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Exhaust Steam Engineering.

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Chairman: H. BAKER, M.Sc., Ph.D. (Principal of the College).

IN my talk to you to-night about Exhaust Steam Engineering, I shall do so from the point of view of a marine engineer, that being the type of engineering with which I am most familiar.

Exhaust steam is, of course, the consequence of the changes that steam passes through, in the cylinders or blading of an engine, where part of the heat which the steam originally contained is converted into mechanical work. Exhaust steam is a mixture of steam and water. The percentage of steam that still remains in the mixture is called the dryness fraction and defines the quality of the exhaust. The quality of the steam at exhaust is not a matter of chance, as it used to be considered —it is pre-determined by three factors, which are:—

(i) The initial pressure and temperature or quality of the steam at the stop valve,
(ii) The back pressure at exhaust,
(iii) The internal efficiency of the engine, also known as "relative efficiency".

The first and second of these factors are terminal characteristics of the steam cycle of some particular engine. The first factor involves two characteristics of the high pressure steam, which need not be either pressure or temperature. The reason for this last statement is that when two things about steam are known, it is possible to calculate all the others. For the second factor the pressure only is required and nothing else will satisfy our requirements.

The significance of these remarks will be appreciated later. For the moment, let us consider our third factor—the relative efficiency.

A steam engine is a machine for converting heat into work. Between the limits as defined by our first two factors a definite known quantity of heat per lb. of steam is available for conversion into work. This is much less than the total heat per lb. of steam.

The heat available for conversion is known as the adiabatic heat drop, and is the measure of the work obtainable from a pound of steam with a perfect engine. A practical engine converts only part of the adiabatic heat drop into work. The ratio of that part actually converted to the whole available heat drop is known as the relative efficiency. Fortunately the relative efficiency is a very stable ratio. It is obvious that any type of steam engine which had a very low relative efficiency would have no chance of surviving, while another type with a particularly high efficiency would sweep all its competitors off the field.

Sir Alfred Ewing tells us that "in practice about 70 per cent. of the adiabatic heat drop may be realized in favourable cases",* so if we do not know what the relative efficiency of any particular engine is, we at least know what it should be if it is to be a good engine.

It is necessary to define the meaning of the back pressure at exhaust. Sir Alfred Ewing's statement applies to heat-drop to condenser pressure, and he probably would not have considered that the marine reciprocating engine would be a "favourable case".

If, however, the back pressure is taken from the indicator card it will be found that the reciprocating engine will come up to Sir Alfred's standard. The turbines may have their heat drop calculated to vacuum pressure. On these terms the turbines took a long time to catch up the reciprocating engines in the matter of relative

* "Encyclopædia Britannica", Fourteenth Edition, 1929, Vol. 21, p. 361.

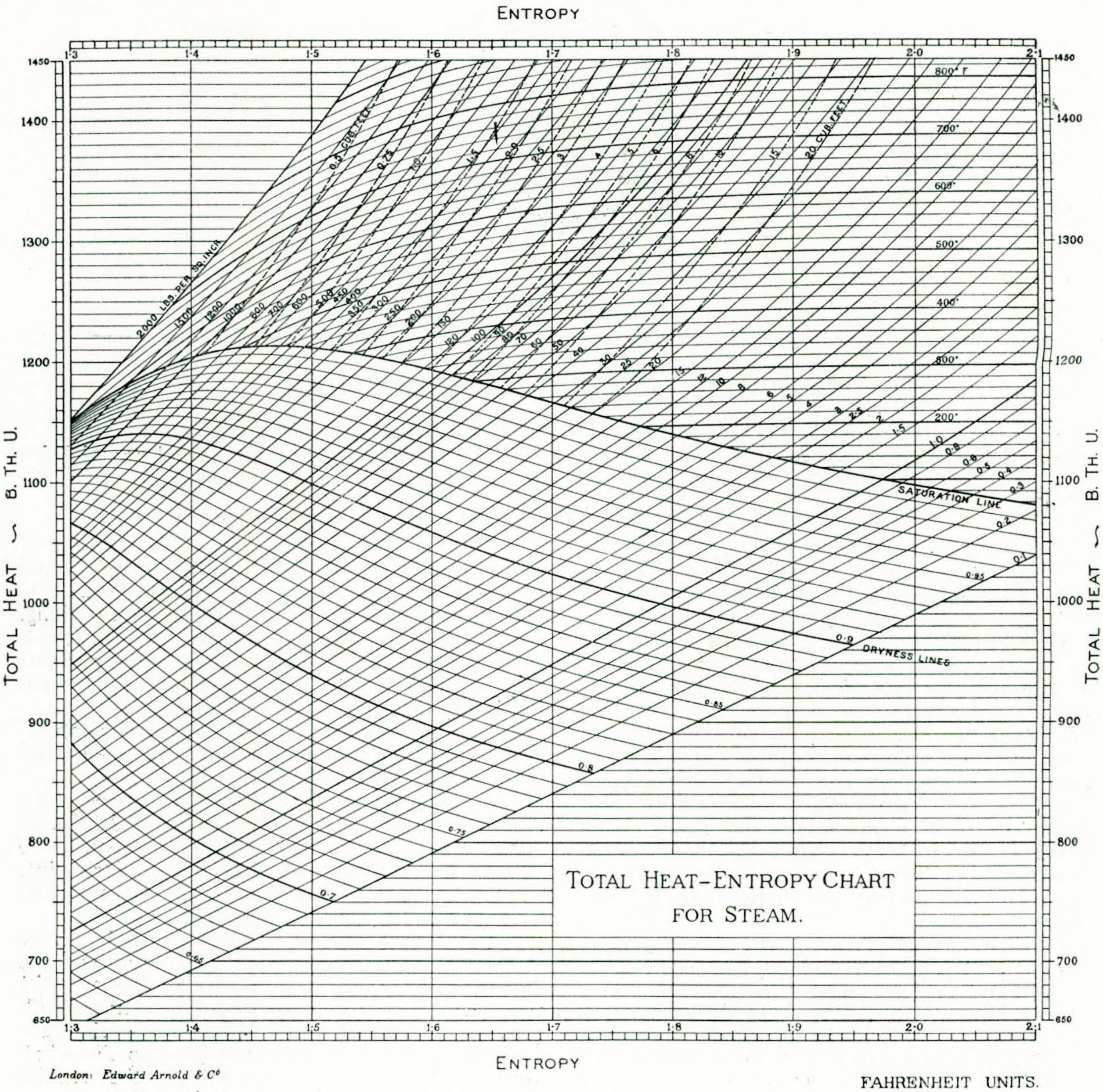


FIG. 1.—Mollier chart.

efficiency, because the latter were independent of condenser defects, but owing to their high mechanical efficiency the turbines were able to hold their own.

Before we go on to the discovery of the heat drop, let us consider how our newly acquired knowledge will be useful.

You know that 2546 B.T.U. equal 1 h.p./hour. This is easily determined by multiplying 33,000 by 60 to get ft. lb. per hour and dividing the product by 777·77, the mechanical equivalent of 1 B.T.U. If this constant, 2546, is divided by ·7 times the adiabatic heat-drop, the result will be the engine's steam rate in lb. of steam per i.h.p. hour. Even this simple formula can be further simplified, for if instead of multiplying the heat-drop by ·7 the constant is divided by that figure, a new constant, 3636·6, is obtained, which is easily remembered. All we have to do then is to divide our new constant by the adiabatic heat-drop and we know the steam rate of the engine in lb./i.h.p. hour.

Fig. 1 is called the Total Heat Entropy or Mollier Chart. It might be described as a steam table set out in the form of a chart, in which total heat and entropy are the co-ordinates. Nobody has yet discovered a form of words which defines entropy either to his own or to anybody else's satisfaction. It is like Hope, a gift from the gods,

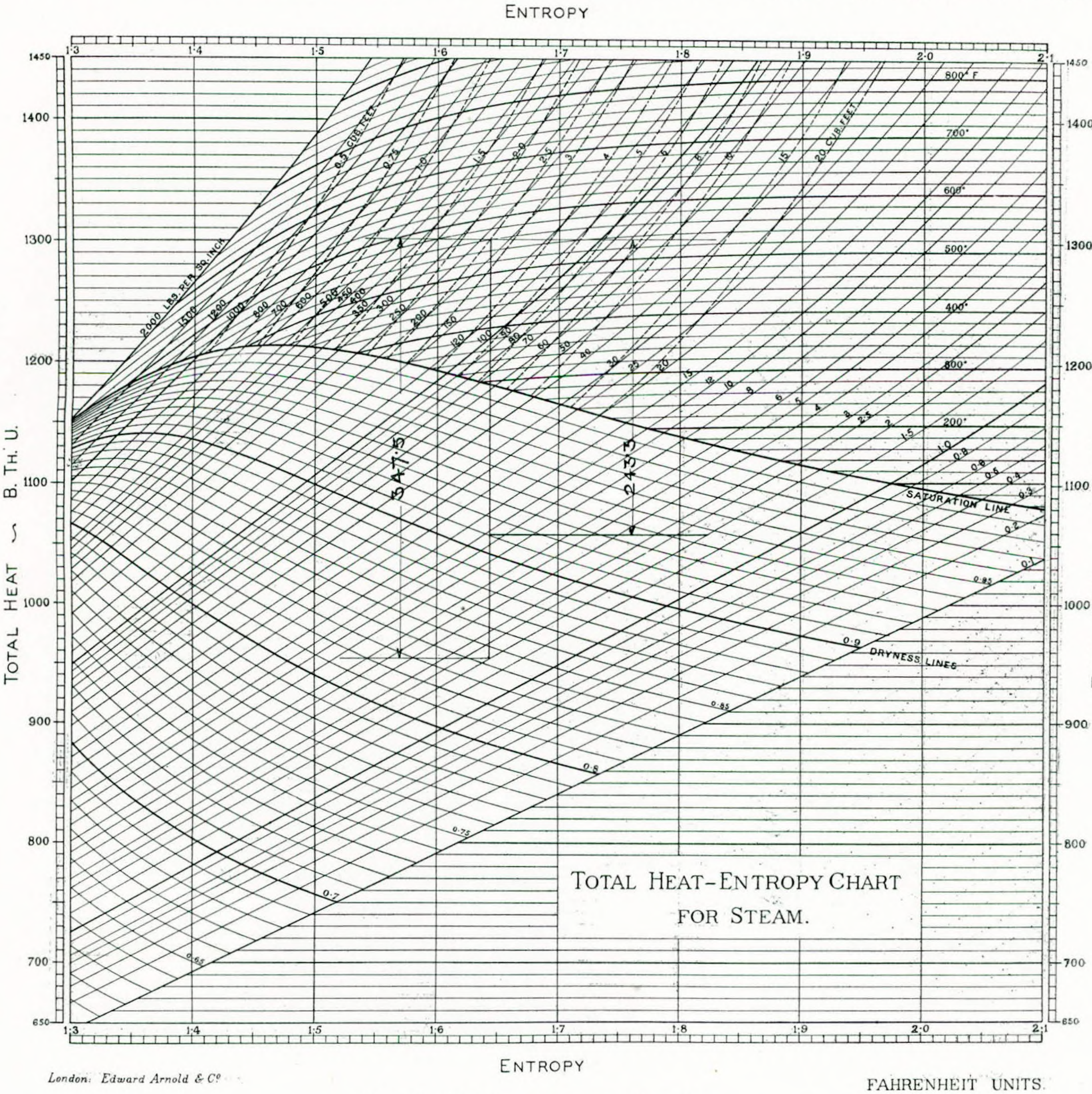


FIG. 2.—Skeleton diagram of engine.

which defies definition but for which we should be thankful. Entropy will be found in the steam tables when required. Even if it was calculated from the known formula it would be necessary to look it up in the tables to check the calculation.

This chart has considerable advantages over the Temperature Entropy Chart, which is better known. All information to be derived from it appears either directly on the face of the chart or requires the measurement of the length of a straight line. Adiabatic heat-drop is shown by a vertical straight line. Let me here say a word about this word "adiabatic". One may sometimes meet the expression "Isentropic heat-drop", which means much the same thing and is more strictly correct. It means heat drop with constant entropy. Isentropic expansion is always adiabatic, but adiabatic is not always isentropic.

Now let us consider the use of the chart.

In Fig. 2 we have the diagram of an engine taking steam at 230lb. per square inch and 560° F. temperature and exhausting against a back pressure of 2lb. per sq. in. This can be regarded as the disembodied spirit of an engine. We know nothing about the size or horse power of the engine. But we already know a lot of other things about it. For instance, we know that the adiabatic heat-drop is 347·5 B.T.U. per lb. of steam. By dividing our

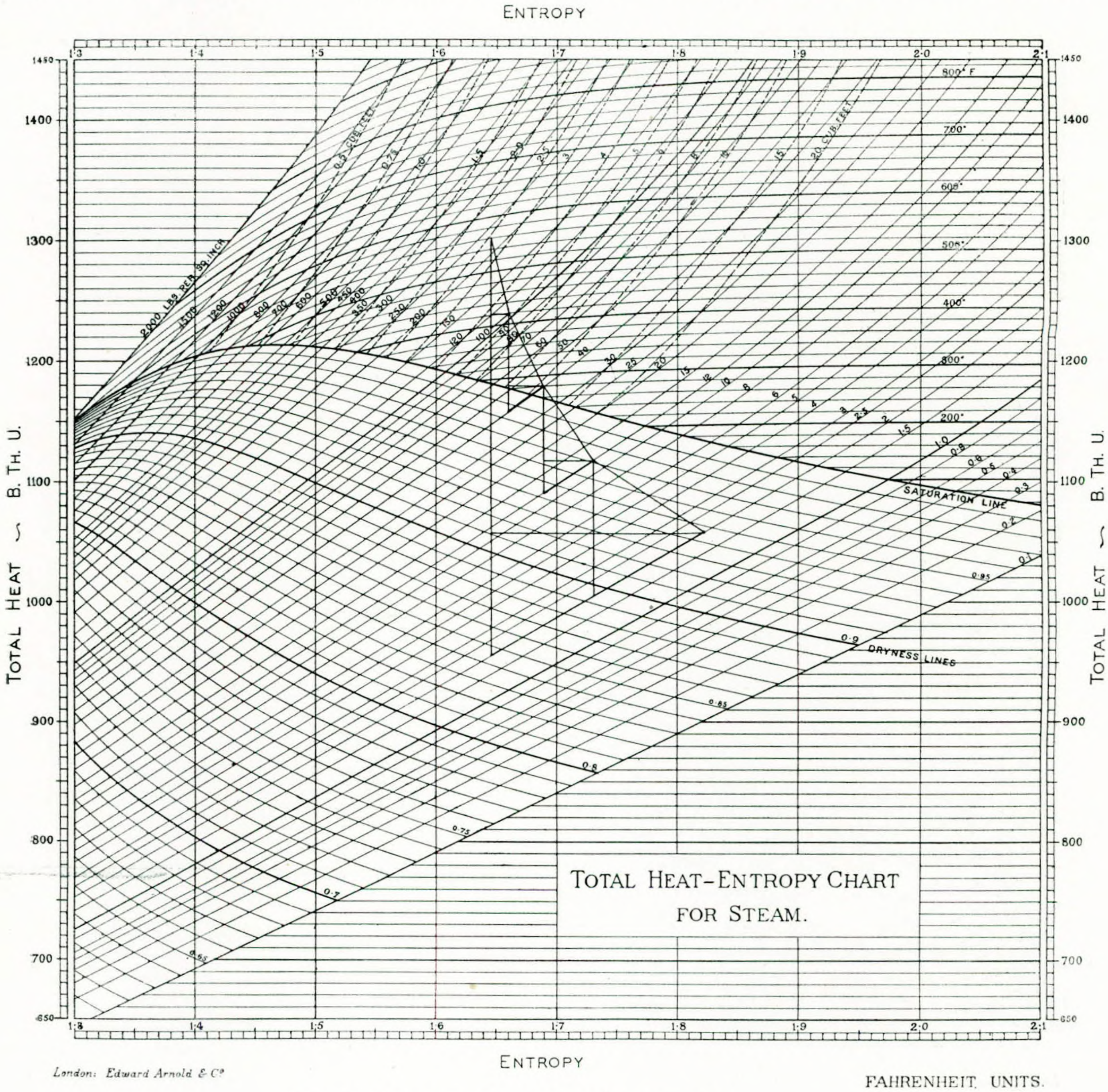


FIG. 3.—Skeleton diagram of engine completed.

constant 3636·6 by this figure we find that its steam rate will be 10·45lb. of steam per i.h.p./hour.

It is useful to multiply the adiabatic heat drop by ·7 and find that the actual heat drop is 243·3 B.T.U. per lb. If we draw a horizontal line through this point to meet the back pressure line we find that the steam at exhaust is 94·6 per cent. dry and contains 1061·4 B.T.U. per lb. and most of these are going overboard with the circulating water unless we can do something about it.

In my interpretation of the title of this lecture, I shall try to show how these thousand or so B.T.U. can be reduced in number. But first perhaps it may be desirable to see a more comprehensive picture of our spectre. Fig. 2 is, to say the least, somewhat sketchy.

In Fig. 3 we see the whole of our spectre. We now discover that it is a quadruple expansion reciprocating engine, and here I may point out what became of the 134 B.T.U. per lb. which did not get converted into work. It will be noticed that each cylinder is represented by a triangle, one side of which is vertical, and for convenience we will call this side the base. In each case a horizontal line is dropped from the apex to the base, dividing the base into two parts. The whole of the base is the measure of the adiabatic heat-drop of that cylinder, and that part of the base which is above the

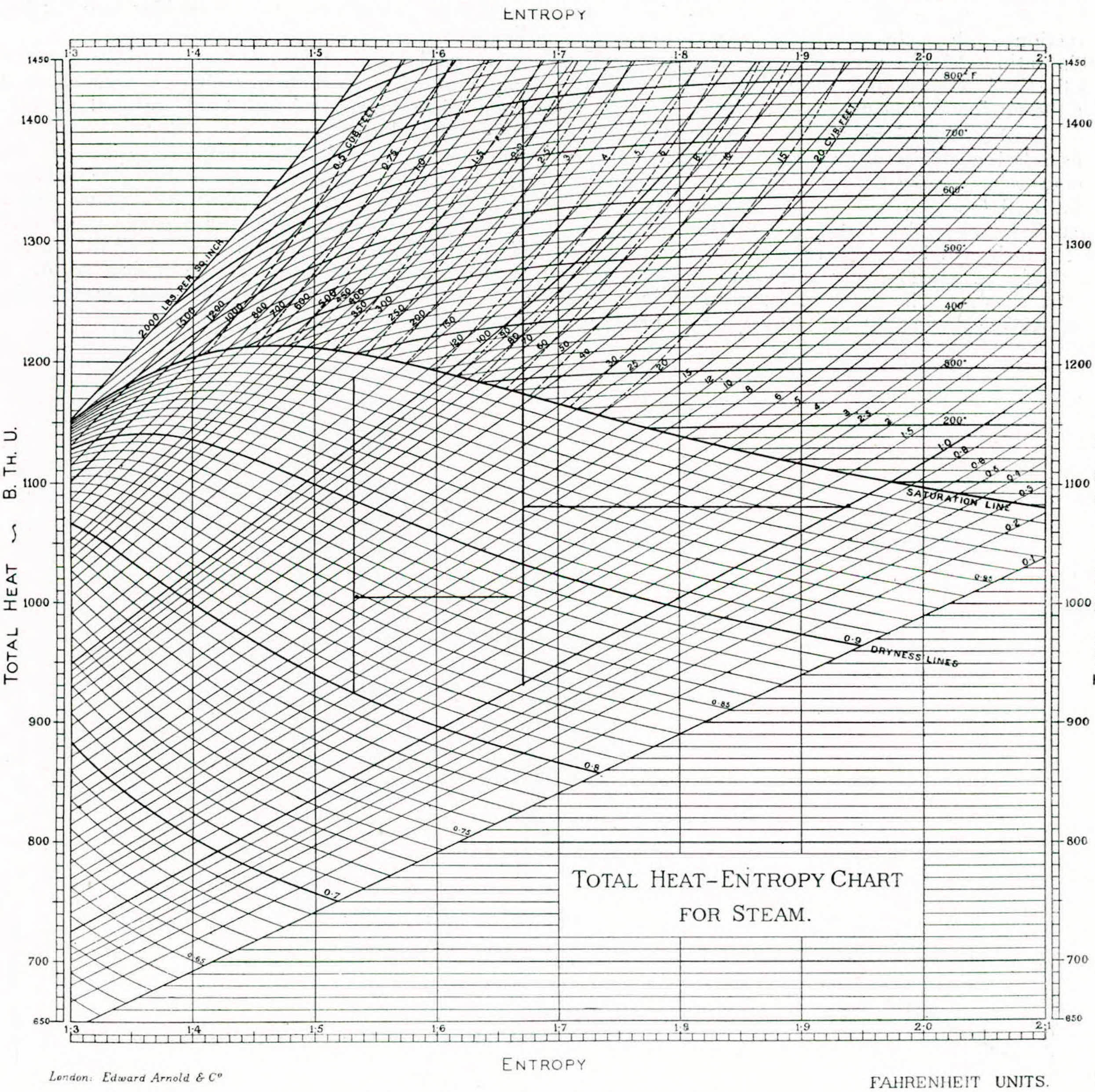


FIG. 4.—Comparison of two engines.

horizontal line is the actual heat-drop and the ratio of these two lengths is the internal efficiency of the cylinder.

From this it will be seen that the heat which is not converted into work in any one cylinder is returned to the steam, and in consequence the next cylinder receives steam which is hotter or dryer than it otherwise would be, but not of a higher pressure. The following cylinders do convert part of it into work, but finally the l.p. cylinder exhausts all the unconverted heat to the condenser, and the onl effect it has at that stage is that the final exh ust is dryer.

Here it will be useful to say something more about the relative efficiency, because it is the dryness fraction at the exhaust which gives a measure of this ratio. If the pressure and dryness fraction at the final exhaust are known, the relative efficiency of the engine is known exactly.

In the case of the steam reciprocating engine the relative efficiency has remained constant for many years, while the marine steam turbine has advanced from 65 per cent. to 70 per cent. in the last 20 years, and I rather suspect that it will not get much further. There are, however, other methods of economising open to it which offer better results. The great advances made in marine engineering during the last 40 years are not due to an improvement in the relative efficiency.

Fig. 4 shows the diagrams of two express liner engines, separated in time by 40 years. They both have an internal efficiency of 70 per cent., yet the modern engine has half the fuel rate of the engine of 40 years ago. It has not half the steam rate, because the heat-drop is not double that of the old engine. The improvements in boilers, condensers and feed heating systems will easily account for extra economy required to halve the fuel rate. Hence it will be seen that the enormous improvement which has taken place in marine engines is not due to any change in the internal efficiency of engines.

Before proceeding, let us consider two further diagrams.

Fig. 5 shows that our heat entropy chart is only part of a much larger chart. This chart is extended to the zero of both total heat and entropy. Here we find that we have a water line which is a flat curve, to which all the pressure lines in the wet field are tangents. The water line extends from the common zero to the critical point, which we rather surprisingly discover to be in the centre of the chart. The saturation line and all the quality lines meet the water line at the critical point. I have not extended these to that point because the chart would become too congested.

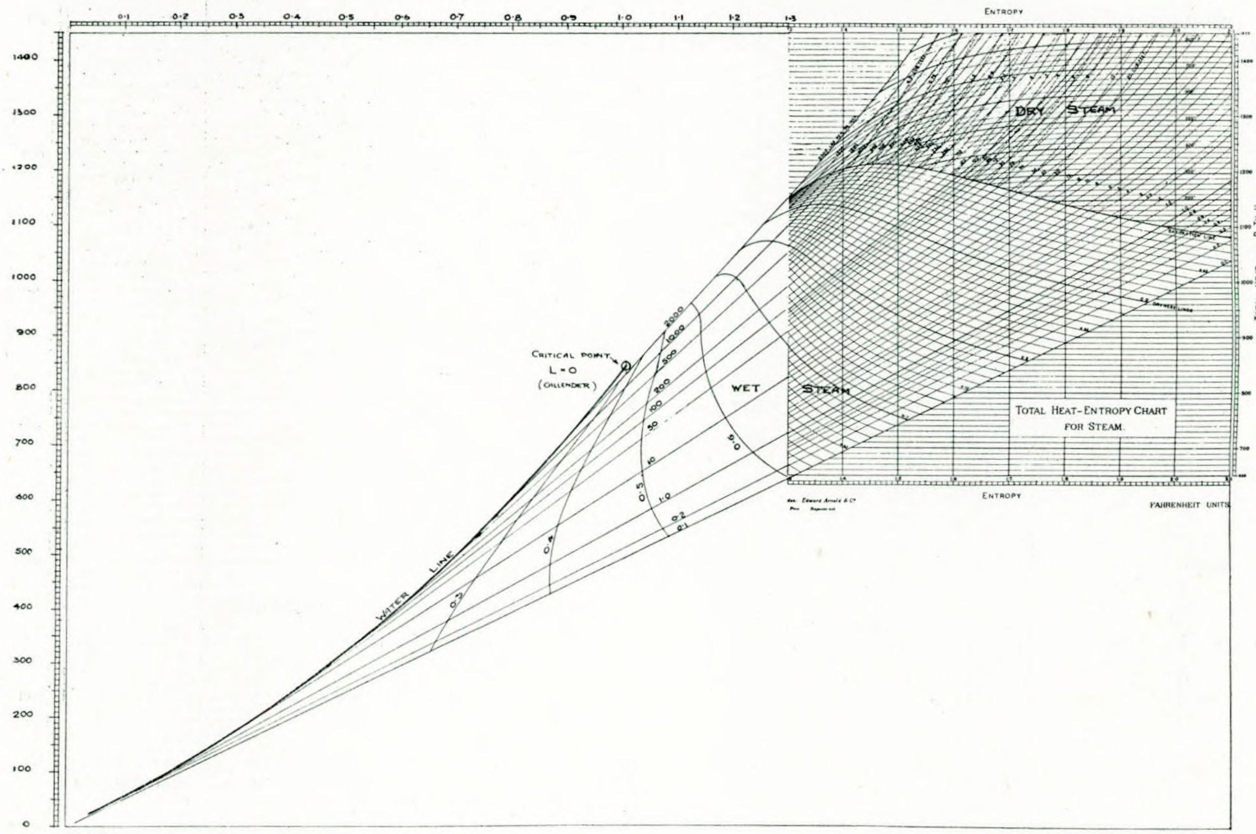


FIG. 5.—Extended Mollier chart.

If you have ever taken the trouble to investigate the pressure lines in the wet field (they are, as you know, straight lines) you will have discovered that they are not parallel, which is fairly obvious, nor do they run to a common radiant point. They appear to have no order at all. This diagram shows that they follow a very definite law. Without discussing the extended chart, let me use it to show the whole cycle of the steam through boilers, engine, condenser and feed system.

In Fig. 6 we have our disembodied spirit once more. It will be remembered that 1061 B.T.U. per lb. were exhausted to the condenser. This shows that 1,000 B.T.U. went overboard, so that 61 B.T.U. were returned with the condensate. Therefore it is necessary to return to the condensate 1243·3 B.T.U. to make good those used or discharged overboard. These 1243·3 B.T.U. can be divided into three sections—feed heating, boiling and superheating. The heat required for the last two of these can only come from the fuel, but that required for feed heating may be derived from other sources. More will be said about this later. Let us now consider two rather insignificant lines in the lower left hand corner. The lower one shows the position the base line would have occupied if the condensate had been at the same temperature as the steam in the condenser. This lowering of the condensate temperature is (or rather "was", because there is no necessity for it now) largely due to air in the condenser. Hot air at low pressure is difficult to pump, and it was the cold water required to cool the air that reduced the temperature of the condensate. It will be seen presently how this trouble was overcome. The upper line would be the position of the base line if the pressure in the condenser was the same as the back pressure. This is not all due to the resistance offered to the steam by the exhaust ports; through some condensers there is a considerable resistance. But that, like the cold condensate trouble, is now a thing of the past.

Now let us see how condenser design has overcome these two troubles. The condenser shown in Fig. 7* is Weir's Uniflux. The name refers to the constant speed of the steam through the condenser, the area available being in proportion to the uncondensed steam left.

It is shown as being a first step in the right direction. This condenser was designed for a turbine, as may be seen from the large size of the exhaust steam inlet. Observe the open pitching of the top rows of tubes to allow the steam to enter the tube bank, but note that the tubes are diagonally pitched. It will also be noticed that the air pump

* From "Turbines Applied to Marine Propulsion", by Stanley J. Reed. Constable & Co., Ltd.

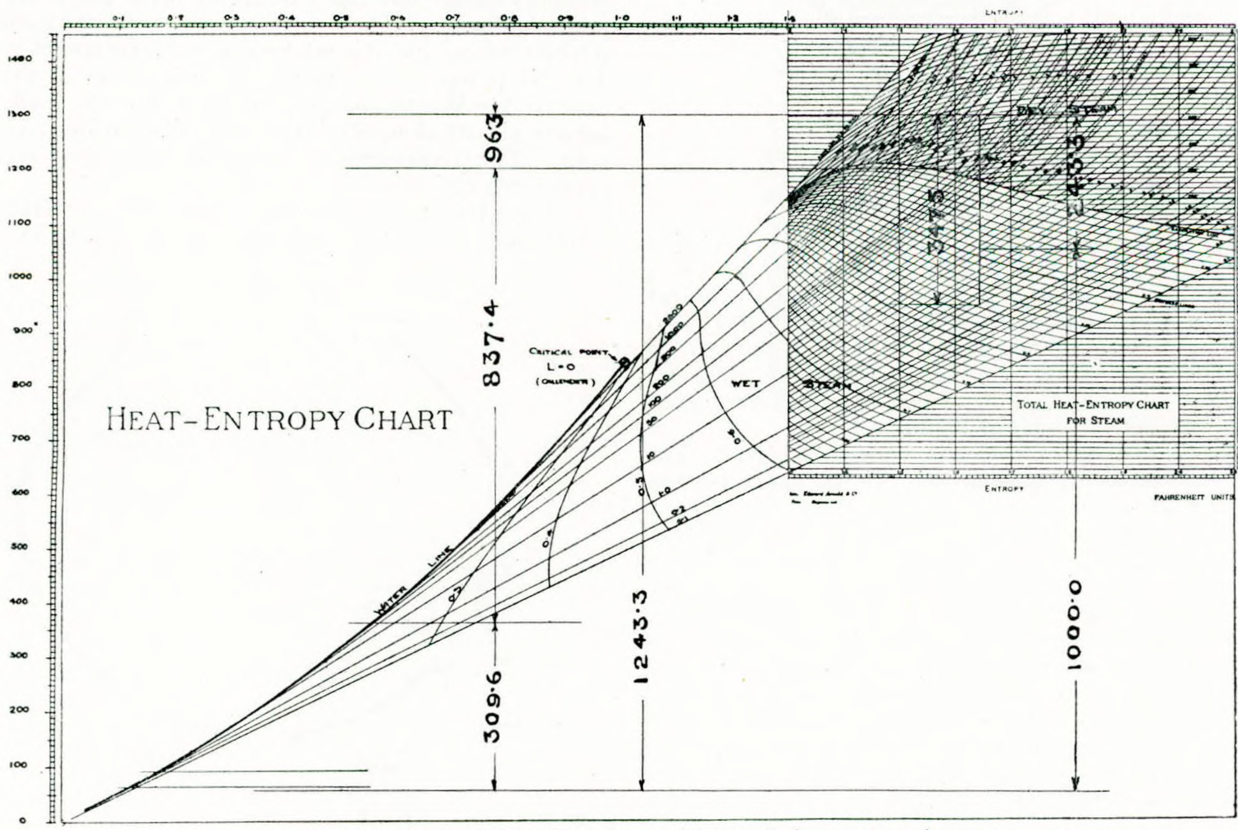


Fig. 6.—Extended Mollier chart with heat balance of engine.

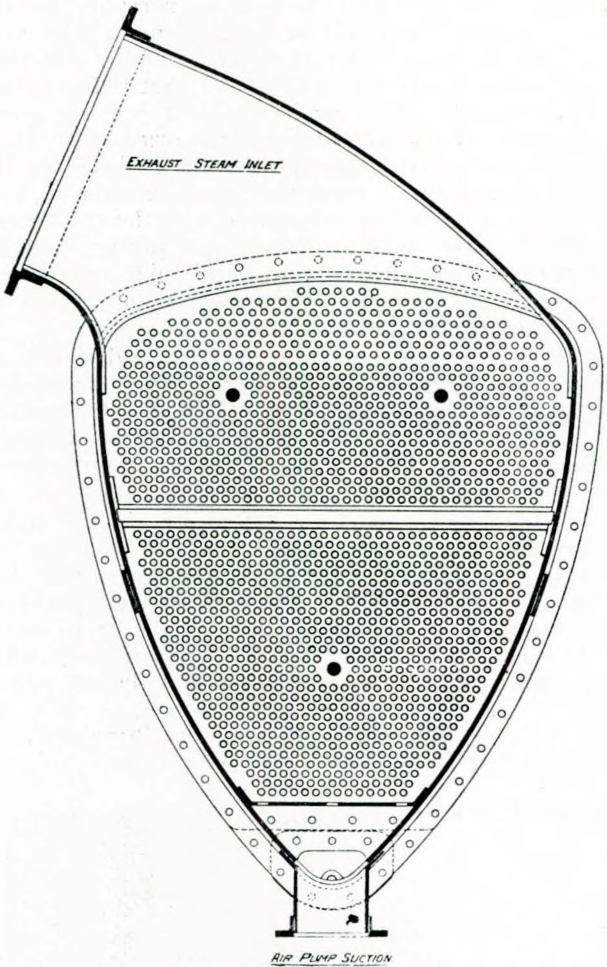


FIG. 7.—Uniflux condenser.

suction is vertical. This means that only a wet air pump was fitted. At a later date the air pump suction was fitted at an angle of 45° and a branch taken from it to a dry air pump, which did really pump air, the air being cooled after it had left the condenser. At a still later date the dry air pump had a separate suction, which was protected from the steam by a special baffle.

*Fig. 8 shows a Westinghouse condenser. This is not a marine condenser; it is included because it presents some early improvements—its date is 1913. The steam was admitted to the tube bank by steam lanes—they were very short, but it will be seen how this idea was elaborately developed. The air pump suction was well up on the side of the condenser and well protected by a baffle, too well in fact, as it formed an air pocket open at the ends only. The condensate was extracted by a separate pump, a water gauge being fitted to show how much condensate was in the condenser. This was required because the extraction pump was hand-controlled. The pitching of the tubes was

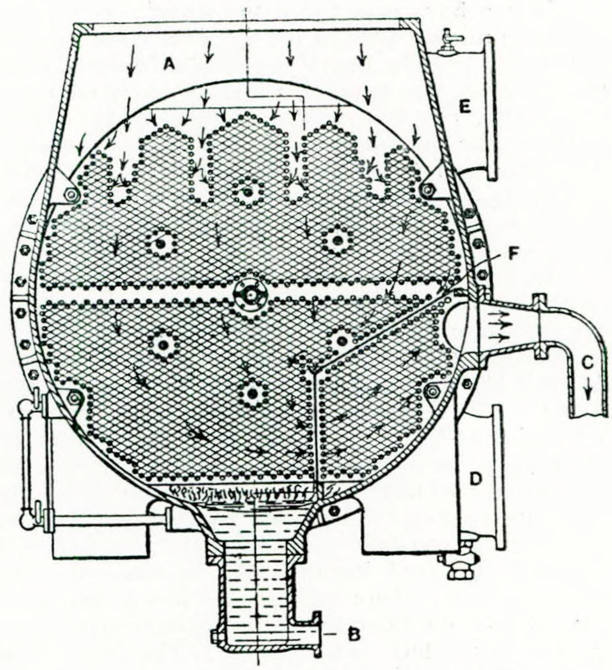


FIG. 8.—Westinghouse condenser, 1913.

diagonal and uniform throughout.

Fig. 9 shows a condenser by Mirrlees Watson, which is rather like the Uniflux in form but wider and shallower. There are more rows of open pitched tubes, and the pitching is now rectangular. The air pump suction is not so high up as in the case of the Westinghouse condenser, but the baffle admits air all along its lower edge like an inverted weir. The remainder of the tubes have diagonal close pitching.

Fig. 10 shows the tube plate of a French condenser—it is an example of a good idea

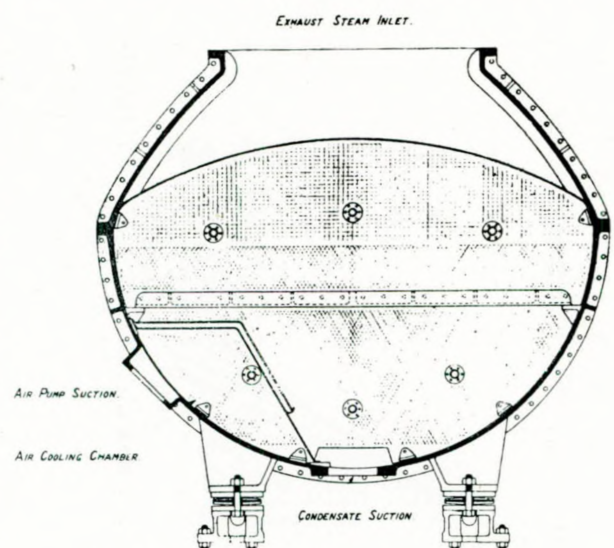


FIG. 9.—Mirrlees Watson condenser.

* Proc. Inst. Mech. Eng'rs., 1914, p. 756.

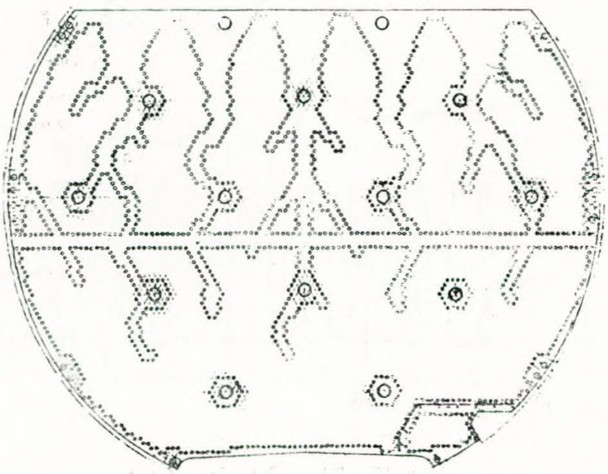


FIG. 10.—French tube plate.

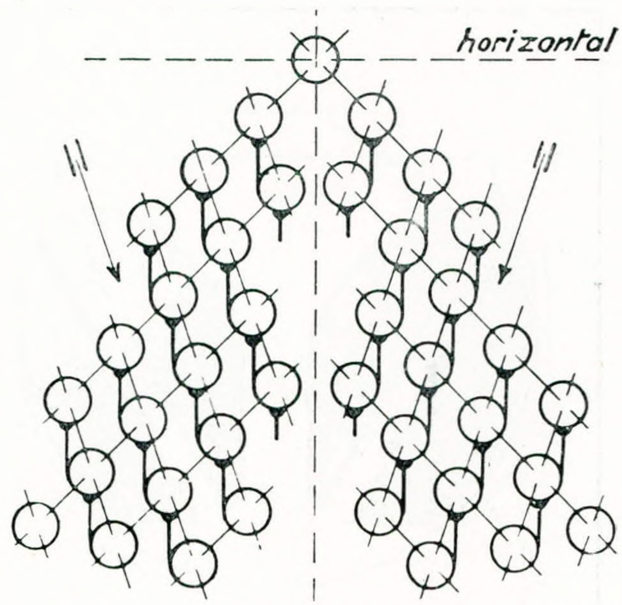


FIG. 12.—Delas condenser tube arrangement.

incorrectly carried out. It seems certain that these steam lanes defeated their own ends by not being straight. The air pocket is too small and too low down. It may possibly have played its part in leading up to a good arrangement.

Fig. 11 shows the Delas condenser made by Mirrlees Watson. This is a really high class modern condenser. The steam lanes are developed into deep V grooves, so that the tube banks become a light screen of tubes which have a very precise pitching.

In Fig. 12 the tubes are so arranged that the condensate falling from one meets the lower tube tangentially and only passes round 90° of it, leaving the remainder of the tube dry. The steam passes through the lanes in straight lines bordered on one side by dry tubes and on the other by a screen of condensate which is constantly being reheated. The condensate in this condenser is well up to vacuum temperature—the difference of pressure is negligible.

Fig. 12a shows a 40,000 sq. ft. Mirrlees-Delas condenser recently built for the Dalmarnock Power Station.

Lastly, in Fig. 13 is shown Weir's Regenerative Condenser. Here the steam lanes give way to a great open trunk road from the top to the bottom of the condenser. The exhaust steam makes direct

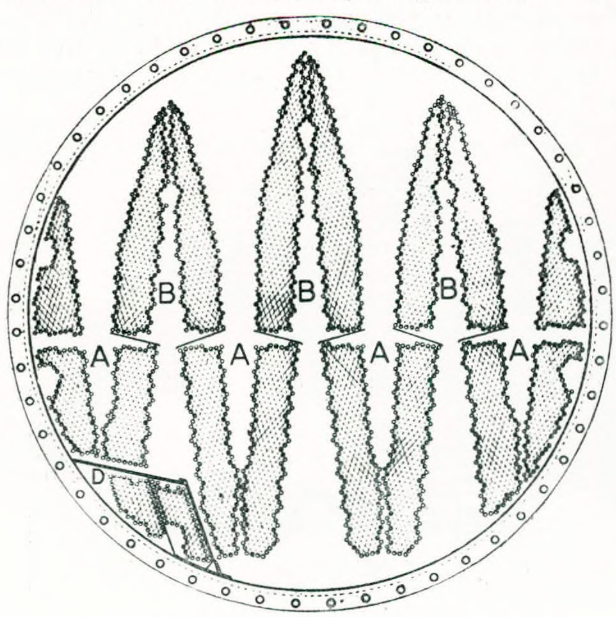


FIG. 11.—Delas condenser.

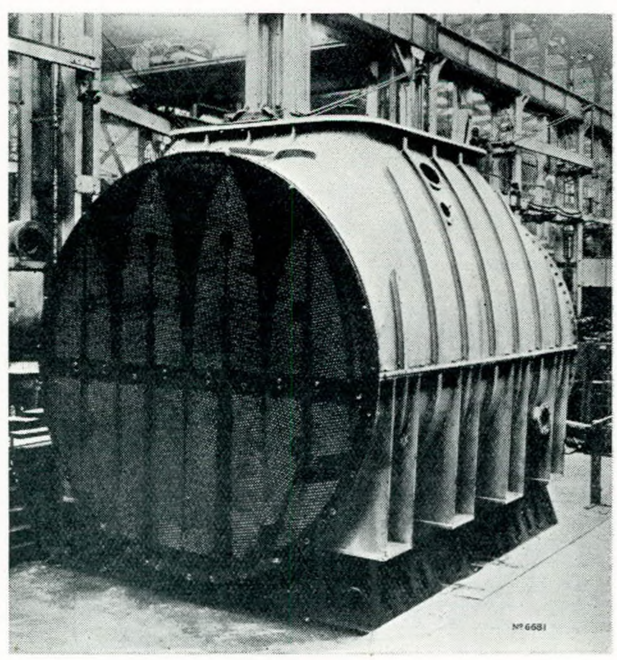


FIG. 12A.—50,000 kW. Mirrlees-Delas condenser.

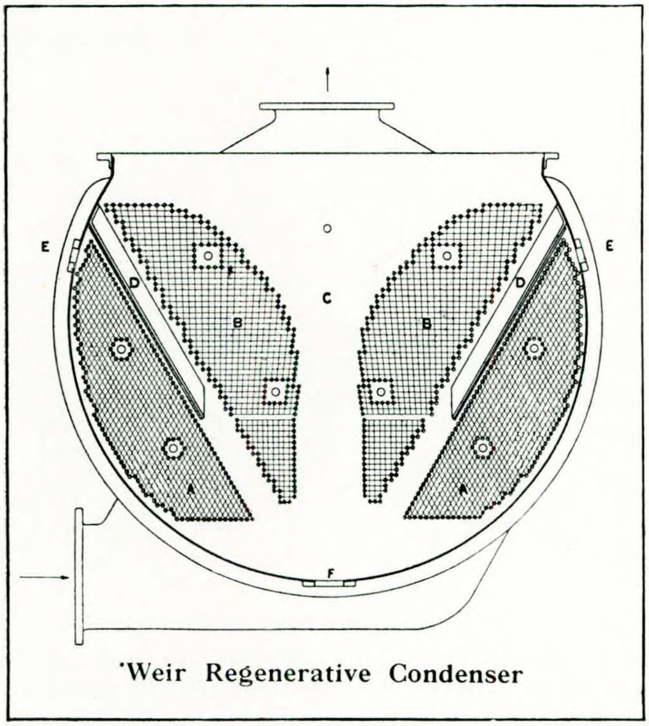


FIG. 13.

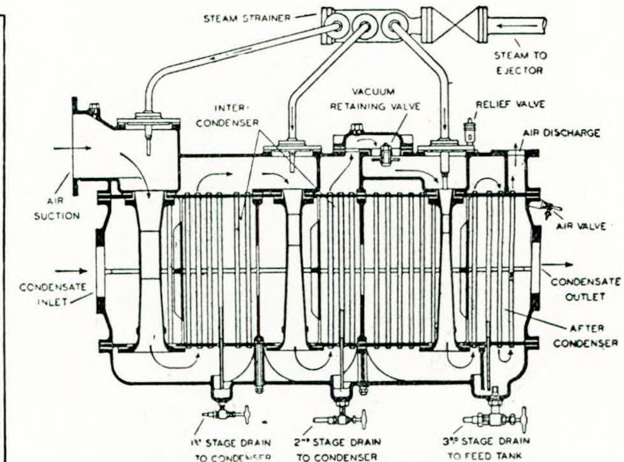


FIG. 15.—Three-jet air extractor.

contact with the condensate and reheats it to full vacuum temperature. The pitching of the first tube banks is very open and rectangular. The air discharges are high up on the sides and behind closely pitched tube banks. The small amount of condensate returning from these is quickly reheated by the exhaust steam.

So far little has been said about air. Air was a terrible trial to engineers in the past; it not only robbed them of part of their vacuum, but their efforts to extract it from the condenser had the effect of supercooling the condensate, and it got into the boilers and set up corrosion. Sir Charles Parsons was, I think, the first person to tackle the problem seriously. Parsons' vacuum augmentor was a method of extracting air by steam jets.

Fig. 14, showing Weir's Paragon air pump, illustrates the system of Parsons' augmentor. The ejector condenser in Parsons' arrangement had sea water circulation and the *air was discharged* to the air pump suction. In this arrangement the air and condensed steam from the jet are discharged above the bucket valves of the air pump. The augmentor condenser is circulated by the condensate, so that the heat in the exhaust from the jet is recovered.

In a modern high class marine condenser plant as shown in Fig. 15 there is no air pump at all. The air is discharged by a multiple steam jet air extractor, with inter-coolers between the jets. The intercoolers are circulated by condensate to recover the heat in the jet exhausts. The water is extracted

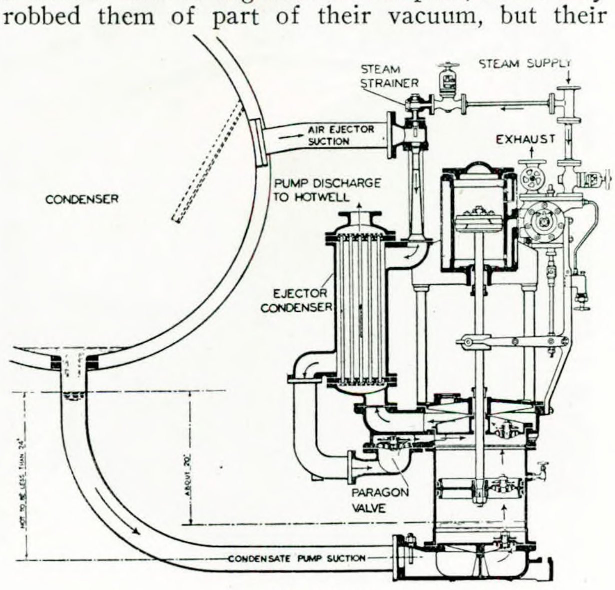


FIG. 14.—Paragon air pump with air ejector.

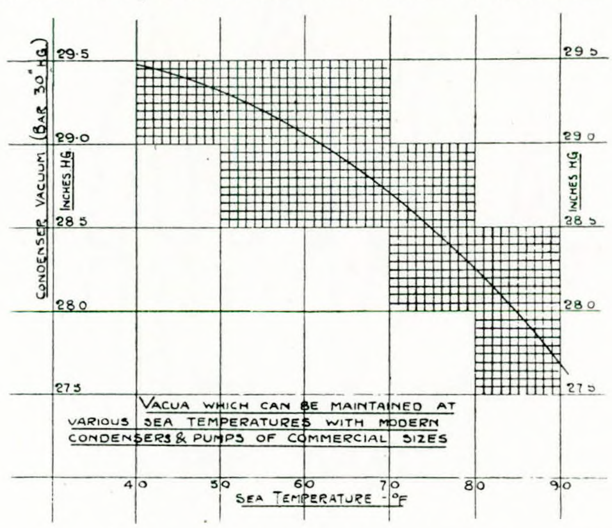


FIG. 16.—Mr. J. Johnson's vacuum—sea temperature curve.

from the condenser by a separate water extraction pump which also does duty as the circulating pump of the air extractor's intercooler. It will be shown later how these are incorporated in the feed system.

The result of these improvements in condensing plant has been to increase the vacuum which can be maintained with a given temperature of sea water to the extent of about $\frac{1}{2}''$ coupled with an increase in the temperature of the condensate of at least 10° F. and the almost complete elimination of air in the feed water. This result has been brought about by the requirements of the turbine. The reciprocating engine was unable to make use of a higher vacuum than 28″, but the turbine can make full use of any vacuum which is available.

It is perhaps not generally realized how much the modern high pressure boiler is dependent on the condenser for its success. Pure air-free feed water is a vital necessity for high pressure and high temperature steam. In the olden times if the condenser tubes leaked the worst of them were plugged and one carried on—if the boiler got too salted, one could always make it a bit fresher by partially blowing down the boilers and filling up with fresh water. In fact there was a formula by which one could calculate how much water had to be blown out in order to recover the moderate degree of freshness that was considered necessary.

A modern high pressure boiler requires 100 per cent. distilled water. There can be no leaky tubes in the condenser. Until an alloy was discovered suitable for making condenser tubes of the standard that was needed, the boiler makers had to mark time. Cupro-nickel appears to have solved that problem satisfactorily.

But to return to the vacuum, I said that it had been improved to the extent of $\frac{1}{2}''$. It will be interesting to know what vacuum can be maintained at various sea temperatures in modern condensers. *Fig. 16 gives this information; it is a photograph of a diagram given to me by Mr. J. Johnson, of Canadian Pacific Steamships Ltd. It was afterwards published in the Transactions of the Institution of Naval Architects. Although it is now five years old, it will still do duty to-day. I would call your attention to the end of Mr. Johnson's title —"pumps of commercial sizes". If a higher vacuum was desired one could have it, but it would be a luxury, which, according to the dictionary, is an "extravagant indulgence".

While studying this diagram for the purpose of this lecture it occurred to me not only that the ocean temperatures are very constant but that they are well known. Many years ago, as a result of the Challenger Expedition, a map was published

* Trans. Inst. Naval Architects, 1932, Vol. LXXIV, p. 56.

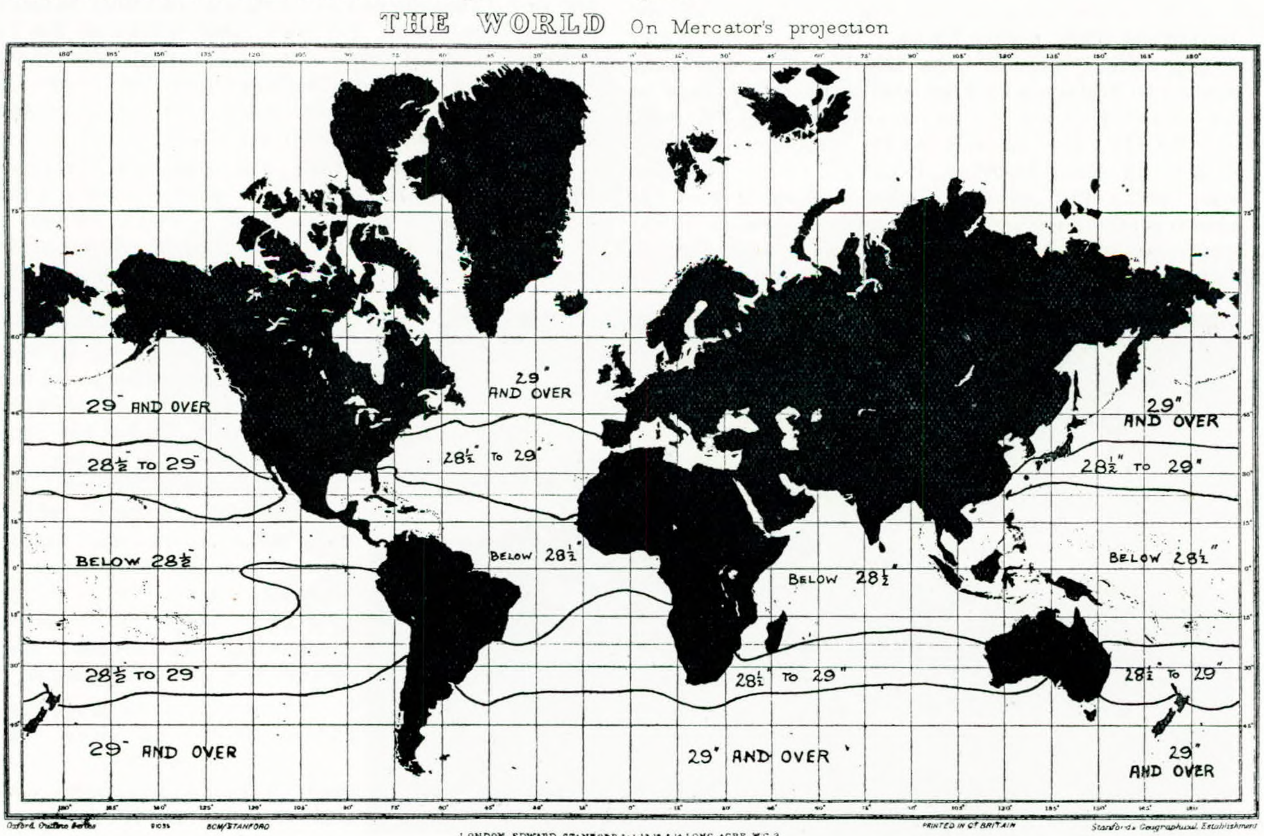


FIG. 17.—Isovacs.

showing the isotherms of the oceans, and this has been confirmed by many subsequent observations. With the aid of the Challenger's map and Mr. Johnson's diagram I have prepared a map (Fig. 17) showing the "Isovacs" of the oceans. From this it will be seen that whereas the reciprocating engine can have its 28″ vacuum practically all over the world the back pressure in a turbine is a question of geography.

It will be noticed that the $28\frac{1}{2}''$ and 29″ isovacs are missing in the northern part of the Indian Ocean. There is a large area extending from the Bay of Bengal to New Guinea in which during certain portions of the year it would be difficult to maintain 28″. However, this district cannot properly be described as open ocean.

The curious kink in the $28\frac{1}{2}''$ isovac to the west of South America is interesting. It is thought to be due to a cold sub-marine current from the Antarctic ice which rises to the surface on the West Coast.

The map being on Mercator's projection gives an exaggerated impression of the area of over 29″. Nevertheless it contains some very important trade routes. Northern Europe, with the more important North American ports on the East Coast, and Northern China and Japan with the similar North American ports on the West Coast, are all within this area.

With the exception of the foregoing all the important trade routes lie very largely in hot water areas, which include both the Canals. The mail routes to India and China and nearly the whole of the Australian mail route are outside the 29″ area.

In the days of sail, ships "ran their eastings down" in the "Roaring Forties", chiefly because they got a fair wind but also because it was the shortest distance. The steamers abandoned this route because it was a bit too boisterous for the heavily laden cargo vessel. It will be interesting to

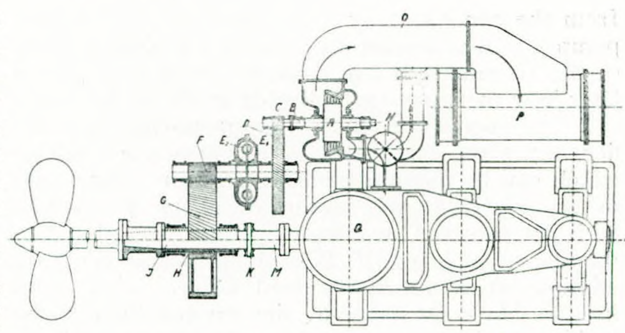


FIG. 19.—Bauer-Wach displayed plan.

see if the turbine steamers take it up again for the sake of 29″ of vacuum.

So much for the part played by the condenser in reducing the heat discharged to the sea. I think it can be agreed that the condenser designers have done all that is possible. There is practically no resistance through a modern condenser and the condensate temperature is well up to the temperature of the vacuum. Indeed one sometimes hears of the condensate being above the vacuum temperature. In that case one fails to see why it does not re-evaporate in the condenser.

What can be expected in the matter of improving the exhaust end of a *reciprocating* engine? The answer is—nothing, not only because any elaboration of the ports and valves of the l.p cylinder is not likely to repay the cost, but chiefly because the exhaust turbine will carry on the expansion of the steam when the reciprocating engine is no longer able to use it.

The exhaust turbine was originally proposed by Sir Charles Parsons. The pioneer ship of this type was the "Otaki", built by Messrs. Denny of Dumbarton in 1908, the turbine being designed by Sir Charles. The "Otaki" showed about 8 per cent. saving in fuel consumption as compared with two sister ships for the same owners. In spite of this, these owners did not repeat the experiment, although other owners did so, the most notable examples being the "Olympic" and "Titanic". In these ships the exhaust turbine took steam from two reciprocating engines and drove a separate propeller without gearing, hence no serious mechanical difficulties were involved in the combination.

As soon as turbines were able to use high pressure steam as efficiently as reciprocating engines the combination engine had no chance of competing with the straight turbine. The combination of reciprocating

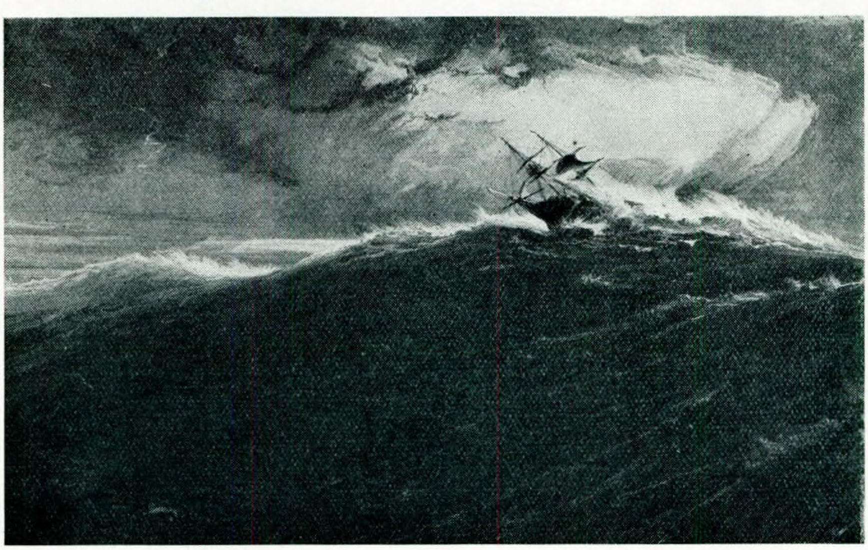


FIG. 18.—The "Roaring Forties", by William Daniel, R.A.

engine and turbine, however, was destined to re-appear a few years later in a different form. About ten years ago a satisfactory proposal was put forward for converting an existing reciprocating steam engine into a combination of reciprocating engine and exhaust turbine. It met a great temporary need because trade was falling off and older steamers were finding great difficulty in competing with newer Diesel-engined vessels on a falling market. This proposal was the combination of several inventions by Dr. Gustav Bauer and Herr Hans Wach, generally known as the "Bauer-Wach Gear".

Fig. 19 shows the various parts of this combination. Q is the l.p. cylinder of a triple expansion engine, which exhausts through a change-over valve N either to the condenser P or through an exhaust turbine A to the same condenser. Attached to the turbine shaft is a pinion C which drives the spur-wheel D. D is attached to the same shaft as the driving part of the clutch E.1. This is the hydraulic clutch, which if it is filled with oil drives E.2. E.2 is attached to the same shaft at the pinion F. F drives the spur wheel G which is attached to the quill shaft H and H is attached at the muff coupling to the main shafting. The several parts between B and H are enclosed in one casing. The arrangement is here displayed diagrammatically. The spur wheel D and the part E.1 of the clutch are as a rule combined in one unit.

The clutch E requires to be filled with oil in order that it may operate; before considering the action of the clutch this should be carefully noted. If the change-over valve was opened to the turbine while the clutch was empty the turbine would start unloaded and would be seriously damaged. This risk is avoided by operating the valve by a servo-motor which uses the same oil as the clutch so that until the clutch is full and under pressure no pressure can come on the servo motor, which has the further advantage that the turbine will be running before the steam comes on.

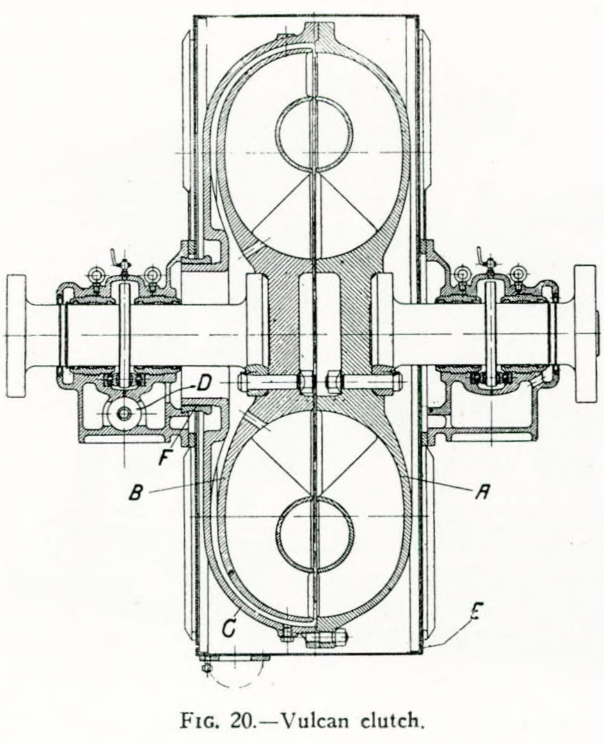


Fig. 20.—Vulcan clutch.

Fig. 20 shows the construction of the Vulcan Clutch. The clutch is in two halves which are, so far as their internal passages are concerned, exactly similar, and they are symmetrical about any diameter, thus either part can be driver and will drive in either direction. The two parts of the clutch form together an annulus with a central core round which the working fluid (oil) flows. They rotate together about a central axis, the shaft; there is a slight slip between them, about 3 per cent., so that the speed of the driver is slightly higher than that of the driven member.

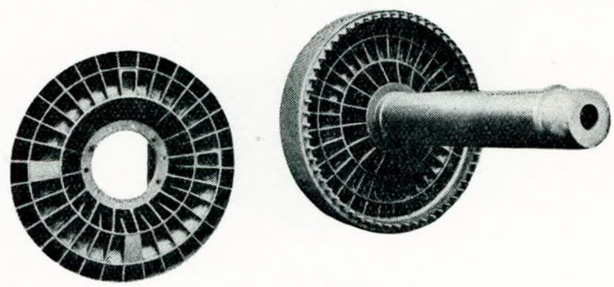


Fig. 21.—Photograph of Vulcan clutch.

Both parts have radial vanes which connect the central core with the outer walls, these vanes are quite flat and as I have just said radial. At the outer edges the vanes have a fairly small clearance, about 10mm., but the alternative inner edges are cut back a long way as shown in Fig. 21. On account of the slight difference in the speed of rotation between the two halves there is a difference in the pressure due to centrifugal force, therefore a flow is set up round the central core. This flow is not resisted by the driven member, the only resistance to be overcome is frictional, due to the viscosity of the working fluid and the liquid friction against the walls.

The vanes, however, impart another type of flow to the fluid, a circular motion about the central axis. This is not a motion relative to the parts of the clutch, but in conjunction with them, and as the oil passes from the driver to the driven member it carries this motion with it, and compels the driven member to rotate, as well.

Fig. 22 is a photograph of a model showing the first of these rotations. It will be noticed that the inner passage is much wider than the outer; on account of this it is often erroneously supposed that the working fluid is accelerated and decelerated as it passes round the inner core. This is not the case; the velocity of the oil about the inner core

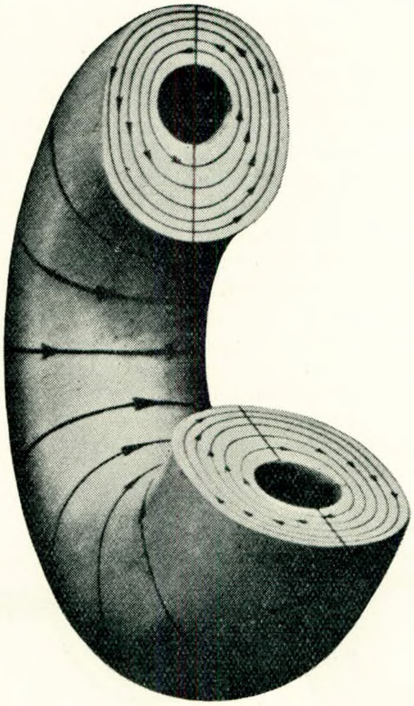


FIG. 22.—Vulcan clutch oil flow.

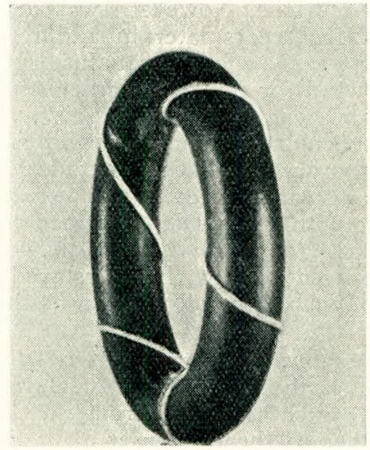


FIG. 23.—Vulcan clutch combined oil flow.

is perfectly uniform. On account of the smaller diameter about the central axis of the inner passage it is shorter than the outer passage, and therefore has to be wider than the outer passage, so that they both have the same area.

Fig. 23 may give a better idea of the path of a particle of oil in space, but it is not quite correct because as the oil passes from the driver to the driven member it changes its velocity about the axis of the shaft, and it is this change of velocity which produces the drive.

Let us now consider the mechanical problems which necessitate what may appear to be an unnecessarily complicated mechanism to couple a turbine with a reciprocating engine. The turning moment of a reciprocating engine is not uniform; this causes the velocity of rotation of the shafting to pass through a regular series of variations during each revolution. This is known as the "cyclical fluctuation of speed". This can be damped out by a fly-wheel, but a marine engine has no fly-wheel. The turbine on the other hand is a constant-speed machine and furthermore it is developing only about one-third of the power of the reciprocating engine. Therefore it will be the reciprocating engine which will set the pace, with disastrous results

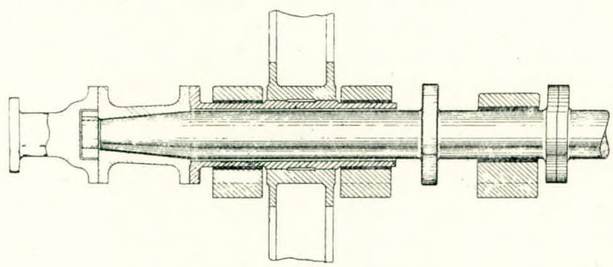


FIG. 24.—Quill shaft.

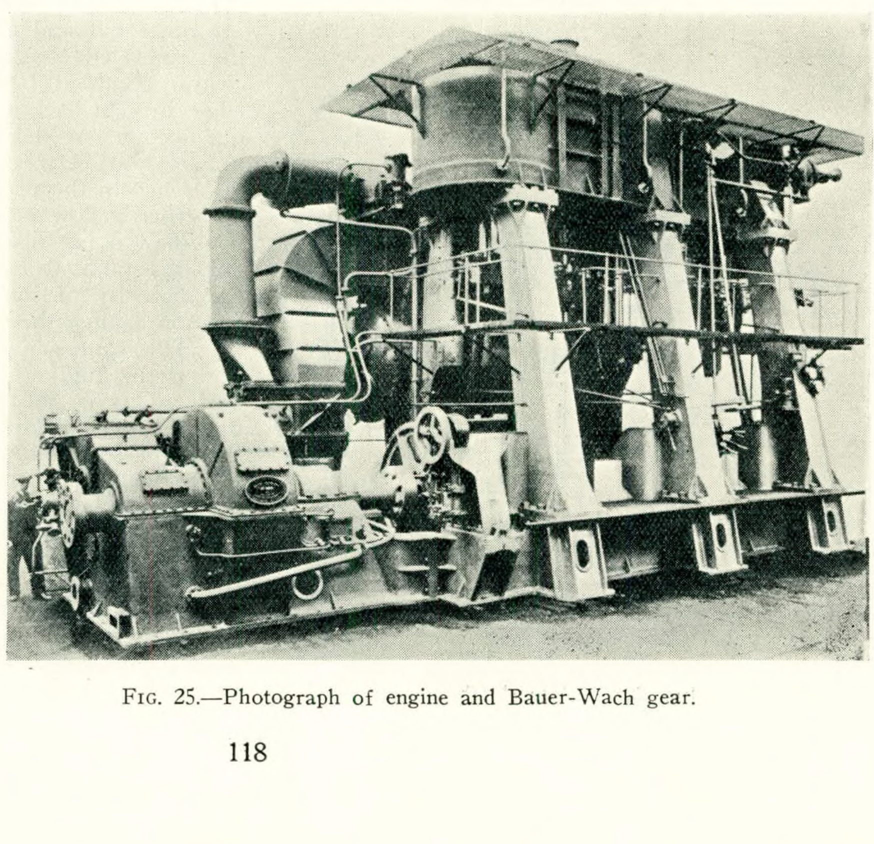


FIG. 25.—Photograph of engine and Bauer-Wach gear.

to the gearing if the turbine and the reciprocating engine are directly coupled. In this combination the elastic Vulcan clutch allows sufficient slip to cushion these shocks. There are other types of slipping clutch on the market, also spring buffer connections. But this is not the only difficulty which has to be overcome in connecting a turbine with a reciprocating engine. The final spur-wheel of the gearing must connect with the main shafting, and the turbine must be near the l.p. cylinder and the condenser, therefore this spur-wheel must be at the forward end of the line shafting, also near the l.p. cylinder and the crankshaft. If it was attached direct to the shaft itself it would be subject not only to the general wear down of the crankshaft, but to a regular oscillation due to the variation of the load on the crankshaft. For this reason the spur-wheel is carried on a separate hollow shaft, called the quill shaft (Fig. 24), which runs in its own bearings and is connected to the line shafting at its after end as far away from the crankshaft as possible. This eliminates all dangerous movement from the gearing, but concentrates a certain amount of stress in the muff coupling, which it has not been easy to overcome.

Fig. 25 is a photograph of a triple expansion engine combined with a Bauer-Wach exhaust turbine.

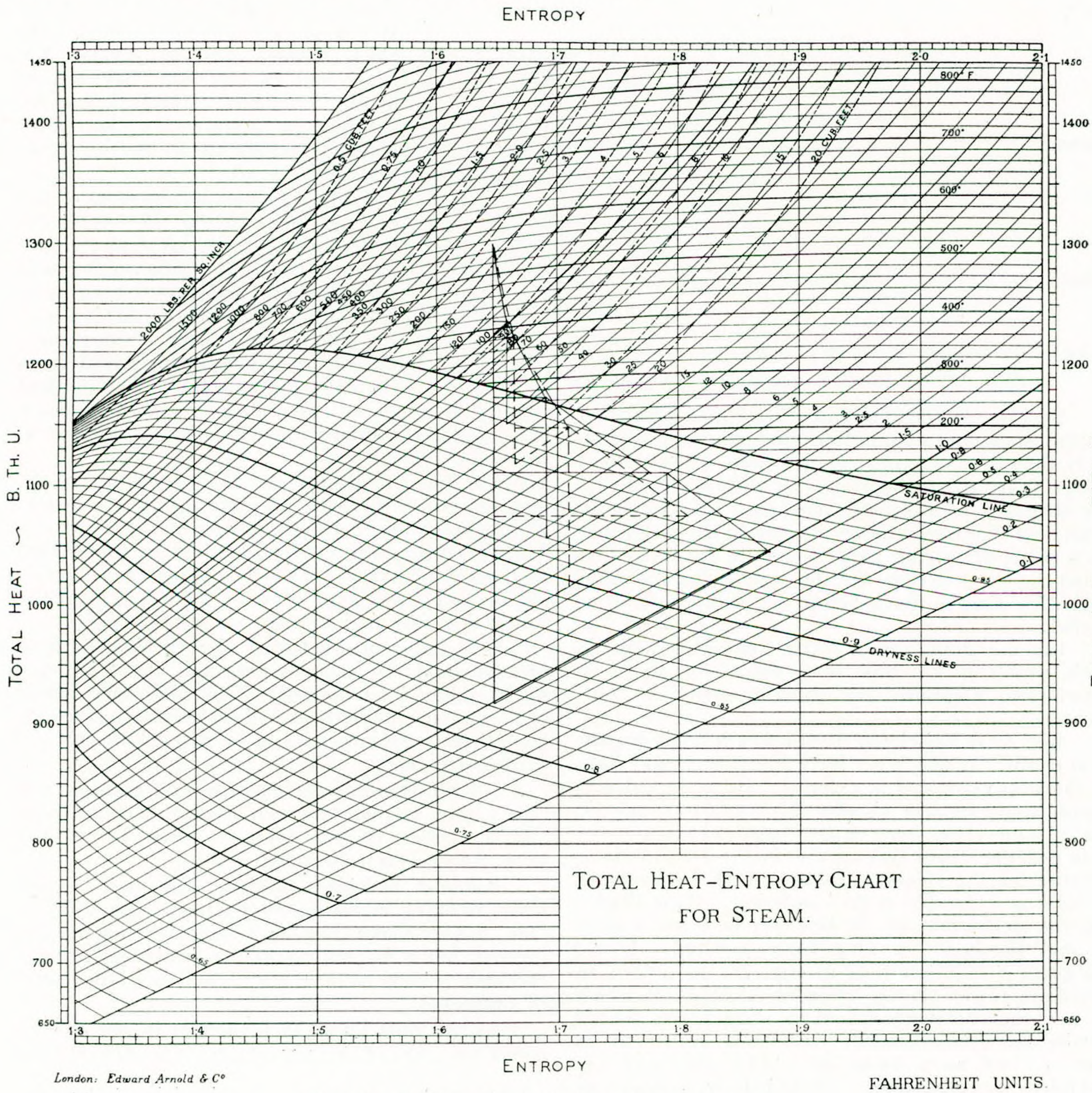


FIG. 26.—Mollier chart with Bauer-Wach diagram.

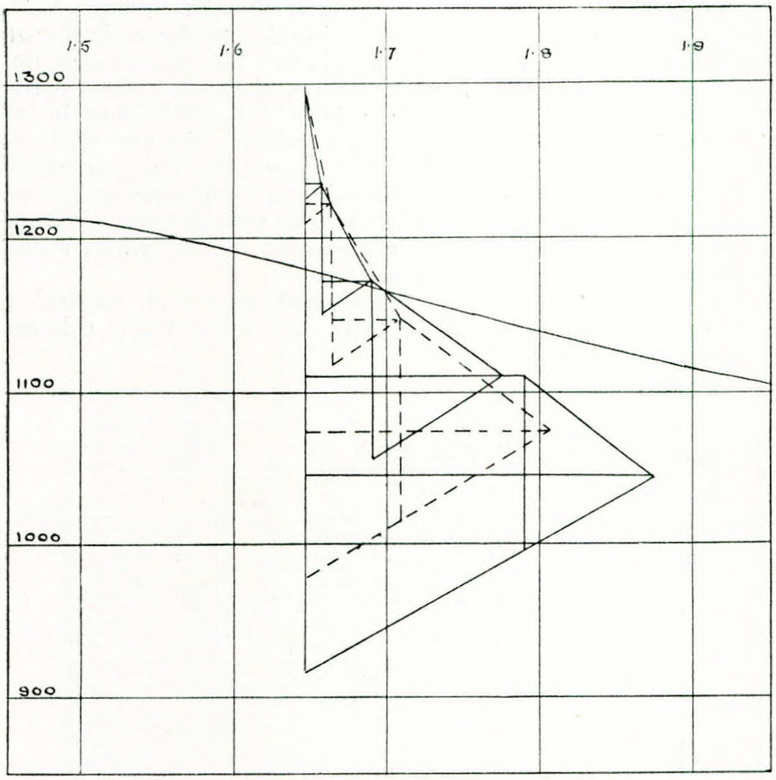


FIG. 27.—Bauer-Wach diagram without Mollier chart.

In Fig. 26 the diagram of a triple expansion engine is shown in chain lines and superimposed on it is that of the same engine when fitted with a Bauer-Wach exhaust turbine. It will be agreed that the diagram is a little confusing. Of course in practice a diagram is never drawn on the chart, but on a piece of tracing paper over the chart, so that when the chart is removed the result is as shown in Fig. 27. This is much better but still a little congested.

Let me show you a better method of dealing with problems of this sort, especially those in which two diagrams are superimposed for comparison.

Fig. 28 is a skeleton diagram of a triple expansion engine, in which we have the steam conditions at the stop valve and at exhaust. Here the exhaust steam engineer has all the information required; for instance, the heat drop (from which may be obtained the steam rate of the engine), the dryness fraction of the steam at exhaust, and the heat per lb. of steam rejected to the condenser. By multiplying the steam rate by the heat per lb. it is possible to find the heat per i.h.p./hour rejected. If we complete the diagram, as in Fig. 29, we note with satisfaction that the l.p. cylinder takes steam which is not superheated, because a slide valve always works more satisfactorily with some water on its face to act as a lubricant. The h.p. and i.p. cylinders are, however, entangled in a mass of lines; most of these are of no value to our present problem. It would be better to expunge all that portion of the chart inside the black frame and to replace only those lines which are going to be useful.

The result is shown in Fig. 30, which gives a much clearer view of the engine. Again we note with satisfaction that the l.p. cylinder takes steam which is above atmospheric pressure, thus preventing the l.p. valve spindle gland leaking air to the condenser.

But what is the use of all this chart surrounding our diagram? We have all we want inside the black frame. Let us abolish the chart and enlarge our diagram, as in Fig. 31. Here we have a diagram without any unnecessary chart lines which will give us all the information we require.

Fig. 32 is the enlargement of the tracing shown in Fig. 27. With the scale increased we can detect errors in our first trial. This engine was actually built and tested both with and without the exhaust turbine, and the data of those tests are on record. Fig. 33 is a diagram prepared from the data as published.* The original engine, as in all my diagrams, is shown in chain lines, and the combination with exhaust turbine in full lines.

The final pressure of 2·2lb. shown on the diagram is not the back pressure on the l.p. piston, which scales on the indicator cards at least 3lb. The lower pressure evidently refers to some point in the exhaust pipe. The data shows a reduction in fuel rate per i.h.p. of over 25 per cent. The conversion was undoubtedly very successful.

The difference between the turbine exhaust pressure and the vacuum pressure, about $\frac{3}{8}$lb. per sq. in., should be noticed. This difference would not occur in a modern plant because the reciprocating engine condenser would be altered to suit the turbine exhaust.

Notwithstanding the expense of a plant of this type a large number has been fitted to existing ships, also a number of new ships are now being built with Bauer-Wach turbines.

Exhaust turbines produce the best results when working in conjunction with a high vacuum. One might make the same remark about any turbine, but it is particularly applicable to exhaust turbines, which have a very small working range of pressures.

As we now know that vacuum is not ours to command, we must accept the vacuum that is

* Trans. Inst. of Eng'rs. and Shipbuilders in Scotland, 1927, Vol. LXXI, p. 255.

available, and that is more likely to be 28″ than 29″. Therefore in the diagrams I am about to show the vacuum is always 28″.

There are other methods of applying the power derived from an exhaust turbine to the main engine. One of them is to compress the steam after it is partially expanded. This system is much cheaper to fit than the Bauer-Wach, but it has the disadvantage that the heat units returned to the steam are subject to the conversion losses in the cylinders which means that not more than 70 per cent. of it will be re-converted into work.

Fig. 34 is a diagram showing this system applied to a triple expansion engine taking steam at 220lb. per sq. in. and 550° F. and exhausting against a vacuum of 28″. The heat returned to the steam is 52·6 B.T.U. per lb.; against this the back pressure of the engine has been increased from 3 to 7lb. per sq. in. There would be a reduction in the steam rate by this conversion, but it would be very small and would not repay the cost of the additional plant.

Fig. 35 shows this system applied to a wet steam engine. In this case we have to get rid of the surplus water before compressing the steam. We find that we can only return 50 B.T.U. per lb. and the weight of steam left to be compressed is 91·7 per cent. of that passing the stop valve. It

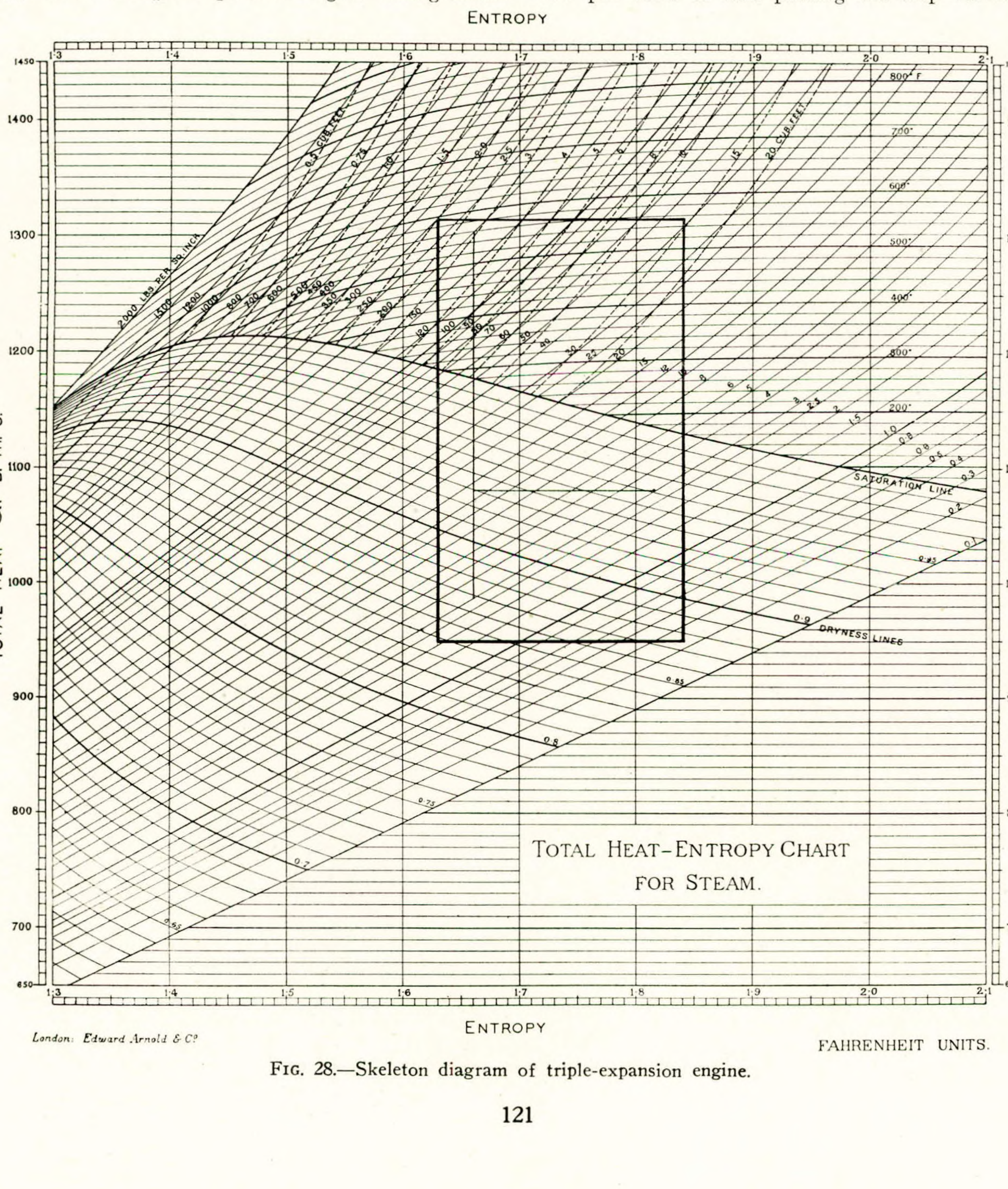


FIG. 28.—Skeleton diagram of triple-expansion engine.

would also be advisable to remove the water from the l.p. exhaust, so that the turbine will have only 86·35 per cent. of the original steam. The water from the l.p. exhaust is, of course, below atmospheric pressure, so it must be blown down to the condenser. It could be passed through a drain cooler to recover the heat.

When comparing the heat drop with that of the original engine, only 91·7 per cent. of the drop in the i.p. and l.p. cylinders must be included. This shows a much better economy than the superheated engine. A reduction of about 11½ per cent. in the steam rate could be expected and a small amount of feed heating from the water separated.

An interesting problem which arises from this diagram is to decide what to do with the water separated from the h.p. exhaust steam. This water has a temperature of 285° F. and is under pressure of 54lb. per sq. in. and contains 255 B.T.U. per lb. If it is returned to the engine at some point where the pressure is lower, part of the water will re-evaporate. The i.p. exhaust is a possible point at which to introduce this water. Let us see what will happen if we do so.

Fig. 36 is a diagram which illustrates this change. The dryness fraction of the l.p. cylinder changes from 98·2 per cent. to 90·47 per cent. We need not be concerned about that, because it will

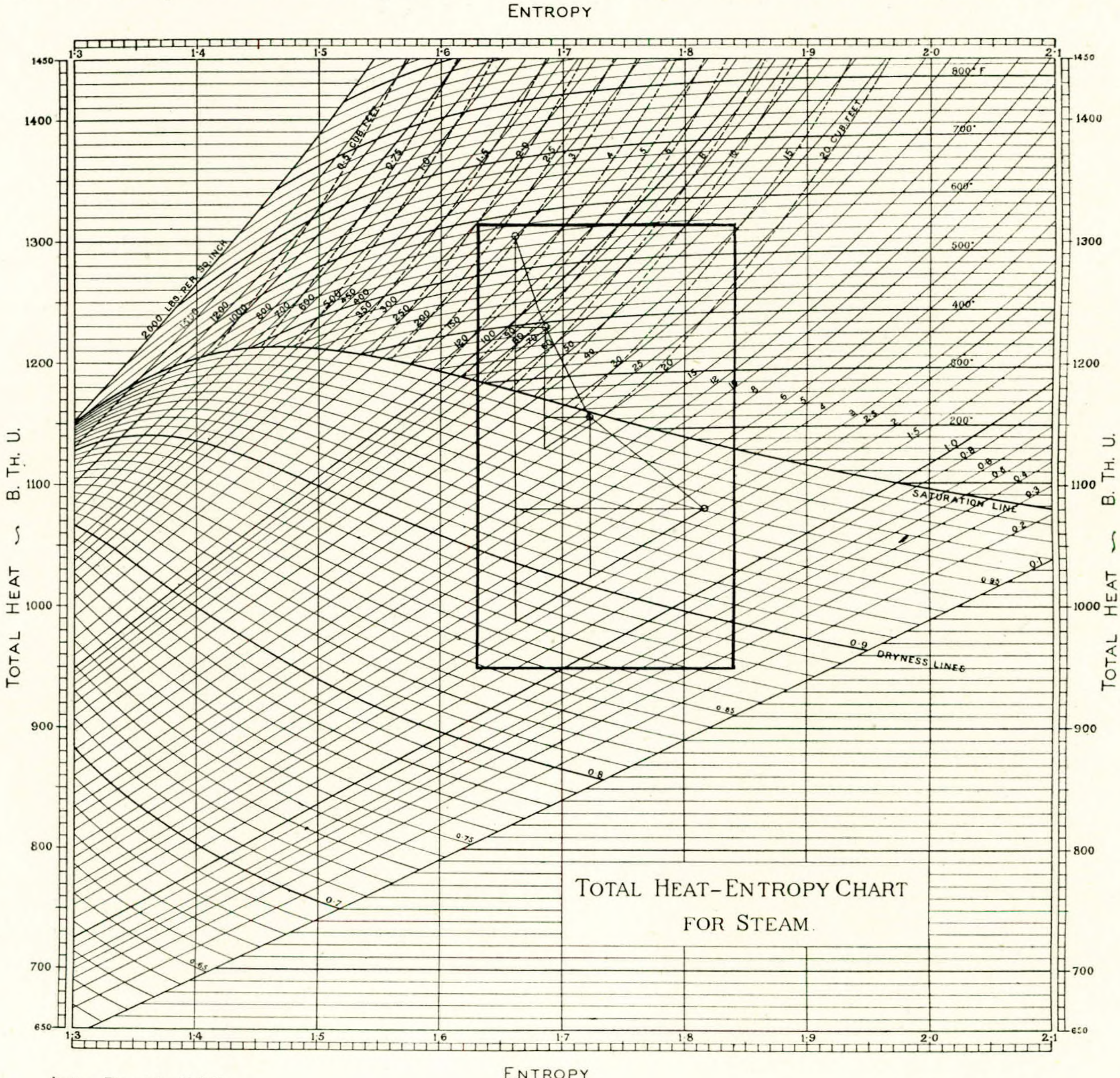
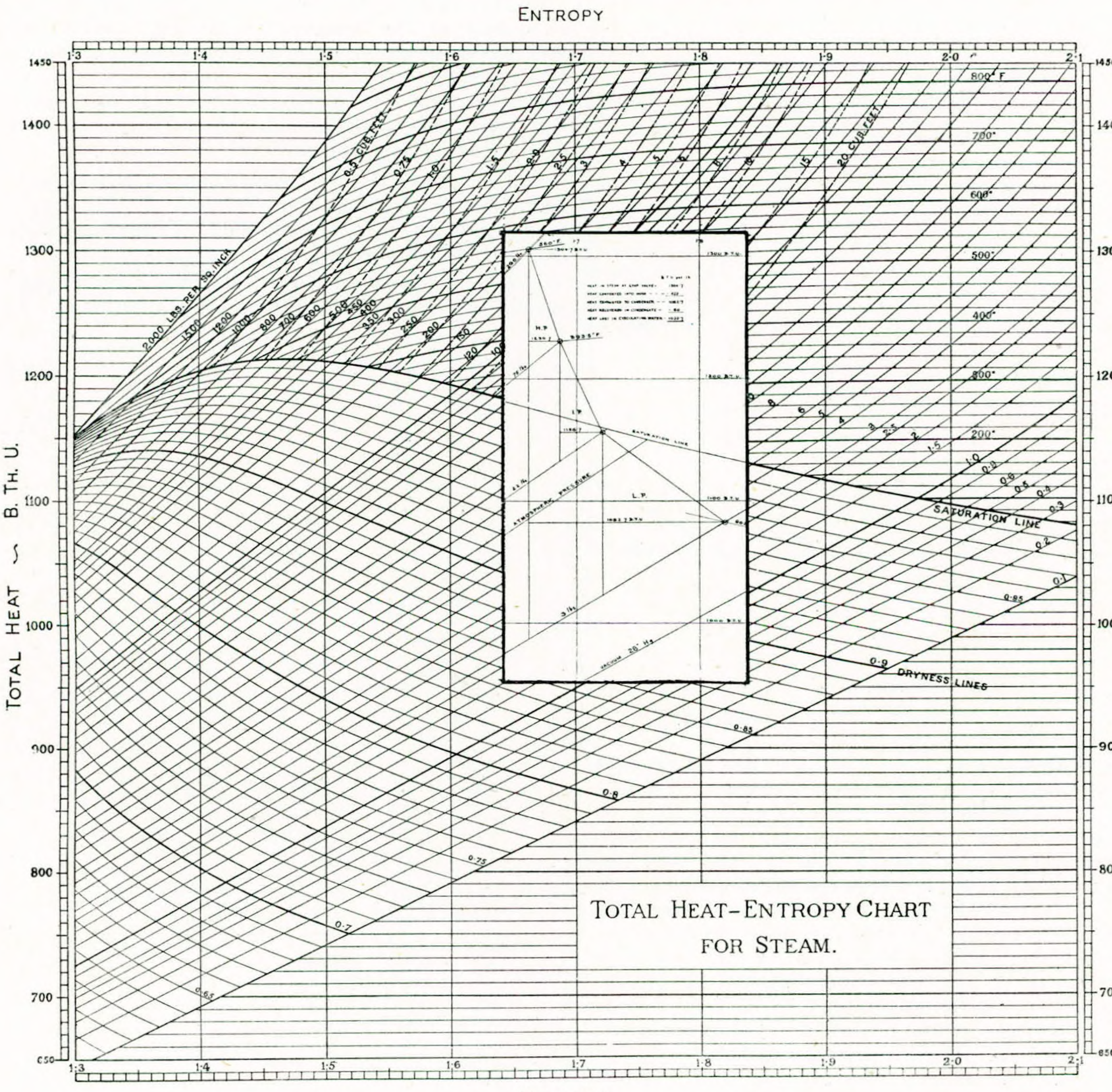


FIG. 29.—Complete diagram of triple-expansion engine.

be noticed that the steam at l.p. admission is drier, is at a higher pressure, and contains more heat than it did in the original engine. How much more work will be obtained from the l.p. cylinder owing to the addition of this hot water? The heat drop of the l.p. with added water is 65 B.T.U. per lb. and previously it was 70 B.T.U., but the 70 has to be multiplied by ·917, so it is only worth 64·2 B.T.U. Hence there is a gain of only 0·8 B.T.U. per lb. Therefore it is not worth doing.

Let us try putting it in at the l.p. exhaust to see if we can get some more work in the turbine from it. We find that the exhaust steam available for the turbine changes from 94·2 per cent. dry to 87·35 per cent. dry, which is exactly the same dryness as if the water had been returned to the i.p. exhaust and then expanded in the l.p. cylinder. This excess water must necessarily be separated and we get the same turbine conditions as before. Of course, no gain will ever result from returning only water to an engine.

Another system, shown in Fig. 37, is to generate electric current with the exhaust turbine and re-heat the steam electrically after it is partially expanded. As the former system appeared to be so much better with wet steam than with



London: Edward Arnold & Co. FAHRENHEIT UNITS.

FIG. 30.—Complete diagram of triple-expansion engine with rectangle cleared.

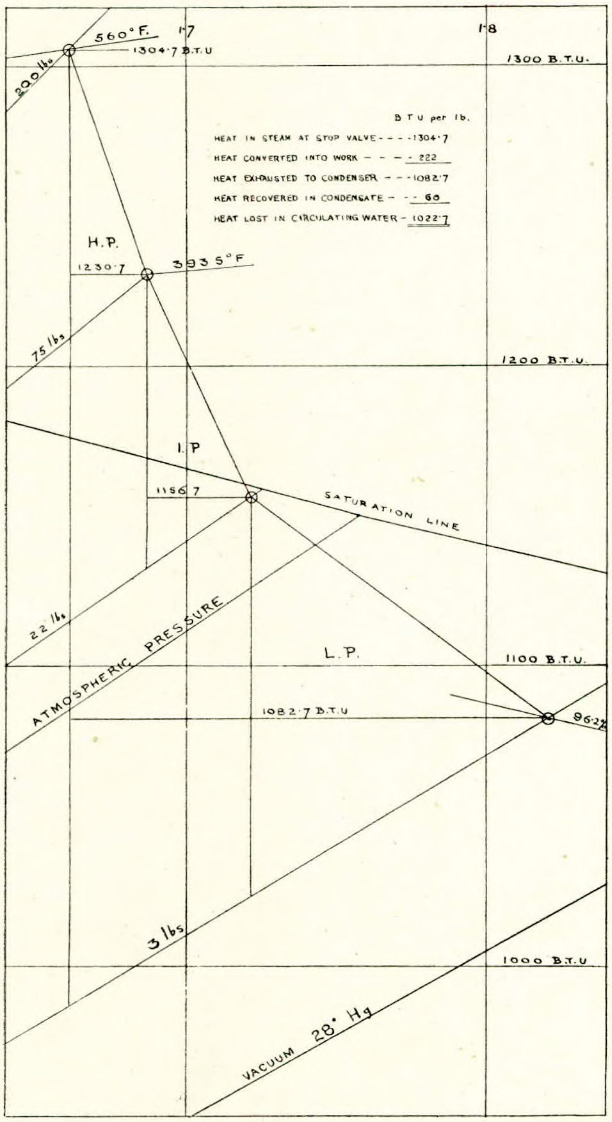


FIG. 31.—Rectangle of Fig. 30 enlarged.

superheated steam, it seemed desirable to try this arrangement under the same conditions. The re-heating added 49 B.T.U. There appeared to be no advantage to be gained by blowing out the water at the h.p. exhaust, but at the l.p. end with the steam 10 per cent. wet it did seem advisable to do so. So far as the power to be derived from the turbine is concerned, it makes no practical difference whether the water is blown out or not, but it is better to do so, because water damages the blades. This result was disappointing, as although there was an increase in the heat-drop it was very small.

There are a great many different ways of employing the power derived from an exhaust turbine. A very obvious one is to generate electricity for lighting the ship.

Fig. 38 shows one system from a different angle. Here the turbine drives the feed pump. If the turbine is cut out for manœuvring the feed pump ceases to take water from the feed tank, but the air pump continues to discharge to the tank, which consequently overflows to the overflow feed tank. The float control in this tank automatically starts the direct-acting stand-by feed pump, which continues to feed the boiler until the turbine starts again, when the pump automatically stops.

We are now in a position to take stock of what has been accomplished in recent years to improve the efficiency of marine engines at the exhaust end alone.

The vacuum has been increased without any reduction of the temperature of the condensate, and the heat units per pound of steam exhausted to the condenser have been reduced. This means that more heat units per pound of steam have been converted into work, and therefore the steam rate of the engine has been reduced.

The effect of these improvements has been a reduction of the fuel rate by some 20 per cent., principally due to the reduction of the heat discharged to the condenser.

Let us consider a concrete case—the engine which was fitted with a Bauer-Wach turbine.

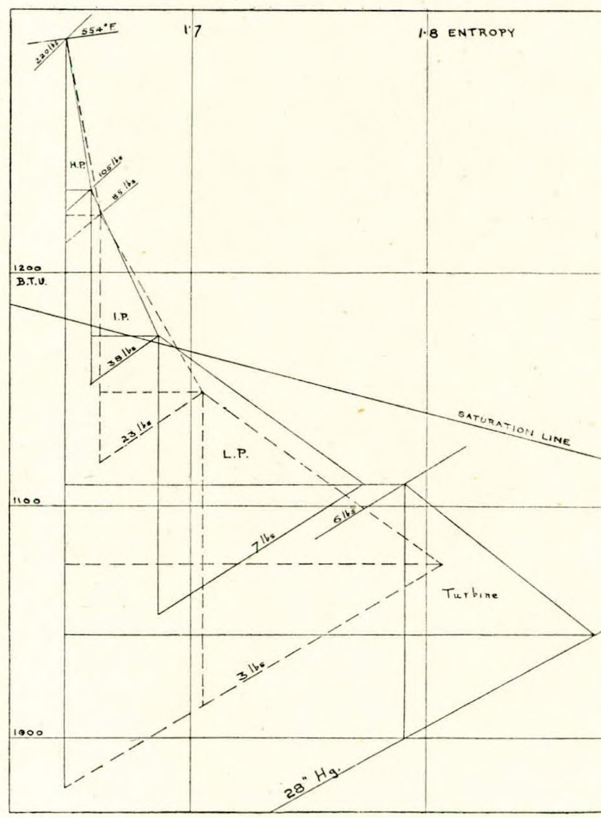


FIG. 32.—Enlargement of Bauer-Wach diagram.

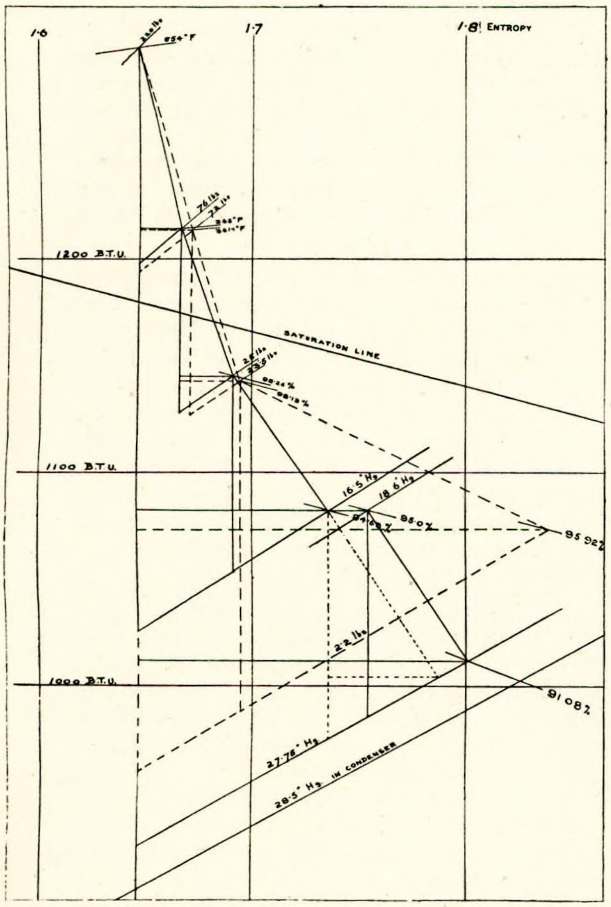


FIG. 33.—Bauer-Wach diagram from published data.

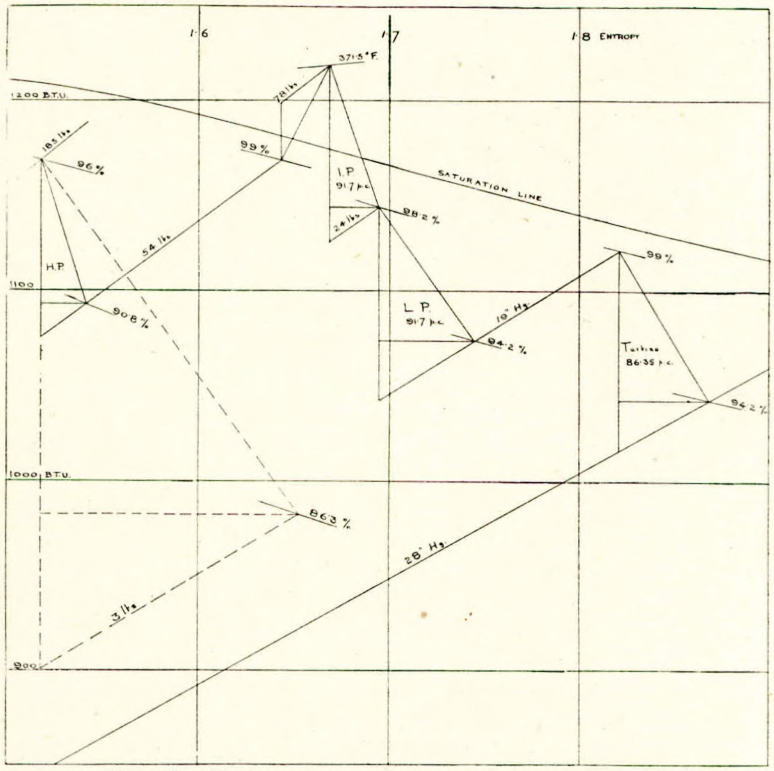


FIG. 35.—Turbo-compressor diagram—saturated steam.

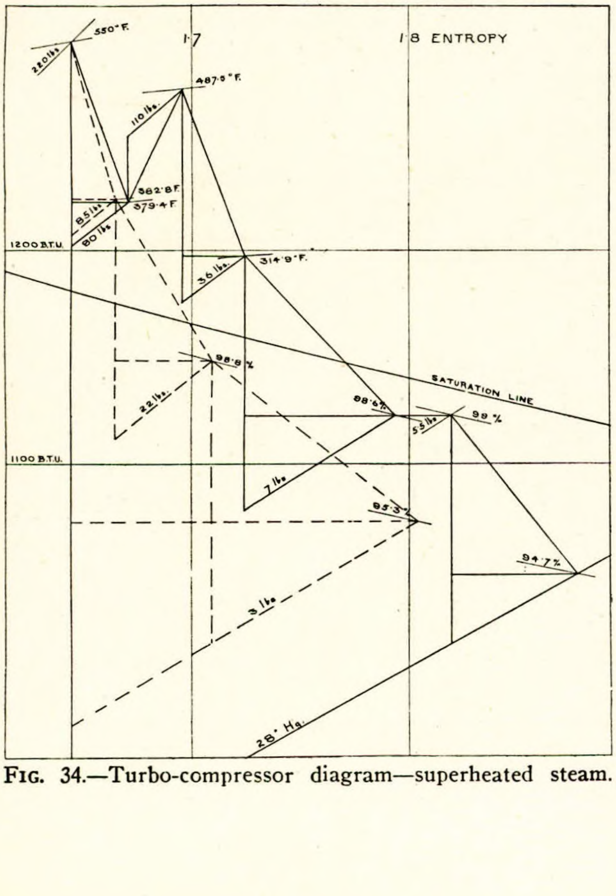


FIG. 34.—Turbo-compressor diagram—superheated steam.

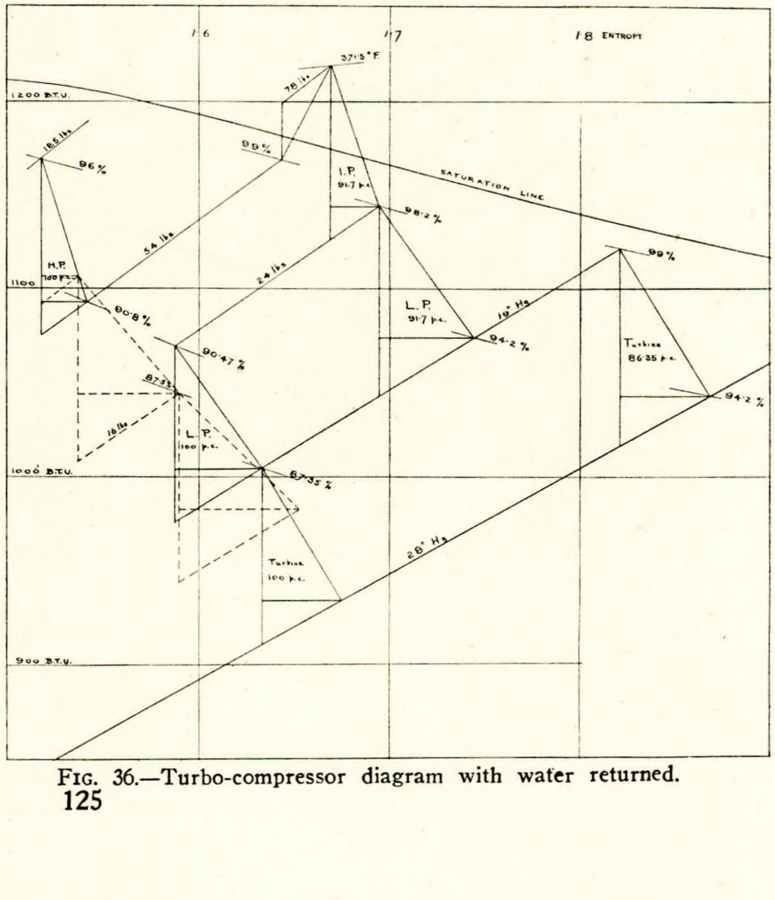


FIG. 36.—Turbo-compressor diagram with water returned.

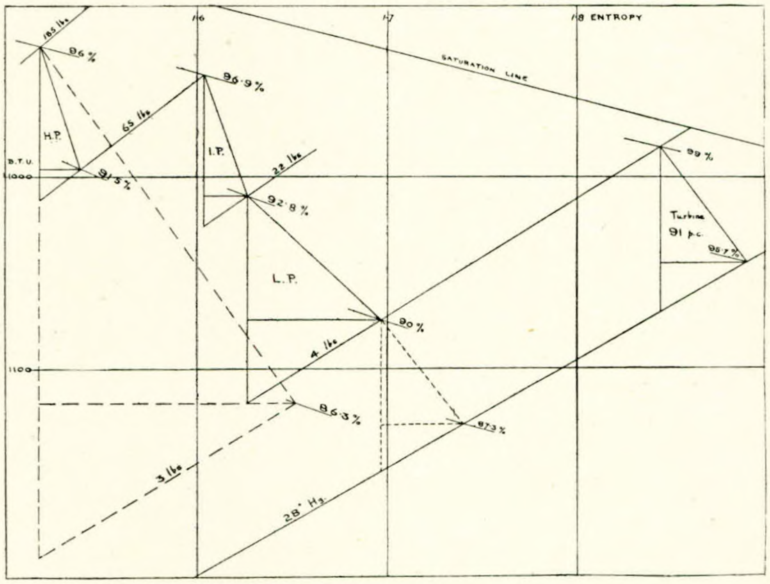


FIG. 37.—Turbo-electric reheater diagram.

The actual heat drop
before conversion was 226 B.T.U. per lb. steam
The actual heat drop
after conversion was 288 „ „
— „ „
Difference ... 62 „ „

Dividing the constant 2,546 by these two heat drops we find :—

Steam rate before
conversion ... 11·25 lb. steam per i.h.p./hour
Steam rate after
conversion ... 8·82 „ „ „

This assumes that a heat unit in a turbine has the same value as in a reciprocating engine; actually it has a greater value, so the error is on the right side.

The total heat in 1lb. of steam passing the stop valve is 1,300 B.T.U. Therefore :—

Heat passing stop valve
before conversion = 11·25 × 1,300
= 14,625 B.T.U./I.H.P. hour
Heat passing stop valve
after conversion
= 8·82 × 1,300 = 11,466 „ „

Difference 3,159 „ „

Also the heat discharge in the circulating water before conversion was 1,024 B.T.U. per lb. of steam and after conversion it was 62 B.T.U. per lb. less, because 62 more B.T.U. were converted into work. Therefore 962 B.T.U. per lb. were discharged after conversion. Therefore :—

Total heat discharged
before conversion
= 11·25 × 1,024 = 11,520 B.T.U./I.H.P. hour
Total heat discharged
after conversion
= 8·82 × 962 = 8,485 „ „

Difference 3,035 „ „

It is not by chance that the reduction of heat passing the stop valve is approximately the same as that discharged in the circulating water. A steady flow to waste of 3,000 B.T.U. per i.h.p. hour has been arrested by fitting the exhaust turbine. 3,000 B.T.U. out of 14,600 B.T.U. = 20 per cent.

To translate this into units which are more easily appreciated, it is equivalent, in a ship of 2,000 i.h.p., to throwing a ton of good Welsh coal overboard every watch, or 6 tons per day. But this is not the end of the story; it would have been more economical to have paid for the coal and left it on shore. It cost money to carry coal, owing to the increase in the displacement of the ship.

So we see that in this ship the fitting of an exhaust turbine alone produced a saving of 20 per cent. This saving, of course, applies to the heat used by the main engine. The heat required by the auxiliaries will not be exactly the same as before conversion. The feed pump for instance will have to pump less water. The circulating pump's duty will be different. There will be less steam to be condensed, and that steam will be colder. But it is not necessarily certain that these conditions will require less circulating water. It will be noticed that the saving is principally due to the reduction in the weight of steam required to produce one h.p. hour. The reduction in heat passed to the condenser per lb. of steam is less than 6 per cent.

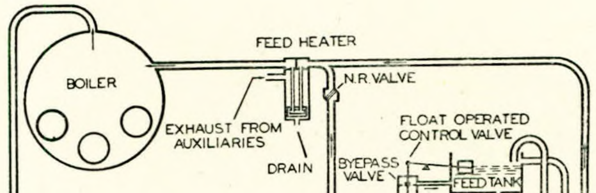


FIG. 38.—Feed system diagram with exhaust turbo-feed pump.

There is no particular virtue in the exhaust turbine. The same result would have been achieved by removing the original reciprocating engine and replacing it by a turbine. The result is due to expanding the steam to a lower pressure, which could not be done in a reciprocating engine.

I have referred at some length to adiabatic heat-drop, how to find it from the Mollier chart, and how to use it; but there is no necessity to refer to a chart to find it. It can be calculated with great accuracy from the data given in an ordinary steam table. Professor Callendar gives a formula which appears to be based on deep thermodynamic mysteries. As a matter of fact, it is nothing of the sort. Its proof depends upon the most simple straight-line geometry. So far as I know this proof is not published in any book. Once seen, the formula is easily remembered.

Fig. 39 is the diagram of an engine taking steam at 200lb./sq. in. at 560° F. and exhausting against a back pressure of 2lb. per sq. in. The adiabatic heat-drop is represented by the straight line joining the centres of the two small circles.

The back pressure line, 2lb., is drawn in heavily. Note the point where this line meets the saturation line; from this point a line is drawn at

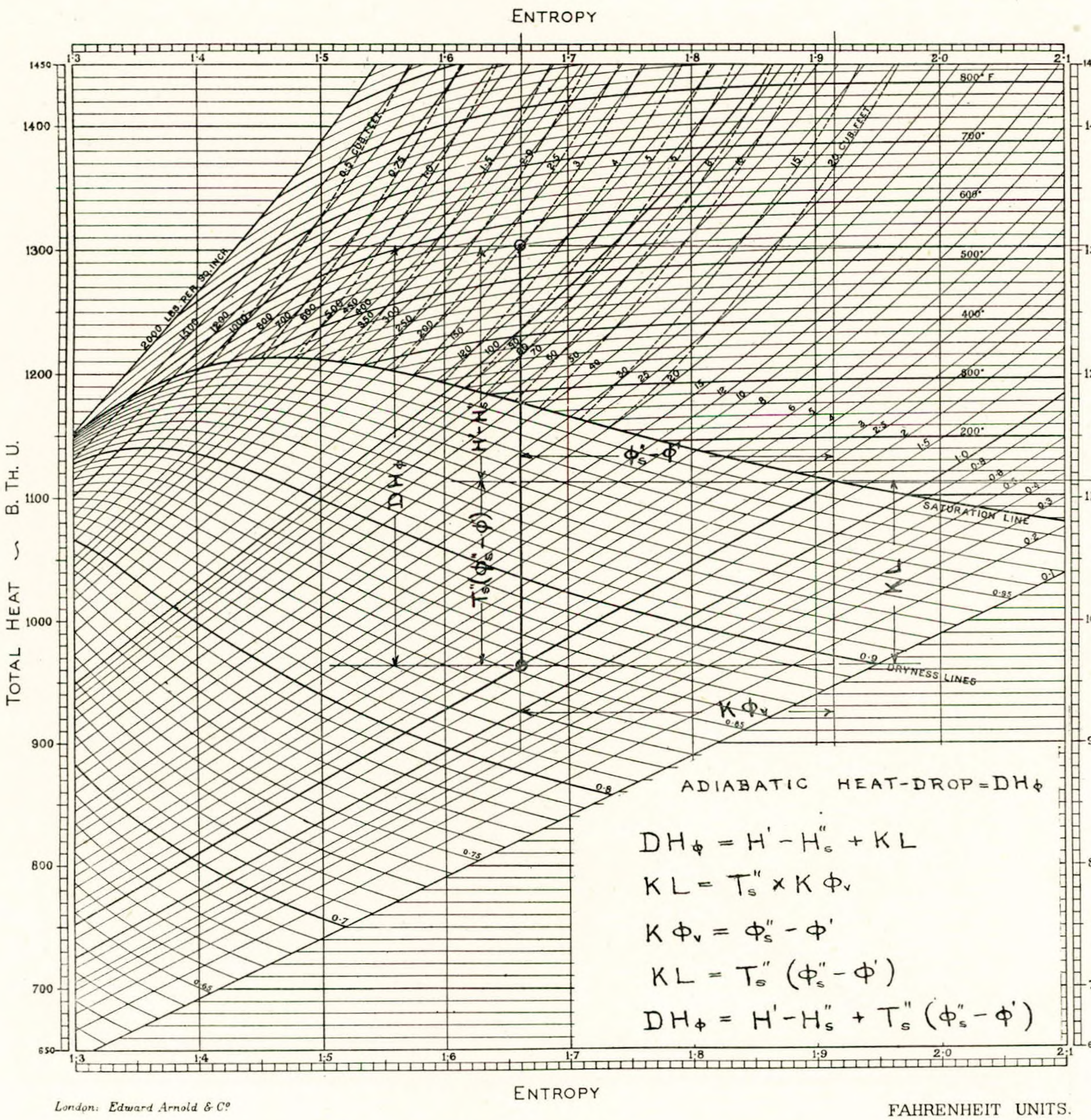


FIG. 39.—Callendar's heat-drop formula.—Diagram I.

right angles to the heat-drop line. This is, of course, a horizontal line, which divides the heat-drop line into two parts. We require to know the lengths of these two parts in terms of heat units. The upper part is easy. Both ends of it are in heat units which can be found in the steam tables. They can be found from the chart by running out lines to the scale on the border.

The lower part of the line requires a little more science. Complete the rectangle of which the 2lb. pressure line is the diagonal. Call the right hand vertical side KL, which means that it is an unknown fraction of the latent heat of steam at 2lb. pressure. The latent heat we could, of course, obtain from the steam tables, but the unknown fraction does not help a little bit; we can, however, write down that:—

$$DH_\phi = H' - H_s'' + KL$$

The single tick to the H means that it refers to the high pressure end and the double tick to the low pressure end. The suffix $_s$ means that the value has to be taken on the saturation line.

We must now examine this term KL on the extended chart (Fig. 40). The first thing to be noticed is that we have a large right-angled triangle of which the hypotenuse coincides with the 2lb. pressure line from the water line to the saturation line. The vertical side is quite obviously a measure of the latent heat and the base is a measure of the entropy of evaporation. This is something quite new to us. It is the increase in the entropy when water is evaporated to dry saturated steam, as can be seen from the construction of the chart.

The point which should be noticed is that the small right angled triangle with the sides KL and $K\Phi_v$ is similar to the large triangle with sides L and Φ_v and therefore $\frac{L}{\Phi_v}$ is equal to $\frac{KL}{K\Phi_v}$.

Now $\frac{L}{T} = \Phi_v$. This is a fundamental equation in thermo-dynamics and is the only bit of this proof which is not simple geometry. T is the absolute temperature at which evaporation takes place.

Therefore $\frac{L}{\Phi_v} = T = \frac{KL}{K\Phi_v}$ and $KL = TK\Phi_v$.

We have found a new expression for KL, but it will be noticed that it still contains the unknown K.

Referring again to our original diagram (Fig. 39) we note that $K\Phi_v = \Phi_s'' - \Phi'$ because they are the opposite sides of a parallelogram, therefore

$$KL = T_s''\ (\Phi_s'' - \Phi').$$

Inserting this value of KL in our first equation we get:—

$$DH_\phi = H' - H_s'' + T_s''\ (\Phi_s'' - \Phi').$$

which is Professor Callendar's equation for heat-drop when the final state is wet. All the terms can be found in a steam table, without reference

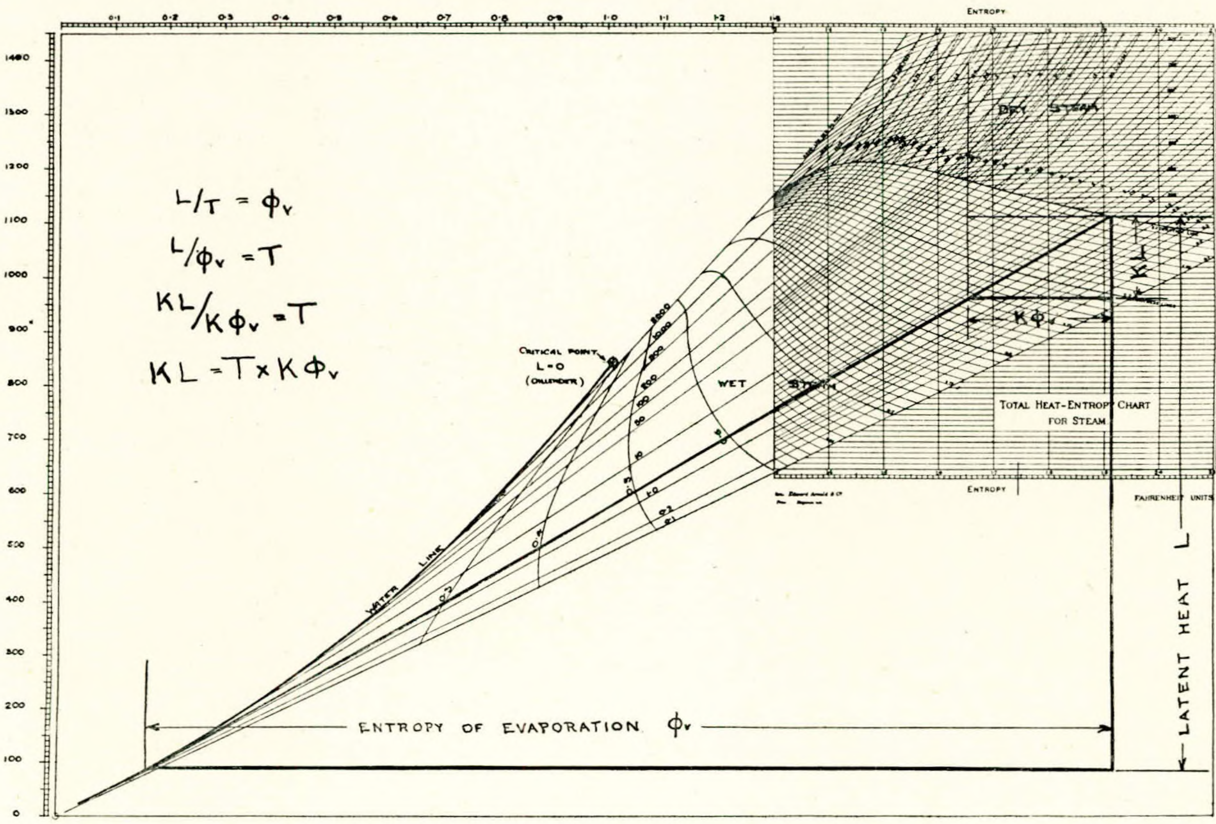


FIG. 40.—Callendar's heat-drop formula.—Diagram II.

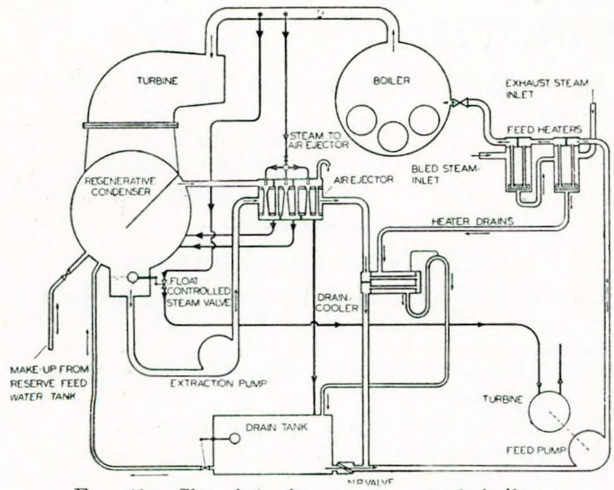


FIG. 41.—Closed feed system for tank boilers.

to a chart.

I have said very little about the important subject of feed heating, but I would like to indicate in what way feed heating is associated with exhaust steam engineering.

There are three main sources from which heat is derived for feed heating. These are firstly, waste funnel gases, which play an important part in power station feed heating, but are not used in this way in marine plants because of the weight and space occupied by the heaters; secondly, auxiliary exhaust steam, which is very largely used in marine installations, but as it plays no part in reducing the heat lost in the circulating water we will not investigate it now; and lastly, steam bled from the main engines. This last system, which is comparatively new to marine engineering and indeed to engineering generally, does definitely save heat from being lost in the circulating water, because all the heat left in the steam at the point of bleeding is returned to the feed water.

No economy is to be derived from bleeding steam which has not done some work. In other words the heating steam must be bled from the engine, and not from some point before the stop valve. It is necessary to emphasise this point because there is a fairly wide-spread belief that feed heating by steam taken from the boiler itself is economical. The Heat Engine Trials Committee have found it necessary to include in their *report a proof that there is no advantage to be gained by doing so.

I propose to show only two diagrams (Figs. 41 and 42) of feed systems which are very similar, and differ chiefly in their methods of automatic control. They are both closed feed systems. The purpose of keeping the feed system closed is to exclude air. Both systems have two heaters and a drain cooler.

In Fig. 41 it will be noticed that the condenser has a sump in it, which is really the hot well. In this sump is a float which controls the steam to the feed pump. There is also a float control on the drain tank which automatically returns to the main system the water derived from the steam bled and that used by auxiliaries. If no steam was lost from the system these controls would be sufficient to maintain a correct distribution. Any deficiency of water will show on the gauge glass of the boiler where there is no control, and must be made up by hand control from the make-up feed water tank.

* Heat Engine Trials Committee Report, 1927, p. 16.

Fig. 42 shows the same system adapted for a water-tube boiler. This type of boiler, being much more sensitive to water level than the tank boiler, has an automatic feed regulator. The float in the condenser operates a double valve which either byepasses water from the air ejector's condenser to the feed tank when there is too much water in the system, or returns water to the condenser from the feed tank when there is too little. The purpose of the pressure controlled valve on the feed pump is to reduce the load on the feed heaters when the regulator valve is partially shut.

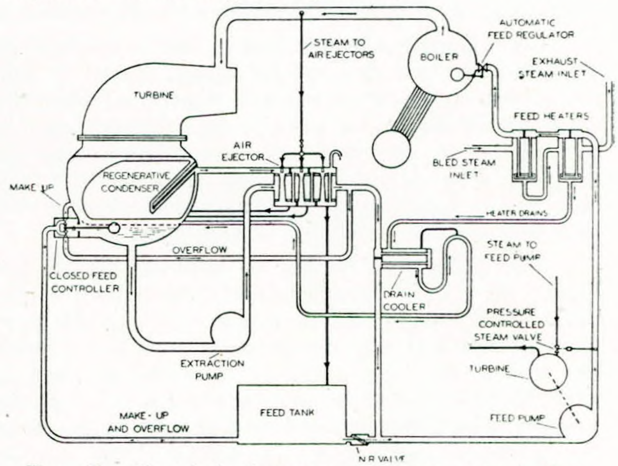


FIG. 42.—Closed feed system for water-tube boilers.

Only one of the two feed heaters in each arrangement uses bled steam. The l.p. heater in each case takes steam from the auxiliary exhaust. These are only skeleton diagrams of feed systems. The complete diagram of the feed system of a modern liner would be merely a confused mass of lines if shown on the screen. We are, however, at present only considering feed heaters which use bled steam because they reduce the flow of heat to the condenser; they do not effect the internal efficiency of the engine. In other words, the steam which does carry on in the engine until it is exhausted to the condenser will have the same dryness fraction at exhaust as it would have had if there had been no bleeding; it is the dryness fraction at exhaust which determines the internal efficiency of the engine.

If in the course of this lecture I have strayed at times from the strict interpretation of my text, I trust you have not found my wanderings unprofitable.

The Council tender acknowledgments to the several Publishers, Engineering Companies, and Kindred Institutions mentioned in Mr. Green's paper for permission to reproduce the various charts, diagrams, photographs and data.—EDITOR.

INSTITUTE NOTES.

GUILD OF BENEVOLENCE—THE COUNCIL'S CORONATION YEAR APPEAL.

The following letter has been sent to all members of The Institute in the hope that those who are not yet members of The Institute of Marine Engineers Guild of Benevolence will join in an endeavour to make this year a memorable one, and the Council look forward with pleasure to receiving from the members of The Institute the support this worthy object warrants.

7th May, 1937.

Dear Sir,

It is felt by the Council of The Institute of Marine Engineers that the Coronation Year of His Majesty King George VI (in which year by the way he has granted his patronage to The Institute) should be marked in some special manner by the members.

The Council consider that no better method of doing so, or one that would appeal more to our Royal Patron, could be adopted than to advance the work of alleviation of poverty and distress so ably carried out by the Guild of Benevolence, and the simplest way of accomplishing this very worthy object is by members of The Institute becoming also members of the Guild of Benevolence.

There are two classes of membership of the Guild, viz.: Life Members and Subscribing Members. Life Membership is attained by a single payment of not less than ten guineas, and the Council are well aware that it follows that the number of Life Members must be necessarily limited, but the support which the Council feel should be given to the Guild to mark this Coronation Year can be given as a Subscribing Member for the minimum sum of 10s. 6d. per annum.

The Council feel confident that if the work done by the Guild in helping members of our profession, their wives and dependants who have, through no fault of their own, fallen on evil days was known to the members of The Institute, there would be a ready and willing response to this Appeal made to you to join the Guild.

Coronation Year and the year of the resumption of The Institute's Royal patronage can be made a memorable one, and this will be fully realised by all the members of The Institute when the following figures are perused :—

Membership of The Institute of Marine Engineers	4,040
Membership of the Guild of Benevolence	71 Life Members. 267 Subscribing Members.

In the hope that the occasion and the object in combination will make a strong appeal to, and have a ready response from the membership of The Institute, an application form for membership of the Guild of Benevolence is enclosed with this letter.

Yours very truly,
ROBERT RAINIE,
Chairman of Council.

ELECTION OF MEMBERS.

List of those elected at Council Meeting held on Monday, June 7th, 1937.

Members.

Ernest Henry Abbott, 13, Ferndene Grove, Heaton, Newcastle.

Reginald Gordon Forsyth, Eng. Com'r., R.N., 35, Thurbern Road, Northend, Portsmouth.

Oliver Thomas Harvey, 1, Chessel Crescent, Bitterne Park, Southampton.

Harold Kidd, 26, Speedwell Road, Birkenhead.

William Robert Pollock, 21, Blanche Street, Elsternwick S4, Victoria, Australia.

Arthur Farquhar Roberts, Muritai, Little Crosby Road, Great Crosby, Liverpool, 23.

George Richard Russell, 17, Bernard Street, Swansea.

Associate Members.

William Crosby, Jnr., 38, Mainsforth Terrace, Sunderland.

Associates.

William Ernest Gransby Brigden, 131, Winchmore Hill Road, Southgate, London.

Henry Carson, Carnaughts, Shankbridge, Ballymena, Co. Antrim.

George Thomas Richardson Campbell, 31, Eastfield Avenue, Monkseaton.

Eric Christian Volke, 85, Mornington Road, Wanstead, E.11.

Transfer from Student to Associate.

James Lawrence Oliver, Elm Vicarage, Wisbech, Cambs.

ADDITIONS TO THE LIBRARY.

Purchased.

Lloyd's Register of Yachts, 1937. Lloyd's Register of Shipping, 42s. net.

"Ship Construction and Calculations", 6th edn., by G. Nicol. Brown, Son & Ferguson, Ltd., 25s. net.

Memorandum on Electric Arc Welding (3rd edn., 1937). Issued by the Factory Department, Home Office, and obtainable from H.M. Stationery Office, 3d. net.

K.R. and A.I. Amendments (K.R.3/37). H.M. Stationery Office, 3d. net.

"The Directory of Directors, 1937". Thomas Skinner & Co.

Presented by the Publishers.

"Welded Joints in Pressure Vessels", by S. F. Dorey, B.Sc. Paper read before The Institution of Civil Engineers.

"A General Review of the Development of the Diesel Engine during 1936". Diesel Engine Users Association.

"Lloyd's Register of British and Foreign Shipping, 1870".

Germanischer Lloyd—Bericht über das Jahr 1936.

Proceedings of The Institution of Mechanical Engineers, Vol. 134, containing the following papers:—

"Modern Forms of Water-Tube Boilers for Land and Marine Use", by Münzinger.

"The Air Resistance of Passenger Trains", by Johansen.

"The Smoke of Cities", by Owens.

"Recent Developments in Hydro-Electric Engineering with Special Reference to British Practice", by Seewer.

"Some Factors Affecting the Design of Heat Transfer Apparatus", by Still.

"The Measurement of Attitude and Eccentricity in Complete Clearance Bearings", by Clayton and Jakeman.

"The Correlation of Impact Tests and the Problem of Standardization", by Warlow-Davies and Southwell.

"Epicyclic Gearing", by Love.

Transactions of the American Society of Naval Architects and Marine Engineers, Vol. 44, 1936:—

"Safety at Sea", by Montgomerie.

"Fire in Passenger Spaces", by Champness.

"Safety of Life at Sea", by de Berlie and Boris.

"Some Observations on the Actual Applications of the Safety and Loadline Convention Rules", by Ikushima.

"Safety at Sea", by Tawresey.

"Some Particulars Concerning the Design of the 'Normandie' and the Elimination of Vibration", by Coqueret and Romano.

"Modern Atlantic Liners", by Rigg.

"Rolling of the s.s. 'Conte di Savoia' in Tank Experiments and at Sea", by de Santis and Russo.

"A Study of Ship Performance in Smooth and Rough Water", by Kempf.

"Power, Speed, Economy and Seaworthiness of Medium-sized Fast Liners", by Foerster.

"Some Experimental Studies of the Sailing Yacht", by Davidson.

"Features of Practice Affecting Design", by Roop.

"Modern River Towboats", by Brodie.

"Oil Tankers", by Hudson.

"Further Developments in the Stability and Rolling of Ships", by Niedermair.

"Velox Steam Generator for Merchant and Naval Vessels", by Meyer.

"Heat Insulation", by Cox.

"Arc Welded Piping in Central Station Steam Plants", by Hirshfeld and Corey.

"Propeller Vibration", by Lewis.

"Neoprene". (Lectures delivered at the Neoprene Exhibition, London). Imperial Chemical Industries.

"Recent Practice in Welding Large Oil Tankers", by Hudson and Jackson, and "Some Effects of Welding on Ship Construction", by Hunter. Advance proof papers issued by the American Society of Naval Architects and Marine Engineers.

"Lessons and Problems in Electricity", by Newell C. Page. The Macmillan Company, 356pp., 161 illus., 12s. net.

This book is the outgrowth of a set of notes used by the author for giving instruction in the fundamentals of electricity to students who had had a year's training in mechanics and the calculus, doubtless in addition to many years' previous tuition in what may be compared to our secondary schools.

The author states that "this book has been developed to meet the needs of students planning to enter the engineering sciences and so may be expected to have a distinct engineering flavour. At the same time the treatment is such that it is believed that a student entering pure science will find it helpful in giving an insight into fundamentals. Principles rather than applications are emphasized . . . . ". Undoubtedly the pure science rather than the engineering features has been stressed throughout the book, and principles rather than applications are more stressed throughout. The treatment of the sections dealt with is very good, and at the end of each chapter are given many problems—mostly with answers—excellent in choice and of great use.

Little previous knowledge of the subject treated is necessary to understand the text, assuming a knowledge of the calculus on the part of the reader, but this fact probably limits the field in which it will be adopted as a textbook. Although a knowledge of the fundamentals treated in the book would undoubtedly be of use to all students, owing to the enormous ground which must now of necessity be covered by a student of electrical engineering in those sections of the subject directly applicable to his work the reviewer considers this work to be far more suitable to the student of pure science. The application to practical work of the fundamentals dealt with in the book gives it, as the author suggests, merely an "engineering flavour" and does not strike the reviewer as a very definite aim for a book. A brief summary of the contents will give a good idea of the parts of the subject dealt with and will serve as a guide to the usefulness of the book to engineering students: Electrostatics, Coulomb's and Gauss' laws; current and electromotive force; Ohm's law; Kirchhoff's rules; resistivity and temperature coefficients; Joule's law; thermo-electricity; magnetism and magnetic fields; instruments; condensers; gaseous conduction and the electron; conduction in high vacua; inductance; a.c. circuits; power and power factor; transformers; oscillating circuits; and units.

"The Most Modern Process of Gear Tooth Production Stage by Stage from Initial Cutting to Final Checking". A Treatise by Charles Churchill & Co., Ltd. Published by Geoffrey Dadd, Ltd., 53pp., illus., 10s. 6d. net.

This treatise presents an impressive advocacy of the merits of hobbing in the initial stage of gear production, followed by machine checking, chamfering, lapping and testing.

To those whose business is solely that of gear production to extremely fine limits, the publication will without doubt be of interest, but it must be emphasized that it is not a text book on the general processes of gear production; for instance, machining problems arising from the use of alloy steels, hard cast-irons and other special metals are not referred to, nor is any helpful mention made of heat treatment. The value of this book would be greatly increased by the inclusion of one or two chapters of more general interest.

"Bearing Metals and Alloys", by H. N. Bassett. Edward Arnold & Co., 428pp., illus., 25s. net.

This work collects and correlates the published information on the subject of alloys for bearings, a field which was badly in need of such a stocktaking. The only comparable book was published seven years ago in the United States; it contained a list of some 1,200 references and its compiler was apparently so impressed by the size of his task that he made little attempt to correlate the information, but presented it in a series of abstracts of important papers. Mr. Bassett, however, in addition to making full use of what obviously must be an excellent indexing system, fills in the gaps from his own experience. For this reason, while the book provides those in industry with a comprehensive work of reference, enabling them to turn readily to the published work of authorities on any particular detail, it can yet be recommended to the student as a readable review.

The properties and applications of all the common alloys are fully treated, due importance being given to the points of view of both the engineer and the metallurgist. There are also interesting chapters on lubrication and on "freak" bearing materials like rubber and stone; unfortunately marine bearings are scarcely mentioned.

In dealing with a mass of material, and no important work appears to have been missed, the author has presented the views of the various writers without expressing critical opinions of his own. Sometimes, general statements are made which, since it is very difficult to generalise about bearing metals, require modification. However, the book fills a vacancy on the engineer's bookshelf as, by collecting and arranging the available information, it helps to clarify the present somewhat involved position of this subject.

"Alternating Current Measurements at Audio and Radio Frequencies", by D. Owen, B.A., D.Sc. Methuen & Co., Ltd., 120pp., 80 illus., 3s. 6d. net.

The aim of this little book is, within a brief compass, to present an account of the methods and procedure used in the measurement, with the high accuracy attainable by the employment of alternating current, of the usual constants of electric circuits, namely, resistance, inductance and capacitance, as well as of frequency. The descriptions are not confined to measurements at low or audio-frequencies but also at radio-frequencies. Particular attention is directed to the use of the alternating-current potentiometer to a wide variety of measurements.

Numerical examples of actual measurements are given in connection with a number of the methods employed, which give useful guidance on such points as the proper choice of the magnitude of standards and the degree of accuracy to be expected. For the benefit of readers who wish to pursue their studies in special directions, useful references to various treatises are given.

The book contains chapters on alternating current theory, and the measurement of self and mutual inductance and capacitance at low frequencies. Bridge measurements at low frequencies are also dealt with. Particular attention is directed to the mode of use and application of the alternating-current potentiometer to a wide variety of measurements. Various types of applications of this instrument are given and described in detail.

A final chapter deals with measurements at radio-frequencies and involves the consideration of capacitance, self and mutual inductance, resistance and frequency.

The book is one of a series written by authors who are actively engaged in research on the subject on which they write, and is intended to supply a compact statement of the modern position of the subject. It may be confidently recommended to the notice of the honours student in practical physics and electrical engineering. The expositions are plain, the figures are well drawn and the general publication is excellent.

EDUCATION GROUP.

Discussion on "A Draft Syllabus for 1st and 2nd Class Certificate Examinations".

A meeting was held at The Institute under the auspices of the Education Group on Friday, May 28th, 1937, at 6 p.m., preceding the Annual Meeting of the Group. Mr. R. F. Thompson, B.Sc., Chairman of the Group, presided.

The agenda on this occasion was confined to a general discussion on a syllabus, drafted and submitted by Mr. T. A. Bennett, B.Sc., as a basis for this discussion, for 1st and 2nd Class Certificate Examinations. Mr. Bennett opened the proceedings by explaining various modifications of the present Board of Trade practice which he had embodied in his draft, copies of which were in the hands of members present, and the Chairman then invited discussion of the draft, taking each subject and section thereof in their appropriate order.

At the close of the meeting agreement was reached upon various amendments to Mr. Bennett's draft, and it was decided to forward the syllabus to the Council with the recommendation that it be sent to the Board of Trade Departmental Committee now in session for consideration by the Syllabus Sub-Committee.

On the proposal of the Chairman, a hearty vote of thanks was accorded to Mr. Bennett for the valuable assistance he had rendered by preparing the draft and opening the discussion.

The recommendation was adopted by the Council at their meeting on June 7th and the syllabus has been forwarded to the Committee above mentioned.

The Coronation. Loyal Address to His Majesty the King.

In the May TRANSACTIONS was published a photograph of the text of the Loyal Address presented to the King by the combined Engineering Institutions of Great Britain. The following is a copy of a letter which has been received from the Home Secretary in acknowledgment of The Institute's participation in the Address:—

"The Home Secretary,
27th May, 1937.

Sir,

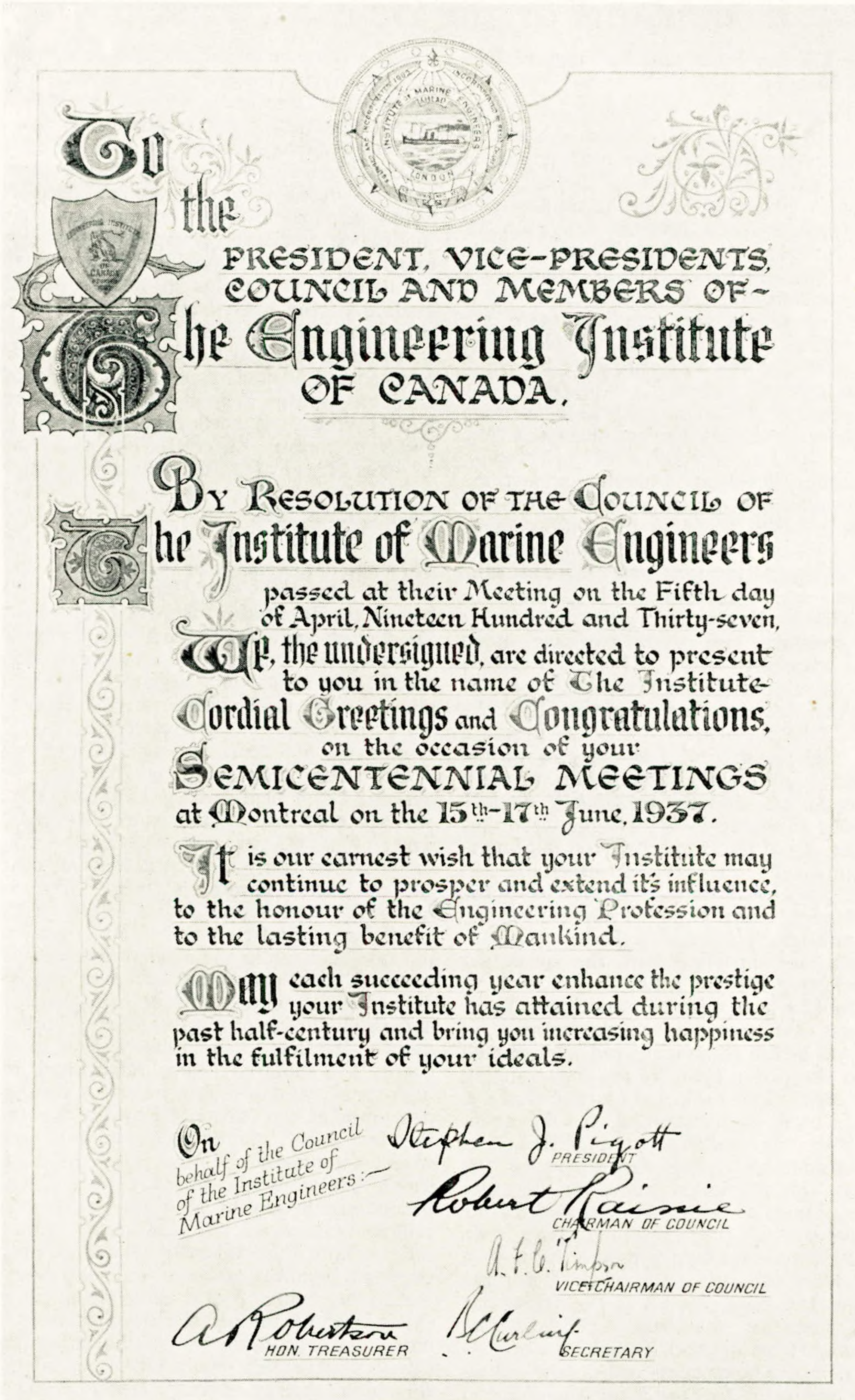
I have had the honour to lay before The King the loyal and dutiful Address from the principal Engineering Institutions and Societies of the United Kingdom of Great Britain and Northern Ireland on the occasion of Their Majesties' Coronation. I have been commanded by The King to convey to you Their Majesties' warm thanks for the assurances of loyalty and devotion to which the Address gives expression on behalf of the Institute of Marine Engineers.

I am,
Sir,
Your obedient Servant,
(Sgd.) JOHN SIMON".

The Engineering Institute of Canada—Illuminated Address.

We reproduce opposite a photograph of the Illuminated Address which was presented by The Institute's representative, Mr. D. B. Carswell, to the Engineering Institute of Canada at the opening meeting of their Semicentennial Convention at Montreal on June 15th.

The Engineering Institute of Canada—Illuminated Address.



To the
PRESIDENT, VICE-PRESIDENTS, COUNCIL AND MEMBERS OF
The Engineering Institute
OF CANADA.

By Resolution of the Council of
The Institute of Marine Engineers
passed at their Meeting on the Fifth day of April, Nineteen Hundred and Thirty-seven,
We, the undersigned, are directed to present to you in the name of The Institute
Cordial Greetings and Congratulations,
on the occasion of your
SEMICENTENNIAL MEETINGS
at Montreal on the 15th–17th June, 1937.

It is our earnest wish that your Institute may continue to prosper and extend it's influence, to the honour of the Engineering Profession and to the lasting benefit of Mankind.

May each succeeding year enhance the prestige your Institute has attained during the past half-century and bring you increasing happiness in the fulfilment of your ideals.

On behalf of the Council of the Institute of Marine Engineers:—

Stephen J. Pigott
PRESIDENT

Robert Raisne
CHAIRMAN OF COUNCIL

A. F. C. Timpson
VICE-CHAIRMAN OF COUNCIL

A Robertson
HON. TREASURER

B C Curling
SECRETARY

ABSTRACTS OF THE TECHNICAL PRESS.

Loads on the Moving Parts and Bearings of Fast Running Motors.

The author presents an investigation of the loads on the moving parts, viz. piston, connecting rod, and crankshaft, and the main bearings of oil engines in which account is taken of the mass and elasticity effects which intervene in the transmission to the main bearings of the pressure acting on the piston. He considers that these factors, while they may be neglected in slow running engines, are of material importance at high speed, especially if this is associated with a very rapid pressure rise during the process of combustion. Under their action the piston top, the gudgeon pin, the crank and the main bearing covers will bend, the cylindrical portion of the piston and the connecting rod will be compressed, and displacements will take place in the oil-filled clearances of the intervening pins. The author develops mathematical expressions for the amplitude of oscillation set up in the connecting rod of such a system for four types of gas pressure curves, namely (1) a cosine curve joined on to an initial straight portion, (2) a sine curve, (3) linear variation from zero to a maximum followed by a linear drop to zero (triangular line); (4) instant pressure rise to a constant value (pressure line parallel to base). These expressions which give the amplitude in terms of the gas pressure forces, the spring constant of the connecting rod, the frequency of the gas pressure fluctuation ω, and the natural frequency ω_e of the rod, are formulated on the assumption of zero damping, the author having previously demonstrated that this influence is of a negligible order. As the force set up in the connecting rod is proportional to the amplitude, the curves of amplitude also represent the force variation. On this basis the author reproduces a diagram of curves showing the variation of the ignition pressure force to a base of crank angle in comparison with the forces set up in the connecting rod for ratios $f=\omega/\omega_e=\cdot4$, $\cdot6$, and $\cdot8$ and sinoidal gas pressure variation. These curves indicate that according to the ratio ω/ω_e the force set up in the connecting rod may exceed that acting on the piston head by amounts ranging up to 70 per cent. The usual assumption that the gas pressure force is transmitted to the main bearings without any modification may thus lead to a very considerable under assessment of the bearing loads. This maximum, moreover, does not occur in the condition of resonance, viz. $f=1$, as one might suppose, for a further plot of the ratio connecting rod force ÷ ignition pressure force to a base of "f" places this maximum at $f=\cdot6$ for the sinoidal and at less than $f=\cdot5$ for the triangular type of gas pressure variation. The diagram further shows that an appreciable excess force is already set up in the connecting rod at low values of f, e.g., 33 per cent. for $f=\cdot2$ for sinoidal and 16½ per cent. for cosinoidal variation. Calculated natural frequencies of the elastic system lying at revolutions far exceeding the service rate are not therefore a guarantee against appreciable excess forces. According to the rapidity of the combustion process, the excess forces set up in the moving parts will rise more or less abruptly to a maximum and fade out in the form of a damped oscillation. The author explains that this characteristic of the force fluctuation explains various forms of trouble experienced in the moving parts and that, more particularly, it results in adverse conditions of loading of the main bearings. He therefore considers that abrupt rises of the ignition and combustion pressures ought to be avoided, that the moving parts should be made as light and as stiff as possible, and that in this respect short stroke engines are more favourably placed than long stroke engines.—*Dr. Ing. Geiger, "Werft, Reederei, Hafen", Vol. 18, No. 11; pp. 163-167. June 1st, 1937.*

Cast Crankshafts.

Referring to the paper entitled "Properties of some Materials for Cast Crankshafts, with Special Reference to Combined Stresses", read before the Institution of Automobile Engineers by Dr. Gough and Mr. Pollard, the writer quotes the following particulars from an investigation made to secure accurate data relating to typical cast materials for crankshafts. Five commercial alloys were selected:—(1) a heat-treated cast steel containing ·32 per cent. carbon, 2·42 per cent. nickel, ·49 per cent. chromium and ·38 per cent. molybdenum; (2) a heat treated 1·75 per cent. copper—·46 per cent. chromium iron; (3) an inoculated iron; (4) a ·42 per cent. chromium—·95 per cent. molybdenum iron; and (5) a 1·87 per cent nickel—·47 per cent. chromium iron. The last three materials were supplied and tested in the as-cast condition. Static tensile and torsional tests indicate that while the steel is essentially a ductile material, each of the four irons has no clearly defined yield points, little deformation at fracture, and negligible absorption of energy under the Izod notched-bar test. The relative fatigue resistances of the five materials were determined in the new high-speed combined fatigue stress testing machine developed at the National Physical Laboratory for the two cases of reversed bending stresses and reversed torsional stresses. The results of the tests show that while the ultimate tensile strengths of the materials vary by as much as 64 per cent., the relative values of the endurance ratio (i.e. the fatigue limit ÷ the ultimate tensile strength) under reversed bending stresses show no very marked variations, being ·5 for alloy steel, ·56 in copper chromium and the nickel chromium iron, ·55 in the inoculated iron, and ·51 for the chromium molybdenum iron, which would all be considered normal for high quality ductile steels. The torsional endurance ratios of the four cast irons which range from ·44 to ·52 are distinctly higher than that of the alloy steel, viz. ·34. Under combined stresses the general behaviour of the alloy steel is thus much akin to that of wrought steel, while the irons develop characteristics more definitely associated with the theoretical behaviour of a class of materials of a more brittle nature. The general conclusion arrived at as the result of the investigation is that the fatigue resistance of each of the five materials examined is high in relation to its tensile strength, a factor of considerable importance in alloys for cast crankshafts.—*"Engineering", 21st May, 1937; p. 584.*

Foundry Progress.

In drawing the attention of readers of "The Engineer" to the supplement dealing with "Modern Foundry Equipment" which accompanies the week's issue, the writer offers some general observations on foundry progress. He states that while for a time the foundry may have lagged behind the machine shop, it is now on an equality with the other departments in a well-organised works, particularly in America, where the scale of foundry operations is much greater than in Europe. This also applies to foundry lay-out, which is not dealt with in the supplement, as well as to foundry equipment. The writer further comments on the excellent work of the British Cast Iron Research Association, under whose aegis cast iron has ceased to be a mixture blended to a "secret" formula and has become a definite metallurgical product, as carefully prepared in the cupola as it would be in the chemical laboratory. With scientific research have come unforeseen improvements in cast iron which has revealed unbelievable qualities, and the old uncertain "rule of thumb" methods are being rapidly replaced by definite scientific processes controlled at every stage. For the operation of foundries by such methods—the writer considers—it is necessary to have men specially trained for the work; and in this connection he particularly commends the degree course in metallurgy of the University of Sheffield together with the practical training given by the British Foundry School.—*"The Engineer", Editorial, May 28th, 1937, p. 625.*

Methods of Shaping Materials Employed in Shipyard Practice.

The author presents a systematic classification of the methods of shaping materials employed in shipyard practice. This survey includes the whole range of operations carried out in the cold state such as pressing, shearing, punching, bevelling, joggling and rolling, shaping in the hot state (i.e. forging operations), the burning out of parts from plates and ingots, building up of parts by electric or gas welding, and the dishing of plates and sections by the application of a burner. The characteristic features of the deformation of the material which takes place in each of these operations is indicated and their respective suitability for the production of parts of the hull structure and ship fittings is discussed. The author draws particular attention to the type of plate bending press used in the sheet metal industry which permits the manufacture of box and tube shaped parts with a reduced number of welded or riveted seams, and to the possibility of following the motor car industry in the use of drop forging methods for a wide range of ship fittings. He recommends the more extended adoption of the electric operation of presses in preference to hydraulic operation on account of the greatly increased cost charges and risk of freezing in cold weather associated with the latter. — *Dipl. Ing. Freudenthal, Werft, Reederei, Hafen, 15th March, 1937; Vol. 18, No. 6; pp. 75-79.*

The Effect of Roughness of the Hull on Ship Resistance.

The author deduces from the trial trip results of a ship, in conjunction with the results of model tests on the hull with and without propellers, and on the propellers in the open, the increase of skin friction resistance due to fouling. Two trials were carried out, one with smoothly-painted bottom, and one with the bottom covered with barnacles averaging 0·2in. diameter and 0·15in. height. He assumes the values of propeller efficiency to be as for the model screw in open water, and the thrust deduction also to be the same as measured for the model, viz. 10 per cent. The wake he deduces from the torque coefficient of the full-size ship, and finds that it rises from 3·5 per cent. for the smooth hull to 9 per cent. for the foul. The Reynolds number of the ship ranges from 0·65 to $1{\cdot}4 \times 10^9$, but in view of the scale effect on appendage resistance, which is a larger percentage of the total at the lower speeds, he regards the figures for the highest speed as the most reliable. These indicate that the skin friction coefficient (expressed in non-dimensional form) has a value 0·0038 for the barnacle-coated surface against 0·0021 for the smooth paint, and 0·00155, the assumed ideal value for a perfectly smooth surface. The excess of the value for the painted surface over that for the ideal case is somewhat less than that obtained by Kempf for the "Hamburg" and less than the other full-scale values available. This is probably due to the fact that the shell plating was welded, and not riveted. The excess in the case of the roughened hull is stated to be in accordance with deductions from experiments on sanded surfaces of corresponding roughness, making allowance for the size of the barnacles as observed. —*H. Amtsberg, Schiffbau, Vol. 38, pp. 135-8.*

The Salary and Status of the Engineer.

The writer, a university professor of engineering who for thirty years has carefully followed the subsequent careers of a large number of engineering students, compares the status and average salary or professional income of the engineering graduate with that acquired by a contemporary who studied medicine, law or commerce. Referring first to the Army, he points out that before the war the pay of the "Sappers" relative to the other branches of the Army was better than it is to-day, and he considers that apart from this consideration, the military engineers have not received the improved conditions that ought to be theirs after an "engineer's war" has been won. In this connection, he suggests that it would be interesting to know how many of the senior military men who would be in the most responsible positions in the British Army if war commenced tomorrow, had received a training in applied science; and he expresses the hope that the appointment of a naval engineer, namely Sir Harold Brown, as Director General of Munitions Production and therefore as a member of the Army Council may mark the beginning of a much needed reform in Government service. Turning to the Royal Navy, the writer finds that since the War the engineering branch has lost nearly everything that it had gained under the "Fisher Scheme". Once more engineers are subordinate to deck officers and cannot hope to reach high rank. Thus in the 1936 Navy List there is only one engineer officer who ranks as a Vice-Admiral on the active list and the Board of Admiralty does not include a naval engineer among its ten members. The writer considers that a blunt description of the situation would come to this: The naval engineer has, during his career, on an average a more strenuous life than the deck officer with far less prospect of reaching the highest ranks in the Royal Navy and with comparatively little real authority. A review of the pay and prospects of engineers in the Public Works Department of a typical Eastern Crown Colony again reveals that these are less favourably placed than the graduates in literature or law who as "cadets" form the staffs of the administrative departments. In addition to a university degree or its equivalent, the A.M.I.C.E. certificate, the P.W.D. engineer must have had practical experience, and the writer states that there are very few candidates who do not require considerable financial help from relatives for at least two years after graduation. In the case of cadet officers, on the other hand, the expense of three years at a university prior to selection is the only financial burden on the parent, selected candidates being sent to either Oxford or Cambridge for a year of post-graduate study, at the expense of the Crown Colony where they will work. The "cadet" can thus be self-supporting three years after he has matriculated, i.e. at 21 years of age. In the Service the salary of the cadet officer will in due course rise to £1,400, at an age of about 45, and in practice it is almost a certainty that he will obtain a class I appointment carrying a salary of £1,500 to £1,800 before he is 50 years of age. The highest grade for an engineer is that of Director of Public Works at £1,800 and there are two or three Assistant D.P.W.'s at £1,400, which is usually the highest pay an engineer can expect. Regarding engineering commercial work, the writer was informed by the director of a large firm that salesmanship was a profitable line, but that it required little technical knowledge. He also mentions the case of a company which paid the chief designer of large steam turbines £750 per annum, while the secretary who was without professional qualification was paid £1,000. He further states that in common experience those who handle money receive higher emoluments than many professional engineers, and considers, finally, that the most practical method of raising the status of the engineer to that of the doctor and the lawyer is to induce statesmen to improve the pay and prospects of engineers in Government service.—*"The Engineer", 28th May, 1937, p. 616.*

The Rhine Passenger Motorship "Albert Leo Schlageter".

The principal particulars of this vessel are:—

Length over all	157ft.
Breadth moulded	22·3ft.
Depth	7·2ft.
Load draft...	3·8ft.
Total maximum b.h.p.	550
Speed upstream	10 m.p.h.
„ downstream	18 m.p.h.
Number of passengers	825

Seating accommodation for about half the total number of passengers is provided in a dining saloon on the fore part of the main deck, and on the superstructure deck, the forward part of which is enclosed. The machinery comprises two 6-cylinder four-cycle Diesel engines, 7·9in. dia. and 9·4in. stroke, running at 725 r.p.m., placed towards the stern. They have exhaust-driven turbo-superchargers running at 22,000 to 23,000 r.p.m., which serve to increase the power some 40 per cent. and decrease the weight per horse-power from 46 to 35lb. per h.p. The engines drive, through 1 : 4·6 reduction gearing, twin six-bladed Voith propellers, having a diameter of 4·6ft., and projecting 3ft. below the flat stern. The propellers run at practically constant speed, and are controllable to give astern or sideways thrust, as required for manœuvring, without reversing or the use of a rudder, no rudder being fitted. Control is from the wheelhouse, by cable and servomotor. On trial a speed of about 14 m.p.h. was attained, relative to the water, in a depth of 16ft., at 490 b.h.p. The corresponding e.h.p. from model tests was 255, a figure which is critically influenced by depth of water and is nearly double the value for deep water. The engines are on spring mountings, isolated as regards vibration from the hull, the necessary elasticity being provided in pipe connections and shafting. The propellers, too, have a rubber mounting, and special silencing and sound-insulation is employed.—*R. Schröter, "Werft, Reederei, Hafen", Vol. 18, p. 154-7.*

Oil Production in Scotland.

Commenting editorially on the Second Report of the Scottish "Oil from Coal" Committee recently issued, the writer draws attention to the passage therein contained in which the authors of the report state the opinion that the tendency of the Scottish shale industry to make the production of solid smokeless fuel its prime object is to be deplored and urge that the use of cannel for the production of oil should be pursued actively. In this connection the writer quotes the paper entitled "Products obtained by the Carbonisation of Scottish Cannel in Continuous Vertical Retorts" read before the Institution of Gas Engineers last year by Mr. James Jamieson and Dr. J. G. King, which describes the use of quite considerable quantities of cannel at Edinburgh gas works. From this it appears that cannels could be used in gas works to-day if they contained but a small percentage of ash and that some gas engineers believe that on a balance the properties of cannel are such that gas works that can obtain supplies satisfactory in ash content would do well to consider its use. From the point of view of oil production the important factor is that the cannel used at Edinburgh has yielded some 40-55 gallons of tar per ton and that this tar was found to be very satisfactory as a raw material for hydrogenation. As regards the quantity of cannel available in Scotland it is believed that there is a total of 25 million tons containing less than 8 per cent. of ash and yielding over 40 gallons of crude oil per ton. Some 80 per cent. of this is in the Lothians area. The amount mined per day could be brought up to 1,438 tons, at which rate, allowing 250 working days per year, the supplies would last for seventy years. The writer considers that the carbonisation of cannel coal in vertical retorts in gas works would assist the project of a Scottish hydrogenation plant, as the more difficult and costly method of hydrogenating raw coal would thereby be avoided. As, however, it is evident that quantitatively the production of oil from Scottish cannel by distillation cannot have any bearing on the national oil problem, the writer endorses the recommendation of the Committee that investigations in this direction should apply to coal as well as to cannel.—*"Engineering", Editorial, 14th May, 1937, p. 555.*

Applications of Monel Metal in Shipbuilding.

The author gives particulars of the composition and the properties of Monel metal in its various forms, viz. hot and cold treated rods, plates, strips, and castings, and presents an exhaustive enumeration of the applications of this metal in marine engines, boilers, their auxiliaries and accessories and in ship fittings of all kinds. He stresses the corrosion resisting qualities of the alloy and claims (1) that while other alloys may be superior to it as regards their tensile strength and yield point, Monel metal exhibits a particularly favourable combination of the properties of yield limit, elongation, impact strength, and strength at high temperatures, and (2) that these are not due to any treatment during its manufacture but to the nature of its constituents and that they are therefore constant. In addition this copper nickel alloy is neither subject to sudden brittleness nor, like the aluminium alloys, to ageing. The applications discussed include internal fastenings of boilers and superheaters, the linings of expansion joints of superheated steam piping, air pre-heater tubes, packing for the joints of steam and sea-water pipes, valve spindles, and the floats, strainers, seats, etc., of steam purifiers. In main and auxiliary engines the blading of turbines is one of the principal applications. Here the author states that 140 vessels including the largest and fastest types, aggregating 2,500,000 tons are so fitted. Monel metal is also widely employed in the manœuvring gear and the governors of the turbines, for cooling pipes of forced lubrication systems, sea-water cooling pipes of Diesel engines and for the intermediate cooling pipes in the air compression and fuel injection systems of such installations, for sea and fresh water pump spindles, and for the valve cones and seats of soot blowers. Other engineering applications are the feed water and de-aëration tanks of closed feed systems, impeller and guide blades of sea-water circulating pumps of the centrifugal type, propeller shafts of motor boats, parts of echo-sounding apparatus, and other parts in contact with sea-water. In ship fittings, Monel metal has been adopted for decorative purposes, e.g. rail stanchions, and for kitchen and cabin, etc. equipment. He states that about 6 tons were so used in the "Monarch of Bermuda", and that it also figured largely in the electric kitchen equipment of the "Queen of Bermuda", "Manhattan", "Bremen", "Europa" and "General Steuben". As regards cost, the author states that Monel metal is somewhat cheaper to us than good corrosion-resisting steel, as it is worked more easily, and the value of the scrap is very high owing to the character of its constituents.—*Dr. Ing. R. Muller, "Werft, Reederei, Hafen", June 1st, 1937; Vol. 18, No. 11, pp. 167-170.*

The Naval Review at Spithead.

The writer mentions the most modern ships of the various types present at Spithead, and compares them with ships now on order. The only post-war capital ships, "Nelson" and "Rodney", were laid down fourteen years ago, but in a few years' time there will be five modern ships of the "King George V" class in commission, and further units under construction. There were four aircraft carriers, but only one, "Hermes", specially designed for the purpose. Five ships of 22,000-23,000 tons are on order, which will have the effect of increasing the Fleet Air Arm from 217 to 400 aircraft. Modern cruisers were represented by "Southampton" and "Newcastle", of 9,000 tons and 32 knots, and eight more are to be built, some of slightly greater size. The writer considers the trial speed given to be conservative as a basis of comparison with foreign warships. These ships have twelve 6-in. guns. There are also to be five 8,000-tonners built, and some of the older cruisers will be modernised. The seven "Dido" class cruisers have been criticised as inferior to foreign ships now building, and for this reason the writer considers them unsuitable for detached duty. The latest destroyer at the Review, "Icarus", of 1,350 tons, belongs to a class now superseded, as regards new construction, by larger types, the "Tribals" of 1,850, and J, K, L class of 1,650 tons. The "Tribals" have a contract speed of 36 knots, and eight 4·7-in. guns, but only four torpedo tubes. The "Thames" class of cruising submarine, present at the Review, is not represented in new construction, but there are 1,520-ton minelayers, 1,100-ton patrol type, and 670-ton coastal type vessels among the submarines in the current programme. Altogether there are five capital ships, five carriers, twenty-one cruisers, forty-nine destroyers, and nineteen submarines, aggregating 545,000 tons, and forty-nine miscellaneous units to be built.—*"The Engineer", 21/5/37, p. 597.*

The Future of the Airship.

The author compares the future prospects of airship and aeroplane for commercial long-distance flights. He states that had the "Hindenburg" used helium instead of hydrogen, disaster would in all probability have been avoided, but the payload would have been reduced by 14 tons, or about two-thirds, thus reducing the commercial value of the airship to a very serious extent. He states that the R101, even with hydrogen, was short of buoyancy, and could not have flown at all on her last trip if helium had been used. No advantage is to be expected by reducing the size of airships, but rather the reverse. The general conclusion reached is that even at its best the airship compares unfavourably with the aeroplane. An aeroplane can be designed to cross the Atlantic at more than 200 m.p.h., with an adequate margin of range, i.e. at two or three times the speed of an airship, which by reason of its slower speed is more at the mercy of adverse winds. While the airship has a greater ultimate range, the ability to make a non-stop journey say to Australia is not of any commercial value, the range of the aeroplane being adequate for the stages likely to be flown on such a route.—*F. W. Lanchester, "Engineering", 28/5/37, p. 613-4.*

Force and Shrink Fits.

The author presents results of investigations carried out on elastic grip assemblies to determine (1) whether the contact film resistance could be regarded as an important factor on the axial tonnage in an assembly of large scale force fit assemblies, and (2) the effect of time on the grip of large scale and small scale elements assembled under different conditions. In the force fit tests, the elements selected for assembly consisted of the 27in. diameter disc tyred wheel and axle as fitted to Glasgow Corporation tramcars, having a wheel centre base 4½in. in length and inside and outside diameter of 4¼in. and 8in. respectively. The wheel centre was forced on to its seat by hydraulic power and the press was regulated to operate at the same rate during each assembly. The mating surfaces were carefully cleaned with solvents before each assembly; after drying they were immediately coated with a film of oil which was allowed to drain off partially before the elements were assembled excepting those cases in which the viscous nature of the lubricant rendered this precaution unnecessary. The pressing-on and back pressure loads were read off a pressure gauge at wheel displacements of ·5in. The tests were carried out with the same wheel and axle having a difference in the free diameters of the mating surfaces of $^{11}/_{10000}$in. and using the following lubricants: (1) neatsfoot oil; (2) Bayonne oil; (3) pure tallow; (4) tallow and 10 per cent. white lead; (5) rape oil; (6) rape oil and 2 per cent. of oil dag; (7) graphited spindle oil with 1 per cent. of oil dag; (8) petroleum jelly with 1 per cent. of oil dag; (9) mercurial ointment. The dimensions of the mating surfaces, measured before and after each assembly at right angles on five different planes showed no signs of change and the surface condition gave no evidence of the slightest hair line marking, thus indicating constant mating conditions. Diagrams of pressing-on loads and back pressure to a base of length of bore clearly show the remarkable influence of the surface contact film on the tonnage necessary to make the fit. The values range from 6 tons for tallow and white lead as a lubricant to 25 tons with Bayonne engine oil, i.e. an increase of about 300 per cent., and this influence is equally operative throughout the dismantling process. Periodic slipping accompanied by a bumping noise was observed when a lubricant without the property of oiliness was used. When mating with a lubricant of pronounced oiliness a gradual slipping of the elements without noise marked the assembly. The author further advances reasons why for a given axial tonnage pronounced oiliness of the contact film will involve greater radial pressure intensity in the mating of the elements than when the assembly is made with a non-oily lubricant and suggests that in the first case some qualities of steel may become overstrained. A plastic or rather semi-plastic range may penetrate well into the wall thickness of the hollow element and thus impair the quality of the grip. A discussion of the hoop stresses induced by different fit allowances leads the author to the conclusion that the current railway practice which fixes the difference in the free diameters of the mating elements of a force fit so that a load of 10 tons per inch diameter is required to press the elements in position without allowing for the effects of the lubricant is open to criticism. To test the effect of time on the grip of large scale assemblies four tramcar wheels were assembled using neatsfoot oil, mercurial ointment, Bayonne oil, and tallow with 10 per cent. white lead respectively, and the pressing-on and the back pressure tonnages were taken at intervals during a period of 31 months. This comparison showed that Bayonne oil permits of a back pressure slip of the same order as the final pressing-on tonnage. Mercurial ointment and tallow with 10 per cent. white lead, which are favoured in some turbine assemblies, increase the axial resistance to slip by as much as 100 per cent. after eight months, and this resistance is still much in excess of the final pressing-on value after 31 months. Within the range of lubricants examined, tallow with 10 per cent. white lead was shown to have offered the least resistance during assembly and separation. The effect of time on grip in expansion and shrinkage fits with different conditions of the mating surfaces was also studied in a number of assemblies, extending over 32 months, of small steel pins and collars, the latter being 1¼in. in depth with 3in. outside and 1½in. inside diameter, while the pins were 2in. in length including a slight taper at each end. The results showed that the axial resistance to slip in different surface conditions—dry and with various lubricants in the expansion fits, with various acid films in the shrink tests—was not materially affected by the lapse of time. Similar small scale tests made to determine the influence of the method of cooling showed that with water cooling a thin film is formed which acts as lubricant, while with air cooling dry mating surface conditions tend to be established.—*R. Russell, B.Sc., Ph.D., "The Engineer", May 28th, 1937; pp. 632-634.*

The Isle of Man Turbine Steamers "Tynwald" and "Fenella".

The Isle of Man turbine steamers "Tynwald" and "Fenella", which were ordered from the Barrow shipyard of Messrs. Vickers Armstrong, Ltd. in 1936, were primarily designed to maintain the daily mail service under winter conditions, when the volume of passenger traffic does not justify the employment of the larger and faster vessels. Accommodation is provided for 1,936 passengers under B.o.T. "Steam 2" certificate or 2,065 passengers under the "Steam 3" certificate for daylight runs in summer only. The principal dimensions are:—length 327ft. 7in. o.a.; 310 b.p. × 46ft. moulded breath × 26ft. moulded depth to the shelter deck. A high raked stem and cruiser stern are new to the Isle of Man Company's fleet, and the stem, though of "soft" construction, is associated with a bar keel. There are eleven w.t. bulkheads and five decks including a promenade deck 160ft. long with a 40ft. extension forming a boat deck surmounted by a 75ft. bridge deck. The rudder is of the Oertz type. The propelling machinery consists of Parsons turbines driving two solid three-bladed manganese bronze propellers through single reduction gearing at 275 r.p.m. for a designed speed of 21 knots. At the full ahead output of 8,000 s.h.p. the h.p. turbines which are of the impulse-reaction type, run at 3,130 r.p.m. and the l.p. turbines, which are of the single flow reaction type, run at 1,935 r.p.m. The maximum astern power is about 70 per cent. of the ahead power. All the turbines have adjusting blocks of the Michell pivoted-pad type, the propeller thrust being taken by a Michell thrust block. The main condensers are of the regenerative type, of 9,420 sq. ft. aggregate cooling surface, and are designed to maintain a vacuum of 29in. with sea-water at 55° F. Steam of 250lb. pressure is supplied by three Babcock and Wilcox oil-fired boilers of 13,560 sq. ft. total heating surface working under forced draught on the closed stokehold system. The forced draught fans, the fresh water pumps, and an emergency bilge pump are electrically driven, the remaining pumps are steam driven. The electrical equipment consists of two main turbo-generating sets of 150 kW. each and a Diesel-driven emergency set of 35 kW.; all three sets supply direct current at 220 volts. On trial the "Fenella" averaged 21½ knots as the mean of two full power runs, and the designed speed of 21 knots was obtained on 8,250 s.h.p. in the 6-hours consumption trial, when the fuel oil consumption was stated to be well within the guaranteed figure.—*"Engineering", 28th May, 1937; pp. 619-620.*

EXTRACTS.

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

The Machinery of the "Cairo" with Automatic Fuel Injection.

"The Motor Ship", May, 1937.

In the motor cargo ship "Cairo" of the Atlas Levante Linie, A.G., is installed the first Krupp two-stroke single-acting engine embodying the latest modifications of airless injection on the Archaouloff system, which has been developed by the builders. The vessel sailed on her maiden voyage on December 15, 1936, and returned to Bremen on February 22, after a trip during which the machinery behaved satisfactorily in every way. After discharging and loading, she sailed again to the East on March 5. Apart from the new features of the machinery, the ship is of interest as being the first of the Levante Line vessels to maintain a service between Hamburg, Bremen and the Eastern ports of the Mediterranean.

The "Cairo" was built to the highest class of the German Lloyd, +100 A.4 (E) "with freeboard" and the following are the main particulars:—

Length b.p.	123 metres—403ft.
Beam	17·2 metres—56·5ft.
Depth to shelter deck	10·3 metres—33·8ft.
Gross register	4,778 tons.
Net register	2,740 tons.
Deadweight capacity	7,350 tons.

She is built according to the Maier form and has a streamlined balanced rudder of the Krupp type. The cargo is carried in five holds with a total capacity of 427,000 cubic ft. In accordance with the needs of the Levante service these holds have no pillars. There are eight derricks capable of a lift of 5 tons, and six designed to hoist 3 tons. In addition, on the foremast is a heavy derrick for a 50-ton lift and, on the after mast, another for a 20-ton lift. Accommodation is arranged for 12 passengers in six single and three double-bed cabins and a dining saloon and smoking-room are provided. In the crew's quarters no cabin accommodates more than two men.

Machinery Details.

The new engine is of the single-acting two-stroke Krupp type with seven cylinders, 650 mm. in diameter, the piston stroke being 1,250 mm., and the output 3,500 b.h.p. at 115 r.p.m.

Whereas in previous adaptations of the Archaouloff principle (in which the pressure for injection is derived from the compression within the cylinder) a mechanical fuel pump has been necessary to deliver the oil to the injection pumps, this mechanical unit is now eliminated.

The cylinder blocks are supported on four cast-iron columns, three of the cylinder blocks comprising two cylinders and one a single cylinder. Combustion pressure is taken by means of tension bolts from the bedplate. The scavenging system is that which has been adopted in Krupp two-stroke engines since 1930, in which the air is supplied through main scavenging ports, whilst small auxiliary suction ports are arranged above. The scavenging air is delivered from four plunger-type pumps arranged at the back of the cylinders and driven from the cross-heads of cylinders Nos. 1, 3, 5 and 7. Hoerbiger suction and discharge valves are utilized.

The Injection Apparatus.

The injection pump, which is screwed into the pressure ring of the cylinder cover, is provided, at the top, with a water-cooled gas cylinder in communication with the combustion chamber of the engine by means of a steel pipe. A cock is fitted in the pipe so that the combustion chamber may be shut off from the pump if desired. This pipe is at relatively high temperature and is, therefore, enclosed, in order to protect the attendants in the event of its being touched by hand.

The lower part of the injection pump comprises the piston and sleeve and the suction valve. This can be regulated by means of the horizontal shaft in front of the engine. When it is turned the suction valves are closed earlier or later, according to the load on the engine, so that the injection of a larger or smaller quantity of fuel follows. The fuel is passed through an Auto-Klean filter on the left of the suction valve casing, and is delivered from the pump pressure chamber to the injection valve during the injection period through the fuel pressure piping.

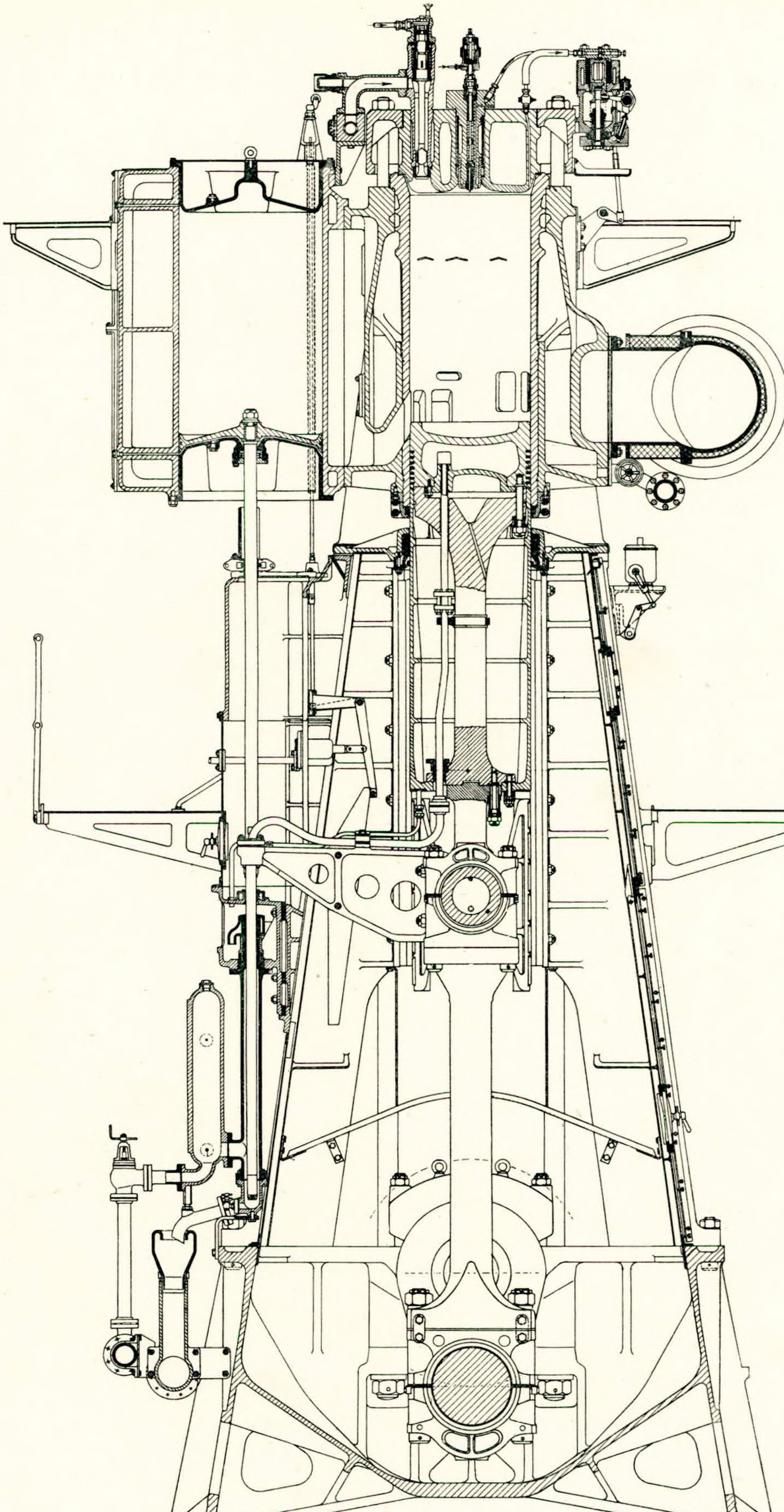
By the employment of this automatically regulated injection pump, the low-pressure mechanically operated pump employed in the engines of the tank ships "W. B. Walker" and "G. S. Walden" ("The Motor Ship", July, 1934, and December, 1935) is rendered unnecessary. Apart from the simplification of the design thereby effected through the direct regulation of the injection pump, there is the important advantage that the quantity of fuel delivered is constant, since the amount delivered from each working stroke of the pump is directly dependent upon the stroke of the injection pump. When the mechanically operated pump delivering oil to the injection pump was employed, it was possible that irregular injection might follow under the influence of air contained in the fuel or because of variation in resistance in the fuel valves.

Air Starting Arrangements.

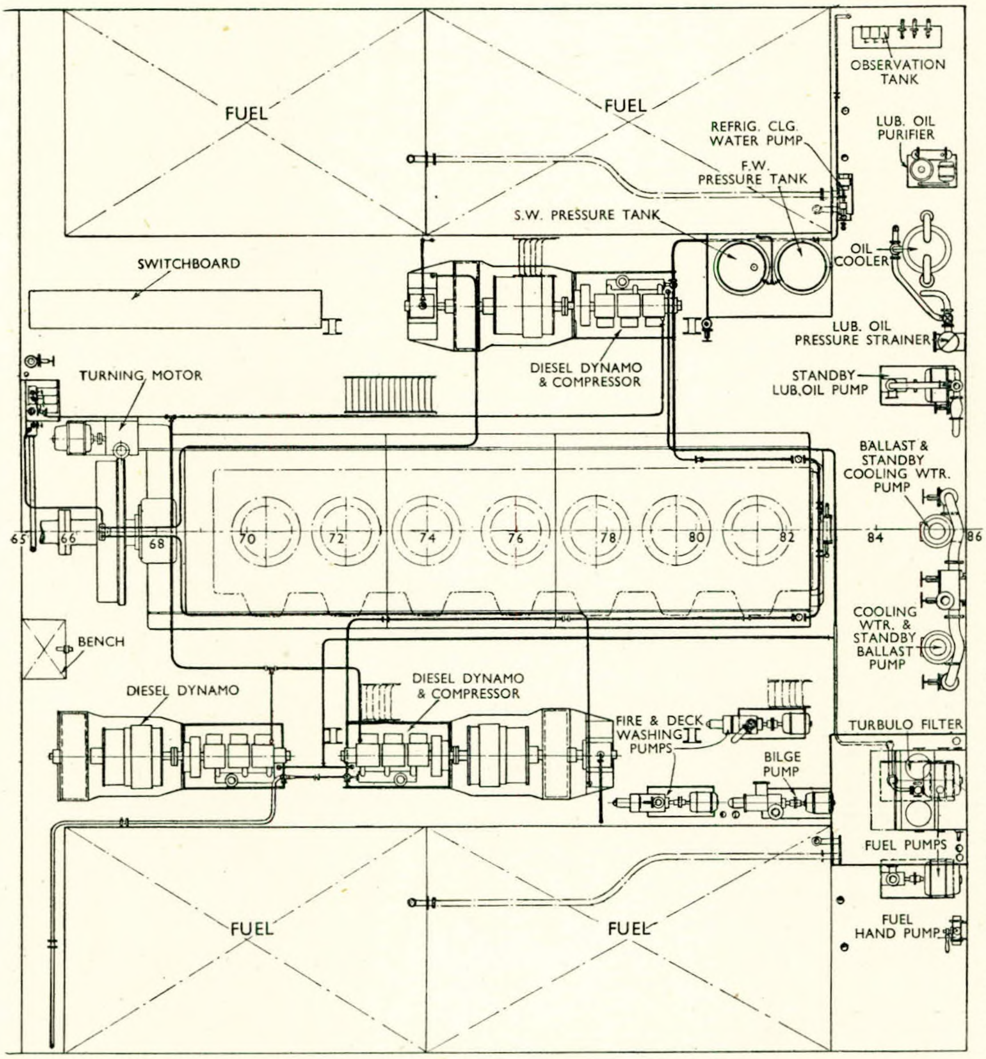
With the elimination of the low-pressure fuel pump, the short vertical control shaft at the after end of the engine has only to serve as an air distributor drive for starting purposes. This is effected in the usual manner with Krupp engines. The full pressure of the actual starting air is maintained at the starting valve during the whole period of manœuvring, but the starting valve is opened only through the intermediary of a pilot valve for a short time when starting. The starting valve is so constructed that it cannot open unless the pressure within the cylinder is lower than the starting air pressure. A back-fire is thereby rendered impossible and the fuel can be delivered at the same time as the starting air is admitted. Hence, the engine starts readily and has a small air consumption.

At the after end of the engine are the controls, which are similar to those used in the engines of the tankers previously mentioned. There are two levers, the fuel control and starting levers respectively, carried on the same spindle. Adjacent to these is the reversing wheel, which alters the position of the cams controlling the air distributor for starting, according to the direction of rotation required. A device between the reversing wheel and the starting lever has the effect of lifting the rollers of the starting valve levers from the cams so that, when reversing, the handwheel may easily be turned.

The engine cannot be started until the reversing wheel is at its end position and reversing is impossible so long as the fuel valve lever is not in the "stop" position. A further locking device prevents the engine being started, except in the desired direction of rotation according to the orders from the bridge.



Sectional elevation through one cylinder.



Engine-room plan of the "Cairo".

The cylinders, covers and pistons are all cooled by sea water, but there are two systems. One supplies the cylinder jackets, from which the water is delivered to the cylinder covers and thence overboard. A small quantity of this flow passes from the cylinder covers to the fuel valves and around the gas cylinders of the injection pumps, and then to a tank in the double bottom. Separate from the cylinder cooling water is a second system for cooling the pistons. The water is delivered to and discharged from the pistons through telescopic pipes. The water passes through open tundishes into the same double-bottom tank as that containing the cooling water from the fuel valves and the injection pump. From this tank the water is pumped overboard by an automatically controlled rotary pump, but water can also be drawn from it by the main cooling water pump, so that, when the air temperature is low, warm water can be mixed with the cold circulating water which is drawn from the sea.

The auxiliary machinery is electrically driven, with the exception of the two starting air compressors, which are coupled to the Diesel-driven dynamos through friction clutches. Each is of the two-stage Krupp type with a capacity of free air of 3·5 cubic metres per minute with an end pressure of 30 atmospheres. The starting air is stored in two electrically welded reservoirs with a capacity of 7·5 cubic metres each.

Three Diesel-driven dynamos are installed. The engines have an output of 155 b.h.p. at 500 r.p.m. and are of the three-cylinder four-stroke trunk piston Krupp type with direct-coupled cooling water pumps, which also supply the water for cooling the compressors, so that the sets can operate without any auxiliaries. An emergency Diesel dynamo compressor of 15 kW. is installed on the main deck, the engine being of the single-cylinder design capable of being started by hand.

The fuel oil is delivered from a daily service tank to the main and auxiliary engines through a Turbulo fine filter and a Krupp centrifugal separator is utilized for the lubricating oil.

It is noteworthy that the cooling water and the ballast pumps, as well as the general service and piston cooling water suction pump, serve respectively as spares for each other, so that the number of pumps is correspondingly diminished without reducing the degree of reliability in the plant. As the vessel will operate for the most part in a warm climate, the refrigerating machinery is somewhat large for the size of the vessel and an ammonia installation is provided of 8,500 k. cal. per hour; it is arranged adjacent to the cooling chambers.

The owners have ordered two similar ships with machinery of this type to be built by Fr. Krupp, Germaniawerft.

Up-to-Date Cargo Steamship.

The "Mulubinba" built by Henry Robb, Ltd., has water-tube boilers and mechanical stokers.

"The Marine Engineer", May, 1937.

The steamship "Mulubinba", built by Henry Robb Ltd., of Leith, for the Newcastle and Hunter River Steamship Company, successfully completed trials in the Firth of Forth on Thursday, April 22. In addition to features of technical interest, the vessel is interesting in that she has continued an association of long standing with Leith. In 1907 Ramage and Ferguson, whose business was acquired by Henry Robb Ltd. some two years ago, built for the Newcastle and Hunter River Steamship Company the twin-screw steamship "Hunter", which is still in service in their fleet, and in the following year also built a smaller vessel.

The "Mulubinba" will be engaged in general trade on the Australian coast between Newcastle and Sydney. She is particularly interesting as being the first British-built coasting cargo vessel to be fitted with mechanical stokers; numerous Dutch-built-and-owned vessels with stoker-fired water-tube boilers have, of course, been in service in the Dutch East Indies for a number of years. Her main dimensions are: length, 220ft.; breadth, moulded, 39ft.; depth, moulded to main deck, 16ft.; depth to upper deck, 24ft. She has been built in accordance with Lloyd's Register's highest class and the Australian navigation requirements. The hull design was tested in the experiment tank at the National Physical Laboratory, and power in excess of that required for the trial speed of $11\frac{1}{4}$ knots, fully loaded, is provided. The steel work and scantlings and structural arrangements are equal to Lloyd's Register's highest class, and in certain cases are in excess of these requirements.

Cargo arrangements.

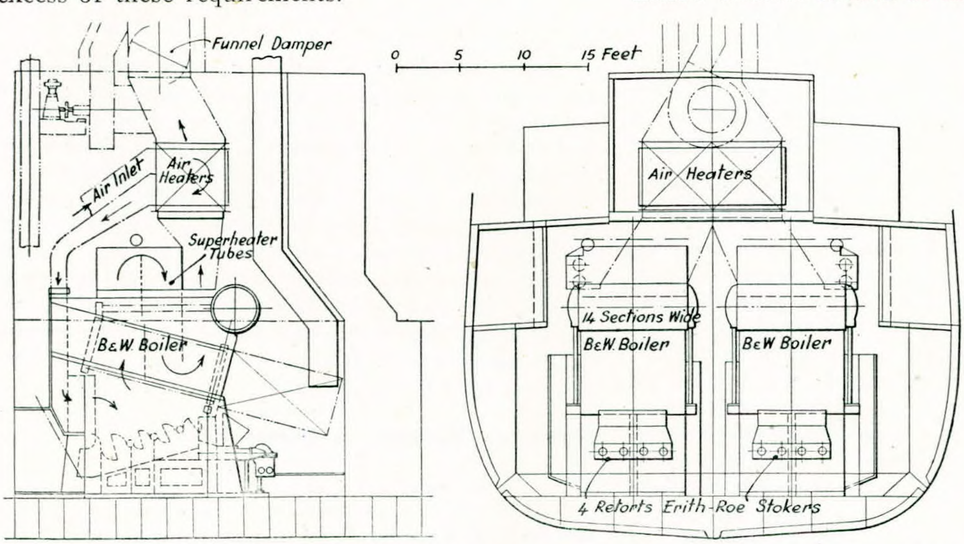
The steam steering gear is of the Wilson-Pirrie type, by John Hastie & Co. Ltd., and is operated by their Atlantic-type engine, it being actuated by telemotor gear from the flying bridge. The compass was supplied by Kelvin, Bottomley & Baird Ltd. In addition to the main cargo hatches there are two large cargo doors on each side of the ship giving easy access to the 'tween decks and facilitating the handling of cargo. The main business in which the ship will be engaged is coal, general cargo, and structural steel carrying. The derricks have capacities of 10 tons on the foremast, $7\frac{1}{2}$ tons on the main mast, and $2\frac{1}{2}$ tons on the samson posts. The deck machinery includes four steam winches, two 8in. by 12in. and two 7in. by 12in., with a steam windlass forward and steam capstan aft. All the derrick gear and cargo-handling appliances are of special design in accordance with the owners' requirements for the quick handling of cargo.

The accommodation for deck and engineer officers and crew is of a high standard. In addition, there are four cabins, each fitted with cot beds and pullman berths, to carry two persons, so that eight passengers may be carried. For these a dining room and smoke room are provided.

Machinery.

The propelling machinery, by the North Eastern Marine Engineering Co. Ltd., consists of a set of triple-expansion reciprocating engines of their latest design taking superheated steam at 215lb. per sq. in. pressure and 550° F. final temperature from two water-tube boilers fitted with mechanical stokers. The machinery has been built to the requirements of Lloyd's Register's highest class and the Australian Commonwealth Navigation Act. The engine cylinders are $16\frac{1}{2}$in., $28\frac{1}{2}$in., and 48in. in diameter and the piston stroke is 33in. The cylinders are supported on strong cast iron columns, of hollow circular section at the front and box section at the back, well secured to the cylinders and bedplate, and are arranged over a three-throw crankshaft with the h.p. cylinder in the middle.

To satisfy the requirements of superheated steam and to improve the engine efficiency, the h.p. and m.p. cylinders are fitted with N.E.M. double-beat poppet valves for steam inlet at top and bottom and for exhaust at top and bottom, the valves being operated by cams on transverse



Showing arrangement of the stoker-fired Babcock & Wilcox boilers in the "Mulubinba".

shafts at the forward and after ends of the engine, actuated by the conventional Stephenson type of link motion. The h.p. cylinder is fitted with a separate liner of Lanz-Perlit iron, and Copeland's metallic packing, to suit superheated steam, is fitted to the piston rod stuffing boxes. The cylinder lubricating arrangements are specially designed to suit superheated steam conditions and to give good lubrication with a minimum consumption of oil, Kirkham T. & K. lubricators being used. Lockwood & Carlisle piston rings are fitted throughout.

The condenser is pear-shaped, built of steel plates, securely bolted to the back columns and designed to maintain 27in. of vacuum with a sea temperature of 80° F. Cooling water is provided by a Drysdale centrifugal circulating pump having 11in. diameter branches, and is driven by an enclosed force-lubricated steam engine. The pump is capable of delivering 1,300 gallons of water per minute through the condenser. The air pump, two hotwell pumps, two bilge pumps, and a sanitary pump are driven by strong plate levers coupled to the crosshead of the h.p. engine. The reversing engine is of the Brown Bros.' patent steam-and-hydraulic direct-acting type and is arranged so that reversing can be accomplished by steam or hand through rack and pinion gearing. A combined North-Eastern balanced type stop and throttle valve of Perlit iron is fitted, with special metal fittings. All controls are placed within easy reach of the engineer on watch. The telegraphs were supplied by Donkin & Co. Ltd., Newcastle-on-Tyne.

The propeller is four-bladed, solid bronze, and is the result of exhaustive tank experiments in order to obtain a design of maximum efficiency.

Babcock boilers.

A special feature of the machinery installation is that steam is supplied by two Babcock & Wilcox sectional type water-tube boilers having a total tube heating surface of 4,200 sq. feet. They are arranged for forced and induced draught, producing superheated steam and employing Erith-Roe mechanical stokers. The water and steam drums are 3ft. 6in. in diameter, 10ft. 2in. long, and the generating tubes are 4in. diameter by 9ft. 9in. long; baffle plates are fitted to conduct the gases three times across the generating tubes. The working pressure is 235lb. per sq. in., giving a pressure of 215lb. per sq. in. at the superheater outlet. The superheaters are of the superimposed type, composed of seamless steel tubes bent into the form of a "U" and expanded into mild steel headers. Dampers are arranged in the combustion space to regulate the steam temperature. Two Howden's fans are provided, one for forced draught and one for induced draught, these fans being driven by Howden-Thermall high-speed steam engines. Although the balanced draught system is used, the induced draught fans are only used at maximum ratings. The furnaces are arranged for burning coal, this being fed to the hoppers on the Erith-Roe retort rear-ashing-type mechanical stokers which are worked by two electric motors of 4 B.H.P. each. Each boiler is fitted with one Clyde blower and four Diamond soot blowers for cleaning the boiler, superheater, and air heater tubes, all blowers being supplied with superheated steam.

A North Eastern gravitational feed water filter is provided to deal with the small extra amount of cylinder lubricant used and as an additional precaution to avoid the possibility of oil entering the boilers, North Eastern oil interceptors are fitted between the l.p. cylinder and the condenser, and also in the auxiliary exhaust range.

Independent auxiliaries.

The independent auxiliary machinery includes a pair of Weir's feed pumps, 6in. by 8½in. by 18in., either pump being capable of dealing with all the feed water required when running at full power with the other pump as spare; one Weir's ballast pump 9in. by 8in. by 18in., capable of dealing with 100 tons per hour; one Weir's auxiliary feed and general service pump 6in. by 8½in. by 18in., and one fresh water pump 5in. by 4½in. by 12in., also supplied by G. & J. Weir Ltd. There is also one Dawson & Downie vertical duplex pump, 5in. by 8in. by 8in., to serve the See's patent ash ejector and the wash deck line; one 10-ton evaporator and one 10-ton distiller supplied by Davie & Horne; and a winch condenser of 400 sq. ft. cooling surface supplied by the North Eastern Marine Engineering Co. Ltd., and there are two electric generating sets supplied by Belliss & Morcom, driven by enclosed force-lubricated compound steam engines. Advantage is taken of the superheated steam for driving all the engine-room auxiliaries, dynamos, and deck machinery; the steering engine is arranged to operate with saturated steam only.

It should be noted that all valves for superheated steam have internal fittings of special metal, the steam pipes being of solid drawn steel, while special attention has been paid to the lagging of the steam pipes and cylinders.

The machinery installation represents the most up-to-date steamship engine-room practice and is economical, reliable, easy to operate, very accessible and will retain its efficiency at varying powers and conditions of service over an indefinite period. We attended the trials of the vessel and were favourably commented upon by many of the engineers machinery. The boiler and stoker plant was favourably commented upon by many of the engineers present, the cool and cleanly stokehold and absence of smoke at the funnel exit being commented upon. A top speed of 12·46 knots was reached—one knot above contract requirements.

Notes on Refitting a Diesel Engine.*

By Lancastrian.

"The Marine Engineer", May, 1937.

It would be unwise to be dogmatic about the interval of time allowed between periodical refits,

* From "The Naval Engineering Review".

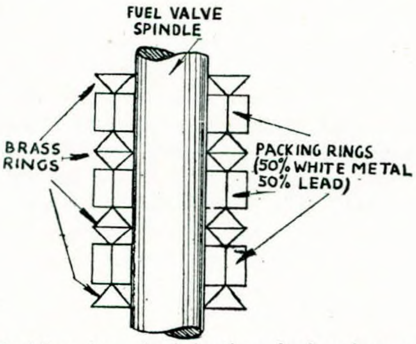


FIG. 1.—Packing for air injection fuel valves which the author has found satisfactory.

but a total of 3,000 to 3,500 running hours should not be exceeded. Just prior to refitting it will be found useful to take a set of indicator cards, which can then be compared with a set taken shortly after the refit. It will also prove useful to measure the compression clearances before the covers are disturbed.

The extent of a refit naturally depends upon several factors—time at one's disposal, number of men available, condition of engine details on opening up, etc. As a rule, however, all top gear and valves are refitted, piston rings renewed as necessary, liners gauged for wear, top and bottom end bearings adjusted, and two or may be three main bearings lifted for examination. The air compressor will require attention, as will also the camshaft drive, governor and fuel pumps. Advantage must be taken of the opportunity of cleaning water and gas passages which are normally inaccessible.

Exhaust Valves.—Remove these by means of lifting bolts and not by banging the top with a big hammer—such methods may be effective, but also make a good job of breaking the spindle end where the split pin is fitted. If the valve is obstinate, take out the induction valve (these usually come out easier) and turn the exhaust valve out with a block of wood placed under it. Examine the cage for bulging and cracks, and the valve for distortion, cracks and bent stem. Resort to skimming as little as possible. See that the spring is of correct length and strength. When assembling take particular care to screw the nut hard up before inserting the split pin—if necessary file the spindle end or use packing pieces. The clearance in the guide is about 7/1000 of an inch. Before tightening down make sure that the cage is free to settle itself squarely on the pocket seating. Tightening down is a matter of experience; if too tight, the cover may crack or the cage may bulge; on the other hand, compression will be lost if it is not tight enough. Some nuts are awkward to get at, and one is repaid by having a special spanner made. It is a good plan to follow up the cover nuts after a run.

Induction Valves.—These should be treated as for exhaust valves. The routine examinations are, of course, not so frequent—being about a third as often.

Air Start Valves.—These give little trouble, but the emulsified moisture present in the starting air may make the spindle a bit sticky. The white-metal stem should, therefore, be worked in the guide to ensure complete freedom. Springs should be examined. It might be mentioned that particulars of springs can usually be found on drawings and it is well worth while looking into the matter. One is apt to neglect springs unless they are actually broken.

Fuel Valves.—If difficult to remove do not risk breaking the casting by using wedges, but remove as for exhaust valves. Examine the gland and seating. Fig. I shows a type of packing which has been quite satisfactory in fuel valve glands. It requires to be tightened and the spindle rubbed in to work up a good surface. See that the pulverisor tube is free—sometimes bits of grit get between it and the spindle and cause the valve to stick open. Look for indications of "masking" at the bottom of the body. An exaggerated effect of "masking", and a simple remedy is shown in Figs. II and III respectively. Flame plates should be renewed if the holes are enlarged, or belled out more than 15 per cent. of their original diameter. The spring in the cap must be of correct strength—if too heavy it may cause the seating to hammer, and if too tight there will be less tendency to leak.

Cylinder Covers.—With the present one-piece type of cover* there is little one can do beyond cleaning up valve pocket seatings and cleaning out water and gas passages. The first thing to do on removing the cover is to clean the underside and examine for cracks, particularly between valve pockets. When one remembers that the cover is arched when tightened down and under load one may be puzzled as to why cracks develop. A suggested reason is—being arched, the underside metal is under a compressive strain. Now if the engine is overheated the metal weakens, and due to this compression the outer surface of the underside may shorten and take a permanent set. On cooling, the

* The author's remarks primarily concern the Mirrlees air injection engine, but are nevertheless of general interest.—ED.

FIGS. 2 (above) and 3.—The effect of fuel valve masking and a simple remedy are shown.

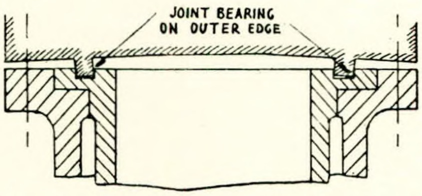


FIG. 4.—The shock plate suggested by the author may form on the outer edge only.

cover will tend to straighten out and a tensile strain will be imposed upon the shortened surface. Repetition of this alternate straining will result in a crack.

To reduce arching, "shock plates" are sometimes fitted between the cover and the breech-end. Opinion differs as to the value of these plates or washers. Where the explosion ring joint is a ground metal to metal spigot, or a ground ring like an un-cut piston ring, shock plates serve a useful purpose in that they prevent the tendency of the joint to form on the outside edge only (see Fig. IV). On the other hand, if the joint is of soft material such as wire-woven asbestos, it is not so much affected. Furthermore, a soft joint sometimes requires to be "followed up" and shock plates hinder this being done, as their removal necessitates wedging up the cover. The only method of fitting plates that the writer has used (on covers with ground joints) consists of putting lead wire round the studs, dropping and tightening down the cover, lifting again and measuring the average thickness of the lead. The plates are then made about 2/1000 of an inch thinner and should be just nipped when the cover is finally down. There is no need to "flog-up" cover nuts until they will not move a fraction further—if the joint is a good one to start with it will not require brute force to make it tight.

Liners.—Examine for signs of partial seizures, score-marks and ridges. In the Mirrlees engine, examine the recesses at the top for cracks. Gauge for wear on the diameter, fore and aft and athwartships at four points, viz.:

(1) At the bottom of the skirt.
(2) Midway.
(3) At the top end of the top ring travel.
(4) At 1in. above (3) if not counterbored.

For a 12in. diameter, wear in excess of 4/1000 of an inch per inch would merit renewal of the liner.* If one allows too much slackness between the piston and the liner one must expect seizures and piston slap. Symptoms of seizure are reduction of engine speed, labouring, and knocking or thumping. There should be no hesitation about stopping an engine if any of the above signs become apparent.

* In our opinion the author's recommendation is unduly cautious. A 12in. liner should run satisfactorily until a wear of about 60 thousandths is recorded. At the period in the liner's life when, in the author's view, it should be scrapped, we consider that Triple-Seal or Double-Seal piston rings should be fitted. In this way another 10 or 12 thousandths of useful life should be obtained from the liner.—ED. M.E.

The main point about removing a liner is to have really stout appliances; it is no use trying with stuff that will bend easily. For example, the bottom strong-back should be a piece of steel about 3in. by 2in. section and turned and recessed at the ends to fit the liner skirt. It will then stand the weight of a "jack" if one is required. Once the initial move has been made, the liner usually comes out easily (unless one forgets to remove lubrication fittings). After removal, the jackets can be cleaned and red-leaded. Before putting the liner back, smear the rubber joints with soft soap; it helps them to enter.

Piston.—Remove all rings and see that the grooves are square and in good condition. Clean and examine the crown for cracks. Before knocking out the gudgeon pin see if it is slack in the body by standing the piston on end and wobbling the connecting rod fore and aft. Do not mistake the top end clearance for slackness in the body, but watch the bore for oil squeezing out. When fitting piston rings, remember the state of the liner; if the liner is worn you cannot afford to have gaps as small as they may otherwise be, or broken rings will result. Though not usually supplied as spares, rings having a lap joint make a better seal than the more common diagonal cut. The side clearance in the grooves is about 2-3/1000 of an inch, usually 3/1000 for top rings. Rings are designed to give a pressure of about 4 to 6lb. per sq. in. on the cylinder wall.

Connecting Rod Bolts.—As the failure of one of these bolts may have very serious results it behoves one to examine them very carefully for flaws. They can be checked for stretching by having a "pop" mark on the side of the head, and at a known distance apart another mark on a flat filed on the thread. Similarly, a line scribed lengthwise on the bolt will reveal twisting. Feathers should be fitted in the side of the head and not in the root of the neck. A piece of advice worth acting upon is—scrap a bolt if there is any doubt about its soundness, as the consequence of fitting a doubtful one may be serious. A bad practice is that of tightening up too much. A little thought and judgment is far better than the merciless use of a heavy hammer. Particularly is this so when it comes to screwing up the locknut. Who has not been sorely tempted to drive it that $\frac{1}{8}$th more to accommodate the split pin? Don't do it—file the nut.

Preparation.—Fill the sump with the required quantity of clean oil and fit a clean strainer in the suction line and have one "stand by". Use the lubricating hand pump until oil flows or drips from all the bearings and see that it reaches the highest point of lubrication (top governor bearing). Note the temperature.

Fuel systems will require flooding and venting, and strainers should be clean. Similarly the circulating water passages require to be flooded and it should be ascertained that the cylinders and gas side of the silencer are free from water. Try blast air on system to locate any stray leaks and rectify as required.

Running.—Blow the engine round for a few revolutions with starting air, and use the lubricating hand pump during the revolutions. If everything appears all right, bar round to a start position and run the engine on fuel for not more than 1½ minutes. This is perhaps the most critical part of the trial. In this short time, bearings, etc., which are incorrectly adjusted often reveal their discrepancies before it is too late. Make no attempt to recharge the air bottles—nine times out of ten this could easily be done—but who is to know that this is not the tenth time? The writer recalls such a "tenth time" when, before recharging was complete, the air compressor L.P. plunger had seized and literally ripped off the big end bearing, and bent the connecting rod. Had the engine been stopped a minute earlier, several days' work would have been saved.

On stopping, remove crankcase doors and feel all bearings and liner skirts, etc. If satisfactory, start the engine again and run long enough to recharge the air bottles. Meanwhile, one's eyes, ears and nose should be applied diligently to the tracing of trouble. As soon as the bottles are re-charged, stop the engine and repeat previous inspection. At this stage, bearings should still be cold and if any are decidedly warm (not necessarily hot) it may be doubtful whether they will run themselves in, although sometimes quite a warm bearing will do so if treated with care in the early stages.

If, after inspection, everything appears to be correct, run the engine without load for about half an hour. During this time keep a careful watch on the lubricating oil pressure and temperature. An undue drop in pressure may be caused by a dirty strainer (there will probably be a fair amount of loose foreign matter in the sump), but the strainer must not be changed whilst the engine is running. During the brief time the strainer is out it is possible—and has in fact happened—for a stray bit of waste to get into the system and cause considerable damage by choking the oil supply. Be alert for any unusual noises, and should a whitish vapour issue from the crankcase vents it most likely indicates a very hot bearing. Do not hesitate to stop the engine if there is any doubt, and if this has to be done because of serious defect in the crankcase, the doors should not be removed for at least five minutes, and no naked lights must be brought near.

Assuming the engine stands up to these trials satisfactorily, it will be reasonably safe to run for half an hour or an hour on about a half to three-quarters load. Before starting up, however, it would be advisable to change the strainer so as to have a clean start. While running on load, keep the same strict watch on oil pressure and temperature and constantly feel round. Some people claim to be able to detect hot bearings by the heat of the crankcase doors; this may be so where the doors are thin, or of an aluminium alloy, but it is very doubtful whether a thick cast-iron door would be so communicative. At the end of the period stop the engine and have a thorough inspection, not forgetting to see that all keeps and fastenings are intact; then, providing everything is satisfactory, there should be no need for a further trial. It would be advisable, however, to have a competent person present during the first three or four hours the engine is on load. The lubricating oil strainer should be changed daily until no trace of dirt remains. Incidentally, the oil should be renewed every three or four months, and the crankcase cleaned.

An American Geared Turbine for Cargo Ships—A Correction.

In connection with the above article, which was reproduced in the May TRANSACTIONS from the April issue of "The Marine Engineer", we have received the following letter from Mr. Alfred E. Jordan (Member) :—

"17, Battery Place,
New York.
June 8th, 1937.

The Secretary,
The Institute of Marine Engineers.

Dear Sir,

I would like to correct a statement appearing on page *64* of the May issue of the TRANSACTIONS, in which it states that the Luckenbach Steamship Company has four vessels which are each propelled by a 2,500 s.h.p. installation of De Laval *double* reduction compound geared turbines, the ships being the 'Harry Luckenbach', 'Lillian Luckenbach', 'Dorothy Luckenbach', and 'J. L. Luckenbach'. As a matter of fact these vessels are twin screw with *two* 2,500 s.h.p. *single* reduction compound geared turbines.

Yours very truly,
ALFRED E. JORDAN,
M.I.Mar.E."

Honour where Honour is Due.

"Syren and Shipping", 2nd June, 1937.

The rank of Commander of the Order of the British Empire bestowed upon Mr. Llewelyn Roberts, "Chief" of the "Queen Mary", has been the subject of singularly little comment in either lay, shipping or engineering journals. The honour is itself singular, and even if it has been merited to an extent that renders emphasis unnecessary—which indeed it has—the fact of an operative engineer being placed in so high a class of a chivalric order has some aspects which call for presentation. Bachelor knighthoods for merchant captains are a commonplace, taking precedent from at least Elizabethan times, when, however, the spur was won rather by the sword than by the compass. Since then there has perhaps never been a time when honour for merchant sailors was more justified or more wise than now; and of all who face the ardours of the sea to-day as officers, it is the engineer officer who is least compelled to do so. He can leave it in times like these without the same prospect of almost certain disaster that faces a deck officer trying his luck ashore. In fact, with that diminishing band who still seek to belittle the importance of the engineering officer it is a stock argument that when he comes ashore it is usually to take a place as a "working man" in an engine shop and that this stigmatises him as something of an impostor in authority at sea. The answer is that the more decorated naval engineer cannot even do that (supposing he wished to) except in a few cases where his training has given him a real, as distinct from official, qualification in manual work. Those who move in an atmosphere of honour or honours think pretty small beer of a knighthood that is not of an Order and will not miss the significance of the particular honour chosen for a marine engineer; it seems to recognise that craftsmanship and chivalry make no antithesis. Is it possible that there has been a desire in high places to settle this genteel nonsense once and for all? We hope so; but in any case Mr. Roberts was the right officer to select for the honour, both for the obvious reason of his position and for his personal work in it. The machinery defects developed in the "Queen Mary" proved to be small, but only a chief operating engineer can understand the anxiety and strain of coping with such concealed derangements in heavy machinery whether ashore or afloat, but especially afloat. And the prestige of Britain was in those turbines. In the more fraternal spirit now growing in the profession, engineers of all sorts and conditions would be glad to see further honours bestowed where due in this peculiarly arduous branch. They need not be confined to the big passenger liners: engineer officers of long or conspicuously good service in the smaller companies should receive proportionate recognition from the State.

Honouring the Sea-going Engineer.

"The Syren and Shipping", June 16th, 1937.

The writer of the following letter is Mr. Llewelyn Roberts, M.I.Mar.E., chief engineer of the "Queen Mary".

To the Editor of "The Syren and Shipping".

Sir,—I have read your paragraph "Honour where Honour is Due" with much interest and pleasure. Personally I am grateful for the honour conferred on me by His Majesty the King in the Coronation Honours List, but I am more pleased that sea-going engineers as a fraternity have been recognised at last by the State.

Your article will be greatly appreciated by all sea-going engineer officers, and it would be pleasing indeed if other newspapers would follow your example in this connection.

May I thank you on behalf of sea-going marine engineers for what you wrote.—Yours sincerely,

LLEWELYN ROBERTS.

Geared Diesel Machinery with Electric Couplings.

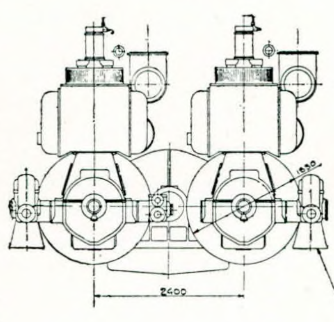
A 4,400 b.h.p. Four-engined single-screw Installation.

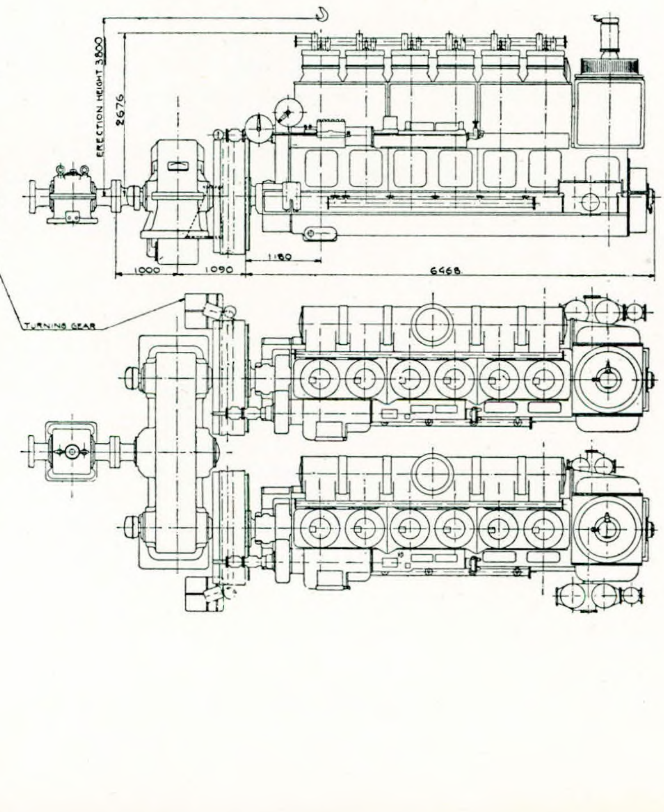
"The Motor Ship", June, 1937.

There will shortly be completed the first large installation of Polar Diesel engines geared to the propeller in conjunction with the new design of A.S.E.A. electric coupling. So attractive has this arrangement appeared to Scandinavian owners that already there are nine ships under construction in which it is being employed, either with two or four engines, and in some cases the four engines are coupled to a single propeller shaft. A few details of the system employed are, therefore, of interest.

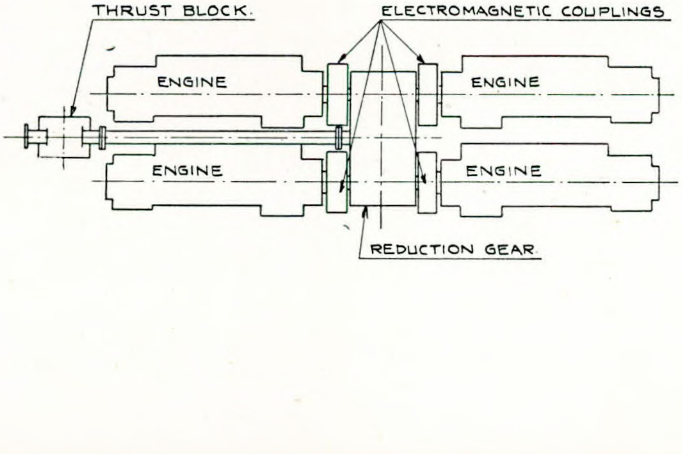
The object of the employment of the electric coupling between the engine and the gearing is to

Elevations and a plan of a single-screw unit with two engines (right).





Arrangement of a 4,400 b.h.p. geared Diesel installation (below).



eliminate high-frequency variations of torque, thus protecting the gearing as well as the shaft against heavy stresses should the propeller be suddenly stopped. The coupling comprises a solid iron rotating part attached to the engine shaft and having a non-insulated short-circuited winding, rotating within a multi-polar magnet, the clearance being between $\frac{1}{8}$in. and $\frac{1}{4}$in. The slip is in the neighbourhood of 1.2 per cent., at normal speed, and the efficiency is about 98.5 per cent. Direct current is employed for the excitation. In the case of two engines driving a single propeller shaft through electric couplings and gearing, they are manœuvred together and a similar arrangement is adopted for four engines coupled to one propeller. Two ships are being built with such a design, each of 9,500 tons, and equipped with four Polar engines developing a total of 4,400 b.h.p. and driving a single screw at 90 r.p.m.

Designs have been prepared for installations of much higher power, including twin-screw ships, each propeller being driven by four engines of 1,460 b.h.p. Triple-screw installations for passenger ships up to 20,000 b.h.p. are also being considered.

Residual Stresses in Welded Plates.

"Shipbuilding and Shipping Record", May 27th, 1937.

It is generally recognised that one of the great difficulties to be overcome before welding can be extensively employed in place of riveting for the strength portion of the hull of a ship is the elimination—or at least the substantial reduction—of the residual stresses set up as a result of the unequal expansion of the various parts of the structure. These residual stresses tend to become greater the more complex is the design of the welded structure, and it is apparent that with the additional stresses due to subsequent loading of the structure in service, fracture may ensue. An interesting investigation into the nature and magnitude of these stresses was given by Mr. H. E. Lance Martin in the course of a paper recently read before the North-East Coast Institution of Engineers and Ship-builders, in which mild-steel specimens which were annealed before welding were welded in the form of unrestricted specimens and measurements made of the strains, both in the direction of the weld and at right angles, by means of an instrument termed a comparator, capable of measuring to 0·000005 of an inch. If the elastic limit of the specimen has not been exceeded, the measurement of the strain permits of the computation of the stress. It is of interest to note that very consistent results were obtained showing that stresses of over 20 tons per sq. in. occur in the weld metal and in the adjacent parent metal in the cases of the unrestricted edge-welded plates.

A Ministry of Marine.

"Shipbuilding and Shipping Record", May 13th, 1937.

Sir Robert Burton Chadwick, Master of the Honourable Company of Master Mariners, and former Parliamentary Secretary to the Board of Trade, said at the annual meeting of the Mercantile Marine Association at Liverpool, on May 4:

"You will never get the best merchant navy in the world while you have this ineffective political service in London. Until we have a Minister and a Ministry of Marine, supported by a department solely concerned with British shipping, we are not doing the best for merchant shipping.

"A Cabinet Minister with a full voice in the Cabinet is needed to lead this country in its tremendous mercantile interests".

Engineers for the Navy.

New Admiralty scheme.

"Shipbuilding and Shipping Record", May 13th, 1937.

The Admiralty announce a new scheme for direct entry of engineer-officers into the Navy from the universities. The following is the text of the proposals:—

A number of commissions as engineer officers in the Royal Navy will be in future offered to university graduates each half-year. Twenty such commissions will be offered next September. Detailed regulations and entry forms may be obtained from the Secretary, Admiralty, London, S.W.1, or the authorities of the following universities: Birmingham, Bristol, Cambridge, Durham, Leeds, Liverpool, London, Manchester, Oxford, Sheffield, Aberdeen, Edinburgh, Glasgow, St. Andrews, Wales, and Belfast.

An applicant must (*a*) have attained the age of 21 and not have attained the age of 25 on January 1 for appointment in the following February, or July 1 for appointment in the following September; (*b*) be unmarried; (*c*) be a British subject of pure European descent and the son of parents who are British subjects at the time of the candidate's entry; (*d*) have undergone a course of not less than three academic years as an internal student (in certain circumstances the first year at a recognised Dominion university will be allowed); (*f*) produce satisfactory evidence that he has been regularly trained in mechanical engineering and has sufficient practical experience; and (*g*) possess an engineering degree (mechanical or electrical) which is accepted by the Institution of Mechanical Engineers as exempting from parts A and B of the examination for associate membership.

Applications must be made on the proper form to the authorities of the candidate's university by June 10.

Valve-seat Wear.

"Shipbuilding and Shipping Record", May 20th, 1937.

It has been found as a result of actual experience in the running of diesel engines at sea, that the incidence of wear of the valve seats—particularly those of the exhaust valves, is not merely dependent upon the quality of the material of which the seat is constructed. We noted in a technical contemporary an account of an investigation of the factors influencing the wear of valve seats, from which it appears that one of the most important causes of wear is the oxidation of the surface of the metal by the exhaust gases. The author is thus led to the conclusion that wear is determined to a large extent by the quality of the mixture of air and fuel led to the cylinder; if the air is in too great an excess, there is an excess of oxygen in the exhaust and owing to the temperature of the gases and also the temperature of the valve seat itself, conditions are conducive to oxidation of the surface with subsequent rapid wear. It is not always possible to use a rich mixture in the cylinder with the sole object of reducing valve seat wear, but the point is worthy of being kept in mind when designing the air intake system of the engine. Other factors to be noted are the efficient cooling of the seats and the elimination of the hammering effect of the valve on its seat due to excessive tappet clearance. The author's investigations were, it should be noted, carried out on motor car and lorry engines, including in the latter those using heavy oil as fuel, but his conclusions should have some bearing on the problem as applied to marine diesel engines.

High-Duty Cast Irons.

"Shipbuilding and Shipping Record", May 13th, 1937.

The term cast iron is used to denote a very wide variety of metals ranging from the material used for, let us say, garden railings, to the cylinder liners of a marine diesel engine. But even so, it is open to question whether the average engineering designer is fully aware of the enormous strides which have been made in recent years in the metallurgy of high-duty cast irons. It will be recalled that a short time ago a paper entitled "Cast Iron and Its Applications in Engineering", by Mr. A. Campion, was read before the Institution of Engineers and Shipbuilders in Scotland, during the course of which the author gave some remarkable data relating to the properties of certain cast irons. In the subsequent discussion a number of speakers were compelled to admit that they were unaware that such irons were available, and the opinion was expressed that the time has come to discard the ideas of cast iron based upon the data of 10 or 20 years ago. It should be recognised that cast iron is now a highly specialised metal capable of being accurately controlled to meet specified service conditions and having properties far in excess of the ordinary grades of cast iron with which the engineer had to be content a few years ago.

It will be of interest briefly to recall some of the more exceptional features of the special grade of cast iron described by Mr. Campion, in order that some idea may be obtained of the extent to which this material is superior to the more ordinary grades of cast iron. In the first place, it should be noted that whereas 10 years or so ago a tensile strength of 16 to 18 tons per sq. in. would have been considered something exceptional, to-day cast iron with tensile strengths of from 25 to 30 tons per sq. in. are regular products of the cupola and strengths approaching 40 tons per sq. in. have been obtained. Mere tensile strength, however, is not the only desideratum; it is when such properties as resistance to fatigue, to the effects of high temperatures, and to the effects of abrasion or erosion are taken into account that the merit of these new cast irons, as alternatives to forged steel can be gauged. A further desirable characteristic is the ability to control the nature of the finished product so that it can fulfil a definite specification. The author was able to give figures showing that in the as-cast condition, the particular type of cast iron which he was describing had a true elastic limit of 6·4 tons per sq. in., with a yield point of 21 tons per sq. in., while in the heat-treated condition these figures were 9·5 tons per sq. in. and 27 tons per sq. in., respectively. The respective moduli of elasticity were 8,940 tons per sq. in. and 11,600 tons per sq. in., while the Brinell hardness numbers were 241 and 302. In order to determine the resistance to the effects of high temperature, similar specimens were tested at temperatures of 15° C. (59° F.) and 400° C. (752° F.), and these showed that the elastic limit was reached at practically the same stress in each case, viz., 6·04 tons per sq. in., although the ultimate stress at the high temperature was only 19·8 tons per sq. in., as against 21·6 tons per sq. in. at atmospheric temperature, and the modulus was 6,182 tons per sq. in. at 400° C., as compared with 7,598 tons per sq. in. at 15° C.

The ability of the material to resist the effects of fatigue shows, to an even greater extent than the figures given above, the superior properties of these high duty cast irons in comparison with the ordinary grade gray cast iron. Whereas specimens of gray cast iron were found to have an endurance limit in the region of 10,000lb. per sq. in., the special iron as-cast gave a figure of 21,000lb. per sq. in., while the heat-treated specimens had an endurance limit of 28,000lb. per sq. in. Attention may also be drawn to the results of impact tests, which have a Charpy value ranging from 1·5 to 6·0ft.-lb. in the as-cast condition and up to 11·0ft.-lb. after heat treatment. It should be remembered that the average Charpy impact value for 25 different cast irons, as determined by the American Society for Testing Materials was 1·26ft.-lb., and that only two showed a value in excess of 2·5. But encouraging as these results appear to be, it is worthy of note that during the course of the discussion the suggestion was made—and the author in his reply expressed his agreement—that from the metallurgical point of view, the production of high duty cast iron is still in its infancy. The British Cast Iron Research Association is, it is stated, on the eve of further very important fundamental discoveries dealing with the control of graphite and the consequent improvement of physical properties, while the question of corrosion, particularly that type known as "graphite softening", is receiving attention by several authorities and a solution is definitely in sight. Of course, it is to be expected that the cost of these special cast irons must be in excess of that of the ordinary grades, but it is gratifying to place on record that the increase is such that it still leaves the final figure well below that of steel, and that, as a consequence, the use of high-duty cast iron, as, for example, for crankshafts, and camshafts, is rapidly extending.

Singing Propellers.

"Shipbuilding and Shipping Record", May 20th, 1937.

The paper on singing propellers by Mr. Harry Hunter (published in our issue of February 18 last), and of the discussion which followed, have been issued as a single volume by the North East Coast Institution of Engineers and Shipbuilders. Included are a great number of written contributions. The matter makes interesting reading. It was to be expected that many different opinions would be expressed regarding the cause of the troubles which have manifested and the cure for these disturbances, but it cannot be gathered from the remarks that have been made that either the cause has been definitely established or the obviation of singing made certain. The consensus of opinion seems to be that singing is caused by the collapse of cavities in the region of the screws, and Mr. Stanley Cook points out that the collapse of quite small cavities would correspond to an appreciable blow. An important point is made by Mr. Cook, namely, that there is no recorded instance of propeller noise during astern running, which fact indicates the probability of the hammer blows being caused by the propeller blades cutting through vortex streams generated by some member ahead of the blade.

The extremely small differences in design which may cause one propeller to sing and another not to, adds difficulty to the task of obviating the disturbance. Sir Amos Ayre instances the case of a bronze propeller which did not sing, but the blades of which were on the thin side from considerations of strength. On a new propeller which was fitted the opportunity was taken to thicken up the blades, but this one provided a particularly bad instance of singing. Sir Amos was informed that a few degrees of helm caused the singing to cease and, later, that the noise had been damped down by the frequent application of paint.

Mr. Neville, whose firm, J. Stone & Co., have had remarkable experience in the design and manufacture of propellers for all classes of ships, expresses the opinion that the solution of the problem of the singing propeller lies in the proper design of the screw rather than in the particular alloy or maker. During the last two years, he says, he has had experience of a system of propeller design which has as its object the highest possible efficiency with freedom from vibration and erosion, and in approximately a hundred consecutive cases covering nearly all classes of vessels and engines there has not been reported a singing propeller—"a fact which should give confidence to all concerned that this phenomenon can be eliminated by correct design". Mr. Neville apparently refers to the Heliston propeller, which has been brought prominently before the engineering world in recent years. It is claimed that this type of screw is so designed that erosion is obviated. Since erosion is caused by the collapse of cavities, it follows that the design gets over cavitation troubles and incidentally "singing" propensities. It may be accepted that in none of the hundred cases to which Mr. Neville refers has singing occurred, but it might be asked whether singing would not take place in some particular case. Since there are many shipbuilding firms which have had no instances of singing screws in their long experience, the exceptional case may arise.

Dr. Baker, in his remarks, points out that information connected with this subject had been collected for the past three years by the William Froude Laboratory at the request of the Shipbuilding Employers' Federation, but, unfortunately, that body has not seen fit to publish the report founded on the data accumulated. In Dr. Baker's opinion, the disturbing noises arise from the propeller, are transmitted to the water, and thence to the ship's plating. The ship's plating, as a rule, sets up the vibration near the propeller, but in at least two ships the noise has been transmitted to compartments well forward in the vessels, the intervening structure being free of any disturbance.

In connection with this theory that the ship's plating may have something to do with the noise set up by a "singing" propeller, it is interesting to note that the case cited by Mr. David Dickie, of San Francisco, refers to that of a diesel fishing boat, which was built of wood. In this particular case there could be no "hull resonance", so that it appears that "singing" is a peculiarity of the screw only although its intensity may be governed by the nature of the structure in its vicinity.

Space does not allow of reference being made to many interesting contributions to the discussion which were made by various authorities on ship propulsion. An enormous amount of data, conflicting in its nature, is given by the author and the various contributors. It may lead to some rational conclusions being drawn if all the evidence from all available sources were carefully assembled and analysed. The appearance of singing screws has been more frequent than was generally known, although, fortunately, the percentage of such, in relation to the total number of screw vessels in service, is remarkably low. The large number of eminent men who have contributed to the discussion of Mr. Hunter's paper is an indication of the widespread interest which has been taken in his able and informative paper.

Pressure Drop in Condensers.

"Shipbuilding and Shipping Record", May 13th, 1937.

The condenser plays a very important part in the economy of the steamship, but it is questionable whether the merit of its performance is assessed with any degree of accuracy in relation to that of the steam generator and the engine. In a paper contributed to the "Journal of the American Society of Naval Engineers", Mr. John R. Weske, who was for five years in the design department of the Bethlehem Shipbuilding Corporation, points out that the weight of the condenser of a naval steam plant is approximately one-quarter of the weight of the entire main propulsion unit, and that as the initial pressure is increased, the relative weight and size of the condenser tend to become larger. These facts serve to emphasise the importance of achieving greater compactness in the design of the condenser, although this can only be done at the cost of increasing the pressure drop which, in turn, has a considerable effect upon the efficiency of the engine. With large units of high capacity the problems of determining the pressure drop becomes one of considerable importance to the designer, and Mr. Weske proceeds to develop a mathematical formula which enables such a calculation to be made. In his proof, a certain number of assumptions have necessarily to be made, some of which are open to question, but it is worthy of note that curves, one obtained mathematically and the other obtained from a special test, show a remarkable degree of similarity.

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 6th May, 1937:—		
Haines, James W. ...	1.C.	Newcastle
Mills, Robert ...	1.C.M.	"
Foster, Ronald N. ...	1.C.	London
Nicholson, Kenneth L. ...	1.C.	"
Darroch, Robert A....	1.C.	Glasgow
Milton, Robert ...	1.C.	"
Walker, John L. ...	1.C.	"
MacKay, William G. ...	1.C.M.	"
Gardner, Norman ...	1.C.	Liverpool
Milligan, Andrew B. ...	1.C.	"
Shephard, Geoffrey ...	1.C.	"
Jones, Francis ...	1.C.M.	"
Tughan, Basil W. ...	1.C.M.	"
Mathews, Charlie G. ...	1.C.M.E.	"
Carson, Henry ...	1.C.M.E.	Belfast
Doherty, William B. ...	1.C.S.E.	"
MacKay, Alexander ...	1.C.S.E.	Glasgow
Stirling, Robert ...	1.C.S.E.	Glasgow
Burdon, William H. ...	1.C.S.E.	Newcastle
Banks, Peter ...	1.C.M.E.	Glasgow
Kelly, James M. ...	1.C.M.E.	"
Moffat, Thomas A. ...	1.C.M.E.	"
Park, John C. ...	1.C.M.E.	"
Burton, James C. ...	1.C.M.E.	London
Groves, Harry G. ...	1.C.M.E.	"
Jessop, Claude S. ...	1.C.M.E.	"
Smith, Andrew C. ...	1.C.M.E.	Newcastle
Groves, Ronald G. ...	1.C.M.E.	"
Brown, Wilfred ...	1.C.M.E.	"
Brown, Robert A. ...	1.C.M.E.	"
Woodbridge, Clarence W.	1.C.M.E.	"
For week ended 13th May, 1937:—		
Gardner, Joseph A. ...	2.C.	Glasgow
Kidd, Arthur E. ...	2.C.	"
Learmonth, William F. ...	2.C.	"
Leishman, Henry ...	2.C.	"
Nicoll, James Y. ...	2.C.	"
Routledge, Thomas A. ...	2.C.	"
Turner, Hugh R. ...	2.C.	"
Geddes, James D. ...	2.C.M.	"
Johnstone, John J. R. ...	2.C.M.	"
McMichael, Hunter T. ...	2.C.M.	"
El Mougy, Ahmed F. M. H.	2.C.	Liverpool
Maudsley, Herbert T. ...	2.C.M.	"
Matthews, Edward J. ...	2.C.	London
Phillips, James B. ...	2.C.	"
Webster, David ...	2.C.	London
Smith, Edward G. ...	2.C.M.	"
Driver, Donald P. ...	2.C.	Newcastle
Hedley, Ernest ...	2.C.	"
Kemp, Arthur W. ...	2.C.	"
Stobbs, Arnold H. ...	2.C.	"
Tate, William ...	2.C.	"
Wears, John D. ...	2.C.	"
Turner, George W. ...	2.C.M.	"
Wallace, James ...	2.C.M.	"
West, John E. ...	2.C.M.	"
White, Malcolm J. ...	2.C.M.	"
Williams, Lawrence T. ...	2.C.M.	"
Wood, Peter ...	2.C.M.	"
Wright, Leonard H. W. ...	2.C.M.	"
For week ended 20th May, 1937:—		
Fleming, Wilson T. ...	1.C.M.E.	Cardiff
Volke, Eric C. J. ...	1.C.S.E.	London
Thomas, Clifford N. ...	1.C.M.E.	Newcastle
Stewart, James ...	1.C.M.E.	"
Raeside, Charles ...	1.C.M.E.	Glasgow
Pollard, Harry ...	1.C.M.E.	Cardiff
Burgess, Thomas S. ...	1.C.M.E.	Newcastle
Nelson, Alfred E. ...	1.C.M.E.	"
Sterling, Sidney W. ...	1.C.M.E.	"
Manser, Leonard ...	1.C.M.E.	Liverpool
Bowlerwell, Walter...	1.C.M.E.	"
Barber, Eric J. ...	1.C.M.E.	London
Elfert, William E. ...	1.C.M.E.	Liverpool
Sage, Arthur L. ...	1.C.	London
Hesketh, Herbert F. ...	1.C.	Liverpool
Spence, Arthur R. ...	1.C.	"
Edwards, William J. ...	1.C.M.	"
Liddle, Kenneth A. B. ...	1.C.	Glasgow
Day, Richard	1.C.	Cardiff
Hooper, Norman E. ...	1.C.M.	"
Bradley, Robert ...	1.C.	Newcastle
Jackson, Albert H. ...	1.C.	"
Marshall, Hugh ...	1.C.	"
Thomson, David B. ...	1.C.	"
Watson, James N. ...	1.C.	"
Brooks, Leslie ...	1.C.M.	"
Johnson, Percival R. ...	1.C.M.	"
Wandless, Joseph A. ...	1.C.M.	"