a buoyancy of 180 cubic metres were fitted at the stern, and securely fastened to the hull; to assist the flow of water their forms were somewhat fined away. Owing to the presence of these caissons the resultant forces of the weight of vessel, and the upward pressure, occurred at a point 5·7 metres above the way ends; in this manner the tipping of the vessel was prevented. The buoyancy of the caissons being at the stern gave a difference of trim of about 1 metre, and a considerable drop was anticipated when the fore edge of the sliding ways passed over the end of the standing ways.

A dynamic coefficient of two was assumed. With this coefficient it was expected that if the launching speed was slow when the vessel dropped off the way ends, the keel would strike the bottom. By rough calculations taking into account the resistance of the water, the movement of the vessel, and sliding ways, also the increased coefficient of friction, the speed of the vessel when at the way ends would be 1 to 1·5 m/sec. By this speed the path of the fore end of the sliding ways would be trajectory, which would practically prevent the danger of a severe blow on the bottom. The speed calculated with a coefficient of resistance of the water, and friction of the sliding ways on the standing ways, was equal to 3·5 to 4 m/sec. The coefficient of friction assumed at the moment of moving was 0·04, when moving 0·03, increasing to 0·1 at the maximum fore poppet pressure. The coefficient of resistance of the water on the moving hull and caissons was taken as equal to :—

$\frac{24 \text{ kg/sec.}^2}{\text{M}^4}$ and that of the ways $\frac{55 \text{ kg/sec.}^2}{\text{M}^4}$

The specific pressure at the way ends when the vessel commenced to rise did not exceed 52 t/m^2; the distribution of the pressure extended over the length of the sliding ways in contact with the standing ways. At the commencement of the fore poppet pressure, the pressure on the ways increased to 118 t/m^2; by calculation the length of the sliding ways transmitting this pressure to the standing ways was 3 metres. The calculated pressure on the fore poppet was 745 tons. This pressure commenced to operate at a distance of 15·7 metres forward of the way ends.

Results Obtained from Observations at the Launch.

The following investigations were carried out at the launch :—

(1) Measurement of the speed by means of special devices placed on board the vessel and on the ways.
(2) Measurement of the speed at observation posts.
(3) Noting the trim at the moment the vessel dropped off the way ends by means of a special device.
(4) Measurement of the compression on the fore poppet.

The observed speed of the vessel at the way ends was 4 to 5 m/sec. From the time the vessel began to move to when she dropped off the way ends, the time taken was 39 seconds, with a mean speed of 3·1 m/sec. By measuring the dip when the vessel dropped off the way ends, the dynamic coefficient of 1·54 was determined, which corresponds to a relative coefficient of resistance U=0·2. On account of the great speed of the vessel at the way ends, no striking of the fore part on the bottom was noticed. The measurement of compression on the fore poppet was effected by means of a compression device. After the launch the compression was measured, and found to be 45mm. By means of this compression it was calculated that the fore poppet pressure was transmitted to the standing ways over a length of 4 metres, against 3 metres as determined by the preliminary calculation.—*"Soudostroienie", No. 9, 1937.*

The Standardization of Parts for Marine Engines.

The system of standardizing the various component parts of a marine engine when accurately carried out and in operation for the whole of the industry possesses a great number of advantages. For example the different parts of one engine may be transferred to another of the same type, without extra work of fitting in place and without detriment to the working efficiency of the engine. The erection of engines constructed of such parts is simplified, spare parts for repairs may be centralized and decreased in number, and less time is required for erection work and effecting repairs. The necessary and most appropriate clearance between parts, where such is essential, could be determined by experiments and analysis; accuracy and quality of the working surfaces could be investigated. Thus, by the use of a system of standardization the efficiency of the engine would be increased, and the construction and repairs greatly facilitated. During the second five-year plan in the U.S.S.R. the work of standardization was commenced at nearly all the works and designing organizations; at the works special departments were formed and equipped with suitable measuring instruments. A special central state organization is now proposed to be instituted for dealing with all standardization work. This department will deal with all work and questions connected with the standardization of engine parts, also with the various problems relating thereto, such as the most efficient clearance and the fitting of the various parts, especially for the present type of quick-running engines; it is also proposed to form an institute for the instruction of special technicians for standardization work. The experiments and research work may be undertaken by the Scientific Standardization Organization with the assistance of those works which possess a sufficiently well-equipped laboratory.

In conclusion certain recommendations are made as regards the work to be undertaken by the above organization, and it is also suggested that all engineering works should be joined up with the standardization department, the works receiving from it all standard measurements for templates, clearances, tolerances, etc. Each works would possess a measuring department equipped with all necessary instruments for checking the work, tools, etc. It would also be necessary that the technology at all the engineering works be thoroughly investigated, and all the latest improvements and designs be introduced so as to bring them up to the level of the best technique existing in engineering. A complete inspection should also be made of all machine tools in the various shops, for the same purpose.—*"Soudostroienie", No. 9, 1937.*

Hydraulic Starter for Diesel Engines.

It appears the present tendency in Diesel engine construction is definitely towards the two-stroke cycle non-compressor type, and it is assumed that four-stroke cycle engines will continue to decrease further in the future; accordingly the two-stroke cycle non-compressor engine only is here considered. Diesel engines are now started by manual, electric, compressed air, or steam power, according to power and revolutions, the main engines being usually started by compressed air. In addition to an air starter, a steam starter is installed in the "Still" steam-Diesel engine. Starting by means of compressed air is usually reliable and without danger. At the same time this method has a number of disadvantages, as follows:—

(1) The high cost of the energy of compressed air intended for starting.
(2) The complicated arrangement of the reversing and starting gears.
(3) Unavoidable leakage of air from the many branches of the pipe line and valves.
(4) A large non-productive loss of starting air in the working cylinder, on account of the late firing and insufficient expansion.
(5) The injurious effect of the coldness of the air (the temperature being sometimes below 0° C.) on the heated surface of the cylinder liner and bottom of piston, by consecutive startings following quickly one after the other, which may be the cause of cracks.
(6) Slowness of starting due to the air and oil not being admitted into the cylinder at the same time, thus increasing the consumption of air.
(7) The retarding effect of the starting air by delaying the warming up of the cylinder.

The foregoing considerations point out that other systems of starting may give quickness and reliability equal to that obtained by compressed air, and a number of its disadvantages may be eliminated. The most rational system would appear to be a rotary starter in preference to one applied to the piston; a very suitable type of starter for this purpose is the Pelton wheel which up to the present has rarely been utilized for this purpose. The advantages of using the Pelton wheel for the purpose of a starter for main Diesel engines are based on its following properties:—

(1) Simple construction bordering on the primitive.
(2) Reliability of its action.
(3) Low initial cost.
(4) Exceptional fitness for working on a small consumption of fluid with a high pressure.
(5) A large turning moment on starting.
(6) The possibility of a considerable increase of power, developed in a short period by increasing the amount of the fluid.

It must be noted, however, that the working parts of the Pelton wheel require some care in preparation, and in particular the nozzle with needle and the inner surface of the buckets must be carefully polished; the remaining parts may be of ordinary material and normal workmanship. The exceedingly simple construction of the apparatus guarantees its reliability and low initial cost. The Pelton wheel fitted at stationary installations has shown the very high degree of efficiency of 85 per cent. working at a pressure of 160 atmospheres, and in some special cases up to 250 atmospheres. The starting torque is nearly twice the running torque, and hence the Pelton wheel compares very favourably with other types of rotary equipment. These properties of the Pelton wheel prove it to be specially adapted for a starting apparatus for main Diesel engines, as at initial starting it is necessary to overcome the friction of the moving parts and the inertia of the dead load. The kind of fluid to be used for a starter in a marine installation is of great importance; it should be of such nature as to be able to replenish any leakage if such should occur, and should not require a large feeding system. Water has shown itself to be the most suitable of three fluids, i.e. water, mineral oil and mercury; it is less costly and may be replenished from the sea, and to prevent corrosion the surfaces of the installation may be galvanized (the specific weights and various particulars of these three fluids where applied to Pelton wheels are tabulated in the article).

The pressure of the jet on the buckets at the moment of motion:—

$$P=2\frac{\gamma}{g}Qw_1$$

Where γ=specific weight of the fluid (Kg/M^3).
g=acceleration due to gravity=9·81 M/sec^2.
Where Q=consumption of the fluid M^3/sec.
w_1=velocity of the falling fluid on the buckets.
$=0{\cdot}97\ \sqrt{2gH}$. M/sec.
Where H=the head of the fluid.

The weight of the supply system, which is a consideration in a vessel, consists of the weight of the reservoir, pipe line and the fluid. It is possible to infer from the foregoing formula, that to reduce the weight of the feed system it is more advantageous to increase the volume of the liquid, i.e. the reserve supply, than to increase the pressure in the reservoir, as by starting with pressure the weight of the reservoir commences to exceed the weight of its liquid contents, and the cost of the reservoir is decidedly higher; it also increases the cost of the air and water pumps. Increasing the pressure has little effect on the increase of pressure of the jet; on this account a pressure of 30 atmospheres is established in practice for a Diesel engine. In cases where the machinery space is restricted, a higher pressure may be adopted. For the purpose of calculating the particulars of a hydraulic starter, it is necessary to obtain the value of the turning moment at starting; this may be found either by calculation or practical experiment. In practice, for calculating the starting apparatus the starting moment M is usually assumed to be 50 per cent. of the normal turning moment M_0, i.e. $M=0{\cdot}5\ M_0$; this value includes the necessary reserve. For reversing at full speed of the vessel, it is necessary to increase the starting moment to overcome the reverse moment transmitted to the propeller by the speed of the vessel. In practice it has been determined that this increase in moment is about 0·35 M_0. The Pelton wheel is quite capable of developing this additional starting moment up to the full capacity of the feed system. It is therefore necessary when calculating the particulars of a hydraulic starter to take the value of the initial turning moment as $M=0{\cdot}5\ M_0$, with a continuous flow of water on the buckets for a mean period of three seconds. This is slightly less than a starter requiring the admission of compressed air in the working cylinder, which has a negative effect by using cold air during the process of starting; this is a consideration when the manœuvring is prolonged. Where a hydraulic starter is installed the following equipment is necessary:—

(1) Two Pelton wheels fitted on a common shaft, one for going ahead and the other astern.
(2) Two jets, one for each wheel, connected to a common casing.
(3) Water reservoirs not less than two in number, of a capacity sufficient for 20 consecutive startings, and able to withstand a working pressure of not less than 30 atmospheres.
(4) Not less than two air receivers calculated for the same pressure, their capacity being two to three times greater than that of the water reservoir, for use in the event of a break-down of the water pump.
(5) A small air compressor for the purpose of the initial filling of the air receivers in the course of 1½ to 2 hours.
(6) A water pump of the triple-plunger type for delivering water to the water reservoir from the reserve tank. The capacity of the pump should be sufficient to pump back the water consumed by one start in the interval between two consecutive starts; the time taken for this in practice is found to be about 20 seconds.
(7) A pipe line connecting the air receiver with the water system, and the water with the nozzles.
(8) Gear to the nozzles and oil fuel pump.
(9) Reserve water tank in the double bottom.
(10) Transmission gearing between the shaft of the Pelton wheel and crankshaft of the main engine; this transmission consists of toothed gearing with a ratio of 1 : 6, and a clutch of the Zinzienatti-Bickford type. The pinion is fixed on the shaft of the Pelton wheel, and the toothed rim fitted loose on the flywheel, and may be connected to it by the clutch.

The starting of the main engine takes place by connecting the fuel pump, which should considerably hasten the starting as compared with that of starting by compressed air. In conclusion the advantages and disadvantages of the hydraulic starter if compared with one of compressed air, are considered to be as follows:—

Advantages.

(1) Abolishing the air valves in the cylinder cover, the various branches of the air pipe line on the engine, and the intricate distributing arrangement for the starting air.
(2) Doing away with two compressors each producing about 100 M^3 of compressed air per hour, and replacing by one of only 5 M^3 per hour and a compressor operated by hand.

(3) The power of each Diesel dynamo may be reduced from 160 h.p. to 100 h.p.
(4) A considerable simplification of the manœuvring arrangement.

Disadvantages.

(1) An additional piston water pump with an electric motor.
(2) An additional toothed transmission gearing with disconnecting mechanism.
(3) Two Pelton wheels with directing devices.
(4) Two additional water reservoirs of about 5·5 M^3 capacity, and the requisite pipe line.

The cost of the items which are not required is slightly higher than the cost of the additional items. The weight of the extra installation is approximately 7 to 8 tons more than the items dispensed with, which is a small matter in a vessel of about 3,400 tons. These considerations permit of the following conclusions: The installing of hydraulic starters in commercial vessels similar to the one described considerably simplifies the construction of the main engine, and at a slightly reduced cost of the installation; at the same time it greatly increases the reliability of the working of the main engine, quickens the starting, and simplifies the manœuvring when reversing. Its application is very advantageous for a two-stroke cycle Diesel engine fitted with an automatic fuel pump with gas injection. For fast-running engines, say up to 1,000 h.p. with revolutions not less than 300 p.m., the hydraulic starter may be fitted without transmission gearing. To illustrate the possibility of a practical application being made of the hydraulic starter as described in the foregoing, a sketch of an installation as actually fitted in a vessel and some of the particulars thereof are included in the article.—*"Soudostroienie", No. 9, 1937.*

The Labyrinth Packing.

The writer observes that of the numerous problems that Parsons had to solve in the development of the steam turbine not the least was that of preventing any appreciable leakage of steam at the glands and over the dummy pistons which he effected by means of labyrinth packings. Here no form of packing which involved rubbing contact was admissible on account of the high peripheral speeds, nor did current practice with steam machinery suggest any alternative. Exactly how he arrived at his idea is not known, but it is reasonable to suppose the phenomena of flow in the bladed portion of the turbine supplied an analogy on which he may have worked. Thus by arranging along the dummy piston a series of pressure drops similar to those which occur in the steam as it passes through the turbine blading, alternatively with means for the destroying of the velocities generated in the process, a limit to the speed of flow could be obtained. It followed that if the speed was so limited, and if the density of the steam was also sufficiently reduced by the successive partial expansions, the quantity leaking past the dummy piston could be kept down to any figure desired. A reduction of the clearances at the points of constriction will of course effect a proportional reduction in the leakage steam, and the packing lends itself readily to such an adjustment if the clearances are arranged in axial direction by altering the longitudinal position of the rotor with respect to the casing. By arranging for the clearances to be radial instead of axial, this facility of adjustment is of course lost, but on the other hand the rotor is free to expand longitudinally without impairing the steam tightness of the labyrinth. Both forms of labyrinth were employed, either separately or in combination, by Parsons, who, the writer considers, left the packing in a form which seems hardly capable of any material improvement.—*Editorial, "The Engineer", 21st January, 1938, p. 83.*

Diesel Varia.

The author briefly reviews the factors which determine the stroke, bore, mean effective pressure, and rate of revolution of oil engines and presents a more detailed discussion of the limits set by considerations of manœuvring and balancing in the choice of the cylinder number. He makes special reference to the balancing of five and seven cylinder engines, indicating the best crank and firing sequences and gives particulars of investigations which proved the existence of bending vibration or "whirl" of the shafting giving rise to bearing trouble in multi-crank engines of comparatively high power running between 200 and 300 r.p.m. In determining the best arrangement of the cranks and the best firing sequence it is therefore essential to secure a certain "distance" between the service speed and the whirling speed of the shaft. In supercharged four-stroke and in two-stroke engines it is further necessary to arrange the firing sequence in such a manner that the top of the pressure wave set up in the exhaust manifold does not pass the outlet of a cylinder which is having its cleaning or scavenging period. The author further elucidates the difficulties which arise in the lubrication of crankshaft bearings owing to the fact that the place of maximum pressure on the bearing changes continually while it remains the same on the crankshaft, so that this particular region of the shaft runs over various parts of the bearing—oil grooves, joints of brasses, etc.—where the oil pressure cannot be maintained, so that the oil film is broken and wear of both shaft and bearing results. Reviewing the three main methods of crank case construction, viz. cast-iron framing with and without tie bolts and welded framing, the author considers the elastic deformation under the alternating forces of combustion and their relative weight when the same top motion is allowed. He discusses sea-water, fresh-water and oil cooling with special reference to piston cooling and reproduces conclusions drawn from systematic tests on the influence of Brinell hardness and composition of material on the wear of cast-iron rings and liners.—*G. J. Lugt, Trans. of North East Coast Institution of Engineers and Shipbuilders, January, 1938.*

Recent Developments in Ship Propelling Machinery.

The article is based on the Thomas Lowe Gray Lecture, delivered before the Institution of Mechanical Engineers. The author reviews progress of various types of engines. As regards reciprocating steam engines, the apparent tendency towards retrogression in the change from triple to compound is due in reality to the use of superheated steam, which has rendered the compound engine as efficient as the triple-expansion. The uniflow principle, cam-operated valves and poppet valves are to be found in modern engines; cam operation makes for mechanical efficiency, but there is doubt as to the advantage of poppet valves over those of the piston type. Reheating by means of high-pressure steam appears to have possibilities, though it has not yet been widely adopted. Temperatures and pressures, particularly the former, have risen, but the main increase in thermal efficiency has been from the addition of exhaust turbines, geared to the main engine. Turbine temperatures and pressures have risen to a greater extent than those for reciprocating engines, and the practical elimination of condenser troubles has encouraged the use of water-tube boilers. 800° F. and 450-500lb./in.^{2} are suggested as limiting figures for coal-burning installations, but oil-burning may lead to higher pressures in the attempt to economize fuel. Auxiliaries are usually electrically driven. Condenser design has been improved, and reduction gearing of simple design is found satisfactory, provided the teeth are accurately cut. Combating of blade corrosion may bring back the impulse type of turbine. The author sees no future for turbo-electric machinery, though many successful installations are in service. The general tendency in Diesel engine design has been towards the elimination of those of the many early types which have proved less successful. Increase in maximum pressure and the use of solid injection have reduced fuel oil consumption by some 20 per cent., to 0·35lb./b.h.p./hr. The reduction in consumption of auxiliaries has been even greater, due to main-engine drive and electric drive having been adopted, and to the utilization of exhaust heat to provide steam. Four-stroke double-acting engines have been superseded by single-acting supercharged engines, but the maximum powers per cylinder are attained in two-stroke engines either double-acting or having opposed pistons. In the case of the latter, 1,600 h.p. is reached. There have been many geared Diesel installations, usually with special couplings, of hydraulic or other type, between the engines and the pinions. The high-speed engines thus made practicable are usually of the trunk-piston design. Electro-magnetic couplings have also been introduced in connection with geared Diesels.—*P. L. Jones, "The Engineer", 14th January, 1938.*

Turbo-Compressors for High Pressures.

The author discusses fundamental design considerations and methods of construction of multi-stage compressors for high overall compression ratios. The compression ratio per stage is limited by the allowable peripheral speed, and whilst the latter can to some extent be increased by the adoption of special types of impellers, such as those having integral blades, the achievement of high outlet pressures involves multiplication of the number of stages. The maximum output of a compressor is governed by the permissible velocity at entry; a high entry velocity increases the energy losses, and decreases the efficiency. Reduced speed of rotation, for a given peripheral speed, increases output. A low output, on the other hand, requires high rotational speeds, and small diameters. It may happen that the upper and lower output limits apply to different stages of the same compressor, necessitating the adoption of different rotational speeds for the low and high pressure stages. Cooling does not present any special difficulties when the pressure is increased, and loss of energy through disc friction of the rotors tends to be reduced. The author analyses the factors influencing efficiency, and estimates the overall isothermal efficiency of a 12-stage compressor designed for an output pressure of 145lb./in.^{2} as 62 per cent. Compressors have been designed for pressures as high as 400lb./in.^{2}, with 26 stages. The constructional features of compressors for high pressures are similar to those of machines for lower pressure but small output. One difficulty is that of maintaining a margin between the speed of rotation and the rotor critical speed; light alloys have been used with this in view.—*W. J. Kearton, Transactions of the Institution of Engineers and Shipbuilders in Scotland, February, 1938.*

Marine Boilers in 1937.

There has been a tendency towards the adoption of boilers giving increased output together with a reduction in size. An interesting replacement was that of the original cylindrical boilers, ten in number, in the "Arundel Castle" and "Windsor Castle", by four water-tube boilers, with an increase in s.h.p. from 14,500 to 26,000. These are of the Babcock-Johnson type, designed for 425lb./in.^{2} and 750° F. Air heaters are fitted, and a combination of forced and induced draught employed. A considerable number of cylindrical boilers of the Howden-Johnson type have been constructed and installed. The first British-built La Mont forced circulation boiler has been installed in a destroyer. It was designed for 300lb./in.^{2} and 688° F., and in shop trials gave an evaporation of 21·1lb. per sq. ft. of boiler surface, per hour, with an efficiency of 73 per cent. There have been several installations of boilers with mechanical stokers, both cylindrical and water-tube. A particular example, that of the "Manchester City", has three single-ended Scotch boilers with Bennis stokers and air preheaters, supplying steam at 225lb./in.^{2} and 630°., the total s.h.p. being 4,000. —*"The Engineer", 14th January, 1938.*

The World's Water Power.

The author surveys briefly the development and available resources of water power. The amount developed is about 60,000,000 h.p., shared mainly by Europe and North America, but the total available is nearly ten times as great, with Africa leading and Asia second, and there are also fairly large undeveloped resources in South America. As regards countries, Italy and France, followed by Norway, Switzerland, Germany and Sweden, are the leading European countries so far as development is concerned, but the largest resources are in Norway and Russia. Canada has more power developed than any European country, and her undeveloped resources are slightly larger than those of the United States. India has developed only a very small fraction of her 39,000,000 h.p., and Brazil has an almost equal amount untapped. About two-thirds of the total of 270,000,000 h.p. available in Africa is in the Belgian and French Congo, and the power developed is negligible.—*"Engineering", 7th January, 1938.*

Engineers and the Public.

The writer discusses the formation by the Institution of Civil Engineers, in conjunction with other bodies, of an Engineering Public Relations Committee. The proportion of the population earning its living through engineering, and the increasing part played by the various branches in the life of the country generally, are contrasted with the slight recognition accorded to engineers by the public, and the lack of information on engineering matters which the latter have available. The work of the Committee will comprise the organisation of lectures and exhibits, and the dissemination of information through the newspapers, broadcasting and the films. It is suggested that the scope might be extended to make the Committee a body capable of speaking for and representing engineers in general, apart from the special question of Public Relations.—*"The Engineer", 14th January, 1938.*

EXTRACTS.

The Council are indebted to the respective Journals for permission to reprint the following extracts and for the loan of the various blocks.

Velox Boiler Installation in "Athos II".

Machinery Alterations Carried out at La Ciotat Resulting in a 60 per cent. Increase in Power.

"Shipbuilding and Shipping Record", 20th January, 1938.

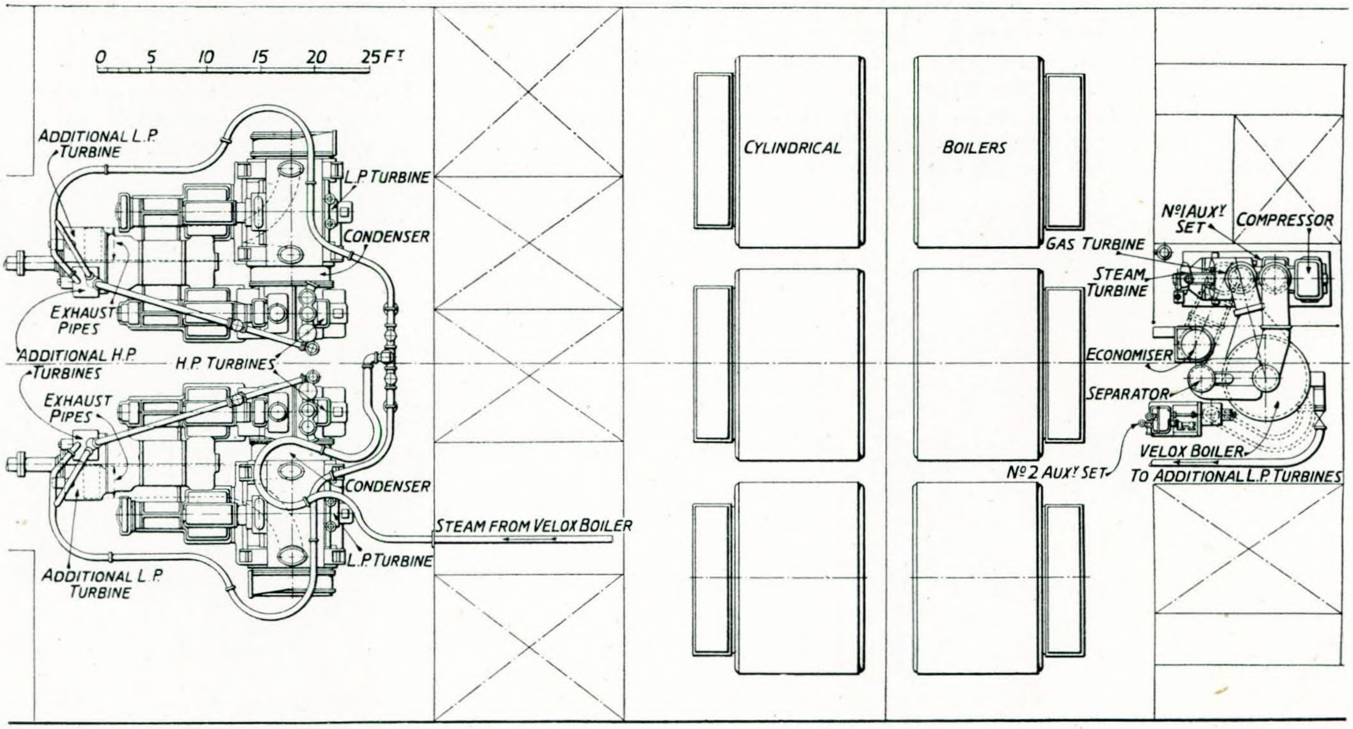
The first installation of a high-pressure Velox boiler in a French ship was recently completed by the Société Provençale de Constructions Navales, La Ciotat, when the passenger steamship "Athos II" had a boiler of this type installed. The "Athos II", which has a gross tonnage of 15,000 tons, was built in 1926 at the Weser shipyard, Bremen, and is operated by the Messageries Maritimes on their Marseilles-China service.

The original propelling machinery of the ship consisted of two sets of turbines comprising an h.p. and l.p. unit, supplied with steam from seven oil-fired Scotch boilers working at a pressure of 14kg. per sq. cm. (199lb. per sq. in.). Each set was arranged for driving its propeller shaft through double-reduction gearing, and the power developed was about 10,000 s.h.p. at 96 r.p.m., which gave the vessel a speed of about 14 knots in service.

In order to increase the speed of the vessel, a new h.p. and l.p. turbine have been added to each set, the steam being supplied by a Velox h.p. boiler working at a pressure of 55 kg. per sq. cm. (782lb. per sq. in.) at a temperature of 450° C. (842° F.). The new boiler has taken the place of one of the old boilers, and supplies steam to the new turbines, while the six remaining Scotch boilers supply steam to the original turbines, and also to the auxiliaries, including those of the Velox boiler.

On leaving the Velox boiler, h.p. steam enters the new h.p. turbines, leaving at a pressure of 14kg. per sq. cm. (199lb. per sq. in.). From these turbines, about 40 per cent. of the exhaust steam enters the new l.p. turbines, where it expands down to the pressure existing in the condenser, while the remainder of the steam mixes with the steam supplied from the Scotch boilers, and is passed through the original turbines.

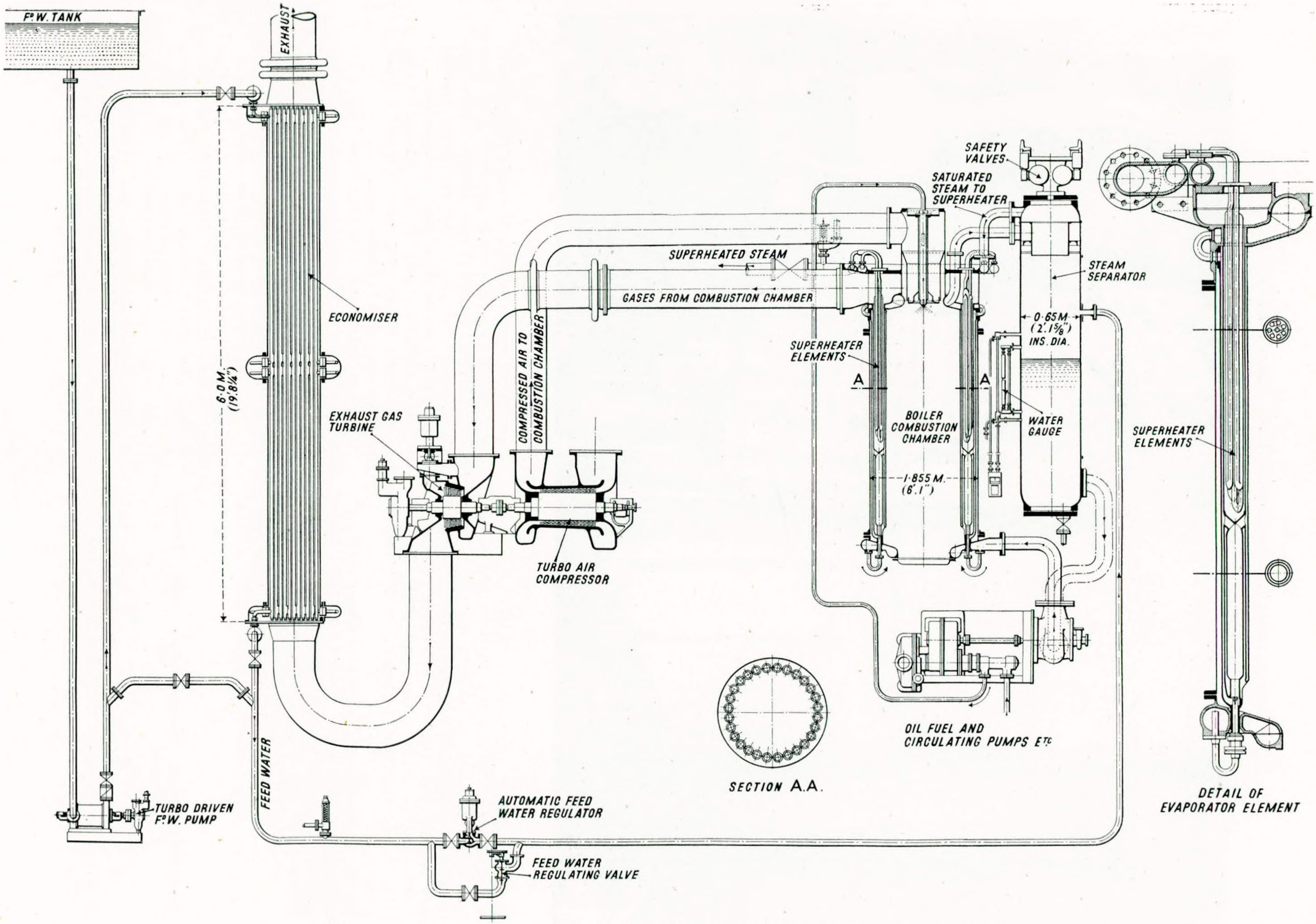
Both l.p. turbines (old and new) exhaust into a common condenser which has an area of 7,426 sq. ft., the cooling area having been increased during the alterations.



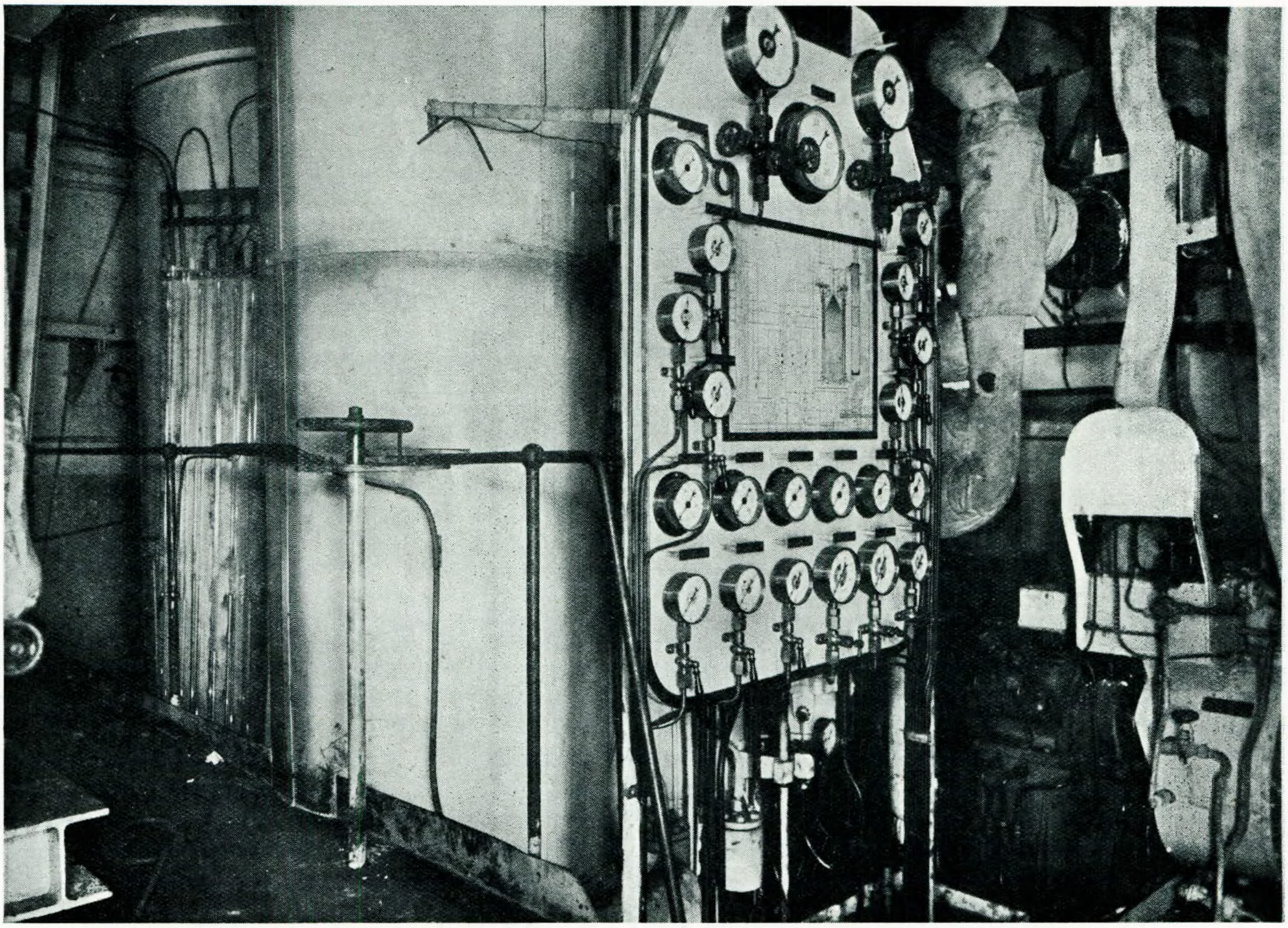
Machinery arrangement of the "Athos II".

When the vessel is running astern, the new h.p. turbines are disengaged from the unit, as they are not arranged with astern blading. Two sets of reduction gearing are provided, the first of which reduces the speed of the new h.p. turbines from 5,500 to 880 r.p.m.; power is then transmitted through a hydraulic coupling to a second set of gearing which reduces the revolutions to the propeller speed of 137 r.p.m.

The machinery alterations have resulted in an increase of 6,000 s.h.p. (about 60 per cent.) in the



Diagrammatic arrangement of Velox boiler and auxiliaries as installed in the "Athos II".



The Velox high pressure boiler installation.

power developed, the total of 16,000 s.h.p. being made up of the two new h.p. turbines developing a total of 2,060 s.h.p. at 5,500 r.p.m., two new l.p. turbines developing 2,740 s.h.p. at the same speed, two original h.p. turbines developing a total of 4,970 s.h.p. at 3,120 r.p.m., and finally the two original l.p. turbines developing 6,230 s.h.p. at 2,800 r.p.m.

In addition to the main propelling machinery, alterations have been carried out to various auxiliaries, while new single collar thrusts have been installed and new propellers fitted.

The French licence for the construction of the Velox boiler is held by the Compaignie Electro-Mécanique, Paris, for Brown Boveri & Co., Ltd., Baden.

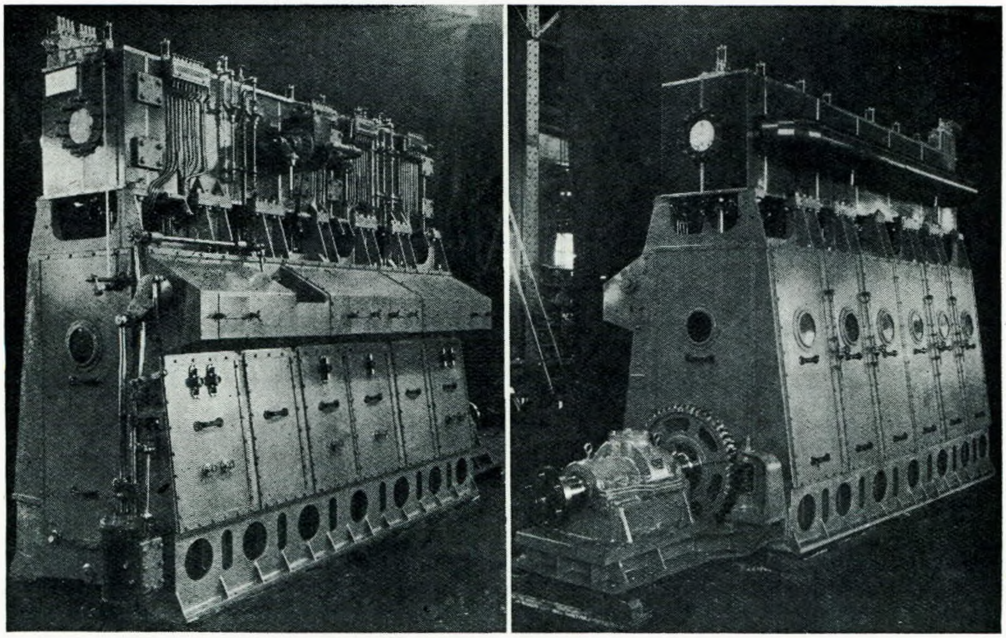
American Steam Reciprocators with Forced Lubrication.

"The Marine Engineer", December, 1937.

We have published several articles during the past few months which have shown that American marine engineers are fully alive to the desirability of producing more economical steam installations. A field which they have made peculiarly their own is that of the turbine-driven tanker; American water-tube-boilered geared-turbine-driven tankers have shown remarkably fine performances in numerous instances, and in the layout of the machinery and boilers real ingenuity and courage have been displayed in many cases.

In the reciprocator field American efforts are not so well known to European engineers, although we have published articles on combination machinery developments and uniflow engines on the other side of the Atlantic which show that these sides of the subject are not being neglected.

The foregoing thoughts lend interest to the illustrations which accompany this article. They show the latest engine types evolved by well-known American marine engineering firms. These engines are not in any sense unusual or particularly advanced from the thermal standpoint, but they are very interesting as examples of the employment of forced lubrication on engines of this type. There is no question as to the desirability of providing an enclosed crankcase and forced lubrication; it is a

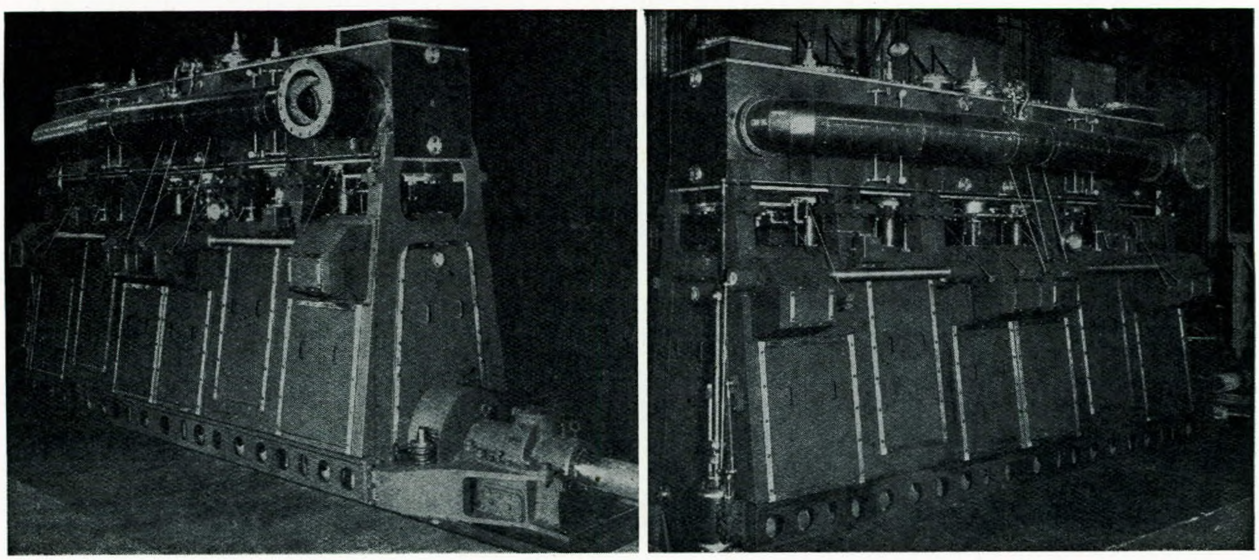


A small four-cylinder triple-expansion engine with forced lubrication and aluminium alloy entablature and bedplate.

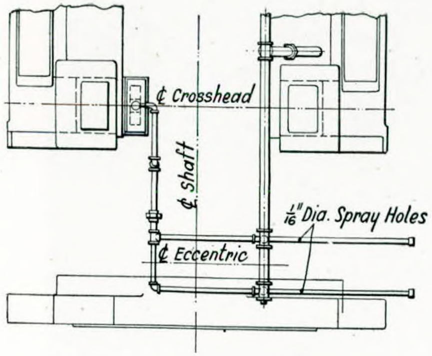
desirable improvement to a marine steam engine, particularly if the revolution speed is higher than average practice. The improvement suffers from one serious drawback—first cost; and at the present time it is unlikely that many British owners would be likely to pay for the feature, however desirable it may be from the maintenance and general mechanical standpoints. While, therefore, it seems unlikely that the enclosed crankcase, forced-lubricator marine steam engine of the low-speed type will make headway over here, the special service American engines illustrated are not without interest. A point in their favour, in passing, is the very neat appearance of these engines, although it might be thought that the numerous oil pipes and oil boxes seen on the cylinders of one of the engines strike an inharmonious note in an otherwise neat and very modern design.

Product of a Diesel Builder.

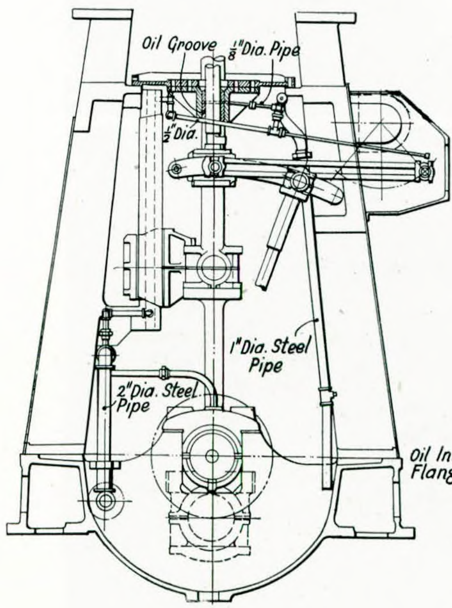
The engine shown in the illustration at the foot of this page is one of several which have been built by the General Machinery Corporation (Hooven, Owens, Rentschler Division), of Hamilton, Ohio,

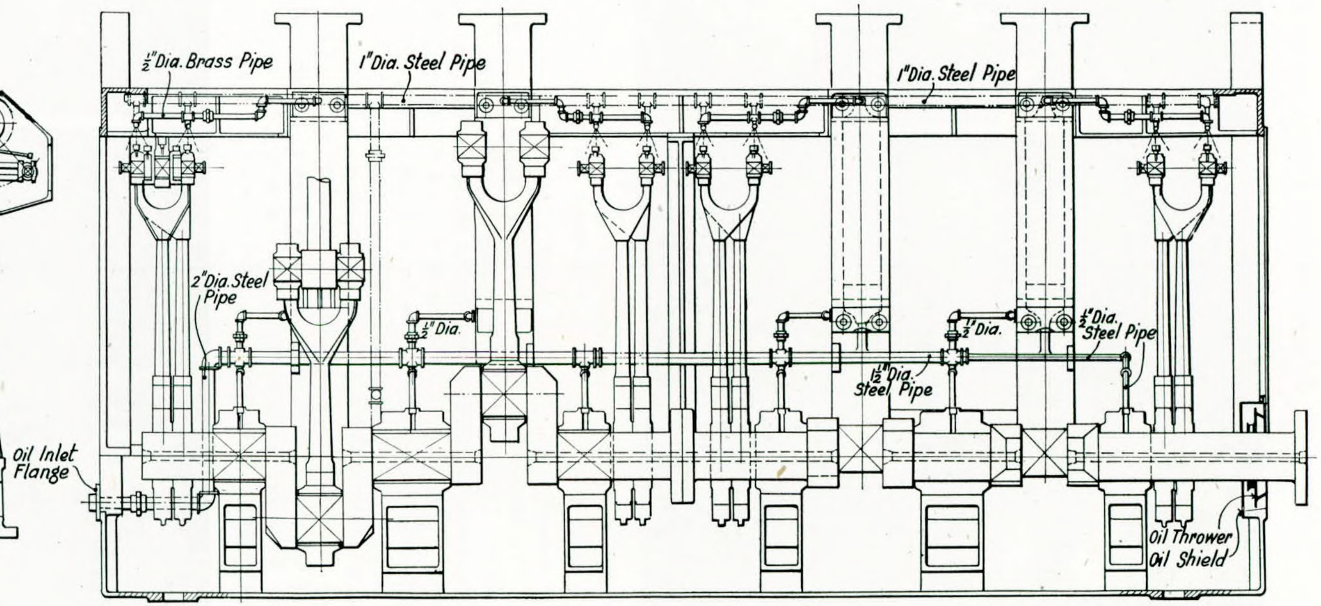


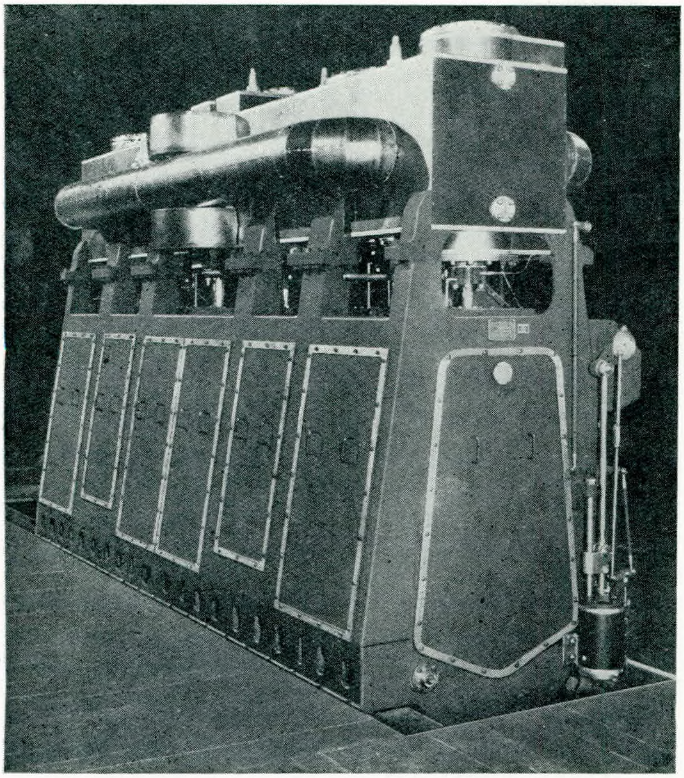
Two views of four-cylinder triple engines of American manufacture which have a very complete system of forced **lubrication.**



As these sectional views show the pressure lubrication arrangements of the Hooven, Owens, Rentschler triple-expansion engine embrace the links, while the accessibility of the oil piping will be observed.







Accessible reversing engine and large crankcase doors of the Hamilton-built engine are shown in this illustration.

and were built for the Federal Barge Line which operates a large number of tow boats on the Mississippi River. For the large boat they required high-speed engines, as the boats were being built for propeller twin-screw drives. For their operation it was required to have high economy, and the engine should operate smoothly without vibration or torsional vibration. It was also required that they be made for forced lubrication, so as to minimise the attendance.

The engines which were built to fulfil these conditions are illustrated, and have cylinder sizes of 16in., 26in., 32in. and 32in. by 24in., and are triple-expansion units developing 1,200 i.h.p. at 200 r.p.m. The steam pressure is 250lb. per sq. in., 100° F. superheat, and operates condensing. The engines are fitted with Stephenson link-motion, and the cylinders are separated from the frame by a division cover. In addition to the stuffing box on the piston rod, there are additional stuffing boxes on the division diaphragm.

The frame of this interesting American engine is entirely enclosed, and has a number of large doors for access. The valve gear is also enclosed in the frame and forced lubrication is used throughout for all reciprocating parts. The lubricating oil is furnished by an independent pump, and is forced through the main bearings, by way of the shaft, to the connecting rod and crosshead slipper. For the oiling of the valve gear separate lines are led up to the parts and lubrication is furnished by spraying oil on the parts only having a very slight motion; no lubrication is provided for the cylinders. The piston rod stuffing box, however, is lubricated by a special timed lubricator.

The first pair of engines of this type was built for the steamship "Illinois", and recently the Hamilton firm built two more for the "Iowa", which is a duplicate of the "Illinois". The engines have performed very satisfactorily, showing high economy and very low maintenance figures. The steam engines are preferred by many American river operators because of their simplicity, and also because of the overload which they can carry at critical times without danger to the parts. Accompanying this article are diagrams of the forced lubrication arrangements of the engine which will no doubt be of general interest.

Aluminium Construction.

The two views of a triple-expansion engine at the head of this article are interesting as they show a very similar design tendency to that of another Ohio engine builder, the American Ship Building Company, of Cleveland.

The general appearance of this engine is good and it is interesting to note that in the interests of weight-saving the entablature and bedplate are cast in aluminium alloy. The engine is a four-cylinder triple, with cylinders 10in., 17in., 21¼in., and 21¼in. by 18in. stroke, and it was built for the Vesta Coal Company, of Pittsburgh, Pa. The working steam pressure is 275lb. per sq. in. (gauge), and the steam is superheated 100° F.

Mechanical Stokers for Scotch Boilers.

"Shipbuilding and Shipping Record", 6th January, 1938.

The insistent demand for economy in the operation of all classes of tonnage is leading to a re-awakening of interest in the possibilities of the mechanical stoker as applied to marine boilers. In particular, on coal-burning steamships of the cargo liner class, it is recognised that a reliable form of mechanical stoker for use in conjunction with the familiar type of Scotch boiler usually installed in these vessels would merit careful attention. With this end in view, an established British firm which has hitherto been associated with marine boilers of the water-tube type has brought forward, through one of its subsidiary companies, a new type of mechanical stoker specially designed for fitting to the furnaces of Scotch boilers. The stoker is of the retort type—a type which has found wide acceptance in conjunction with large water-tube boilers on land—in which the coal is fed by means of rams into retorts formed between an arrangement of tuyere grates. There are two retorts to each furnace, the coal being forced upwards through the incandescent mass above, this arrangement, it is claimed, leading to smokeless and therefore highly efficient combustion. The upward movement of the rams at the same time pushes the incandescent mass forward and the ash and clinker ultimately fall over into the bottom of the furnace, whence they are raked out in the usual way. The hoppers in front of each furnace are hand fed, so that the mechanism which is steam operated is of the simplest type possible.

High-speed Diesel Generators.

"Shipbuilding and Shipping Record", 6th January, 1938.

The gradual adoption of higher speeds of rotation for diesel engines is particularly noticeable in those units which are employed for driving electrical generators. A good illustration of this fact is to be found in the type of engine which, we learn, is to be employed for driving the two emergency generators on No. 552, the sister ship to the "Queen Mary". The engines are being supplied by a firm which has usually been associated with diesel engines for land purposes, and they are of the six-cylinder type, the generator having an output of 75kW. when the engine is running at 900 r.p.m. Actually, the complete range comprises engines having from two up to eight cylinders, the latter developing 200 b.h.p. at a speed of 1,200 r.p.m. The cylinder head is of the Ricardo-Comet type, the pistons being of aluminium alloy with ample ribbing to ensure good heat transference. The bed-plate and column form a single casting, but each cylinder is also a separate casting, with removable liners, there being large cooling water spaces and two doors for cleaning purposes. The fuel pump is of the Bosch monobloc type, and Bosch automatic fuel valves are fitted. Forced lubrication is employed for all the main bearings, the bearings throughout being of exceptionally large area in proportion to the loading. A feature of the set is that it is entirely self-contained, the complete unit comprising engine generator and radiator, with electrical self-starting equipment, the push button being mounted on a panel with the oil-pressure gauge and the tachometer.

The Shipowners' Problem.

To Modernise or to Build?

By A Special Correspondent.

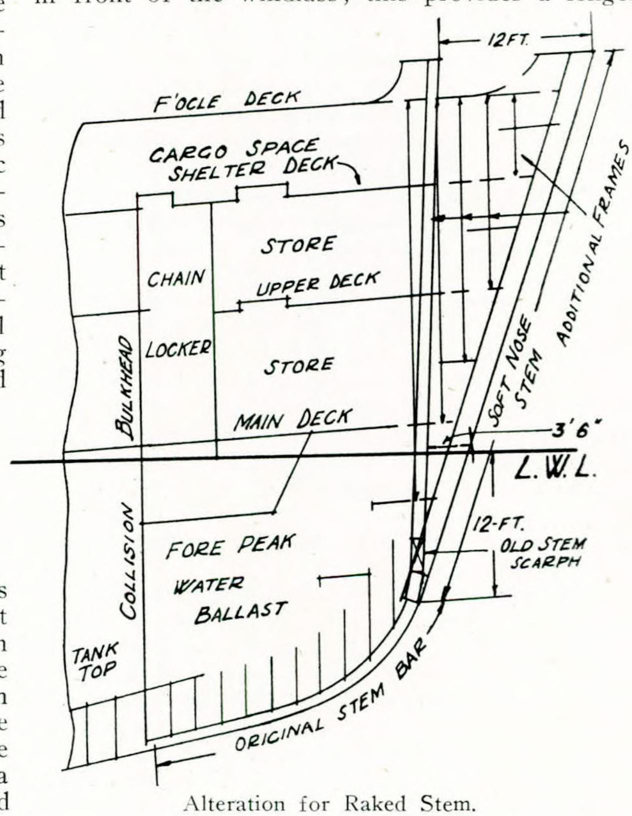
"The Shipping World", 26th January, 1938.

In these days of higher shipbuilding costs many shipowners are considering whether it is not cheaper to modernise existing vessels rather than build new tonnage in order to compete with the latest types of vessels which have recently been built. Several ship and engine repairing firms are well qualified to carry out such work. The personnel and facilities available have reached a high standard of efficiency in the repair works, and extensive alterations are common. The modern vessel is an advance on former practice, and hull and engines have been greatly improved. The skill necessary for the successful reconditioning of a vessel is as great as that required for building one. Often a vessel may have the length increased or decreased, or have decks, bulkheads, and other important structural members altered. The engines and boilers can be taken out of a vessel, and new reciprocating engines, turbines or oil engines, installed, or she may be converted from coal burning to oil fuel. A ship is often not recognised as the same vessel after extensive alterations, for her appearance may be quite different after the reconditioning.

At present stems of the soft-nosed description are much in favour, and many vessels have the old straight stem bar removed, and a raked plate stem fitted above the light water-line. From the accompanying illustration it will be observed that the alterations required are not so large as might be anticipated for the fitting of a new stem. In many respects the raked soft nose stem is preferable to the straight stem bar. It provides a better run and ending for the forward waterlines, and in the case of a head-on collision, protects the under-water portion of the hull. When repairing damaged shell plates in way of the stem, it is a great convenience.

Raked or Straight Stem.

With the raked stem more space is available in front of the windlass; this provides a longer



Alteration for Raked Stem.

run for the cables between the deck lip of the hawse pipes and the chain pipes, and the hawse pipes can lie at a large angle, thus preventing a quick nip to the cable at the deck. In the past accidents have happened through the hawse pipes having been placed too close to the windlass and being too straight, thus causing a quick nip to the cable. In the illustration shown the bar stem is retained at the fore foot up to the scarph, which is situated above the round of the stem. The top stem bar from this scarph to the spirkitan plate is removed and the plate stem is fitted from about 12 feet below the load waterline to the top. The bottom stem bar laps into the bosom of the plate stem, and additional frames are necessary forward owing to the large rake of the stem. To suit the new design at the fore end the existing fore peak frames are altered, and the shell plating overlaps are altered to suit the increased length of plating. On the panting stringers and lower decks wide breastworks are fitted and on the upper decks the stringers are increased in length as well as the plating, if necessary.

There is no limit to the demands for increased speed on land, sea, and in the air. Passenger and cargo vessels are now designed for higher speeds than formerly, and the present-day ship design reduces the hull resistance and is more economical on fuel for a specified speed. Is it more economical to construct a new vessel and scrap the old one, or to modernise the old vessel? This is the question which shipowners may ask, and the answer depends on freights and building costs. Because of the present high building prices it may be more profitable to modernise the old vessel. The increased deadweight carrying capacity of the modern vessel, compared with vessels built a few years ago, is not great, but there is a difference in speed. If the vessel has been well maintained during her life, it may be an economical proposition to recondition her engines to increase the speed, at the same time altering the hull to suit modern standards.

Marine Engineers' Examinations.

"The Engineer", 7th January, 1938.

The Report of the Departmental Committee on Examinations of Engineers in the Mercantile Marine, under the chairmanship of Mr. Maurice Gibb, which was appointed on 6th January last by the President of the Board of Trade, and referred to in a Journal note of 22nd January, has now been published by the Stationery Office. The Committee finds that the standard of the examinations is an appropriate one, and should not be lowered, but it makes a number of proposals which should, in its view, make the examinations more suitable for the testing of engineers responsible for the operation and maintenance of marine engines, and bring them more into line with the increasing tendency in modern education towards the avoidance of unnecessary examinations. The Committee recommends that Steam and Motor Certificates, and Endorsements, First and Second Class, should continue to be issued, but a candidate who has had adequate experience on both types of ships should, if he so desires, be allowed to take a single examination for a Combined Certificate enabling him to serve in his appropriate grade in either steam or motor ships. Candidates possessing a Second Class Certificate of either kind, steam or motor, should be allowed to proceed to the examination for a First Class Certificate for the other type of ship, provided that the necessary sea service in ships of the type for which the certificate is desired has been obtained. The normal period of workshop service should be retained at not less than four years, but candidates who have been engaged on work suitable for the training of a mechanical engineer in the manufacture of machinery should not be obliged to serve additional time in marine engine workshops or on regular watch at sea. The present period of qualifying sea service for Steam or Motor Certificates should be retained at eighteen months, but the conditions regarding the kind of training required during that period should be revised, and provided the necessary experience has been obtained on the main propelling plant, and on boilers in the case of candidates for Steam or Combined Certificates, service on auxiliaries, run in conjunction with the main propelling machinery and essential to its running, should be allowed to count at full rate, and, further, service on other auxiliaries and, in certain circumstances, "day work", should, subject to conditions, be allowed to count at half rate. For a Combined Certificate the minimum period of qualifying sea service should be twenty-one months. Service on home trade ships should continue to count at two-thirds rate compared with foreign-going service, but time served as third and fourth engineer, where these engineers serve as senior engineers in charge of the entire watch, should be allowed to count at this two-thirds rate. Various suggestions as to the examination are made.

Symposium on Corrosion Testing.

"The Engineer", 24th December, 1937.

The American Society for Testing Materials has issued in book form a symposium on corrosion testing procedure which was held at its 1937 meetings. The book contains seven papers by twelve authorities, and covers the principles of corrosion testing, atmospheric testing, salt spray testing, methods for copper alloys, soil corrosion, liquid corrosion, and electrical resistance method of determining corrosion rates.

The Importance of Sound Insulation.

"The Shipping World", 1st December, 1937.

The importance of quiet running turbine reduction gearing is generally recognised, as is the value of quiet deck machinery in vessels carrying passengers. The noise in the engine-room of the average motorship leaves much to be desired, however, and particularly does this apply to auxiliary Diesels. To silence the exhaust is a fairly easy problem, but it is surprising how often this is not satisfactorily accomplished. The scavenge pump and cylinder air inlets of two-stroke and four-stroke Diesel machinery respectively are in many instances not effectively silenced, although low-cost suction air silencers are available for this purpose. So far as the mechanical, combustion, and other noises accompanying a marine Diesel are concerned, one way of minimising these is to enclose the engine in a special sound-proof bonnet or casing, as has been done in one or two high-speed-engined Diesel yachts and in certain stationary plants. This is an effective measure, but one which may not appeal to the average superintendent engineer. Something of a compromise is achieved by enclosing the valve gear of the engine (usually the source of most of the mechanical noise) in a light, easily-detachable casing lined with sound-deadening material. This leaves main engine noises uncared for, and it would seem that the methods suggested for small and comparatively small engines are not practicable where large main propulsion Diesels are concerned. Probably the best plan to adopt in order to prevent noise transmission to passenger quarters is to line bulkheads and casing with sound-insulating panels. Several varieties are approved by Lloyd's Register and the other classification societies, and fireproofness is one of the qualities of these products. Such a plan of sound-insulation has been carried out in a recent cross-Channel motorship, and certain other motorships have been greatly improved from the passengers' standpoint by fitting such insulation since they went into service.

Marine Economisers.

"The Shipping World", 1st December, 1937.

On shore the employment of the economiser as an aid to economy in fuel expenditure has long been established. It is strange, therefore, that its employment on shipboard is a comparatively new development. As yet the economiser is principally employed by American and German marine engineers, while at least one well-known British firm —the makers of the widely-used Green economiser —is prepared to fit it into ships. Briefly explained, the economiser consists of a special form of heating surface in the path of the flue gases from the boiler through which the feed water passes on its way to the boiler. Surely, it may be argued, the economiser can achieve no real economy if the boiler plant is already fitted with an air preheater? In actual practice the economiser is complementary to the air preheater. It is known that pre-heating of the furnace air is not advisable beyond a certain point; economy is then obtained at the expense of boiler upkeep costs. At this point in the search for further economy, the economiser comes into its own. Not only is boiler maintenance not increased, despite a useful gain in economy, but that gain is obtained without the further expenditure of live steam on feed heating. The first cost of the economiser, having regard to the useful degree of economy obtained, is not high. The economy credited to the economiser is of the "high grade" type, at the top end of the scale, where additional savings are hard to make and are usually costly to achieve. In its modern form the marine economiser is compact, moderate in weight and does not occupy space which might otherwise be usefully employed. In the Foster economiser, which has been used so successfully in several ships and which is made under licence in this country by Messrs. E. Green & Son, Ltd., the tubular surface of the economiser is enormously increased by fitting cast iron gills or fins to the outside of the tubes forming the economiser.

Elimination of Cylinder-liner Wear.

"The Motor Ship", February, 1938.

It is always a delight to escape from stereotyped ideas, even in the presentation of technical papers, and Mr. G. J. Lugt (who is responsible for the design of the Werkspoor engines) called a new tune not only in the title of his paper ("Diesel Varia"), reproduced at some length in this issue, but also in its matter. One cannot exclaim with the Queen of Denmark, "more matter and less art", for the paper, if presented with some art, is full of matter. To wander in Mr. Lugt's discursive way on some of the various, and always intriguing, problems which are facing the designer must be attractive in the extreme to one who is daily confronted with the complex questions still arising in the production of modern Diesel machinery.

We suggest that all superintendent engineers and Diesel-engine designers, as well as those shipowners who can spare the time to interest themselves in their machinery, should study Mr. Lugt's paper with care. In this note we refer only to one point which he raised, namely, the old bogy of cylinder-liner wear. He remarked that experience gained with chromium-coated liners seemed to corroborate Mr. Ricardo's suggestion that what we call liner wear is not only wear but also erosion. Chromium is more or less immune against erosion, hence the success of the chrome-hardening process. Apart from this, however, and referring to the normal cast-iron liner, Mr. Lugt emphasizes the importance of the relative hardness of liner and ring.

Last month we referred to a discussion on the subject at the Diesel Engine Users Association, when a well-known engine designer stated his preference for a ring softer than the liner, whereas a piston-ring manufacturer preferred it the other way round. Mr. Lugt carried out some experiments on the subject, as a result of which he concluded that the best results are attained if both materials are of about the same hardness and are of an identical nature, contrary to experience with metals other than cast iron. If there is a difference in hardness, the harder material wears to the greater extent. The addition of chromium and phosphorus reduces the rate of wear, whereas nickel and manganese have no influence. Titanium and vanadium seem to have a favourable effect.

Welding in Shipbuilding.

Practices Adopted in Hull and Diesel Machinery Constructon.
By E. Dacre Lacy.

"The Motor Ship", February, 1938.

It has been an acknowledged fact for several years that, as far as electric arc welding in ship construction is concerned, this process is particularly applicable to the construction of oil tankers. The late Sir Joseph Isherwood, an advocate of welding as a means of connecting the various structural members of a ship's hull, stated: "Welding should be used particularly in oil carrying vessels, where every rivet is a potential source of leakage". Welding is now being used quite considerably in the construction of oil tankers, especially in the U.S.A., on the Continent, and to a somewhat lesser extent in this country.

Large tankers have been built in which all the transverse and longitudinal bulkheads, including the connections to the hull, are welded; and two tankers are at present being built, each about 500ft. in length, where the hull is entirely welded to the exclusion of riveting. The reason why welding is the ideal method of construction of such vessels is because the structure can be made into one jointless piece instead of a large number of parts, thus eliminating leakages at seams, rivets and other connections.

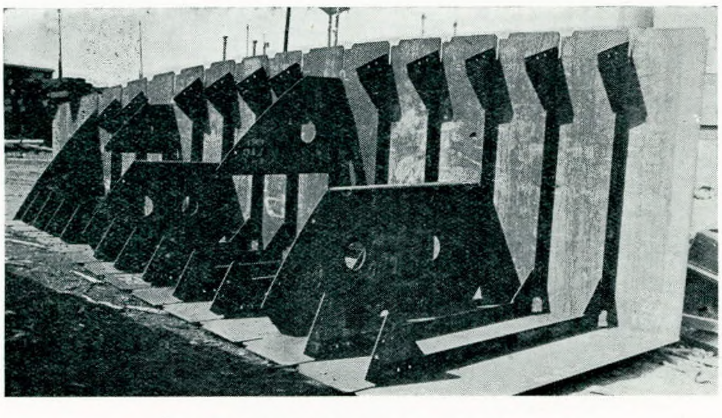
Leakages are, of course, caused by faulty rivets, faulty material or workmanship, and corrosion. The replacing of rivets is a costly matter, and the maintenance of a riveted vessel causes a considerable delay and hence expense in the upkeep, of a small ship, while on the other hand, as another well-known authority has stated: "With welded bulkheads there should be no leakage; therefore, different grades of oil can be carried in adjacent tanks without danger of contamination. The internal surfaces of the tanks, being clear of rivet heads, are easy to clean, which saves considerable time, also by keeping the surfaces clean corrosion is arrested and reduced to a minimum; there should be little, if any, corrosion caused by galvanic action, as all the materials should be of about equal potential quality. The life of a welded tanker should therefore be longer than that of one riveted, under equal conditions of service. It should be possible to prepare a welded oil tanker for carrying light or clean oils, after carrying heavy or dirty oils, more quickly and effectively than a riveted tanker, because there are no joints to harbour oil or allow seepage. In welded construction, the plating is not weakened by rivet holes, so that the full strength of the plate is maintained and there should be no weak points for the starting of cracks. In order that the welded structure shall be as solid as possible and to reduce the stresses on the welding, it is necessary that the parts should be fitted to bear metal to metal before welding. It is advantageous for strength to use the minimum number of parts as well as to reduce the cost of fitting and welding, and upkeep and repairs in case of damage".

The use of welding allows the shipbuilder to reduce the cost of building his vessel owing to the constant saving he can make in steel material by using this method. On the other hand, the owner obtains increased deadweight and a reduction in maintenance and repair costs—he can carry his cargo of oil without risking the danger of damaging parts of the ship's structure.

Welding in the M.S. "Port Jackson".

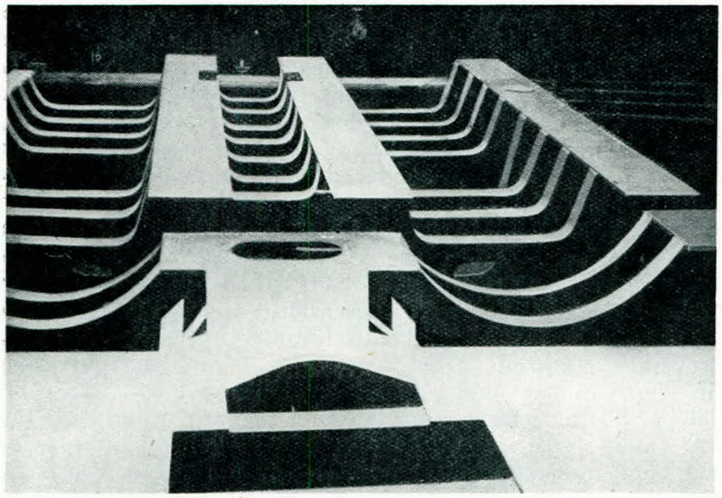
The following details relating to a partially welded motor ship, and to two all-welded oil tankers indicate the trend of modern welding development.

Messrs. Swan, Hunter & Wigham Richardson, Ltd., of Wallsend-on-Tyne, recently built the twin-screw motor ship "Port Jackson" to the order of the Port Line. The dimensions are as follows: length b.p., 495ft. 6in.; breadth, moulded, 68ft.; depth moulded to upper deck, 41ft. 6in.; depth moulded to second deck, 33ft. 5in.



A welded side ballast tank.

She was built under the special survey of Lloyd's Register for classification 100 A.1, with freeboard and to fulfil the Board of Trade requirements for cargo vessels, and is designed for the carriage of refrigerated and general cargo. Electric welding was extensively used in the construction. The main items include the main watertight bulkheads, oil fuel bunkers, engine casings, deckhouses, deck plating pillars, shaft tunnels, generator seats, auxiliary engine seats, masts and outriggers, derrick posts, fresh-water and sanitary tanks, skylights, ventilators, and all gas-tight bulk-



All-welded engine seating for a twin Diesel-electric installation.

heads throughout the cargo spaces.

The bulkheads are plated transversely and are fabricated in halves on skids, a vertical lap weld being arranged close to the centre line to assist in fairing on the ship, thus giving a good shell-tight connection with the boundary bar.

Bridge and Boat Decks.

The butts and seams of these decks were joined by electric welding and the beams were riveted thereto. The decks were erected complete, and bolts through the beams kept them in position. The beams were riveted after the decks were welded, thus combining riveting and welding. The studs used for holding wood decks, etc., were welded to the plating.

Deckhouses.

Large panels of plating were welded complete with "flat" stiffeners, the method of welding on skids being similar to that of the bulkheads. A foundation bar 4in. by 3½in. was fitted all round the houses to provide the deck connections.

Auxiliary Engine Seats.

These seats were marked off on the ship and were cut and prepared by the plater, then erected and tack welded. A large quantity of overhead and vertical welding was required when finishing the seats. After completion, they presented a neat appearance and were very strong.

Generator Seats.

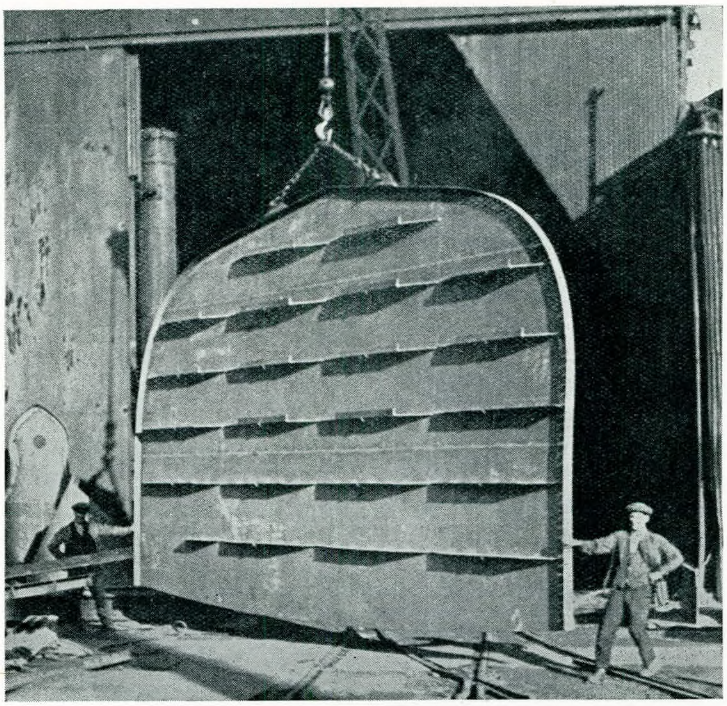
These are three in number, and are approximately 23ft. long by 5ft. 6in. wide by 2ft. 3in. in height, and are subject to constant vibration; thus a strong and efficient construction is necessary. The top plate is 0.875in. thick, the side plates are 0.75in. thick, and the athwartship plates and brackets are 0.67in. thick. The plates were cut to shape and assembled with tack welds, and the finish welds made in the "downland" position whenever possible.

When completed, the seats showed a considerable saving in weight over riveting, as well as giving a strong and neat appearance.

All-welded Ships.

A description of the construction of two electrically arc-welded Canadian tankers was given recently in a paper entitled "The Electrically Welded Ships 'Bruce Hudson' and 'Transiter'", by E. R. Macmillan, of the British Corporation Register, Montreal. These vessels are the first ships built in Canada in which all the connections were electric arc-welded. The builders were the Horton Steel Works, Ltd., of Fort Erie, Ontario.

The "Bruce Hudson" was specially built for shallow-draught service to the order of Lloyd Refineries, Ltd., of Port Credit, Ontario, and was designed for the carriage of heavy and light oil in bulk on the Great Lakes and canals of Canada. The dimensions of this vessel are 165ft. in length, 30ft. in breadth, and 11ft. 3in. in depth, and she has a cargo capacity of approximately 875 tons on a draught of 9ft. 6in. She is framed transversely throughout, and has four main cargo holds, each



All-welded oil-fuel bunker bulkhead.

having a longitudinal oil-tight bulkhead on the centre line, making a total of eight tanks in all.

The "Transiter" was also built for the Great Lakes and canal service, and was designed by Messrs. Lambert and German, Montreal. The dimensions of this vessel are as follows :—

Length 180ft., breadth 34ft. and 15ft. 6in. depth, moulded. She has a trunk 2ft. 6in. in height and 32ft. in width extending the full length of the cargo compartments. Her deadweight is approximately 1,450 tons on a draught of 12ft. 7in. She is framed transversely and the frames are supported by longitudinal girders on the sides and bottom, combined with transverse webs spaced about 8ft. apart, diagonal struts being a feature of the web construction in way of the bilges, which is in accordance with the designers' conduit-bilge construction. The four main cargo holds are subdivided by a longitudinal oiltight bulkhead on the centre line, making a total of eight compartments.

Both the above-mentioned vessels were entirely welded, and their design shows a remarkable development in the utilisation of arc welding not only for parts of the superstructure but for their actual hulls.

Welding has for a number of years been making considerable strides in the field of marine engineering. One of the pioneers, as regards using this process in the building of marine-engine parts, is Smith's Dock Co., Ltd., South Bank-on-Tees. In a vessel now being built by this concern, over 200ft. in length, there are three decks almost completely welded, and the deckhouse, main-engine seats, bulkheads and oil-fuel tanks are also welded.

The practice of fitting welded main-engine seatings has proved entirely satisfactory, both for steam and Diesel machinery, the welded seatings being more rigid than when riveted and resulting in a greater steadiness of the running machinery. Welded oil-fuel bunkers, ballast tanks, feed and fresh-water tanks have shown a greater degree of tightness than when riveted, resulting in the speeding-up of tank testing during construction. It is for this greater immunity from leakage that steel decks over accommodation are welded; in no case has there yet been a report of leakage in the vessels referred to. Deckhouses, casings and hatches are also welded.

As regards Diesel-engine structures, electric welding is now playing no inconsiderable part in their construction. The first all-welded unit was the Doxford opposed-piston Diesel engine, and welding has made it possible for the builders to install this engine in small vessels where it would not otherwise have been possible.

Five-cylinder Diesel units built by Messrs. Barclay, Curle and Co. have also been completely fabricated by electric arc welding in their own works.

Dry Ice.

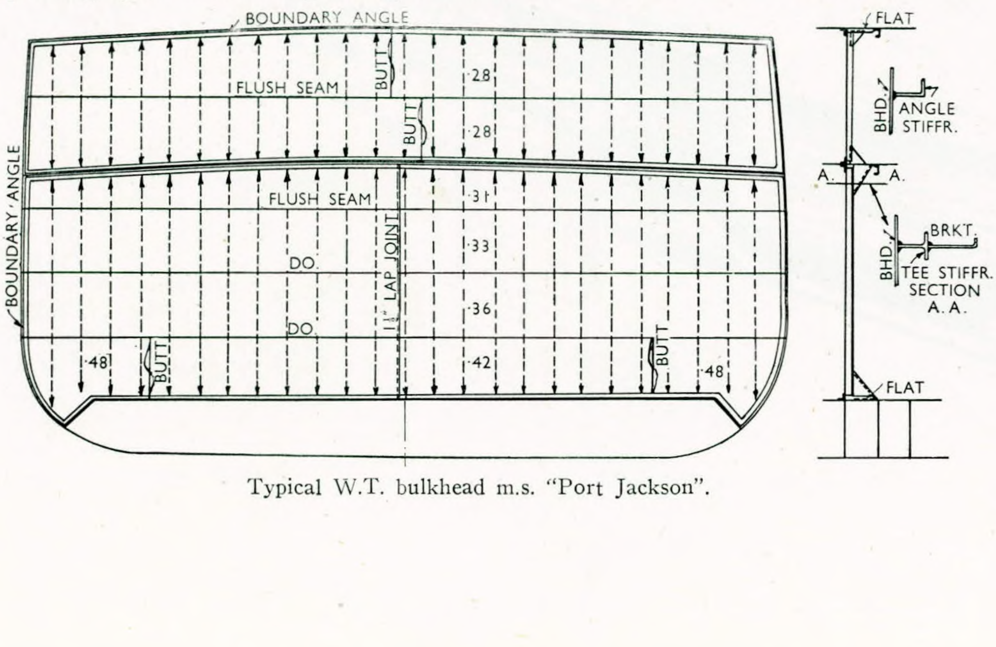
"Ice and Cold Storage", September, 1937.

During recent years there has been an increasing demand in this country for solid carbon dioxide. At one time, its use was practically confined to the ice cream trade; it then entered more fully into the field of refrigerated transport for the carriage of such perishables as frozen and chilled meat, fish, dairy produce, eggs, fruit, flowers, etc.

Other Uses.

Other uses to which dry ice is now put include the production of liquid or gaseous CO_2, as an auxiliary to other refrigeration systems, because of its beneficial effect on certain food products, and for dispensing cabinets, small display cases, and vending machines. It has also entered into a vast industrial field of usefulness, and is used in connection with the cold shrinking of metal parts, the cold treatment of special steels, the freezing of ice cream and foods, the grinding of dyes and gummy products, for cooling and holding aluminium alloy rivets, the manufacture of neon lights, the manufacture of radio valves, the manufacture of golf balls, the trimming of rubber products, concentration of ether or solvents by freezing out the water content, and the cheapest source of carbon dioxide for the gas storage of foods in ships, cellars, and refrigerated and non-refrigerated compartments generally.

The process, Carbo-Ice, starts with the combustion of coal and coke in a specially designed gas producer, which results in the production of a combustible gas which is burned in a furnace where



Typical W.T. bulkhead m.s. "Port Jackson".

steam and smoke are produced. The steam is used to supply the power to operate the entire factory, as well as to supply the heat to work the chemical processes, while from the smoke the raw material (carbon dioxide) is extracted from which the two products, Carbo-Ice and Carbo-Gas are made. The smoke is subjected to various cleaning and purifying operations which have been perfected and developed by the engineers of the company, and it is due to the efficiency of these processes that the high degree of commercial purity is attained.

A New Impeller Pump.

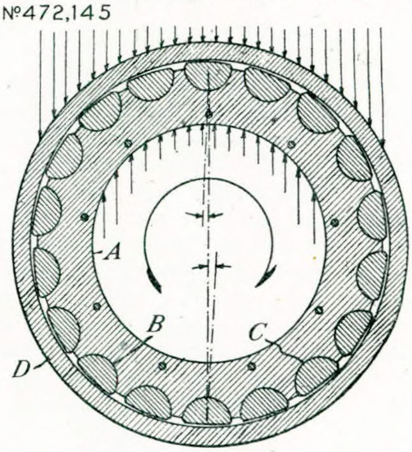
"The Engineer", 26th November, 1937.

An interesting type of butterfly valve impeller pump, invented by Captain Louis le Clezio, is being developed in this country by International Technical Developments, Ltd., of Thames House, Millbank, London, S.W.1. As may be seen from our drawing, the working principles of the pump are very simple. It comprises a piston-rod which reciprocates coaxially within the pump barrel. At the end of the piston are set two flaps on a common hinge, which permits them to move freely. Stops at the end of the piston prevent the flaps opening out sufficiently for their peripheries to touch the sides of the pump barrel. The piston reciprocates at 500 to 1,000 strokes a minute, and once it is in operation it has been found that the flaps do not touch the barrel sides, even if the stops are removed. This is due to the fact that the moving column of liquid keeps flowing in the same direction, even on the return stroke, and at the end of the forward stroke considerable hydrostatic pressure is built up in the annular space between the flaps and the barrel, and this pressure is balanced by opposing forces in the flaps. In this manner, a practically frictionless hydrostatic seal is formed between the flaps and the pump wall, without actual metallic contact taking place between them. We are informed that the pump has a mechanical efficiency of about 75 per cent.

Oil Film Bearings.

"The Engineer", 5th November, 1937.

This journal bearing has some of the characteristics of the Michell bearing, in that it depends for its lubrication on the maintenance of a series of wedge-shaped films of oil between the moving parts. The journal A is provided with semi-circular

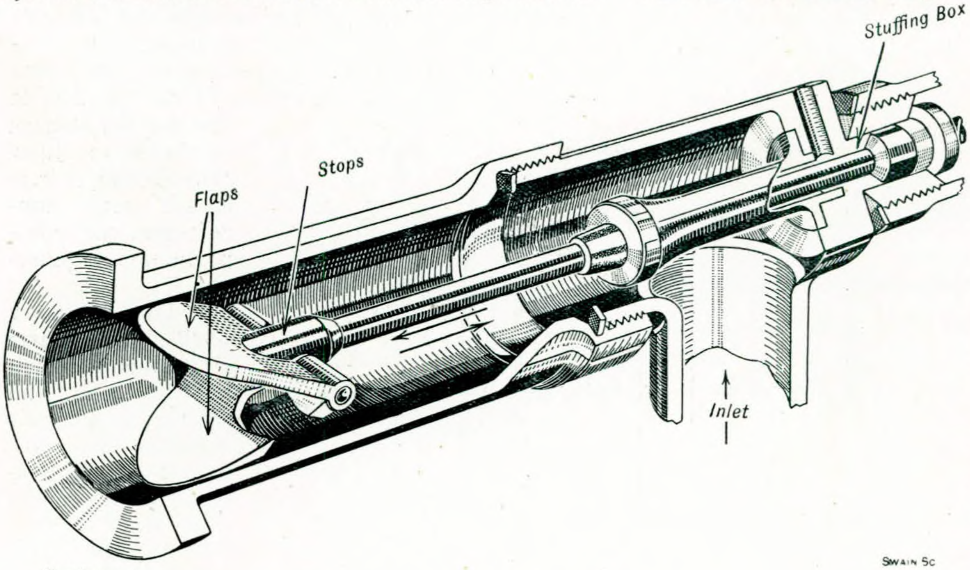


grooves B in which there are lodged semi-circular bearing blocks C that are an easy fit in the bearing D. It is suggested that the load is carried by the blocks through an angular zone of 120°, and that during the remainder of their revolution they are free to pick up a fresh supply of lubricant.—17th September, 1937.

Fuel and the Diesel Engine.

The Oil Suppliers' Point of View.
"The Motor Ship", February, 1938.

[In view of the various discussions which have lately taken place on the subject of the provision of suitable fuels for marine Diesel engines and the articles which have been published in "The Motor Ship" the following contribution representing the views of one of the leading oil supplying concerns in the world will prove of interest and value to shipowners and their engineers.—Ed.]



Butterfly valve impeller pump.

In "The Motor Ship" for October, 1937, appeared an article under the above heading dealing with the desirability or otherwise of making modifications to engines for different classes of Diesel fuels. The case was presented by the writer "L.J.H." somewhat from the chief engineer's point of view, which makes it all the more interesting since that functionary has to bear the brunt of any difficulties that arise when engine and fuel are in disagreement. He is at some pains to explain that the frequently used term "unsuitable fuel oil" refers in his writings at any rate to fuel which has characteristics impairing its efficient utilization in the particular engine under consideration. He admits "The same oil may meet the requirements of any other engine admirably, or, alternatively, the first engine might burn it efficiently with suitable modifications to the combustion arrangements".

Let us take the latter half of this sentence first. It is undoubtedly agreed that the adjustment desirable when changing from one fuel to another in the normal course of events should be an absolute minimum. Things would be easier if all manufacturers' designs made certain adjustments possible with the engine running, instead of by dismantling and resetting, but experience shows that even such simple adjustments are looked upon with disfavour by many marine superintendents. The reason for this is that they feel general knowledge of the combustion process is not sufficiently complete for the average engineer appropriately to vary the combustion arrangements to suit a particular grade of fuel, even if it could be done without loss of time. It is worth considering whether or not this view is necessarily correct.

Looking at the problem from the fuel point of view, it is a little difficult to understand just what solution "L.J.H." is endeavouring to present. Thus he evidently deplores the slightest variation in "analysis" whilst making it clear that a given fuel, unsuitable for one engine, may be excellent for another.

Economic Considerations.

The fallacy of the view that supplies of identical analysis should always be available at any given port or ports is, of course, that it takes no account of fluctuations in freights and market values, which sometimes combine to make one source of supply more economic than another. The fuel supplier is expected to tranship his products from the supply source which enables him to lay down a suitable quality at the most reasonable prices. It is a very moot point whether the purchaser would be prepared to see his price soar in return for a guarantee of continuity of supply from the same source when, from elsewhere, the oil company could supply more cheaply an equally suitable grade, though possibly of slightly different analysis. This question is, of course, considerably complicated when a vast number of bunkering stations are concerned; for, even if it were possible to keep an absolutely identical fuel in one port for years, it would be entirely contrary to reason to tranship that same fuel to all other ports, where it might arrive charged with freight costs far in excess of the price of other—and equally suitable—fuels from a more logical supply centre. Moreover, there is no single grade of Diesel fuel available in sufficient quantities to satisfy the world's motor ship bunkering demands.

A Common Misconception.

Just as it is natural that those not wholly conversant with all sides of the problem should be apt to overlook important factors bearing upon the "analysis" of supplies received, it is also abundantly clear that they are prone to serious misconceptions in their interpretations of such figures. When "L.J.H." complains, for example, about the inability to obtain fuel with a constant specific gravity at the same port, he perpetrates the very error which has given so much trouble in the past. Specific gravity is important from the point of view of fuel measurement; it is of no significance as a criterion of quality or from the engine performance point of view. An oil company responsible for the provision of Diesel bunker grades of reasonably constant burning quality will certainly endeavour to maintain within acceptable limits those specification points (such as viscosity) which its researches have shown to be of practical importance. But to supply such fuel in the most economical way it will, in many cases, not be found possible to give close guarantees for the less important points, such as specific gravity, which may naturally vary when it is sought to make the same type of fuel from crudes of different origins. Nevertheless, cases are by no means unknown where a ship's engineer has been so insistent in his demands for a certain constant specific gravity that he has been willing—and, in fact, anxious—to have quantities of boiler fuel oil discharged into his Diesel fuel tanks to obtain the specific gravity alleged to "give the best results"!

The ironical aspect of the problem is, however, that even if the impracticable fuel distribution scheme—the dream of so many ships' engineers—were attained, we should still not have disposed of the difficulties presented by those engines for which the universally available fuel of entirely constant characteristics would be "unsuitable". Moreover, in actual practice the best that can be done is to lay down a few grades of which the essential qualities remain reasonably constant. Now, whilst it is true that many engines are happily quite insensitive to the inevitable slight differences between individual consignments of a fuel grade, others are not entirely so. As we have seen, however, elaboration of fuel specifications by the process of adding to the number of physical or chemical characteristics which the buyer desires to stipulate within narrow limits means, in many instances, that the Diesel fuel must be manufactured at what is not necessarily the most economical supply point. This result, since it involves higher prices, would not be fair to those customers not requiring the stricter guarantees; whereas charging all the extra costs only to the few involved would put up their fuel bill considerably.

It would not be very far wrong to state that the "trouble-making engines" are usually amongst those types that have been developed and tested in the manufacturers' works on the very lightest fuels. Such a procedure has been a common practice, of which more and more engine builders see the danger nowadays, and it is in the best interest of the user to be aware of these facts. After all, an engine is useless without its fuel and, even though the engine is bought first and the fuel later on, the combination of the two should be in the minds of makers and users alike from the beginning. Estimating an engine's cost at £10 per horse-power, it will consume its own worth of fuel in only a few years' time. It must obviously be as illogical to buy an engine without due regard to the fuel it will have to use in service as, say, to buy a gun of abnormal bore.

So we must have fuel qualities within suitable, economically realizable ranges; but we must also have engines that will run on all such fuels without adjustment and without any appreciable variation in efficiency. Systematic work on the combustion qualities of Diesel fuels is a development of the last decade or so and finality is not reached. Attempts have certainly been made to grade fuels into different categories, and, of these, the new specifications of the British Standards Institution probably represent the best that can be done at present to define the suitability of a fuel for an engine and vice versa. The value of these would, however, be greatly enhanced if, instead of formulating lengthy special specifications of their own, engine manufacturers could, as a result of their tests, indicate the ability of their units to operate satisfactorily on the appropriate B.S.I. standardized grades.

No doubt such a procedure would tend to shift the present somewhat undue emphasis in marine work from the fetish of high specific power output, so that buyers would not be so influenced by low weight, size, and cost if, for these virtues, the digestive powers of a unit were obviously being sacrificed. There are certain large motor-ship fleets having engines which, whilst not, perhaps, the most compact units available, do show themselves capable of using satisfactorily all the normal Diesel fuel bunker grades, and it should be the aim to combine this inestimable versatility with the most modern possibilities in design.

BOARD OF TRADE EXAMINATIONS.

List of Candidates who are reported as having passed examinations for certificates of competency as Sea-Going Engineers under the provisions of the Merchant Shipping Acts.

For week ended 30th December, 1937:—

Name.	Grade.	Port of Examination.
Milne, George C.	2.C.M.	Newcastle
Sibbit, George E.	2.C.M.	"
Soden, Alfred E.	2.C.M.	"
Chard, Richard	2.C.	Liverpool
Drysdale, Edward R. ...	2.C.	"
Cronan, Jack	2.C.	"
Logan, Henry C.	2.C.	"
Sarginson, Keith M. ...	2.C.	"
Thompson, John	2.C.	Liverpool
Jones, David	2.C.M.	"
Lewis, William	2.C.M.	"
Howes, Edward J. L. ...	2.C.	London
Lindop, William E. B. ...	2.C.	"
Potts, Ralph D.	2.C.	"
Sellers, John	2.C.	"
Dobbie, James	2.C.	Glasgow
Lyon, Richard C.	2.C.	"
Miller, James	2.C.	"
Robbie, John C.	2.C.	"
Smart, Alexander B. ...	2.C.	"
Smith, Alfred	2.C.	"
Williamson, David J. ...	2.C.	"
Downie, Alexander D. ...	2.C.M.	"
Kerr, Robert	2.C.M.	"
MacIntyre, Duncan	2.C.M.	"
Hutchinson, Edward N. ...	2.C.	Newcastle
Hutchinson, Robert C. ...	2.C.	"
Larsson, John E. V.... ...	2.C.	"
Miller, Walter O.	2.C.	"
Ross, James	2.C.	"
Wallace, Neil	2.C.	"
Fittzen, James W.	2.C.M.	"
Gray, John G.	2.C.M.	"
Madden, Edward	2.C.M.	"
George, Alexander L. ...	2.C.M.E.	London

For week ended 6th January, 1938:—

Name.	Grade.	Port of Examination.
Brown, Charles H. W. ...	1.C.	London
Williamson, Leslie	1.C.	"
Colwell, Edwin W.	1.C.M.	Newcastle
George, Wilfred	1.C.M.	"
Peck, Francis H.	1.C.M.	"
Crawford, Andrew A. ...	1.C.M.	Glasgow
Robb, Fleming	1.C.M.	"
Newall, Robert	1.C.M.E.	"
Ferguson, James C.	1.C.M.E.	"
Small, John D.	1.C.M.E.	London
Huxley, Edmund	1.C.S.E.	Newcastle
Abbey, Edwin S. B.	1.C.S.E.	"
Ashley, William R. ...	1.C.M.E.	London

For week ended 13th January, 1938:—

Name.	Grade.	Port of Examination.
Hobbs, Alfred W.	2.C.	London
Laker, Arthur G.	2.C.	"
Logan, Henry C.	2.C.M.E.	Liverpool

For week ended 20th January, 1938:—

Name.	Grade.	Port of Examination.
Carter, Ronald	1.C.M.	Hull
Barrett, Maurice J.	1.C.	London
Herbert, Charles A.... ...	1.C.	"
Scotchmoor, John W. ...	1.C.	"
MacMillan, Murray A. ...	1.C.M.	"
Fagan, John	1.C.	Liverpool
McCormick, Archibald P....	1.C.	"
Turner, Norman	1.C.	"
Verity, Claude H.	1.C.	"
Cowen, George W. C. ...	1.C.	Newcastle
Heads, William T.	1.C.M.	"
Rees, Richard H.	1.C.	Cardiff
Trenchard, Lewis D. ...	1.C.M.E.	"
Waygood, Alan G.	1.C.M.E.	"
Wilson, Edward	1.C.M.E.	"
Short, George F. J.	1.C.M.E.	Liverpool
Kirby, John H.	1.C.M.E.	"
Boyle, Thomas C.	1.C.M.E.	"
Kennedy, William	1.C.M.E.	"
Grant, Wilfred E.	1.C.S.E.	London
Lawrence, Mortimer J. ...	1.C.M.E.	"
Lenaghan, Henry P. ...	1.C.M.E.	"
Powell, Francis J.	1.C.M.E.	"
Evans, William J.	1.C.S.E.	"
Proctor, James	1.C.M.E.	"
Scarth, George R.	1.C.M.E.	"
Cowell, George	1.C.M.E.	Newcastle
Peel, Robert A.	1.C.S.E.	"