The INSTITUTE of MARINE ENGINEERS
Transactions

1949, Vol. LXI, No. 11

PRESIDENTIAL ADDRESS

Rt. Hon. LORD ROTHERWICK

First I should like to say how much I appreciate the honour you have paid me in asking me to become your President—a position that has been occupied by so many distinguished people in the past.

About the time when I first began to be interested in ships, the propelling unit almost universally adopted was the triple expansion engine supplied with steam from Scotch boilers, and working at a pressure of about 160lb. per sq. in. There were very few gadgets and not much auxiliary machinery, sometimes not even a dynamo. The power developed was generally governed by the firemen and the quality of the coal.

By present day standards, those engines would be regarded as very uneconomical, but they jogged along, turning at a comfortable 65 r.p.m. or thereabouts.

Nobody worried very much about high efficiency, but I understand that now and again, and at very infrequent intervals, a great event took place in the engine-room.

The Chief Engineer had decided to take indicator diagrams. It was expected (I had almost said arranged) that the results would show a figure of about 1·5lb. of coal per i.h.p. per hour. While the diagrams were being taken, it was quite likely that the intermediate stop valve would be opened a little more, and as a result, the diagrams when worked out, would show a power somewhat higher than the average for the trip. The main thing was to show a figure somewhere in the region of 1·5lb. of coal per i.h.p. per hour, which the head office expected.

I have even heard it said that some of the indicator diagrams had never been in close contact with an indicator, but so long as they were of good shape, everyone was satisfied. Certainly there was less attention paid to the scientific side of marine engine operations in those days, and I feel quite sure that the word "vacuum" was less understood than it is now.

At any rate, there is a true story in connexion with one of our own chief engineers who has now departed from us.

It was during the time when we were fitting many of our vessels with the Bauer-Wach exhaust turbine. You will, of course, understand better than I the importance of a high vacuum when an exhaust turbine is fitted. In this particular case it was noticed that the chief engineer was recording on his abstracts a figure for the vacuum in the condenser higher than the barometer. When he was questioned about this, he stated that this was the figure shown on the gauge and he saw no reason, if the condenser was perfectly tight and no leakage anywhere, why he should not obtain a figure higher than the barometer. I think we have made some progress since then, as I do not think any of our seagoing engineers today would hold an opinion such as this. There is no doubt that great strides in economy have been made since those days, but I often wonder whether as much progress has been made in reliability. In my view, reliability is of paramount importance.

We have only to consider the enormous present-day cost of building a vessel, compared with even pre-war days, and this represents a very heavy annual depreciation figure, and consequently the daily cost of operating a vessel is correspondingly high.

Any delays, therefore, incurred through repairs and upkeep, apart from the present high cost of such repairs, is much more serious than in days gone by. This is a major reason why I would lay stress on reliability.

When we speak of economy, we may mean economy in fuel, economy in weight and space or economy in first cost. All these are naturally very attractive to the shipowner, and I would be the last to attempt to put a brake on progress in any direction, provided that reliability is always kept in the foreground.

With regard to economy in fuel in steam engines (the shipowners' ambition), the normal avenue of approach has been to widen the gap between the initial and final pressures and temperatures.

The introduction of the exhaust turbine to the reciprocating steam engine about twenty years ago, gave an economy in fuel of nearly 25 per cent by making use of the steam at the lower end of the scale beyond the limts of the reciprocating engine. At the upper end, pressures and temperatures were gradually increased until at the present time, pressures of 650lb. per sq. in., and steam temperatures of 850 deg. F. and even beyond, are by no means rare. These are outside the economic limits of the normal reciprocating steam engine, but are within the scope of the steam turbine.

I recollect that when we introduced superheated steam in our reciprocating engines, we found that the economic limit of temperature was about 580 deg. F. Beyond this, we found that any thermal gain which we hoped for, by increasing the superheat temperature, was more than counterbalanced by wear on the cylinders and piston rings.

We also found that there was very little gain, if any, to be obtained in economy by supplying auxiliary machinery with superheated steam. Here again, the more rapid wear on the internal parts more than counter-balanced the gain which could be expected with superheated steam, and in addition there was more tendency for oil to find its way into the boilers, and as we all know, lubrication has its merits in its proper sphere, but we are heading for trouble if we try to lubricate the inside of the boilers, particularly if these be of the water-tube type.

Turning now to economy in weight and space by increasing the pressure range through which the engine operates, we naturally reduce the size of the engine, and this can be reduced still further by increasing the speed of rotation of the engine, but this necessitates more positive and more thorough lubrication of the moving parts.

High-pressure boilers call for greater purity of feed water and these things, together with the various gadgets, including feed regulators, feed heaters, feed filters, etc., call for more alertness and skill on the part of the engineers in charge.

Economy in first cost, whether by the employment of less expensive materials or by a higher rating of the power unit, generally means that the machinery is less reliable for long periods of service. If the machinery, whether steam or Diesel, is called upon to operate constantly at, or near, its maximum capacity, there will surely be more likelihood of a breakdown than if it were operated under easier conditions. If we decide to operate the machinery something below its rated power, the first cost would be greater, but in my view, the extra reliability more than justifies the extra cost.

The question of how much below the maximum capacity machinery should be run can only be decided by each one for himself, the best guide being experience. I suppose almost everything in engineering is a compromise and at the one extreme we have the heavy low-rated, extremely reliable but uneconomical engine, and at the other end, we have the high-rated, fast-running, highly efficient and economical engine.

I would only repeat that I regard reliability as a most important factor in deciding the size and type of machinery to propel a vessel.

There is a tendency at the present time to increase the speed of rotation of auxiliary machinery, and while there may be no objection to this in the case of turbines, I feel that we may lose some reliability through too high a speed if the engine is of a reciprocating type.

There is also a tendency nowadays towards electrically operated auxiliaries, and I think this is a step in the right direction, both from the point of view of reliability and of low operating costs. The latter applies particularly when the vessel is in port. Unfortunately, in many ports abroad, there are now very serious delays and vessels are immobilized for long periods waiting their turn to discharge cargoes.

In pre-war days, it was customary for a cargo liner to spend about two-thirds of a year at sea and a third in port; we can practically say now that the reverse holds good. For this reason, economical operation of machinery in port is more important than in pre-war days, and although the first cost of electrical auxiliaries is considerably greater than the steam equivalent, the fuel cost of operating electrical auxiliaries shows a considerable economy, taken over the life of a vessel, and this is particularly noticeable in port.

The controversy between steam and Diesel still goes on. There are advocates for the steam engine who have not a single good word to say for the Diesel; and on the other hand, the Diesel supporters are firmly convinced that the internal combustion engine is gradually displacing the steam engine. It was at one time thought that the Diesel engine would almost entirely displace the steam engine, owing to the low fuel consumption of the Diesel compared to the steam engine. Considerable improvement in efficiency has, however, been made in the steam engine since the advent of the Diesel, and it still remains a serious competitor to the latter.

For instance, in one class of vessel we built before the war, the average speed was 15 knots on a consumption of 60 tons of oil per day. In the case of a vessel of about the same tonnage, recently completed, we anticipate a speed of 16½ knots on the same consumption, namely 60 tons per day.

The different types of Diesel engines now available are almost confusing to the non-technical mind. We started off with the simple single-acting 4-stroke type, which, although somewhat heavy for the power developed, was more reliable than some of the later types. Then came the single-acting 2-stroke and the double-acting 4-stroke, and later the double-acting 2-stroke. I can recollect some experience with the double-acting 4-stroke type and the repair bills were certainly heavy. There is no doubt that if we want reliability, which I consider is of the greatest importance, we must not rate the power of our Diesels too high.

We have recently placed into service sister vessels, some of which are driven by steam turbines, and others by Diesel engines. It is much too early yet to attempt to make a definite comparison, but there should be some interesting information in due course.

Generally speaking, however, the competition is less keen in the higher powers. As the power increases, the pendulum swings more in favour of the turbine. At the other end of the scale, our old well-tried friend, the simple triple expansion engine is still a keen competitor. It may be that difficulty in obtaining engineers is a factor, as whatever type of engine we adopt to propel our vessels, we still require engineer officers to operate the machinery.

Where the horse-power is low, the difference in fuel rate between the modern high efficiency plant and the simple, but more inefficient type of engine is less marked than at the higher powers. For the higher powers obviously the efficiency factor becomes of primary importance.

I am told that it is very difficult, at the present time, to obtain experienced engineers, particularly in the higher grades, and I think the reason for this is fairly clear. There is undoubtedly a heavy demand ashore at present for experienced engineers and the attractions of a fuller home life may often outweigh the attractions of a sea-going career, and the question arises as to what are we doing to offset this tendency, as it is obvious that we must have engineer officers to operate the machinery of our vessels.

There is no doubt that accommodation has vastly improved compared to the standard thirty to forty years ago. Officers' cabins are now larger and better furnished and it is normal practice today for each engineer officer to have his own room. Officers' smoke rooms are provided and better means of ventilation. One of the most important factors is, of course, the rate of pay, but I feel that with the improvements in basic rates and a more generous allowance of leave at the end of the voyage, plus a good pension scheme, a seagoing career can now be considered very attractive.

We have also tried to improve the working conditions on board our vessels. I am a strong believer in providing good ventilation in the engine-room, also good lighting, and I do not believe in having the engine-room too cramped. There is a tendency to cut down the engine-room as much as possible in some quarters, so as to provide more space for cargo. This is quite understandable but it is possible for this to be overdone. My feeling is that an engineer will be more alert and efficient, and more content if his working hours are spent in a well-ventilated and well-lit engine-room, and where he has some floor space to carry out any overhauling which may be necessary.

Although I know that you are chiefly concerned with the machinery in the engine-room, I should like to say a few words about the holds and cargo equipment, for after all, cargo liners are built to transport cargo and a shipowner might perhaps be forgiven if he were inclined to look upon the engine-room as a necessary, but non-earning part of the equipment. A general cargo carrier has, of course, to deal with a variety of cargoes as distinct from ships which are specially equipped for the transport of particular cargoes, such as bulk oil, grain and refrigerated produce.

One class of cargo which a general carrier is called upon to deal with, may almost be considered as special cargo, as it involves special equipment. I refer to what are known as heavy lifts, and it would appear that the manufacture of some heavy pieces of equipment for transport abroad is only limited in weight by the facilities the shipowner can offer in the way of lifting gear.

It does not seem very many years ago since a 30-ton lift was considered as a really heavy piece of cargo, but lifting gear has increased in size until we can now take on board, and discharge with the ship's own gear, single lifts up to about 125 tons in weight. It will be clearly understood that the transport of heavy pieces of cargo does not only involve the actual lifting of the weight itself, but there is also the problem of stability in many cases.

More attention is paid in modern vessels to the illumination of cargo spaces and I can speak from experience that a well-lighted hold for working cargo during the hours of darkness, has proved to be of great benefit.

A matter which is causing me great concern is the present cost of building vessels. During the past few years, the cost of building tonnage has continued to show a progressively upward trend. From experience I am able to quote rises in building costs for sister vessels. For a vessel delivered in 1947, the increased cost over a similar vessel delivered in 1946 amounted to 6 per cent, apparently a small percentage, but which amounts to a considerable addition to the total cost. For a third sister vessel delivered in 1948, the increase in the building cost over the 1946 vessel amounted to no less than 20 per cent. We find today that the cost of building is three times the cost of a similar vessel in 1938.

These increases do not take into account the increased delays in the completion of vessels when considerable sums in course have been expended which show no return to the shipowner. These delays are increased by late delivery of machinery and fittings of various kinds.

It is difficult for shipowners to visualize the position when cargoes are not so plentiful and freight rates are reduced in the future, and how they will be able to cover the necessary depreciation on these inflated values, apart altogether from the question of earning any return on the capital invested by them. I think that you will agree that owners will require to have considerable faith to continue to build vessels at present costs, and that a halt must be called sooner or later.

Hard necessity will ultimately drive all those concerned in the building of tonnage to bring down the costs somehow to a figure which will induce owners to undertake the construction of new tonnage.

Shipowners will follow with great interest the development of the gas turbine, as their aim is naturally to obtain increased speed, and at the same time reduce operating costs, particularly fuel consumption. In this respect, we hope the gas turbine will carry economy, with greater speed, a stage further before many years have passed, but here again, one of the most important factors is reliability.

The Heart of a Tanker*

H. NICOL (Graduate)

A young engineer beginning his career in marine engineering is at a great disadvantage, considering that most of the present day technical papers are written by highly skilled engineers, and in terms with which a boy is not familiar in the early days of apprenticeship. It is the aim of this essay to outline clearly, simply and briefly the propelling machinery required for a typical motor vessel.

More than ever before, there is today a great demand for oil, both for machinery lubrication and power purposes. On account of Britain's oil reserves having fallen to a critically low level, due to the enormous demands made during the years of war, and not having any workable home sources she has become solely dependent on other countries for her oil supply. As a consequence oil tankers are required to convey the crude oil in bulk from the oil fields to the refineries at home. To meet this demand for crude oil and to replace war losses an extensive expansion of the existing oil tanker fleet is an imperative necessity.

It would be reasonable to suggest, that of all the oil tankers being built at present, the most popular is the 12,000-ton class. With oil tankers more than any other type of vessel, it is desirable to have as much cargo space as possible (see Fig. 1) with the result that they are more generally powered

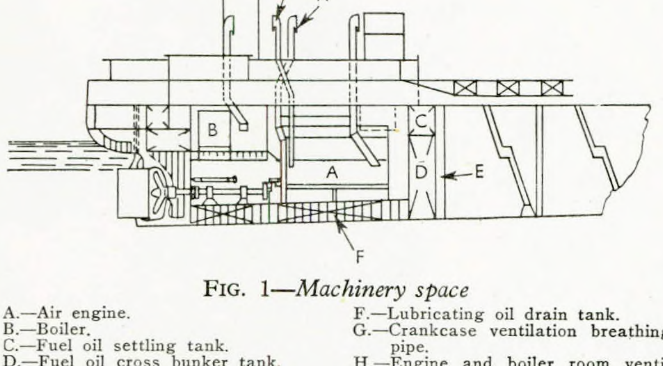

FIG. 1—*Machinery space*

A.—Air engine.
B.—Boiler.
C.—Fuel oil settling tank.
D.—Fuel oil cross bunker tank.
E.—Coffer dam.
F.—Lubricating oil drain tank.
G.—Crankcase ventilation breathing pipe.
H.—Engine and boiler room ventilators.

by Diesel machinery, thus giving a shorter machinery space in comparison with other types of propulsion, such as steam turbine and steam reciprocating engine installations. Another salient feature of the motor tanker is that it has no standby losses in port when the turn about is only forty-eight hours; this is not the case with the steamer, considering that the steam supply cannot be allowed to go down for such a short term of rest.

The propelling machinery for these 12,000-ton tankers, if being built for British owners, is often built and installed to the requirements and survey of Lloyd's Register of Shipping or British Corporation (now incorporated in Lloyd's) and if built for foreign companies to their selected and named authorities.

*This essay won the combined Jacobs, Murdoch and Robertson Award for 1948.

It would be appropriate at this stage, before carrying on further with the propelling machinery, to give a brief description of the initial proceedings between shipping company and ship and engine builders. The first thought of building a ship naturally originates with the oil tanker company, who notify the ship and engine builders of their desire. The ship and engine builders then submit tenders to the oil tanker company for their consideration and possible acceptance. The order having been placed the contract specifications are drawn up and a conference is arranged between the owners, and the ship and engine builders' representatives. At this meeting the contract is discussed, and notes of proceedings made of any additional requirements of the purchasers, together with the extra costs involved, the keel laying, launching and commissioning dates, and any other points which may require agreement.

After this the working specifications are printed, and issued to the various departments directly concerned. The drawing office prepares the necessary drawings, adhering to the requirements stipulated in the specification, the other departments, e.g., the ordering department arranges for the buying in of any accessories, which may have been requested by the owners, and attends to the complicated proceedure of issuing the drawings and ordering from the manufacturing departments, all the parts detailed thereon. The estimating department, accounts department, electrical department, engine works, etc., etc., and owners and shipbuilders each receive copies and if any changes take place in the original order the specifications issued to the departments enumerated above are returned to the machinery specification engineer and are amended in accordance with his latest records. After being brought up to date, they are re-issued to their respective departments.

The selected ship, a 12,000-ton tanker, is required to operate under average service conditions at a mean speed of about 11·5 knots under fully loaded conditions. An allowance of 20 per cent power is made above the power that is required for fair weather and clean bottom conditions. The service power output and engine speed of a Diesel engine fulfilling these requirements is not less than 3,000 b.h.p. and generally not more than 3,500 b.h.p. at within 110 to 120 r.p.m. This power would be obtained with an indicated mean effective pressure, within the limits of 88 to 115lb. per sq. in. This vessel is of the single-screw type, propelled by a four-stroke Diesel engine totally enclosed, forced lubricated, direct reversing, arranged with airless injection of fuel and designed for under-piston pressure induction. The engine is coupled direct to the propeller shafting, the rotation for ahead motion being suitable for a right-handed propeller. The cylinder liners, covers and exhaust valves are fresh or distilled water cooled and the pistons are oil cooled from the forced lubrication system.

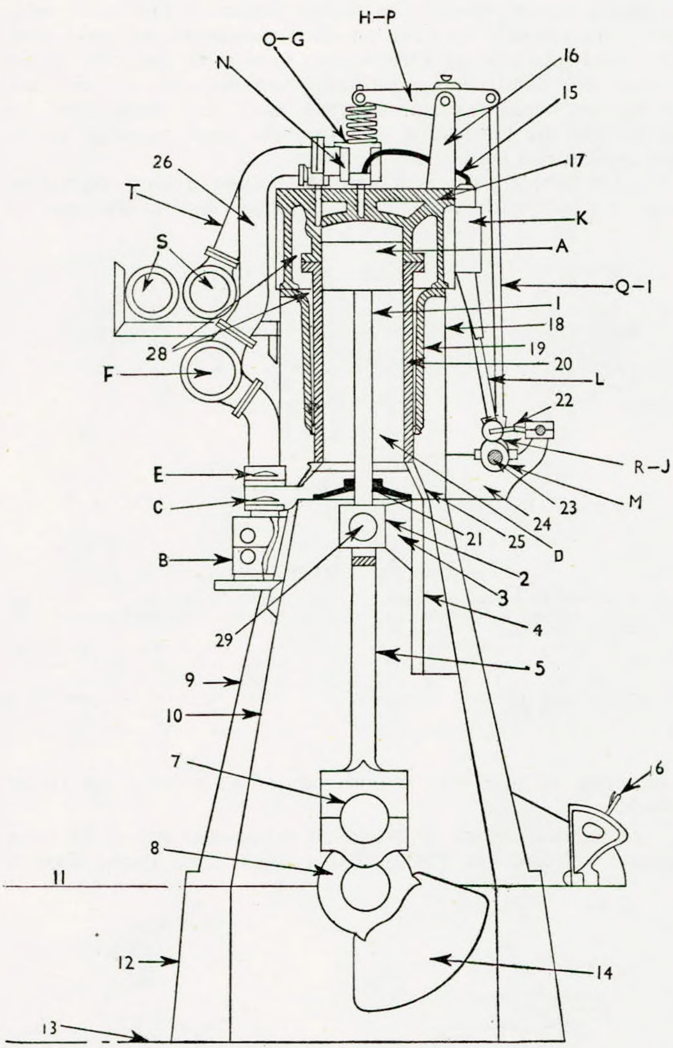

FIG. 2—*Section of four-stroke, single-acting engine with underpiston supercharge*

A.—Piston.
B.—Air inlet trunk.
C.—Two suction valves.
D.—Supercharging chamber.
E.—Two delivery valves.
F.—Supercharged air manifold.
G.—Inlet valve. This valve is not shown, but is similar to the exhaust valve and is similarly placed only on the forward side of the fuel valve.
H.—Inlet valve lever, same as exhaust valve lever but placed forward of it.
I.—Inlet valve push rod, similar to exhaust valve push rod but placed forward of it.
J.—Inlet valve cam, similar to exhaust valve cam but placed forward of it.
K.—Fuel pump.
L.—Fuel pump push rod.
M.—Fuel valve cam.
N.—Fuel valve.
O.—Exhaust valve.
P.—Exhaust valve lever.
Q.—Exhaust valve push rod.
R.—Exhaust valve cam.
S.—Exhaust gas manifolds.
T.—Exhaust gas trunk.

1.—Piston rod.
2.—Crosshead.
3.—Crosshead shoe.
4.—Crosshead guides.
5.—Connecting rod.
6.—Starting handle.
7.—Crankpin.
8.—Crank web.
9.—Engine frame.
10.—Crankcase doors fitted between engine frames.
11.—Floor line.
12.—Engine bedplate.
13.—Tank top.
14.—Balance weight.
15.—Fuel supply pipe led from fuel pump to fuel injection valve.
16.—Valve lever bracket.
17.—Cylinder cover.
18.—Distance piece.
19.—Cylinder jacket.
20.—Cylinder liner.
21.—Piston rod scraper box.
22.—Manœuvring crankshaft. (Regulating shaft.)
23.—Camshaft.
24.—Camshaft bracket.
25.—Segmental casing.
26.—Supercharged air trunk.
28.—Cooling water space.
29.—Gudgeon pin.

The principle on which the engine works is as follows. When the piston A (see Fig. 2) is on the upward stroke, a partial vacuum is created underneath it, inducing free air to flow from the atmosphere through an inlet trunk B and two suction valves C, into the cylinder space D below it. The air is compressed on the following downward stroke of the piston and is forced through two slightly loaded delivery valves E into the supercharging manifold F and then to the inlet valve G.

The inlet valve is opened by the lever H and push rod I actuated by the cam J, thus admitting the air into the cylinder D on the top side of the piston when it is on the downward stroke. Shortly after the bottom dead point the inlet valve is closed and on the upward stroke the air is compressed to high pressure with incandescent temperature. Fuel is then injected under pressure, through the fuel valve N, by the fuel pump K which is operated by the push rod L and cam M, slightly before the piston reaches top dead centre and combustion takes place during the next downward stroke. This is the working stroke as all the energy of the expanding gas is converted into mechanical work. When the piston approaches bottom dead centre the exhaust valve O is opened by the lever P and push rod Q actuated by the cam R so that, on the following upward stroke of the piston, the products of combustion are expelled from the cylinder in succession through the exhaust valve, exhaust trunk T, exhaust manifold S and then through a silencer into the atmosphere.

Thus the piston traverses the cylinder four times to complete one cycle of operation.

The pumps required for main engine services, namely, sea water circulating, fuel supercharging, fresh or distilled water cooling and forced lubrication, are driven by the main propelling unit and are either of the vertical reciprocating or rotary screw displacement type. Steam-driven stand-by pumps for all main engine services are provided for manœuvring purposes. All stand-by pumps are of sufficient capacity to maintain operation of the main engine at normal service power output in the event of breakdown of any of the main engine driven pumps. The engine is started by means of compressed air at a maximum pressure of about 350lb. per sq. in. taken from air reservoirs situated in the engine room.

The exhaust gases from the main engine are generally utilized in boilers which thus provide power for the steering gear and, as far as possible, for all other steam driven units in normal operation at sea. The exhaust manifold is efficiently lagged, but if exhaust gases are not utilized for boilers, water cooling may be adopted as an alternative.

Auxiliary boilers of multi-tubular marine or water-tube type are fitted, each arranged for being fired by oil fuel and exhaust gas together and independently. When using exhaust gas only, the boilers are capable of supplying sufficient steam for steering gear and all steam-engine driven auxiliaries in normal operation at sea.

Two steam-driven electric generators, or one Diesel-driven and one steam-driven electric generator, are generally installed, each capable of supplying the power for electric lighting, fans, motor-driven auxiliaries, wireless machinery and heating.

The usual shop and mooring (or basin) trials are carried out. During the shop trials the b.h.p. and i.h.p. are measured and the mechanial efficiencies calculated therefrom are used in determining the b.h.p. from the indicator diagrams taken when the engine is installed in the ship.

The full power shop trial, usually of six hours' duration, or as may be required by the owner's representative, is carried out with the engine operating at the various power outputs specified by the owner's in their contract agreement.

The mean indicated pressure is obtained during the shop trials while the engine is coupled to a hydraulic brake. The specific fuel consumption per b.h.p.-hr. and per i.h.p.-hr. is calculated over a range of speeds and powers corresponding approximately to propeller conditions. The owner's acceptance trials over the measured mile include a double run to record the speed of the vessel when the engine is developing its service output, and other speed requirements agreed upon. One pair of runs over the measured mile during the acceptance trials are made at 50, 75, 100 and 110 per cent of service power. The mean indicated pressure during the trials at sea is obtained and checked by calculation from the fuel consumption.

During the acceptance trials the engine manœuvring gear is tested and the engine is required to reverse from full speed ahead to full speed astern. The capacities of the manœuvring air storage reservoirs are tested to ensure that there is sufficient air for at least twelve starts in ahead or astern directions in accordance with survey requirements.

The engine is required to be capable of running dead slow regularly at about 40 to 50 r.p.m. at sea without stopping and with all cylinders firing. All trials of the main engine are usually carried out on standard marine Diesel fuel On completion of satisfactory sea trials the vessel is handed over to the owners.

Particulars of the principal parts of the engine will now be given, commencing with the bedplate and following as close as possible the assembly of the various engine parts.

The bedplate is made of good quality steel (see Fig. 3) of

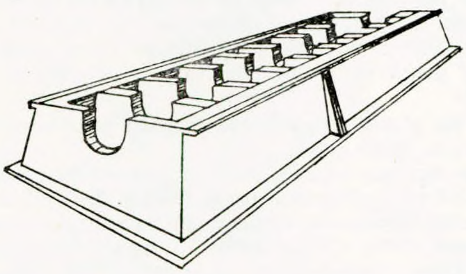

FIG. 3—*Bedplate*

fabricated construction, or it may be of cast iron according to circumstances of production, etc. The thrust block is incorporated in the bedplate with provision for bolting to strong seatings secured to the double bottom of the ship. All joints and bearings surfaces are accurately machined and fitted metal to metal; flanges of the bedplate are machined smooth on the under-side with a slight taper towards the centre of the engine, i.e., flanges increasing in thickness from the outside, to permit the easy fitting of chocks. The cross-members of the bedplate are arranged to take the main bearing bushes, which are of circular section, made of cast steel, and tinned and lined with white metal. The bores of the main bearing bushes are machined slightly eccentric to the outer part of the cast steel shell so that the bushes can be easily removed while the crankshaft is in position. The top halves of the main bearing bushes are secured by keeps made from either welded steel plate, cast steel or cast iron. All bearings below cylinder tops are forced lubricated and an oil-tight sump is incorporated in the bedplate for the purpose of collecting the used lubricating oil. From the engine sump the oil is drained by gravity to a lubricating oil drain tank situated below the engine bedplate.

The crankshaft is of the fully built type (see Fig. 4) made in two sections coupled together by solid flanges and fitted bolts. The cranks are arranged with balance weights to ensure minimum vibration, to eliminate unbalanced forces and couples, and

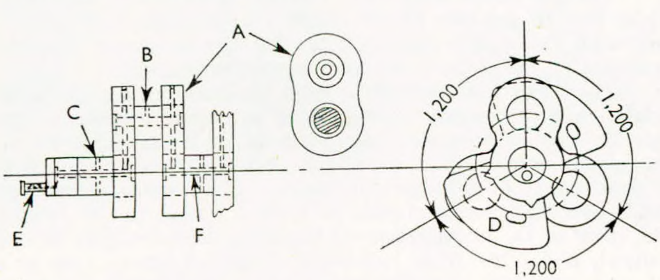

FIG. 4—*Crankshaft*

A.—Crank web without balance weights.
B.—Crank pin.
C.—Journal.
D.—Balance weight.
E.—Crank of lubricating oil and circulating water pumps. (Reciprocating type.)
F.—Lubricating oil holes.

to give a turning moment as even as possible. The crank webs, pins and journals are made of good quality forged mild steel, pins and journals are shrunk into webs with one heat of the webs. No dowel pins are fitted, shrinkage only is relied on. Holes are bored through journals, webs and crank pins for conveying the lubricating oil from the main bearings to the connecting rod bottom end bearings.

The thrust shaft is forged of the best quality ingot steel (see Fig. 5) with solid half couplings and collars and is

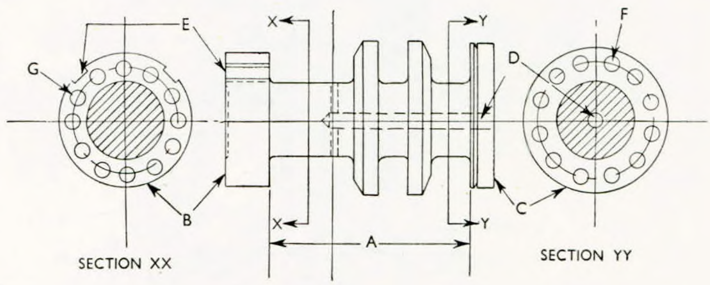

FIG. 5—*Thrust shaft*

A.—Machined to a smooth finish.
B.—Half coupling for joining up to tunnel shafting; arranged to support the turning wheel.
C.—Half coupling for joining up to crankshaft.
D.—Lubricating oil hole conveying oil to outer bearing.
E.—Keys for securing turning wheel to half coupling are fitted into keyways machined parallel with shaft.
F.—Holes for fitted bolts are rough bored and afterwards finished reamered with crankshaft.
G.—Holes for fitted bolts are rough bored and afterwards finished reamered with tunnel shafting

machined all over and smooth turned in way of the thrust block.

The thrust block, incorporated in the after end of the main engine bedplate (see Fig. 6) has a single fixed thrust shoe of

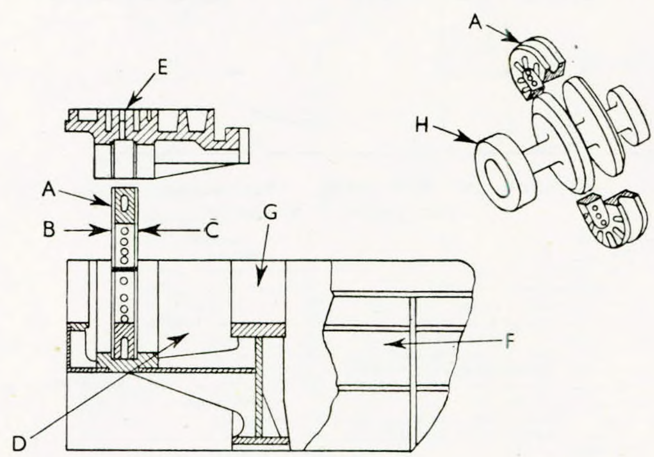

FIG. 6—*The thrust block incorporated in the engine bedplate*

A.—Thrust shoe (in halves).
B.—Bearing surface taking ahead thrust.
C.—Bearing surface taking astern thrust.
D.—Thrust block.
E.—Thrust block cover.
F.—Bedplate. (Fabricated.)
G.—Main bearing housing.
H.—Thrust shaft.

ample dimensions and capable of taking the maximum thrust. The thrust shoe is lined with white metal, on both ahead and astern faces, suitably grooved to ensure efficient lubrication and cooling of the working faces. The thrust block is lubricated and cooled from the main engine forced lubrication system and has gunmetal gland baffles fitted to prevent oil leakage at the shaft. Suitable thermometers are fitted to indicate the temperature of the oil entering and leaving the block.

The turning wheel, which serves as a light flywheel, is made of cast iron, in halves and is fitted on the thrust shaft (see Fig. 7) and has teeth cast in the rim for engaging with the

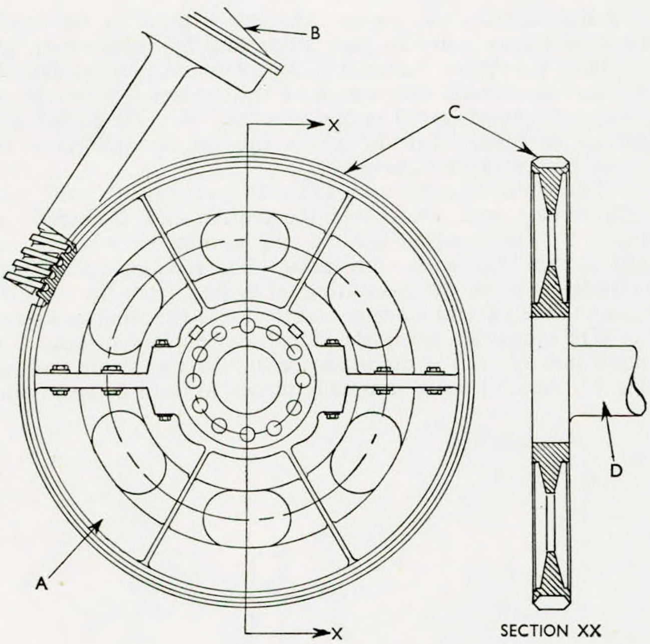

FIG. 7—*Turning wheel*

A.—Turning wheel (in halves).
B.—Turning gear.
C.—Teeth cast on wheel rim.
D.—Thrust shaft.

turning gear. It is marked in degrees on the side of the rim, with top and bottom positions of all cranks specially indicated. A permanent pointer is fitted on the engine structure to register the angular movement of the wheel. A suitable guard is fitted over the turning wheel for safety purposes.

The turning gear engages with the teeth on the turning wheel. It is arranged so that the engine can be turned by an electric motor in either direction one complete revolution in ten minutes; hand power is also arranged. Provision is made for keeping the turning gear securely in and out of engagement with the turning wheel, and usually a warning board marked "Turning Gear In" is hung up near the starting handle to inform the operator when such is the case.

The frames and distance pieces are arranged for supporting the cylinders and are made of either mild steel of fabricated construction or cast iron (See Fig. 8). They are bolted to the

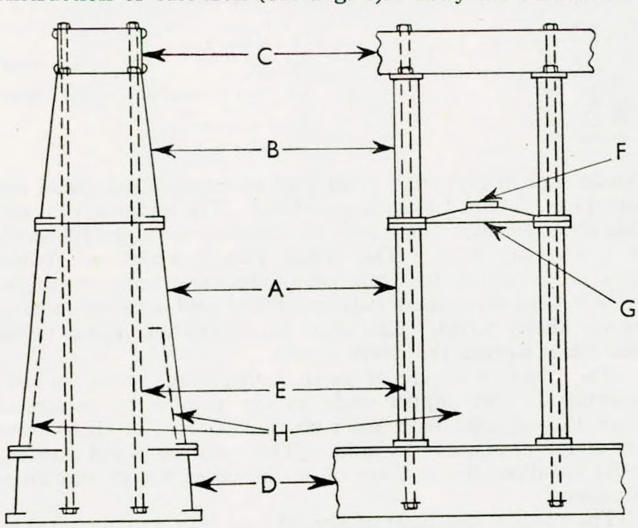

FIG. 8—*Frames and distance pieces*

A.—Frame.
B.—Distance piece.
C.—Cylinder cover.
D.—Bedplate.
E.—Tension bolts.
F.—Piston rod scraper box.
G.—Diaphragm plate.
H.—Inspection doors.

bedplate, and are connected to each other by cast-iron crosshead guide plates. Closed in at the tops by means of diaphragm plates, fitted with piston rod scraper boxes and by large inspection doors front and back, the frames complete a closed-in crankcase.

Crankcase ventilation is effected by means of a suitable breathing pipe (see Fig. 1) led from the top of the crankcase to the boat deck, with goose-neck and wire gauze covering on the boat deck end.

The cylinder liners, made of special hard close-grained cast iron (see Fig. 2), are machined on the outside as well as the inside. The bore of each liner is machined smooth and cylindrical to give a good wearing surface. Each liner is arranged with a heavy flange at the top by which it is securely fastened to the underside of the cylinder cover. The joints at the bottom of the liners, i.e., those made between liners and jackets, are designed to give free longitudinal expansion.

The cylinder jackets are of cast iron and arranged for fresh water, or distilled water cooling (see Fig. 2). The jackets are bolted to the cylinder covers and are made watertight at the lower ends by means of rubber sealing rings fitted into grooves. Access to the cooling space between the cylinder jackets and liners is provided by means of hand holes.

The cylinder covers are supported by distance pieces which are bolted to the top of the frames (see Fig. 9). Covers and

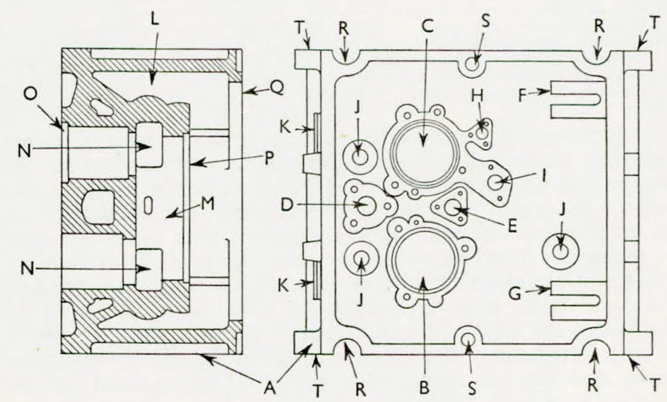

FIG. 9—*Cylinder cover*

A.—Cylinder cover.
B.—Inlet valve pocket.
C.—Exhaust valve pocket.
D.—Starting valve pocket.
E.—Fuel valve pocket.
F.—Facing for exhaust valve lever bracket.
G.—Facing for inlet valve lever bracket.
H.—Cooling water connexion.
I.—Safety valve pocket.
J.—Core hole.
K.—Hand hole to water space.
L.—Cooling water space.
M.—Combustion space.
N.—Pocket arranged to allow for travel of valve spindle head.
O.—Recess to suit removable exhaust valve seat.
P.—Cylinder liner securely fastened to this face.
Q.—Cylinder jacket securely fastened to this face.
R.—Tension bolt hole.
S.—Lifting hole.
T.—Covers fastened together by means of fitted bolts passing through flange.

distance pieces are secured by tension bolts passing through the frames to the underside of the bedplate (see Fig. 8). These bolts take the full reaction of the load on the cylinder head and complete the mechanical circuit through bedplate, crankshaft, connecting rod, piston rod and piston. The covers are made of special close-grained cast iron, of strong square-shaped design, with pockets cast-in for inlet, exhaust, safety, starting air and fuel valves and are arranged for fresh water, or distilled water cooling.

Fresh and distilled water is used for cooling the cylinder liners, covers and exhaust valves; the water is circulated in a closed system by the cooling water pumps. The cooling water pumps discharge through a heat exchanger to the engine cooling system (see Fig. 10) from which the water is returned to the suction side of the pumps. The delivery main (or cylinder cooling water inlet pipe), has a branch fitted with shut-off valve, leading to each cylinder; the outlet pipe from each cylinder has a thermometer fitted. A fresh or distilled water

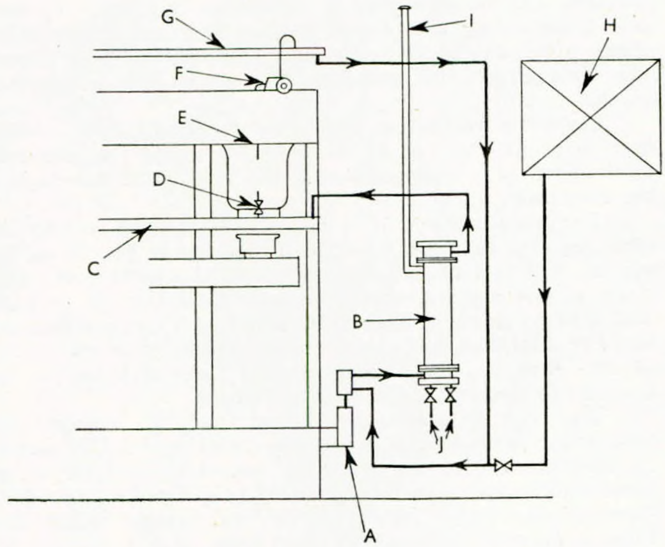

FIG. 10—*Engine cooling water system*

A.—Main engine driven cooling water circulating pump.
B.—Fresh water heat exchanger.
C.—Cooling water inlet pipe.
D.—Branch pipe conveying water to cylinder jacket. (Valve fitted.)
E.—Pipe conveying water from cylinder jacket to cylinder cover.
F.—Pipe conveying water from cylinder cover to exhaust valve.
G.—Cooling water outlet pipe.
H.—Fresh water make-up tank.
I.—Air breathing pipe.
J.—Circulating sea water connexions.

heat exchanger is fitted, with a cooling surface capable of ensuring that under full power conditions the temperature in the system will not exceed 130 deg. F. with sea water at a temperature of 90 deg. F. If two heat exchangers are fitted they are arranged in parallel, thus allowing each or both to be in service as required. The heat exchanger tubes are made of aluminium brass and are arranged so that the cold sea water can circulate through them, thus extracting the heat from the fresh or distilled water of the closed system. Thermometers are fitted to each outlet pipe. An air pipe is led to the top of the engine room and an adjustable relief valve is incorporated in the delivery pipe, with overflow to the supply tank. Steam heating arrangements are provided in the jacket cooling water system, to enable the temperature of the water to be raised to 130 deg. F. before starting the engine. A tank is fitted in the engine room casing to make up any water lost by evaporation, etc. Oil from the forced lubrication system is used for cooling the pistons. A detailed description of this system is given in the paragraph on pistons. The fuel injection valves are cooled with fuel oil to ensure that the nozzle tips are maintained at the lowest possible temperature.

The connecting rods are made of good quality mild steel, with the top end forked and the bottom end T-shaped (see Fig. 11). The crankpin and top end bearings are made of cast steel and are lined with white metal. The rods are bored longitudinally to allow the lubricating oil to pass from the crankpin bushes to the top end bearings and thence to the crosshead shoes.

The crossheads are made of high carbon forged steel. The centre part of each is made square to take the piston rod (see Fig. 12) the ends being accurately turned to form gudgeon pins.

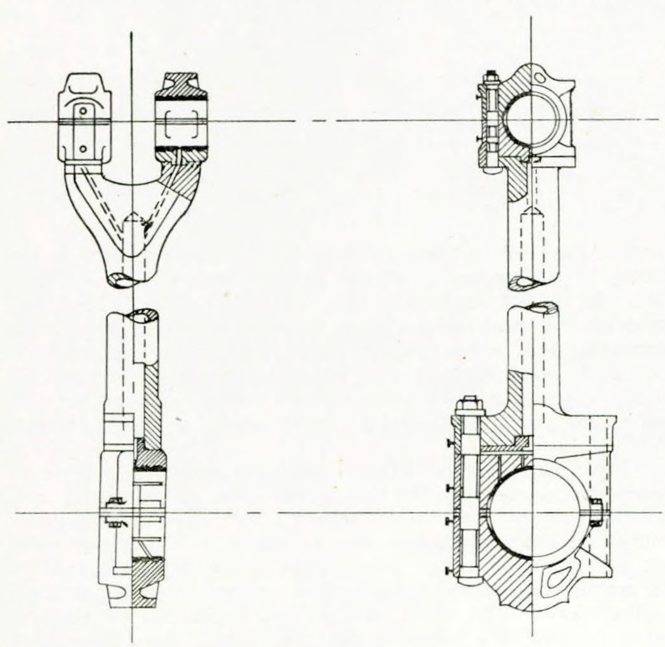

FIG. 11—*Connecting rod*

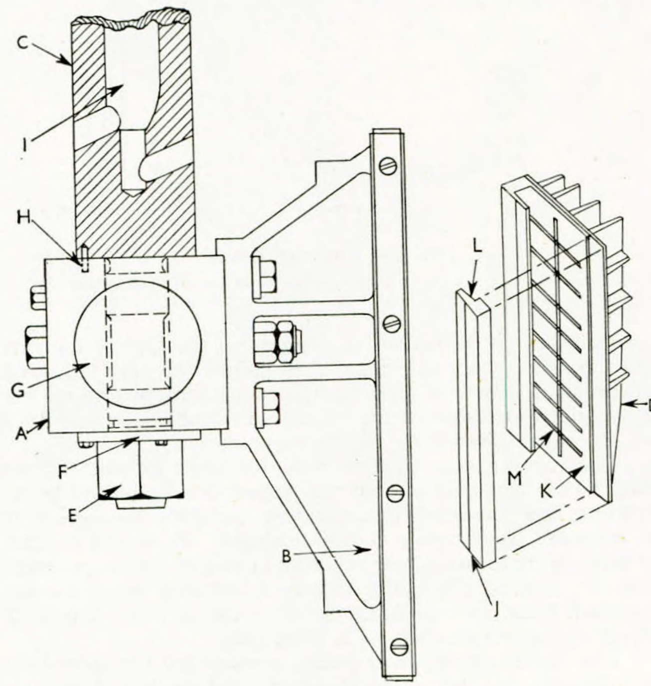

FIG. 12—*Crosshead guide*

A.—Crosshead.
B.—Crosshead shoe.
C.—Piston rod.
D.—Crosshead guides.
E.—Piston rod nut.
F.—Locking plate.
G.—Gudgeon pin.
H.—Dowel pin.
I.—Piston cooling passage.
J.—Guide bar.
K.—Whitemetal face taking ahead thrust.
L.—Whitemetal face taking astern thrust.
M.—Oil grooves.

A guide shoe of cast steel, lined with white metal on ahead and astern faces, is bolted to each crosshead. The gudgeon pins and guide shoes are lubricated with oil supplied through the bored-out connecting rods. The guide plates, which are bolted between the frames on their port sides, are made of special cast iron, and have accurately machined and scraped faces to take the ahead thrust. Cast iron guide bars are bolted to the guide plates to take the astern thrust.

The piston rods are of good quality forged steel and are connected at their upper ends to the pistons by means of flanges forged solid with the rods and secured at their lower ends to the crossheads by nuts. The rods are bored concentrically to allow the cooling oil to be conveyed to and from the pistons.

The pistons are made of special cast iron and designed to withstand the working temperatures and pressures without distortion. They are attached to the piston rods by means of steel studs and wrought iron nuts (see Fig. 13). Cast-iron piston rings are fitted. The pistons are oil cooled from the forced lubrication system, the oil being led to the pistons by

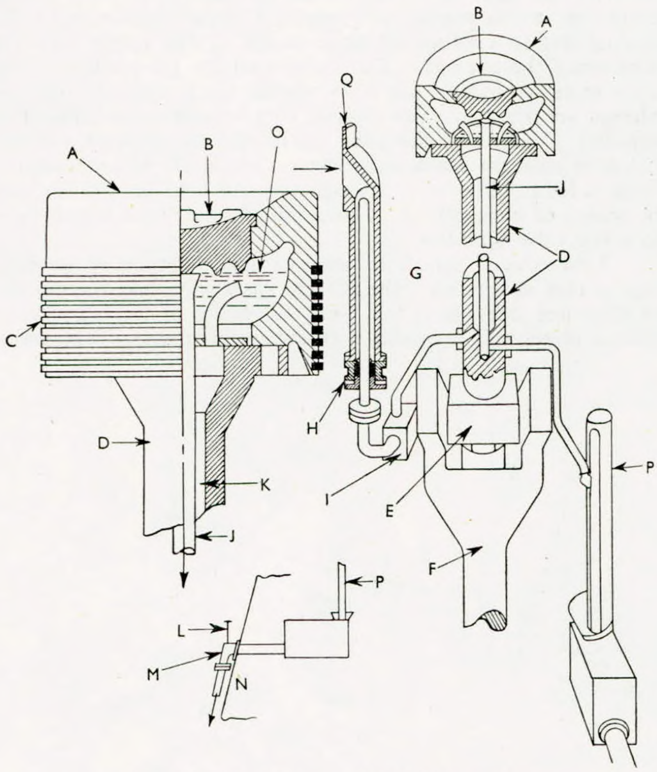

FIG. 13—*Piston and oil cooling system*

A.—Piston.
B.—Piston plug.
C.—Piston rings.
D.—Piston rod.
E.—Crosshead.
F.—Connecting rod.
G.—Telescopic pipes.
H.—Gland
I.—Cooling oil box fitted to gudgeon pin of crosshead.
J.—Cooling oil outlet pipe.
K.—Cooling oil inlet through bored out connecting rod.
L.—Thermometer indicating outlet temperature of piston cooling oil.
M.—Visible flow sight box showing piston cooling oil return.
N.—Engine frame.
O.—Cooling oil.
P.—Slotted stand pipe.
Q.—Pipe fastened to engine frame.

means of telescopic pipes. The moving members of the pipes are mounted on a bracket attached to the crosshead. The oil passes from the inlet telescopic pipes, through the bored-out piston rods, to the pistons and is returned, through a tube within the bored-out rods, through a nozzle, passing up and down a slotted stand pipe, to the outlet pipes and thence to the return main through a visible-flow sight box.

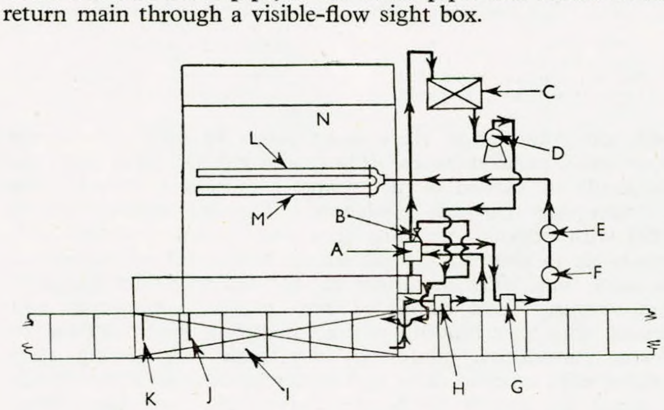

FIG. 14—*The forced lubrication system*

A.—Main engine driven lubricating oil pump.
B.—Relief valve.
C.—Elevated tank.
D.—Purifier.
E.—Alarm.
F.—Cooler.
G.—Filter.
H.—Strainer.
I.—Lubricating oil drain tank.
J.—Drain connexion from engine sump.
K.—Drain connexion from thrust block.
L.—Piston cooling oil supply main.
M.—Engine bearings and thrust block supply main.
N.—Engine.

The forced lubrication system is arranged to supply oil under pressure to all bearings below cylinder tops in the main engine, also oil for cooling the main pistons (see Fig. 14). The oil is circulated by the main engine-driven pump, with an independent steam-driven pump as standby; the pumps draw through a strainer from the drain tank under the main engine, and discharge through a filter and cooler into the lubricating oil main and the piston cooling oil main on the engine. The supply mains have an adjustable relief valve fitted with overflow to the drain tank. The pumps also discharge to an elevated lubricating oil tank, from which the oil passes through a purifier into the drain tank. A hand pump is fitted, and arranged to draw from the lowest point in the lubricating oil drain tank for discharging dirty oil or water into an oil refuse

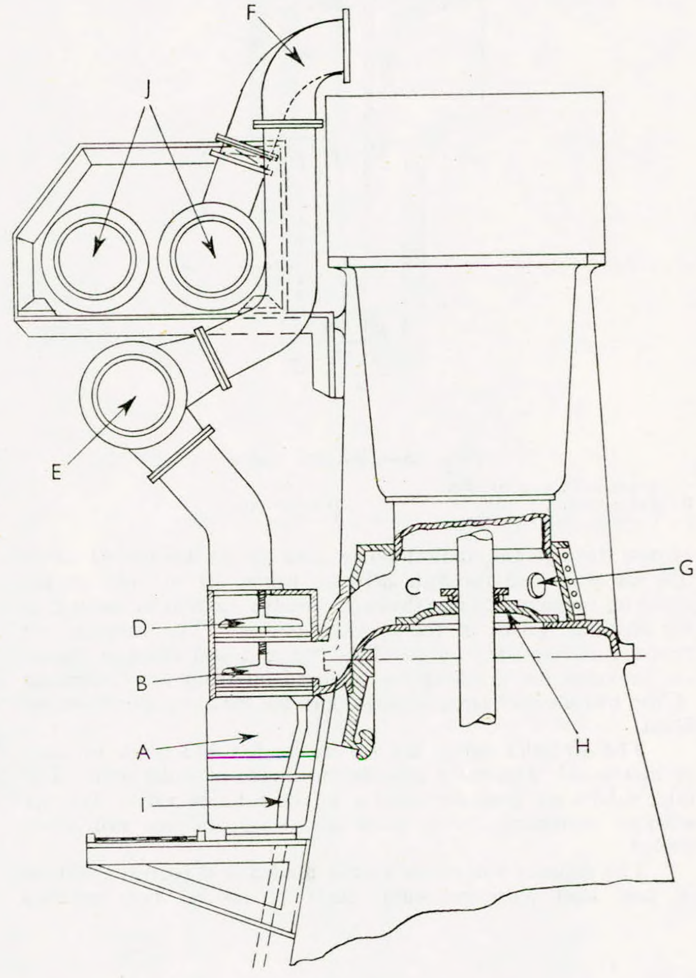

FIG. 15—*Arrangement of pressure induction*

A.—Air inlet manifold.
B.—Two suction valves.
C.—Supercharging chamber.
D.—Two delivery valves.
E.—Supercharged air manifold.
F.—Air pipe connecting supercharged air manifold to air inlet valve.
G.—Access door arranged in segmental casing.
H.—Piston rod scraper box.
I.—Drain pipe.
J.—Exhaust manifolds.

tank in the engine room. A pressure alarm is fitted in the forced lubrication system to give warning when the oil pressure falls below a safe value. All the necessary pressure gauges and thermometers are incorporated. The cylinders are supplied with oil from mechanical lubricators, generally driven by roller chains from the camshaft, having a visible flow of oil which can be regulated down to the smallest quantity; two or three point discharge is arranged for each main piston.

The space below the pistons is utilized for pressure induction (see Fig. 15). On the upward stroke of the pistons a partial vacuum is created on the underside, thus inducing air

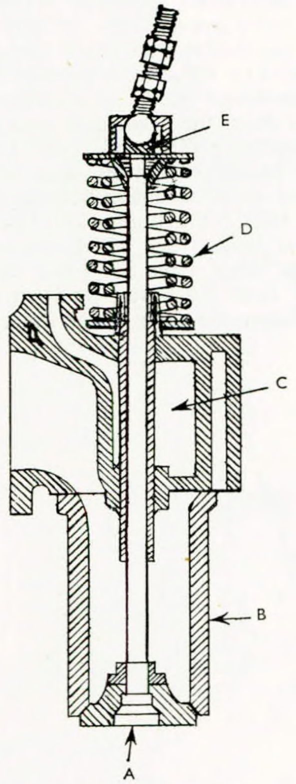

FIG. 16—*Exhaust valve*

A.—Valve head and spindle.
B.—Valve seat.
C.—Cooling water space in valve body.
D.—Springs.
E.—Hardened cup.

to pass through the suction valves, and on the downward stroke this air is forced through delivery valves set to open at the requisite pressure, thence through suitable air trunks leading to the air inlet valves in the cylinder covers. The scavenge air pressure is regulated by hand as required and efficient means are provided for draining the oxidized oil from the undersides of the pistons and large access doors for cleaning purposes are fitted.

The air inlet valves are spring loaded and work in cases of cast iron. The valve spindles are made of mild steel. The inlet valves are generally similar to the exhaust valves, but are without removable valve heads and seats and are not water cooled.

The exhaust valves are spring loaded and work in casings of cast iron with renewable seats of special heat-resisting material (see Fig. 16). The valves are secured to the cylinder covers by studs which are protected from contact with the cooling water. Casings are water-cooled at their upper parts for protecting the springs. The valve spindles are made of mild steel with removable heads of special heat-resisting material. Sleeves are fitted in valve casings to give maximum support to spindles. Valve spindles have hardened cups to work in conjunction with the hardened spherical heads of the lever adjustment screws. The valve spring caps are held on the spindles by means of cone collets. Grease nipples are fitted for lubricating the valve spindles.

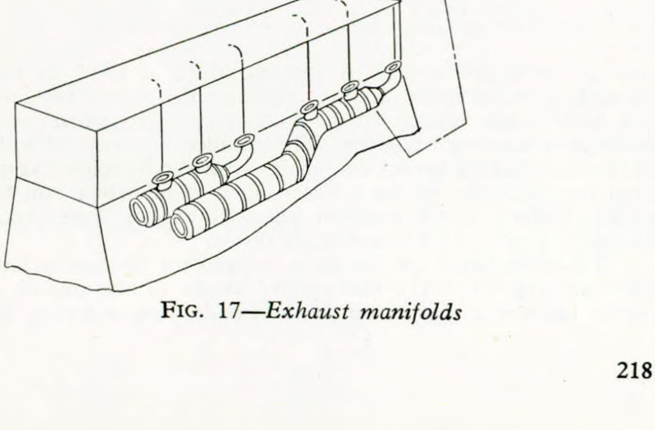

FIG. 17—*Exhaust manifolds*

Two exhaust manifolds are arranged, to improve scavenging, so that each receives the exhaust gases from half the number of cylinders (see Fig. 17). They are made of welded steel or riveted plates and provided with expansion pieces. The manifolds are connected to the exhaust valves by means of branch pipes and expansion pieces. The engine exhaust pipes from the manifolds are carried to the exhaust heat boilers, with by-pass to atmosphere through a silencer. The changeover valve is fitted with external operating gear and so arranged that it is impossible to shut off the exhaust to boiler and atmosphere at the same time. The manifolds are covered with high temperature resisting slabs embedded into plastic composition and secured with wire binding, supercoated with high temperature plastic composition reinforced with mesh wire netting, then covered with asbestos cloth and finally secured with steel bands arranged at a suitable pitch. Supporting plates are fitted along the underside of manifolds. Galvanized sheet steel cleading plates are fitted in way of expansion joints, lined with two thicknesses of white asbestos cloth.

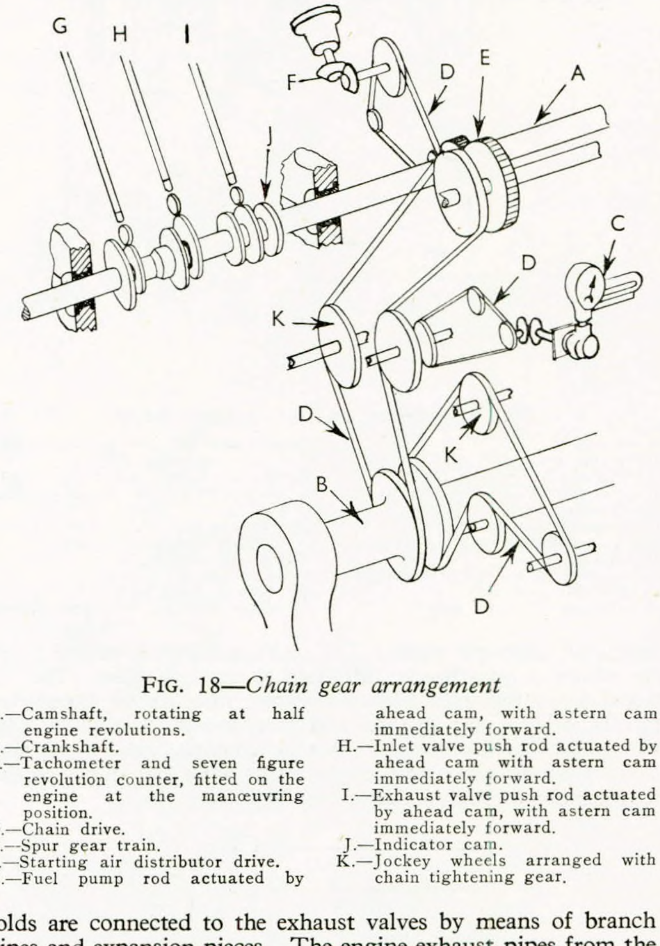

FIG. 18—*Chain gear arrangement*

A.—Camshaft, rotating at half engine revolutions.
B.—Crankshaft.
C.—Tachometer and seven figure revolution counter, fitted on the engine at the manœuvring position.
D.—Chain drive.
E.—Spur gear train.
F.—Starting air distributor drive.
G.—Fuel pump rod actuated by ahead cam, with astern cam immediately forward.
H.—Inlet valve push rod actuated by ahead cam with astern cam immediately forward.
I.—Exhaust valve push rod actuated by ahead cam, with astern cam immediately forward.
J.—Indicator cam.
K.—Jockey wheels arranged with chain tightening gear.

The camshaft is made of high quality mild steel and runs in gunmetal bushes which are supported by cast-iron brackets, bolted to the engine frames (see Fig. 18). Oil from the engine forced lubrication system is supplied to the camshaft bearings, oil baffles being incorporated to prevent loss of oil along the

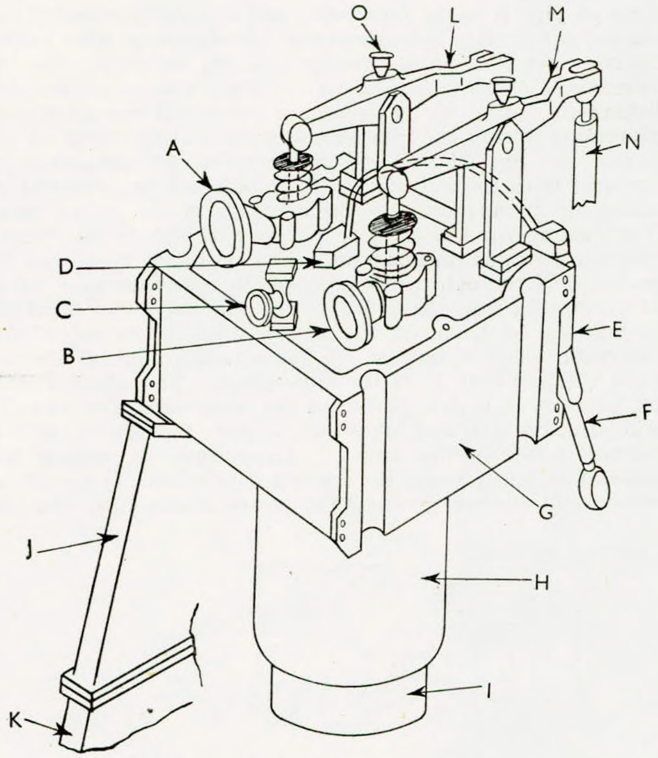

FIG. 19—*Inlet and exhaust valves*

A.—Exhaust valve.
B.—Inlet valve.
C.—Air starting valve.
D.—Fuel valve.
E.—Fuel injection pump.
F.—Fuel pump push rod.
G.—Cylinder cover.
H.—Cylinder jacket.
I.—Cylinder liner.
J.—Distance piece.
K.—Frame.
L.—Exhaust valve lever.
M.—Inlet valve lever.
N.—Inlet valve push rod.
O.—Oil cup.

shaft. The camshaft is driven from the engine crankshaft by means of accurately machined gearwheels and superior quality roller chain, so that vibration, chattering, etc., is minimized as much as is possible practically. The chain is provided with a jockey wheel for adjusting the tension.

The inlet and exhaust valves are operated by means of cast steel rocking levers (see Fig. 19). One end of each lever bears on the valve spindle and the other is connected to the upper end of a steel push rod. The lower end of the push rod has a roller engaging with the valve cam. Two cams are fitted on the camshaft for each inlet valve, exhaust valve and fuel pump, one for ahead and one for astern motion. The cams are specially formed to give quiet running. The rollers are made of hardened steel and ahead cams have renewable hardened steel toe-pieces. Each lever fulcrum is fitted with a bronze bush, which is supplied with lubricant from an oil cup lubricator of the wick drip feed type. A mild steel manœuvring crankshaft supported by the camshaft brackets (see Fig. 2), is connected to the lower ends of the push rods by short connecting rods. When this crankshaft is rotated by the reversing gear, the rollers of the pushrods are withdrawn clear of the cams and the camshaft, also involved in the sequence of events, is then moved axially by the reversing gear.

The main-engine-driven fuel surcharge pump is arranged to draw from the oil fuel service tank through a filter and discharges to the suction side of the engine fuel pumps.

The fuel injection pumps are of the single unit type, designed for solid injection of fuel, and arranged one for each cylinder (see Figs. 19 and 20). The pumps are connected to the manœuvring lever and governor by links and levers in such a way that the fuel supply to the cylinders can be regulated by adjusting the effective stroke volumes of the pumps. Each pump is operated by a cam and push rod. A pressure gauge is fitted to indicate that the pump is doing its work properly, special

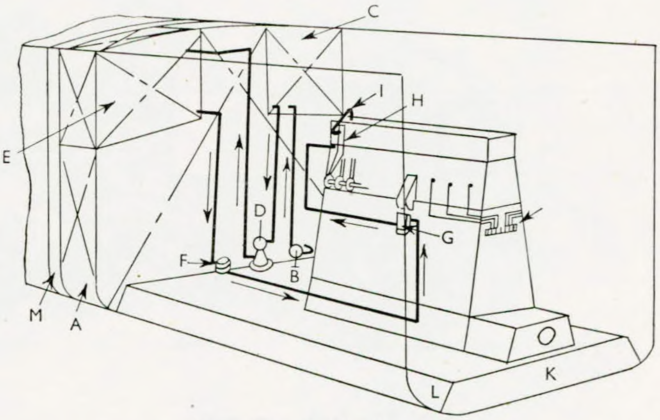

FIG. 20—*Fuel oil system*

A.—Fuel oil cross bunker.
B.—Fuel oil transfer pump.
C.—Unpurified fuel oil settling tank.
D.—Fuel oil purifier.
E.—Purified fuel oil settling tank.
F.—Fuel oil filter.
G.—Fuel surcharging pump.
H.—Fuel injection pump.
I.—Fuel valve.
J.—Cylinder lubricators.
K.—Double bottom tanks.
L.—Bilge.
M.—Coffer dam.

provision being made for damping out fluctuations at the gauge.

One fuel injection valve is fitted in each cylinder cover. They are of the automatic type, suitable for airless injection and operated by the pressure in the fuel pumps. The valves are cooled with fuel oil by-passed from the surcharge supply to the fuel injection pumps.

Steam-driven air compressors of approved make and number are fitted in the engine room. Each compressor has an adequate capacity of free air at a maximum pressure of about

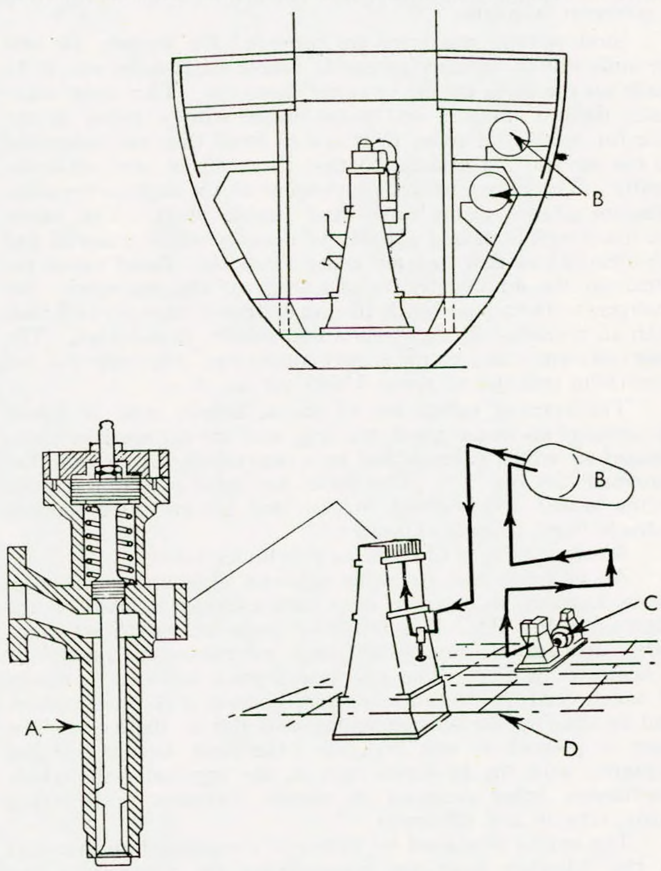

FIG. 21—*Starting air system*

A.—Starting air valve.
B.—Starting air reservoirs.
C.—Air compressor.
D.—Floor plates.

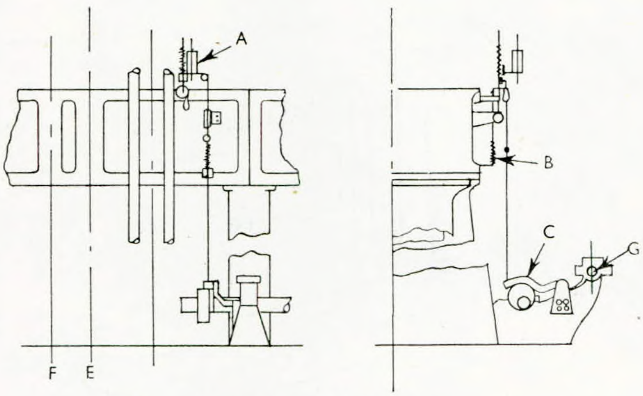

FIG. 22—*Indicator gear*

A.—Indicator.
B.—Operating cord tension spring.
C.—Lever arranged with compression spring, to keep roller on cam face.
D, E, & F are centre lines of frame, fuel pump and cylinder respectively.
G.—Manœuvring crankshaft.

350lb. per sq. in., when running at the designed speed and is capable of maintaining sufficient pressure in the reservoirs to start the main engine during normal manœuvring. The compressors are usually of the water-cooled, two stage type with intercoolers and drains provided between the stages. The steam engines are fitted with approved piston rings, governor, drains from steam cylinders and steam pipes, also by-pass connexion with valve from steam to exhaust line. The air charging line runs direct from compressors to air reservoirs only, and the air starting line runs direct from the reservoirs, through valves, to the main engine. The engine and compressor are mounted on a common bedplate.

Steel storage reservoirs are provided for starting air and are sufficient in capacity to enable twelve starts from rest to be made on the main engine in either direction. They have manholes flanged inwards and doors made with a recess in the face for joints. If more than one is fitted they are connected to the air starting system, so that they can be used independently. The reservoirs are provided with the necessary valves, pressure gauges, drain valves and fusible plugs. The valves are fitted with seats and spindles of non-corrodible material and are arranged so that they are easily accessible. Relief valves are fitted on the air compressor discharge to the reservoirs. An emergency starting air bottle having approved capacity complete with all necessary fittings, valves and gauges, is provided. The reservoirs are tested to the survey authorities' requirements, for a working pressure of about 350lb. per sq. in.

The starting valves are of the automatic type of ample dimensions to ensure quick starting, and are operated by compressed air which is controlled by a distributor driven from the camshaft (see Fig. 21). The valves are made of stainless steel, spring loaded and working in cast steel bodies. One starting valve is fitted to each cylinder.

A safety valve is fitted on each cylinder cover.

An indicator gear to enable indicator diagrams and 90 deg. phase diagrams to be taken from each cylinder is fitted on the engine (see Fig. 22). The indicator cocks are to British Standards Institution dimensions thus permitting a variety of indicator to be used. The indicator drum is actuated by means of tough cord, to minimize stretch, passing over brass pulleys and fastened to the tension spring and also to the lever. The lever is pivoted at one end, the other end having a roller engaging with the indicator cam on the camshaft—the whole mechanism being designed to permit diagrams to be taken easily, rapidly and efficiently.

The engine is started by means of compressed air supplied to the cylinders from the manœuvring air reservoirs. The starting air is controlled by the various valves and hand gear shown in Fig. 23. Before starting the engine, stop valve R is opened. A small opening of the valve R permits air to pass through pipe B to the pilot valve and into compartment L of the automatic valve through pipe F (see section of pilot valve). As the valve R is opened further, starting air passes into the chamber M of the automatic valve. While there is air pressure behind the piston K the automatic valve will remain closed, preventing starting air from reaching the starting valves on the engine (see Fig. 21). The starting valves are pneumatically operated from the starting air distributor and are arranged to admit air automatically to the cylinders at the proper time. The starting hand lever Q is now moved over to the "start" position indicated on quadrant. This operation moves rod W and lever X through trip V and thereby opens the pilot valve. It will be seen from the section of the pilot valve that the effect of this will be to cut off the supply of air to the top of the automatic valve K and at the same time to release the air from compartment L to the atmosphere. The effective load on the valve K is now greater on the underside. The valve K will therefore lift and allow air to pass through N to the starting valves on the engine. Immediately the engine has attained sufficient speed, the starting handle lever Q should be moved still further to bring the engine under fuel. As the

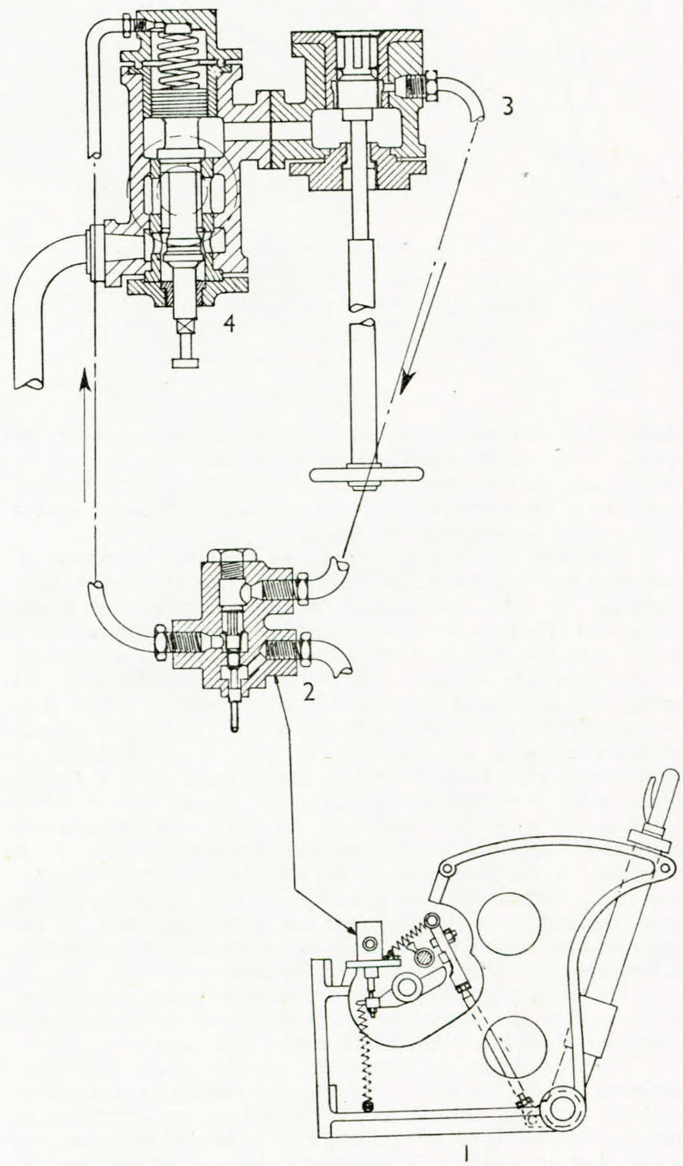

FIG. 23—*Diagrammatic arrangement of starting gear*

1.—End view of controls.
2.—Section through pilot valve.
3.—Starting air stop valve.
4.—Automatic valve.

starting hand lever is pushed further over, the adjusting screw A mounts roller C, the trip V disengages, and lever X will then jump back to its original position under the influence of the spring. The pilot valve will close instantly and the air will again pass from the stop valve, through the pilot valve, to the compartment L of the automatic valve. The load on top of the piston of this valve will now exceed that on the bottom, and the valve will close. At the same time the compressed air in the pipes on the engine will escape through P to the air silencer, which is open to atmosphere allowing the starting air distributor piston and rotating valve disc to be lifted clear of the distributor body. When starting it is always advisable to move the starting lever Q from the start position to the running position as smartly as possible. When the ship is under way, the starting air stop valve is kept closed. A drain cock (not shown) is provided on this valve and is opened frequently to drain away any water which might accumulate in the system. The starting gear is examined and cleaned occasionally to ensure perfect working at all times. A special portable lever, kept in a handy position, is provided and is used frequently for easing the automatic valve, so that it will always be in efficient working order.

The starting air distributor and switch cock is shown in Fig. 24. After the starting hand lever has been moved to the start position the starting air is permitted to pass through the automatic valve to the starting air valves. These valves are controlled by means of pistons actuated by air from the starting air distributor and switch cock, which take their air from the

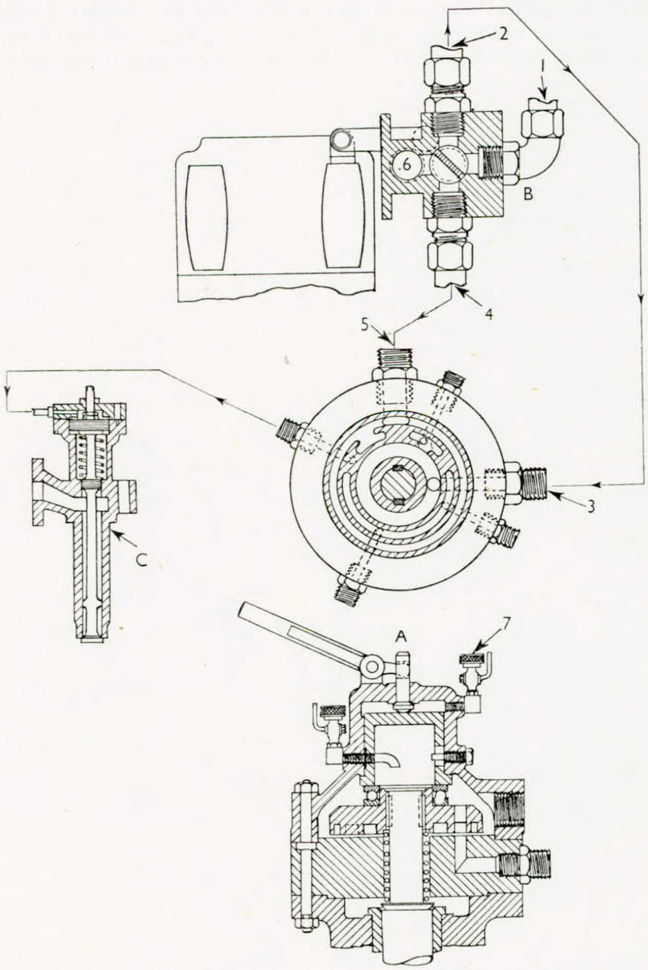

FIG. 24—*Air distributor and switch cock*

A.—Air distributor.
B.—Switch cock.
C.—Air starting valve.
1.—Air inlet.
2.—Ahead outlet.
3.—Ahead inlet.
4.—Astern outlet.
5.—Astern inlet.
6.—To atmosphere.
7.—Lubricator.

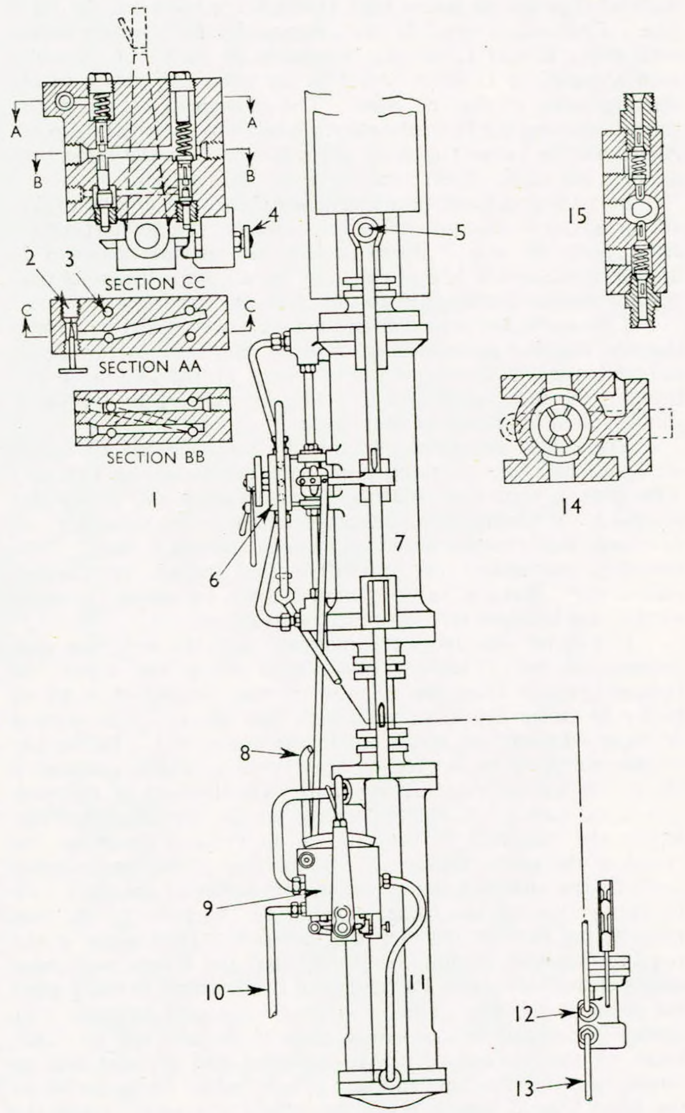

FIG. 25—*Reversing gear*

1.—Sections through air control valve.
2.—Air inlet.
3.—Air outlet.
4.—Trigger.
5.—Rack (enclosed).
6.—Switch cock for reversing gear hand pump.
7.—Oil brake cylinder.
8.—Branch from starting air.
9.—Air control valve.
10.—To air silencer.
11.—Air cylinder.
12.—Hand pump.
13.—Suction from oil system.
14.—Regulating cock for reversing gear oil cylinder.
15.—Section through switch cock.

starting air main at a point between the automatic valve and the cylinders.

From the air main, air passes through a T-piece and is split-up between the distributor and the switch cock in such a way that piston A in the distributor moves down and presses valve disk B into contact with the distributor body C before air is admitted to C via the switch cock. This ensures that no starting air will be lost by escaping between the working faces of the valve disk and distributor body.

Air passing through the switch cock may be directed to either the ahead or astern side of the air distributor by means of a valve D, which is actuated by means of links, by the travelling bearing E, which, in turn, is under the control of the main reversing gear.

Air leaving the switch cock by either the ahead or astern outlets is directed through the distributor body to the appropriate channels, F or G respectively, in the rotating valve disk. The outlets H to the starting valves are also in the distributor body so that the air passes back through the body via the valve disk. Channels F and G are continually in communication with ports K and L, so that whenever an outlet H coincides with either K or L air is passed to the operating piston in the starting valve of that cylinder. The movement of the piston permits starting air from the main's supply to enter the cylinder. Air cannot be present in both ports K and L at the same time as these are under direct control of the switch cock.

Ports K and L are of a length suitable to the engine timing and arranged so that one cylinder is always open to starting air. After ports K and L have passed, the exhaust channel N uncovers the outlets H and exhausts the air from the pipes thus permitting the operating piston to close the starting air valve.

Two oilers are provided and are employed when it is likely that the starting air equipment will be required. A few drops only are needed to ensure free working of the piston A and between the faces of B and C. The freedom of the piston A can be tested by means of the handle M.

The engine is reversed by means of compressed air acting on a suitable type reversing gear, with oil brake (see Fig. 25). The gear is provided with a hand-operated oil pump for reversing the engine when compressed air is not available. A direction indicator for ahead and astern motion is fitted. The reversing gear moves the camshaft in fore and aft directions in such a way, that the valve-operating cams for ahead or astern motion are brought into operation as required.

It will be seen from the diagram that the reversing gear consists of two cylinders, one placed below the other, the bottom cylinder being for air, and the top one, which is an oil buffer or brake cylinder, being kept full of oil. The pistons in these cylinders are connected to the piston rod. To the top of this common piston rod is fitted a rack, which gears with the pinion on the manœuvring shaft. On the back of this rack is a lineal cam piece, which, acting on the travelling bearing, moves the camshaft endways so as to place it in either the ahead or the astern position. The turning of the manœuvring shaft by the rack and pinion causes the rollers of the push rods to swing clear of the cams. Owing to the form of the cam piece at the back of the rack the camshaft travels along to the required position within the period that the rollers have been swung clear of the cams. Particulars of the principal working parts are given as follows. The air control valve unit contains four spring-loaded non-return valves, namely, an inlet and an outlet valve for the top end of the air cylinder, and an inlet and an outlet valve for the bottom end. These valves are operated by the hand lever C which, when put over to "Astern", opens the inlet valve for one end and the outlet valve for the other end simultaneously. When the hand lever is put over to "Ahead" the other two valves are opened and the astern valves are closed. The oil regulating cock (Fig. 25(a)), with double ported plug, and lever M is actuated by hand lever C through lever N and connecting rod. The cock is arranged so that when the hand lever is moved over to either the ahead or astern positions the oil is free to pass from one end of the brake cylinder to the other. The indicator rod fixed so as to move alongside the piston rod is provided with stop pieces engaging with the lever M when the gear moves either to the ahead or astern position alternatively brings back the plug L to neutral position with handle C as shown in Fig. 25(a). The ports in the plug are in this way closed, and further flow of oil from one end of the brake cylinder to the other end being prevented, the desired braking effect is brought about. In order to keep the gear in astern position, a trigger has been arranged in conjunction with hand lever C on air control valve, so that when the hand lever is returned to mid position and the gear is set for running astern, the inlet valve to the bottom and the outlet valve from the top of the air cylinder are held slightly open. When the lever is moved over to "Ahead" the trigger automatically goes out of action and the gear is secured in the ahead position under the influence of its own weight, i.e., at bottom of stroke. When the air control valve has been dismantled or disconnected for any purpose, care must be taken that the trigger is in "OUT" position when the reversing gear is set for going ahead. The valve A is a stop valve, and must be kept open while manœuvring or going astern. In the event of the air cylinder or the air control valves breaking down, the reversing can be accomplished in the brake cylinder by means of the hand reversing oil pump worked in conjunction with the switch cock and gear, section view of which is shown in Fig. 25(b). The pump draws oil from the lubricating oil system and delivers it to the switch cock, which has connexions to both top and bottom of the oil cylinder and to an overflow pipe which leads to the engine crankcase. Nameplates marked "Ahead" and "Astern" are fitted indicating the direction in which the handle is to be moved in order to put the cams into the ahead or astern position. At sea, or when the hand pump is not in use, the switch gear handle is kept in the neutral position indicated on the nameplate. It is important that the switch gear is set correctly for if it is not the oil will drain from oil brake cylinder through hand pump. The brake cylinder is tested daily. This can be done by opening the air cock (not shown) on top of the oil cylinder and giving the hand pump a few strokes to make sure that the cylinder is full of oil. The importance of this will be appreciated when it is fully realized that reversing without oil in the brake cylinder may lead to serious damage to the gear.

A governor of the inertia type is fitted to the engine to regulate the fuel supply if the engine tends to race (see Fig. 26). The arrows beside the diagrams, in Fig. 27, show the direction in which the governor is moving, diagrams 1, 3 and 5 being the upstroke and diagrams 2, 4 and 6 being the downstroke. When the engine is running at normal speed the weights are in the positions shown on diagrams 1 and 2, i.e., with the knife edge of the bottom weight A in its inner position and the knife

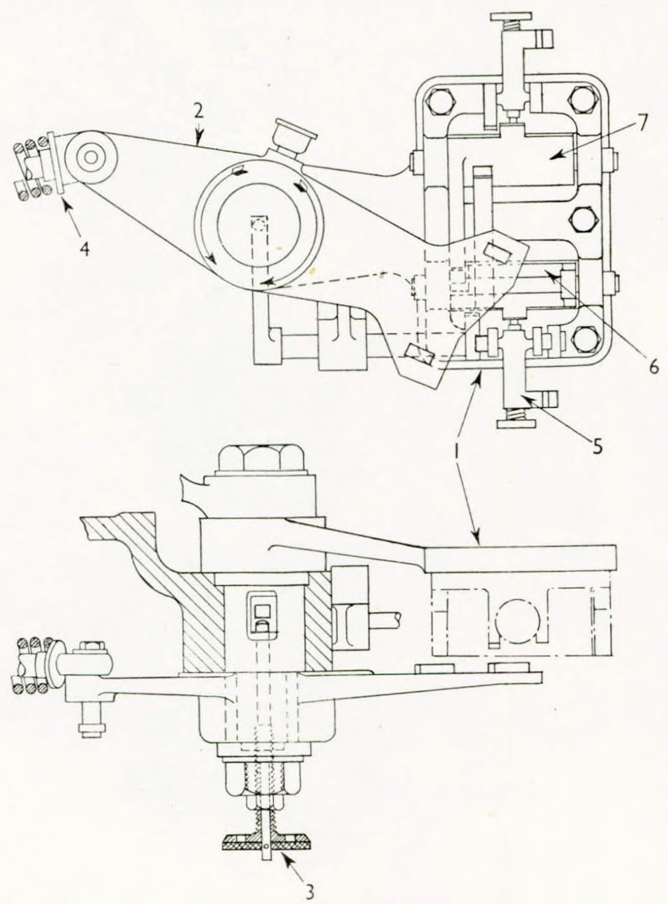

FIG. 26—*Governor*

1.—Governor arm in mid position.
2.—Normal position of regulating arm.
3.—Disc set to "0".
4.—Retaining lever.
5.—Spring box.
6.—Bottom weight.
7.—Top weight

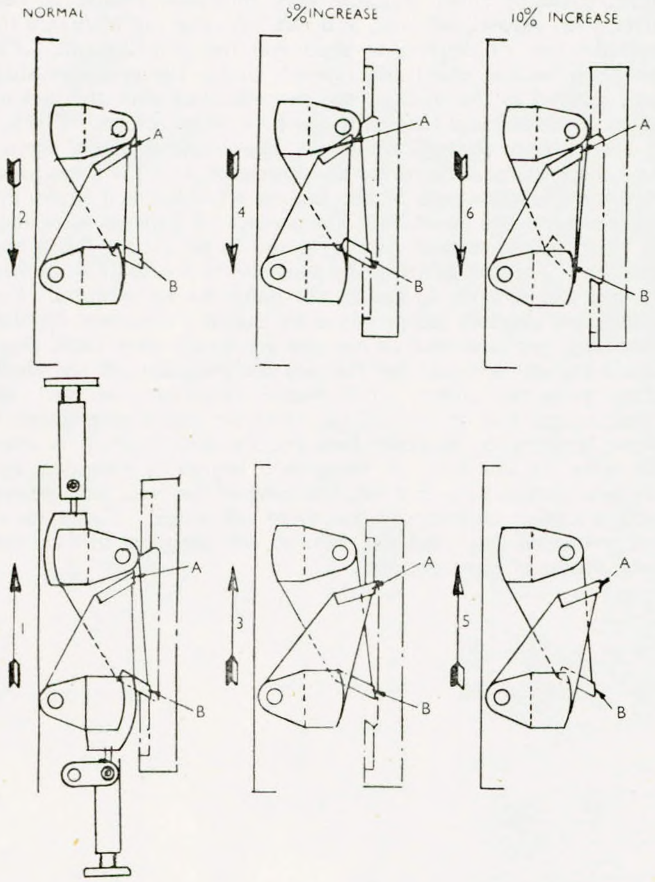

FIG. 27—*Diagrammatic sketch of governor operation 1, 3, 5, upstroke, 2, 4, 6, downstroke*

edge of the top weight B in its outer position, during both the upstroke and the downstroke. These positions are maintained as long as the engine speed does not rise 5 per cent above normal. On the engine speed increasing to 5 per cent above normal, the governor comes into action in the following manner. At the beginning of the upstroke of the governor the knife edge of the bottom weight moves into its outer position, diagram 3, engages with the regulating arm and lifts it to its upper position. At the commencement of the downstroke, diagram 4, the knife edge of the top weight (which is set to remain in its outer position until the speed increases to 10 per cent above normal) engages with the regulating arm and returns it to its original position. At the beginning of the next upstroke, the knife edge of the bottom weight again moves into its outer position and engages with the regulating arm and the operation is repeated. Thus, the regulating arm is kept moving up and down with the governor as long as the engine speed varies between 5 and 10 per cent above normal, and only those fuel pumps which have their injection period when the regulating arm is in its upper position are cut out, the rest being more or less reduced. Should the engine speed increase to 10 per cent above normal, the knife edge of the bottom weight moves into its outer position at the beginning of the upstroke as before, diagram 5, but at the commencement of the downstroke the knife edge of the top weight moves into its inner position, diagram 6, and the regulating arm instead of being returned to its normal position remains in its upper position until the revolutions drop below 10 per cent above normal, when the regulating arm will again move up and down with the governor and finally the speed having dropped to below 5 per cent above normal, the regulating arm will then remain in its lower position. When the regulating arm is held in its top position the fuel pumps are not cut out entirely, but reduced to approximately 10 per cent of normal full load. Hand adjusting gear is fitted so that the setting of the governor may be altered while the engine is running. The spring box of the bottom weight is fitted to a small crank, the shaft of which is turned, through a lever, by the graduated disk. Thus, should it be desirable to delay the action of the governor, by turning the disk in the direction of "Increase revolutions", the spring box is moved upwards and so increases the pressure of the spring on the weight so that more force is required to bring the knife edge into its outer position. Similarly by turning the disk in the direction of "Decrease revolutions" the spring box is moved away from the weight and less force is required to move the knife edge outwards. The final setting of the governor after test is such that the normal 5-10 per cent setting is obtained with the hand adjusting gear set at "0".

Alarms of approved type are fitted in the main sea water circulating, fresh water circulating, and the lubricating oil systems.

All similar parts such as cylinder liners and covers, pistons, bearings, fuel valves and pumps, etc., are strictly interchangeable so that spare parts can be fitted readily. This condition applies to all main and auxiliary machinery.

The plummer blocks are made of cast iron and have white metal in the top and bottom half bushes (see Fig. 28). Oil save-alls are fitted on bearings and the covers have lubricating boxes for oil and grease. The bearings rest on hardwood packing wedges and are secured by steel bolts and nuts. Water service pipes of galvanized steel are fitted, with cocks and conductors to each bearing.

The intermediate shaft is made of forged ingot steel, with solid half couplings, and is in excess of the survey authority's requirements for strength. The shaft is increased in diameter, and machined smooth in way of bearings. The half couplings of adjacent shafts are carefully fitted and bolted together with mild steel fitted bolts. Guard plates are arranged over shaft couplings, projecting on either side.

The propeller shaft is made of forged ingot steel with solid half coupling and is in excess of the survey authority's requirements for strength. It is turned all over and cased with a continuous gunmetal liner with bearing of ample length, water grooves are provided near forward end of liner. The screw thread at the outer end of the shaft is left-handed, for a right-handed propeller, and the taper for the propeller boss is $\frac{3}{4}$ inch per foot; a forged nut with locking plate is fitted and a rubber ring is incorporated to prevent ingress of water at the shaft. The key in the propeller shaft runs the full length of the propeller boss. A spare propeller shaft is securely fastened

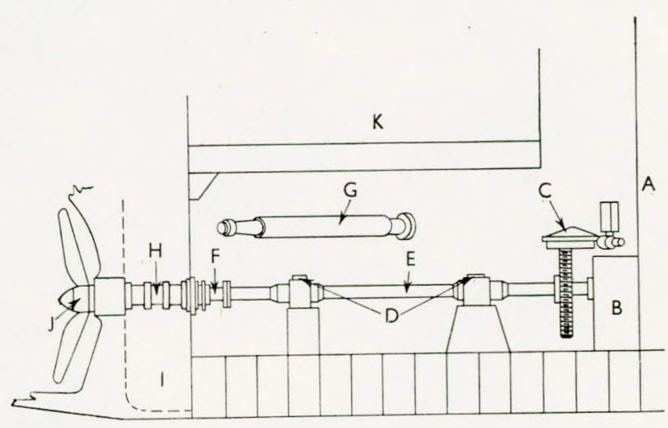

FIG. 28—*Propeller and shaft system*

A.—Main engine.
B.—Thrust block.
C.—Turning gear (with electric motor and starter).
D.—Plummer block.
E.—Intermediate shaft (diameter increased in way of bearings).
F.—Propeller shaft.
G.—Spare propeller shaft (fastened to ship's side).
H.—Stern tube (passing through after peak bulkhead and stern frame).
I.—After peak.
J.—Propeller boss and cone.
K.—Boiler room.

in the after end of engine room in a convenient position for ready substitution. The builders generally guarantee that the shaft system is free of all dangerous stresses due to torsional vibrations, inside the working speeds. A diagram showing critical stresses is usually sent to the owners, or their consultants, for their approval.

The stern tube is made of cast iron and strongly proportioned; secured at the outer end to the boss of stern post by a round wrought-iron nut and stopper pin, and at the inner end by a flange bolted to the peak bulkhead. The outer end has a long bush of gunmetal fitted with strips of lignum vitæ, with end-grain wood in the lower half, the wood being prevented from turning by gunmetal strips attached to the bush and so arranged that it can be drawn for renewal. A brass check plate, secured by studs, is fitted at after end of bush to keep the wood in place. The inner end of the stern tube is fitted with a brass neck bush and gland of cast iron, lined with brass. The gland studs are made of rolled naval brass, two of them being left long enough to allow the stuffing box to be packed at sea without the gland having to be taken right off the studs. A cock with pipe is fitted to allow water to play on the stern gland, also a connexion from the sea water cooling system for washing out stern tube.

The propeller is right-handed and of the solid four-bladed type, made of either stainless steel, maganese bronze or cast iron, with streamlined boss, and nut. A cone cap to match the outside taper of the boss is fitted over the propeller nut. The propeller boss is bored and tapered to suit the propeller shaft and secured to the shaft by the propeller nut with the locking plate dovetailed and secured to the boss by set screws. The key is carried right through the boss. The diameter, pitch, surface and shape of the sections are in accordance with the latest practice for the attainment of the highest efficiency and immunity from erosion and vibration. The propeller is finished accurately to pitch in the highest class style and inspected for finish and balance. The nut securing the propeller to the shaft is screwed to suit and securely locked in the hardened up position. Eye plates and shackles are provided on the ship's counter for dismantling purposes and strong cast or forged steel caps, studs and nuts are provided for starting the propeller off the shaft. Zinc protection plates, when bronze propellers are used, are fitted to the hull in way of the propeller and a rope guard is fitted between the propeller boss and the stern frame. A spare propeller of cast iron or manganese bronze is provided and securely fastened on deck aft, the cone of the boss well greased and protected against corrosion from salt water. Templates of the propeller cone and the keyway are supplied to facilitate easy fitting of replacements.

The Diesel Driven Heat Pump Evaporator

S. B. JACKSON (Member)

INTRODUCTION

In view of the increasing importance of the economic operation of all types of marine vessels, attention is drawn to a new type of Diesel-driven heat pump evaporator which offers a method of distilling fresh or sea water more economically than by any other method.

Regarding the smaller cargo vessel, a case is made for a large building programme during 1950-1954* involving construction of 4,000,000 tons gross, stressing that owners will only place orders provided that operating efficiency and cost can be greatly improved. Much of this tonnage will require steam propulsion, subject to the problems of the water side of modern water-tube boilers being solved.

The type of evaporator to be described requires no chemical knowledge, can be operated by crews of cargo ship standard and, if necessary, can be completely automatic, possesses highly economic characteristics combined with ease of installation and maintenance, with minimum repair cost. While labour costs will continue to rise, under such conditions first cost is not so important. The effluent from the evaporator may be used in complete confidence for high-pressure boilers, making the advantages of high-pressure steam available to the small ship.

The fresh water now to be carried on small cargo vessels owing to new regulations is at least four times that of pre-war. The principal disadvantage is the reduction of cargo deadweight capacity which this entails. Any device providing drinking water cheaply as and when it is required while providing boiler feed would have its initial cost offset against increased cargo carrying capacity during relatively few years of service. Supposing a vessel has now to carry 500 tons as compared with 125 tons, 375 tons of cargo are lost. At 15s. per ton, this amounts to £280 per voyage, approximately. With 100 voyages in twenty years life, increased total earnings of £28,000 are obtained. The present method of evaporation in association with simple treatment methods after evaporation lends to the production of good potable supplies.

Recently† the thermodynamics of heat pump evaporators were given within the limits of such occasions. A new heat pump evaporator is in course of development which will have numerous marine applications. The objective here is to expand the work then noted and to indicate its commercial potentialities.

FUNDAMENTALS OF HEAT PUMP EVAPORATION

The principles of the heat pump evaporator are given in Fig. 1 where steam vapour at any pressure and temperature was compressed by the vapour compressor, the compressed steam being returned to the evaporating vessel via the heat exchanger where it released its latent heat to the boiling water and was condensed. Contraflow principles of heat exchange were employed. The evaporator was eminently suitable for distilling sea water for marine purposes.

The energy expenditure and cost of evaporation were only a percentage of that of even quadruple effect evaporators. The amount of driving power depended upon the temperature difference between the compressed steam and boiling water temperatures. By adequate design these could be kept low, for the smaller the temperature difference the greater the performance efficiency of the cycle. The temperature difference need be no higher than was necessary to effect the heat transfer in the evaporator vessel.

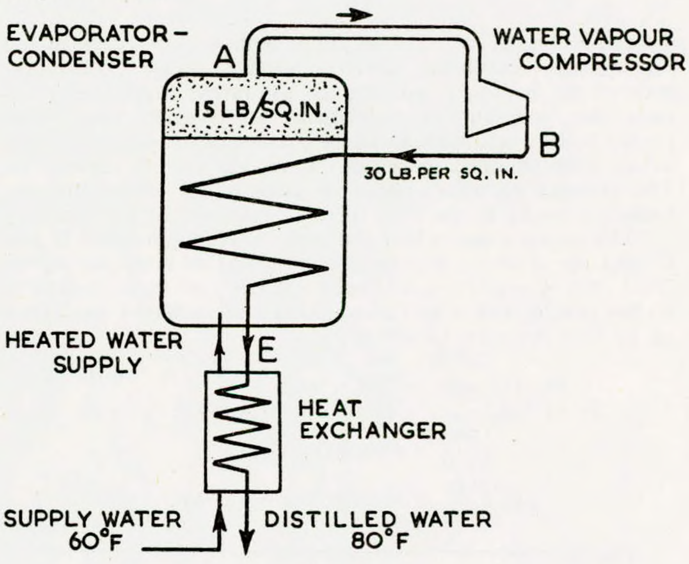

FIG. 1

By evaporating at atmospheric conditions no special sealing was required in the compressor, but the economy when evaporating at moderate pressures was tempting, considering that the engineering difficulties were negligible.

The theoretical coefficient of the heat pump evaporator is

$$\frac{T_1}{T_1 - T_2} = \frac{T_1}{\Delta T}$$

where the general temperature-entropy diagram indicated in Fig. 2 showed these values. It would be appreciated that such evaporators were not limited to the evaporation of water, but all types of solution could be catered for.

The actually realized coefficient of performance was obviously smaller than the above value, for account must be taken of the efficiency of the compressor, the temperature differences required for heat transfer, the pressure losses in the vapour pipes which should be as large as practicable, the influence of the height of vapour over the water evaporative area and any restrictions due to say, elimination plates, steam

*Shipping World, 5th October 1949, vol. CXXI, No. 2936, p. 377.
†JACKSON, S. B. 1946 Contribution to discussion on "Notes on Steam Jet Refrigeration for Marine Purposes" by W. Sampson, Trans. I.Mar.E., Vol. LVIII, No. 11, p. 216.

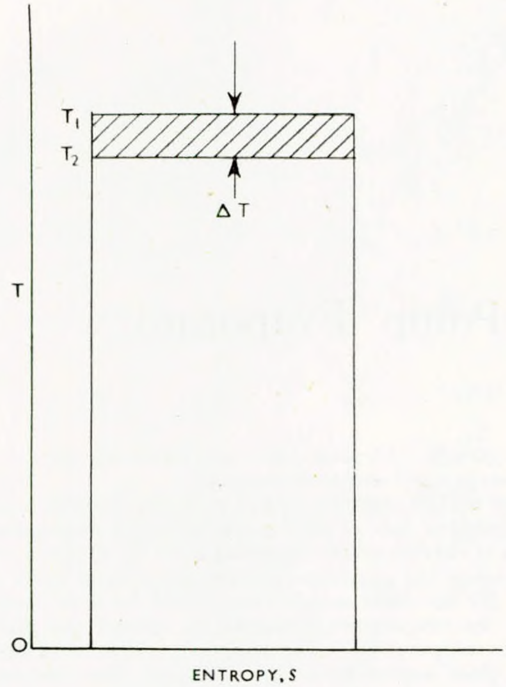

FIG. 2

separators, bends, etc. For solutions boiling with difficulty and consequently involving correspondingly greater differences between the boiling point and the saturation vapour temperatures, the coefficients of performance are relatively low. With readily boiling solutions, as in the present case, water, very high values were obtained, dependent upon the factors referred to. The presence of non-condensible gases or air affects the performance owing to the high thermal resistance of gas films.

Assuming a case where the water is evaporated at 213 deg. F. and the resulting vapour is compressed to 30lb. per sq. in. 250·3 deg. F., with an adiabatic efficiency of compression of 80 per cent η_c and a heat conduction and radiation loss factor η_{hl} of 92·5 per cent, then

$$\frac{250{\cdot}3 + 460}{(250{\cdot}3 + 460) - (213 + 460)} = \frac{710{\cdot}3}{37{\cdot}3} = 19{\cdot}10$$

$$19{\cdot}10 \times \eta_c \times \eta_{hl} = 19{\cdot}10 \times 0{\cdot}80 \times 0{\cdot}925 = 14{\cdot}10$$

$$\frac{970}{14{\cdot}10} = 68{\cdot}8 \text{ B.Th.U. per lb.}$$

$$\text{and } \frac{2545}{68{\cdot}8} = 37{\cdot}05\text{lb. per b.h.p.-hr}$$

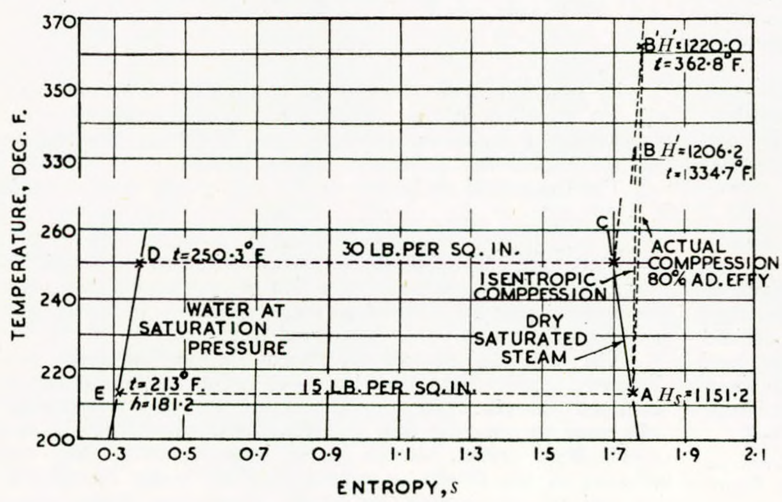

FIG. 3

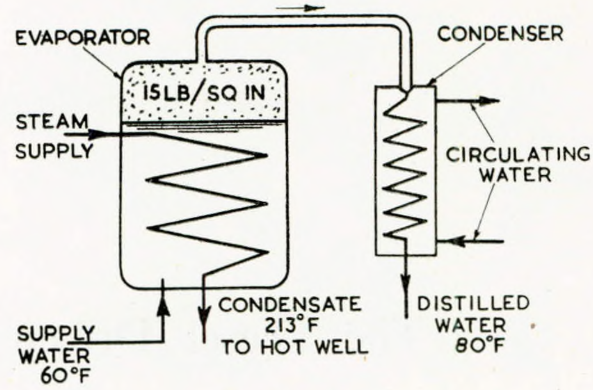

FIG. 4

Fig. 3 shows the temperature-entropy diagram* for this evaporator in Fig. 1 where the water was introduced at 15lb. per sq. in. abs. and 213 deg. F. and the heating steam again 30lb. per sq. in. abs., 250·3 deg. F., the raw water inlet temperature being 60 deg. F. Each pound of evaporated water has a final total heat of 181·2 + 970·0 = 1,151·2 B.Th.U. per lb. It thus obtains 1,151·2 = (60 – 32) = 1,123·2 B.Th.U. per lb. from the steam at higher pressure.

Assessing the isentropic efficiency of the compressor as 80 per cent, the condition of the evaporated vapour is assumed to be dry saturated at point A. Adiabatic compression is shown by the isentropic AB and produced an increase of total heat of 55 B.Th.U. per lb. Compression at 80 per cent adiabatic efficiency is shown by AB^1 giving the total heat of $\frac{55 \times 100}{80}$ = 68·8 B.Th.U. per lb. so that the superheated steam at B^1 has a total heat of 1,220 B.Th.U. per lb. When passed to the distiller this releases heat to evaporate the incoming water following line BCDE leaving at 213 deg. F. with a sensible heat of 181·2 B.Th.U. per lb. having given up 1,220 – 181.2 = 1,038·8 B.Th.U. per lb. The raw water accepts 970 B.Th.U. per lb. of this furnishing the latent heat of evaporation, leaving 1,038·8 – 970 = 68·8 B.Th.U. per lb. for sensible heat.

The raw water requires 181.2 – (60 – 32) = 153·2 B.Th.U. per lb. as sensible heat and 68·8 B.Th.U. of this can be released in the distiller leaving 153·2 – 68·8 = 84·4 B.Th.U. per lb. to be supplied. The distilled water leaves at 213 deg. F. contains 181·2 – (80 –32) = 133·2 B.Th.U. per lb. to dispose of in the cooler or heat exchanger giving the reserve of 133·2 – 84·4 = 48·8 B.Th.U. per lb. to meet the losses. The losses may be reduced by conventional methods applicable to evaporators.

The energy output from the Diesel engine which is presumed to be driving the compressor is equivalent to 68·8 B.Th.U. per lb. This is only $\frac{68{\cdot}8}{1{,}095{\cdot}2} = \frac{1}{15{\cdot}9}$ of the energy required to evaporate 1lb. of water in the simplest single stage comparable distillation plant as shown in Fig. 4 where the same evaporating conditions were observed. Live steam was supplied in the ordinary way as the heat conveying medium. The vapour from the water being distilled passed to a condenser where it gave up its latent heat received from the primary steam. Raw water at 60 deg. F. and distilled water at 80 deg. F. was assumed in both cases. While the steam thermo-compressor increased the ratio of distillate/steam used, it could not approach the result given.

It is desirable to use a low temperature differential and comparison ratio in order to reduce the power requirements. Further, as high an inlet pressure to the compressor as may be convenient results in low specific volumes handled permitting the reduction of

*DAVIES, S. J. and WATTS, F. G., 13th September 1948. Section G, British Association, p. 285, reprinted in *Engineering*, Vol. 166, No. 4312, p. 309.

the size of the compressor.

In the case given above, the specific volume of steam at 15lb. per sq. in. abs. is 26·29 cu. ft. per lb. and at 30lb. per sq. in. abs., 13·74 cu. ft. per lb. By raising the inlet pressure to say, 40lb. per sq. in. abs. a specific volume of 10·49 cu. ft. per lb. is obtained. If this is compressed to 55lb. per sq. in. abs., 7·82 cu. ft. per lb., which corresponds to the same pressure elevation already considered, it is observed that a very much smaller compressor can be employed for a given total output.

In this case, where the pressure and temperature rise are appreciable, $\frac{2,545}{68\cdot8} = 37\cdot05$lb. water are distilled per b.h.p.-hr. Consider a typical 25 b.h.p. Diesel engine; this delivers $\frac{25 \times 2,545}{68\cdot8} = 925$lb. of distilled water per hour.

TABLE 1

	Per cent.	B.Th.U. per
Power	32·0	63,500
Water jackets	31·5	62,500
Exhaust gas	30·0	59,500
Miscellaneous	6·5	12,900
Total	100·0	198,400

For such a machine, the heat balance is given in Table 1. Distilled water of high purity may now be circulated in the water jackets giving $\frac{63,500}{925} = 68$ deg. F. temperature rise. It will be noted that the temperature at the outlet to the water jackets is 148 deg. F. which is moderate, and conforms to good engine practice. Continuing through the exhaust heat boiler, an average recuperation for small types is about 65 per cent with an outlet exhaust gas temperature at about 275 deg. F. Hence, $\frac{0\cdot65 \times 59,500}{925} = 42$ deg. F., the final temperature and the hotwell temperature being 190 deg. F. The fuel consumption of such an engine using fuel of 19,350 B.Th.U. per lb. was 0·41lb. per b.h.p.-hr. making 10·3lb. per hr. for the engine at full load corresponding to 90lb. distillate per lb. of fuel consumed.

Consideration of the make-up requirements of a vessel of 50,000 s.h.p. maximum rating and 40,000 s.h.p. normal rating at 600lb. per sq. in., 850 deg. F., with an average vacuum of 28·5 inch Hg. shows that make-up demand would be 7,500-8,500lb. per hour for the propulsion machinery, indicating a standard evaporator of 10,000lb. per hour.

For a representative triple-effect steam thermo-compression evaporator (10,000lb. per hour) the heat balance is given below:—

		B.Th.U. per lb.	B.Th.U. per lb.
Heat Input to Evaporator			
High pressure steam			
1,360lb. per hr., 600lb. per sq. in., 850 deg. F....	= 1,360 ×	1,435 =	1,950,000
Evaporator feed			
9,100lb. per hr., at 60 deg. F.	= 9,100 ×	28 =	255,000
Circulating water			
5,200lb. per hr. at 60 deg. F.	= 5,200 ×	28 =	145,000
			2,350,000
Heat Output from Evaporator			
Distillate			
10,000lb. per hr. at 170 deg. F.	= 10,000 ×	138 =	1,380,000
Circulating water			
5,200lb. per hr. at 150 deg. F.	= 5,200 ×	118 =	612,000
Blow-down			
430lb. per hr. at 213 deg. F.	= 430 ×	181 =	78,000
Conduction and radiation losses			280,000
			2,350,000

As the water is at 60 deg. F. datum again, $\frac{1,950,000}{10,000} = 195$ B.Th.U. per lb. in steam and $\frac{10,000}{1,360} = 7\cdot4$lb. distillate per lb. steam supplied. Considering the boiler room efficiency 82·5 per cent, pipe loss efficiency 97·5 per cent, auxiliaries and miscellaneous loss factor 95·0 per cent, $\frac{1,950,000 \times 100}{82\cdot5 \times 97\cdot5 \times 95\cdot0} = 2,540,000$ B.Th.U. per hr. which with fuel at 19,350 B.Th.U. per lb. calorific value to compare with the Diesel, gave 131lb. fuel per hr.

It should be noted that the distillate from the heat pump was at 190 deg. F., whereas it was only 170 deg. F. for the triple-effect type. The performance should be corrected in favour of the heat pump.

An indication of the operating performance of a Diesel-driven evaporator distilling 10,000lb. per hr. with inlet raw sea water at 60 deg. F. is given by a coefficient of performance of 17·0 and 51·0lb. per b.h.p. hr., or 50 B.Th.U. per lb. distilled, requiring actually 196 b.h.p. necessitating, say, a 200 b.h.p. engine which could carry appreciable emergency overloads if required. Now the fuel consumption for such a machine approximates 0·375lb. per b.h.p. hr., giving on 196 b.h.p. the value of 73·5lb. per hr., denoting a fuel economy of 57·5lb. per hr. or 44·2 per cent.

The equipment thus comprised a distiller-hot water boiler unit providing hot feed of high quality to the boilers, or, if necessary, cold water supplies for domestic and other services. The daily output of the 25 b.h.p. machine was approximately 2,250 gallons, and gave some indication of the potentialities in special reference to the small cargo vessels referred to earlier. For turbine drives, the economy of recuperation of engine waste heat was not available. This limitation may be an incentive to oil engine drives.

Table 2 gives particulars of the actual operation of relatively small distilling units.

TABLE 2—PERFORMANCE OF HEAT PUMP COMPRESSION EVAPORATOR WHEN EVAPORATING SEA WATER

	Temperature, deg. F.	Weight, lb.
Sea water, inlet to heat exchanger	60	1·25
Sea water, outlet from heat exchanger	207	1·25
Steam, to compressor	212	—
Steam, from compressor	222 (18 lb. per sq.in.) abs.	—
Distilled water, inlet to heat exchanger	221	1·00
Distilled water, outlet from heat exchanger	70	1·00
Blow down, inlet to heat exchanger	213	0·25
Blow down, outlet from heat exchanger	80	0·25
Water evaporated per b.g.p.-hr.	—	42·50
Coefficient of performance	—	14·7

As a triple-effect evaporator has been compared with the Diesel-driven heat pump type, it is evident that owing to the low relative distillate production of the usual single-effect low pressure type serious attention will be directed to the Diesel-heat pump combination, which will compete effectively in all spheres of evaporation.

As remarked earlier, the Diesel driven machine will be of great value in the small ship since it brings the benefits of extremely high economy obtainable only on the largest land and marine installations.

VAPOUR COMPRESSORS

One machine of a series which has been developed is shown in Figs. 5, 6 and 7. It is of the positive rotary displacement type consisting essentially of a fixed casing or stator in which a rotor is fitted with blades free to move radially. As the rotor is smaller in diameter than the bore of the stator or casing and

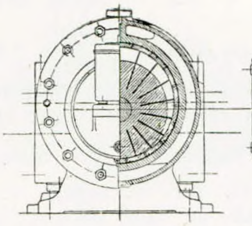

FIG. 5
Cross section, showing floating rings

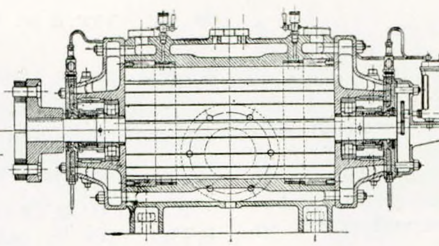

FIG. 6
Longitudinal section

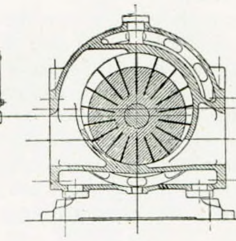

FIG. 7
Cross section, showing blades and ports

is placed eccentrically within it, there is a crescent shaped space between them. By rotation of the rotor, the blades are maintained by centrifugal force in contact with the floating rings and casing.

The space is divided by the blades into cells, which in turn expand from a minimum to a maximum volumetric capacity (suction period) and then decrease to a minimum volumetric capacity (compression period) the action of which will be understood from the cross-sectional elevation.

Floating rings near each end of the rotor revolve in unison with it and take the centrifugal thrust of the blades restricting and controlling their pressure. The consequent rubbing velocity, low friction, absence of reciprocating parts and valves allow of high speed without vibration or perceptible wear over long periods of continuous use.

Metallic glands of special design dispense with the use of stuffing boxes and packing; oil lubrication is automatic and extremely low.

The machine has the advantage of continuous aspiration and delivery with high adiabatic efficiency (80 per cent) mechanical efficiency (90 per cent) and volumetric efficiency (93 per cent). High peripheral speeds can be obtained. As compared with reciprocating machines for the same pressure range, higher speeds and smaller sizes are permissible for a given volume handled. All parts are designed to meet the conditions of evaporation and are highly resistant to corrosion. When, it is noticed that the machine handles virtually pure water vapour with probably a trace of free oxygen (O) or carbon dioxide (CO_2) confidence can be placed in the machine.

The provision of suitable moisture elimination plates or steam separators and the liberal design of evaporator surface area obviates carry-over to the machine. Owing to the partial vacuum existing over the evaporative surface the tendency to vaporize liquid molecules is appreciably increased. However, the raising of the temperature of the vapour in the compressor readily vapourizes the negligible amounts of liquid which may be carried to it.

The efficiency is unsusceptible to appreciable changes of speed or suction pressure, high values being obtained over a wide range of operating conditions. The design was such that a minimum of floor space, weight and supporting arrangements were involved, all of which were important in marine installations. Consequently, the erection cost was a minimum, for complete integral units could be supplied. Its low noise level rendered it suitable for applications where this feature assumed importance. It was suitable for long periods of uninterrupted duty.

Regarding the remainder of the evaporator variables, the temperature in the boiling vessel could be thermostatically controlled to ensure that the machine was supplied with vapour at the correct state at all times. The main factor which changed was ΔT which was controlled by the rate of scaling of the heat transfer surfaces, but this could be controlled extremely simply by chemical treatment referred to later.

Another compressor is of the centrifugal type, consisting of a single stage impeller operating at a speed permitting 5-10lb. per sq. in. pressure rise. One design takes in steam at 42lb. per sq. in. abs., 270 deg. F., 10·0 cu. ft. per lb. compressing to 47lb. per sq. in. abs., 277 deg. F. plus superheat at 9·1 cu. ft. per lb. The impeller has seventeen radial vanes and is of aluminium alloy forging heavily chromium plated which combines lightness with high surface hardness and desirable chemical properties as have been found necessary in the evaporation of certain liquids other than water. This design permits continuous operation at steam temperatures of 450 deg. F., which is far above those in the practical range of distillation. It operates at high speed and is dynamically balanced, and driven through suitable fabroil gearing. The adiabatic efficiency approaches 70 per cent. In this case close speed control is essential, yet the overall economy is high, owing to low ΔT.

A suitable steam labyrinth seal is mounted on the shaft to the rear face of the impeller. The ball bearing is the only part requiring lubrication. This is in a resilient mounting. Oil from the bearing is prevented from leaking into the steam stream by an oil seal which may be pressurized by steam tapped off the delivery volute and led to the seal by internal passages in the casing. Various angles of compressor inlets may be arranged to satisfy particular locations of the respective components of the complete equipment.

The lubrication system is designed to comply with Lloyd's specification in that it operates satisfactorily permanently tilted 15 deg. in any direction and also with a temporary tilt of 22·5 deg. such as might be experienced in marine work during a roll. The basis of the design is supported by many years of aircraft and marine service.

The incoming water may be circulated through the jackets of both types of compressor.

TABLE 3—ANALYSIS OF RAW WATER AND HEAT PUMP EVAPORATOR DISTILLATE

Condition at Time of Analysis

	Raw water	Single stage heat pump distillate
Appearance (after filtration	Clear	Clear
Colour	15 Hazen	5 Hazen
*p*H	7·3	6·3
Electricity conductivity	385·0	5·0
	Parts per 100,000	Parts per 100,000
Lime (as CaO)	9·5	—
Magnesia (as MgO)	1·8	—
Iron (as Fe) Total	0,005	—
Silica (as SiO_2)	0·6	0·012
Chloride (as Cl)	1·2	0·01
Sulphate (as SO_3)	1·0	—
Nitrate (as N_2O_5)	1·0	—
Phosphate (as PO_4)	—	—
Free carbon dioxide (as CO_2)	2·2	—
Total solids at 180 deg. C.	23·5	0·15
Bicarbonate alkalinity ($CaCO_3$)	17·6	0·7
Temporary hardness	17·6	—
Permanent hardness	3·1	—
Total hardness	20·7	—

Grains per gallon = parts per 100,000 × 0·7

To ensure that the steam is absolutely free from oil traces, in spite of precautions in compressor design, an oil filter may be included. Filters of very high efficiency as used extensively in food industries are available. As regards drinking water, no apprehension in this connexion need be felt.

CHEMICAL PERFORMANCE

Table 3 shows analyses* of raw water supplied to a heat pump evaporator and its distillate. These are the only details of heat pump evaporator chemical performance so far made and published in Great Britain.

It should be noted that the heat pump evaporator was operating under overload conditions at the time of test. Nevertheless, the distillate produced is of excellent quality for boiler feed purposes, 99·4 per cent of the original total solids being removed. While the *p*H of the effluent is low, this is attributable to a certain amount of oxygen (O) or carbon dioxide (CO_2) in solution. This condition can be corrected by the well-known methods of tri-sodium phosphate (Na_3PO_4) and or alternatively caustic soda (NaOH) according to the *p*H value or boiler conditions required. Alternatively, the gases may be released in a degasifying column. For marine work the former process is preferable. Consideration of the respective solubilities of the various constituents indicates that the carry-over did not exceed 2·5 per cent.

The removal of 98·2 per cent of silica (SiO_2) suggests the almost complete elimination of encrustation of superheaters and turbine blading due to carry-over through these components with consequent improvement of boiler and turbine efficiency.

It has recently been pointed out to the author by an eminent water conditioning chemist that the 0·012 part per 100,000 of silica in the final water could have dissolved from the glass Winchester quart while flying the sample from Switzerland to this country. This feasibility is known by the author to result in rapid etching of boiler gauge glasses in power stations using very high quality boiler feed.

Regarding the electrical conductivity, this is a good indication of the efficiency of distillation and the condition of the effluent. Better values may be obtained with conventional multi-stage evaporators, but in the author's experience of land industrial installations and modern power station evaporators they are only improved by triple and quadruple effect evaporators under more favourable specific evaporative surface area loadings with pre-conditioned raw water of zero hardness. When it is remembered that the machine is only a single stage type under overload, supplied with raw water, the performance denoted by the almost complete absence of values in the right hand column appears in its true perspective as highly creditable. Considering that a multi-stage type could be built in conjunction with individual "effect" vessels with large evaporative area, the performance of such a machine would no doubt be better than steam compression evaporation. But the performance indicates that such a design is unnecessary.

It should be noted that the total solids after evaporation are only 0·15 parts per 100,000. This is well within the published limits of 1·5 parts per 100,000 given by manufacturers of triple-effect plants.

The employment of such evaporators, which are entirely automatic, places the marine engineer in the position of merely being able to specify a given capacity knowing that the machine will do the job. He does not have to acquire the extensive chemical knowledge and practical experience which are essential properly to control chemical types of water conditioning plants. Such a claim is not only of interest to owners of small vessels, but to those of the highest powers.

The evaporator permits raw water to be taken in and treated on board ship. The desirability of this has already been discussed in reference to cargo ship considerations, but it may be that it is required in certain circumstances. If a base exchange zeolite plant preceded the evaporator, this could be supplied at negligible cost, automatically regenerated by sea water, the efficiency at all times over very long periods would be extremely high. In such circumstances, the boiler compound-starch treatment referred to later was unnecessary. Distillate of the highest purity was obtained with consequent improvement of boiler and turbine operation. The absolute freedom from scale resulted in high boiler operating efficiencies and maintenance attributable to descaling was eliminated. There would thus be considerable incentive to install such evaporators for marine purposes in both naval and commercial service.

With the zeolite pre-treatment, permanent and temporary hardness are removed. The water obtained was of zero hardness as compared with the lime-soda process which gave 2-3 parts per 100,000 hardness in the effluent. As, however, the process converted all the scale forming chemicals into solubles the total solids in the effluent may be higher than those in the raw water. The combination of evaporation of pure water and the continuous supply of water containing solids necessitates careful blow-down control in order to limit the concentration in the boiling vessel.

It will be observed that one machine could deal effectively either with fresh or salt water, from which boiler feed, domestic and drinking supplies could be obtained as required.

OPERATING CONSIDERATIONS

Apart from inadequate water straining, the principal cause of fouling of distilling plants is due to scale deposition. The rate of formation is affected by the brine density, brine chemical characteristics and operating temperatures. In order to keep the scale formation at a minimum, the design of the evaporator could be based on producing the rated capacity at sub-atmospheric pressures so as to avoid the tendency to precipitate in relation to temperature. This, however, involves consideration of the compressor and economic characteristics. It is felt that these factors should predominate. Having regard to this, chemical treatment will be required.

Starch injection has been applied to evaporators successfully.* United States boiler compound (alkali-sodium phosphate type) was added in the proportion of 1lb. compound and 4lb. starch daily. The result was that for two years tube nests have not required cleaning or descaling and only the finest possible scale was deposited on the coils and surfaces, despite the presence of a higher than normal proportion of scale forming matter. It would appear that starch provides a colloidal inhibition.

In a recent American survey of heat pump distillers† very little scaling was obtained in certain cases, while in others scale ranged from 0·025 inch to 0·25 inch thick. Reports showed that the evaporative capacity was not seriously affected by the larger amount.

It should be noted that only the water preheater surface area is affected by scale, as it is in this vessel where precipitation occurs. The evaporative vessel is entirely free from deposits.

The ships used a mixture of United States boiler compound in varying proportions and starch injected into the feed and these solutions were again found most satisfactory. Other mixtures have been developed in this country. The author is of the opinion that a combination of chemical and blow-down control provides the solution of the scaling problem. Since its nature is so well understood there need be no undue apprehension in this connexion. Referring to reliability of the distillate, Benke remarks that when using such methods indicated in his article, all make-up feed can be distilled from the sea direct.

Having regard to the foregoing considerations, it will be noted that the amount of blow-down can be maintained at a very low value without any adverse effect. It will be noted from Table 2 referring to the performance of a heat pump com-

*Made by S. B. Jackson from water provided by Professor Eischelberg, Technical Institute, Zurich 1948.

*Benke, A. L. 1946. Trans.I.Mar.E., vol LVIII, p. 161, "Starch Injection and Evaporator Output".

†Bethon, H. E. 1949. Society of Naval Engineers, vol. 61, p. 469, "Fouling of Marine Type Heat Exchangers".

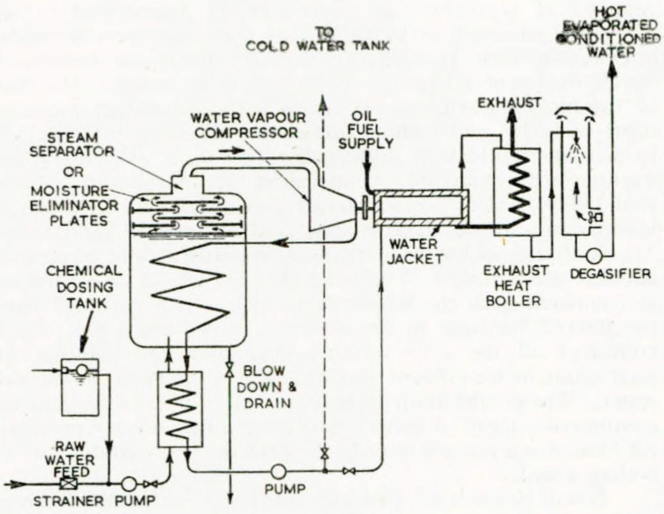

FIG. 8

pression evaporator when evaporating sea water, that the amount of blow-down is $\frac{0 \cdot 25}{1 \cdot 25} = 20$ per cent in the absence of chemical control methods. As the heat in blow-down is readily recuperated by means of a heat exchanger, the possibility of its reduction by chemical treatment of the water with the corresponding fuel economy is more attractive.

Fig. 8 showed a general arrangement of a Diesel-driven heat pump evaporator, which was self explanatory. A starting heater could be included in the evaporator vessel. Electric heaters may be used or sources of waste heat could be employed.

Reference to the various papers on the engining of cargo vessels of high power* shows that the amount of heat to evaporators varies between 1·75–4·0 per cent of the total heat in the boiler fuel of turbine driven ships. Steam is partially expanded through the turbine to a point, usually the intermediate pressure branch, at which it is applied to the evaporator. It is found that when using a vapour compression evaporator driven either by an independent turbine or electric motor supplied from the ship's mains, the overall economy is improved. Therefore, it will be found even more economic to expand all of the steam down to the lowest absolute pressure through the turbine making better use of it in the form of work on the propellor, in association with a separate Diesel-driven evaporator.

This permitted a reduction of boiler capacity of the order given above, and allows the maximum power generation of the turbine. It cannot be said that the contemporary cost of boiler and turbine plant is negligible and any reduction would be appreciated. Further, owing to the various factors obtaining there will be a lower overall oil consumption with a more flexible and independent evaporator arrangement. This latter characteristic enabled the evaporator to be used in harbour for filling boilers after overhaul, etc.

In the case given on p. 227, 131lb. fuel represents 1,800lb. per hr. reduction in actual boiler capacity (and weight) required and some improvement of turbine efficiency, in addition to the saving of 25·8 tons of oil fuel for each 1,000 hours operation at sea. The reduction in weight as compared with triple-effect is about 45 per cent.

EXPERIENCE AND PROGRESS

There is adequate experience in the operation of Diesel-driven units† which were extensively used during the recent war by the United States Navy. A total of over 10,000 were produced in sizes of 750, 1,000, 1,500, 2,000, 3,000 and 6,000 gallons per day of fresh water from sea water. Designs in the same country are being completed for capacities of 25,000, 50,000 and 100,000 gallons per day. Portable plants of up to 6,000 gallons per day capacity have also been developed and manufactured in quantities for military purposes.

In Switzerland a large number of heat pump evaporators of very large capacities have been produced for a diversity of purposes. The experience gained suggests that it is not a transitory product of the boom, but a valuable instrument of thermal economy for the present as well as the future.

A demand exists in this country not only for marine service but for numerous purposes on shore. It is reported that a large unit is to be installed in a power station. Further, other British developments both for land and marine applications are under active consideration, and one firm is initiating manufacture.

The evaporator could be used on shore stations in many remote parts of the world to distil fresh water for ships' and local purposes. In such locations where the arrival of fuel stores may be erratic or unreliable, due to the low fuel requirements of the machine under review it would tend to replace conventional evaporating plants now in use.

*Trans.I.Mar.E. 1948. Vol. LIX, p. 243, Symposium on "The Engining of Cargo Vessels of High Power.

†SPORN, PHILIP, AMBROSE, E. R. and BAUMEISTER, T. "Heat Pumps". (New York, John Wiley & Sons, Inc. London, Chapman and Hall Ltd.)

INSTITUTE ACTIVITIES

Minutes of Proceedings of the Ordinary Meeting held at the Institute on the 11th October, 1949.

An ordinary meeting was held at the Institute on Tuesday, 11th October 1949 at 5.30 p.m. The Chairman, Mr. G. Ormiston (Vice-Chairman of Council) in opening the meeting said: "As most of you are aware our Chairman of Council, Mr. Logan, is on a business trip to the Far East and is therefore unable to be with us on this occasion.

This is the first meeting of the Session and I have great pleasure in introducing to you our President, Lord Rotherwick, who has kindly come along this evening to present to us his Presidential Address.

Our President is a member of a famous shipping family. His father founded the Clan Line in the year 1878 and Lord Rotherwick has been a director of that well-known line of ships since 1903. He has also been a director of Cayzer Irvin and Co. since its incorporation in 1907.

Lord Rotherwick holds numerous other important appointments in connexion with shipping and I think you will agree that he is fully qualified to follow the eminent shipowners, who in the past have honoured us by occupying our Presidential Chair. I have much pleasure in calling upon our President to deliver his address".

The President, the Rt. Hon. Lord Rotherwick, then delivered his address (see p. 209).

In proposing the vote of thanks to the President, Mr. Ormiston said: "I think you will all agree that we have had a very interesting address from our President and even if a discussion were allowed, there would be few speakers who would disagree with the points Lord Rotherwick has so ably put forward.

There is one point, however, which I imagine would raise an argument and that is contained in one word reliability. There is a very old saying in this world which runs 'You can't get something for nothing'—and when the triple expansion engine with one auxiliary feed pump and one ballast pump, is compared with an up to date installation with high-pressure turbines or Diesel engines and the numerous necessary auxiliaries, I think most marine engineers will agree that the first was more reliable than the second. But if one of our President's modern vessels returned a consumption figure of 1·5lb. of fuel per horse power, then his superintendent engineer would not be asked if the job was reliable. As a superintendent engineer, however, I think that designers can do with an occasional reminder that reliability is *the* most important word in a marine engineer's vocabulary.

Marine engineers are a very conservative body and it is very evident that our President takes a great interest in their difficulties. We are in the habit of looking sideways at any drastic changes and I think it is safe to say that with our sea-going experience, we make sure that apart from any economy which may be gained, we have to be absolutely sure that reliability will not be affected.

It is evident from Lord Rotherwick's remarks that his company is feeling the affects of shortage of engineers. As you all know, numerous committees have attempted to solve this problem and the Council of this Institute has spent many weary hours also trying to find a solution. At a recent council meeting it was agreed that a scheme which had been prepared by a sub-committee should be forwarded to the Ministry of Transport for their consideration.

I think it is true to say that engineers conditions at sea today are better than at any time in the history of shipping, yet in spite of this we are short of sea-going staff. We can only hope that one of the many schemes which have been put forward will one day bear fruit.

Lord Rotherwick's closing remarks regarding present day costs is also one of the troubles of marine engineers and apart from this there is the difficulty in obtaining spare parts to keep ships in commission. I am sure there are many here today who have been advised by suppliers that they can deliver a simple component in six or eight weeks, when the ship is scheduled to sail in fourteen days. These conditions cause endless worry in the running of ships and we can only hope for improvement in the future.

On your behalf, I would like to thank our President for a most interesting address and I now call on our Hon. Treasurer, Mr. Robertson, to second the vote of thanks.

Mr. A. Robertson (Honorary Treasurer) seconding the vote of thanks said: "Your Chairman asked me to second the vote of thanks which he has so ably proposed. We are reminded tonight that our President is the second of his family to occupy the Presidential chair.

When your brother, My Lord, delivered his presidential address in 1930, our then Chairman, Mr. H. J. Vose, in introducing him, said that he came to them as a representative shipowner in a sense, rather like a 'Daniel in a lion's den' and expressed the hope that the members would accept him in terms of complete friendship and co-operation. We undoubtedly did so and further we believe he developed a very keen affection for this Institute and its members. We can assure you, My Lord, that you will command our esteem, admiration and affection, as your brother did in 1930-31.

It is interesting to note the differences between his address in 1930 and your address today. Sir August Cayzer gave a survey of various types of propelling machinery for ships from the old paddle steamer onwards to triple and quadruple engines, turbines, turbo-electric drive and internal combustion engines. He dealt particularly with the Bauer-Wach exhaust turbine which your firm were one of the first to install extensively. He dealt with the use of coal and oil in boilers, as well as pulvirized fuel, and expressed the view that the high capital costs of internal combustion engines was preventing British owners from placing orders. What would he have said of present day costs?

In your address today, My Lord, your theme has undoubtedly been efficiency and economy, but not at the expense of reliability. There are too many superintendent engineers in the audience this evening for me to enter into any expression of opinion on such a controversial subject. They should know what a shipowner wants and your remarks upon the subject will not go unnoticed.

I can only conclude by endorsing thoroughly the points made by our Chairman in proposing this vote of thanks and assuring you that we look forward to a very satisfactory year under your Presidency.

It gives me great pleasure to second this vote of thanks".

The vote of thanks was accorded with acclamation and the meeting then terminated at 6.35 p.m. There were seventy-five members and visitors present.

JUNIOR SECTION

Lecture at West Hartlepool

The lecture hall at West Hartlepool Technical College was filled to capacity on Wednesday, 19th October 1949 to hear Mr. Calderwood, M.Sc. (Member of Council) deliver his lecture on "The Combustion Turbine". Mr. W. E. Loveridge (Local Vice-President) introduced the lecturer and pointed out that he was an expert on his subject and had spent many years in connexion with gas turbine developments. As regards the future for the gas turbine, the chairman expressed his view that, judging by previous experience of other prime movers, there would be a fairly lengthy period during which various initial troubles would be eradicated but that it was almost certain success would be achieved sooner or later.

Mr. Calderwood opened his lecture by dealing with the historical aspect, tracing the basic idea of a combustion turbine back to the very early days of Hero's turbine, and went on to describe the first experiments connected with the running of a reciprocating oil engine in conjunction with the turbine system. He then showed diagrammatic arrangements of the various cycles which were being developed by many firms today, dealing with the advantages and disadvantages of each particular type. The problems to be solved were very often a question of metallurgy and as better materials became available, so would progress be accelerated.

The lecture was listened to with great interest by the audience, a considerable proportion of which were students from the College, together with a number of leading engineers of the district.

At the conclusion of the lecture, questions were invited, and it was evident from these that Mr. Calderwood had succeeded in stimulating a keen interest in his subject. The various points raised by the questioners were satisfactorily dealt with and the meeting concluded with a very hearty vote of thanks to Mr. Calderwood.

Lecture at Gillingham

At the Junior Section lecture on the "Launching of Ships" given by Mr. R. S. Hogg at the Medway Technical College, Gillingham, on 27th October 1949, the Chair was taken by the Deputy Principal, Mr. Tucker, B.Sc., in the unavoidable absence of the Principal, Mr. C. Colles, B.Sc., A.M.I.Mech.E.

Mr. Hogg gave a thoroughly planned lecture combining British, American and Canadian methods of launching ships, which proved most interesting to an audience which filled the hall, consisting of approximately eighty shipwrights and apprentices, students, draughtsmen and marine engineers. Mr. Hogg followed the lecture with a discussion period and answered all queries in a masterly manner. At the end of the proceedings on the proposal of the Chairman, a hearty vote of thanks was given to Mr. Hogg, and Mr. Owen, who was representing the Council, expressed his appreciation and thanks of the Institute to the Principal for the arrangements which had ensured such a successful lecture.

Lecture at Poplar Technical College

Mr. R. S. Hogg (Member) gave a lecture entitled "Launching of Ships" at Poplar Technical College on Tuesday, 15th November 1949. The Chair was taken by the Principal of the College, Mr. W. Laws, M.Sc., A.M.I.E.E., who during his introduction of the lecturer mentioned the cordial relationship which had existed between the Institute and the College for so many years and expressed his pleasure that the College had again been given the opportunity of hearing another Junior Lecture.

Mr. Hogg's lucid descriptions, his genial manner, and his knowledge of the subject was greatly appreciated by a mixed audience which filled the main lecture hall to capacity, and who proved their interest by the number of questions they asked.

At the finish of the lecture Mr. T. W. Longmuir (Member of Council) expressed to Mr. Laws the appreciation of the Institute for the facilities provided by the College and to Mr. Jackson, a member of the staff, for his able assistance with the lantern. He then proposed a vote of thanks to Mr. Hogg for the time he must have spent in preparation of such an interesting and instructive lecture.

Lecture at West Ham Municipal College

Mr. J. Calderwood, M.Sc. (Member of Council) delivered his lecture, "The Combustion Turbine", at West Ham Municipal College on the 18th November 1949, and the Principal, Dr. E. A. Rudge, in introducing the speaker expressed the deep appreciation of the Governors of the College of the policy of the Institute in providing lectures for students on subjects of current interest. He emphasized the great value to students in the opportunity to hear speakers who were authorities in their subjects, and pointed out that the college was very fortunate in having Mr. Calderwood give his lecture.

The lecture was most successful and was attended by over 100 students drawn from the main groups of courses held in the engineering departments of the College, such as First Degree and National Certificate courses. The evident appreciation of the lecture was shown by the wide variety and large number of questions put to the speaker in the discussion which followed.

A vote of thanks was proposed by Mr. Pratt and seconded by Mr. Levell, endorsed by the Head of the Department, Mr. H. C. Oliver and passed with acclamation.

Lecture at Belfast

A large audience of over 200 attended the lecture on "Photo Elasticity" by J. Ward, Ph.D., B.Sc. (Member) at the College of Technology, Belfast, on the 18th November 1949. The audience consisted mostly of young students but several teachers and members of technical staffs were also present. The lecture was of particular interest owing to the new installation of laboratory equipment in the new department on this subject. Dr. Ward held a lively discussion and cleared up many abstruse points of questioners. Mr. W. E. McConnell (Member) represented the Council, replacing Mr. Pounder (Vice-President), who was unavoidably called out of town.

Lecture at Liverpool

Approximately fifty students attended Dr. Ward's lecture on "Photo Elasticity" at the City of Liverpool Technical College on the 25th November. Dr. Grundy, Head of the Mechanical Engineering Department, was in the Chair. An interesting discussion ensued. Mr. G. Pickering (Member) represented the Council and proposed a vote of thanks to the lecturer and the College.

Lecture at Dartford

Mr. A. G. Crouch delivered his lecture on "Steam Generation for Power Stations" at Dartford Technical College on the 29th November to an audience of about forty-five. Mr. Crouch illustrated his lecture with twenty slides showing the development of water-tube boilers from their earliest days to the present time. A film entitled "Steam" followed the lecture and showed the manufacture of various components and the erection of large boilers for power stations. The meeting was highly successful and aroused great interest. Mr. J. D. Farmer represented the Council.

LOCAL SECTIONS

Cardiff

The First Annual Dinner of the Cardiff and District Local Section was held on Friday, 21st October at the Royal Hotel, Cardiff, when 133 members and guests attended. Mr. Ivor J. Thomas (Vice-President), was in the Chair. After the toast

"The King", Mr. Daniel Skae (Member) proposed the toast "Our Guests" to which the response was made by Mr. R. G. M. Street, who stressed the fact that in these days of strife and uncertainty, the engineers were the one body of people in the world who could be relied upon to carry on with efficiency. Mr. John E. Church (Member) then proposed the toast "The Shipping Industry" and in his speech said that the future of the industry in the Cardiff area depended entirely on the ability to export the only raw material of the country—coal. Continuing, Mr. Church said it was unthinkable that they should buy and transport from foreign oilfields fuel for shipping when beneath their feet there was the finest fuel in the world. Mr. John W. Roberts responded to his toast. After Mr. A. E. Brown had proposed a toast to the Cardiff and District Section of the Institute, Mr. Colin Moffatt (Member), in his reply, gave a brief *résumé* of the work done by the Section during his year of office as Chairman and said that he was pleased with the enthusiasm that had been shown by all members in making the section so successful.

Mr. Insley Blackmore (Member), proposed a toast to the Vice-President and Mr. Ivor J. Thomas in his reply said it was very gratifying to have Mr. W. J. Ferguson (Member of Council) and Mr. J. P. Campbell (Member of Council) with them and he hoped that they and others in London would honour their functions by their presence on future occasions.

The proceedings terminated at 10.15 p.m.

Sydney

About fifty members and thirty guests attended the dinner of the Sydney Local Section at the Carlton Hotel on the 16th November 1949. Mr. S. A. Smith (Member of Council) was in Sydney in the *Himalaya,* and with Mr. Hugh Livingstone, the chief engineer, attended the dinner. Mr. H. A. Garnett (Local Vice-President) was in the Chair and briefly welcomed the visitors. This was followed by the Toast of the Institute of Marine Engineers, ably proposed by Engineer Rear Admiral A. B. Doyle, C.B.E., who had recently retired from the position of Third Naval Member of the Australian Naval Board. This was replied to by Mr. W. G. C. Butcher, and following this, the Toast "Our Guests" was proposed by Mr. G. F. Ross and replied to by Mr. W. R. Hebblewhite, the President of The Institution of Engineers (Australia) and by Mr. S. A. Smith.

Other guests included Mr. H. G. Conde, the Controller of Electricity in Sydney; Mr. C. R. Bickford, the Chairman of the Sydney Division of the Institution of Engineers (Australia); Mr. A. E. Denning, Director of the Technical University, Sydney; Captain(E) E. A. Good, Engineer Manager, Garden Island; Professor MacDonald of the Sydney University; and numerous others whose interests were in or allied to marine engineering.

A visit to the new P. and O. liner *Himalaya* was arranged for 10th November, and about fifty members attended and were received by Mr. Smith and Mr. Livingstone, and given a most instructive tour of the engine room department, as well as a view of the ship. Great interest was aroused in viewing the lay-out of a new modern engine-room.

MEMBERSHIP ELECTIONS

Elected 31st October 1949

MEMBERS

Edwin Charles Bucknall, Lieut.(E), R.N.
William Geoffrey Cowland, Rear-Admiral(E)
Joseph Graham
Robert Johnson
Terence Kehoe, D.S.C.
John Carrick Kennaugh
Frederick James Moule, Com'r(E), R.N.
David Alexander Kean Nelson
Olaf Sinclair Olsen
Joseph Page
Cecil William Randall, Lieut.(E), R.N.
Eric Read, Com'r(E), R.N.
James Sydney Renfrew
John Marshall Scholefield
Frederick Irvin Sharman, Lieut.(E), D.S.C., R.N.
Lionel Meredith Sharp
John Robert Short, Lt.-Com'r(E), R.N.
George Storey
William Wilson

ASSOCIATE MEMBERS

Edward George Hutchings, B.Sc.
William Saint Lonsdale
Ian Donald McPherson Scott
Thomas John Strand, B.Sc.

ASSOCIATES

Arthur Thomas Bramley
Leslie Harris
Edward Bax Hough
Ewen Haddon Kennedy
John Desmond McCarten
Kenneth Reay
Kenneth Richardson
William Shillitoe
Charles Derek Smith
John Oswald Sykes
Alec Hugh Weeks

STUDENTS

Mark Varvill
John Ward
Reginald John Mayer

TRANSFER FROM ASSOCIATE TO MEMBER

Allan Francis Dearn
Percy Roland Gill
George Rowell Head
Ross Edward Kavanagh
John Grieves McInnes
Paul Chinniah Martin
Henry Ridley
Douglas Rhodes West

TRANSFER FROM ASSOCIATE TO ASSOCIATE MEMBER

Kenneth William Berry
William Richard Wallis

TRANSFER FROM STUDENT TO ASSOCIATE

John William Ellis Mansfield

Elected 5th December 1949

MEMBERS

Arthur John Cyril Blight
Humphrey Bott, Com'r(E), R.N.
David Young Brown
Ronald Blackwood Collin
Ernest Ralph Corke
William Andrew Edwards
Eric Victor Fewster
James Murray Graham
Alexander John Greenwood, D.S.C., Lieut.(E), R.N.
Frank Leslie Haines, Lieut.(E), R.N..
William Lawrence Horn
Henry Edwin Vaughan Joyner
Mathew Mather
Charles Alexander Maxwell, Lieut.(E), R.N., M.B.E., D.S.C.
Thomas Wilson Melville
Sydney Murie
Lewis Mickael Olsen
Horatio Nelson Pemberton
James Watt Rasmussen

Charles Edward Simms, D.S.O., Capt.(E), R.N.
Ralph Thompson
Ronald Laker Tucker

ASSOCIATE MEMBERS
James McEwan Anderson
Robert Dent Scotson

ASSOCIATES
Gordon Riddick Brown
Dennis John Capel
Robert Chambers
James Duncan
William Graham Howie
Thomas Victor Ingram
Walter Okeeffe Middleton
Kenneth Pike
Samuel Abraham Samson
Derek John Smart
James Stevens, B.Sc.
John Thomas Thirlaway
Edward George White
Cyril George Wood
Peter William Yarwood

GRADUATES
Nicholas Charles Harvey
Muhammad Akbar Yusuf, B.Sc.

STUDENTS
Michael Curtis Eames
Roger James Gates
James Andrew Russell

TRANSFER FROM ASSOCIATE MEMBER TO MEMBER
Eric Ronald Bathurst

TRANSFER FROM ASSOCIATE TO MEMBER
Edwin Arnold
Edward Lyndon Buls
Alexander Ferguson Carrie
John Anderson Duncan
Joseph Benjamin Hayes
Gordon Whitfield Hedley
Richard Woof

TRANSFER FROM ASSOCIATE TO ASSOCIATE MEMBER
Henry Arthur Littlewood

TRANSFER FROM STUDENT TO ASSOCIATE MEMBER
Thomas Leigh Taylor, Lieut.(E), R.N.Z.N.

OBITUARY

HEMSLEY LAWSON RONALD BELL (Member 4398) was born in 1883 and served his apprenticeship with the Fairfield Shipping and Engineering Co., Ltd. He was, since 1921, managing director of Messrs. Hemsley Bell, Ltd., tanker owners and was the founder member of this firm, which, incorporated in 1915, owns three vessels and is engaged in coastwise traffic and oil bunkering in the port of Southampton. During the war he managed a shipyard for the Admiralty in Oban, Scotland. He was killed in a road accident on the 6th November 1949, when his car was involved in a collision with a lorry on the Portsmouth road. He was elected a Member in 1921 and was also a member of the Institution of Naval Architects. A keen sportsman, interested in shooting and fishing, he was a member of the Royal Southampton Yacht Club, the Island Sailing Club and the Royal Southern Yacht Club. He was also a "messmate" of the Southampton Master Mariners' Club. He leaves a widow and a daughter.

HECTOR DICK BUCHANAN (Member 4243) was born at Greenock in 1881 and served his apprenticeship with Scott's Shipbuilding and Engineering Co., Ltd., Greenock. He served with the Argyle and Sutherland Highlanders for two years and was overseas during the South African War. He was employed by Swan, Hunter and Wigham Richardson, Ltd., Wallsend-on-Tyne, as a draughtsman for seven years until he joined the staff of Newport News Shipbuilding and Dry Dock Co., Newport News, Va. in 1913. After five years with this firm he was appointed a Ship and Engineer Surveyor to Lloyd's Register of Shipping in 1918 and stationed in New York. In 1919 he was transferred to Japan and served in Yokohama, Kobe and Nagasaki until the end of 1939 when he returned to the United Kingdom and was stationed at Birmingham. In 1943 he was transferred to the Society's office in Vancouver, B.C., and retired at the end of 1946. His services in the United States, Japan and British Columbia coincided with periods of great shipbuilding activity and expansion. He was elected a Member in 1921. He died on the 28th September 1949, after a short illness, aged 68 and was cremated at Vancouver.

JOHN CRIGHTON (Member 1623) was born at Dunkirk in 1867. He served his apprenticeship partly in Dunkirk and finally at Liverpool with Messrs. Griffiths and Co. He commenced his sea-going career with the Frederick Leyland Line and after service on many of their vessels, rose to the rank of chief engineer. He saw active service in the 1914-18 war and was torpedoed several times. In connexion with the sinking of the S.S. *Russian* in 1916 he was commended by the Admiralty for good service and devotion to duty. He was severely wounded in the sinking of the *Armenian*. In 1918 he was appointed Lloyd's surveyor at Bordeaux and was later transferred to Dunkirk, where he remained until his retirement in 1930. In 1920 he was awarded the Order of the British Empire for his war services. During the 1940 air raids, his home was severely damaged and his health undermined and after a long illness he passed away on the 14th October 1946 at the age of seventy-nine. He was elected a Member in 1902.

WILLIAM COWAN HILL (Member 3103) served his apprenticeship with Messrs. Scott and Co., Greenock, and G. McCartney and Co., Greenock. He spent five and a half years at sea and in 1916 joined the staff of the Singapore Oil Mills as Chief Engineer. He was elected a Member in the same year. In 1945 he was repatriated after internment in the Far East and returned in 1946 to his former employers, now the Ho Hong Oil Mills (1931) Ltd., Singapore. He died in the colony on the 10th June 1949. He was elected a Member on 18th April 1916.

WILLIAM NORMAN IMRAY (Member 8022) was born in 1879 in the U.S.A., and served his apprenticeship in the U.K. with the British Aluminium Co., and Messrs. Hall, Russell and Co., Aberdeen. He had eighteen years sea-going experience with the Union Castle Line and was for twelve years their resident engineer in New York, later being transferred to their Cape Town and Durban offices in the same capacity. From 1901 onwards he was in charge of their repair works at Blackwall, retiring in 1945. He was elected a Member in 1935. He died on the 25th August 1948, at the age of sixty-nine.

TOM JEFFERSON (Member 5852) was born at Carlisle on 10th April 1885 and was educated at Leeds Institute and University. After eight years with Kitsons and Co. (1901-9), three years in the shops and five years in the drawing office, he went to the West Coast of South America in 1910 as chief draughtsman of the Antofagasta and Bolivia Railway, Chile. He served with this concern for more than eight years, from 1910 to 1918, during the war period was acting as Works Manager for two years. In 1918 and 1919 he was in charge of locomotive erection and repairs for the Chile Exploration Co. From 1919 to 1923, he was Assistant Locomotive Superintendent with the Anglo-Chilean Nitrate and Railway Company at Tocopilla. From 1923 to 1931 he was employed by the Nitrate Railways at Iquique, first as Locomotive Superintendent until 1926, and afterwards as Chief Mechanical Engineer. In 1932, Mr. Jefferson joined the Central Railway of Peru as Chief Mechanical Engineer, a position which he held until his death on the 2nd April 1948. He was a Member of the Institute of Locomotive Engineers. The burial ceremony was held from the Masonic Temple, Lima. He leaves a widow and three daughters.

EDGAR LIONEL LULY (Associate Member 5604) was born in Tenby, Pembrokeshire in 1903 and received his early education at the Tenby County School. In 1919 he began his apprenticeship as a shipwright at H.M. Dockyard, Pembroke, where he continued his studies in the Upper School and gained an Open Science Scholarship to Swansea University College in 1923. In his last year at Swansea he was engaged under Professor Bacon in special work on fuels and combustion. He took his B.Sc. degree in Mechanical Engineering in 1926. The whole of his active career was spent with Messrs. Babcock and Wilcox, Ltd., whose Testing Department he joined in 1926. After scarcely more than two years he successfully carried out a mission to the Malay States to investigate the combustion properties of Malayan coals. He subsequently held various positions in the London Office of the Testing, Projects and Research Departments, where his fertile mind and unusually keen analytical powers were seldom brought to bear on a problem without some valuable result. The Babcock-Luly steam cycle diagram, for instance, was evolved as a means of lightening his own labours in the course of an investigation into the value of high pressures and temperatures. In 1944 he was appointed manager for the Birmingham area and held this position at the time of his death on the 25th September 1949, after an operation. He was elected an Associate Member in 1926 and was forty-six.

WILLIAM ROBERTSON PEARSON (Associate 10065) was born in 1920 and educated at Dumbarton Academy, obtaining a Higher Certificate, and later at various technical colleges and evening institutes. He served his apprenticeship with Messrs. Barclay, Curle and Co., Scotstoun. He commenced his seagoing career with Messrs. H. Hogarth and Sons, Glasgow, and was 4th, and later 3rd, engineer on their vessels. In May 1943, he joined the staff of Messrs. J. and J. Denholm, Glasgow, as 3rd engineer and obtained his 2nd class steam certificate in 1944. In December 1943, he became 3rd engineer with Chellew Navigation Co., Cardiff, and in June 1944, he was appointed 2nd engineer with his old firm of Messrs. J. and J. Denholm, of Glasgow. He was elected an Associate in 1944. He was torpedoed in 1944 and spent eight days in a lifeboat, with broken ribs, and as a direct result of these experiences contracted tuberculosis, spending three years in a sanitorium, and finally had a lung removed. His health was undermined by his war injuries and his promising career ended on 28th October 1949.

JOHN SINCLAIR (Member 97) was born in 1858 and served his apprenticeship with John Brown and Co., of Clydebank. He joined the staff of the P. and O. Steam Navigation Co. as a junior engineer and rose to the rank of chief engineer. He retired in 1916 at the age of fifty-nine as Commodore Chief Engineer through ill health. He later accepted a post with the Ministry of Production and served with this organization until the end of the 1914-18 war, when he retired permanently. He died on the 24th February 1948, at the age of ninety.

DEANS CARSON SMITH (Member 9985) was born in 1900 and educated at a private school at Hull and later in Sunderland, serving his apprenticeship with Messrs. Richardson, Westgarth and Co., Ltd., of Sunderland from 1916-1921. From 1921 to 1937 he rose from fourth to chief engineer and in 1937 joined the staff of Messrs. Harte, Lindley and Co., as assistant surveyor. He was responsible for supervising construction upkeep and damage repairs of a large fleet of steam and motor vessels. He held First Class Board of Trade Steam and Motor Certificates. He was elected a Member in 1944 and was also a Member of Society of Consulting Marine Engineers and Ship Surveyors. He was involved in a fatal accident when travelling on business to Cardiff on the 9th September. He leaves a widow and three children.

MOSES SOAR (Member 1600) was born at Cinderhill, Nottingham, in 1850 and educated locally. He began his career as a colliery engineer in this district and later became chief engineer of the Newton Chambers Collieries in the Chapeltown district for many years. He then joined the staff of Messrs. Buckleys, Ltd., of Sheffield, manufacturers of patent piston rings, as an agent and later became supervisor to the fitting of the piston rings, making several business trips to the Continent. He accepted an appointment with the Midland Manufacturing Co., Sheffield, as one of their directors and later with Mr. Unwin, Works Manager, started their own business as colliery agents and so formed the Unwin Soar Agency. He was still engaged on developing this venture until he retired at the age of eighty-five. In his private life he was an active church worker and was for many years Chapeltown Sunday School Superintendent. He was elected a Member in 1935 and died at the age of ninety-five on the 25th April 1945.

WALTER SPEEDY (Member 958) passed away peacefully on the 30th March 1949, at the age of ninety.

WILLIAM LAMOND STEVEN (Member 6473) was born in Glasgow in 1885 and educated at Abbotsford Public School and served his apprenticeship for two years in the drawing office and three in the engineering works of Messrs. J. H. Carruthers and Co., Ltd., Glasgow. He later studied at Glasgow Technical School and gained many prizes. After several years at sea with various companies he obtained his first class Board of Trade certificate and spent six years with the Glasgow Corporation Electricity Department as power station engineer. In 1922 he took an appointment with the British Phosphate Commission at Ocean Island and two years later as resident engineer he transferred to Naunu Island in the Central Pacific. On returning to the U.K. he accepted a post with Insurance Engineers, Ltd. After being released from the Army in the 1939-45 war he worked on aircraft and afterwards accepted an offer with a local industrial firm. He was an Elder in the Presbyterian Church of Scotland and a member of the British Legion. He was elected a Member in 1930 and died on the 22nd August 1949, at the age of sixty-four leaving a widow.

PERCY COLLINGWOOD STROTHER (Member 10416) was born in Newcastle is 1881 and was educated at a local school. At the outbreak of the 1914-18 war he was Chief Engineer on one of the Tyneside Steam Shipping Co.'s ships and was interned at Hamburg for the duration of the war. After the war he emigrated to Canada where he qualified as a Chief Engineer and was employed on a ship trading on the Great Lakes for several years. He returned to the U.K. and served at sea for a considerable time. Prior to the 1939-45 war he secured a post on the staff at the Woolwich Royal Arsenal, which post he retained until his health failed and he was obliged to take up a maintenance position with Messrs. Herbert Dees, Ltd. In 1947 his health completely broke down and after several operations he died at Croydon on the 8th May 1948, and was cremated there. He was elected a Member in 1945.

JOHN WARD (Member 2666) was born in 1880 and educated at Fettes College, Edinburgh, and Glasgow University and served his apprenticeship with Messrs. Denny and Bros. He commenced his sea-going career on T.S.S. *Monaki* of the Union Steamship Co., and served in various vessels of this company until he obtained his chief engineer's certificate. On his return to Dumbarton he joined the staff of Messrs. Takato and Co., and paid two visits to Japan, on his way home representing Lloyd's in Calcutta. On his return to the United Kingdom he was appointed outside manager and finally manager of Messrs. Hardic and Gordon. He died on the 16th July 1946 aged sixty-six.

DENZIL GEORGE WILLIAMS (Associate Member 6949) was born in 1897 in Swansea. He served his apprenticeship with the Premier Gas Engine Co. and Crossley-Premier Engines, Ltd., at Sandiacre, Notts, under the late Mr. J. H. Hamilton. He joined Messrs. Petters Ltd., at Yeovil in 1927 and was appointed chief draughtsman in 1937. In January 1939 he transferred to Loughborough to the Engine Division at Brush Electrical Engineering Co., Ltd., which post he retained until 1st August 1939 when he joined the staff of Messrs. J. and H. McLaren at Staines. He held this appointment up to the time of his death when he was drowned in a bathing accident at Bude, Cornwall on the 24th August 1948. He was elected an Associate Member in 1932 and was also a member of the Institution of Mechanical Engineers. He leaves a widow and daughter.

W. A. WOODESON (Member 2278) was born in 1869 and served his apprenticeship with Messrs. Alfred Grant and Co. In 1890 at the age of twenty-one he joined the staff of Clarke, Chapman and Co. He soon displayed a marked gift for invention, and in the earliest days of his association with the Gateshead firm designed the slow speed feed pumps with the characteristic distribution steam valves which bear his name. In 1900 as a departmental manager he turned his attention to steam raising plant. The company had manufactured a "Petersen" water-tube boiler for the *Buteshire* and he sailed with her to Australia and New Zealand on a guarantee trip. In 1901 he developed for land purposes the "Woodeson" patent water-tube boiler with straight tubes all of equal length expanded into embossed tube plates. He was appointed general manager and a director of the company in 1910. After serving as a Captain in the Royal Engineer Volunteer Regiment during the 1914-18 war, he returned as managing director of the Gateshead business and was made chairman of the company in 1928, a position he held until his death. Between the two wars he paid especial attention to the development of the company's range of marine auxiliaries. He was a keen student of the application of pulverized coal for steam raising and his patent burners, developed in 1928 and 1929 became well known. He took a direct and personal interest in his company's attempts to pioneer the use of pulverized coal at sea and his burners and associated equipment were installed on *Berwindlea*, the first vessel in the world to be built for pulverized fuel firing. He was elected a Member in 1909 and was also a member of the Institution of Civil Engineers, the Institution of Mechanical Engineers, the Institution of Naval Architects, and of the Society of Naval Architects and Marine Engineers, New York, U.S.A. He was President of the North East Coast Institution of Engineers and Shipbuilders in 1940-42 and was a Liveryman of the Worshipful Company of Shipwrights and was also a Freeman of the City of London. He died on the 30th September 1949.

INDEX

Papers and Discussions, Annual Report and Institute Notes

Index

Papers and Discussions, Annual Report and Institute Notes

Election of Members

MEMBERS

* Transfer

* Transfer

Index

ASSOCIATE MEMBERS

* Transfer

* Transfer

Index

GRADUATES

STUDENTS

*** Transfer**