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Modern Marine Condensing Plants and Feed Systems.

READ

By H. HILLIER

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CHAIRMAN: Mr. T. R. THOMAS, B.Sc. (Chairman of Council).

Synopsis.

IT is the purpose of this paper to show briefly the basic characteristics of the more important units in the various feed systems in use in modern steamships and to indicate the fundamental ideas underlying present day practice without attempting to go into a great deal of detail or to give full thermodynamic calculations.

The paper deals with the primary considerations and functions of the condenser, with a brief explanation of the regenerative type now in general use wherever high vacuum is required. Various types of reciprocating air pumps and steam jet air ejectors are discussed. The principles of the various feed systems in use at the present day are examined, particular consideration being given to the basic ideas of the closed feed systems used in conjunction with Scotch boilers and water-tube boilers.

In a steamship, the lower limit of the efficiency of the steam engine cycle is determined by the temperature of the sea and the efficiency of the condensing plant. Sea temperatures vary at different seasons and in different parts of the world, the approximate extremes to be taken into consideration being 45° F. and 85° F. In determining the size of condensing plant for a ship it is necessary, therefore, to ascertain the temperature of the sea in which the ship will do most of its work. For ships sailing in the North Atlantic, a temperature of 55° F. is usually assumed, whereas for ships running through the tropics, the temperature is generally taken as 80° F. or 85° F. Within the limitations of space available and permissible weight, the condensing plant should be designed to give the highest possible vacuum that the prime mover is capable of using efficiently.

Modern steamships are fitted exclusively with surface condensers and the broad principles governing the design of surface condensers have been well known for many years. Recent years have, however, seen a marked advance in the technique of design, and intensive experimental and research work has culminated in the wide adoption of what is known as the regenerative condenser.

The primary function of a condenser is to produce the lowest possible temperature at the exhaust steam inlet to the condenser or, in other words, at the exhaust branch of the engine or turbine, it being assumed that the prime mover is designed so that it will take advantage of the vacuum corresponding to such temperature. In discharging this function, the condenser should deliver the condensate to the condensate outlet at the same temperature as the steam entering the condenser exhaust branch, and it is in this feature that the regenerative condenser shows such a marked advance. The vacuum temperature is, of course, higher than the sea temperature, the difference being of the order of 16° F. to 25° F., depending upon the requirements to which the condenser is designed.

Assuming that a condenser has been designed to maintain a certain vacuum with a specified sea temperature when condensing a specified quantity of steam, Fig. 1 shows the vacua which will be maintained by such condenser with different sea temperatures, all other conditions remaining the same. For example, a condenser designed to give a vacuum of 28·65in. Hg. with the sea at 55° F. will give a vacuum of 27·5in. Hg. with the sea at 80° F., the quantities of circulating water and steam condensed remaining the same.

If the engine is developing a constant power, the quantity of steam entering the condenser will be reduced or increased as the vacuum rises or falls respectively and, as an approximation, and assuming a constant quantity of circulating water, the difference between the vacuum temperature and the temperature of the sea will vary in direct proportion to the quantity of steam condensed.

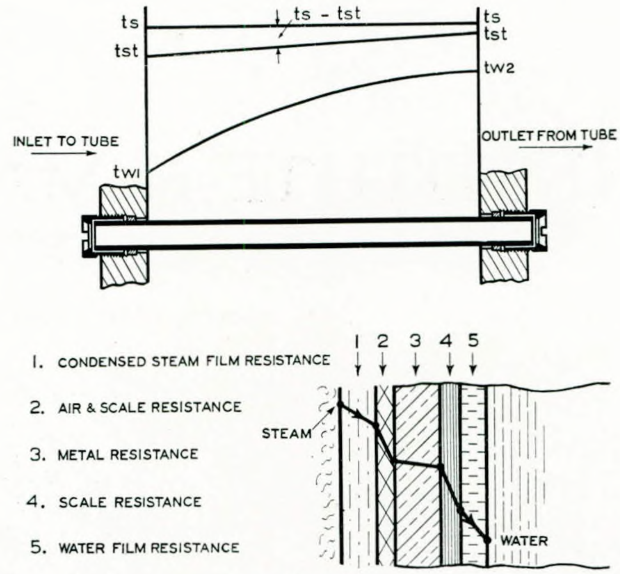


FIG. 2.—Temperature gradients in a surface condenser.

A condenser operates at its maximum efficiency when the difference between the temperature corresponding to the vacuum maintained in the condenser exhaust inlet and the temperature of the cooling water at the outlet from the condenser is a minimum, and when the temperature of the condensate leaving the condenser corresponds to the temperature of the steam entering the condenser.

The transmission of the heat in the steam to the water flowing through the condenser tubes is dependent upon the temperature gradient between the steam and the water and the resistance against which the heat flows. The temperature gradient and the resistances vary throughout the condenser. Taking any tube in a condenser, the temperature gradient can be shown diagrammatically as in Fig. 2; t_s represents the temperature of the steam at the inlet to the condenser, t_{st} is the temperature of the steam in the vicinity of the tube, the ordinate t_s-t_{st} representing the drop in pressure between the condenser exhaust branch and the tube, neglecting non-condensable gas pressure. The cooling water temperature rises from t_{w1} at the inlet to the tube to t_{w2} at the outlet from the tube. The values of t_s–t_s and t_{w2}–t_{w1} vary throughout a condenser. The value of t_{st} is affected considerably in the vicinity of the air outlet by the presence of non-condensable gases, which, for the sake of brevity, will be referred to hereinafter as air.

The lower part of Fig. 2 shows diagrammatically the various resistances to the heat flow. The heat passes through a film of condensate formed on the tube by the condensing steam, scale or dirt on the outside of the tube, the material of the tube, scale or dirt on the inside of the tube, the film of

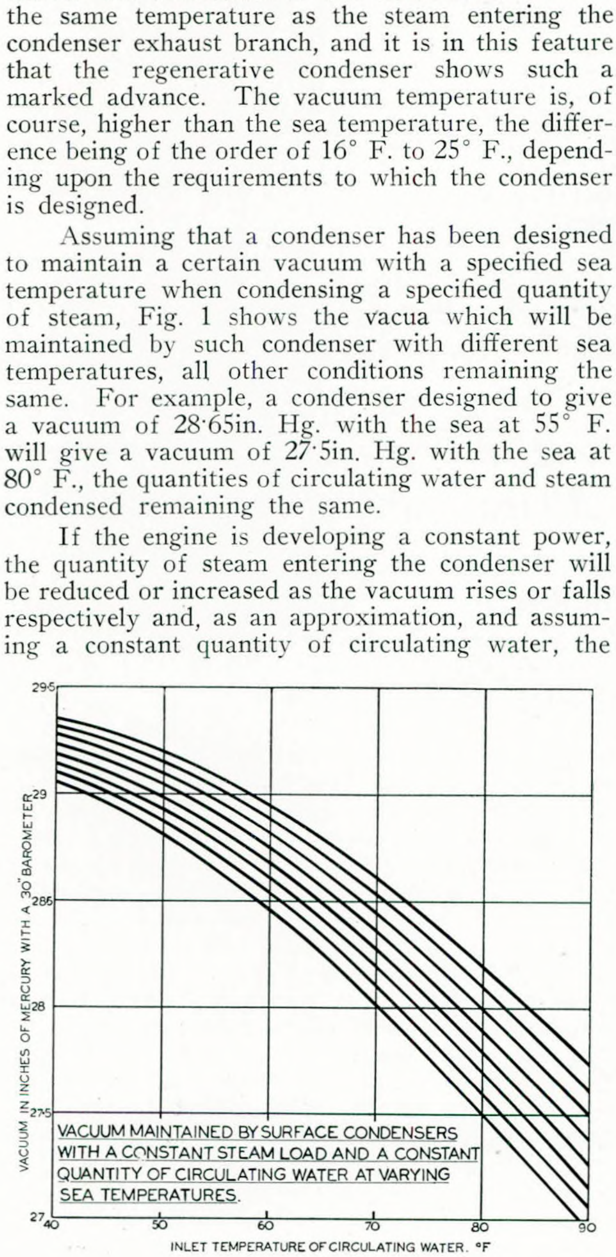


FIG. 1.—Variation in vacuum due to varying sea temperature.

cooling water in contact with the inside of the tube, and from this film to the main body of water passing through the tube. The average relative values of the various resistances in a reasonably clean condenser are shown in the diagram.

The laws correlating the various factors which affect these resistances have been carefully determined by numerous scientific experiments. To enable condensers to be designed to give desired performances, the known laws of heat flow require to be applied as determined by the careful examination of and comparison with the data obtained from actual condensers under widely varying conditions.

The performance of a condenser can be judged by comparing the overall heat transmission coefficient with the coefficient which would be obtained assuming the condenser tubes to be clean and the condenser to be entirely free from non-condensable gases with a uniform steam temperature throughout.

The overall heat transmission coefficient can be calculated from the formula

$$K = \frac{W_s \times (H-h)}{t_m \times S}$$

where W_s is the total weight of steam entering the condenser in pounds per hour.

H is the total heat per pound of steam at the exhaust inlet to the condenser in British thermal units.

h is the sensible heat of each pound of condensate in British thermal units.

t_m is the mean temperature difference between the steam and the cooling water in degrees Fahrenheit.

S is the total cooling surface of the exterior of the tubes of the condenser in square feet.

K is the overall heat transmission coefficient in British thermal units per hour per square foot per degree Fahrenheit difference in temperature.

The mean temperature difference is usually calculated from Grashof's formula assuming uniform steam temperature throughout the condenser.

$$t_m = \frac{t_{w2} - t_{w1}}{\log_e \left(\frac{t_s - t_{w1}}{t_s - t_{w2}}\right)}$$

where t_s is the saturation temperature of the steam entering the condenser.

t^{w1} is the inlet temperature of the cooling water.

t^{w2} is the outlet temperature of the cooling water.

The total resistance to heat flow from the steam to the water is the summation of the separate resistances and can be stated as follows:—

$$\frac{1}{K} = \frac{1}{K_s} + \frac{1}{K_m} + \frac{1}{K_w}$$

where K is the overall heat transmission coefficient.

K_s is the heat transmission coefficient from the steam to the metal of the tube.

K_m is the heat transmission coefficient across the metal of the tube wall.

K_w is the heat transmission coefficient from the metal of the tube to the water.

K_s varies throughout the condenser, dependent upon the presence of non-condensable gas and dirt on the outside of the tube.

K_w varies with the velocity and temperature of the water flowing through the tubes and is affected by the presence of dirt inside the tubes.

For a single tube swept by steam in motion, the resistance to heat flow from the steam to the metal is of the order of 1/2,500. Taking this value, the known resistance value for an Admiralty brass tube 18 L.S.G. (·048in.) thick and the water film resistance as determined by Eagle and Ferguson, the overall heat transmission rates for a clean tube are as shown in Fig. 3. It is impossible for each tube in a condenser to give a performance in accordance with Fig. 3, and it is necessary to allow a margin to cover the possible blanketing of many of the tubes with air, the presence of dirt and scale on the inside and outside of the tube surface and the pressure drop through the condenser.

Fig. 4 shows the values which can be expected of condensers for marine service with the tube surface in a reasonably clean condition and with efficient air withdrawal.

A condenser is a vessel through which a continuous flow is taking place due to a difference in pressure existing between the branch at which the steam enters the condenser and the outlet from which the air and the associated steam leave the condenser. The difference in pressure between these

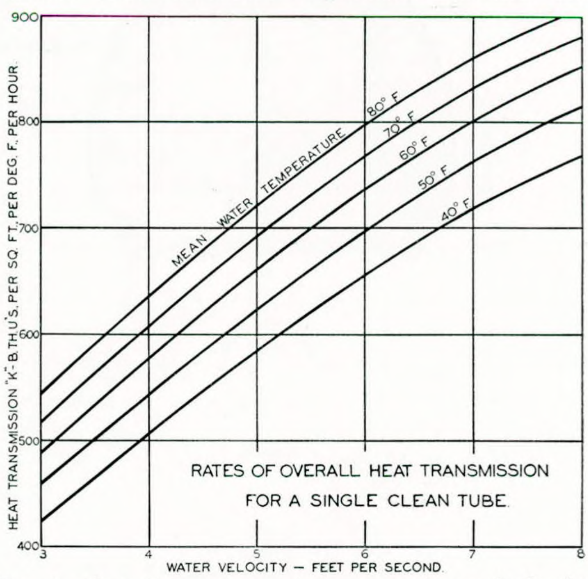


FIG. 3.—Overall heat transmission rates for a single clean condenser tube.

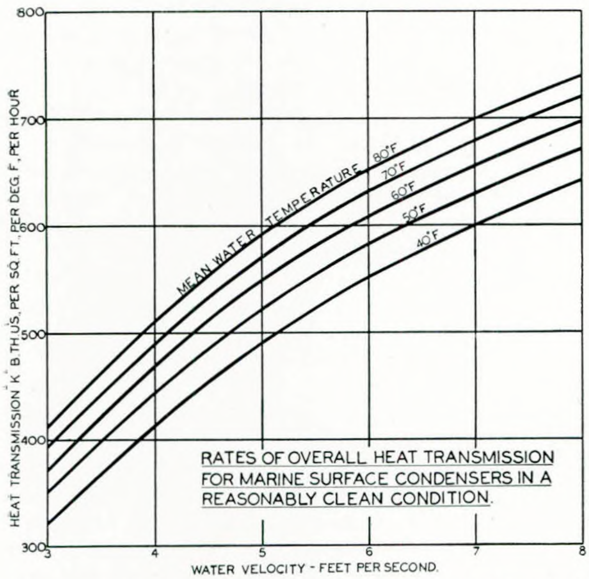


FIG. 4.—Marine surface condenser overall heat transmission rates.

two points should be reduced to a minimum by suitably arranging the tubes to obtain steam penetration of the tube banks with as little pressure loss as possible, and in this connection it is important that the banks of tubes with which the steam first comes into contact should be suitably arranged so that the kinetic energy of the steam entering the condenser

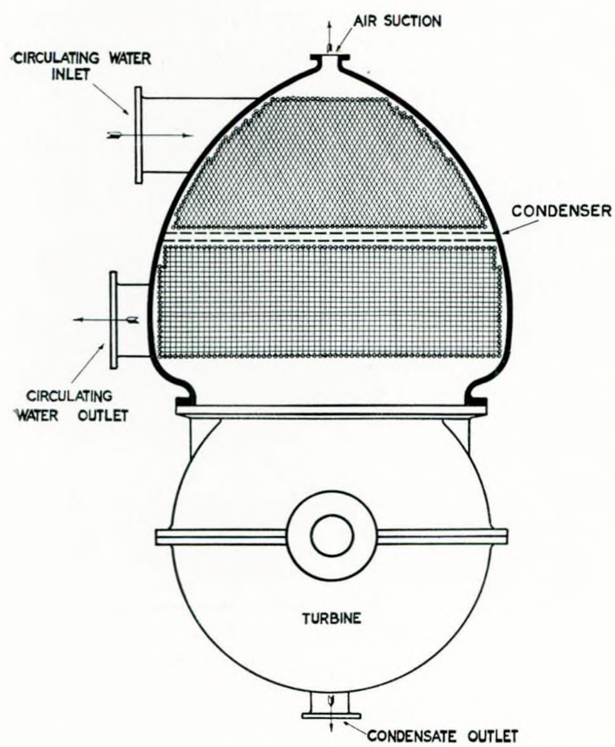


FIG. 5.—Fundamental idea of regenerative condenser.

is used efficiently in promoting steam flow through the tube banks.

The two chief improvements in present day condensers are the withdrawal of the air separately and remotely from the condensate and the regenerative feature previously mentioned. The condensate withdrawn from the older designs of condensers was frequently 10° F. to 15° F. below the temperature of the steam entering the condenser. A difference of 10° F. between these temperatures represents a fuel consumption of 1 per cent. and is a loss which the regenerative design has eliminated. In a bled steam heating installation, the fuel value of this condensate depression is less than the figure given, but it is still an appreciable saving.

Assuming it were possible to arrange the condenser above the turbine as shown in Fig. 5, it is

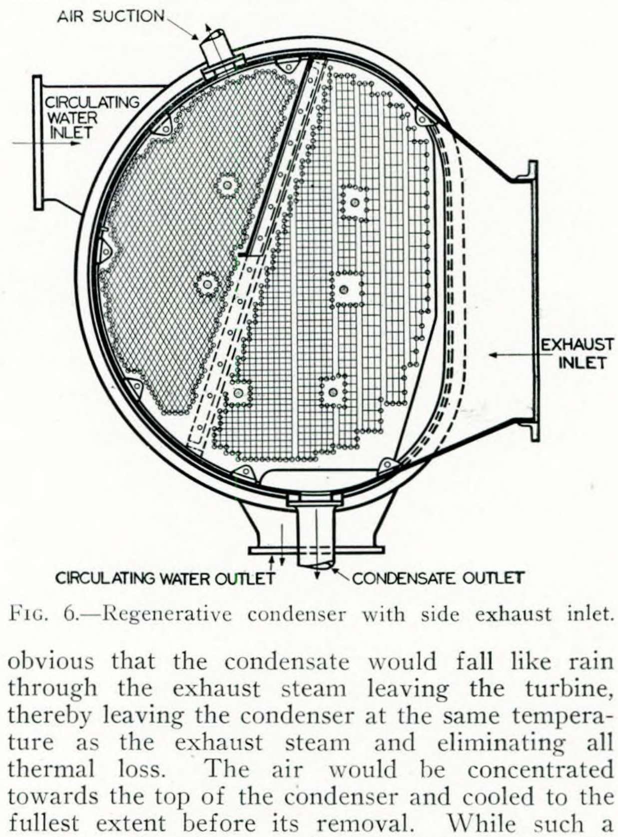


FIG. 6.—Regenerative condenser with side exhaust inlet.

obvious that the condensate would fall like rain through the exhaust steam leaving the turbine, thereby leaving the condenser at the same temperature as the exhaust steam and eliminating all thermal loss. The air would be concentrated towards the top of the condenser and cooled to the fullest extent before its removal. While such a condenser has been used in experimental work, practical difficulties prevent the adoption of the arrangement shown, but the essential principles involved have been embodied in present day designs.

Fig. 6 shows a condenser as fitted in steamers where the height of the engine room is limited, such as cross channel steamers. The condenser is arranged at the side of the turbine, the condensate must pass through the exhaust steam as it enters the condenser and the air is withdrawn at a point remote from the condensate outlet, the tube bank being provided with a division plate to form a gradually reducing cross section to concentrate and

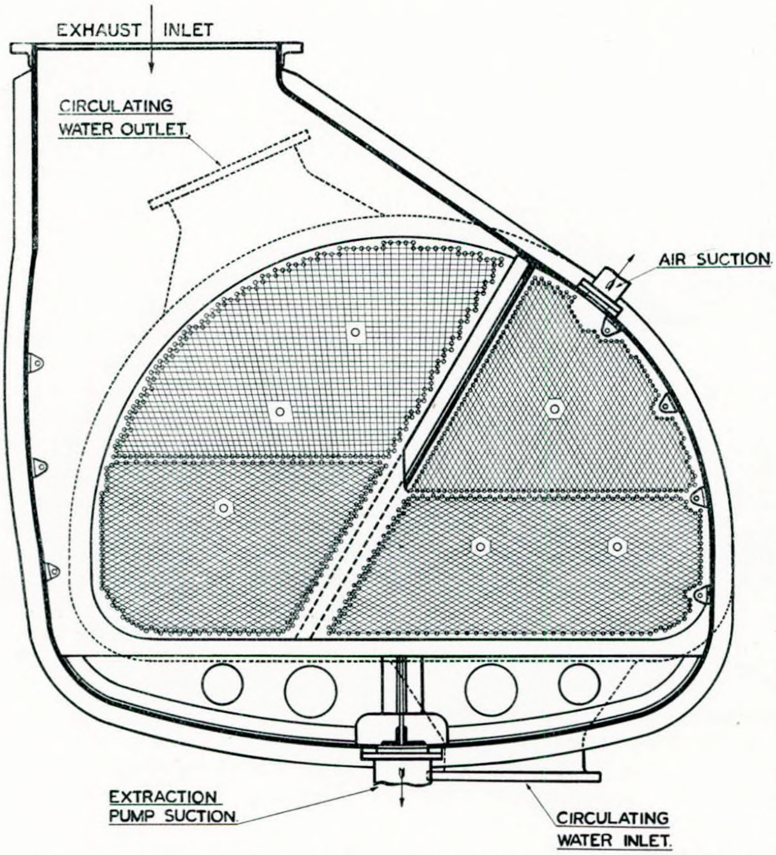


FIG. 7.—Regenerative condenser with side regenerative passage.

cool the air before it is removed. In a large number of ships, the condenser is arranged underneath the turbine, in which case the design of the condenser embodying the same principles is as shown in Fig. 7. A large passage is provided at one side of the condenser so that the steam leaving the turbine can flow direct to the bottom of the condenser. The pressure drop through such passage is negligible and the condensate, therefore, leaves the condenser at the temperature of the exhaust steam. The division plate provides a decreasing cross section which concentrates and cools the air efficiently before it leaves the condenser. This design is the practical embodiment of the condenser shown in Fig. 5. Where the height available is restricted so that the condenser is wide compared with its height, the principles explained above are incorporated in a layout as shown in Fig. 8, in which a central regenerative passage is provided with remote air suctions at each side of the condenser. It will be seen that, in a condenser of the type shown in Fig. 8, advantage can be taken of the central regenerative passage to arrange the end doors in two halves, thereby facilitating cleaning operations.

In the examples shown, it will be appreciated that the provision of a steam lane of generous proportions leading to the condensate outlet not only eliminates condensate temperature depression but ensures that a large number of tubes are swept by the incoming steam and are operating at a maximum heat transmission, thereby assisting the condenser as a whole to operate at a high efficiency.

Coincident with the development of the regenerative condenser, air pump design has also been materially improved. The presence of air in a condenser very seriously reduces the heat transmission on the steam side of the tubes due to the reduction in temperature caused by the presence of air, since the absolute pressure in the condenser is the sum of the water vapour pressure and the air pressure. The temperature corresponds to the water vapour pressure and is, therefore, less than the temperature corresponding to the vacuum. In addition, the condensation of steam at the surface of a tube concentrates the air round the tube and the heat flow is

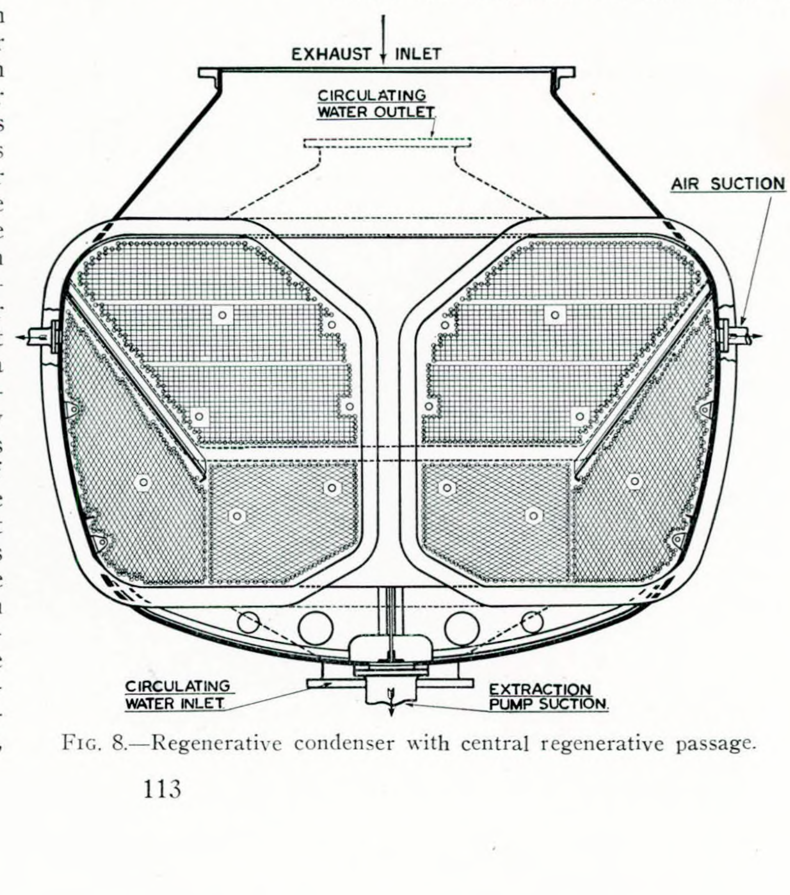


FIG. 8.—Regenerative condenser with central regenerative passage.

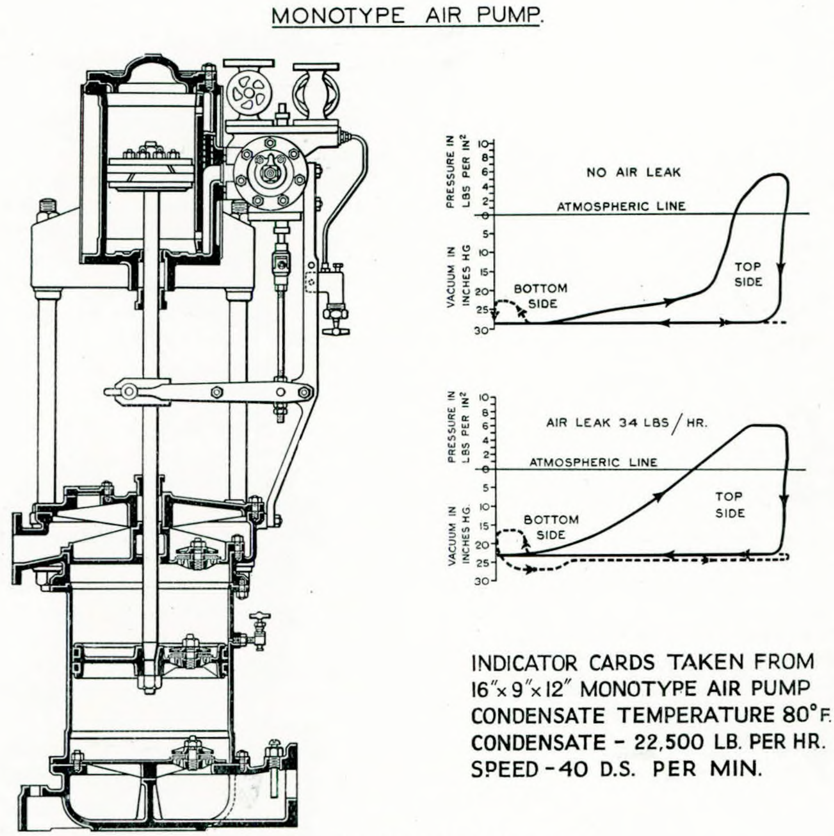


FIG. 9.—Monotype air pump.

impeded due to the steam having to diffuse through the air blanket to reach the tube surface. Present day condenser designs concentrate the air at the air outlet and localise the bad effect as much as possible.

Air pumps in marine practice to-day are usually of the reciprocating type, steam ejectors or a combination of a steam ejector operating in conjunction with a reciprocating pump, an arrangement which is usually adopted for small power installations.

Fig 9 shows the typical three-valve type of air pump which withdraws air and condensate together. The performance of this type of pump is shown in Fig. 10. This pump is quite unsuitable for operating with a regenerative condenser, because the air pressure at the condensate outlet is negligible and its air withdrawing capacity under such conditions is zero. The air withdrawing capacity is dependent on the volume swept by the bucket, less the volume of the condensate handled, and the pressure of the air which is dependent on the depression of the condensate temperature below the vacuum temperature.

As will be seen from Fig. 10, the pump cannot handle air at pressures below approximately $\frac{5}{8}$in. Hg. This is due to the presence of air in the water left in the clearance volume at the end of the stroke, the water in the suction pipe and the friction loss through the suction pipe and foot valves into the pump barrel.

At low air pressures, the volume of a given weight of air is extremely large and the volumetric efficiency of the pump is very low. Fig. 10 shows an analysis of the results of a test on a 16in. by 9in. by 12in. Monotype air pump, the volumetric efficiency being calculated on the basis of the air pressure existing in the condenser and the volume swept by the pump bucket, less the volume of the condensate. Curves A and B show clearly the limitations of the ordinary three-valve type of pump. As a matter of interest, curve C has been added to show the inherent pressure loss associated with this type of pump. This represents a direct loss in vacuum quite apart from the impaired heat transmission due to the presence of air in the condenser.

The Dual air pump, shown in Fig. 11, was designed to reduce the losses associated with the ordinary three-valve type of pump by removing the air separately

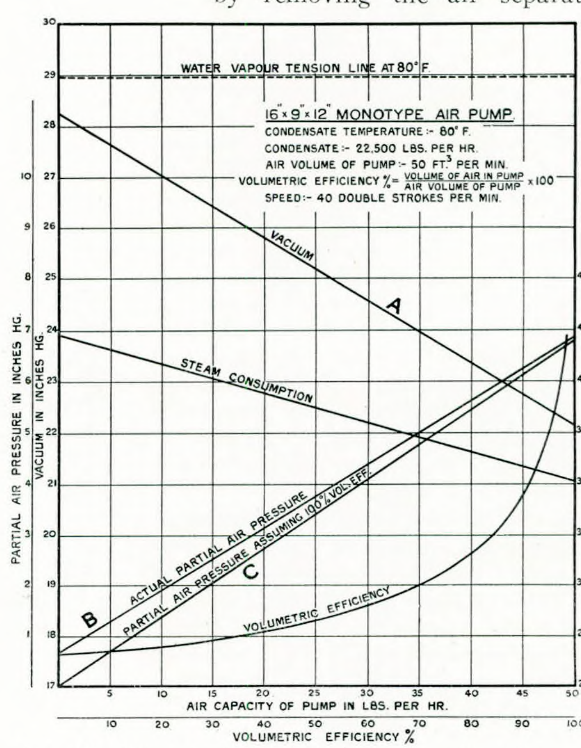


FIG. 10.—Characteristic performance of Monotype air pump.

from the condensate and cooling the air below the temperature of the condensate. Barrel A handled condensate direct from the condenser, while barrel B withdrew air only from the same branch on the condenser in the original arrangements, and subsequently from a separate air outlet from an air cooling compartment in the condenser. The barrel B was provided with a separate sealing water circuit, which included a sea water circulated cooler wherein the sealing water was cooled to a much lower temperature than the condensate temperature, the air being cooled subsequently by the sealing water to approximately the same temperature. This arrangement shifted the datum temperature on which the air pressure was based, but the extreme limit was, of course, the temperature of the sea. Fig. 12 shows the performance of a Dual air pump with the injection water cooler circulated with water at 60° F. It will be seen that at any reasonable air leakage there is a considerable air pressure in the condenser.

While the reciprocating air pump is extremely inefficient when handling air at low pressure, its efficiency is very good when the air is at a pressure of 2in. Hg. or more. It has also the advantage of adjustability and reserve capacity, since its speed and, therefore, its air handling capacity can be increased to meet adverse conditions when they arise. In the ordinary three-valve air pump, the down stroke is used solely to transfer air and water from the underside of the bucket to the top side. The indicator cards in Fig. 9 show clearly that there is practically no resistance to be overcome during the down stroke. In the case of the Dual pump, the discharge from barrel B was led into the top of barrel A, since this arrangement preserved a closed circuit for the cooled sealing water, thereby avoiding unnecessary heat loss and reducing the pumping load. Barrel B is a single-acting pump, but barrel A is double-acting since it draws water from the condenser during the up stroke of the bucket and air from the discharge of barrel B during the down stroke of the bucket, the air and condensate being discharged to the feed tank on the succeeding up stroke.

To obtain the best results from a reciprocating air pump, it should be arranged to deal with air at a pressure of, say, 2in. Hg. or more and should

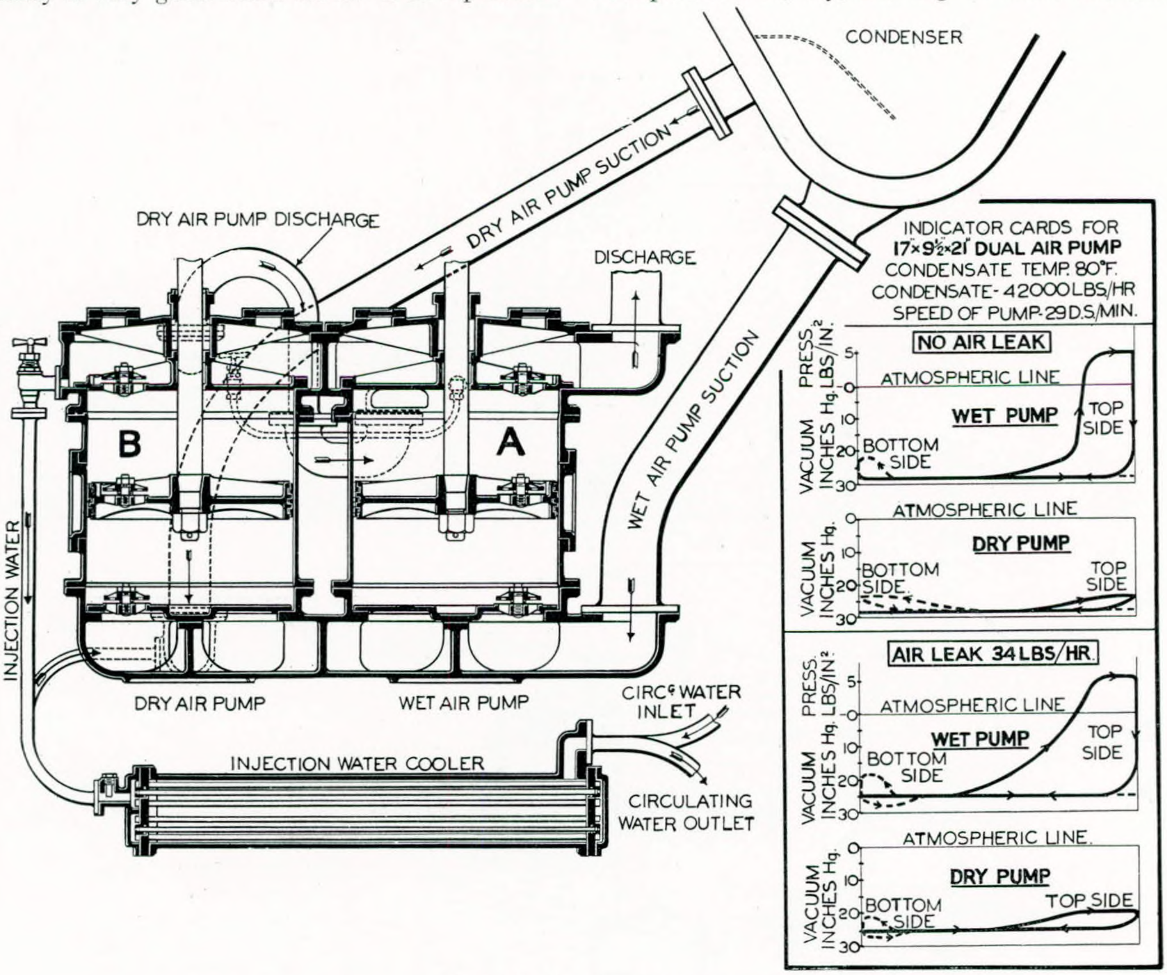


FIG. 11.—Dual air pump.

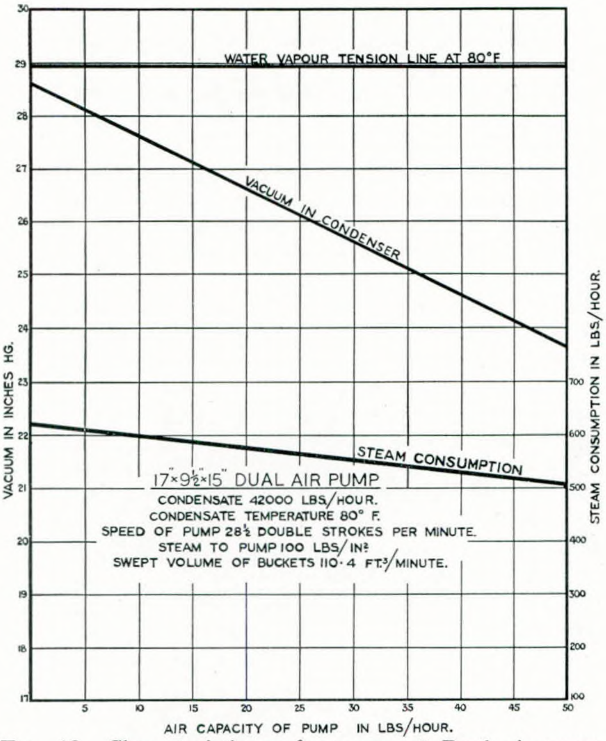


Fig. 12.—Characteristic performance of Dual air pump.

be operated double-acting so that the down stroke as well as the up stroke is utilised. The modern reciprocating air pump is usually provided with a steam-operated air ejector which withdraws the air from the condenser and compresses it to 2in. Hg. or more, the reciprocating pump drawing the air after compression by the steam ejector and discharging it to the atmosphere.

Modern steam installations of any considerable power are now almost invariably fitted with steam-operated air ejectors, usually of the two or three-stage type. In a steam ejector, steam is expanded from boiler pressure and temperature in a convergent-divergent nozzle. During the expansion, the steam acquires a high velocity of the order of 4,000ft. per second at the exit from the nozzle and produces at this point a vacuum which is slightly higher than exists in the condenser by the pressure drop necessary to cause the water vapour and air to flow from the condenser to the air ejector. This pressure drop is much smaller than in the case of a reciprocating air pump, since the flow of the air is not impeded by the presence of solid water; there is no clearance volume or valve loss to contend with, and a continuous flow is maintained. Further, the actual volume of the air to be handled is not the limiting factor as in the case

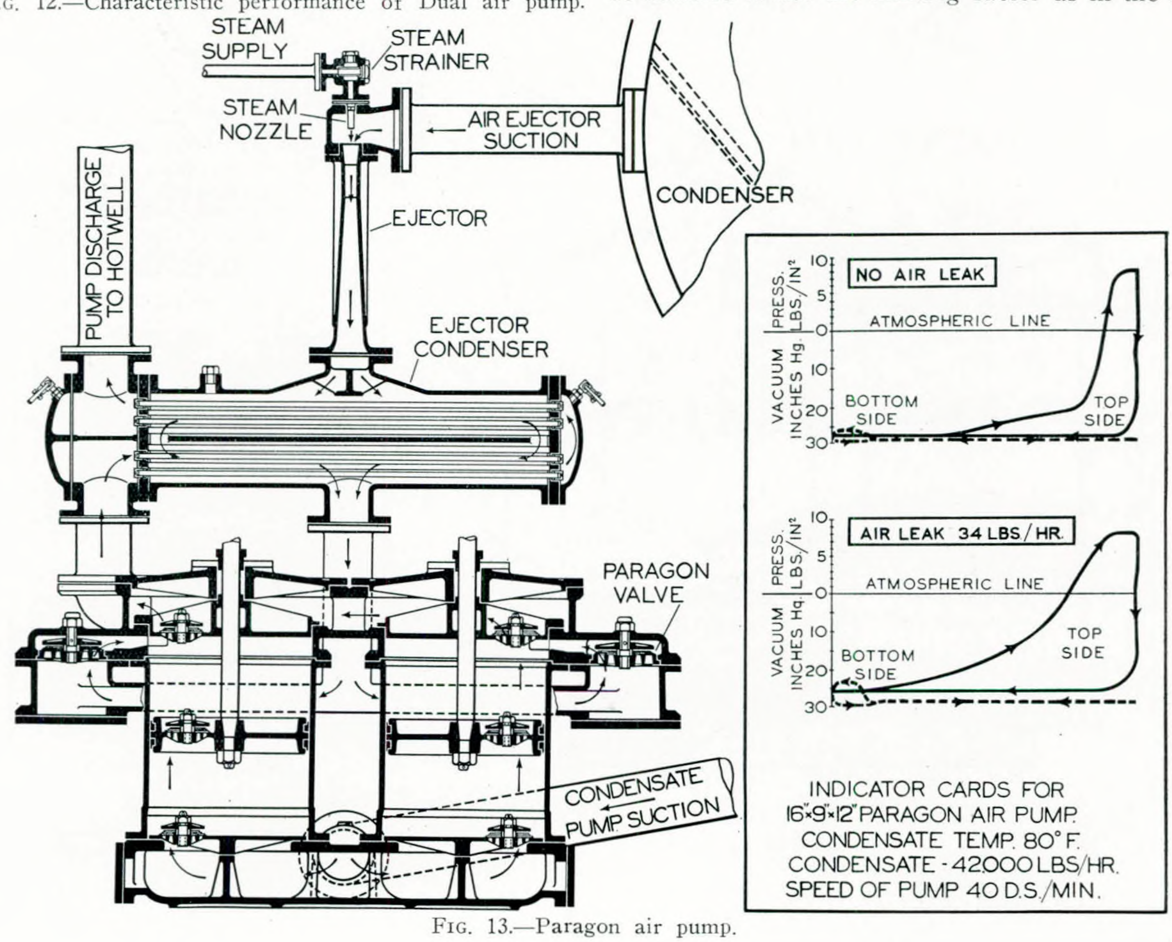


Fig. 13.—Paragon air pump.

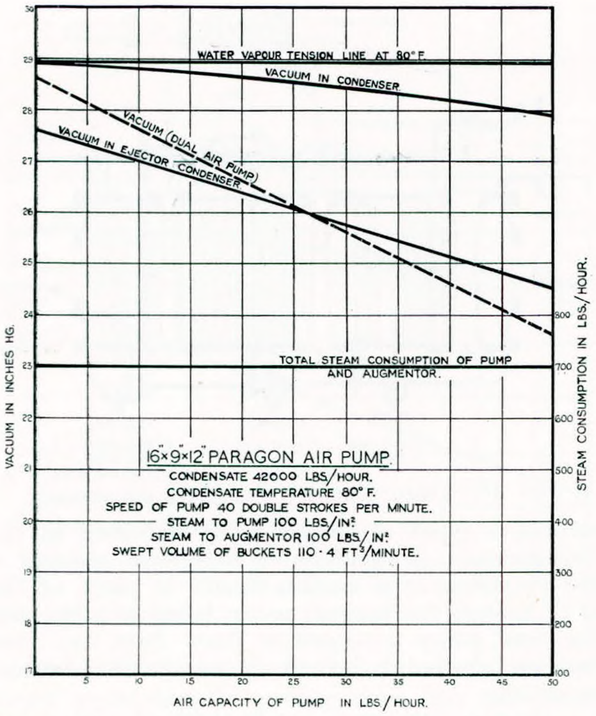


FIG. 14.—Characteristic performance of Paragon air pump.

of a reciprocating air pump, the limiting factor being the total weight of the air and water vapour. The steam ejector has the inherent advantage, as compared with the reciprocating pump, of being able to deal with large volumes so that the air pressure is low and the pressure drop between the condenser and the air ejector is small.

The Paragon air pump is the most widely used modern pump, in which the advantages of the steam ejector are combined with the advantages of the reciprocating air pump. Fig. 13 shows a section through a Paragon pump which is, to a large extent, self-explanatory. The two barrels operate on exactly the same cycle. During the up stroke, condensate is drawn through the foot valves. The steam ejector withdraws the air from a separate air suction and discharges the air and operating steam into a condenser in which the ejector steam is condensed. The air passes from the condenser through the Paragon valve into the pump above the bucket, while the bucket is making its down stroke. At the end of the down stroke, the condensate is transferred to the top side of the bucket and the air and condensate are discharged through the head valves and the ejector condenser to the feed tank during the succeeding up stroke. The heat of the ejector steam is thereby conserved in the feed water.

The indicator diagrams taken from a Paragon air pump (*see* Fig. 13) show clearly the pressures prevailing on both sides of the bucket, during the up and down strokes. Fig. 14 shows the performance of a Paragon pump with twin barrels, its superiority over the dual air pump being shown clearly by a comparison with the dotted curve which shows the vacuum maintained by a dual air pump of the same size.

The marked advance made in reciprocating air pump design is illustrated by the following comparative data. Assuming that the buckets of each of the three types of pumps sweep through 110 cubic feet per minute and withdraw 42,000lb. of condensate per hour at a temperature of 80° F. from a condenser into which there is an air leakage of 10lb. per hour, the comparative figures are:—

	Monotype.	Dual.	Paragon.
Vacuum maintained in condenser in inches of Mercury (Bar. 30in. Hg.)	27·3	27·65	28·83
Volume of air withdrawn from condenser in cubic feet per minute	40	52	500

Another form of comparison can be made by assuming each of the three types to maintain a vacuum of 28in. Hg. (Bar. 30in. Hg.) when withdrawing 42,000lb. of condensate per hour at 80° F. from a condenser into which there is an air leakage of 47·5lb. per hour, in which case the volume to be swept by the buckets of each type is as follows:—

	Monotype.	Dual.	Paragon.
Volume swept by the buckets in cubic feet per minute	1,020	663	110

Many ships are now fitted with reciprocating engines and exhaust steam turbines where the highest possible vacuum is required. The independent air pump of the Paragon type has the advantages of flexibility and independent operation which enables the vacuum to be maintained when the main engines are not running. In many cases, however, the air pump is of the Edwards type or a three-valve type driven by levers from the main engines, and in such cases a steam ejector arrangement is

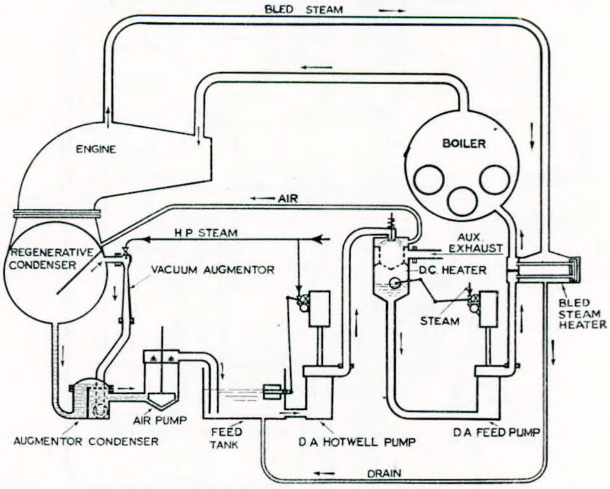


FIG. 15.—Feed system for reciprocating engines with engine-driven air pump.

to the boiler. To ensure satisfactory operation of the feed pump, the condenser must be arranged several feet above the pump, a requirement which is impracticable as a rule where the main propelling machinery is concerned, although the arrangement shown is used for the harbour operation of an auxiliary system on board ship. The fundamental circuit of the closed-feed system now widely used for any considerable power, is as shown in Fig. 22, the duties of condensate extraction and boiler feeding being performed by separate centrifugal pumps because the low head usually available between the condenser and the condensate pump necessitates the condensate pump being operated at a comparatively low speed, while the discharge pressure of the feed pump requires the latter to be operated at a high speed in order to obtain a satisfactory design. Further, it is advantageous to arrange air ejectors, drain coolers and feed heaters in the pipe line

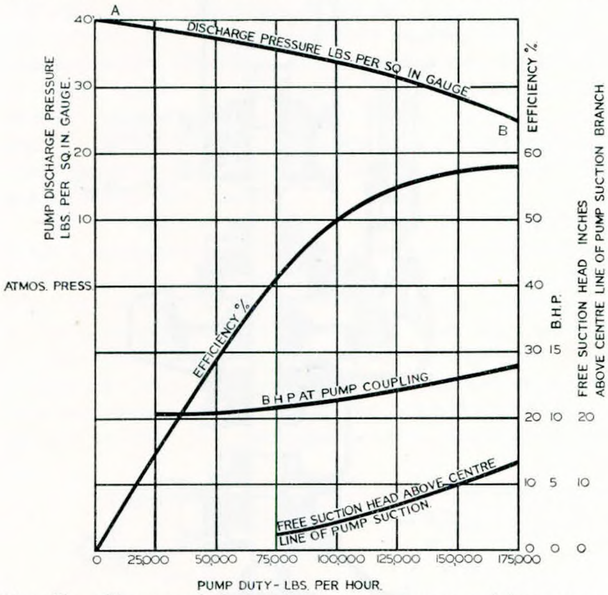


FIG. 20.—Characteristic performance of motor driven two-stage condensate pump.

between the condensate and feed pumps, since they are then subject only to the low discharge pressure of the condensate pump and can be made with comparatively light scantlings.

The fundamental circuit shown in Fig. 22 requires amplification for practical use on board ship. First, all marine installations have auxiliary plant using steam, and provision must be made to return the resultant condensate to the boilers; second, no provision is made to admit water into the system to make up for the losses of feed water which are inevitable, due to leakage from the glands of turbines, pumps, etc., and third, because, in the case of water-tube boilers, the weight of water contained in the boiler varies considerably at different steaming rates due to the fact that the volume of steam present in the water increases very rapidly as the boiler load increases and is a considerable volume

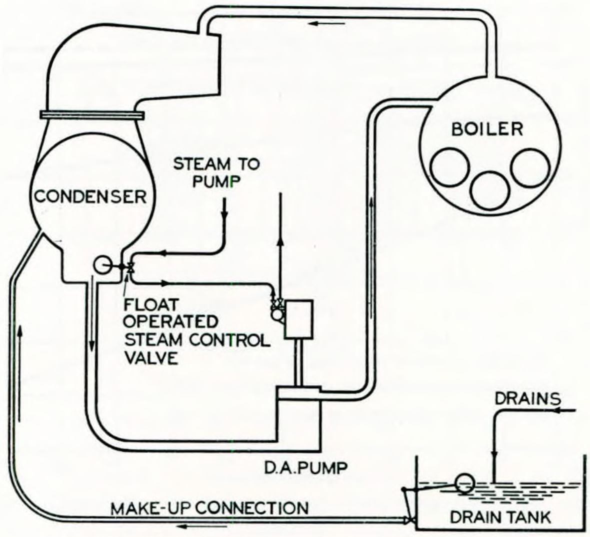


FIG. 21.—Simplest form of closed feed system.

compared with the water volume at the maximum rate of evaporation.

These facts necessitate the provision and control of three water levels—the water level in the boiler, the water level in the condenser and the water level in the feed tank. The control of these three levels differs with the type of boiler. In the case of a tank boiler, such as a Scotch boiler, the water volume is large and a considerable quantity of water can be discharged into the boiler for a comparatively small rise in the water level. Since the losses from the system are small, a closed circuit of the type shown in Fig. 22 with a Scotch boiler could be operated for a considerable time before any appreciable fall in the boiler water level would occur, and no direct control of the level is, therefore necessary.

Fig. 23 shows a closed-feed system for boilers of the tank type.

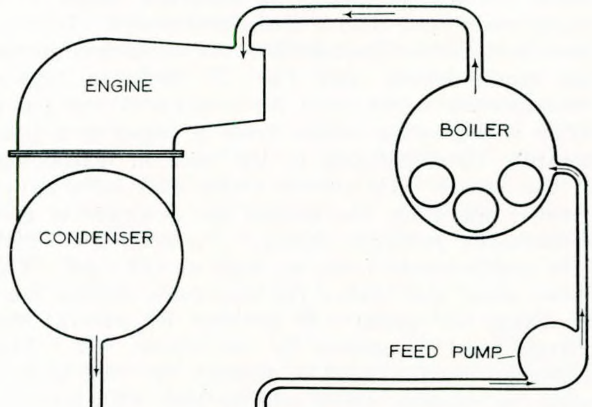


FIG. 22.—Fundamental circuit of modern closed feed systems.

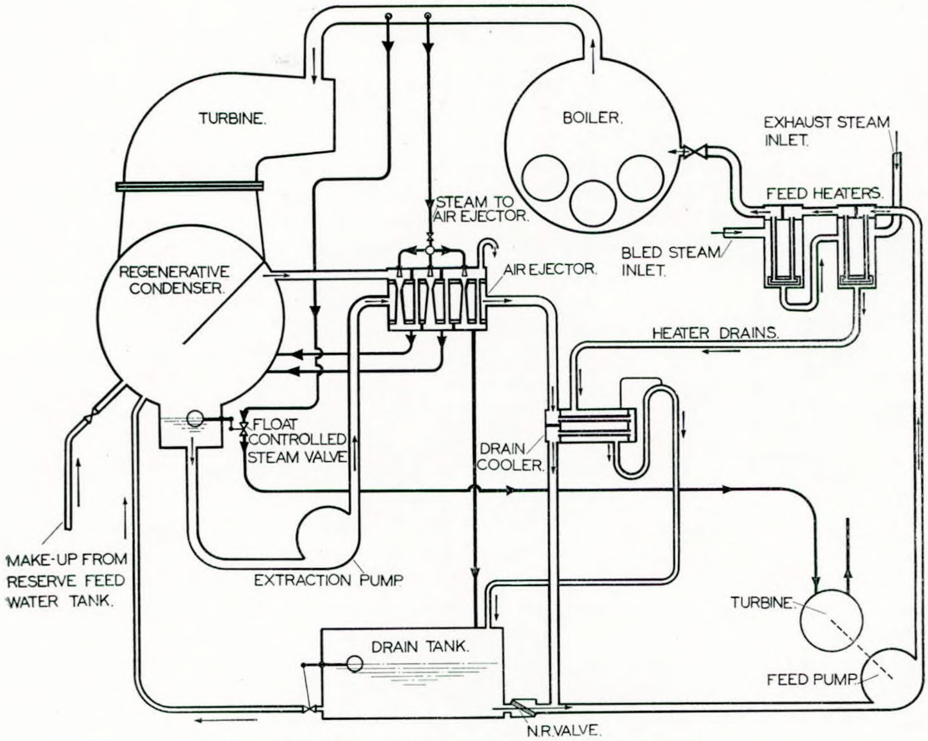


FIG. 23.—Closed feed system for tank boilers.

open. When two or more boilers are in service, the check valves on the respective boilers are hand-regulated to distribute the water between the boilers. Frequent adjustment of the check valves is not necessary due to the large quantity of water required to cause a small change in water level.

A feature of note in this system is that the feed pump is required to develop only sufficient pressure to overcome the boiler pressure and the static and frictional resistances to feed flow, the feed pump steam consumption being a minimum for the work to be done.

Fig. 24 shows a closed feed system for water-tube boilers in which the water content is relatively small. A great advantage obtained with the system is that all units are entirely automatic at all powers. The controlling water level in the system is the water level in the boiler which is maintained between predetermined levels, the

The drainage from the auxiliary exhaust and feed heating system is led to the feed tank which is provided with a float-operated valve which passes the drainage water to the condenser when the water level rises in the feed tank. As the water level in the condenser rises and falls, it actuates a float which controls the steam supply to the feed pump, the steam supply being opened as the water level in the condenser rises, and closed as the water level in the condenser falls. The whole of the steam condensed in both the main and the auxiliary systems is passed through the condenser wherein it is de-ærated before it is passed to the boilers. Make-up feed passes from the reserve feed tank to the condenser under control of a manually-operated valve, which is opened as necessary to maintain the desired water level in the boilers. Where one boiler only is in use, the feed check valve is left wide

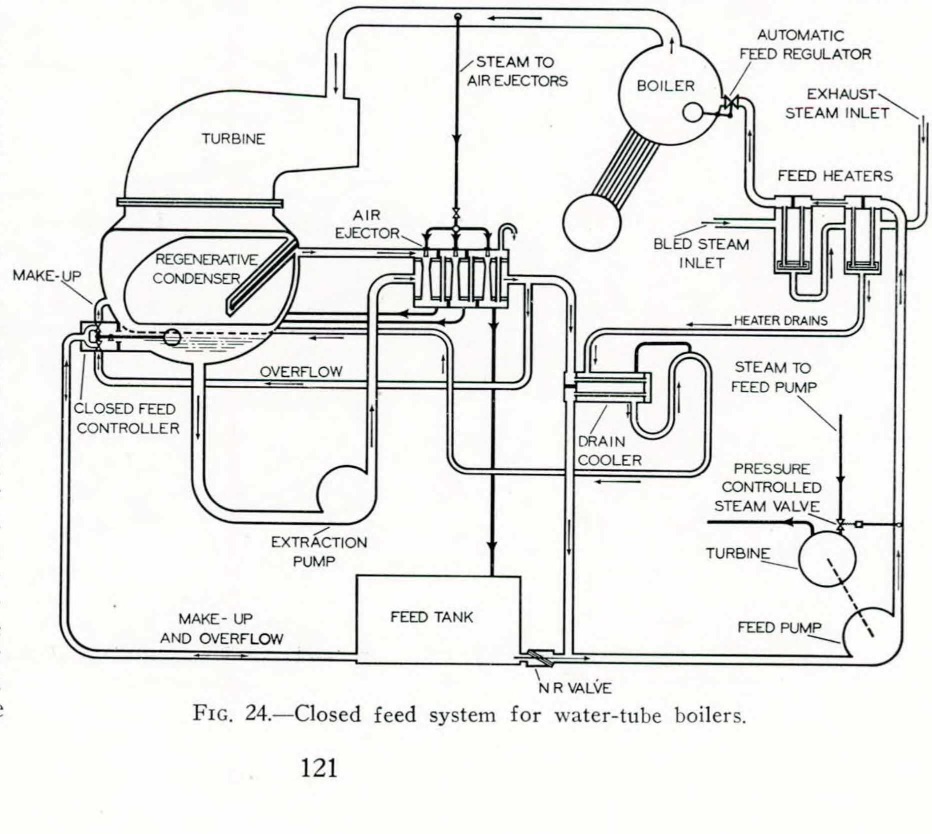


FIG. 24.—Closed feed system for water-tube boilers.

level at full power being two to three inches lower than the level at no evaporation. Variations between the rate at which water is fed into the boiler and steam is condensed in the condenser, alter the water level in the base of the condenser. The water level in the condenser is automatically regulated by a closed feed controller (*see* Fig. 25) which

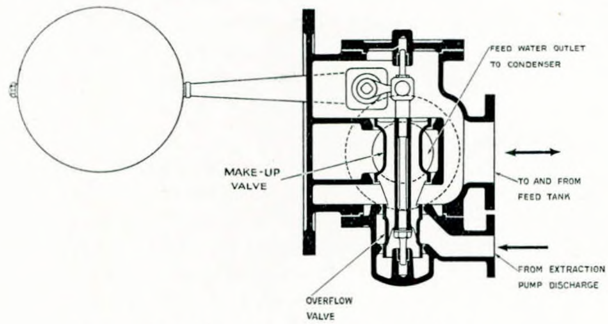


FIG. 25.—Closed feed controller.

comprises two valves actuated by a common float, one valve being an overflow valve and the other a make-up valve. If the water level rises, the overflow valve opens and water is discharged to the feed tank, while a fall of water level causes the make-up valve to open and water is drawn into the condenser from the feed tank. Under steady running conditions, the make-up valve is slightly open, just sufficient to pass the quantity of water necessary to make good losses from the system. The water level in the feed tank must be kept under observation and water drawn from a reserve tank as necessary to maintain the feed tank about two-thirds full under normal running conditions. In some ships, evaporators using bled steam are fitted in the feed system and are operated as necessary to maintain the feed tank level.

Fig. 26 shows a characteristic diagram of a closed feed system as shown in Fig. 24. The total travel of the closed feed controller float is divided into three zones. The upper zone covers the gradual opening of the overflow valve from the shut position to the full open position. When the float is in the middle zone, both the overflow and make-up valves are closed while the lower zone covers the opening of the make-up valve from the closed position to the full open position. The line AB shows the lowest level at which the condensate pump will operate without cavitation and is the free suction head of the pump. The lines CD and EF show the controlled suction head with the overflow and make-up valves in operation respectively, the difference between these lines and the line AB representing the margin over the free suction head to prevent cavitation in the pump. At the maximum output of the pump, the controlled head approaches the free suction head and the margin is FB. This condition is reached very occasionally—only, as, for example, when a sudden stoppage of the main engines occurs.

The arrangement of the condensate pump in relation to the closed feed controller and the condenser requires careful consideration in each case to ensure the satisfactory operation of the condensate pump when the ship heels when turning or rolling, or due to a beam wind.

The effect of heel can be eliminated if the condenser, closed feed controller float and condensate pump are all arranged on the same fore and aft line. This is not always practicable and it is usually necessary to compromise between the immediate effect of a heel and the effect of a continuous heel, the latter being the more important. In Fig. 27 the observer is assumed to be looking aft on a condenser with the condensate pump and closed feed controller float arranged on a different fore and aft line. On even keel, the controlled suction head for steady running is MR, while the minimum controlled suction head, if conditions

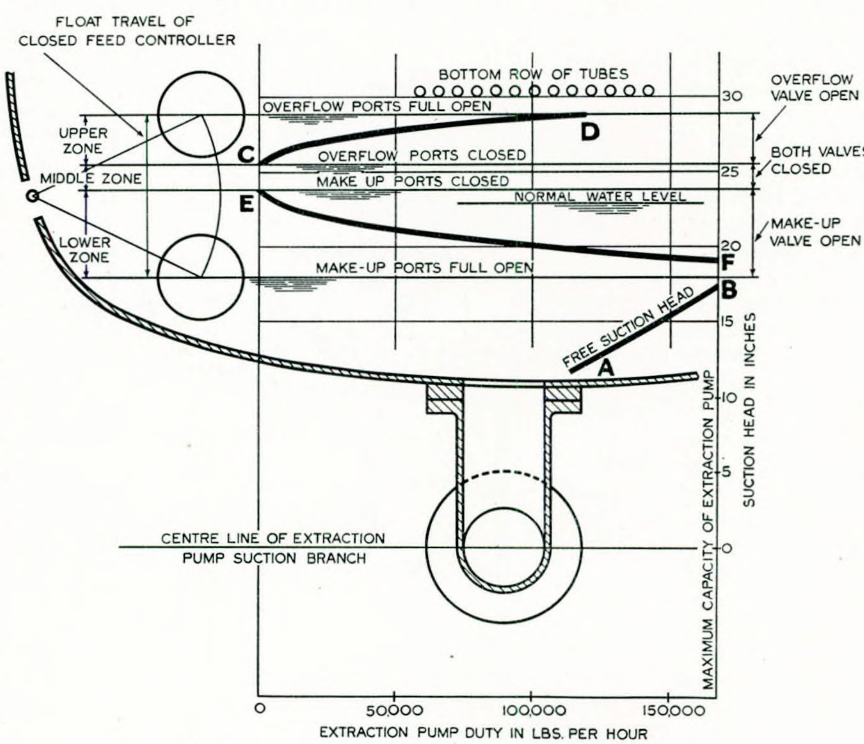


FIG. 26.—Characteristic diagram for closed feed controller.

necessitate the make-up valve opening wide, is MQ. If a heel occurs, the line TN represents the water level which obtains immediately and the suction head on the condensate pump is reduced to MN. When this occurs, the closed feed controller opens the make-up valve wide and water is drawn from the feed tank into the condenser until, if the heel is continuous, the suction head becomes MP. In the extreme and unlikely condition of the condensate pump requiring water at maximum capacity during a continuous heel, the suction head available would be MW. The head MN which occurs immediately a heel takes place is determined by the position of the condensate pump relative to the condenser, while the suction heads prevailing under all other conditions are determined by the position of the closed feed controller float in relation to the condensate pump.

A study of the diagram will show that it is preferable to arrange the float and the condensate pump on the same fore and aft centre line and as near as possible to the fore and aft centre line of the condenser.

Fig. 24 shows a closed feed system with one boiler, one condenser and one feed tank. In most ships there are several boilers, and two or more condensers and feed tanks. The boilers take water from the discharge of the feed pump or pumps as

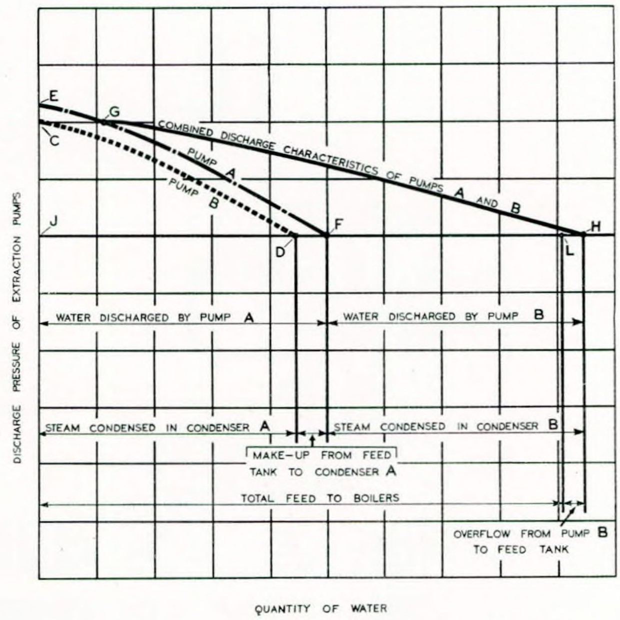


Fig. 28.—Diagram illustrating parallel operation of closed feed system units.

they require it, and can be regarded as acting as one boiler. With two or more condensers there are two or more condensate extraction pumps and, in general, these pumps have to discharge into a common pipe line connected to the suction of one feed pump or the common suction of a number of feed pumps. Each condenser provided with its own closed feed controller. It is necessary for the condensate extraction pumps to discharge in parallel and, to avoid surging, each pump must have a stable pressure capacity characteristic with the discharge pressure falling continuously from no load to full load when running at a constant speed. The various pumps should have the same characteristic and should be operated at the same speed, so that they share the required feed quantity between them. The condensate pumps may, however, be arranged in separate engine rooms with different friction losses between their discharge branches and the common point of discharge, or they may be running at different

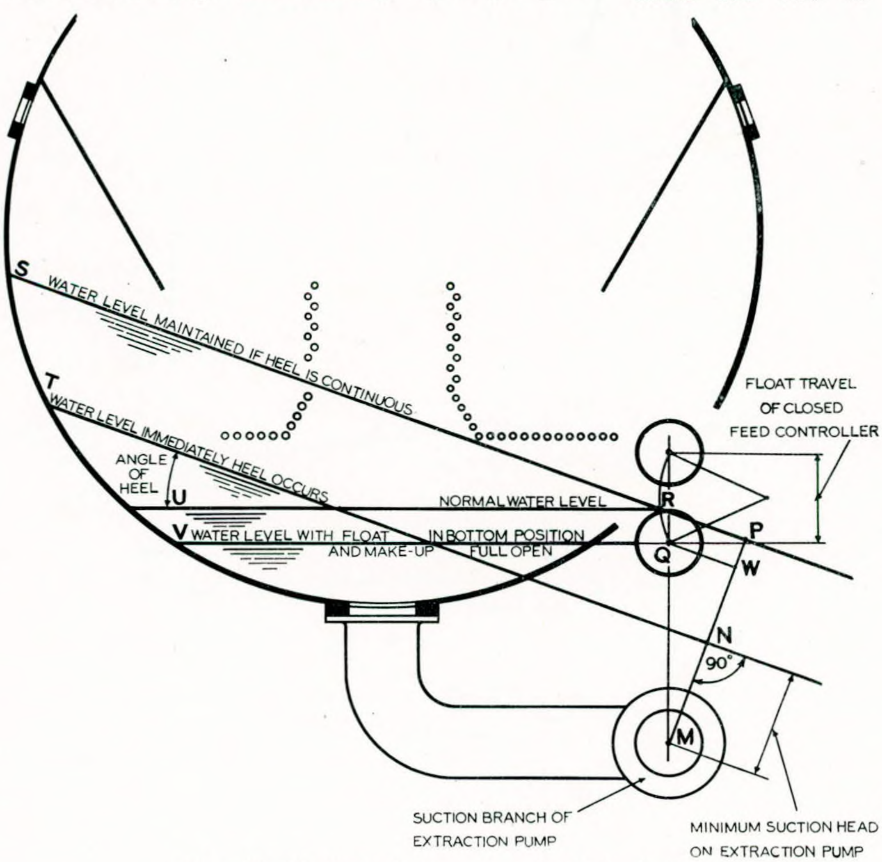


Fig. 27.—Heeling diagram for closed feed system.

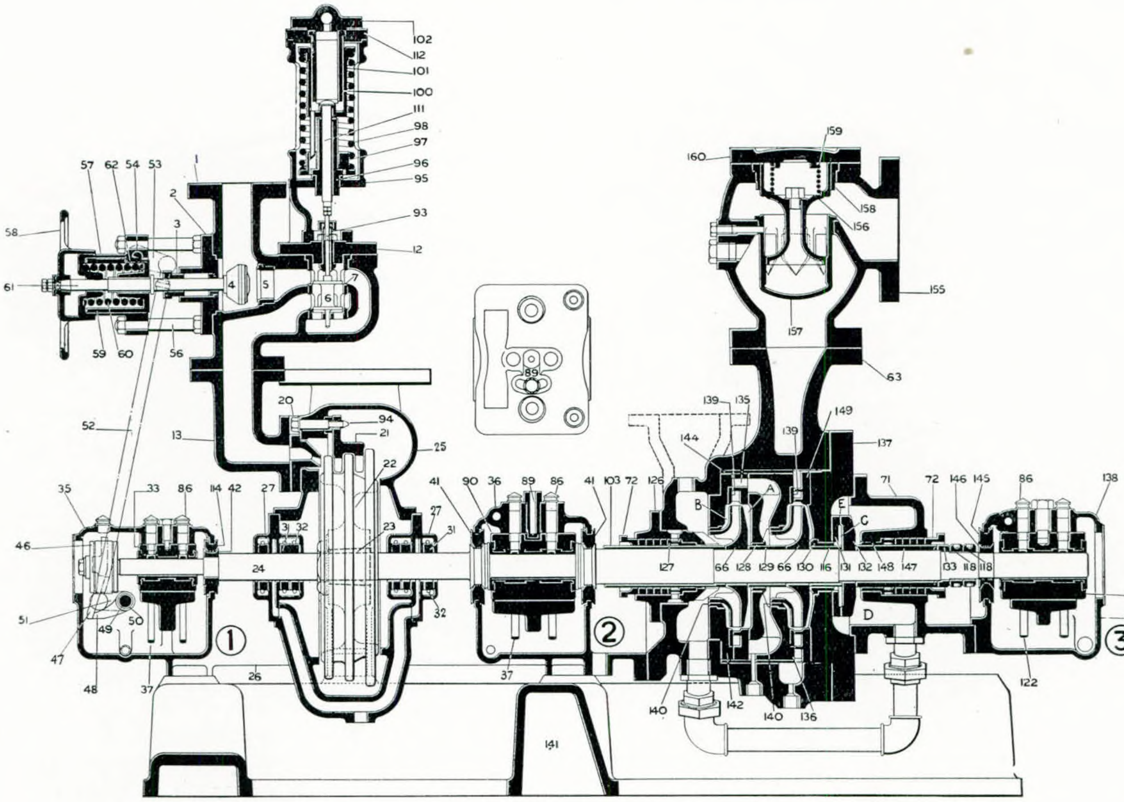


FIG. 29.—Two-stage turbo-feed pump.

speeds due to maladjustment; further, the vacua or the quantities of steam condensed in the respective condensers may be widely different. It is of interest, therefore, to examine how the closed feed system automatically adjusts itself should such adverse conditions occur. For simplicity, two condensers with two pumps only will be considered, because three or more units operate in parallel in exactly the same way.

Referring to Fig. 28, assume the two pumps A and B are developing different discharge characteristics EF and CD at the common point of discharge due to any of the reasons mentioned above, then the combined discharge charactertistic of the two pumps can be shown by the line EGH. The discharge pressure of both pumps will be the same and will fall until pump B is capable of discharging at a capacity equal to the rate at which steam is being condensed in condenser B. This will cause the water level in condenser B to rise until the overflow valve opens and passes a small quantity of water LH to the feed tank. At the same time, the water level in condenser A falls and the make-up valve opens as necessary to pass a quantity of water equal to DF. When the two pumps are developing the same characteristic and the duties of the two condensers are alike, no overflow occurs and the make-up is divided between the two condensers. When conditions vary, however, the closed feed controllers automatically adjust the water flow to maintain the feed to the boilers. Where there are two or more feed tanks, the cross connections on the feed tanks should be opened and the feed tanks operated as a common tank.

For the present, water-tube boilers are being

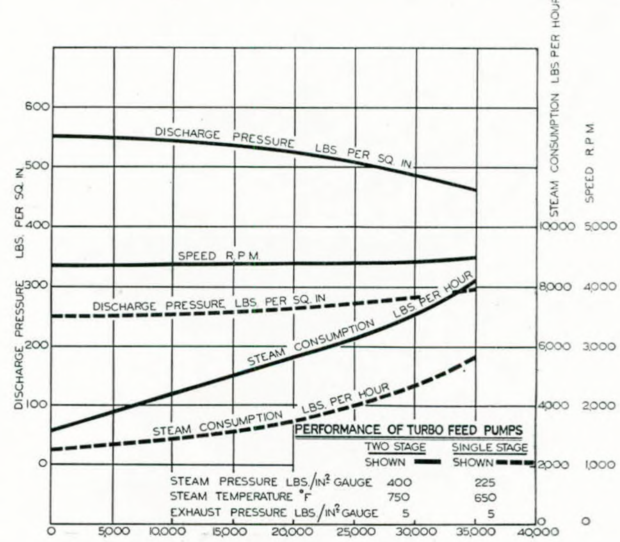


FIG. 30.—Characteristic performance of turbo feed pumps.

operated at pressures in the neighbourhood of 400lb. per sq. in., and Fig. 29 shows a section of a turbo-driven feed pump of the two-stage type required to meet such boiler pressures. A typical performance of such pump is shown in Fig. 30. The discharge pressure capacity characteristic, it will be noted, is stable and falls continuously from no load

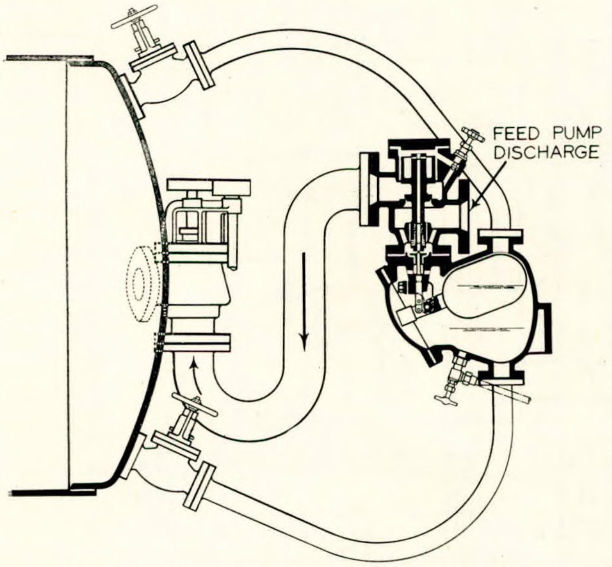


FIG. 31.—"Robot" boiler feed water regulator.

to full load, so as to ensure stable and satisfactory operation, free from surging, when operating solo or in parallel with other feed pumps. The exhaust steam from the turbine is led to a feed heater at about atmospheric pressure. Fig. 30 also shows the performance of a single-stage feed pump as operated in a feed system of the type shown in Fig. 23.

The remaining important unit in the feed system is the boiler feed regulator. The response of centrifugal condensate and feed pumps to a demand for water is immediate, and the pumps can change from no load to full load practically instantaneously. The boiler feed regulators control the demand for water, and it is very necessary that the control should be smooth and such that the supply of feed water to the boilers is continuous and not intermittent, since an intermittent supply would not only cause unnecessary variations in the operation of the condensate and feed pumps, but would interfere with the steady steaming of the boilers.

The Robot feed regulator shown in Fig. 31 is the latest development in regulators and meets all the requirements of a closed feed system. The regulator comprises a float which actuates a pilot valve in accordance with the rise and fall of the water level in the boiler. The pilot valve acts as a servo-motor with the feed water as the hydraulic medium controlling the movements of the feed regulator valve by varying the pressure on top of the piston to maintain the feed regulating valve in a partially open position corresponding to the position taken by the pilot valve. The piston is made twice the area of the valve and, for equilibrium, the pressure on top of the piston is half-way between the feed pressure at the inlet to the regulator and the pressure at the outlet from the regulator. When the float rises and causes the pilot valve to open, the pressure on the piston increases and the feed regulating valve moves downwards until the pilot valve opening reaches the equilibrium position. When the float falls, the pressure on the piston falls and the regulating valve opens until the pilot valve opening establishes equilibrium pressure on the piston again. The operating water leaking past the piston flows into the boiler, eliminating the thermal and pumping losses which occur with regulators in which the leakage water is led back to the feed tank.

The various diagrams of feed systems given herein show the fundamental parts only, and require to be amplified to include all the subsidiary requirements which vary for different ships and owners. To illustrate this point, Fig. 32 shows the complete piping diagram for a closed feed system of a twin screw installation with water-tube boilers.

DISCUSSION.

Mr. S. A. Smith, M.Sc. (Member), opening the discussion, said that he would like to congratulate Mr. Hillier on his excellent paper, but as the author had confined himself to well-tried marine condensing installations there was, in the speaker's opinion, nothing to criticise. He would therefore content himself by stating a few facts on the running of feed systems in some of the latest ships of the Company with which he was associated.

On page 109 the author stated that for ships running in the tropics the sea-water temperature was generally taken as 80° to 85° F. From a large number of records of voyages to Australia at all times of the year, the average sea-water temperature over the whole voyage was found to be about 72° F. This would also be about the average voyage temperature on the China run. In the Red Sea the mean monthly temperatures in June, July and August ranged from 80° F. at Suez to 89° F. in the Gulf of Aden.

With a temperature of 72° F. the average vacuum over the voyage should be 28½ins. In the P. & O. service the condensing plant was designed to be capable of 28ins. vacuum (30in. bar.) with sea water at 82° F., giving a temperature difference between steam and inlet water of 19° F. Allowing 10° F. difference in temperature between the exhaust steam and overboard discharge, the temperature rise across the condenser would be 9° F. Knowing the quantity and quality of the steam to sufficient to deal with fairly high vacua. Would the author state at what vacuum it was found necessary to change over from the two-stage to the three-stage ejector.

Mr. G. Thompson, M.Eng. (Member of Council) said that in reference to Fig. 16 illustrating the open feed system with independent air pump, a point which had always worried him was that the air pump and ejector air-charged the feed tank and it was necessary to depend upon the direct contact heater to get rid of the air. In many older ships there was an air vent in the top of the direct contact heater, and he wondered how effective the direct contact heater was in eliminating air. He had a feeling that the air still remained at the top of the direct contact heater and found its way into the boiler. Had any quantitative tests of air content been made before and after the direct contact heater, and was there any simple apparatus which could be used on board ship to measure the oxygen content of the feed system?

Mr. W. A. Christianson (Member) said that Fig. 15 showed the steam air ejector discharging into the main condensate and asked how the arrangement was dealt with under manœuvring conditions. During the stoppages there would be no condensate flowing through to condense the ejector discharge, and one could not rely upon manual shutting-off and opening-up of the steam supply to the ejector to coincide with stopping and starting of the main engine.

On the proposal of **Mr. Sterry B. Freeman, C.B.E., M.Eng.** (Vice-President) a very cordial vote of thanks was accorded to the author.

The Author's Reply to the Discussion.

The Author, in reply, stated that the operating data supplied by Mr. Smith were very interesting and thanked him for his confirmation of the merits of the closed feed system.

The use of a low pressure evaporator, mentioned by Mr. Smith, utilizing low pressure bled steam, was the most economical method of producing distilled make-up water, and had the additional advantage that evaporation at low temperature eliminated scale formation to a large extent.

Where h.p. evaporators were fitted, a feed heated making use of some of the evaporator vapour to heat up the raw water before it entered the evaporator, reduced considerably the steam consumption required for a given quantity of make-up.

Regarding Mr. Smith's remarks concerning erosion of condensate extraction pump impellers and his suggestion that the impellers should be of Monel, the Author stated that the first stage impellers of the condensate extraction pumps supplied by the firm with which he was associated were now made of Monel metal.

The air pumps shown on pages 113 and 114 were included to illustrate the reasons underlying the development of the present combination of steam ejector and reciprocating pump which was widely used for vessels of small powers.

As regards the suggestion that air ejectors of larger capacity should be fitted in order to reduce the time taken to obtain the necessary vacuum for starting purposes, it was pointed out that this would result in an oversize air ejector being operated continuously throughout the life of the ship. Where it was desired to obtain a vacuum quickly, the practice in present day power stations of using a separate starting ejector might be adopted. The total weight of steam required to obtain a given vacuum was approximately the same, irrespective of the size of ejector used. A large single-stage ejector could be fitted, taking steam at a comparatively high rate and operating for a short length of time. In power stations, the starting ejector was usually capable of obtaining a vacuum of about 25″ hg. in four to five minutes. Suitable provision would, of course, require to be made on board ship to condense the operating steam used in a starting ejector.

The Author agreed that Mr. Green was correct in his suggestion that the large free water surface was the reason why a considerable quantity of water could be discharged into a Scotch boiler for a small rise in the water level. The looseness of the statement made in the paper arose from the fact that a large free water surface was invariably associated with the large water volume characteristic of a tank type boiler, such as a Scotch boiler.

With regard to Mr. Gillies' remarks, the make-up feed in a closed feed system was usually introduced above the air baffle and sprayed so that every opportunity was given for the air to be liberated from the make-up water. In some cases, the make-up water was led into a perforated pipe inserted through the bottom of the condenser and so arranged that the make-up water was sprayed through the perforations into the steam lane where the air was driven off and the water de-aerated before falling to the bottom of the condenser.

Regarding Mr. Jones' remarks, the Author stated that the three-stage ejector was considerably more economical than the two-stage ejector for high vacua of 29″ and above, and that it was advantageous to fit it for vacua above 28½″. Three-stage air ejectors were usually fitted in marine service because the sea temperature conditions permitted of high vacua being maintained at some part of the voyage or at some time of the year.

Regarding Mr. Thompson's remarks, most of the air discharged with the water into the feed tank was liberated in the feed tank and the oxygen content of the feed water passing to the direct contact heater was of the order of 4 - 6 cubic centimetres of oxygen per litre, depending on the temperature conditions and the mechanical treatment the water had received. When the feed water was passed through a direct contact heater, adequately vented, it was de-aerated to the extent of about 0·15 to 0·2 cubic centimetres of oxygen per litre. The Winkler test was the best known means of measuring the oxygen content of feed water, and the apparatus required was comparatively simple and inexpensive.

With regard to Mr. Christianson's remarks concerning the arrangement shown in Fig. 15, the Author stated that the general practice was to operate an engine-driven air pump without the steam ejector during manœuvring conditions; this arrangement provided sufficient vacuum for manœuvring purposes, the steam ejector being put into operation only when steady sea-going conditions were obtained.

Where the air pump was of the independent type, a byepass connection was sometimes provided which permitted a flow of feed water from the feed tank into the condenser, from where the air pump completed the circulation of such water through the air ejector condenser back to the feed tank. The circulation of this byepass water enabled the air ejector to be maintained in service during manœuvring conditions, the byepass being closed as soon as steady sea-going conditions obtained.

A Symposium on High Pressure Boilers.

A further contribution to the discussion on the series of papers published under the above heading in the April, 1935, Transactions (Vol. XLVII, No. 3, pp. 63-99) is published below together with the authors' replies thereto.

Mr. F. A. Pudney (Member) wrote that this interesting collection of papers dealing with steam generation made one realize that, whether one liked it or not, alternatives to the systems at present in general use were now available. Steam raising for central station work, for naval purposes, and very possibly for trans-Atlantic work (rather than for tramp and long-distance routes), appeared to be the chief aim at present. With modern methods of tube construction and the successful means adopted for jointing work, reliable service could no doubt be assured.

A Diesel exhaust gas boiler was indicated in Fig. 11 of the La Mont boiler paper, and from the author's remarks the writer gathered that in practice the construction had not given satisfaction, despite the fact that the draught appeared to be good. Apparently the nest of tubes was such that a virtual "Davy lamp" effect had been created, which would account for the poor operation with the exhaust "flame".

The arrangement in Fig. 5 of the Loeffler boiler paper might be interesting for special purposes, but there seemed little advantage in complicating a submerged part in order to simplify the prime mover. It was doubtful if a shipping company in this country would seriously consider reversing propellers for powers up to 50,000 s.h.p.

The construction shown in Fig. 4 of the Sulzer boiler paper was impressive. Could the author say what was the average length for a run of tubing before welded construction had to be resorted to, and how many welds there were in the generator tubing illustrated in Fig. 4? What local work could be carried out with this wound construction, as compared with the nest and bundle mounting? At the top of the illustration he had referred to there appeared to be a burst—or was it a blemish on the photograph?

The remark in the Wagner-Bauer boiler paper that "Finally it may also be pointed out that the most suitable design of the fittings contributed to the good work of the boiler" was most important and applied to all the boilers which were the subject of these papers.

The writer had not noticed any reference to the necessity for and method of cleaning, so that the attached view (Fig. 1) of an interesting tube

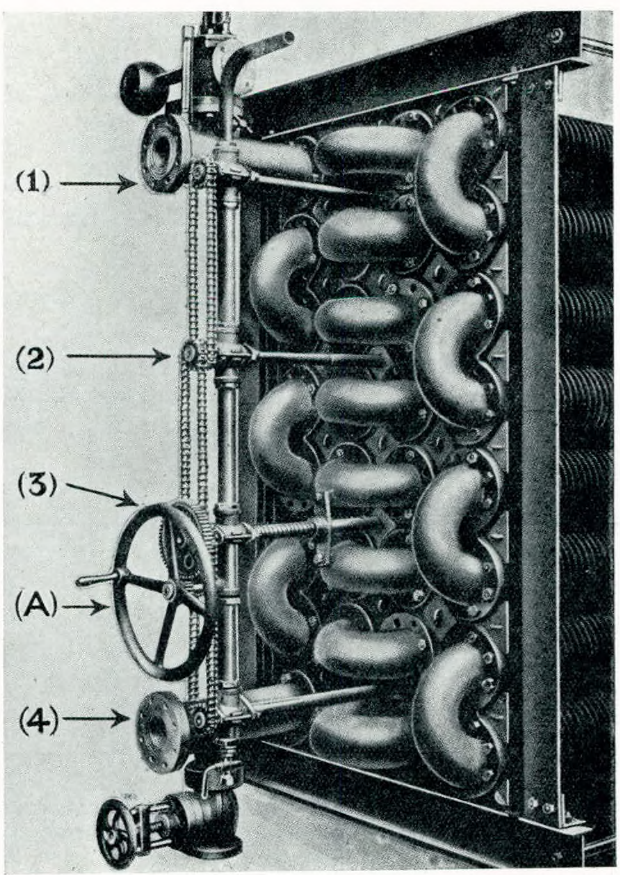


Fig. 1.

cleaner might be of value. The scheme indicated an application to some economisers viewed recently in Italy, and was known as the Taje Patent. The chief feature was that all tube cleaning could be carried out without opening doors or fronts, and the absence of all knuckles and swinging parts. As would be seen, the rotation of the handwheel (A) operated the chain which in turn rotated the steam carrying tubes (1), (2), (3) and (4), and by means of the nozzles arranged on those tubes very complete and quick cleaning could be obtained at all times, the whole operation being carried out from the platform level.

These comments on cleaning did not, of course, apply solely to the Wagner-Bauer boiler, but also to a number of the other types of generators described.

Dr. Ing. E. Goos, in reply, expressed his agreement with the remarks of Mr. Pudney, who pointed out in regard to the comments on boiler cleaning that Mr. Pudney had apparently overlooked the statement in the Benson boiler paper that the tubes could be kept clean easily by steam blowers and that, therefore, the efficiency was nearly always at the same level. Of course the steam blowers must be designed to meet the special requirements, but in this respect no difficulties had been experienced.

Dipl. Ing. D. W. Rudorff, replying to Mr. Pudney, stated that in its application as a Diesel exhaust boiler the La Mont boiler had given highly satisfactory results, as, for example, in the installation of the tanker "Toulouse" of the Wilh. Wilhelmsen Line, Oslo, Norway, which was shown in the accompanying illustration Fig. 2). The La Mont Diesel exhaust boiler installed in this boat had been subjected to an efficiency test during the trial run the results of which were quoted in the following short digest.

The boiler installed was of the type shown in Fig. 11 of the writer's paper. Its gas-touched heating surface was 646 sq. ft. and the operating pressure 115lb./sq. in. gauge. Exhaust could be supplied

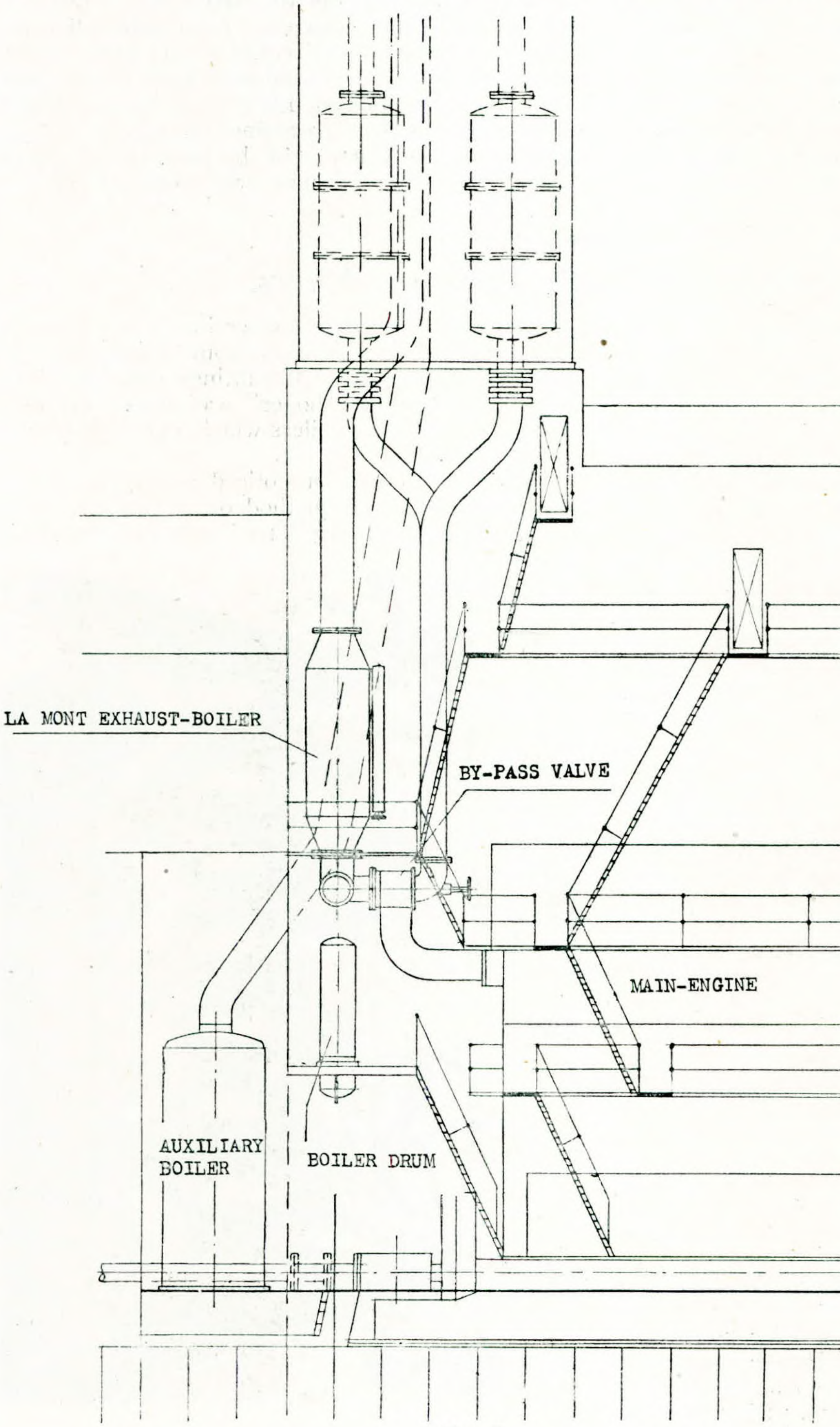


Fig. 2.

from either main engine (each of which developed 3,400 b.h.p.) or from both jointly. The feed water was supplied to the boiler at 122° F. During the trial run, exhaust from one engine delivering 3,050 b.h.p. entered the La Mont exhaust boiler at 729° F. and left at 426° F. Under these conditions 3,000lb. of steam were delivered from the boiler per hour at 115lb./sq. in. gauge pressure; the specific steam output amounted therefore to 0·984lb. per b.h.p. per hour. Based on the gas-touched heating surface, 4·65lb. of steam were raised per hour per sq. ft. of boiler heating surface. This high specific output, being more than twice that obtained in a similar installation of the cylindrical exhaust boiler type, showed that very high rates of heat transfer were obtained in the La Mont boiler. A computation showed that an average heat transfer co-efficient of 24·4 B.Th.U.'s per hour per sq. ft. gas-touched surface per degree Fahrenheit temperature difference was reached, which was a remarkably high figure for a waste-heat boiler of this kind. The draft loss through the exhaust boiler was ascertained at 1·73 inches H_2O, while a computation based on the draft loss formula given by Reiher in his well-known treatise would indicate no less than 4·95 inches H_2O. The author had already referred to this discrepancy, until now unexplained, in his paper.

From the drop in gas temperature and the steam output the boiler efficiency could be found as 86·0 per cent. In this particular installation 42·6 per cent. of the heat contained in the exhaust gas was utilized by the boiler. It would have been quite easy to increase this percentage by increase of the boiler heating surface if there had been any use for a higher steam output. In the "Toulouse", however, the useful amount of steam was rather limited, as steam was used mainly for preheating of the fuel oil. Provision was made to use some steam for heating purposes, but the heating period during each trip was rather short as the boat was used in the Europe-Australia service.

It could be shown that an increase in boiler heating surface by 161 sq. ft. or by 25·0 per cent. would have increased the total steam output to 3,260lb. per hour at an exhaust gas temperature of 392° F. at the boiler outlet. One could even go further and provide an economiser after the boiler; thus the feed water temperature could be raised from 122° F. to 266° F., which would cause the temperature of the exhaust gas at the boiler outlet to fall to about 327° F. The total steam output would then amount to 3,790lb. per hour. If each main engine of the "Toulouse" had been equipped with such exhaust boilers, about 530 b.h.p. could have been generated from the exhaust or 8·7 per cent. of the output of the main engines.

The installation in the "Toulouse", like those in other vessels, had given a highly satisfactory performance since the first day of operation and no trouble whatsoever had been reported by the owners.

Mr. J. Calderwood, M.Sc. (Member), in reply, stated that in the Sulzer boiler the tubing was obtained in normal commercial lengths, the exact length being chosen so that the welds came in a convenient position for access if repairs were required. The number of welded tube joints would therefore depend upon the size and arrangement of the boiler, but would generally be over 300. In the wound construction of the combustion chamber shown in Fig. 4 local repairs could be effected by bending a damaged tube inwards at the place of its two end welds, cutting these out and welding in a new length of tube which could then be bent back into place. The mark referred to at the top of Fig. 4 was a sight door opening; the tubes were bent to a form which allowed a small opening in the furnace wall.

Mr. O. Jebens, replying to those of Mr. Pudney's remarks which related to the Wagner-Bauer boiler, stated that in connection with the cleaning of the side exposed to combustion gases it had been found necessary to provide the air preheater and the nest of downcomer tubes with an adequate number of soot blowers. These, numbering five altogether, were shown in Fig. 2 of the paper. Two were placed inside the nest of downcomer tubes and three in the air preheater. These soot blowers were composed of heat-resisting tubes which were provided with a number of nickel nozzles. In order to sweep all gases as uniformly as possible the soot blowers, as they turned, were moved in an axial direction by a screw thread to the extent of one tube pitch each time.

Under normal working conditions it had been found advisable to blow the soot for a short time every three to four hours. On account of the high temperature of the steam used in the blowers, the heating surfaces were completely freed from soot without any accumulation in the nests caused by entrained water. Any permanent deposit with consequent reduction in the degree of efficiency of the boiler never occurred, even after a long period of service.

INSTITUTE NOTES.

EDUCATION GROUP.

An inaugural meeting of members of the Education Group was held at the Institute on Friday, May 10th, 1935. The Chairman of Council, Mr. T. R. Thomas, B.Sc., presided, and there were 23 members present.

The Chairman, after some introductory remarks, invited Mr. T. A. Bennett, B.Sc. (Member of Council), on whose initiative the Council had decided to form this section, to open a discussion on the plan of operations to be adopted by the Group. A lengthy discussion ensued, in which the comparative importance of technical and practical training for marine engineers figured prominently, as did also the question of the amount of academic training to be required of Probationer Students under the new By-Laws. The controversial nature of the main issues raised emphasised the desirability of coordination, through the medium of The Institute, of the educationists and the marine engineering industry, and it was generally agreed that, as the Chairman had remarked, the Council's action in forming this Group had been most opportune.

An Executive Committee was elected, consisting of Messrs. T. A. Bennett (Chairman), C. J. M. Flood, J. B. Harvey, T. W. Longmuir, F. Reid, H. Scott, R. F. Thompson, C. A. Walker, G. J. Wells and B. C. Curling (Secretary), this Committee being charged with the drawing up of an agenda for a further meeting from the material arising from the discussion.

ELECTION OF MEMBERS.

List of those elected at Council Meeting held on Monday, May 13th, 1935.

Members.

George Henry Alford, 6, Exeter Road, South Harrow, Middlesex.

George Anderson, Edenville, Grant Street, Burghead, Moray.

Charles Walter Blackwell, 225, Roman Road, East Ham, E.6.

Henry Hebblewhite Cuttle, 95, Brookvale Road, Southampton.

Harry William Delamare, 170, Humber Road, Blackheath, S.E.3.

William Pearson Farquharson, 26, Harcourt Terrace, S.W.10.

Norman Douglas Vandendriesen Ferdinands, Brookside, Blake Road, Borella, Ceylon.

Framroze Kaikhushro Jaboolee, 12, Todivalla Road, Poona, Bombay.

James Stuart Jolly, c/o Anglo-Iranian Oil Co., Ltd., Abadan, S. Iraq.

Hugh Alexander Livingstone, 88, Marlborough Avenue, Hull.

John McAfee, Lloyd's Register of Shipping, 71, Fenchurch Street, E.C.3.

Leo Joseph Louis Menezes, Chief Engineer, s.s. "Victoria Marie", c/o The Merchant Steam Navigation Co., Musjid Bridge, Bombay, 3.

John Grant Munro Millar, 3, Howard Place, Edinburgh.

Samuel Spiller Nickels, Chief Inspector of Boilers, Shillong, Assam.

Robert Allan Pratt, 11, Creswell Avenue, North Shields.

Harry Rainford, 21, Carlton Street, Prescot, Lancs.

Aeneas Dundas Singer, c/o Anglo-Persian Oil Co., Ltd., Naft Khaneh, Khanaqin, Iraq.

John Oliver Thompson, 32, Lanville Road, Aigburth, Liverpool, 19.

Associate Members.

David William Beeken, Ardgowan, 30, Palmerston Road, Buckhurst Hill, Essex.

Ronald Rutherford Henderson, Delaware, 5, Leigh Park Road, Leigh-on-Sea, Essex.

Associates.

Leonard Henry William Davis, No. 2 Riverside Cottages, Alfred's Way, Barking, Essex.

Wilfred Emile Gooday, Consulting Petroleum Technologist, Brettenham House, Wellington Street, W.C.2.

Transfer from Associate Member to Member.

James Renwick Leslie Hilson, The New Zealand Refrigerating Co., Ltd., Picton, New Zealand.

Charles Victor Lewis, Toowong, Hallowes Lane, Dronfield, near Sheffield.

Frank Lewis Turner, 8, Levendale Road, Forest Hill, S.E.23.

Charles Henry Stanbridge, Electricity Department, Port Elizabeth Municipality, Box 369, S. Africa.

ADDITIONS TO THE LIBRARY.

Purchased.

Lloyd's Register of Yachts, 1935.

Presented by the Publishers.

"Heavy-Oil Engine Performances as Recorded on Log Sheets", by W. A. Tookey. Diesel Engine Users Association.

Annual Reports of the Society of Chemical Industry on the Progress of Applied Chemistry, 1934, Vol. XIX, containing the following:—

"General, Plant, and Machinery", by Richards and Vowler.
"Fuel", by Raine and Winterbottom.
"Gas, Carbonisation, Tar, and Tar Products", by Hollings and Voss.
"Mineral Oils", by Goulston.
"Intermediates and Colouring Matters", by Rodd and Coffey.
"Textiles, Fibres, and Cellulose", by King.
"Pulp and Paper", by Underhay.
"Bleaching, Dyeing, Printing, and Finishing", by Speakman.
"Acids, Alkalis, Salts, etc.", by Parkes.
"Glass", by Robertson.
"Refractories, Ceramics, and Cements", by Lynam and Rees.
"Iron and Steel", by Hudson.
"Non-Ferrous Metals", by Powell.
"Electro-Chemical and Electro-Metallurgical Industries", by Cuthbertson.
"Oils, Fats, and Waxes", by Hilditch.
"Paints, Pigments, Varnishes, and Resins", by Members of the Oil and Colour Chemists' Association.
"Rubber", by Dawson.
"Leather and Glue", by Woodroffe.
"Soils and Fertilisers", by Crowther.
"Sugars, Starches, and Gums", by Eynon and Lane.
"The Fermentation Industries", by Hopkins and Norris.
"Foods", by Cox.
"Fine Chemicals, Medicinal Substances, and Essential Oils", by Stedman.
"Photographic Materials and Processes", by Horton.
"Explosives, 1933-34", by Weir.
"Sanitation and Water Purification", by Garner.

The British Electrical and Allied Industries Research Association. Sub-Committee J/E: Joint Committee: Steels for High Temperatures: The

Creep Strength (Dauerstandfestigeit) of Steels in Relation to Alloying Constituents and Heat Treatment, by Paul Grünl. Bibliography of Literature on the Behaviour of Steels at High Temperatures, furnished by "The Engineering Index Service".

"Scientific Research and Social Needs", by Professor Julian Huxley. Watts & Co., Ltd., 287pp., illus., 7s. 6d. net.

In order to prepare the material for a series of lectures given under the auspices of the B.B.C., the author paid a number of visits to various government and commercial laboratories, and an account of that survey of experimental work is given in this book. As the whole field of scientific interests was more or less covered in the course of those visits, the index of the volume contains references ranging from matters pertaining to agriculture to others relating to yellow fever. Having in mind the part played by engineering science in present-day affairs, the record naturally includes many comments on work that is being done in laboratories attached to some of our larger engineering establishments; since information on this aspect of the subject is given from time to time in our professional journals, many engineers will be well acquainted with what Professor Huxley has to say on the experimental interests of engineers at the moment. For obvious reasons, the pages devoted to biological and allied sciences constitute the most instructive—as well as constructive—part of the work.

The author is a whole-hearted believer in the efficacy of science as a necessary and sufficient instrument for the improvement of mankind, since on page 177 he remarks: "I shall not be able here to say much about sociology, except to point out that if we want to control the development of society in an efficient, orderly way, we had better trust to science instead of the so-called common-sense opinion, blind economic forces, politics, or revolutions". In this connexion it should be noted that the history of science, as well as the evidence before us of large-scale experiments that are being conducted along "scientific" lines at the present time in various quarters of Europe, make manifest the fact that science can be as biassed as politics in its outlook. Indeed, it may be said that the society of men is suffering not from want of science, but from the common-sense that permeates such agelong treatises as Plato's *Republic* and Aristotle's *Rhetoric*, or, in short, what the scholar calls the humanities. If the scheme outlined in these chapters were effected, everything would be "planned", to use a favourite word of the author's, from the cradle to the grave, so that on finishing the book one is impelled to ask where does the individual spirit enter, apart from forming a cog in the machinery that the author would erect for the enlightenment of scientists and their laboratory assistants.

There is another, and obvious, danger associated with the point of view taken in the volume, which is hinted at by an observation on page 270: "Well, look at the absence of scientific progress in other periods and places—in India for instance, or in China". Had the writer remembered that those two countries in particular form the source of a considerable part of the world's philosophy, he would have saved himself the longing sigh for the time when we should be "able to open and shut our minds to each other at will" (page 125). Actually, the philosophy and practices of the Yogi, in India, claim to have achieved this and much more, and the origin of that school of thought is almost lost in the mists of antiquity. While the reviewer holds no brief for the science of Yoga, he is of the opinion that scientific inquiry demands an open mind.

Part of the subject-matter consists of discussions between Professor Huxley and his friends, but these talks are rather academical in content, which doubtless would have been avoided to a certain extent had the circle included such minds as are exemplified in the person of Gilbert Murray, or his American counterpart, Nicholas Murray Butler. After all, if social needs are under consideration, it is well and just to glance at the message written along the corridors of time, to the effect that the spirit of man is greater than his parts.

In conclusion, it should be added that there is much of interest in the volume before us, even if the author's conclusions regarding the possible service of science to humanity are sometimes anything but constructive when seen through the eyes of an ordinary engineer.

"Planning, Estimating and Ratefixing", by A. C. Whitehead. Sir Isaac Pitman & Sons, Ltd., 293pp., illus., 10s. 6d. net.

This book is intended to meet the needs of production engineers and students, in which categories the author includes mechanics, draughtsmen, estimators, ratefixers, planning engineers, cost accountants, foreman, managers, and others who now or who one day will control production.

The writer confesses that when he promised to review this book he anticipated several hours of dry reading on a hackneyed and often maltreated subject. The author, however, quite departs from orthodox presentation and appears to know, contrary to the authors of most other volumes on this subject, that when the stop watch has done its best, the human element is the most important factor. In consequence, the book soon becomes most interesting, especially to a works manager, such as the writer, who has to strike a balance between the time-study man and the man who has to do the job.

All the ordinary workshop processes are studied, and the various chapters contain hints on workshop planning and machine layouts, including a description of how to arrange for conveyor controlled output. The methods of estimating real, as compared with nominal productive capacity, together with capacity charts, are also considered.

One of the chief merits of the book is that the treatment of time study is based on the author's own conclusions. To quote his preface: "What I have written I know to be true, and I have tried to be so definite and clear that there is no ambiguity anywhere".

To enable the book to be used by all those for whom it is written the author has refrained from using difficult mathematics. It can be strongly recommended to anyone interested in costing processes, whether they practise the whole art of time study or not.

"Engineering Applications of Electron Physics", by G. Windred. The Draughtsman Publishing Co., Ltd., 96, St. George's Square, S.W.1, 46pp., illus., 2s. net.

This is one of the technical publications of the Association of Engineering and Shipbuilding Draughtsmen, and it is what it purports to be, a brief outline of the principles of modern electron physics and the application of these to industrial practice.

The author suggests that his object is to show the value in engineering practice, but this is only true of special branches of engineering, particularly electrical.

The pamphlet is well written and, although it deals with matters with which the marine engineer is not familiar, can be read with interest and profit by him. The principles with which the author deals form the basis of a great many practical applications and it is to be expected that these will tend largely to increase in the future.

The engineer who takes the trouble to devote the limited time required to this publication will be well repaid.

"The Ventilation of Ships", by F. L. Bullen. Charles Birchall & Sons, Ltd., 17, James Street, Liverpool, 148pp., illus., 7s. 6d. net.

The problem of maintaining suitable air conditions throughout the various compartments into which a ship is divided, is one which requires treatment along different lines according to the nature of each individual space. Some compartments provide stowage for cargo, sometimes of an inert character and sometimes productive of gases which must be carried away for safety. Other compartments, machinery spaces for instance, and refrigerated compartments, provide ventilation problems involving considerations of temperature gradients and so on. Accommodation spaces require the provision of refinements in ventilation —given the title "air-conditioning"—which call for more or less elaborate schemes of piping and plant, according to the extent to which a simple meeting of the actual needs of the case may be influenced by the desire to obtain a very high standard of individual comfort.

Mr. Bullen's book provides a clear and concise picture of the nature of the considerations involved in these several types of ventilation problems, and does this in a manner which is authoritative and easy of understanding. The volume is furnished with numerous graphs, tables, and drawings, and appears likely to prove of great value to all technically interested in the maintenance of proper atmospheric conditions throughout ships—either from the designing or constructional standpoint, or from the point of view of sea-going responsibility. The book is admirably set out, and ranges competently over the whole subject from both the theoretical and practical aspects. It should prove a very successful addition to the library of technical works which is gradually being built up by these publishers.

"Liquid Fuels", by Harold Moore, M.Sc. The Technical Press, Ltd., 263pp., illus., 21s. net.

This book consists of six parts and covers a wide field which is dealt with in a very practical manner. The subjects are put forward in a way to interest both chemists and engineers who are concerned with the manufacture, properties, utilization and analysis of oil.

Parts 1 and 2 are chiefly concerned with the chemistry and preparation of liquid fuels, including not only petroleum oils but also products such as shale and coal oils, vegetable and animal oils, etc. The information given is concise but sufficient for those who wish to make a general survey of the subject without going into too much detail.

Utilisation of fuels in engines is dealt with in Parts 3 and 4. Some of the author's previous book on "Liquid Fuels for Internal Combustion Engines" is repeated, but the text has been greatly amplified and brought up to date. The latest methods of testing petrol for octane rating and Diesel fuel for ignition quality or cetene value are described.

Part 5 is devoted to fuels for external combustion, and a useful chapter is included on the chemistry of combustion, while a further chapter gives details of various types of domestic and furnace fuel oil burners.

Methods of analysis of oil fuels and the significance of the tests occupy Part 6, and all tests likely to be met with in any fuel specification are fully described.

The book also contains, in an appendix, a number of graphs, charts and tables referring to oil constants which are extremely useful for reference purposes.

JUNIOR SECTION.

Debate:—Should the Marine Engineer have a University Training.

The final meeting of the 1934-35 Session took place in the Reading Room of The Institute on Thursday, April 18th at 7 p.m., for which occasion a debate had been arranged on the question "Should a Marine Engineer have a University Training?" Due probably to the date being the eve of the Easter holidays, the attendance was much below the average for these meetings, while two of the three principals—Mr. E. F. Spanner, who had undertaken the duties of Chairman, and Mr. H. R. Tyrrell, B.Sc. (Associate Member), who had entered the lists as spokesman for the "Noes" in the debate—were prevented from attending the meeting, due to a business emergency and illness respectively. Mr. G. J. Wells (Vice-President) deputised as Chairman, and Mr. E. W. Cranston, Wh.Sc. (Associate), after only one day's notice, took Mr. Tyrrell's place in opposition to the advocate of a University training—Mr. K. P. Harman (Student).

Mr. Harman began by confining his interpretation of "a University training" to the normal degree course in Mechanical Engineering, coupled with a more detailed study of Marine Engines, the course being taken at an institution such as the City and Guilds Engineering College, Armstrong College, Newcastle, or one of the London Polytechnics. A course of evening study, he suggested, could not be considered as a possible substitute for the training to which he was referring.

Marine Engineers of the present generation were responsible for the efficient operation of such a wide range of complicated machinery that the knowledge required far exceeded that which sufficed for the marine engineers of a generation ago. To be efficient, a marine engineer to-day must have been trained in the application of physical laws, and must have considerable mathematical ability. He must also have a sound knowledge of thermodynamics, chemistry and electro-technology, to enable him to diagnose correctly faults in Diesel engines or electrical machinery, or corrosion troubles. The University training gave the requisite knowledge of fundamental principles underlying the action of any of the various types of machine he would be called upon to handle, and what was more important, a mind trained to apply underlying principles quickly and accurately, and to grasp essential facts with the minimum effort.

The acquisition of such theoretical knowledge, he considered, was not possible by evening study after fatiguing work in the machine or fitting shops. The enforced neglect of recreation and of education outside technical spheres which this plan necessitated could but adversely affect both the physical and mental health of such students.

The man who served a normal works apprenticeship and then went to sea had little or no opportunity for experimental work such as was carried out in the University laboratories. In his opinion such experience was of great value and importance to a marine engineer. Again, a young engineer whose experience was limited to that gained in the works to which he was apprenticed might lack a sense of proportion, in that his outlook might be that of a narrow-minded specialist, whereas a University training would have made him familiar with the underlying principles of other branches of engineering.

For research work a University training was essential; research work was a necessary preliminary to any invention of major importance. Hence the speaker argued that unless marine engineers were equipped with a University training, research and invention, the most interesting part of marine engineering, would be taken right outside their scope and left to people on shore, who had not had the opportunity to study existing engines over long periods under all working conditions. This would be a most unsatisfactory state of affairs, and would seriously retard the development of the science of marine engineering.

Mr. Harman further argued that a University training offered more opportunities to meet men of widely differing ideas and varying temperament than did the normal life of an apprentice in a workshop, and that with due respect to the social welfare schemes existing in some of the larger firms, the advantages from the social point of view lay with the University student, with resultant benefits to his personality and character. He advocated two years' experience in the workshops *after* the University training.

In conclusion Mr. Harman suggested that the professional status of marine engineers would be raised if a University training were made compulsory or became more general.

Mr. Cranston prefaced his remarks by defining a Marine Engineer as an engineer responsible for the care and maintenance of the machinery on board a ship. His observations would apply to these officers, and not to the small minority engaged as superintendent engineers or less directly connected with marine engineering in the practical sphere. A very high standard of technical ability was not required, but a high degree of practical skill and experience combined with a fair amount of knowledge of engineering theory was essential. It had been argued that marine machinery was becoming so complicated that more and wider technical knowledge was necessary on the part of the marine engineer, but in the large vessels to which this argument referred there was sub-division of responsibility and there were various classes of engineers in addition to those in charge of the propelling machinery, such as refrigerating engineers and electrical engineers in charge of certain sections of the auxiliary machinery, and no one man was expected to be an all-round expert.

It was agreed that an apprenticeship of at least four years served in the engine shops, with certain alternatives, was the essential preliminary to a marine engineering career. How was it possible for a would-be marine engineer to serve such an apprenticeship, qualify to enter a University and complete a University training? He might (a) while serving his apprenticeship attend evening classes in order to qualify and subsequently enter a University or (b) qualify at school and proceed thence to a University, and serve his apprenticeship after obtaining his degree. There were disadvantages in both schemes In the first plan while at the University he would lose much of the practical skill he had acquired during his apprenticeship, while, conversely, in the second plan the University-trained man would lose a great deal of his highly theoretical knowledge during his shop training. In both cases a serious disadvantage was the length of time required to obtain the University training and serve the apprenticeship, not to mention the cost of such a combined training, which for many young men would be prohibitive, especially in plan (b) where a premium would probably have to be paid for the apprenticeship.

The course Mr. Cranston advocated was that by which a youth left school and started his apprenticeship at the age of sixteen, with the addition of evening study at a technical school (or, as some firms allowed, at part time day classes). In this way practical skill and a fair technical knowledge were gained hand in hand, and the cost of obtaining them was comparatively small.

The speaker was of opinion that with few exceptions, present rates of remuneration of sea-going engineers were not commensurate with the cost involved in obtaining the combined University and apprenticeship training previously described as schemes (a) and (b). He could not support Mr. Harman's suggestion that compulsory or more general University training would raise the professional status of marine engineers. He doubted whether a University-trained man would remain a marine engineer, and thought it more probable that after obtaining his Certificates of Competency, he would leave sea-going for some more congenial post ashore, which might have no connection with marine engineering. If this became general the machinery of the majority of ships would be in charge of comparatively inexperienced engineers. He considered that if University-trained engineers entered the marine service in large numbers, marine engineering as a whole would suffer in that respect. In conclusion he asserted that designers of marine engines did not depend on a great amount of highly technical calculations, but preferred to rely on the results of past experience—the type of experience which could not be obtained at a University.

The Chairman then declared the meeting open for a discussion, in which almost everyone present participated, and which brought out an important factor in the problem, i.e., that a single educational principle could not be laid down as applicable to all marine engineers alike, owing to the wide variety of types and sizes of ships in the Merchant Navy, from the largest liner to the humblest tramp or coaster. One could visualise a selective system being established in the near future, which would provide for the absorption by the Merchant Navy of marine engineers trained on both the lines indicated by Mr. Harman and Mr. Cranston. Such a solution was apparently favoured by those who, when the Chairman called for a division, abstained from voting. The remainder divided in the proportion of approximately two to one against a University training.

Votes of thanks to Messrs. Harman and Cranston, and to Mr. Wells for his admirable chairmanship concluded a very interesting and instructive meeting.

ABSTRACTS.

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

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.

Name.	Grade.	Port of Examination.
For week ended 11th April, 1935 :—		
Thomas, Noah G.	2.C.	Cardiff
Demery, Jack	2.C.	"
Bates, Hubert O.	2.C.	"
Swain, Robert F.	2.C.	Glasgow
Rankin, Stuart M.	2.C.	"
Morison, George G.	2.C.	"
Lee, John L.	2.C.	Liverpool
Foggitt, Thomas V.	2.C.	"
Curphey, George J. S.	2.C.	"
Abbott, Benjamin W.	2.C.M.	"
Wilson, Herbert J.	2.C.	London
Waddell, Robert D. F.	2.C.	"
Murchison, Roderick I.	2.C.	"
Scaife, Roy	2.C.	Newcastle
Raine, Daniel N. P.	2.C.	"
Raddings, John L.	2.C.	"
Johnson, William E.	2.C.	"
Hawdon, Wilfred	2.C.M.	"
For week ended 18th April, 1935 :—		
Jaboolee, Framroze K.	1.C.	Liverpool
Jones, Robert E. E.	1.C.	"
Preece, Frederick C.	1.C.	"
Whitehead, Thomas	1.C.M.	"
Faulds, Edward	1.C.	Glasgow
Bell, Robert A.	1.C.M.	"
Mackay, Alexander	1.C.M.	"
Hewitt, Clark	1.C.M.	Newcastle
Johnson, Thomas F. E.	1.C.M.	"
Kirton, Reginald M.	1.C.M.	"
Moffat, William J.	1.C.M.	"
Rawlings, Robert	1.C.M.	"
Blackwell, Charles W.	1.C.	London
Dowling, Thomas	1.C.	"
Lusher, Ernest	1.C.M.E.	Newcastle
Robertson, George T.	1.C.M.E.	"
Gane, Gilbert R.	1.C.S.E.	London
McMullan, George	1.C.M.E.	"
Edwards, John A.	1.C.M.E.	Liverpool
Beattie, Robert P.	1.C.S.E.	"
Wilcox, George J.	1.C.M.E.	Cardiff
For week ended 25th April, 1935 :—		
Barker, John	2.C.	Newcastle
Craig, Frederick C.	2.C.	"
Davison, Joseph H.	2.C.	"
Dobby, Stanley	2.C.	"
Lynch, John R.	2.C.	"
Marshall, William F.	2.C.	"
Miller, George S.	2.C.	"
Naisbitt, Robert H.	2.C.	"
Souter, William D.	2.C.	"
Almond, George N.	2.C.M.	"
Little, Thomas R.	2.C.M.	"
Taws, Henry H.	2.C.M.	"
Taylor, William S.	2.C.	Liverpool
Weaver, Douglas R.	2.C.	"
Birnie, William F.	2.C.M.	"
Jones, Francis	2.C.M.	"
Morris, Frederick T.	2.C.	London
Turner, Archibald S.	2.C.	"
Weller, Herbert C. G.	2.C.	"
Williamson, Leslie	2.C.	"
Haddow, Reginald A.	2.C.M.	"

Name.	Grade.	Port of Examination.
Simmons, George D.	2.C.M.	London
Beattie, Lewis H. H.	2.C.	Glasgow
Esler, Andrew	2.C.	"
Hamilton, George	2.C.	"
Kerr, Robert	2.C.	"
McKiernan, James J.	2.C.	"
Turnbull, George	2.C.	"
Patterson, John W. T.	2.C.M.	"
Scott, Samuel S.	2.C.M.	"
For week ended 2nd May, 1935 :—		
Boxwell, Leonard M.	1.C.	London
Kirkwood, Robert C. C.	1.C.	"
Stratta, Joseph F.	1.C.	"
Downey, William A.	1.C.M.	Newcastle
McPhie, Leonard B.	1.C.	"
Hearn, Maurice	1.C.	"
Ashton, George	1.C.	"
Robinson, Kenneth O.	1.C.	Liverpool
Simpson, Hector S.	1.C.	"
Todd, Robert	1.C.	Glasgow
Phillips, Thomas	1.C.	"
Copeland, George A.	1.C.	"
Potts, William S.	1.C.M.E.	Liverpool
Davies, Evan T.	1.C.M.E.	"
Thomson, James R.	1.C.M.E.	Newcastle
Mackintosh, George Y.	2.C.S.E.	"

World Power Conference, Annual Report, 1934.

Prepared by the Central Office of the World Power Conference, 36, Kingsway, London, W.C.2.

The World Power Conference was founded in Great Britain in 1924, when the first Plenary Meeting was held at Wembley. Since that year, as is well known, the Second Plenary Meeting has been held at Berlin and a number of Sectional Meetings have taken place. In response to a widespread demand for a Report, which will cover briefly but comprehensively not only the Meetings held under the auspices of the International Executive Council, but the numerous other activities of the World Power Conference as a permanent organisation, the enclosed Annual Report has been prepared by the Central Office. It is the first of a series, to be issued in March of each year.

SUMMARY.

SCANDINAVIAN SECTIONAL MEETING.

The papers presented at the Scandinavian Sectional Meeting, 1933, centred round the power problems of large-scale industry and of land and sea transport. The Transactions were published during 1934 in seven Volumes, uniform with the Transactions of previous Conferences.

CHEMICAL ENGINEERING CONGRESS.

The Chemical Engineering Congress will be held at the Central Hall, Westminster, between June 22nd and June 27th, 1936, as a Sectional Meeting of the World Power Conference. The Technical Programme was issued at the beginning of March, 1935.

STATISTICAL YEAR BOOK OF THE WORLD POWER CONFERENCE.

After preparatory work extending over seven years, Tables have been approved for the collection of international statistics on a strictly comparative basis dealing with the natural and developed resources of the world in fuels and in power. It was found to be necessary, in order to ensure complete uniformity, to prepare elaborate definitions of the terms and units employed in the headings to the Tables.

The blank Tables have now been distributed for actual completion, and it is expected that the first Statistical Year Book of the World Power Conference will be published at the end of 1935.

"WORLD SURVEY: PUBLISHED UNDER THE AUSPICES OF THE WORLD POWER CONFERENCE".

A new monthly periodical, with the above title, will begin publication during April, 1935. It will contain articles and economic and statistical data, centring round but not confined to the power and fuel industries. It will also contain a highly selective International Power and Fuel Bibliography, in which the individual Bulletins published in the past by several of the larger National Committees (including the British National Committee) will be merged.

Although not the official "organ" of the World Power Conference, "World Survey" will be able to rely upon the active collaboration of the 49 member countries. This fact will, it is hoped, enable the new Journal to make an unique contribution to the solution of world—and national—problems.

STANDARDISATION.

The World Power Conference does not itself undertake standardising work, but as representative of the "users" of standardisation it brings before the appropriate body or bodies matters in respect of which it considers standardisation desirable. The World Power Conference has also met with success in its endeavours to co-ordinate the work of the different standardising authorities. It has prepared a confidential Report on the present position of the organisation of standardisation, both internationally and nationally, and a revised edition of this Report is shortly to be placed on sale as an official publication of the World Power Conference.

INTERNATIONAL COMMISSION ON LARGE DAMS OF THE WORLD POWER CONFERENCE.

The Transactions of the First Congress on Large Dams, held in Sweden in 1933, were published during 1934 in five Volumes. Discussion is now proceeding regarding the Programme of the Second Congress on Large Dams, the date and place of which have not, however, yet been decided upon. The Commission is promoting research upon the subject of special cements for use in the construction of dams. It is also preparing an International Register of Dams retaining more than 50ft. depth of water.

The Lewis Single-drum Water-tube Boiler.

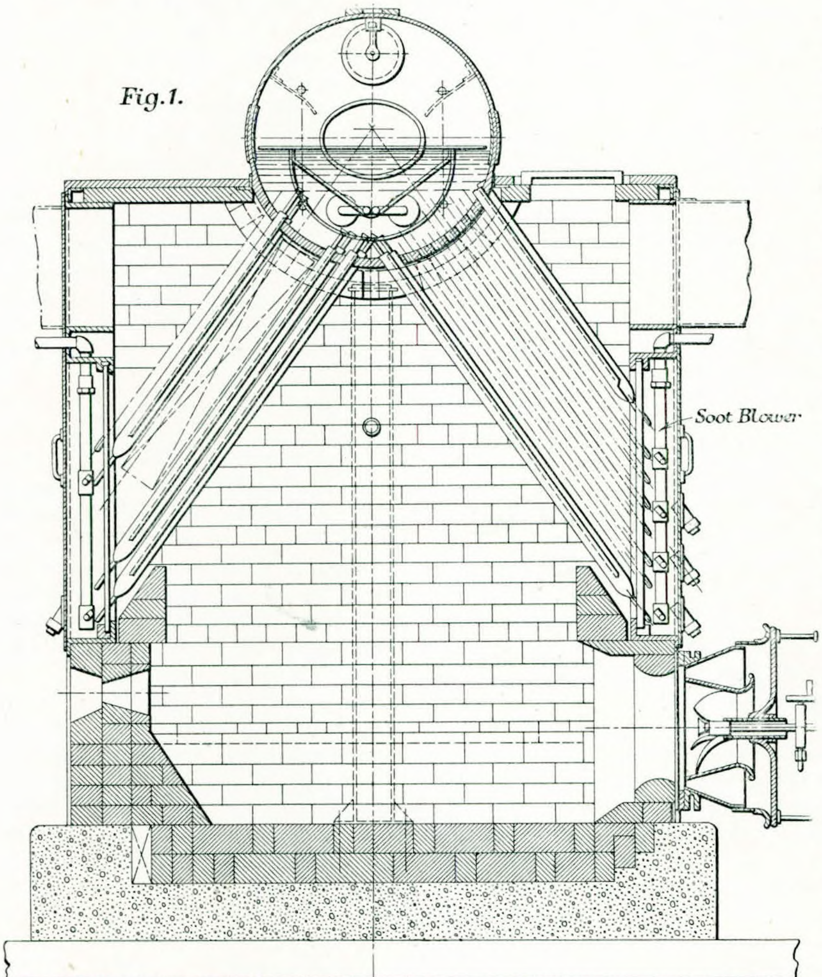
"Engineering", April 26th, 1935.

We illustrate on page 39, a type of water-tube boiler designed by Mr. W. Yorath Lewis, M.I.Mech.E., No. 3 Engineering Centre, British Industries House, Marble Arch, London, W.1, and having, it is claimed, several advantages in comparison with some of the existing types. The general arrangement of an experimental unit installed in the Northampton Polytechnic Institute, London, is illustrated in the transverse section of the boiler and furnace given in Fig. 1, while Fig. 2 (not here reproduced) shows the boiler in position. As will be seen from the former, a single horizontal steam and water drum is employed and the tubes connected to this are arranged in two banks forming an inverted V. The lower ends of the tubes, however, are free, the bottom drums commonly employed with this disposition of the tubes being eliminated. It is pointed out that with this arrangement the number of expanded tube joints is reduced, the weight of the water drums and their contents is eliminated and the width of the boiler for a given heating surface is reduced by the diameter of one water drum.

This form of construction has been rendered possible by the use of a special form of U-tubes for the heating surface. Each U-tube is made up of a pair of tubes, 1½in., 2in., or more in diameter, which are straight for practically the whole of their length, but are slightly curved at the bottom where they are forged or welded together, forming a spear-head end similar to that of some superheater elements. The two legs of the U thus formed lie close together and parallel and their upper ends are expanded into the steam and water drum in the usual way. Into each leg of the U-tubes is fitted another tube of smaller diameter terminating just before the bend, these tubes acting as downcomers for the circulating system. As will be clear, on reference to Fig. 1, they extend beyond the upper ends of the U-tubes and are expanded into a specially designed chamber inside the steam and water drum, in which chamber the water freed from steam is collected. This "solid" water descends through the downcomer tubes and ascends through the annular space between the downcomers and the walls of the U-tubes where evaporation takes place. The circulation, it is pointed out, is assisted by the fact that the density of the water in the downcomers is greater than that of the mixture of steam and water in the annular spaces. In connection with the cirlation system it should be mentioned that the bends at the lower ends of the U-tubes are each divided into two compartments, so designed that the water descending through the downcomer in one leg of the U-tube ascends through the annular space in the other leg. Thus an abrupt change in the direction of flow, such as occurs in an ordinary Field tube, is avoided. Moreover, the arrangement of the steam and water drum is such that the steam and water mixture rising from each heating-tube annulus is kept entirely clear of the heavier steam-freed water supplied to the downcomer tubes, and this arrangement, it is claimed, ensures a steady water level, dry steam and freedom from priming to which the cover of the central chamber and the baffles indicated in Fig. 1 also contribute.

It is claimed that the velocity of circulation is sufficient to sweep round the bends of the U-tubes any precipitation that might occur, but as the use

Fig. 1.



of pure feed water is now general there is little likelihood of any deposit occurring. In other cases, it is stated, the use of a continuous blow-down system in conjunction with softeners and treating apparatus would be sufficient to ensure freedom from injurious scaling or choking of any internal passages or surfaces. For special conditions of pressure and capacity and for extreme economy in weight, space and fuel consumption, it is pointed out that the internal chamber in the steam and water drum could be completely closed and connected to a circulating pump delivering into it but fed from outside. In this way, it is claimed, very high rates of heat transmission can be reached. These may be facilitated by the insertion of corrugated plate baffles longitudinally between the heating tubes giving an extra long traverse of the gases lengthwise and across the tubes with high gas velocities throughout. No difficulty, it is stated, arises in providing the forced-draught pressures necessary to produce such gas velocities.

Although only one arrangement of the heating surface is shown in Fig. 1 it will be clear that the arrangements can be modified to suit particular requirements. For instance, water walling of the end or side walls of the furnace could be effected and all the tubes could be of equal length, if required, instead of diminishing in length from the furnace side outwards as shown. A useful feature rendered possible by the elimination of the bottom drums relates to the soot blowers. As shown in Fig. 1, steam or air jets can be arranged in a simple and convenient manner for blowing upwards along all the tubes and covering practically the whole of the heating surface.

The boiler illustrated, which, as already stated, is installed at the Northampton Polytechnic Institute, has a normal evaporative capacity of 2,000lb. per hour, from and at 212° F., and a working pressure of 200lb. per sq. in. The heating surface of the main bank of tubes is 180 sq. ft., and that of the other bank, in which the superheater will be located, is 90 sq. ft., giving a total heating surface of 270 sq. ft. It should be noted that the superheater is not yet fitted and there is no flue on this side of the boiler; when in service a superheat of 200° F. will be imparted to the steam. The overall dimensions of the boiler with 14in. brickwork are 7ft. 6in. by 5ft., by 8ft. 6in. in height to the centre of the drum, the volume of the combustion chamber being 75 cub. ft. The weight of the pressure parts, including the drum, mountings and tubes is 4,000lb., while that of the water in the drum and tubes is 1,500lb. The weight of the supports, framework, casing doors, burners, etc., is 3,000lb. and that of the brickwork 15,000lb., so that the total weight is 23,500lb., or 10½ tons. In a trial carried out by Dr. Docherty and his staff, we are informed, the efficiency, at an evaporation rate of 7½lb. per lb. per hour per sq. ft. of heating surface, was 66 per cent. on the higher calorific value of the fuel and 70 per cent. on the lower value.

The Exhaust Turbine for Trawlers.

"Shipbuilding and Shipping Record", 14th February, 1935.

The development of a special type of exhaust turbine, mounted vertically above the propeller shaft and eliminating hollow shafts, has reduced the cost and size of the equipment and greatly facilitated its use where the total power is less than 1,000 h.p. Replacing the low-pressure cylinder of a triple-compound engine by an exhaust turbine effects yet further reduction in space and weight, and the steam consumption of a 750 h.p. "turbo-compound" set is about 8·25lb. per i.h.p.-hour, with steam at 617° F., and 97 per cent. vacuum.

The special small-power equipment is claimed to be particularly suitable for trawlers. More than 45 vessels of this type have already been fitted with exhaust turbines, and all the new steam trawlers at present under construction in Germany are said to be so equipped. The extra power available gives the vessels a greater radius of action, facilitates trawling, and expedites marketing. Also, the inertia of the turbine and gears checks racing of the propeller in rough weather and enables fishing to be continued when it would otherwise have to be suspended. Returns published by "Hansa" show that trawlers with exhaust turbines bring in the best catches, and comparisons of catches before and after the installation of exhaust turbine show an increase of 41·5 per cent. in the mean daily catch. The smallest set yet built is the 140 i.h.p. exhaust turbine of the Hochseefischerei N. Ebeling's trawler "Hamburg" which, with a reciprocating engine of 340 i.h.p., develops a total of 480 i.h.p. The coal costs of a reciprocating engine with exhaust turbine and cylindrical boiler are given as about 7s. per 1,000 i.h.p.-hour (at 20 marks=£1), falling to 4s. 8d. with a high-pressure, superheated engine, exhaust turbine, and pulverised coal water-tube boiler.

Efficient Modern Ships.

"Shipbuilding and Shipping Record", 28th February, 1935.

Some remarkably good results have been obtained by the recently-built vessels for the frozen and chilled meat trade between this country and New Zealand. On trial at light draughts, speeds approaching 18½ knots have been obtained, and on service, high constants have been recorded both on the power, displacement and speed basis, and on the fuel consumption, speed and displacement basis. These results indicate a distinct advance on the records of somewhat similar vessels built a few years ago for the same services. The new vessels carry only the twelve passengers allowed on the cargo ship basis, but are able to show a clean pair of heels to many of the passenger ships trading between the same ports. It is disconcerting to passengers on liners to observe cargo ships forging ahead of their vessels. It is possible, therefore, that the advance made in fast cargo ship design may necessitate the construction of faster passenger ships for the same routes. The suggestion has been made that the discovery of some new type of form more efficient than can be obtained along normal lines of development would be of great benefit to shipbuilding by rendering obsolete much of the existing tonnage. It appears as if the progress which is being made will very soon have the same result without having recourse to abnormality in form or in type of machinery installed. The latest productions are promising in this connection.

A Remarkable Performance.

"The Marine Engineer", March, 1935.

In recent years French naval contractors have earned for themselves quite a reputation for the very high speeds attained by certain of their vessels of war. Some little time ago, for example, the French flotilla leader "Cassard", built and engined by the Ateliers et Chantiers de Bretagne, attained a maximum speed over the measured mile of 43·4 knots, with a mean speed of 42·9 knots. Remarkable as this performance undoubtedly is, it has just been improved upon by the new flotilla leader "Le Terrible", which was built by the Chantiers Navals Francais at their Blainville yard, near Caen. During her speed trials over the Penmarch measured mile the vessel attained the wonderful maximum speed of 45¼ knots on one run, with a mean maximum speed of 45·06 knots. Moreover, during the ship's endurance trials a speed of 43 knots was maintained for eight hours and the machinery was then worked up to allow of a speed of 44 knots being maintained for some time. These results are almost incredible, but we have no reason to doubt their authenticity seeing they have been given fairly extensive publicity, as has been the policy for some little time across the Channel in regard to naval performances.

It is interesting to note that the turbines employed in this French warship are of the Rateau type, built under licence by the company who engined the "Cassard", to which reference was made at the beginning of this comment. It is understood that the power developed on trial was no less than 100,000 s.h.p., which figure again seems most noteworthy. The displacement, incidentally, was 2,830 tons and the length of the ship is 435ft.

The figures which we have quoted undoubtedly give us serious food for thought, and all concerned with these fine performances are deserving of sincere congratulation. Incidentally, wider publicity for the performances of British-built warships would prove distinctly worth while, we suggest, in assisting our own firms in obtaining foreign naval orders.

Testing a 5,500 b.h.p. Sulzer Engine.

"The Motor Ship", April, 1935.

After the completion of the first unit for the "Dorset" at Sulzer Bros.'s works (Fig. 1) in Winterthur, some very complete tests were carried out on the plant.

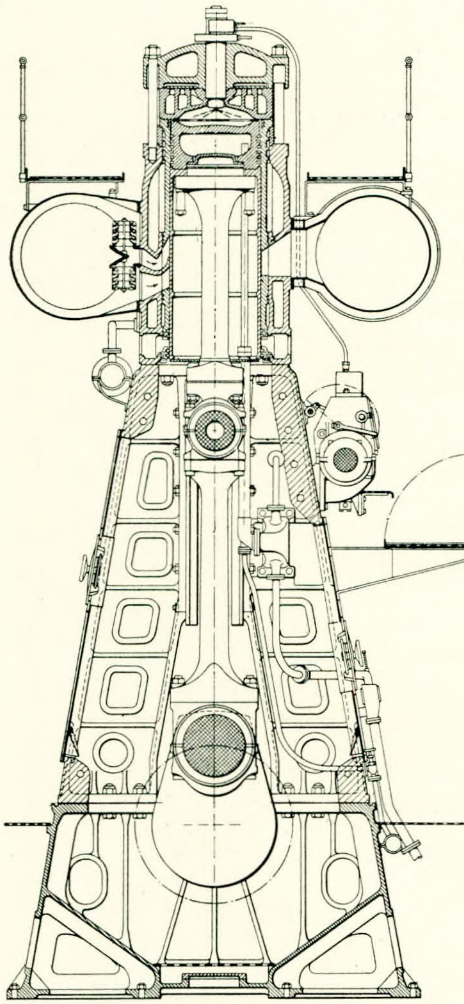


FIG. 1.—Sectional elevation through a cylinder of the engine.

The results of these trials which were made by the builders, were published in this journal*, and they revealed some low figures, the minimum fuel consumption, for instance (at three-quarter-load) being 0·335lb. per b.h.p., and at 10 per cent. overload, 0·338lb. per b.h.p.

In view of the results, it was decided to have independently controlled tests carried out on the second engine for the "Dorset". These were made by Dr. Ing. G. Eichelberg, Professor of the Technical University at Zurich, and they confirmed the builders' figures, whilst giving interesting details relating to efficiency and performance.

Reasons for High Efficiency.

A combination of causes has presumably brought about the low fuel consumption. The combustion chamber form in conjunction with the arrangement of the dispersal of the fuel spray through 10 holes in each atomizer, has apparently allowed of perfect combustion at all loads. The scavenging pump absorbs but 5 per cent. of the power, which is low, and the mechanical efficiency is high. The piston speed of 1,030ft. per minute (5·24 metres per second) is higher than normally adopted with Diesel engines, and partly in consequence of this, the cooling water losses are reduced, representing only 20 per cent. of the heat supplied to the engine.

The liner is uncooled below the scavenging ports giving a reduced heat loss and it is considered that piston ring friction is diminished by decreasing the sliding surfaces of the ring, also the initial tension. This is accomplished by having a special joint on the two upper rings to prevent any blow back of the gases, and allowing the four following rings to be narrower than normal.

The engine is of the two-stroke single-acting Sulzer design with two rows of scavenging ports, admission to the upper row being controlled by automatic disc valves in the scavenging trunk. The eight cylinders have a diameter of 720 mm. (28¼in.) and the piston stroke is 1,250mm. (49in.), the designed output normally, at 126 r.p.m. being 5,500 b.h.p. The stroke-bore ratio is thus approximately 1·74.

Complete details of the design of the engine may be obtained in the issues of "The Motor Ship" to which reference has been made, and only new particulars will now be included. Scavenging air pressure at 2 metres water gauge allows a mean effective pressure in the cylinder of 4·8 kilogs. per sq. cm. There are two double-acting reciprocating scavenge pumps in tandem, direct driven from the crankshaft; they have a bore of 1,660 mm. and a

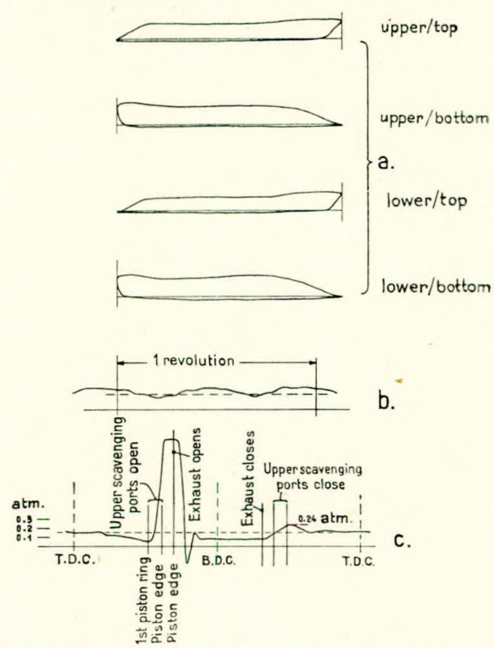


FIG. 3.—Scavenging pressure diagrams at full load.

a. Scavenging pump.
b. Pressure in scavenging receiver.
c. Pressure between non-return valves and upper scavenging ports.

* "The Motor Ship", August, 1934.

stroke of 750 mm. The swept volume is 1·59 times the total volume of the working cylinders. Allowing for the volumetric efficiency of the pumps, the scavenge air volume is 1·35 times that of the working cylinders.

Pressure Fluctuation during the Scavenging Process.

At full load the suction pressure is in the neighbourhood of 180 mm. water gauge, as indicated in Fig. 3. In this illustration the fluctuation of pressure in the scavenging receiver is shown, and

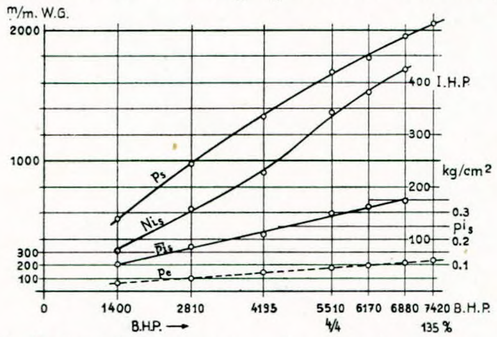


FIG. 4.—Scavenging pressure and scavenging pump power. Ps. scavenging pressure mm. W.G. Ni_s scavenging pump i.h.p. $\bar{pi}_s$ of the scavenging pump kg. per sq. c.m. (referred to the working-cylinder). p_e Exhaust back pressure mm. W.G.

the variation in pressure between the scavenging valves and the ports. At the end of the expansion stroke of the piston, the pressure in the cylinder causes pressures up to 2½ atmospheres in the space

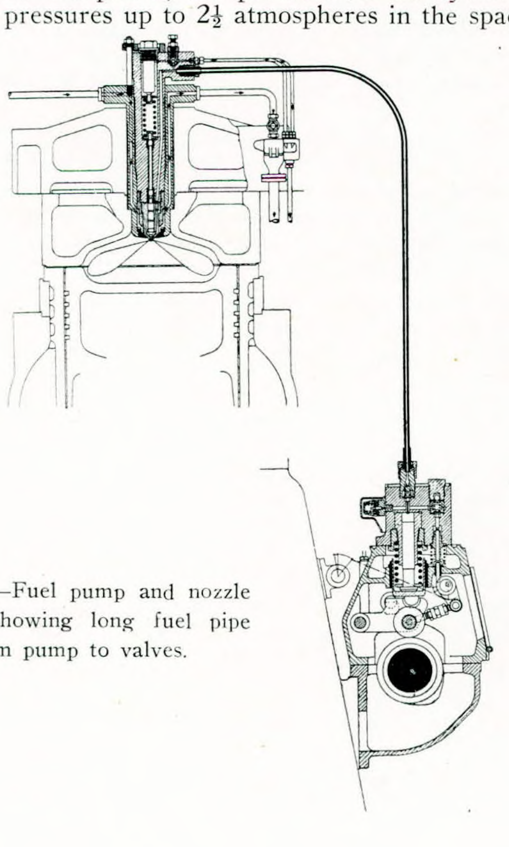


FIG. 5.—Fuel pump and nozzle valve showing long fuel pipe from pump to valves.

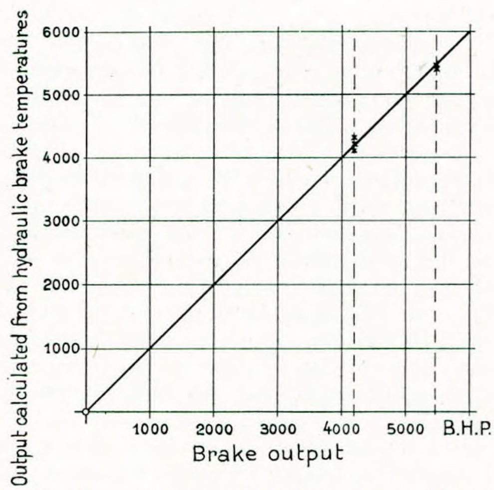


FIG. 6.—Graph showing results of tests to check the output of the brake used during the trials.

between the scavenge valves and charging ports. When the exhaust port is open, this pressure falls quickly, whilst during the admission of scavenging air, the pressure is intermediate, between that of the receiver and the exhaust back pressure. When the exhaust ports are closed by the piston, during the scavenging stroke, there is a rise of pressure in the space due to after-charging.

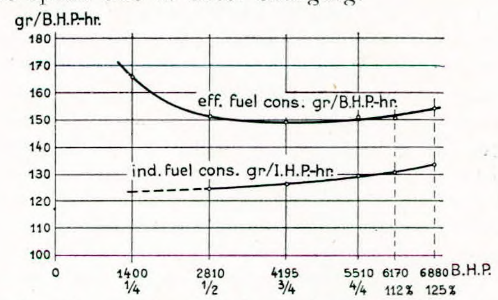


FIG. 7.—Curves showing the fuel consumption results.

Fig. 4 shows the scavenging pressure and the power required to drive the scavenging pumps. It will be observed that at normal output the mean pressure of the scavenging pumps referred to the working cylinder is about ·3 atmosphere/cm. and that the power required to drive them is about 340 i.h.p.

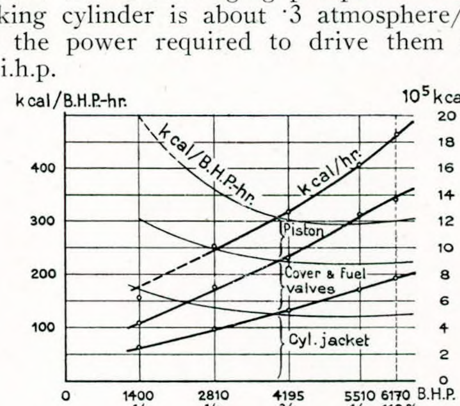


FIG. 8.—Graphs recording the distribution of heat in cooling-water.

The Fuel System.

It was recorded in the description of the engines that one cam is utilized for operating each fuel pump (one per cylinder) both for ahead and astern running. An explanation of this is of interest, since the injection of fuel commences before the piston reaches the top dead centre, and does not end until a considerable crank angle has been passed through, and a cam operating a pump causing direct injection in this way does not lift and fall symmetrically to the dead centre. It cannot, therefore, be employed both for astern and ahead rotation. In this engine use is made of the lag in injection occurring on account of the finite velocity of the propagation of the pressure waves in the fuel piping.

Unlike the arrangement in some Diesel engines where the fuel pumps are located as close as possible to the fuel valves, a long pipe is arranged between each fuel pump and nozzle valve. This has the effect of providing a momentary accumulation of the energy of injection in the pressure wave along the pipe. The pump cams can, therefore, operate somewhat earlier than with a short pipe. It is found that with a pipe 10 metres in length, and a pressure wave, corresponding to that of sound (1,400 metres per second) a lag of injection of 5·4 deg. crank angle is obtained at the normal running speed of the engine (126 r.p.m.).

The lag is so arranged than the end of the delivery of fuel occurs at top dead centre, hence the cams may be constructed symmetrically around the dead centre, and nothing has to be reversed on the fuel side. The direction of rotation of the engine, therefore, depends only on the point of admission of the starting air. The suction stroke of the pump is as rapid as the delivery stroke, and a vacuum is caused in the pump, but this has no effect, on account of the perfect finish possible with modern pump plungers. The delivery of fuel is suspended when the engine is not rotating in the direction indicated by the engine-room telegraph.

The Trials.

The summarized tables give the main details of the results of the trials, which took place on July 24th to 26th, 1934. There was a 12-hours' full load run, then tests at reduced load, next four hours at 12 per cent. overload, and short periods at 25 per cent. and 35 per cent. overloads respectively. Reversing and manœuvring tests were made and

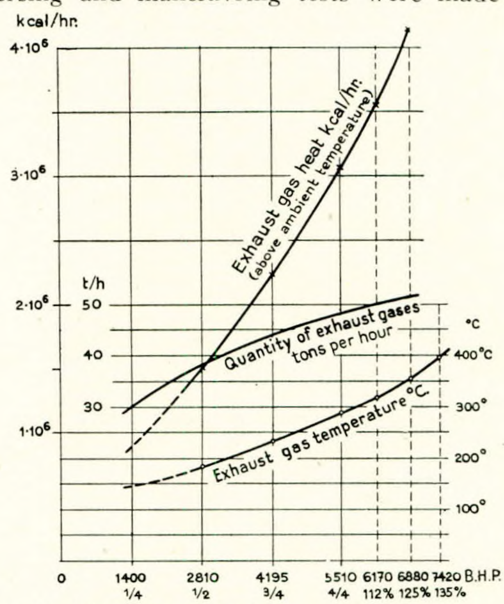


FIG. 9.—Temperatures and heat of exhaust gases.

low speed trials, which, as we have already recorded, showed a somewhat remarkable result. The minimum speed was 16 r.p.m., with all cylinders firing.

It was desired to confirm all the figures in view of the low results anticipated, and besides utilizing an ordinary hydraulic brake, measurements were taken of the extent the water was heated in passing through the brake, and the brake power, therefore, fairly closely ascertained. The brake

TABLE I.—Summary of Results of Test on 5,500 B.H.P. Sulzer Engine.

Load.	1/4	1/2	3/4	4/4	112%	125%	135%
Speed, r.p.m.	80·8	102·0	116·1	126·6	131·6	135·9	139
Brake output, Ne B.H.P.	1,400	2,180	4,195	5,510	6,170	6,880	7,420
Indicat. output Ni B.H.P.	—	3,400	4,950	6,440	7,100	7,940	—
M.E.P., pe kg./cm.²	1·92	3·05	4·00	4·82	5·20	5·61	5·9
M.I.P., pi kg./cm.²	—	3·69	4·72	5·64	5·98	5·48	—
Mech. effic., mech. %	—	82·5	84·7	85·5	86·8	86·2	—
Max. diagr. pressure, kg./cm.²	40·0	50·1	56·3	60·8	61·9	64·3	—
Scavenging pressure, mm. W.G.	550	970	1,330	1,665	1,770	1,940	2,030
Indic. scav. pump output, Ni_{Sp} H.P.	76	157	226	342	380	420	—
$Pi_{scav.\ pump}$ referred to working cylinder ...	0·104	0·170	0·215	0·30	0·32	0·34	—
$P_{fric.} = P_i - P_e - P_i$ scav. pump	—	0·47	0·505	0·52	0·46	0·53	—
Ind. fuel cons. gr./I.H.P.—h.	—	125·0	126·8	129·3	131·0	133·6	—
Eff. fuel cons. gr./B.H.P.—h.	165·4	151·1	149·3	151·4	150·6	154·2	—
Thermal efficiency referred to brake output (Hu = 10,135 cal./kg.) th.	37·7	41·2	41·7	41·2	41·4	40·4	—
Exhaust gas pressure, mm. W.G.	50	95	145	173	189	210	230
Exhaust temp., t_A °C	—	186	234	290	318	356	398
Exhaus. quantity, t/h	30·2	38·4	43·9	48·0	49·95	51·6	—
Exhaust heat, kcal./h	—	$1.51.10^6$	$2.24.10^6$	$3.08.10^6$	$3.56.10^6$	$4.15.10^6$	—
Exhaust heat in % of heat introduced	—	35·0	35·2	36·4	37·8	38·7	—

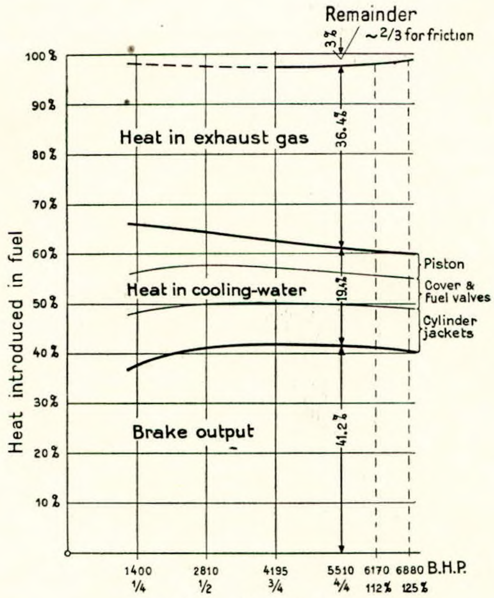


FIG. 10.—Heat balance.

figures were very closely confirmed, as seen in Fig. 6.

The fuel consumption figures were almost precisely the same as those referring to the previous engines, and detailed in the August, 1934, issue of "The Motor Ship". They are published in Table I. The fuel oil was analysed and found to have a lower calorific value of 10,138 K.cals. per kg., and to bring this to a standard basis of 10,000 K.cals. per kg. two grammes per b.h.p.-hour should be added to the consumption. An amount of 1 gr. per b.h.p.-hour should also be added for the power required for driving the lubricating oil pump and the cooling water circulating pumps as they were driven separately by electric motors, and the power needed was 36·5 h.p.

Table III and Fig. 7 give the fuel consumption results, with these modifications, and it will be seen that over a range from half load up to 12 per cent. overload the average fuel consumption is about 0·338lb. per b.h.p.-hour, including the fuel needed for driving the scavenging pump, the circulating fresh water and sea water pumps, and the lubricating oil pump. It may be added that the ships in which this type of engine is installed utilize exhaust

TABLE III.

Fuel consumption results based on oil with a lower calorific value of 10,000 K.cals. per kg. and including power for driving, circulating and lubricating oil pumps.

	Grammes per b.h.p.-hr.	Lb. per b.h.p.-hr.
Half-load	154·1	·34
Three-quarter-load ...	153·3	·336
Full load	154·4	·34
12 per cent. overload ...	153·6	·338

gas boilers, so that the actual fuel consumption from the commercial standpoint represents an even better figure than that recorded in the table.

The reduction in the loss of heat to the walls already referred to was confirmed by the cooling water measurements, Table II, and at full load the

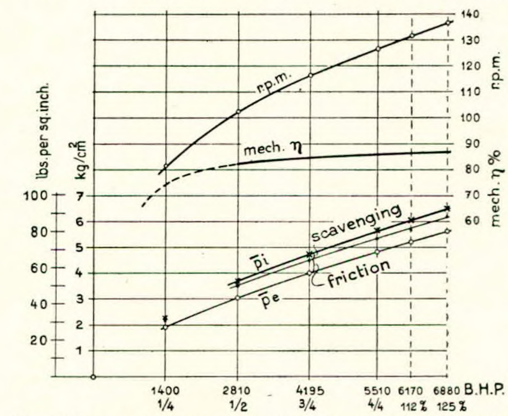


FIG. 11.—Mean pressure, speed and mechanical efficiency.

loss was 300 K.cals. per b.h.p.-hour, or 19 per cent. of the heat introduced. Fig. 8 shows the details of the measurements of the various loads, distributed between liner, cylinder cover and piston.

Particulars of the loss of heat in the exhaust

TABLE II.

Load.		1/4	1/2	3/4	4/4	112%	125%
Cooling-water temp.							
Inlet to jacket	°C.	28·75	29·75	31·5	28·6	30·2	35·75
Inlet to cover	°C.	35·0	37·5	42·0	42·1	45·0	47·0
Outlet from cover	°C.	39·5	43·6	49·5	52·75	56·2	57·5
Inlet to fuel valve	°C.	28·5	30·5	28·5	29·4	29·3	35·0
Outlet from fuel valve	°C.	34·5	36·0	35·3	35·5	34·6	42·3
Outlet to piston	°C.	36·0	42·2	40·0	40·2	42·0	49·0
Inlet from piston	°C.	29·0	31·1	29·0	30·0	29·2	34·2
Outlet from piston leak water ...	°C.	35·3	41·5	39·5	39·4	40·5	48·0
Cooling-water quantity.							
Jacket and cover...	kg/h	38,900	49,800	50,250	51,000	51,600	66,000
Fuel valve	kg/h	2,640	3,080	3,030	5,970	5,400	4,105
Piston	kg/h	25,000	25,840	29,520	33,420	35,200	38,200
Piston leak-water	kg/h	2,400	2,880	2,640	3,960	3,600	3,360
Heat in cooling-water.							
Jacket	...kcal/h	243,000	386,000	528,000	688,000	764,000	743,000
Cover	...kcal/h	175,000	304,000	377,000	543,000	578,000	627,000
Fuel valve...	...kcal/h	15,850	16,900	20,600	36,400	28,300	29,950
Piston	...kcal/h	191,080	317,200	357,200	374,800	489,600	608,200
Total	...kcal/h	624,930	1,024,100	1,282,800	1,642,200	1,859,900	2,008,150
Total % of heat introduced		26·6	23·75	20·2	19·4	19·7	18·7

gases are given in Fig. 9, and the heat balance (Fig. 10) shows that over 41 per cent. of the heat supplied to the fuel is converted into useful energy.

A New Method of Supercharging.

The Wibu System which may be Utilized in Conjunction with other Supercharging Plant.

By Prof. Dr. Ing. LUDWIG EBERMAN.

"The Motor Ship", April, 1935.

So far as the four-cycle engine is concerned, the present trend of development lies in the increase in power with a given cylinder-stroke volume. For this reason, widespread attention is being given to the question of supercharging.

The most common method adopted is the exhaust-gas turbo charging system, but, on account

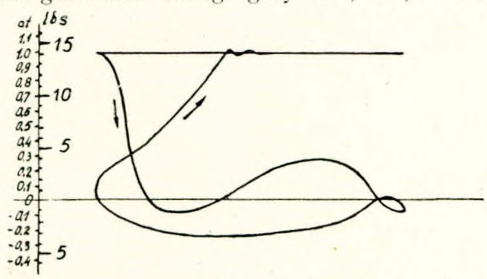


FIG. 1.—Pressure curve in the cylinder during exhaust and suction strokes.

of the relatively high cost, it is more suitable for larger than for small powers.

A method of supercharging, suitable for all powers, should present the following characteristics:—

(1) Low capital cost.

(2) Simplicity.

(3) Equal adaptability for all engine powers.

(4) A supercharging pressure of 0·2 to 0·3 atmospheres.

These conditions appear to have been met with a new supercharging system, developed by two Polish engineers, Adam Wicinski and Jakob Bujak. It is named the Wibu supercharging method and depends upon the use of the mass inertia of the air column in corresponding suction pipes and differs from the previously known methods,* in which the resonant oscillations of the air column are utilized.

Method of Operation.

The operation of the Wibu system is as follows:—In the first portion of the suction stroke the suction valve is slightly opened, and as a result a partial vacuum of about 0·3 atmospheres is created. At about the middle of the suction stroke the suction valve is fully opened, hence the vacuum causes a high air velocity up to 200 metres per second.

Under these conditions the air column in the suction pipe shows a much greater increase in kinetic energy than with a resonant oscillation. During the Wibu supercharging experiments with the object of determining the effect of the utilization of

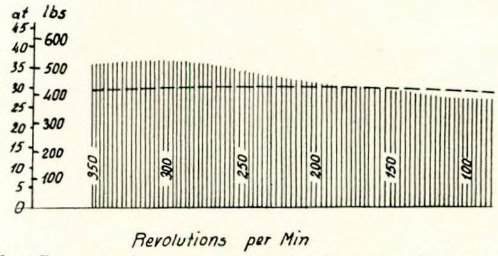


FIG. 2.—Compression pressure at various revolution speeds.
- - - - - without supercharging.

the resonant oscillations, the advantage gained was so small that the system did not appear to enter into practical consideration.

The next experiments were made on a single-cylinder Diesel engine with a cylinder diameter of 300 mm., the piston stroke being 450 mm. and the speed 300 r.p.m. An increase in power from 60 b.h.p. to 75 b.h.p. was obtained and the mean pressure was increased from 5·65 to 7.1 kg. per sq. cm. Fig. 1 shows the variation in pressure in the cylinder and it is seen that compression begins at 0·18 atmosphere. Fig. 2 shows the compression pressure at varying speeds of revolution. The dotted line indicates the pressure without supercharging, and at normal load. Fig. 4 shows the corresponding power, fuel consumption and exhaust-gas temperature for the single-cylinder motor at 300 r.p.m. with and without Wibu pressure-charging.

Results in Service.

After these first experiments several engines were fitted with the Wibu supercharging system. The first was a four-cylinder motor with the same

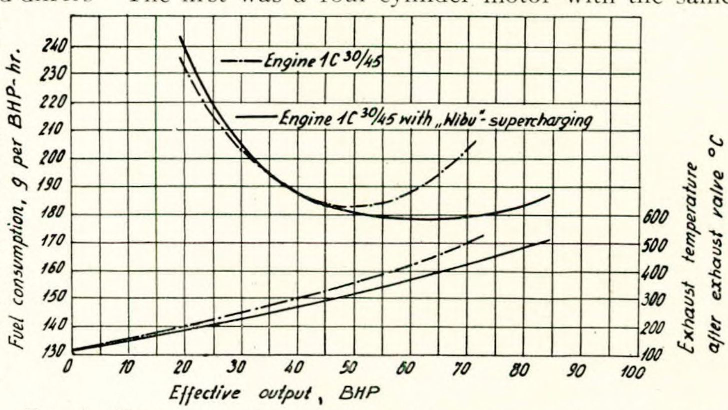


FIG. 3.—Fuel consumption and exhaust gas temperature with and without supercharging.

* Voissel: Resonanzerscheinungen in der Saugleitung von Kompressoren und Gasmotoren Forschungsarbeiten Nr. 106 (1911).

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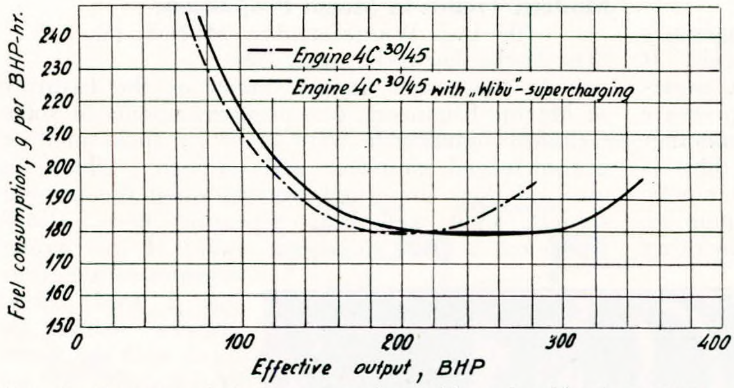


FIG. 4.—Power and fuel consumption with and without pressure charging.

cylinder dimensions and running at the same speed as the experimental engine. An increase of power from 240 b.h.p. to 300 b.h.p. was obtained and the mean effective pressure rose from 5·65 kg. per sq. cm. to 7·1 kg. per sq. cm.

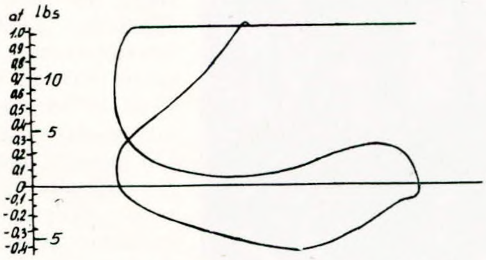


FIG. 5.—Pressures in the cylinder with high degree of supercharging.

Fig. 3 shows the fuel consumption before and after conversion to supercharging.

A three-cylinder motor of another construction was also converted. This had a diameter of 290 mm. and a piston stroke of 400 mm. At 300 r.p.m. it gave an increase of power from 150 b.h.p. to 188 b.h.p. with corresponding mean pressures to those stated previously. The engine works 24-hours daily under full load, and the economy in fuel consumption over a period of eight months was such as to pay off the capital cost of the conversion.

A six-cylinder engine with a diameter of 180 mm. and a piston stroke of 250 mm., running at 800 r.p.m., showed an increase of power from 180 b.h.p. to 240 b.h.p.

Simultaneously with the development of the Wibu system, further tests were carried out with the object of obtaining a higher increase in power than has previously been reached. The experiments were made in the works of Spolka akc. Wielkich Preców i Zakladów Ostrowvekich, at Warsaw, and the results were favourable, utilizing a supercharging pressure of 0·3 atmospheres. Fig. 5 shows the pressure curve in the cylinder and Fig. 6 that in the suction pipe.

In comparison with other supercharging systems, the cost of conversion of existing engines is small and the amount of work insignificant. It is only a question of changing the inlet valve cam and the construction of corresponding suction pipes, the dimensions of which are deter-

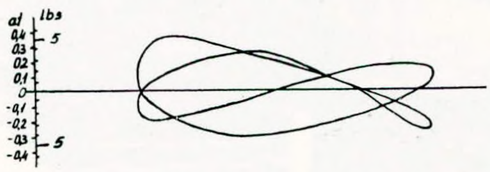


FIG. 6.—Pressure in the suction pipe with high degree of supercharging.

mined on a basis of empirical formulæ, the results of experimental investigations. Even smaller than the cost of conversion is the additional cost of the Wibu supercharging system when building new engines.

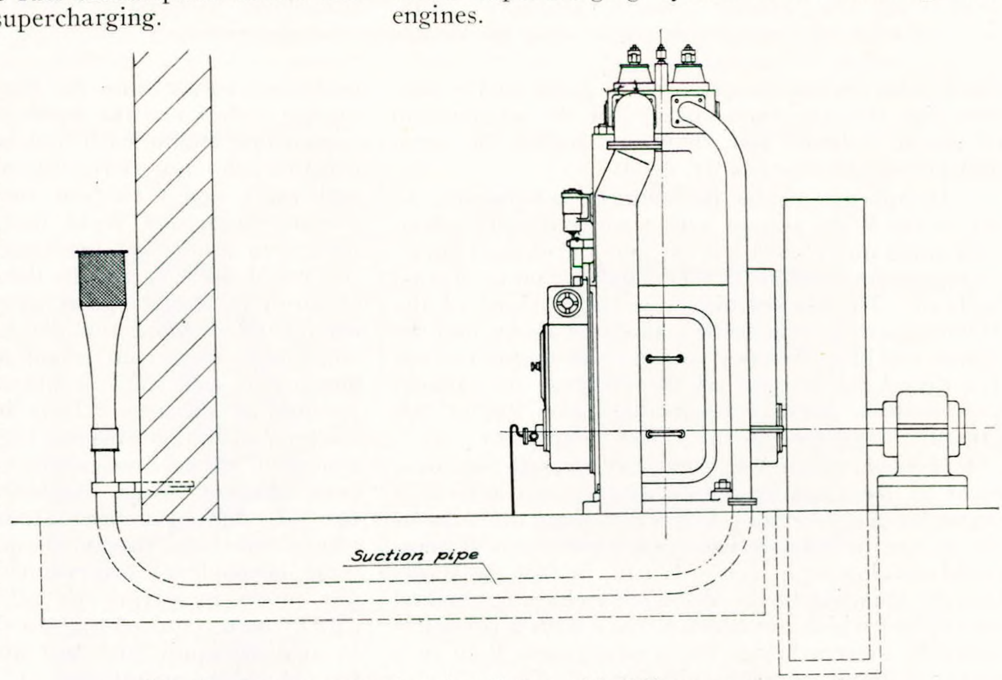
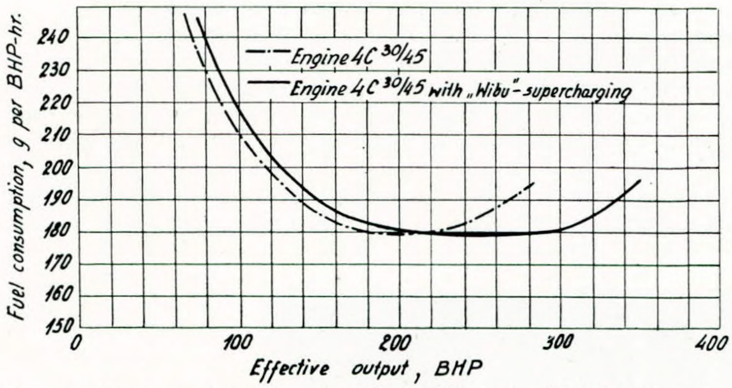


FIG. 7.—Arrangement of an installation with Wibu pressure charging.



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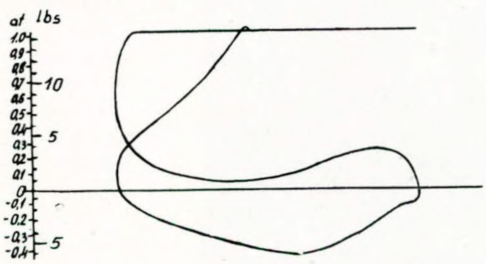

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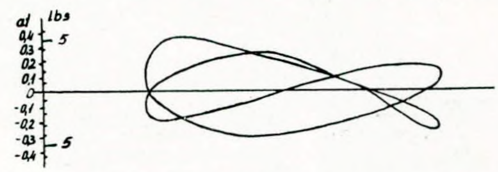

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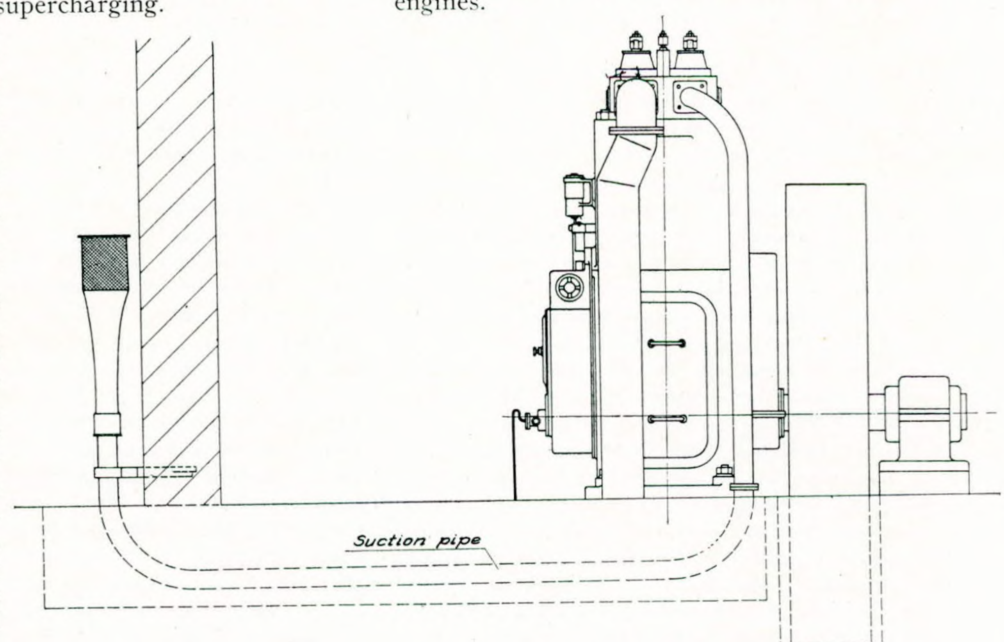


FIG. 7.—Arrangement of an installation with Wibu pressure charging.

Combination with Existing Systems.

The Wibu system can, however, advantageously be combined with other supercharging methods. It is to be noted that the efficiency of the Wibu supercharging method is greater the higher the pressure of the air supplied from the blower of another system with which it may be combined. This is due to the fact, on the one side, that the kinetic energy of the air column in the suction pipe increases with the weight of the air, and, on the other hand, with higher air pressure in front of the suction pipe there is more energy for the acceleration of the air column, and, therefore, higher air speed and greater kinetic energy result.

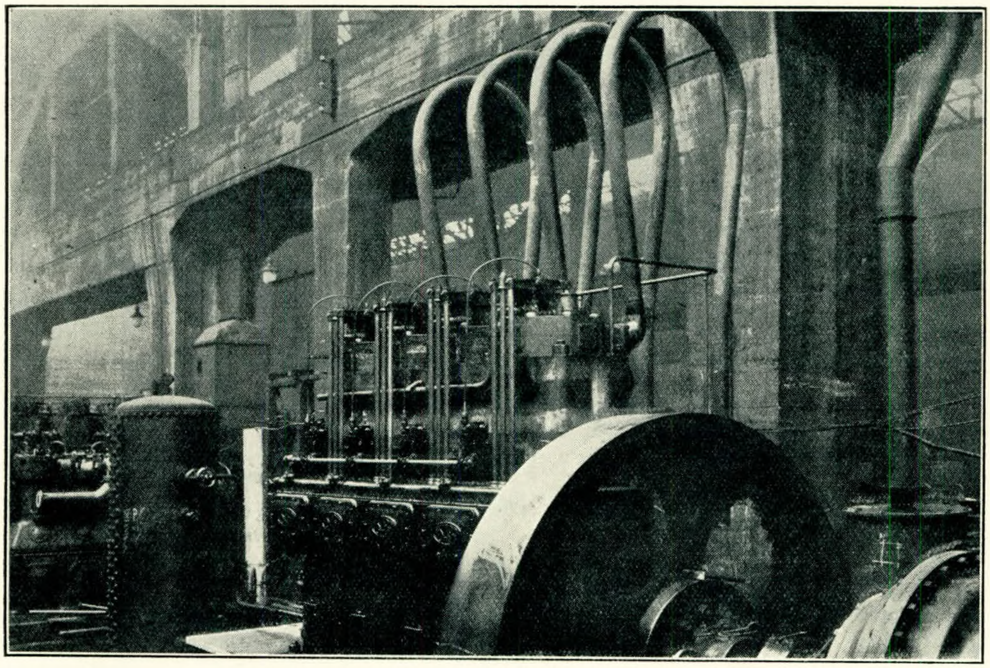


FIG. 8.—A four-cylinder engine using the Wibu supercharging system.

It appears to be particularly satisfactory to utilize the Wibu system with a supercharging plant, combining an exhaust-gas turbine and turbo blower, since an improvement in fuel consumption is effected by both. The blower improves the working of the Wibu supercharging system, as stated above, and the power and the efficiency of the exhaust-gas turbine are raised on account of the increase of exhaust gas content, and, consequently, the higher air pressure developed in the turbo compressor.

It follows, therefore, that the overall improvement in operation by a combined system with both supercharging methods is greater than the sum of the power increase which each of the two systems could develop separately. It can, in fact, be stated that the adoption of the Wibu supercharging method on engines which are fitted already with a pressure-charging system brings more advantages than to a motor without pressure charging.

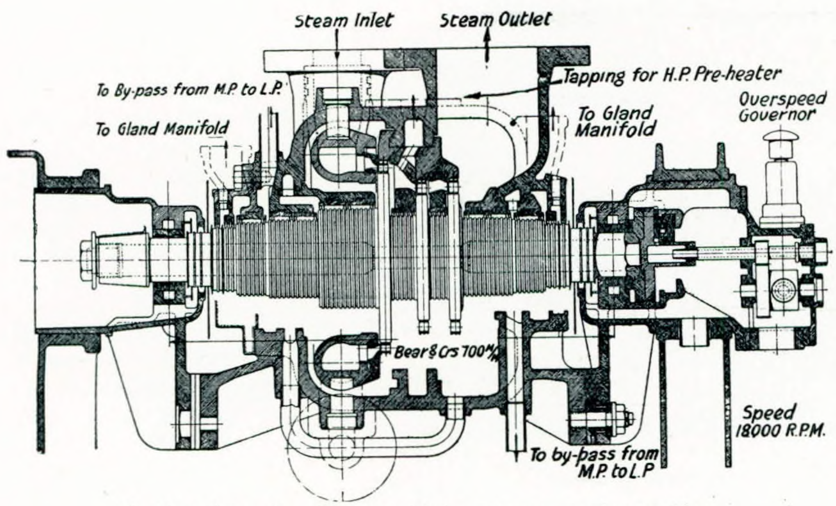
Modern Trends in Steam Propulsion.

By D. S. WHITEFORD, B.Sc., M.I.N.A., Etc.

"The Marine Engineer", April, 1935.

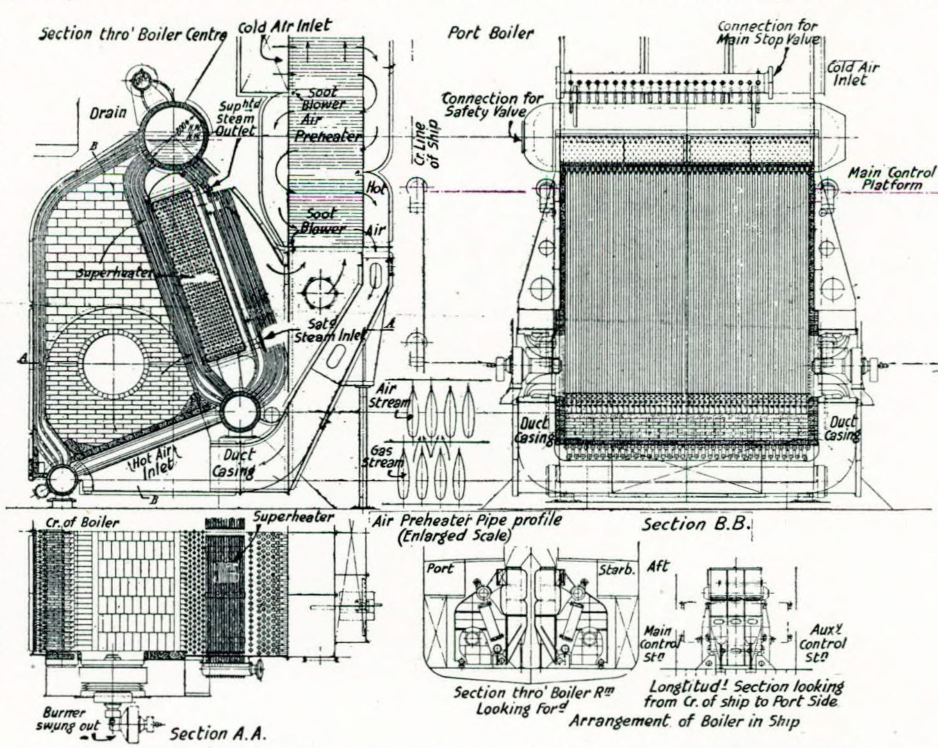
At the recent annual dinner of the Institute of Marine Engineers, two pronouncements of some technical significance were made by spokesmen of acknowledged eminence and authority, which, in view of their important bearing upon the present subject, may well bear repeating here. Lord Essendon, a great shipowner, said: "I was wrong when a little time ago I said Diesel engines would do away with the steam engine. All can see to-day that with the improvement in turbines and auxiliaries coal is moving forward and can rival oil". Lord Weir, a shrewd engineer, said, at the same function, "Technical developments have saved British yards. Coal is as good as oil to-day—at sea. Oil is easy, but coal is difficult to use; new devices and systems will alter that position".

This question of the relative merits of coal and oil fuel is one which has given rise to much controversy ever since the burning of liquid fuel in marine boilers and the development of the internal-combustion engine each first began to merit serious consideration for mercantile shipping some twenty odd years ago. Tactical factors with which the present discussion is in no way concerned have tended to render oil practically the universal fuel for naval vessels, and in the larger high-powered steam-driven luxury passenger liners all the operating factors in which fuel plays any part are so overwhelmingly in favour of oil that a return to coal-burning in such ships is difficult if not actually impossible to visualise. It is indeed in such ships, working with high-pressure high-temperature steam, generated in oil-fired water-tube boilers, that the most efficient steam machinery performances so far recorded have been attained, and it must be recognised that without oil such results would have been incapable of achievement. Having conceded this much in favour of oil, it should however, equally be recognised that the disadvantages of solid as against liquid fuel, are mainly concerned with the physical operations of handling, firstly in



Section through a Wagner high-pressure marine turbine.

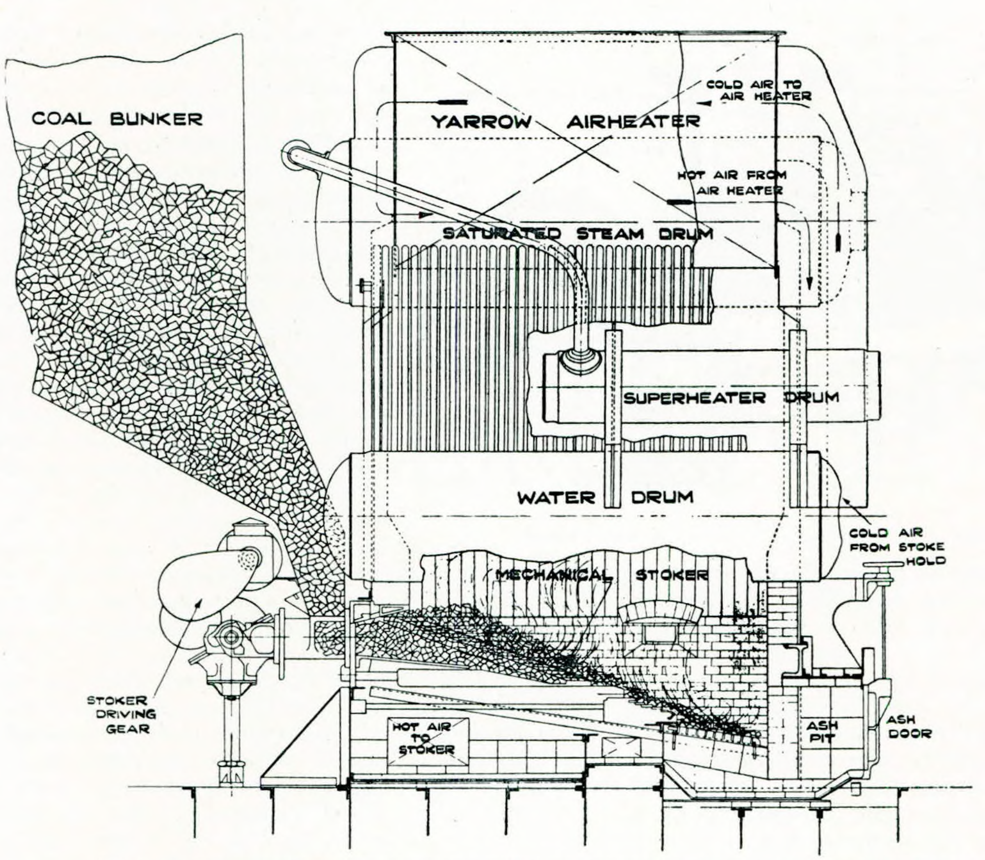
bunkering, and secondly in firing. It has for long been appreciated that the hand-firing of coal in marine boilers is a crude and inefficient process, and yet it must surely be something of a paradox that in such a mechanical age as the present well-nigh 99 per cent. of the world's coal-burning tonnage still relies upon the century-old shovel feed. Efforts have been made, mainly since the war, to substitute for hand-firing alternative mechanical systems. Of these, pulverised coal-burning on a variety of systems was vigorously applied on several ships a few years ago, but, it must be confessed, with but little success. The majority of the pioneer installations were unfortunate in being associated with multiple furnace Scotch boilers, where restrictions in the length of the flame, inadequate combustion chamber volume, and the difficulties of obtaining balanced distribution of the coal dust fuel, proved to be problems of a much more complex character than would have been encountered had the more suitable unit furnace of the water-tube boiler been selected as the starting point for what at one time appeared to be a most promising coal development.



Arrangement of Wagner high-pressure water-tube boilers. Generating plant of somewhat similar type is to be used in the new N.D.L. turbo-electric liner "Scharnhorst".

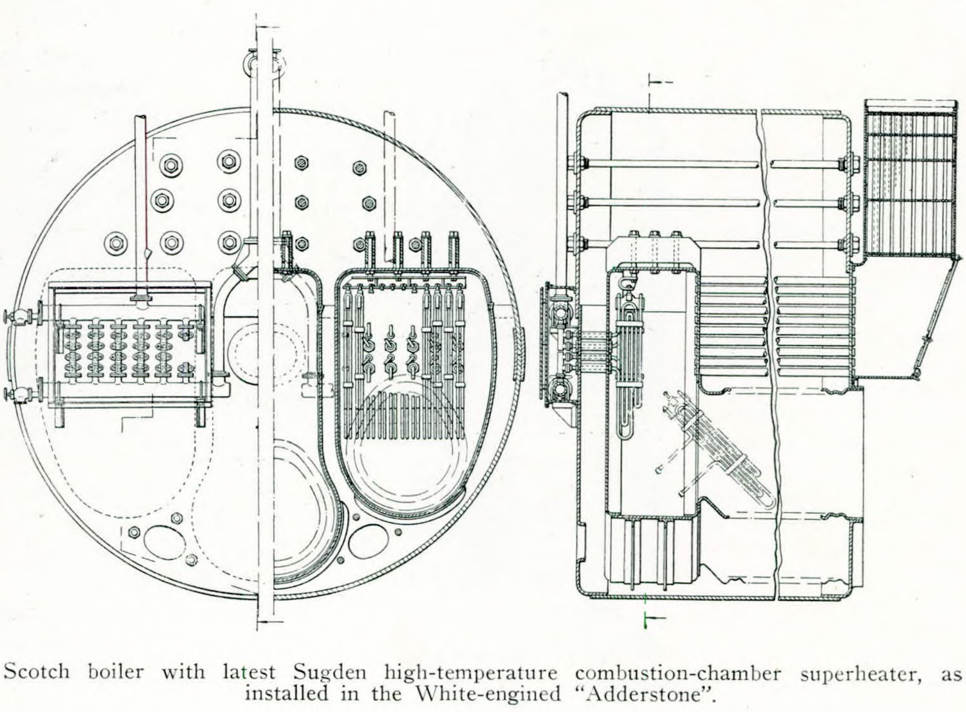
Mechanical Stoking.

In another and more successful direction, hand coal-firing has been superseded in an increasing number of ships by what may be collectively described as mechanical stoking systems, including chain grate and multiple retort types. Although it was at one time held to be practically impossible in a rolling ship to maintain the even distribution of the fire over the grate surface requisite for the attainment of the same satisfactory performance as was obtained with shore station mechanical stokers, this difficulty would appear to have been now overcome. In this connection the progressive policy of the L.M.S. Railway Co. may be noted. The first of their Irish cross-Channel steamers to be fitted with mechanical stokers was the "Duke of Lancaster", in which the existing boilers were converted. In this conversion it was found that the output of the boilers could be maintained at 25 per cent. in excess of the rating possible with hand-firing, and with considerable increase in boiler efficiency. So encouraging were these results that mechanical stokers were incorporated in the new "Princess Maud", placed on the Larne-Stranraer service about a year ago. In both these ships the coal is fed by hand to the stoker-hoppers, but in the latest L.M.S. steamship just launched—



Yarrow three-drum boiler with Yarrow integral superheater, airheater, and Taylor retort-type stoker. Equipment of this type is installed in the latest Southern Railway train ferries.

the "Duke of York", for the Belfast-Heysham service—the mechanical stokers will be fed direct from self-trimming bunkers with the total elimination of manual labour. In the Great Western Railway Co.'s Fishguard - Waterford cross-Channel steamship "Great Western", delivered early last year, mechanical stokers were also fitted in the Babcock & Wilcox watertube boilers. The Southern Railway Co., in their three new cross-Channel train ferry steamers recently completed, have also adopted mechanical stokers in the Yarrow water-tube boilers. The ultimate decision to fit mechanical stokers in these train ferry ships was only arrived at after careful consideration of a number of alternative machinery types, and the fact that Kent coal—mined within a few miles of the Dover terminal—was available was no doubt a powerful economic factor in favour of coal. In the five 10,000-ton Canadian Pacific "Beaver" class ships, as well as in about thirty steamships belonging to the Dutch K.P.M. Line, mechanical stokers have already proved themselves in all weathers over a period of years. The accumulation of operating experience and the evolution of improved stoker types, as well as con-



Scotch boiler with latest Sugden high-temperature combustion-chamber superheater, as installed in the White-engined "Adderstone".

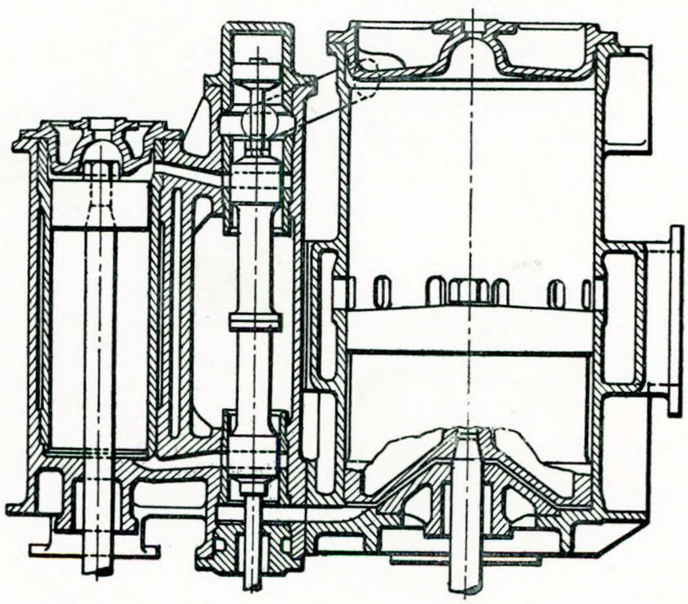
centration upon the design of self-trimming bunker compartments, must tend to widen the potential scope of mechanical coal-firing, even to the extent of embracing a large volume of tonnage of moderate size which is meantime oil-burning. By the greater economic efficiency thus attainable, this promising development must inevitably strengthen the competitive position of the marine steam engine, particularly where native coal supplies are cheap and abundant.

When the history of the past ten years of marine steam engine progress comes to be written, it seems safe to say that emphasis must undoubtedly be given to the great improvement effected in the design and output of marine boilers. In the well-tried Scotch boiler, which is still preferred for the majority of medium-sized steamers, constructional considerations limit the safe working pressure to under 300lb. per sq. in. Improved designs of superheater have now made it possible to attain steam temperatures of the order of 750° F. without difficulty, and this development combined with greater attention to insulation, feed-heating, and preheating of the combustion air have raised the average efficiency of this boiler type by as much as 15 to 20 per cent. since the war.

Howden-Johnson Boiler.

One of the most important departures from standard Scotch boiler design is the Howden-Johnson boiler, special features of which are the single dry back combustion chamber, fitted with a water wall of curved tubes, designed to promote rapid circulation, and the surrounding of the entire boiler with an enveloping air casing which serves the dual purpose of insulating the boiler and heating the combustion air. For a given steam output it is claimed that the weight is 33 per cent. less and the first cost 20 per cent. less than that of the normal Scotch boiler. while its evaporative efficiency is also substantially higher at 88 per cent. Three Howden-Johnson boilers have been fitted in the new Booth liner "Crispin"*, now completing at Birkenhead.

The inevitable trend towards higher working pressures has been responsible for a widespread increase in the adoption of water-tube boilers with pressures of the order of 400 to 450lb. per sq. in. and for the powering of the largest ships of the express liner class, this type of boiler has been firmly entrenched for several years past. In several merchant ships such boilers have been fitted with individual outputs equivalent to 7,000 s.h.p. and more and with efficiencies of the order of 88 per cent. Remarkable progress has recently been made with new types of high rating steam generators, and not only have these tended greatly to enhance the competitive position of steam, but the indications of contemporary developments actually suggest that we are as yet merely on the threshold of this highly promising phase. The recent re-engining of the Royal Mail motor passenger liners "Asturias" and "Alcantara", with single-reduction geared turbines taking steam from Johnson two-drum water-tube boilers, with a power output 75 per cent. greater than the original Diesel machinery and within the existing machinery space bulkheads, is merely one example of the possibilities opened up by this new steam technique. Two only of the three Johnson boilers fitted in each ship are adequate, under normal service conditions, the steam pressure being 425lb. per sq. in., and the temperature 700° F. A somewhat parallel steam generator development is seen in the new Norddeutscher-Lloyd 21-knot turbo-electric liner "Scharnhorst", now completing at Bremen, in which the maximum power of 32,000 s.h.p. will be derived from four Wagner-Bauer drum-type water-tube boilers. Here the working pressure is rather higher at 720lb. per sq. in., and the steam temperature is 878° F. In the sister ship, building at the Blohm & Voss Hamburg yard, the boiler plant comprises four Benson-type tubular boilers, working at 1,150lb. per sq. in. The Benson boiler is alone among the super-pressure tubular-type boilers so far proposed or developed, which has behind it several years' sea experience in ocean-going tonnage. The 3,200lb. per sq. in. Benson unit in the Hamburg-Amerika ship "Uckermark" has now been on service almost five years, and although some inevitable "teething" troubles were encountered at the outset, and despite the fact that the working pressure has been substantially reduced, this boiler has definitely established itself as being wholly suitable for marine service.

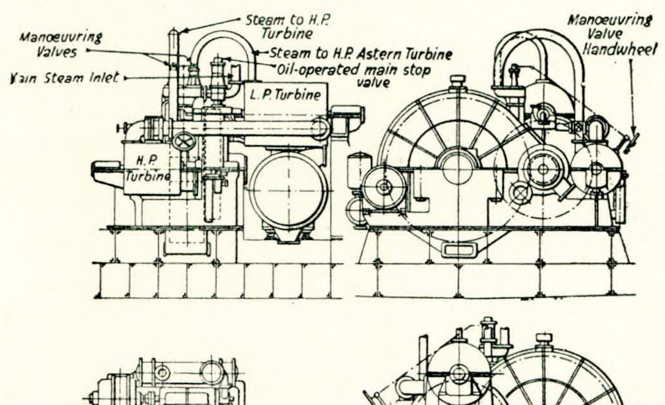


Section through the Fridrikstaad steam "motor", a double-compound high-economy reciprocator which is typical of modern practice.

The first installation of the Sulzer single-tube

* See descriptive article in this issue.—Ed., *M.E.*

boiler or steam generator, on a sea-going ship, will shortly be made on the thirteen-year-old Rotterdam-Lloyd liner "Kertosono". A new primary pressure steam turbine will be added to the existing double-reduction geared steam-turbine plant. This primary turbine will take steam from the Sulzer oil-fired boiler at 882lb. per sq. in. pressure and 707° F. temperature, and will exhaust to the existing high-pressure turbine. These alterations will raise the power output from 4,500 s.h.p. to 5,800 s.h.p., and the sea speed, after hull lengthening, from 13 knots to 15 knots. The new Sulzer boiler, it may be noted, will do the work previously done by four Scotch boilers, the fifth of the original Scotch boilers (converted to oil-firing) being retained for supply-



Two arrangements of Brown-Boveri marine Turbloc machinery for vessels of moderate power. The lower layout is intended for vessels up to about 10,000 s.h.p. The manner of driving the dynamo and other auxiliaries will be noted.

ing steam to the auxiliaries.

In yet another type of tubular boiler, the Velox steam generator, the principle of supercharging is used by means of which combustion takes place under a pressure far exceeding that used in normal practice. The highest steam pressures for which existing materials are suitable can be adopted, but for marine purposes the conservative pressure of 450lb. per sq. in. appears likely to be favoured. This boiler is extremely light and compact, and, as steam can be raised from cold in five minutes, it is obviously suited to naval requirements. In this connection it should be recorded that a Velox generator has already been ordered by the Admiralty from Richardsons, Westgarth & Co., Ltd., representing the sponsors, Brown, Boveri & Co., Ltd., of Baden, Switzerland. Space does not permit of reference being made in detail to such other special boiler types as the La Mont, Loeffler, Atmos and Zoelly generators—this branch of the subject, in fact, merits a separate and detailed treatment of its own.

When we come to the question of steam machinery, contemporary progress is so many sided as to render it possible merely to touch the fringes of the subject in what follows here. Primarily, it should be noted that according to the latest Lloyd's Register statistics, the steam reciprocating engine is still responsible for the propulsion of 69·5 per cent. of the total world tonnage—evidence sufficient in itself to show that this tried and trusted engine type is still very much alive. The development of satisfactory methods of mechanically-combining steam reciprocating machinery with an exhaust turbine, on a single line of shafting, first practically demonstrated in 1926 on the German Bauer-Wach-engined trawler "Sirius", of 750 s.h.p., has done more to perpetuate the popularity of the steam reciprocator than any other single improvement for many years past. The success of the Bauer-Wach exhaust turbine system is largely attributable to the Vulcan hydraulic coupling, which provides the most satisfactory means, so far evolved, of connecting the high-speed exhaust turbine to the slow-speed reciprocator. When fitted to an existing reciprocating engine the power output can be increased by as much as 25 to 30 per cent. or, alternatively, at the normal original output, a reduction of fuel consumption of the order of 25 per cent. may be obtained. The Clan Line, the first British shipping company to have the enterprise and courage to adopt the Bauer-Wach system, have already twenty-two ships so fitted, and have recently ordered a further 10,000-ton, 7,000 s.h.p. twin-screw vessel, with similar exhaust turbine drive. Upwards of 300 Bauer-Wach exhaust turbine sets have already been placed in service or are on order, and the success of the pioneer turbo-compound installation in the British trawler "Kingston Cornelian", has been followed by orders for four similar installations.

The White combination system, in which a high-speed reciprocating engine and its exhaust turbine are each geared to the line shafting, has been outstandingly successful in the "Adderstone", and a further small installation of about 500 equivalent i.h.p., is now on order. In the Gotaverken system the exhaust turbine is coupled to a compressor, in which the h.p. exhaust steam has its pressure and temperature raised, and thereafter does added useful work in the reciprocator. This system has also met with substantial success, and has been fitted to a number of British ships. In the Metropolitan-Vickers and British-Thomson Houston exhaust turbine systems the turbine generates electrical energy which is applied to the line shafting through an electric motor, whose armature is mounted on the shafting direct. Of exhaust turbine systems in general there are at least eight separate types, the distinctive features of which are mainly concerned with the method of transmitting the energy of the turbine to the propeller.

Of improved pure steam reciprocating engines, one of the most interesting is the British Stephens tri-compound engine, comprising one h.p. cylinder between two l.p. uniflow cylinders. The h.p. cylinder is fitted with independent steam and exhaust valves of Andrews & Cameron patent quadruple-opening balanced type, using superheated steam and all valves are hydraulically operated. In a 2,000 i.h.p. engine the power absorbed by the valve gear is only 1·3 h.p., and the overall mechanical efficiency of the engine is 96 per cent. Electrically-welded steel sections are used for the columns and framework of this engine, which is an extremely strong, compact and light unit. In the 12,500-ton 11-knot steamship "Loch Ranza", recently fitted with a 2,000-i.h.p. Stephen engine, a specific fuel consumption of 1·13lb. Virginian coal has been recorded on service, the equivalent coal coefficient being 31,000.

The performance of steam reciprocating engines in general has been greatly improved during the last year or two by a variety of means. Of these, the most important are the use of superheated steam; improved insulation of steam pipes, engines and boilers; the use of patent valves and valve gear; better distribution of steam with resultant reduction in receiver losses; higher feed-heating and combustion air temperatures, etc. Apart entirely from the incorporation of these principles in new ships, numerous existing vessels have had their machinery converted or rejuvenated on these general lines. In certain cases the h.p. cylinder has been replaced with a new cylinder fitted with cam-operated independent steam and exhaust valves, and this, with the addition of superheating and other associated modifications, has resulted in reductions in fuel consumption of the order of 20 per cent. or more. In one British cargo carrying fleet alone, thirty-two ships have had their steam reciprocating machinery modernised on these lines at an expenditure reported to be in the vicinity of £150,000. Whether in new or existing machinery installations, it no longer admits of any doubt that the incorporation of these technical improvements is justified on the highest economic grounds. Before leaving the steam reciprocating engine it may be appropriate here to refer to what is probably one of the most meritorious performances ever recorded with this type of engine. With a deadweight of 7,075 tons and on the 6,000 odd miles run from Cardiff to Buenos Aires, the Isherwood-Arcform steamer "Arcgow" made a non-stop run last year on one boiler, at an average speed of 8·5 knots, on a daily coal consumption of only 9·26 tons. This remarkable result was achieved with a "straight" triple-expansion reciprocator, using superheated steam, the only material departure from the orthodox being the provision of Andrews & Cameron separate cam-operated steam and exhaust valves.

The remarkable success which has attended the introduction of the exhaust turbine system during the last few years has tended to direct efforts towards the development of a simple form of geared steam turbine suitable for medium power moderate speed cargo steamships. Such a tendency has been encouraged by the belief held in certain quarters, and no doubt properly so, that a well-designed pure geared steam turbine installation of, say, 1,500 to 2,500 s.h.p., should show a lower specific fuel consumption than a reciprocating engine alone, or combined with an exhaust turbine of equivalent aggregate power. Proceeding on these lines the Parsons Marine Steam Turbine Co. have recently developed their Simplex unit double-reduction geared turbine, and have completed a demonstration unit of 2,000 to 2,250 s.h.p., at 80 r.p.m. This unit comprises an h.p. and l.p. ahead and astern turbine each connected through flexible couplings to opposite ends of a common primary pinion; while a secondary pinion on the same shaft as the primary wheel drives the main wheel on the propeller shaft. The steam conditions are 270lb. per sq. in. pressure with 750° F. temperature, and at 2,000 s.h.p. output the steam consumption is 8¾lb. per s.h.p. per hour, or equivalent to 7¾lb. per i.h.p. For a vessel of 8,500 tons deadweight such a plant, working in conjunction with Howden-Johnson boilers, is estimated to be capable of realising a fuel consumption of less than 1lb. of coal per i.h.p. for all purposes. On figures such as these it must be obvious that coal-fired steam machinery will not be lightly displaced for the propulsion of moderate power medium speed cargo ships. A broadly similar simplified geared turbine installation for equivalent power range has also been developed under the name of the Turbloc system by Brown-Boveri & Co., of Baden, Switzerland.

For the propulsion of large passenger liners of the luxury type at high sea speeds, steam machinery, having regard to the enormous strides recently made, and yet to be made with modern steam generators, appears likely to hold the field successfully against the reciprocating Diesel engine. In such ships, silent, smooth-running machinery, with guaranteed freedom from disturbing hull vibration, is increasingly becoming a *sine qua non*. The attainment of these *desiderata* with internal-combustion machinery is not always a simple problem. The steam turbine, whether mechanically geared, as in the "Queen Mary" and "Empress of Britain", "Bremen", "Europa", "Rex" and "Conte di Savoia", or in association with electric drive as in the "Normandie", "Strathnaver", "Strathaird", "Viceroy of India", "Monarch of Bermuda" and "Queen of Bermuda", and others, is at once a highly efficient and wonderfully dependable instrument of propulsion. That such ships should—for the physical reasons already outlined—burn liquid instead of solid fuel does not render them any the less the glorious stronghold of steam.

The future of coal—whether as a solid fuel or colloidal with oil—and of the steam engine, need in no wise be regarded with pessimism. Ten years ago the performance standards being regularly achieved to-day would have been regarded as beyond practical attainment, and he would be bold indeed who would say that technical ingenuity has in any wise expended itself. To the informed student of contemporary development the indications must indeed point to a wholly contrary conclusion.

Explosion from a Cast Iron Stop Valve Chest in the Steamship "City of Paris".

Abstract of Report of Preliminary Inquiry.

Mercantile Marine Department,
Board of Trade,
18th January, 1935.

Date and place of the explosion.—The explosion occurred at 1.50 p.m. on the 5th December, 1934, in the steamship "City of Paris", the vessel being at sea and about 8 miles from Plymouth on a voyage from Plymouth to Antwerp.

Description and principal dimensions of the stop valve chest.—The stop valve was the auxiliary steam supply portion of a combined main and auxiliary steam stop valve chest fitted to the forward port boiler and of the form indicated in Fig. 2. The main steam portion served to control the passage of steam from the boiler to the inlet headers of the superheater, the oil fuel installation and to the inlet side of the auxiliary stop valve. The chest and covers were of cast iron, the valves and seats of "Monel" metal and the spindles of naval bronze. The main valve was of screw lift type and the auxiliary valve of non-return type, each $5\frac{1}{4}$in. diameter. The inlet branch from the boiler was $5\frac{1}{2}$in. diameter and the delivery branch of the auxiliary stop valve was 5in. diameter. The thickness of the chest was $\frac{15}{16}$in. generally and $\frac{7}{8}$in. at the auxiliary delivery branch. The valve covers were $11\frac{1}{2}$in. diameter, $1\frac{1}{16}$in. thick and each was secured by 8 studs $\frac{3}{4}$in. diameter on a pitch circle $9\frac{3}{4}$in. diameter. The valve spindles were $1\frac{1}{2}$in. diameter, screwed 6 threads per inch and fitted to external bridges each secured by two columns screwed $1\frac{1}{8}$in. diameter.

Nature of the explosion.—The explosion was of a violent nature. The valve cover was fractured around the bossed portion forming the stuffing box and radially, in five places, through the jointing flange; the centre portion of the cover was blown out, allowing the steam from the four boilers which were at work to escape, by way of the auxiliary steam pipes, through the orifice formed. Owing to the escaping steam the stokehold was untenable, the ship's machinery was stopped owing to the shutting of the boiler stop valves by the engineers in their attempts to rescue two of the personnel who were known to have been trapped in the stokehold, and the ship was put in jeopardy by reason of the heavy weather and her proximity to the shore.

Cause of the explosion.—The explosion was caused by water hammer action.

General remarks.—The steamship "City of Paris" was built in 1921 by Messrs. Swan, Hunter and Wigham Richardson, Limited, Newcastle-on-Tyne, and fitted with machinery by the Wallsend Slipway and Engineering Company, Limited, Wallsend-on-Tyne. She was classed ✠ 100A1 in Lloyd's Register of Shipping and held a passenger certificate issued by the Board of Trade. The propelling machinery comprised one set of turbines coupled by double reduction gearing to a single screw. Steam for the propelling machinery and the auxiliary services was supplied by five single-ended marine multitubular boilers working at a pressure of 225lb. per sq. in. Superheaters of the smoke tube type were fitted to each boiler to superheat the steam for the main turbines to a temperature of about 560° F., but the steam supplied to the auxiliary services was not superheated. The steam pipe arrangements fitted when the ship was built provided main steam pipes from each of the five boilers and auxiliary steam pipes taking steam from the two forward boilers only. In 1929 additional steam pipes were fitted to connect the three after boilers to the previously existing auxiliary steam pipe range in order that any of the five boilers could be used to supply steam for auxiliary purposes. The arrangements of the auxiliary steam pipes from 1929 were as indicated in Fig. 1.

In January, 1934, the owners' superintendent engineer considered the steam pipe arrangements in relation to the manner in which it was usual to operate the boilers and in order to reduce the risk of water hammer action he arranged for the fitting of isolating valves at the junctions of the main steam pipes from the forward boilers, and this was done. The provision of isolating valves at the junctions of the auxiliary steam pipes from the forward boilers was also given consideration, but, having regard to the short length of "dead end", which would be formed by one of the forward boilers being shut off, it was considered they were not necessary. The original designs of the steam pipe arrangements were approved by the Board of Trade and when the alterations were made in 1929 and in 1934 the owners submitted designs for approval and had the alterations completed to the satisfaction of the surveyors. Prior to the explosion the ship had been 13 years in service and there had been no damage caused by water hammer action in any part of the steam pipe systems and no sign of such action had been observed by the engine room staff.

Leaving Bombay on the 12th November, 1934, the vessel steamed to Marseilles by using four boilers, the forward port boiler not being in use. From the 28th November, 1934, to the 5th December, 1934, while steaming from Marseilles to Plymouth, only three boilers were used, the forward port and the after starboard boilers being shut off and not under steam.

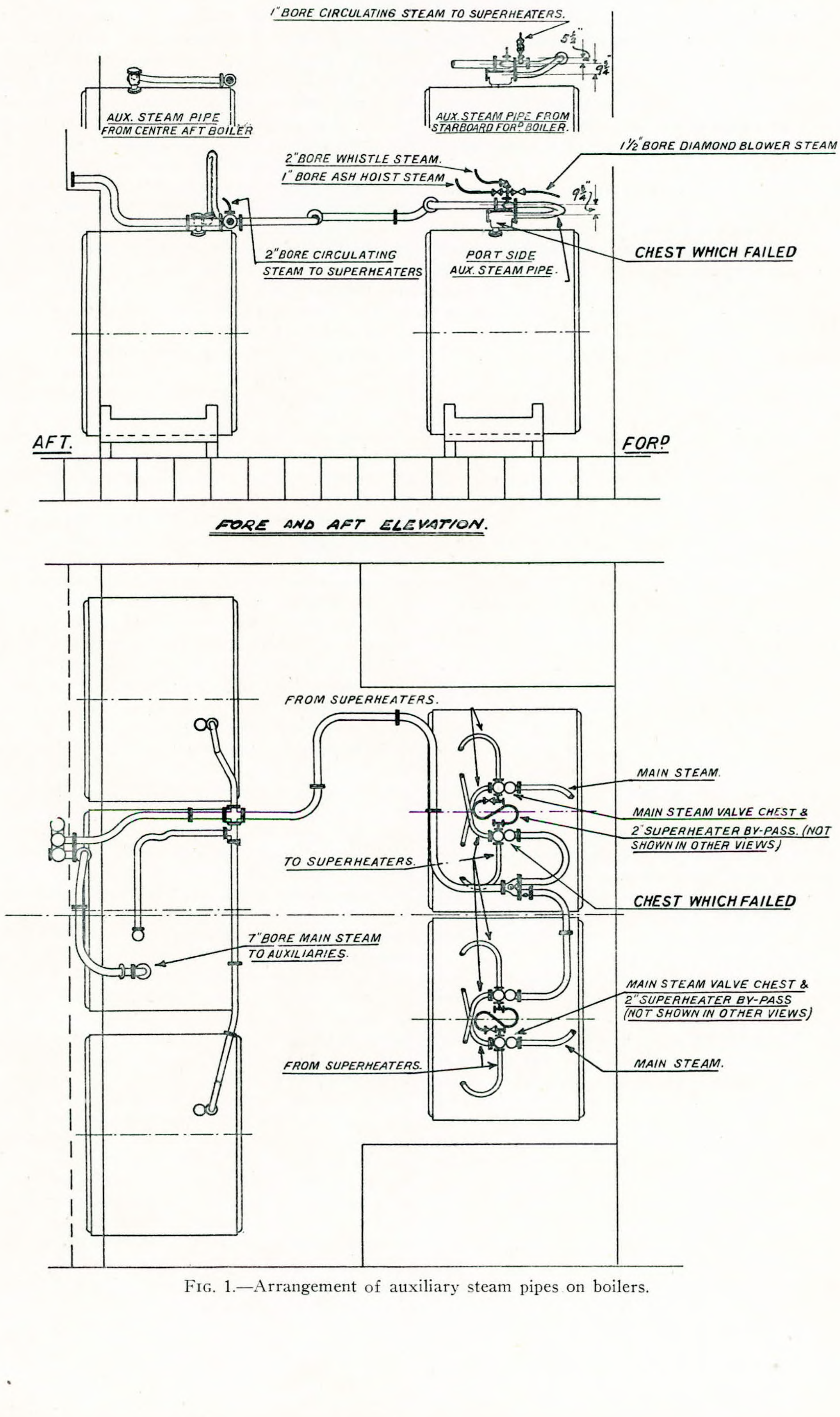


FIG. 1.—Arrangement of auxiliary steam pipes on boilers.

The ship arrived and anchored in Plymouth Harbour at 1.15 a.m. on the 5th December, 1934, and, as it was desired to obtain more speed to proceed from Plymouth to Antwerp, steam was raised on the after starboard boiler and, at about 10.45 a.m., this boiler was coupled in on both the main and auxiliary ranges of steam pipes, with the other three boilers.

During the forenoon 54 men of the stokehold crew were replaced by men who were transferred from the s.s. "City of Exeter", and to enable the new men to settle into their quarters and their work the ship's departure was delayed. At 12.16 p.m. all was in order, "Stand By" was rung at 12.28 p.m. "Slow Ahead", followed by various orders up to 1 p.m., when, the ship being outside the breakwater and the pilot discharged, the engines were put "Full Away". The ship proceeded on a course S. 40° E., and there being a gale of force (7) blowing from S.W. by W., she soon ran into the high sea and heavy south-westerly swell which was running. At about 1.20 p.m. the ship commenced to roll fairly heavily and continued to do so to an increasing degree up to 1.50 p.m. when the explosion took place.

From the time of leaving Plymouth the machinery had worked under normal conditions, apart from the abnormally heavy rolling of the ship, and there was no sign of any steam leakage or defect when, at about 1.40 p.m., Mr. Oswald Alexander Levack, the 3rd engineer, went into the stokehold to instruct the firemen in the use of the ash ejector, and apparently he was still so engaged at 1.50 p.m., as his body was found adjacent to the port ash ejector which had been in operation immediately before the explosion took place.

The 6th engineer, Mr. Charles Steven, who was on watch with the 3rd engineer, was in the engine room and heard no sound of the explosion. He was first made aware of the mishap by the crowding of the firemen into the engine room with the man who was injured by scalding. The 6th engineer immediately tried to get into the stokehold but found it impossible to proceed beyond the forward end of the passageway which terminated at the forward end of the after boilers.

The attention of the engineers who were not on watch was directed to the mishap by the noise and the sight of steam issuing from the funnel casing and they proceeded to the engine room without delay. It was realised that the 3rd engineer and some of the native stokehold hands were missing and, assuming these men to have been trapped within the firing space, the chief engineer, Mr. Edward Harrower, the 2nd engineer, Mr. John McGlashan, the 4th engineer, Mr. Alfred Park, the 5th engineer, Mr. William Starkey, and the 7th engineer, Mr. Peter Robinson, each endeavoured to enter the stokehold, but none was able to get beyond the forward end of the passageway between the after boilers, where the protection afforded by the roof of the passage and the guard plates attached to the front of the boilers ceased. It was soon apparent that a considerable reduction in the density of the steam in the stokehold would be necessary before an entrance to the firing space could be made, and to effect this the chief engineer gave orders that an attempt to close the stop valves on the after boilers should be made. The main engines were kept running with a view to reducing the steam pressure as rapidly as possible and the 2nd and 4th engineers proceeded to enter the space on the top of the after boilers from the upper engine room grating by the doorway in the screen bulkhead. The space was filled with steam of high temperature and they worked together in order that one might render the other assistance should it become necessary. At their first attempt they closed the main and auxiliary stop valves on the centre boiler before the conditions of heat and the difficulty of breathing forced them to retreat to the engine room. Then, with sacking wrapped around their heads, they made several further attempts before the valves on the starboard boiler were closed. The density of the steam was much greater on the port boiler, and considering it impossible to get to its valves at this stage they went to the lower platform in the engine room. The 4th engineer then went into the starboard side of the stokehold firing space where the steam had been rendered less dense, probably by the ventilator having been turned to the wind, and found the body of the coal trimmer, Enzadulla Asmutulla, lying under the ventilator and in front of the starboard boiler. He dragged the body to the passageway, and after passing it to some of the native firemen who were there, he went back into the space, searched the starboard side, shut the draught valves on the furnaces and swung open the fire doors of the foreward starboard boiler. The conditions in the port side of the firing space were still too bad to allow of his proceeding into it. The 2nd and 4th engineers then returned to the top of the after boilers and, the steam having cleared somewhat on the port boiler, they closed the stop valves. The forward starboard boiler was the only one on which the steam valves were now open and the pressure in it was being lowered rapidly. The lubricating oil pumps then stopped for lack of steam and made it necessary to shut down the main engines, but, to continue the reduction in the steam pressure, the safety valves of the forward boiler were eased by means of their easing gear. With the lowering of the boiler pressure and the ventilators having been put to the wind, the conditions in the firing space were greatly improved. The body of the 3rd engineer was then found and removed to the engine room. Although at that time a considerable volume of steam was still issuing from the fractured stop valve chest the native firemen were ordered to draw the fires of the forward starboard boiler. They obeyed the order without hesitation and worked with highly commendable steadiness and zeal. Shortly after this it was possible to gain access to the top of the

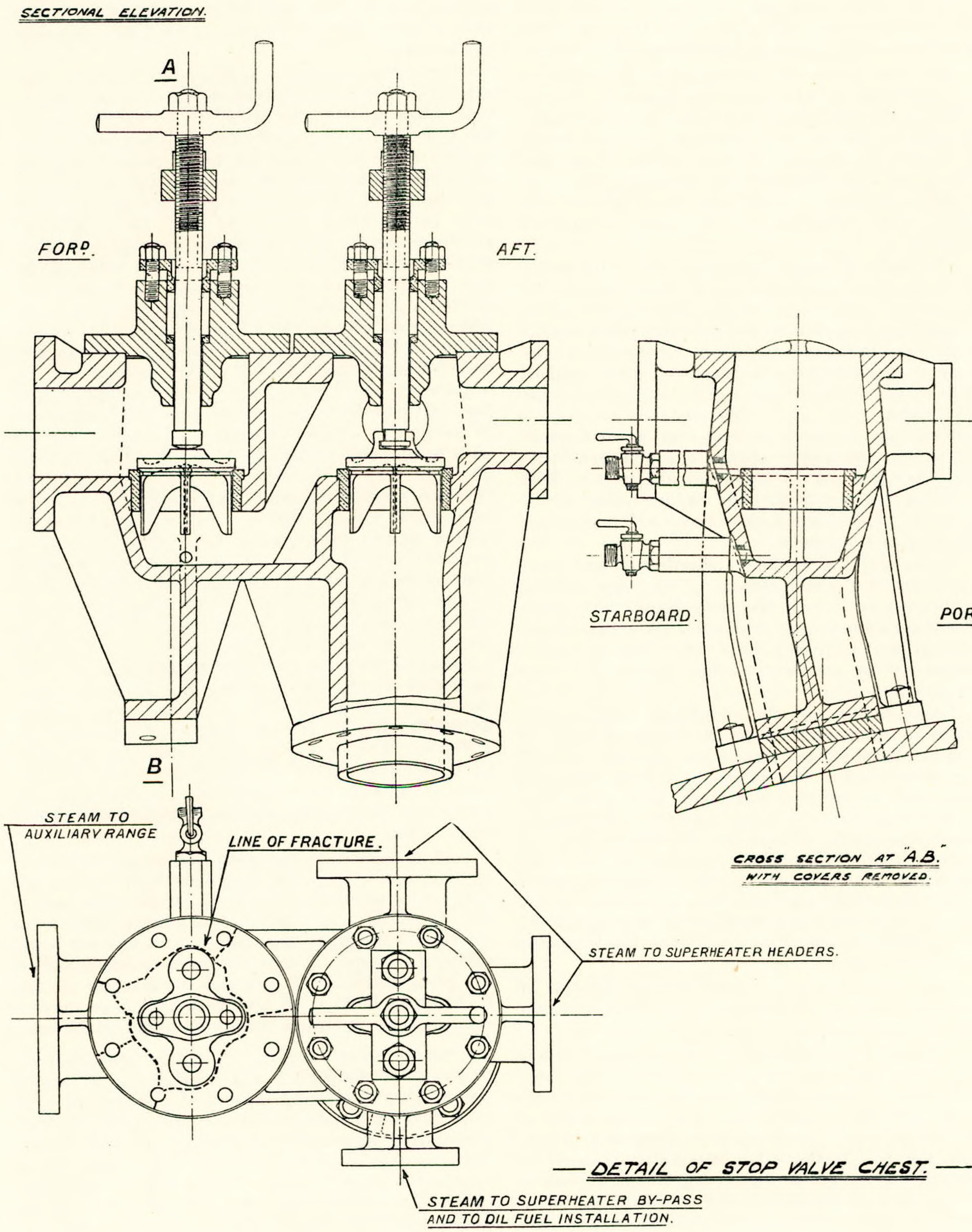
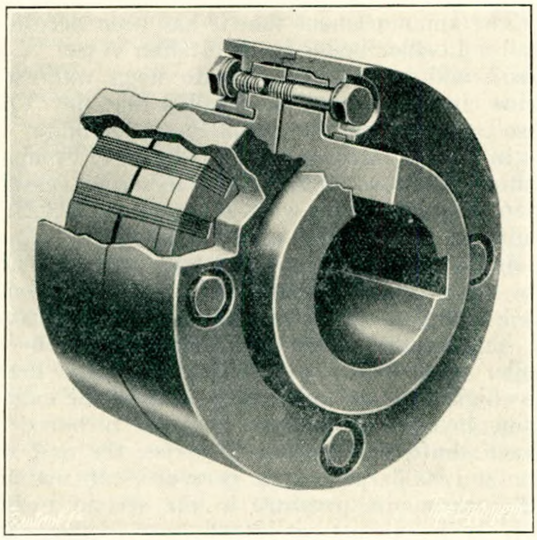

SECTIONAL ELEVATION.
A
FORD.
AFT.
B
STARBOARD.
PORT.
CROSS SECTION AT "A.B."
WITH COVERS REMOVED.
STEAM TO
AUXILIARY RANGE
LINE OF FRACTURE.
STEAM TO SUPERHEATER HEADERS.
— DETAIL OF STOP VALVE CHEST. —
STEAM TO SUPERHEATER BY-PASS
AND TO OIL FUEL INSTALLATION.

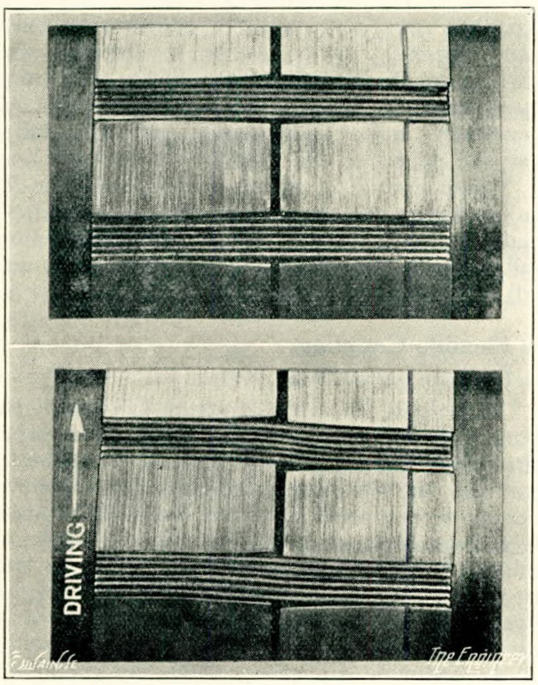
Fig. 2.

In addition to absorbing shock and vibration and so protecting the gear teeth, the coupling also permits the gears to adjust themselves axially—a very necessary condition for the satisfactory operation



Construction of flexible coupling.

of double helical gearing. As will be seen from the engraving, the coupling consists of driving and driven hubs having laminated springs arranged in axial slots to connect the driving and driven members. The slots are shaped so that the springs are adequately supported whilst retaining their resilience, and the springs are held in position in the



Position of springs under no load and overload.

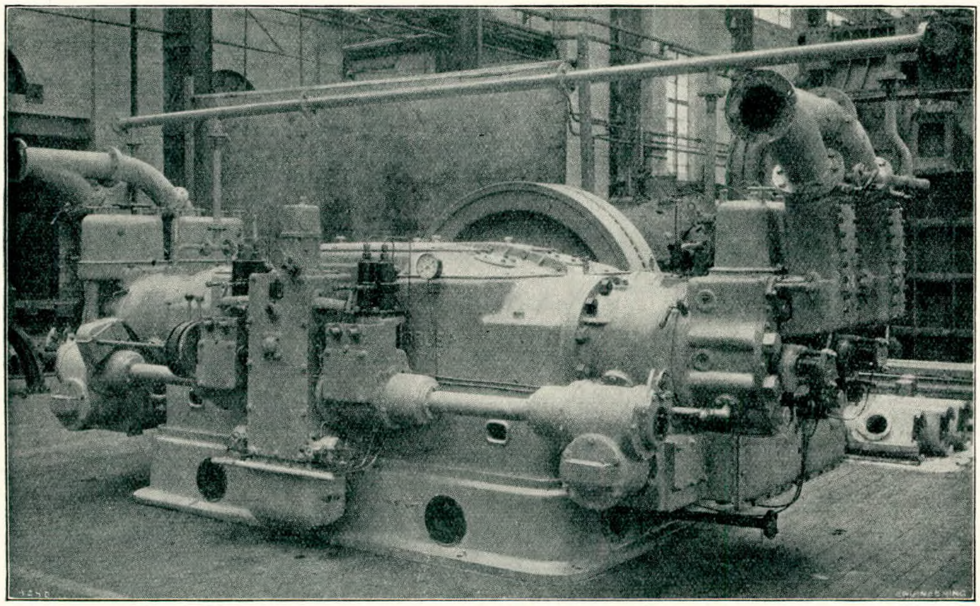
slots by shroud rings which are spigoted together. The shroud ring over the driven half-coupling is located from the shroud ring on the driving half-coupling, and is free from all contact with the driving hub. The second engraving shows the laminated springs with the coupling at rest and subjected to maximum overload. It will be observed that the slots are so shaped that at "no-load" there is a vee-gap between the springs and the slot walls, which closes up on the "driving" and "driven" faces respectively as the torque increases. Thus, the active load-bearing surface of each spring is extended inwards as the torque increases to maximum. Should it be desired to uncouple the driving portion once a coupling has been arranged in position, it is only necessary to part the shroud rings and remove the springs. As the successful functioning of the coupling depends upon the maintenance of adequate lubrication, a grease gun nipple is provided for the systematic application of the lubricant, and a soft packing ring is fitted to prevent the leakage of grease past the driven hub.

Refrigerating Plant on M.S. "New Zealand Star".

"Engineering", 22nd March, 1935.

A contract for the supply of six four-cylinder Diesel engines has recently been completed. All the engines are for driving CO_2 compressors, four of the latter being supplied by Messrs. J. & E. Hall, Limited, Dartford, and two by Messrs. The Liverpool Refrigeration Company, Limited, Liverpool. Two each of the compressor units have been fitted on the "Imperial Star", the "New Zealand Star" and the "Australian Star", three new vessels built for Messrs. The New Zealand Shipping Company, Limited, for operation between this country and New Zealand. A repeat order for six similar engines has also been placed.

One of the oil engines for the "New Zealand Star", which left London on her maiden voyage yesterday, is shown in the Brush works in the photograph reproduced on next page. It may be recalled that the engine already described was of 160 brake horse-power, but the model shown in the illustration is of larger size, designed to develop 300 brake horse-power at 290 r.p.m. All six engines have, however, been slowed down to operate at a speed of 250 r.p.m., at which speed the output is 250 brake horse-power. The chief characteristics of the engine are the excellent balance, due to the opposed-cylinder design, the reduction of torsional oscillation to a negligible amount on account of the short crankshaft, exceptional accessibility, and low fuel consumption. The design lends itself particularly well to marine installations, as the low overall height enables the engine to be fitted in 'tween decks spaces with ample head clearance for inspection and overhaul. It is important that engines designed for compressor operation should be flexible, so that the speed can be adjusted to suit the load with a view to maintaining economical consumption and reducing maintenance, and tests carried out in our presence on the "New Zealand Star" showed that the Brush units were somewhat outstanding in this respect. In this country, where the sea water is cold, the engines will be required for the most part to run at low revolutions, but during the course of the voyage, the engines will be gradually speeded-up to suit the increasing sea-water temperature, and will run at full capacity in the tropics. There will also be occasions when the ships are being loaded in dock with the hatches open, when the full capacity will also be required. In order to cater for the wide speed range thus called for, two governors with a changeover device are fitted to each engine, one governor controlling the engine from the lowest speed of 65 r.p.m. up to 150 r.p.m. and the second controlling the speed between the latter figure and the maximum.



The controls are extremely simple, and are grouped in a convenient central position. There are two levers, one being the control lever for starting and stopping, while the other regulates the advance and retard of the fuel-injection period. The latter feature is patented, and ensures clean running at various speeds and on different qualities of fuel. It is stated that the provision of this control was one of the main technical considerations which led to the order for these engines being secured. A feature which we believe to be unique for horizontal Diesel engines is that the exhaust valve is fitted at the top and the inlet valve at the bottom of the breech end. The inlet valve, being at the bottom, is at the coolest part of the breech and in a position where the coolest water enters, thus ensuring a maximum air charge. A further advantage of the arrangement is that the spray valve remains cooler, since no heat rises from the hot exhaust bend, and so on, to impinge upon it, and finally, should the sprayer dribble due to the presence of grit or to distortion of the needle valve, the fuel cannot drip on to the hot exhaust valve to produce excessive carbon deposit and other troubles. With the arrangement adopted, any drop from the sprayer falls into the incoming air, which is moving at high velocity, and forms a combustible mixture which is burned with the ordinary charge. The position of the exhaust valve at the top also enables it to be removed complete with its cage without disturbing the inlet valve. It is possible to exchange a complete exhaust valve, sprayer, or fuel pump while the engine is running, so that the danger of damage to the cargo due to the stoppage of the refrigerating plant is rendered exceedingly remote.

The cooling-down trials were entirely satisfactory on the two completed vessels, the "Imperial Star" and the "New Zealand Star". Tests made with a heavy marine fuel in the shops before delivery gave a consumption of 0·38lb. per brake horse-power hour, and it is anticipated that a still lower figure will be recorded when the engines are thoroughly run in. During the cooling-down trials, one of the fuel pump racks showed a tendency to stick, and a spare pump was substituted in five minutes with the engine running at full torque on three cylinders.

Condenser Tubes.

"The Marine Engineer", April, 1935.

In recent years the subject of tube materials for surface condensers has happily not received that degree of ventilation which was so distressingly in evidence some few years ago. For this state of affairs our thanks are due to the tube manufacturers and those independent research experts who tackled what, at the time, was undoubtedly a very grave and heart-breaking problem and solved it with reasonable rapidity and commercial insight. At one time it was undoubtedly true to say that the condenser tube was the "Achilles heel" of the modern turbine installation, both marine and stationary types. When, about 1922-23, the first 80-20 cupro nickel tubes were made commercially available it was fairly evident that the end of the problem in its most acute form was in sight. Onwards from that time, of course, further advances have been made, the amount of nickel in the alloy has been increased with beneficial results, and other very satisfactory tube materials have been evolved.

To-day the condenser tube problem is interesting engineering history.

These remarks are inspired after a perusal of a most interesting paper on this subject which was read recently before the Scottish Section of the Institute of Metals by Mr. A. Spittle. In connection with cupro nickel tubes Mr. Spittle revealed in his paper some full-scale sea service experiments which were carried out by a well-known North Atlantic superintendent engineer in the year 1924. It was decided to re-tube the condensers of two twin-screw sister ships with the then new cupro nickel tubes. In one ship tubes of 80-20 mixture were utilised; in the other vessel special cupro nickel tubes were evolved of 95-15 and 90-10 quality alloys, these tubes being fitted in alternate rows in each condenser. The vessels were run under similar conditions for a year, when the condensers were opened up for inspection, a complete row of tubes on the vertical centre line of each condenser withdrawn and split open for examination. The 90-10 tubes showed general corrosion; the 90-15 examples were in better condition, although they had suffered from a certain amount of general corrosion; and the 80-20 tubes appeared to be as good as when they had been installed 12 months earlier.

This result was particularly striking, because with the brass tubes previously used tube failure from corrosion could normally be expected after about seven months' service, and the trouble would steadily increase until replacement of the whole of the tubes was imperative at the end of 12 months.

These practical tests clearly indicated that nothing less than 20 per cent. of nickel in the tube mixture was of much benefit from the corrosion standpoint, and accordingly the low-nickel tubes were replaced by 80-20 mixture tubes, while 100 tubes of 70-30 quality were inserted at the same time. Subsequent results indicated beyond any shadow of doubt that the last-mentioned type of tube material was quite the best, and readily justified its high cost in the immunity from tube trouble which it ensured over reasonably long periods of service. From that time forward the cupro nickel condenser tube as we know it to-day increased in popularity, and for a number of years practically every vessel (with one or two notable exceptions), in which water-tube boilers have been used, has been provided with cupro nickel condenser tubes. This is a striking tribute to the success of these tubes, for it must be borne in mind that their cost is not by any means low, although, of course, in actual practice they prove economical because of their relatively long life.

With a view to producing a cheaper type of tube of greater durability than the ordinary Admiralty brass tube, the aluminium brass condenser tube was evolved some few years ago. The cost of this is very little higher than that of ordinary 70-30 brass and Admiralty brass tubes, and in a comparatively short time tubes of this class have been widely employed in a variety of different types of vessel. In fact, to-day the ordinary brass and Admiralty brass condenser tube is fast becoming obsolete, and before many years have passed, the aluminium brass tube will probably be universal where first cost is of importance, while cupro nickel tubes will continue to hold the field for all high-powered vessels using water-tube boilers and having what might be termed high-duty condensers.

In his paper we see that Mr. Spittle mentioned that over a thousand vessels are now provided with aluminium brass condenser tubes; actually, the number is considerably in excess of 1,000, and is actually nearer 1,500. Aluminium tubes in the aluminium bronze range are, perhaps, but the logical development from aluminium brass, and in the comparatively short period of three years, they have established a sound reputation for dependability. In due course they will doubtless be very extensively applied in many cases, although it is unlikely that they will ever oust any of the proved alloys introduced some few years earlier, and to which we have referred above. They are of particular value, for instance, where a vessel has to operate in estuaries and docks containing water which is liable to have a strong scouring effect on the tubes. Under these conditions aluminium bronze tubes have been found to stand up remarkably well.

Increased Speed Without Extra Fuel Consumption.
"The Motor Ship", April, 1935.

For new construction the air-injection Diesel engine is obsolete. The question must, therefore, necessarily arise in the minds of shipowners whether it is not desirable to convert the engines in their older ships to airless injection.

This can be quickly done, and some results of conversion have already been published in "The Motor Ship". An example is given on another page in this issue. Two voyages were undertaken by a ship, one before converting the machinery and the other after the change was made. Under precisely similar weather and loading conditions, an additional speed of 0·4 knots was attained, with a somewhat lower fuel consumption.

There is no doubt whatever regarding the accuracy of these figures, which were provided by the shipowner. The conversion of existing air-injection engines to the airless type can be utilized either for increasing the power and speed in the way it was carried out in the vessel in question, or the fuel consumption may be reduced. In the latter case it is certain that in practically every ship a reduction in the fuel bill of between 7 per cent. and 9 per cent. can be attained.

No such conversions have been carried out in this country, although on the Continent at least 30 vessels have had their engines modified. We are informed that as the results have been so successful, dozens of ships are likely to have the same work carried out on them. The modifications needed are small, the work only takes a week or two, and the cost appears to be reasonable. We suggest that British shipowners investigate the matter, since nearly every owner is very desirous of obtaining a little more speed out of his ships.