Development of the Götaverken Diesel Engine

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Götaverken, having been concerned over a long period with the construction of various arrangements of marine propulsion machinery, developed their own design of two-cycle single-acting slow revolution Diesel engine in 1939. During the past twenty years this design has been produced in different sizes to meet the demand of increased outputs. An arrangement for running on heavy fuel has been made and turbocharging has been adopted in various engine sizes. To meet requests for machinery developing 20,000 b.h.p. and more on one propeller shaft, alternatives have been considered before deciding upon a Diesel engine of a type similar to the previous one, but with a larger cylinder bore and stroke. Some indications as to future developments are given.

The engine which today is known as the Götaverken engine is as a design only about 20 years old. Götaverken has, however, manufactured engines ever since the foundation of the company in the 1840's, the first steam engine for marine use being designed in 1861. The manufacture of marine Diesel engines was not taken up, however, until 1915, when manufacture under licence from Messrs. Burmeister and Wain was started. In the course of time very good collaboration was established with this company, a relationship which still exists, although the manufacture of the Burmeister engine at Götaverken has now ceased.

Parallel with this licence for the manufacture of propulsion and auxiliary engines in the 1920's and 1930's, Götaverken developed Diesel engine designs of their own, of which the best known is the so-called power gas machinery. This engine type, which was introduced† to this Institute in 1939, is a two-stroke Diesel engine with an automatically varying compression ratio so that it can deliver exhaust gas at a pressure of about 4 kg./cm.^{2} (57lb./sq. in.) above atmospheric pressure to a propulsion engine, either a piston engine or a turbine. Actually, tow boats fitted with these engines have been in service until quite recently, and during the 1930's a minelayer was rebuilt and equipped with machinery consisting of 4 power gas generators—gasifiers—(Fig. 1) and 2 gas turbines to the following particulars:

Gasifiers
Bore: 370 mm. (14·6in.)
Stroke: 480 mm. (18·9in.)
6 cylinders
R.P.M.: 325
Exhaust gas output: 2,175 gas h.p. corresponding to a mean effective pressure of 7·42 kg./cm.^{2} (106lb./sq. in.) based upon the turbine output.

Gas turbine
R.P.M.: 3,600
Shaft output: 3,320 s.h.p.
Overall efficiency of the machinery: 33 per cent
Total propeller output: 2 × 3 250 = 6,500 s.h.p.

In the early 1930's, 2 four-stroke auxiliary engines of different sizes were developed, which, with some minor alterations, are still included as standard practice in ships built by Götaverken and its subsidiary company, Oresundsvarvet.

The larger of these types has the following particulars:
Bore: 300 mm. (11·8in.)
Stroke: 450 mm. (17·7in.)
R.P.M.: 350-360
Cylinder output non-supercharged, 60 b.h.p.; and in a supercharged version, 90 b.h.p.

When, towards the end of the 1930's, several shipowners requested a single-acting, low revolution, two-stroke Diesel engine for their ships, the only way this could be met was for the company to design its own engine type.

The first of these engines was completed in 1938 and was designated 680/1500 VG-6, whereby the first figures indicate the cylinder bore and the second figures the stroke in mm. The letters VG represent the type of engine and 6 the number of cylinders. This engine was designed to develop 5,200 i.h.p. or 4,300 b.h.p. at 112 r.p.m. with a corresponding mean indicated pressure of 6·5 kg./cm.^{2} (92lb./sq. in.).

The uniflow system of scavenging was employed with a poppet exhaust valve in the cylinder head, the valve being operated by rods connected to levers in the crankcase, actuated by cams fitted on the crankwebs, there being no separate camshaft for the valve.

The cylinder covers were round and were secured to the cylinder blocks with heavy studs. Round covers were adopted for ease of dismantling, thus facilitating the overhaul of the pistons.

The fuel pump plungers were actuated from a camshaft driven from the crankshaft by a "Simplex" chain. This camshaft was light, as its work was confined to the operation of the fuel injection pumps.

The fuel injection was of normal solid injection type, with two fuel valves for each cylinder.

The pistons were oil cooled and the cylinders were cooled by fresh water.

These features have all been retained up to the present time.

The scavenging system was based on a common tandem piston-type scavenging pump placed at the forward end of the engine. A method later employed, using the underside of the working piston as a scavenging pump, led to the development of the present system, i.e. separate pumps for each cylinder

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† Abstracts of the Technical Press, Trans.I.Mar.E., Vol. 51, p. 52. "The Motor Ship", March 1939; Vol. XIX, pp. 450-451.

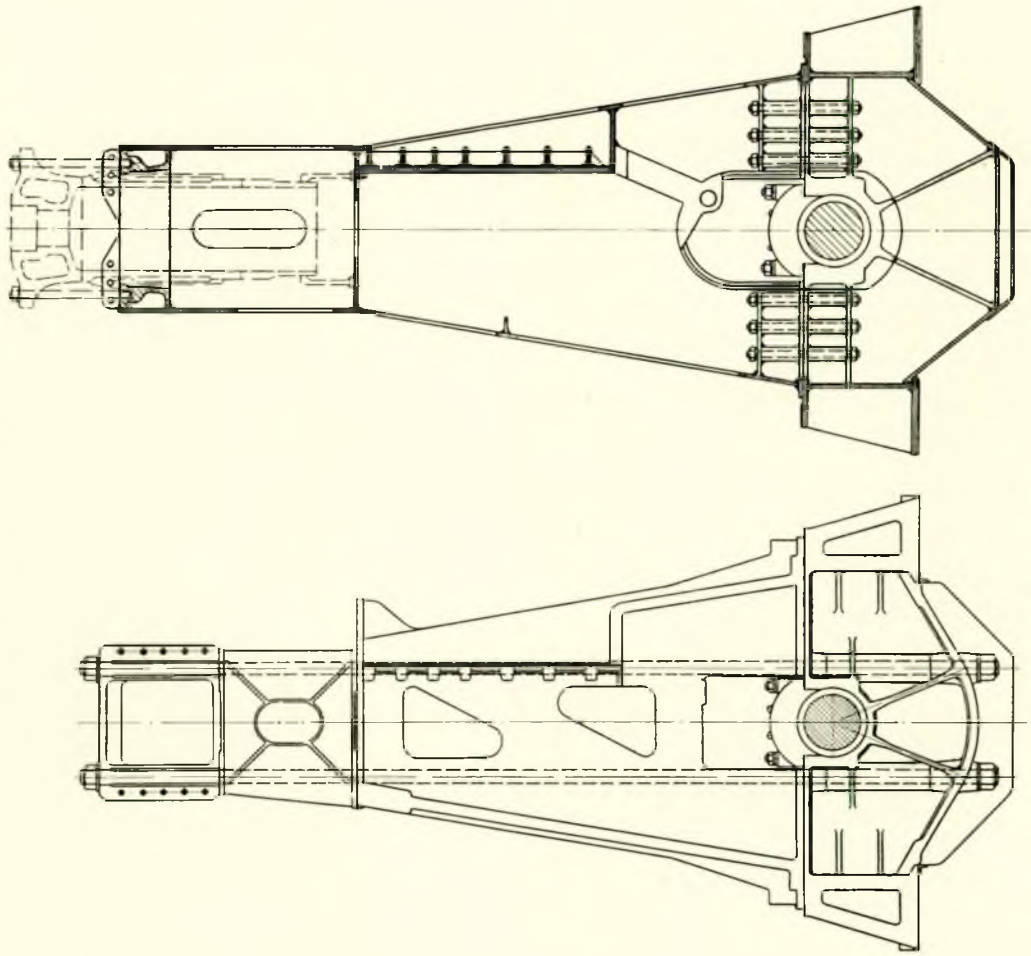


FIG. 2—*The cast entablature (left) and the all-welded arrangement (right)*

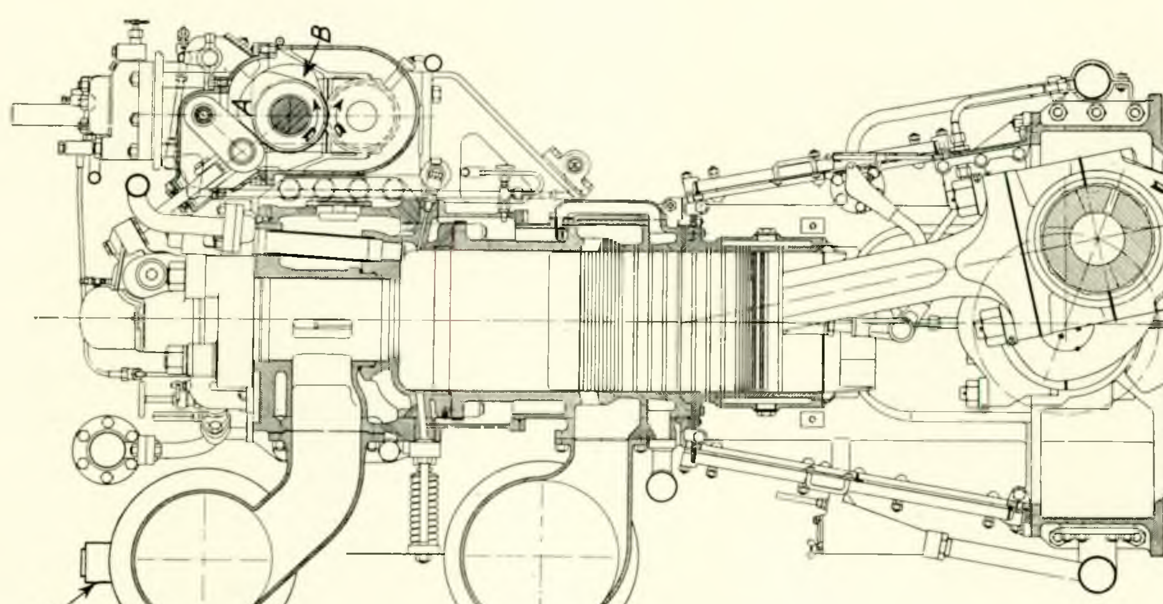


FIG. 1—*Cross section of a Götaverken gasifier*

built into the upper part of the engine entablature and driven from the crosshead.

This first engine type was supplemented a few years later by a slightly smaller one, designated 630/1300 VG. With a cylinder bore of 630 mm. (24·8in.) and a stroke of 1,300 mm. (51·2in.) this new engine developed 625 b.h.p. per cylinder at 125 r.p.m. In other respects the design was in principle the same as for its predecessors.

In the meantime a need for an even smaller engine had arisen, and in 1944 the first engine of this new series, 520/900 VGS, was tested, the particulars being as follows:
Bore: 520 mm. (20·5in.)
Stroke: 900 mm. (35·5in.)
R.P.M.: 160
Cylinder output: 375 b.h.p.

The letter S after VG indicates that the engine is all-welded, this design being chosen in order to reduce the weight further. The main features of the welded design compared with the cast iron design can be seen in Fig. 2.

Due to the development in size and speed of ships contracted after the war, higher outputs were required than the 680/1500 VG could offer. Thus, about a year after the war a new engine, 760/1300 VG, was designed with the following particulars:
Bore: 760 mm. (29·9in.)
Stroke: 1,300 mm. (51·2in.)
R.P.M.: 125
Cylinder output: 905 b.h.p.

One year later a slightly larger version with a lower number of revolutions, 760/1500 VG followed, i.e. with the same cylinder bore, but with an increased stroke of 1,500 mm. (59·1in.). The engine developed 935 b.h.p. per cylinder at 112 r.p.m.

All these types, with the exception of 520/900 VGS, had cast iron bedplates and entablatures. The experience with the welded type, however, from a running as well as from a manufacturing point of view, had been so satisfactory that welded versions of the 630/1300, 680/1500 and 760/1500 engines were also constructed.

Parallel with this conversion from cast iron to welded engines, the use of heavy oil was introduced as standard in the early 1950's. Thus, at present practically every engine is arranged to burn heavy fuel oil.

In order to bring this short summary of the Götaverken engine's development up to date, it should be mentioned that turbocharging was introduced as standard practice in the middle 1950's, and the great majority of the company's engines now being delivered or contracted for are turbocharged.

The largest engine so far installed is a 760/1500 VGS12U which develops 15,000 b.h.p. at 112 r.p.m. (Fig. 3).

DESCRIPTION OF THE PRESENT GOTAVERKEN ENGINE DESIGN

The various types of Götaverken engines described above are, as already mentioned, in principle similar in design, whether welded or cast. Therefore in describing the current engine, the size at present most commonly manufactured has been chosen, the 760/1500 (760/1500 VGSU when welded and turbocharged). This description will therefore deal with such engine features as are typical in the Götaverken design. (Figs. 4 and 5).

A guiding principle in designing has been primarily to meet the demands for operational security, facility in handling, and at the same time to keep the maintenance costs as low as possible. Another factor which has had a great influence on the engine design is the high wage level in Sweden, especially in the shipbuilding industry. This has forced them to reduce the number of man hours, for instance by avoiding too many machining surfaces and by avoiding large working tolerances, and by designing the components for ease of handling when erecting the engine in the workshops and when installing it on board.

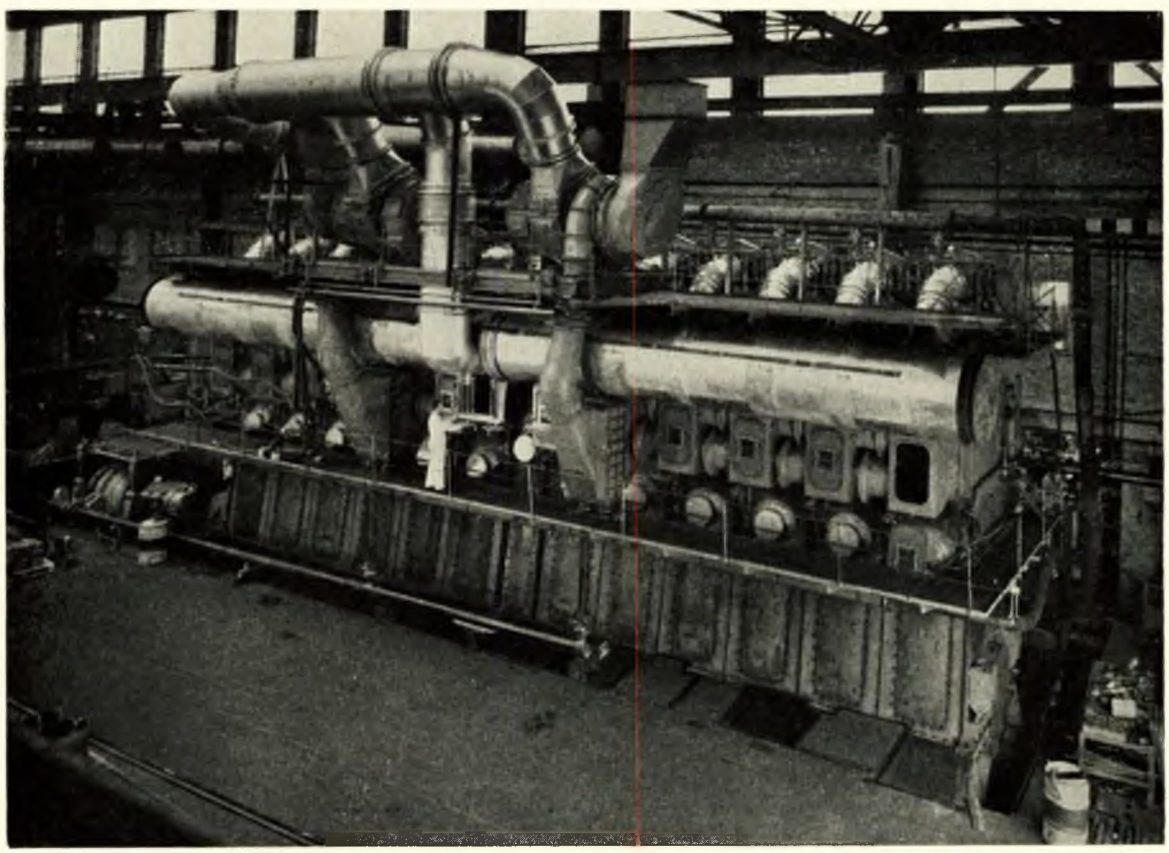


FIG. 3—*760/1500 VGS-12U Götaverken engine built in 1959, developing 15,000 b.h.p. at 112 r.p.m.*

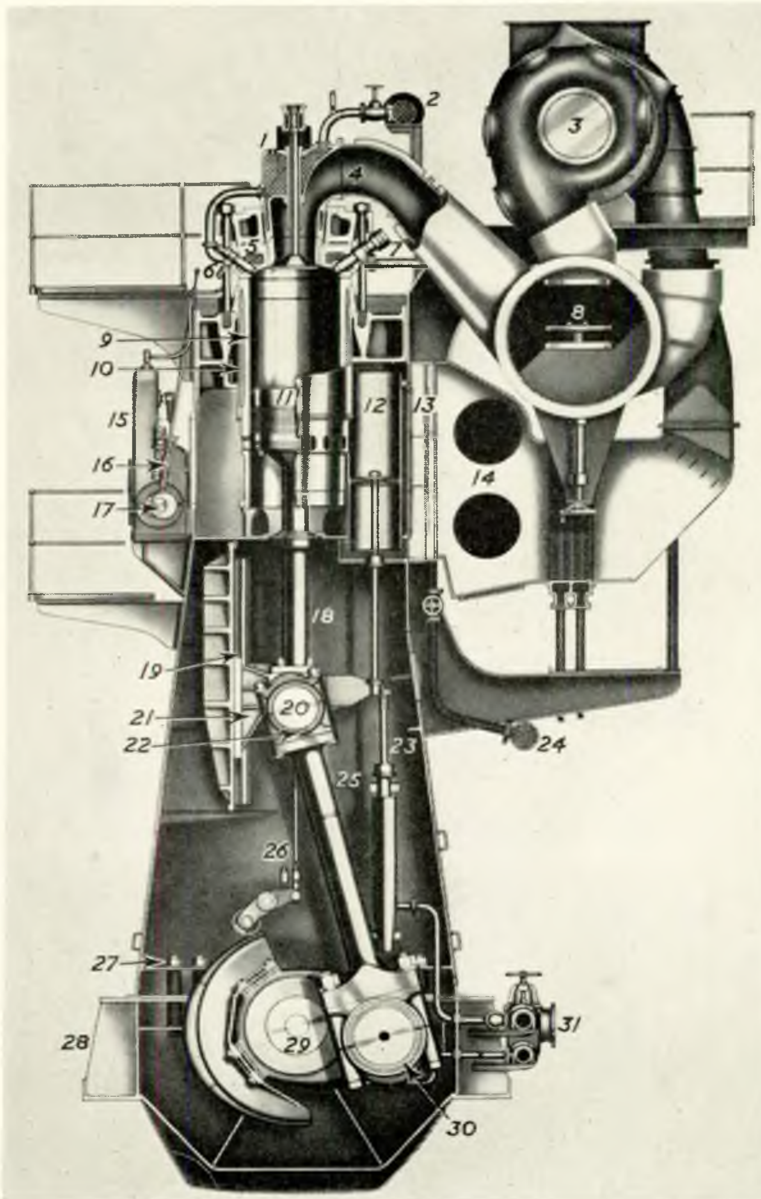


FIG. 4—*Cross section of type 760/1500 VGS-U*

1) Exhaust valve; 2) Cooling water outlet; 3) Turbocharger; 4) Exhaust pipe; 5) Cylinder cover; 6) Fuel injection valve; 7) Starting air valve; 8) Bypass valve; 9) Cylinder liner; 10) Cooling water jacket; 11) Working piston; 12) Scavenging air pumps; 13) Scavenging air valves; 14) Scavenging air receiver; 15) Fuel injection pump; 16) Indicator movement; 17) Camshaft; 18) Piston rod; 19) Crosshead guide; 20) Crosshead pin; 21) Crosshead shoe; 22) Crosshead bearing; 23) Telescopic pipe for piston cooling; 24) Cooling water inlet; 25) Connecting rod; 26) Pull rod for exhaust valve; 27) Entablature; 28) Bedplate; 29) Crankshaft; 30) Big end bearing; 31) Lubricating and cooling oil inlet

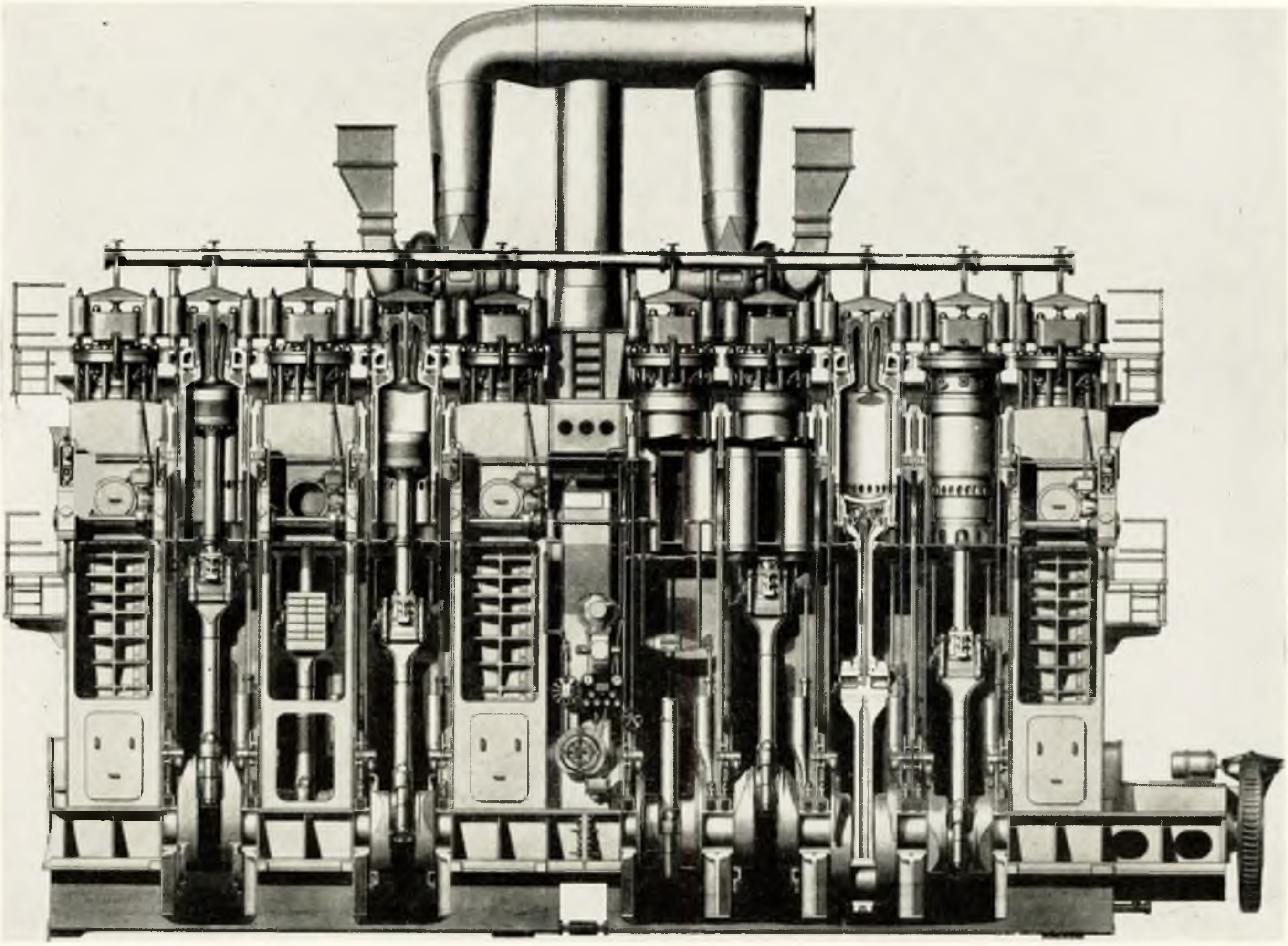


FIG. 5—*Longitudinal section of the engine*

Bedplate; Entablature

The bedplate is of all-welded steel (Fig. 6), divided into a suitable number of sections and assembled with fitted bolts. Each section consists of two longitudinal stiff girders with cross members of "I" section welded between them. The cross members are fitted with welded cast steel saddles which carry the main bearings.

Furthermore, the cross members are fitted with heavy bolt connexions which transmit the forces from the entablatures directly to the bedplate. The aft part of the bedplate carries the thrust bearing of Michell type.

A sheet steel oil tray, into which the lubricating oil drains from the various lubricating points of the engine, is welded to the underside of the bedplate.

The all-welded steel entablatures are designed as box type units, one for each cylinder (Fig. 7). A rigid steel casting forms the top of each entablature into which it is welded. This casting is provided with heavy fitted bolts, connecting all the entablatures, forming a longitudinal girder of great strength. Furthermore, it is fitted with long studs to hold down the cylinder cover. The upper part of the entablature forms the scavenging air receiver and extends down to the top of the crankcase, from which it is completely isolated by means of a horizontal plate carrying a piston rod stuffing box of the oil ring type. As the scavenging air receivers are interconnected, a common scavenging air manifold is formed.

When the welded entablatures were first introduced, cracks were experienced in certain cases on some of the engine types. It was considered that annealing of entablatures should be adopted in order that static stresses remaining after welding should not be superimposed on the existing basic stresses and thereby cause a reduction of the safety factor. It is also possible that static stresses may, after a time, give rise to deformations, causing the load distribution to alter, with consequent overloading of certain elements. As the early entablatures were made from ordinary ship plate the quality varied, and when cracks occurred it was thought that they might be caused by brittle material. A high tensile steel was therefore introduced, and this measure, combined with annealing, has proved successful. As appears from Fig. 8, no cracks have occurred since then.

The ahead and astern guide plates are of cast iron, provided with shims for the adjustment of guide clearance and secured to the entablature by means of fitted bolts.

Cylinder Liner; Cooling Water Jacket

The cylinder liner is of vanadium titanium alloyed cast iron with great wear resistance at high temperatures. The top of the liner is provided with a robust flange with ground faces and is sandwiched between the cylinder cover and the cooling water jacket. The lower part of the liner is provided with equally spaced scavenging ports round the periphery, which admit scavenging air when the piston is approaching bottom dead centre. The cylinder liner and cooling water jacket form a complete unit and as such are withdrawn together. (Fig. 9).

Cylinder Cover; Exhaust Gas Valve

The cylinder cover is in two parts and consists of a cast

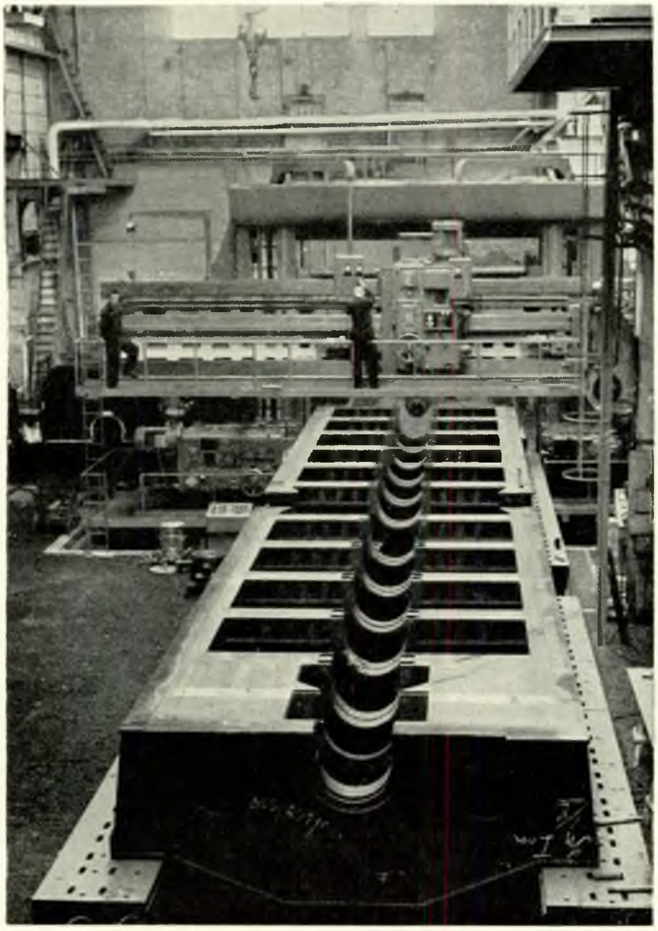


FIG. 6—*Welded bedplate in a combined milling and planing machine*

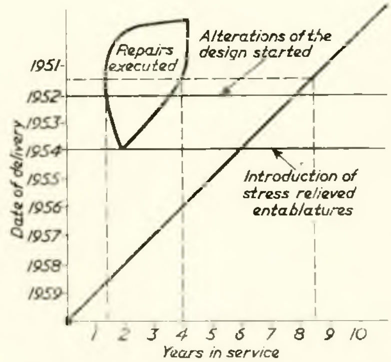


FIG. 8—*Diagram showing casualty and service experience with Götaverken welded engines*

iron water cooled lower part and a cast steel uncooled upper part. The lower part is the cylinder cover proper, which seals against the combustion chamber and contains all valves. The exhaust gas valve is located centrally in the cylinder cover, which ensures the most efficient scavenging of the cylinder and at the same time results in a symmetrical design of the cylinder cover. The other valves, fuel injection, starting and safety valves, are mounted on the periphery, the two fuel valves being placed diametrically opposite each other. The two parts of the cylinder cover are bolted together into one unit by four heavy studs, which at the same time serve as holding-down bolts for the exhaust gas valve.

The exhaust gas valve is designed and dimensioned to ensure the least possible flow resistance during the scavenging and exhaust period. The valve housing is of cast iron and water cooled. The lower end of the housing is provided with a replaceable valve seat of cast iron. The valve stem is of mild steel with a disc of heat resisting material. To facilitate the movement of the valve, the stem is spherically mounted in the cast steel yoke.

Crankshaft

The crankshaft is in two parts coupled together with fitted

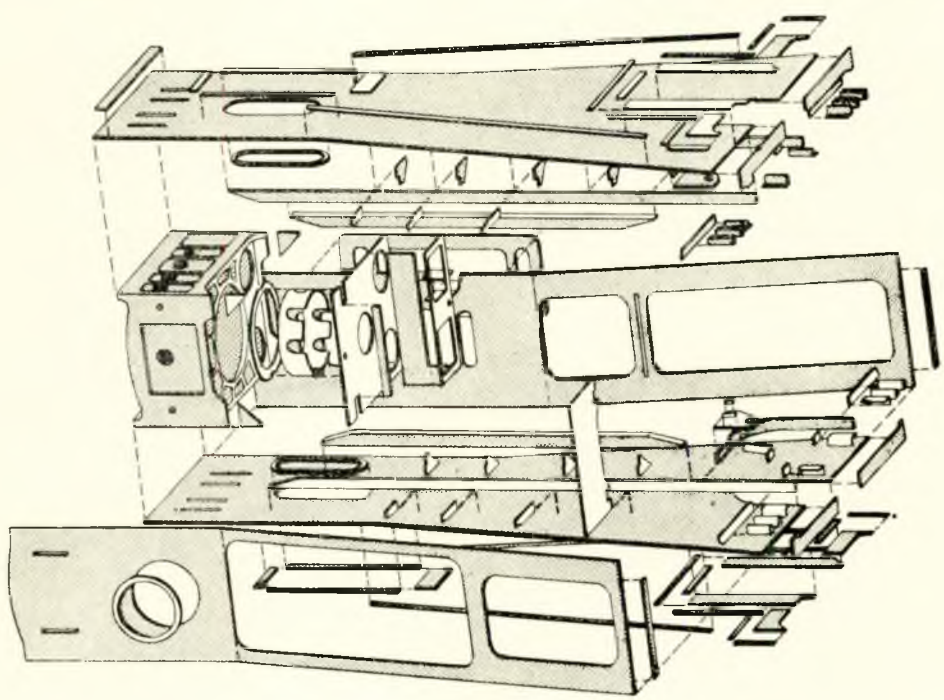


FIG. 7—*Exploded view of entablature in position for welding*
(This type of drawing is commonly used in the welding shops)

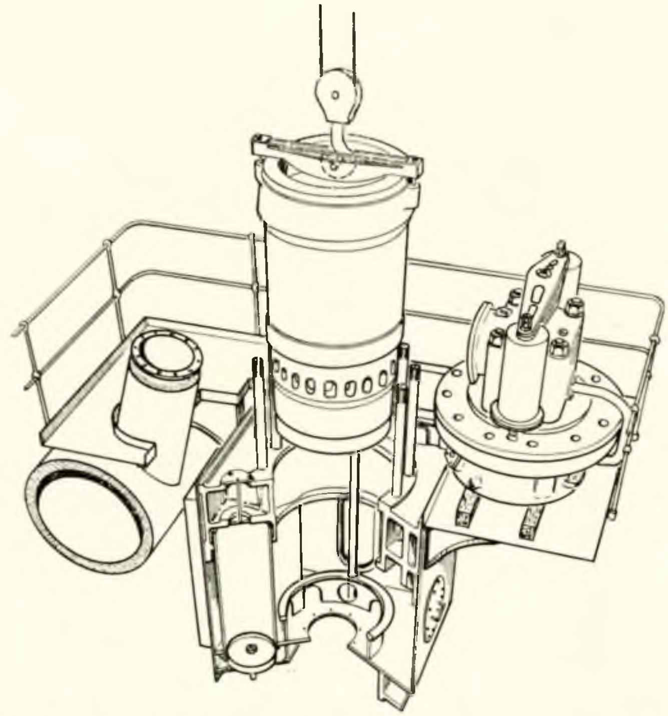


FIG. 9—*Dismantling arrangements for the cylinder cover and liner*

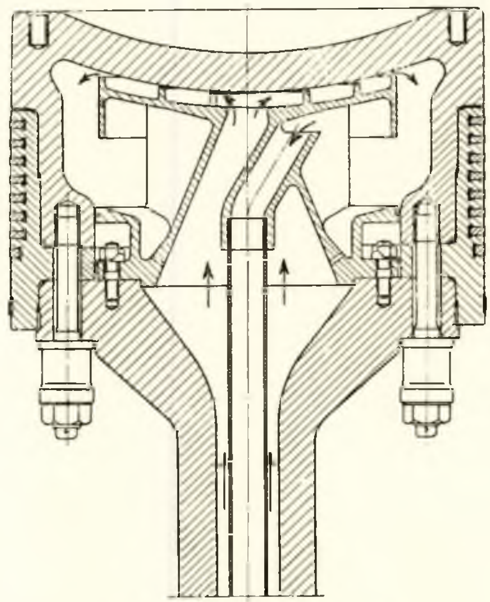


FIG. 11—*Working piston*

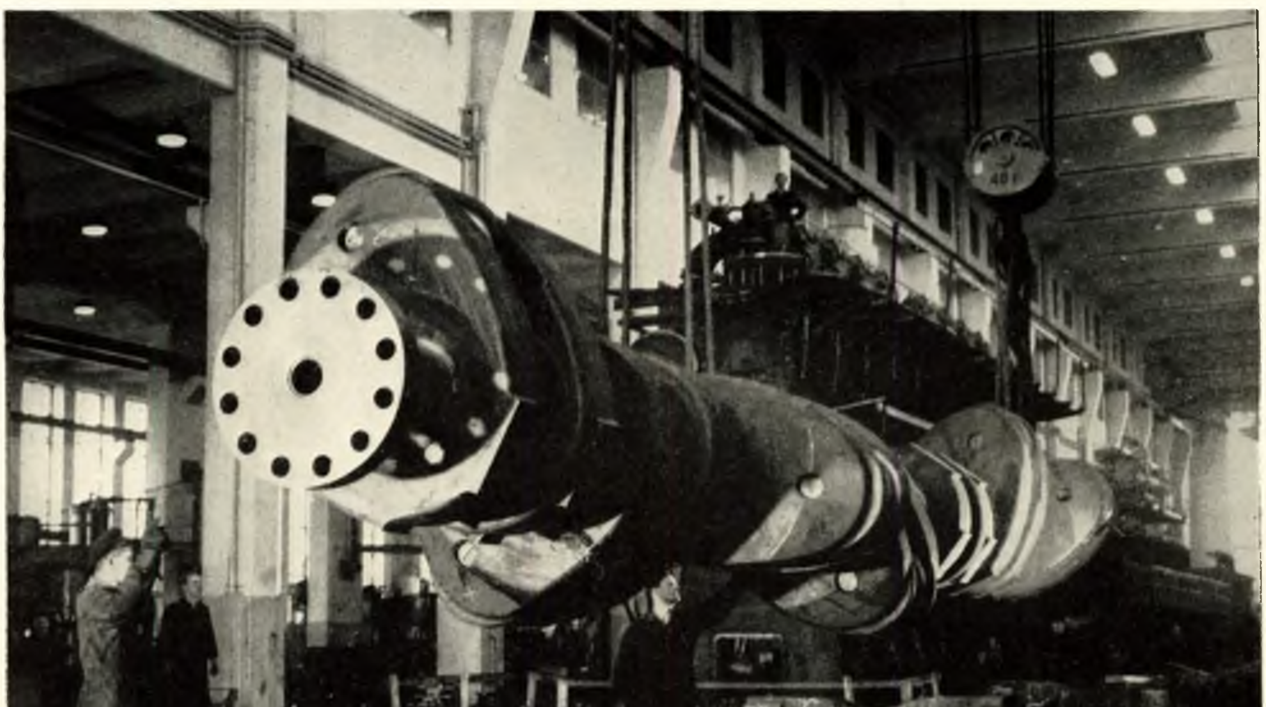


FIG. 10—*Crankshaft for a seven-cylinder engine*

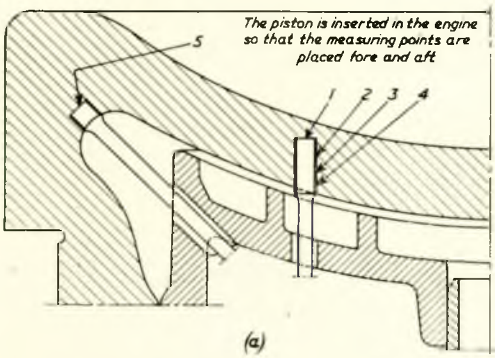
bolts (Fig. 10). At the aft end it is fitted with coupled bolts to the thrust shaft. The crankshaft is of the semi-built type, with crankpin and webs of cast open-hearth steel, into which the main bearing journals are shrunk. An ample radius is provided between each crankpin and web to reduce the concentration of stresses in the material. In order not to reduce the bearing length of the big end bearing, these radii are located in the webs. Each crankpin and main bearing journal is drilled to reduce weight, the drilled holes being used for the purpose of lubrication. Each crank web is fitted with a cast steel cam holder, to which cam segments for the operation of the exhaust valve are bolted.

Working Piston

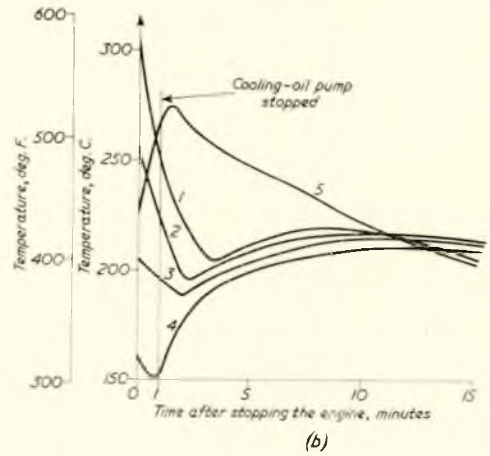
The structural part which is perhaps the most important piece when developing a Diesel engine towards higher power is the working piston (Fig. 11). Trouble was experienced in the beginning, cracks occurring in the piston crown. However, with improved fuel injection equipment, by adopting chromium molybdenum steel instead of regular carbon steel in the piston crown and by improving the piston cooling, the trouble was eliminated. In connexion with the introduction of turbocharging, detailed investigations of the running conditions of the piston have been made on an experimental engine.

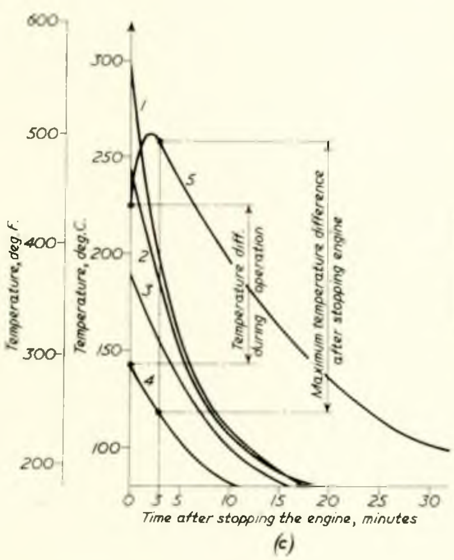
With the aid of thermocouples placed in the piston crown, the exhaust valve disc, the cylinder liner, etc., it has been possible to measure the temperature drop in the structure under different running conditions to an accuracy of about 2 deg. C. (3·6 deg. F.). These arrangements have provided hitherto unexpected opportunities to obtain reliable readings on the engine; for instance, when running with different rates of excess air, with different atomizers and fuel injection pumps, etc. This makes it possible to decide with great accuracy at which degree of turbocharge the same thermal load is obtained on the piston as at normal full load with a non-turbocharged engine.

Fig. 12(a) gives an example of the location of the thermocouples in a piston crown. The temperature is measured at 10-mm. intervals at the points 1, 2, 3, and 4, through the wall of the piston. The thermocouples are placed in the part of the piston where experience has shown that burning of the surface generally takes place. Thus measurements are taken on the part of the piston crown which carries the heaviest thermal loading. The element 5 has been placed in that part of the piston where coke formation generally occurs. Looking at the cooled surface at 5 and the heat absorbing surface in the same area it is clear that the cooling surface in relation to the heat absorbing surface is small compared with the points 1—4

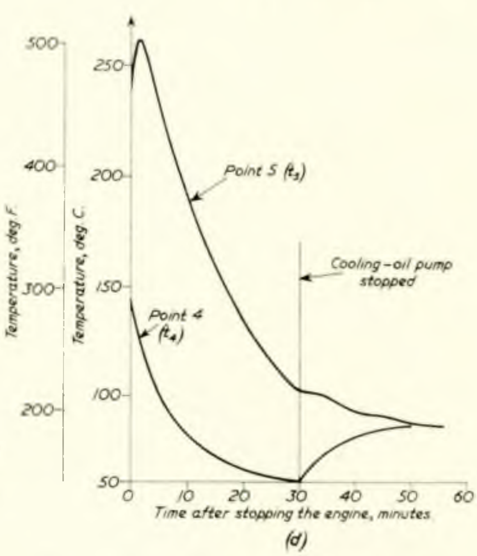


Location of measuring points in piston crown





Temperature curves for a piston without turbocharging



The cooling oil pump was stopped thirty minutes after stopping the main engine. Temperature curves for different points in the piston crown showing temperature variations when engine is stopped after having run on full load

FIG. 12(*a*)-(*d*)—*Measuring temperature in the piston* where the heat surface and the cooling surface are equally large. This must mean a rise in temperature at 5. Thus it can be seen what happens when the engine is stopped.

Fig. 12(b) shows what occurs when there is no cooling insert in the piston and the engine at full load is suddenly stopped. At measuring point 5, which was in the periphery, the temperature was about 225 deg. C. (437 deg. F.) when the engine was stopped. After about 2 min. this temperature had risen to 275 deg. C. (527 deg. F.). Measuring point 4 showed about 160 deg. C. (320 deg. F.), which soon rose to about 210 deg. C. (410 deg. F.). It can be seen from Fig. 12(b) how a higher temperature is obtained at measuring points 3 and 4 as the heat from measuring points 1 and 2 penetrates through the piston crown. After about 10 minutes the entire piston had an average temperature of about 210 deg. C. At this temperature the oil carbonizes on the cooling surfaces. If the engine is started at this point, cold oil is thrown up towards the overheated surfaces, and shrinking stresses, which can cause cracking in the piston crown, occur.

In Fig. 12(c) the temperature process in a piston with a cooling insert is seen. Also, in this case the temperature at measuring point 5 rises from about 225 deg. C. to 260 deg. C. (437 deg. F. to 500 deg. F.), which is slightly less than in the previous case. This is due to the fact that the cooling device cannot press the oil up into the moulding of the piston, so this remains dry. It can be seen how the temperatures at all points in the piston crown fall rather rapidly. After 20 minutes the temperature of the piston crown is well below 200 deg. C. (392 deg. F.). Thus, there is here no risk of carbonizing. In Fig. 12(d) mainly the same temperature process can be followed. In this figure measuring points 4 and 5 only have been inserted and these are situated on the inside of the piston. In this case the engine was very heavily loaded and then stopped suddenly. After the engine had been at a standstill for 30 minutes the oil pump was stopped. A quicker levelling of the temperatures occurs. The piston reaches an average temperature of about 85 deg. C. (185 deg. F.), neither is there any risk of carbonizing. If the engine had been run another 10 to 15 minutes on low load, much lower temperatures would have been obtained, and it would have been possible to prevent the high temperature obtained at measuring point 5.

The figures show that with an extremely good safety margin against carbonizing, the oil pumps can be stopped at any time 30 minutes after the engine has been stopped.

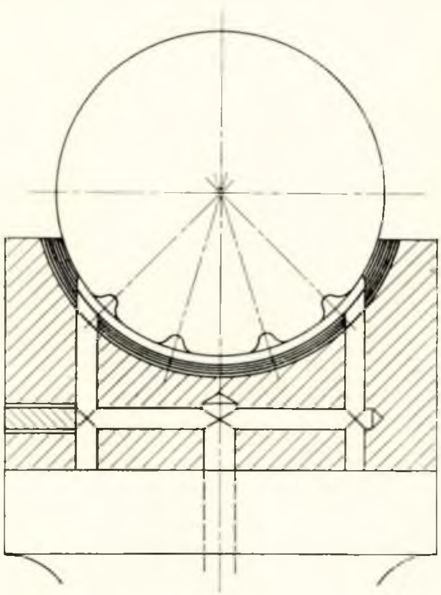


FIG. 13—*Crosshead bearings with oil grooves in the pin*

Crosshead; Crosshead Bearing (Fig. 13)

The crosshead is a steel casting, incorporating a white metal faced guide shoe, and is drilled for the passage of lubricating oil to the shoe. A cast steel bracket with two arms is bolted to the crosshead and carries the scavenging pump piston rods. The crosshead pin of special steel, hardened, carefully ground and polished, is drilled throughout its length to reduce weight and to provide a passage for the lubricating oil to and from the piston.

The crosshead bearing consists of one bottom half and two top halves of cast steel, white metal lined, provided with shims for adjusting the bearing clearance. The bottom half bearing extends for the full length of the crosshead pin, ensuring the largest possible bearing surface. Thorough investigation has been made on the bearing condition in view of the still increasing loadings.

In order to improve the oil distribution in the bearing, tests have been made with oil grooves in the crosshead pins instead of in the white metal. The tests showed that it was possible to reduce the bearing surface by 50 per cent without difficulty. As a further demonstration of the design's success, the firing pressure was increased by 20 per cent, the mean pressure was raised by 35 per cent and the lubricating oil pressure reduced from 2 kg./cm.^{2} (28lb./sq. in.) to 1 kg./cm.^{2} (14lb./sq. in.). Under these conditions the specific bearing pressure was 225 kg./cm.^{2} (3,200lb./sq. in.). The crosshead pin concerned was flame hardened and the bearing was lined with Götaverken standard white metal.

A bearing and pin made according to this idea but with a full bearing length has now been in use for more than 2 years and has proved so superior to the standard bearing that the shipowner has decided to fit bearings of the new design as soon as a change of bearings is required. This modified bearing design has increased the safety of operation so that the safety margin is considered to be quite sufficient to allow a higher degree of turbocharge.

Connecting Rod

The connecting rod is an open hearth steel forging, drilled throughout to reduce weight and to provide passage for lubricating oil to the crosshead bearings and guides. The top end of the connecting rod is flanged to seat the crosshead bearing, and the bottom end is arranged for connecting to the big end bearing.

Scavenging System; Exhaust Gas Valve Movement (Fig. 4)

Scavenging of the cylinder is on the uniflow principle, with scavenging air entering through the ports at the bottom of the liner and with the exhaust through the valve in the cylinder cover. The shape of the scavenging ports is such that the air is given a rotary movement during scavenging. This system achieves a good scavenging efficiency, the design resulting in a simple and symmetrical cylinder liner.

Scavenging air is supplied by double-acting reciprocating pumps incorporated in each entablature and driven by cast steel arms mounted on each crosshead. The suction and delivery valves are of simple design and are mounted, for easy access, at the back of each entablature at the middle grating level.

Each exhaust valve is operated by the cam segments mounted on each crank web, the movement being transmitted to the yoke into which the valve stem is fitted by means of rollers, levers and pull rods (Fig. 14).

Chain Transmission (Fig. 15)

The camshaft is driven from the crankshaft by means of a "Simplex" roller chain of special steel, the whole transmission being mounted in a special entablature situated between the fore and after cylinder groups.

The camshaft, which is mounted at the front of the engine, is fitted with symmetrical cams which are adjustable within limits to enable the fuel pump timing to be altered to suit the fuel being used.

The camshaft is driven by a dog clutch with a large clearance between the dogs, corresponding to the reversing angle of the camshaft. This arrangement permits the same cams to be used for ahead and astern. There is also an air operated lock which moves longitudinally on the camshaft and locks the camshaft and the driving sleeve of the same during normal operation of the engine.

Fuel Pump; Fuel Valve (Fig. 16)

The pump plunger is actuated from the fuel injection cam by means of a lever with a roller and a spherically mounted push rod. Each fuel pump barrel is provided with a special, hardened steel liner, which is a press fit and into which the plunger is lapped. A cylinder with a ground-in spring loaded piston acts as a shock absorber at the end of each stroke, when the high pressure oil is returned to the pump suction. The cylinder housing also incorporates the fuel oil supply to the pump. The fuel pump plunger is of special steel and is provided with two helices which communicate with an axial hole in the plunger opening into the pressure chamber of the barrel.

The quantity of fuel delivered by each pump is regulated by the relative position of the edges of the helices to the oil feed holes in the pump body and liner during the discharge stroke. This relative position is varied by rotating the fuel pump plunger until the required engine speed is obtained.

The fuel valves, two for each cylinder, are located diametrically opposite each other and are situated in the water cooled lower cylinder cover, and fed from a common fuel line. Each one is a spring loaded needle valve with a small lift, hydraulically controlled and oil cooled. All parts of the valve are of steel or special steel, accurately ground where necessary to ensure perfect sealing at the highest possible working pressure.

Manœuvring; Reversing System

The engine is controlled by means of a single hand wheel, by which starting, reversing and speed regulation are effected (Fig. 17). This wheel is mounted on a control shaft and situated at the front of the intermediate entablature at floor

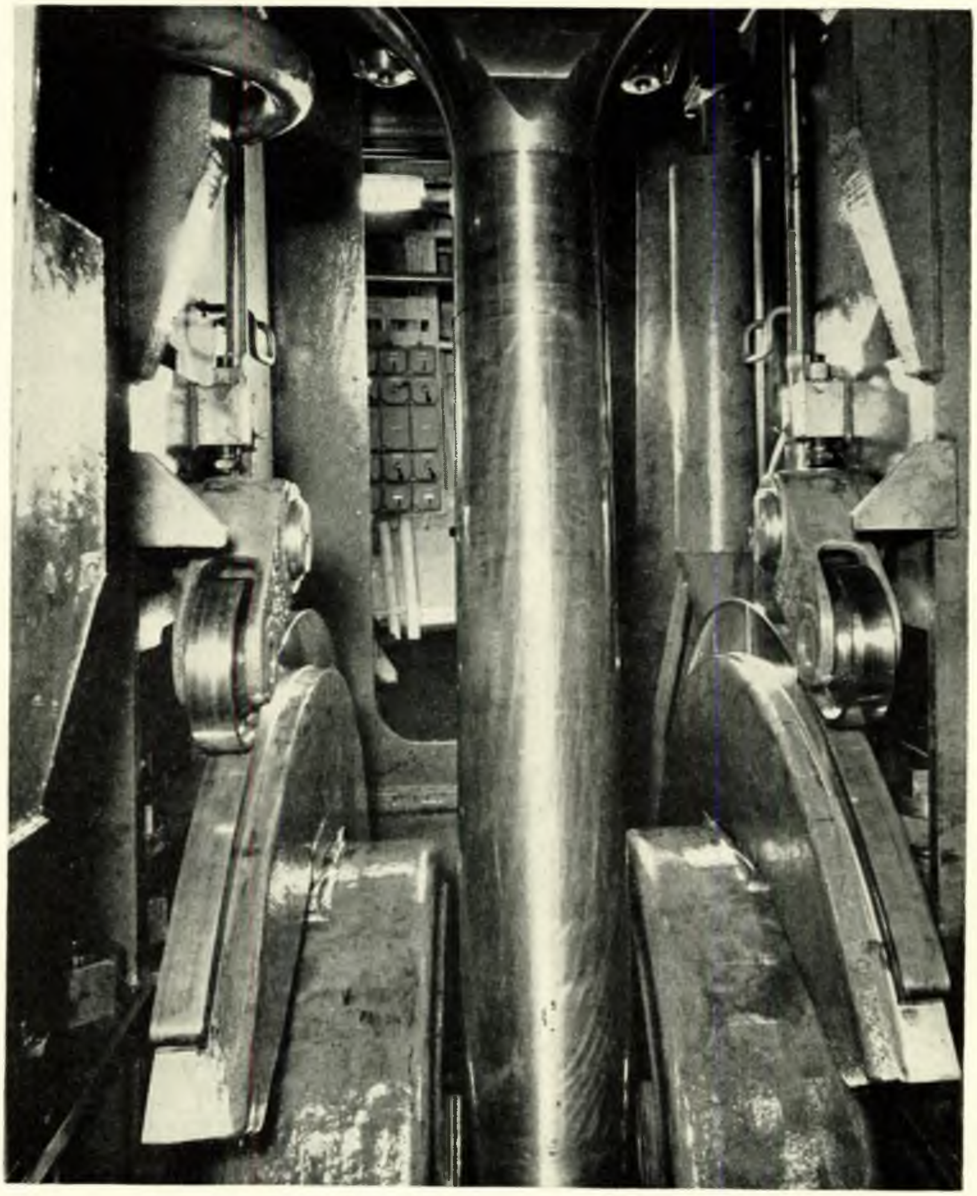


FIG. 14—*Interior view of crankcase, showing cam segments, rollers, levers and pull rods for exhaust valve*

plate level. The hand wheel is provided with a pointer which moves round a graduated back plate. The graduations are duplicated on each side of the "stop" position and indicate "ahead" and "astern". The graduations correspond to the rotary movement of the fuel pump plunger, by means of which speed regulation is obtained. An eccentric mounted on the control shaft transmits the movement of the control wheel to the fuel pump plungers by means of links and rods.

The whole of the control system, including the starting air slide, starting air distributor, starting air valves and the air operated lock on the camshaft, are operated by compressed air. The air is controlled by mechanically operated valves, actuated by cams mounted on the control shaft. When starting in either the ahead or astern direction, the control wheel is turned to the starting position indicated on the back plate, and immediately starting air is admitted to the starting slide, which, in turn, admits air to the operating side of the

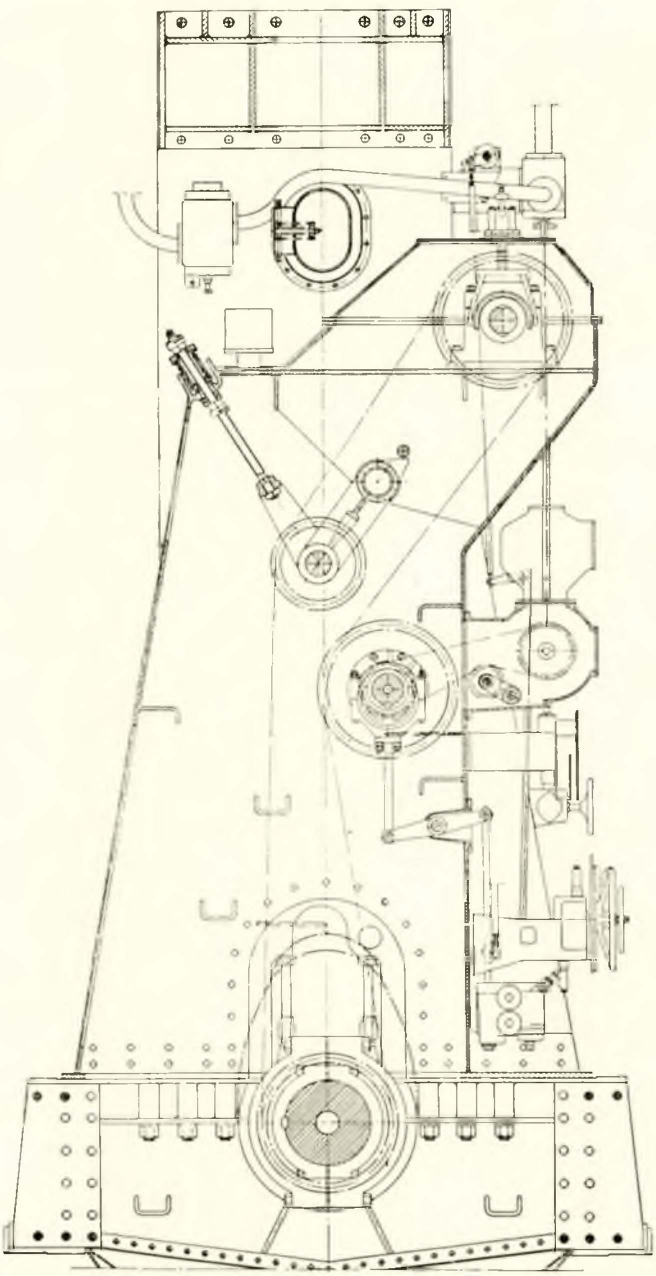


FIG. 15—*Chain transmission from crankshaft to camshaft*

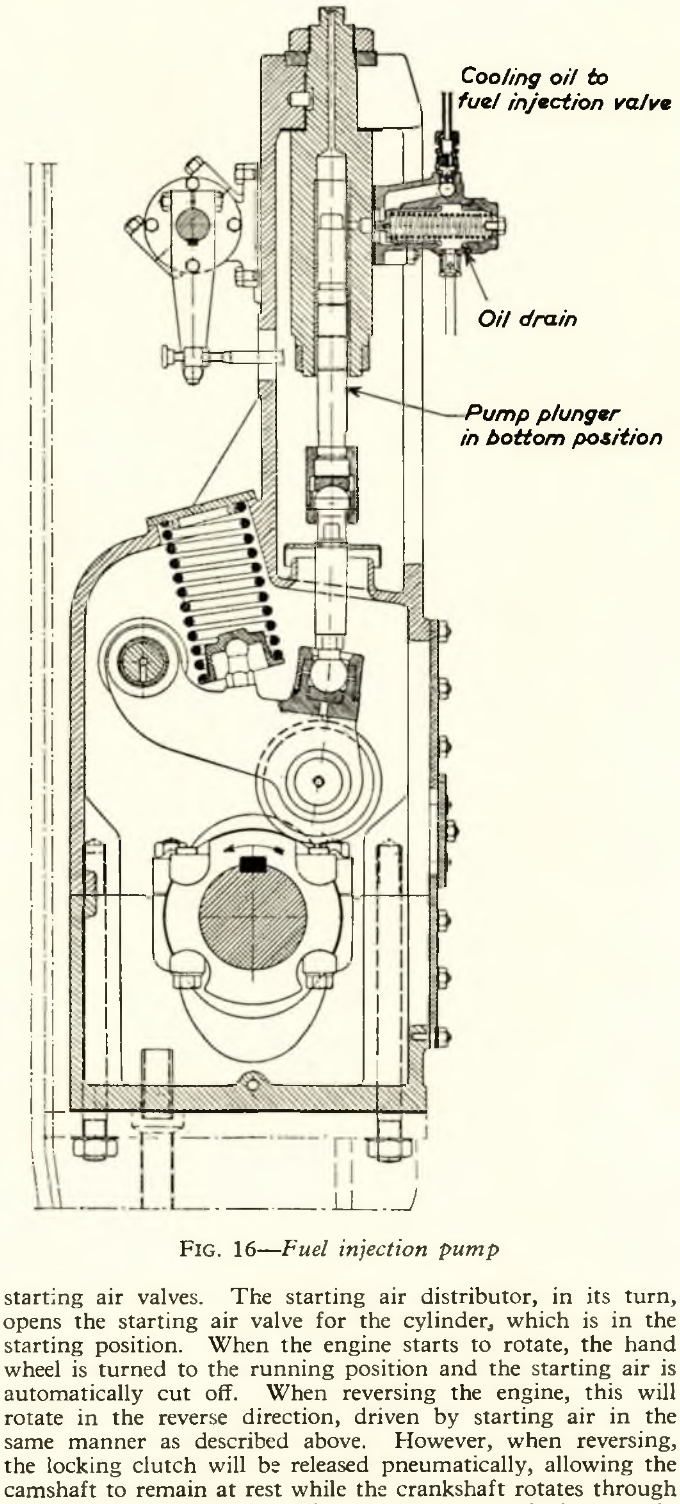


FIG. 16—*Fuel injection pump*

starting air valves. The starting air distributor, in its turn, opens the starting air valve for the cylinder, which is in the starting position. When the engine starts to rotate, the hand wheel is turned to the running position and the starting air is automatically cut off. When reversing the engine, this will rotate in the reverse direction, driven by starting air in the same manner as described above. However, when reversing, the locking clutch will be released pneumatically, allowing the camshaft to remain at rest while the crankshaft rotates through the reversing angle. In this way the camshaft is reset in relation to the crankshaft for operating the engine in the opposite direction. The cams for the fuel pumps will thus be brought into the correct position in relation to the new direction of rotation.

Lubricating Oil and Cooling Water Systems

Lubricating oil for the engine and for piston cooling is stored in two double-bottom tanks. The electrically driven lubricating oil pumps draw oil from these tanks and discharge through filters and coolers to the engine. Each pump is fitted

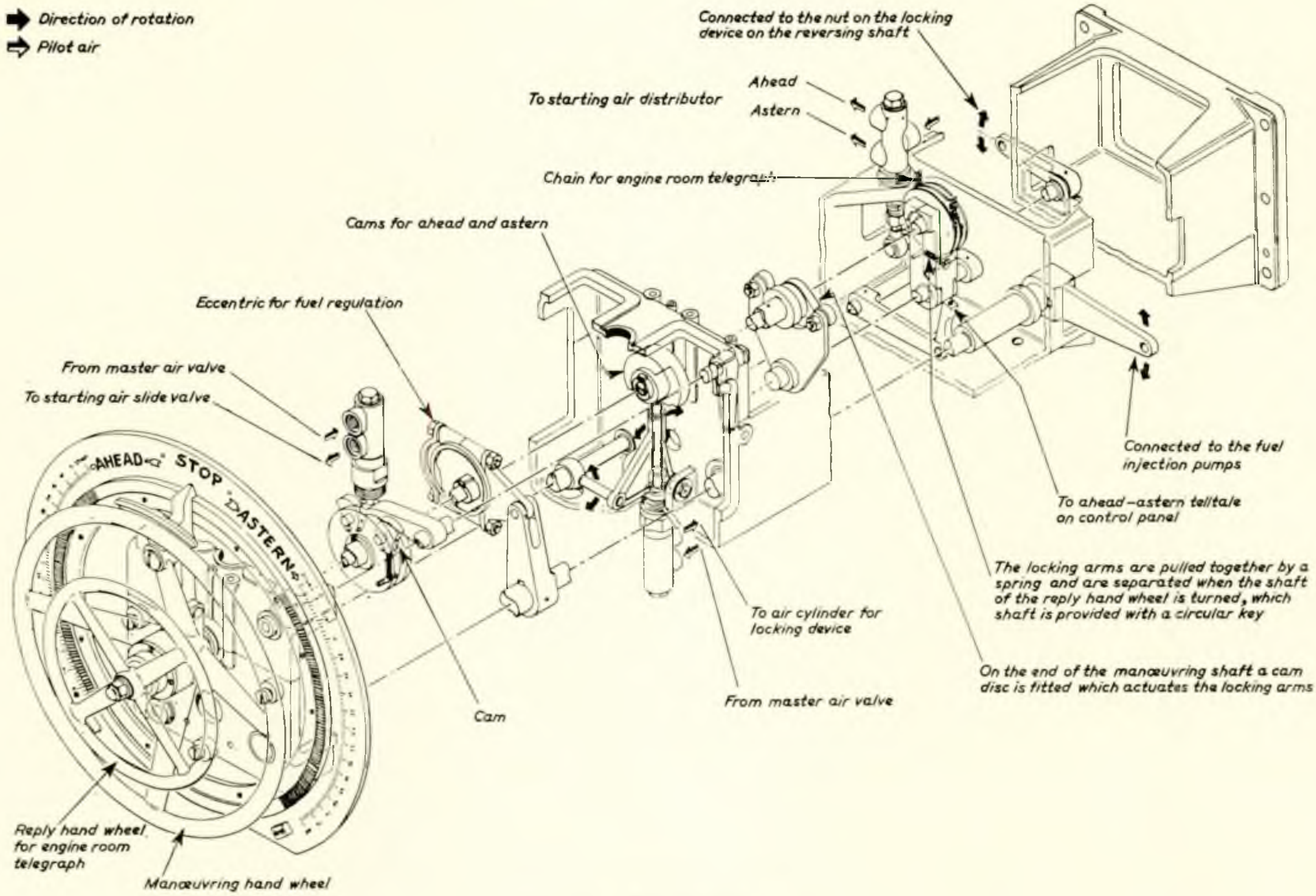


FIG. 17—*Manœuvring case*

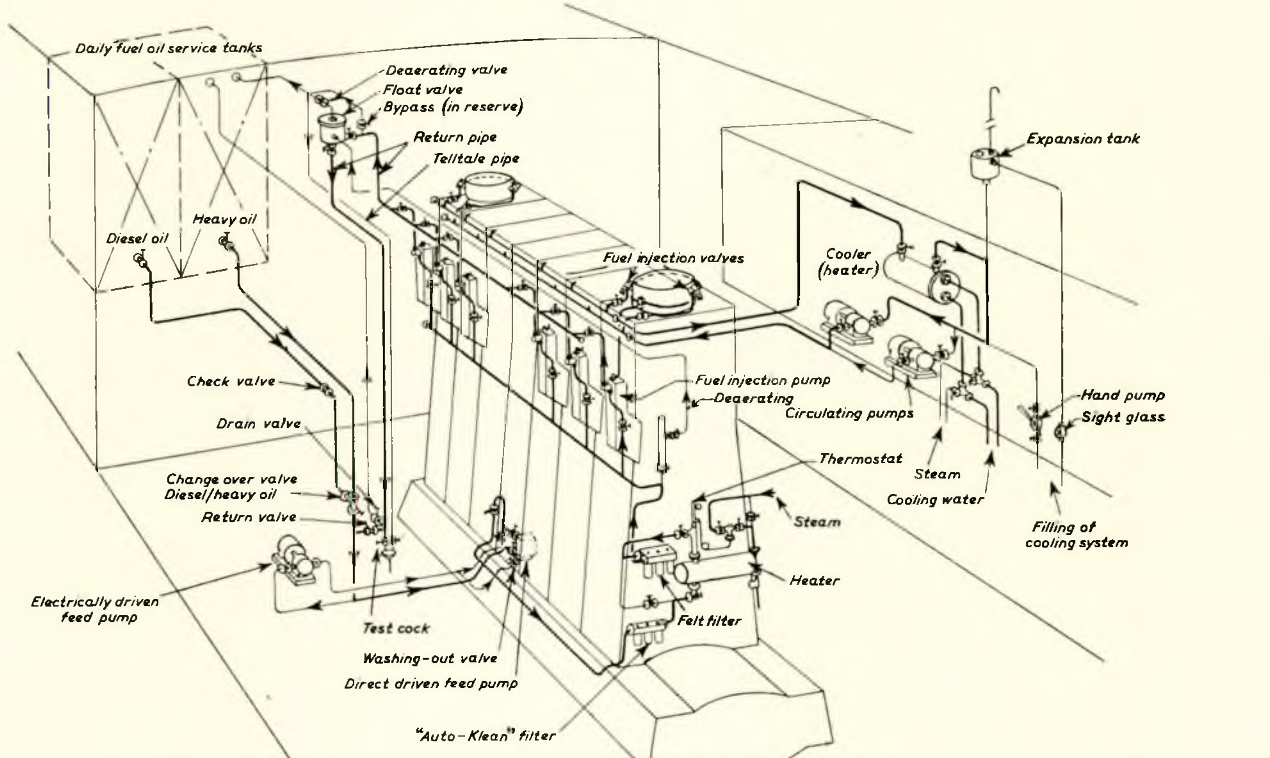


FIG. 18—*Arrangement for heavy oil operation*

with a spring loaded overflow valve by means of which the discharge pressure can be regulated within wide limits. It also acts as a safety valve.

The lubricating oil coolers are of normal multitubular design and are provided with bypasses for water and oil. Thus, alternative methods of temperature control are available.

The main engine is fresh water cooled on a closed system by means of electrically driven centrifugal pumps.

Fuel Oil System (Fig. 18)

Fuel oil is pumped by means of the transfer pump from the bunker tanks to the settling tanks, and from there is usually led to the fuel oil separators, which discharge to one of the daily service tanks. An engine driven pump is fed by gravity from the daily service tank and discharges the oil through filters to the fuel pumps. This pump is dimensioned so as to supply a certain excess of fuel oil, used for cooling the fuel valves, from where it is returned to the daily service tank. A hand pump is also provided for priming the fuel system.

For operating on heavy fuel oil the engine is provided with a separate pump and heat exchanger operating on a closed circuit for cooling the fuel valves.

The heavy fuel oil is stored in steam heated bunker tanks which, as a rule, are deep tanks, and is pumped by means of the transfer pump to the steam heated heavy oil settling tanks.

The oil is then fed through heaters and usually two separators in series to the steam heated heavy oil daily service tank, then by gravity either to the engine driven supply pump or to an electrically driven fuel pump.

The electrically driven supply pump also serves as a standby for the engine driven unit, or can be used for flushing and circulation of the main engine fuel system.

The pumps are overdimensioned to provide an abundant excess supply, which facilitates control of the oil temperature in the fuel system of the engine.

Discharge from the pumps is made through a multi-element "Auto Klean" filter to a steam heated heater fitted with thermostatic control and oil bypass connexions; thence through a fine felt element filter to the fuel injection pumps. The excess oil from the fuel injection pumps is fed back through a float valve, either to the suction side of the supply pump or to the daily service tank. The float valve automatically deaerates the system.

A spring loaded pressure valve fitted in the return line from the float valve ensures that the fuel injection pumps are correctly filled.

The cooling system for the injection valves is completely separate from the fuel system of the engine, and, for this reason, water can be used as a cooling medium if desired, but when running on medium heavy fuel oil ordinary Diesel oil should give sufficient cooling.

The cooling system consists of an electrically driven circulating pump and a standby unit, a heat exchanger and an expansion tank connected to the suction side of the pumps. The heat exchanger is cooled by sea water, but if necessary it can be heated by steam.

The cooling medium is circulated in series through the two fuel valves in each cylinder, through the heat exchanger back to the circulating pump.

Turbocharging System (Fig. 19)

For its turbocharged engines, Götaverken has adopted the constant pressure system. The regular scavenging system is retained, with the pumps connected in series to the blower. One or two exhaust gas turbines, each driving a compressor, increase the mean indicated pressure from 6·75 kg./cm.^{2} on a normally aspirated engine to 8·8 kg./cm.^{2} (96lb./sq. in. to 125lb./sq. in.).

The turboblower unit is made up of a single-stage centrifugal compressor and a single-stage axial flow turbine mounted on the same shaft.

The unit is cooled by fresh water connected parallel to the ordinary system of the Diesel engine.

From the compressor outlet of the turboblower unit (see Fig. 3) the compressed air is led through a welded pipe to an intermediate air cooler, which is cooled by sea water. For this reason, the ends and distribution baffles are made of corrosion proof non-ferrous alloy.

From the intermediate air cooler the air is discharged to the external scavenging air receiver, thence to the reciprocating scavenging pumps. The receiver is made up of a number of welded mild steel casings fitted to the scavenging valve frame of each engine entablature. The casings have inspection covers which enable the scavenging air valves to be inspected and are interconnected by means of pipes fitted with expansion joints. They are also fitted with automatic non-return valves of the same type as those used for the scavenging pumps. These valves remain closed as long as the pressure in the receiver is the same as, or higher than, that in the engine room. When the turboblower unit is disengaged, however, the reciprocating scavenging pumps draw air at atmospheric pressure both through these valves and through the unit at rest.

The dimensions of the exhaust gas receiver have been increased somewhat, as compared with those of the receiver on non-turbocharged engines. In addition, it is fitted with a bypass, centrally located at the intermediate entablature of the engine. This bypass is of welded mild steel fitted with a mushroom valve, consisting of four plates on a threaded steel spindle and a guide tube, between two seats. One of the seats is connected to the turbine inlet and the other directly to the exhaust manifold after the turbine. When the flow of gas through the turbine is shut off, the compressor slows down the rotor which, in the course of a few minutes, comes to rest. The exhaust gas receiver can be divided into two parts by the bypass unit, which in that case is fitted with two bypass valves. As the scavenging belt and the external scavenging air receiver also are divided at the intermediate entablature, the two halves of the engine can operate as separate units, one operating with turbocharge and one without. The separate controls for the fuel injection system under these conditions is provided for by means of a divided control shaft.

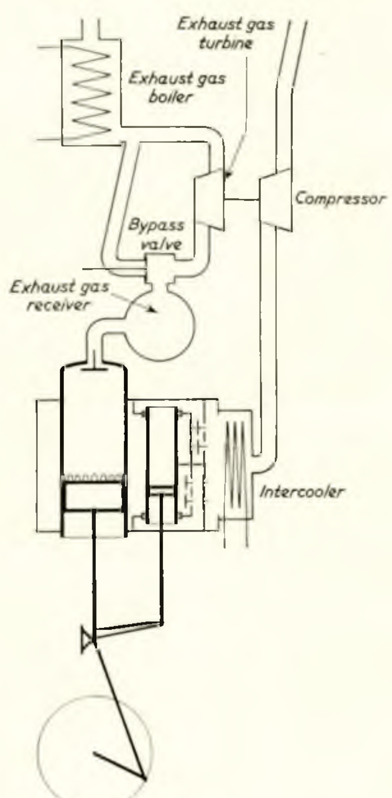
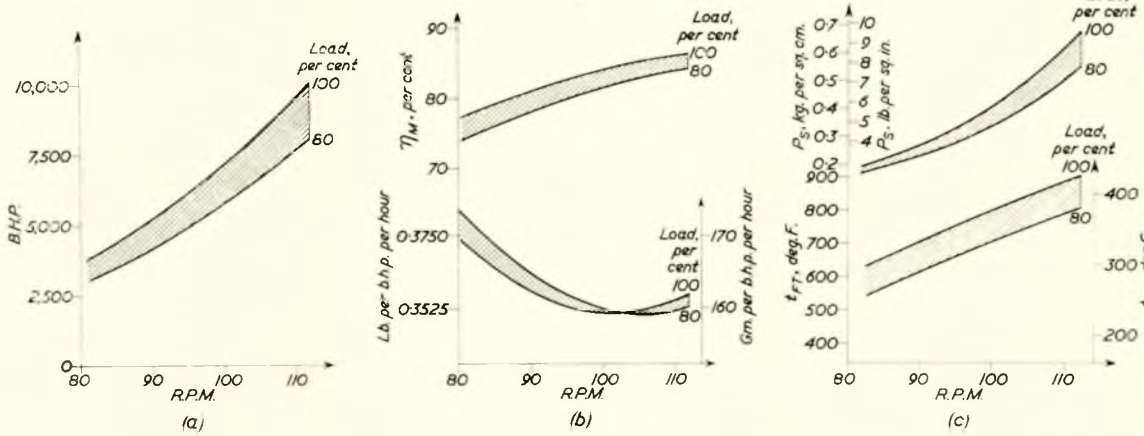


FIG. 19—*The Götaverken constant pressure turbocharging system*



a) Power in b.h.p. at different loads in relation to the r.p.m. for a 760/1500 VGS 8U engine

b) Mechanical efficiency and fuel consumption at different loads and r.p.m.

c) Scavenging air pressure P_S and temperature before turbine t_{FT} at different loads and r.p.m. Temperature after air cooler 30 deg. C. (86 deg. F.)

FIG. 20

With the constant pressure system there is no difficulty when increasing the rate of turbocharge to control the process so that the exhaust gas temperatures are maintained, or even lowered (Fig. 20). Exhaust gas temperatures measured in the exhaust bends remain below 350 deg. C. (665 deg. F.) at 112 r.p.m. This leaves a satisfactory safety margin for the valves. Accordingly, the thermal load on the engine is not increased by turbocharging.

FURTHER DEVELOPMENTS

When a few years ago, due to the ever increasing size of ships, especially of tankers and ore carriers, it became more and more desirable to reach propeller outputs exceeding 15,000 b.h.p., and as most of the orders received specified Diesel propulsion, a thorough investigation was made as to how higher outputs could be achieved.

In many cases when twin-screw arrangements were decided upon, the existing engine types were quite satisfactory. A large number of twin-screw tankers have been delivered, or are being built, with outputs up to 20,000 b.h.p.

With single-screw direct propulsion, an engine with larger cylinder bore and stroke than had been manufactured hitherto by Götaverken would be needed. Before the design work on such an engine was begun, an investigation was made to find out whether or not a multiple engine installation would provide a better solution.

The following three systems were therefore studied:
1) Diesel electric propulsion.
2) Geared engines.
3) Power gas propulsion.

The company already had a certain amount of running experience with Götaverken machinery in all three systems.

Diesel Electric Propulsion

A highly turbocharged, two-stroke, opposed piston engine had been developed for the Swedish Navy. This engine, which was described* at the CIMAC Congress in Zürich in 1957, has the following main data:
Bore: 185 mm. (7·3in.)
Stroke: 2 × 230 mm. (2 × 9·06in.)
Number of cylinders: 10
Maximum output: at 1,000 r.p.m.
3,250 b.h.p. corresponding to a mean effective pressure of 11·83 kg./cm.^{2} (168lb./sq. in.)
Weight: 8,300 kg. (18,260lb.)

* Johannson, E., and Thulin, L. G. 1957. "A Highly Supercharged Two-stroke Lightweight Diesel Engine". CIMAC Congress, Zürich.

If the number of revolutions and the mean pressure were reduced to correspond to a continuous output of 2,500 b.h.p., it would be possible with 10 engines, each coupled to one generator, to feed an electric propulsion engine or about 22,000 b.h.p.

The Diesel generator units would be light enough to be lifted ashore from time to time for periodical overhaul. The overhaul as well as temporary repair work could also be done with one unit shut down.

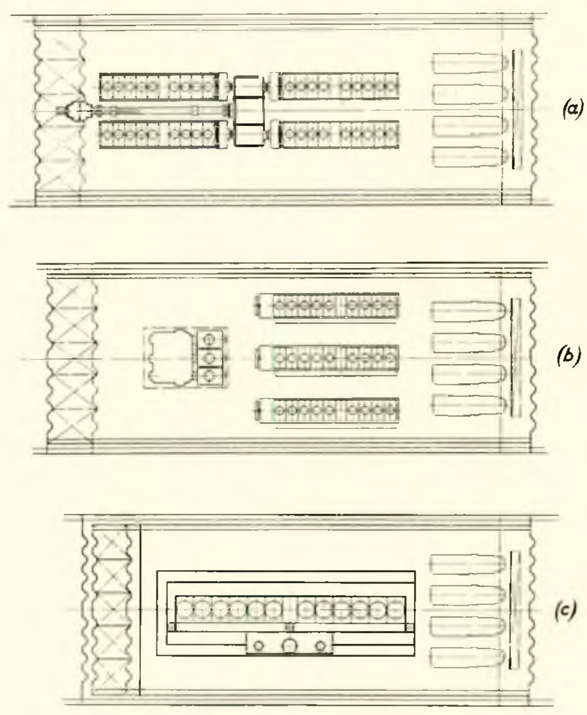


FIG. 21

a) Geared Diesel installation: four engines of 5,620 b.h.p. each at 200 r.p.m.; engine room length, about 22 m. (72ft.)

b) Power gas installation: three gasifiers of 9,090 gas h.p. each at 200 r.p.m.; engine room length, about 21·7 m. (70ft. 6in.)

c) Diesel engine, direct drive: 22,000 b.h.p. at 115 r.p.m.; engine room length, about 20 m. (65ft. 6in.)

As, however, in this case it was a question of designing standard machinery for transocean service, and running on heavy oil was a major condition, the project was rejected at an early stage since the difficulties in burning heavy oil were considered to be too great.

Geared Diesel Engines (Fig. 21(a))

Shortly after the war Götaverken delivered the passenger liner m.s. *Saga* for the Göteborg-London route. The machinery consisted of four eight-cylinder, four-stroke Diesel engines of the trunk type, each developing 1,675 b.h.p. at 270 r.p.m. with electromagnetic slip couplings and suitable gearing; an output on the propeller shaft of about 6,500 b.h.p. at 120 r.p.m. was obtained.

Apart from certain teething troubles, experience of this machinery arrangement was very good.

To obtain an output of 22,000 b.h.p. and have at the same time satisfactory conditions for running on heavy oil, the engine type 520/900 VGSU, with a higher number of revolutions in this instance, was considered, the machinery to have the following particulars:

4 Diesel engines
Bore: 520 mm. (20·5in.)
Stroke: 900 mm. (35·5in.)
Number of cylinders: 9
Shaft output: at 200 r.p.m., 5,620 s.h.p. each
Mean indicated pressure: 8·8 kg./cm.^{2} (125lb./sq. in.)
4 electro-magnetic slip couplings
1 reduction gear, propeller speed 115 r.p.m.
Propeller output: 21,800 s.h.p.

Power Gas Propulsion (Fig. 21(b))

Here the same engine type, 520/900 VGS, would be used as a gas generator, ("gasifier"), without turbocharger but equipped with the necessary additional compressors.

3 gasifiers
Bore: 520 mm.
Stroke: 900 mm.
Number of cylinders: 9
Exhaust gas output: at 200 r.p.m., 9,090 gas h.p. each
Gas pressure: 4·0 kg./cm.^{2} (57lb./sq. in.)
Gas temperature: 380 deg. C. (710 deg. F.)
1 gas turbine and reduction gear, propeller speed 115 r.p.m.
Propeller output: 21,600 s.h.p.

The power gas alternative was judged to be superior, as it gives a relatively free choice for the placing of the engines, i.e. the gasifiers, and offers a rather robust power transmission. The limited astern performance of the gas turbine, however, is a drawback, the alternative being to use a variable pitch propeller.

Comparing such power gas machinery with a large Diesel engine for direct coupling, the main advantages and disadvantages can be summed up as follows:

Power Gas Operation

Advantages

1) Smaller cylinder dimensions—more easily handled units and lower thermal stresses.
2) Gasifier units can be disconnected for repairs, especially during ballast trips.
3) Less weight.

Disadvantages

1) Great number of cylinder units: more time and attention needed for maintenance.
2) High number of revolutions with greater cylinder wear and difficulty in obtaining good combustion.
3) Limited astern performance of gas turbines.
4) Complicated regulators for manœuvring and for part load.

Direct Operation

Advantages

1) Fewer cylinder units—easier supervision.
2) Low number of revolutions—less cylinder wear; better combustion conditions.
3) Normal Diesel engine astern performance.
4) Simple manœuvring and regulating gear.

Disadvantages

1) Larger units—more difficult handling—higher thermal stresses.
2) Adjustment or repair of engine can only be performed to a limited extent without having to stop the propeller.
3) Greater weight.

Regarding the disadvantages of direct operation mentioned here, the following can be said:

a) The larger and heavier units are of course more difficult to handle, but in the case of engine size 520/900 mechanical lifting gear are provided for the piston overhaul and many other operations. In view of this the disadvantages are limited to any difference in the rate of work and the risks involved.
b) With about 400 Götaverken engines now in operation the design has gradually been perfected and it has been possible to reduce the work on vital parts at sea to a minimum. On the other hand, one or more cylinders can easily be temporarily disconnected if necessary.
c) The greater weight of the machinery may reduce the pay load. On the other hand, as shown by Fig. 21, in the case under discussion the length of the engine room is reduced.

The considerations summarized above convinced the author's company that, when dealing with Diesel propulsion of ocean going ships, slow running direct coupled engines are the best solution also for higher outputs than those in service up to the present.

Accordingly, in the early part of 1957 it was decided that an engine with the necessary increase in cylinder output should be designed. The first engine of this type, designated 850/1700 VGSU, is now under construction and should be ready for service next year.

When laying out this new main engine type, the aim has been to cover in the first instance outputs up to approximately 22,000 b.h.p. The cylinder bore will be 850 mm. (33·5in.) and the stroke 1,700 mm. (67·0in.). Limitation of the bore to 850 mm. (33·5in.) will result in the saving of longitudinal space in the engine room. The dimensions of an 850/1700 VGS8U compared with a 760/1500 VGS12U, both having an output of 15,000 b.h.p., are indicated in Fig. 22. With the bigger bore and stroke the space in the casing is better utilized, which becomes even more apparent when compared with the multiple

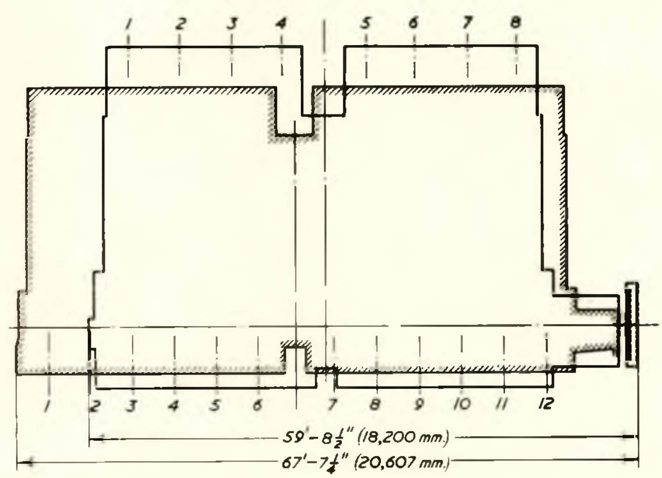


FIG. 22—*Comparison in size between 760/1500 VGS-12U and 850/1700-8U (both of 15,000 b.h.p.)*

engine alternative. With these dimensions and an indicated mean pressure of 8·8 kg./cm.^{2} (125lb./sq. in.), which has already been achieved in the present turbocharged engines, 22,000 b.h.p. will be obtained at 115 r.p.m. with a twelve-cylinder engine. In order to limit the propeller diameters at these large outputs, a speed of 115 r.p.m. has been chosen, which gives a mean piston speed of 6·5 m/sec. (21ft./sec.).

Like the previous types, the new engine will be a two-stroke, single-acting, crosshead engine, with uniflow scavenging and scavenging ports in the lower part of the cylinder liner, and one exhaust gas valve in the cylinder cover.

The constant pressure system for turbocharging has been retained, as it makes it possible, without extensive modifications, to increase further the rate of turbocharge.

When using the constant pressure system for turbocharging there must be some kind of scavenging pumps or fans. Each working cylinder is thus combined with a double-acting scavenging pump of piston type, which is actuated from an arm on the crosshead. An important advantage with such scavenging pumps is the fact that it is possible to operate the engine without turbocharge if there is a breakdown of the turbochargers. The reciprocating pump is simple, safe in service, cheap, and causes no manufacturing or servicing problems. The power required with these pumps is low compared with other air compressors.

The bedplate is of all-welded construction, with steel plates and steel castings. The frames are of cast iron and the engine is provided with stay bolts which transmit the gas forces from the cylinder to the bedplate. If the weight of a cast iron engine is compared with the weight of a welded engine, the great saving in weight is to be found in the welded bedplate (Fig. 23). For this reason the combination of welded bedplate and cast iron frames, which gives an engine of a comparatively low weight, has been decided upon.

At the time this engine was being planned, the classification societies had received a great number of reports on breakdowns of welded engines of various makes and types and were preparing new rules for the manufacture and design of welded elements, and the choice of cast iron frames was made with this in view. Due to the company's good experience of welded bedplates, in which not a single crack had been found, it was obvious that no objection would be made to the new design. On engines which have been in service during the last five years, with annealed entablatures of the present design, no cracks have been detected, for which reason the question of repair of cracks, especially in the transverse plates, was no longer of the same importance (Fig. 7). On the other hand it seemed to be inadvisable to design an entirely new welded engine without knowing the rules it would have to comply with as a finished product. The new classification rules for the design and manufacture of welded engines are now known, however, and do not stipulate that the modification of earlier design principles is necessary. An all-welded version of the 850/1700 will therefore follow.

The design with an upper cylinder cover of cast steel and a lower cover of cast iron is retained.

The cylinder liner and cooling water jacket are assembled into one unit before being fitted in the cylinder block, in the same way as in the company's present welded engines.

The crankshaft is made in two halves with crankwebs of cast steel, and the main bearing pins of forged steel are shrunk into the crankwebs in the same manner in which their main engine crankshafts are manufactured.

The calculations of the torsional vibrations of the crankshaft used to be a time-consuming task, and with orthodox methods the possible combinations on a great number of cylinders were too numerous to allow them all to be studied. The rapid development of electronic computers and the advantage of having an electronic digital computer installed in Gothenburg, however, have enabled Götaverken to make extensive torsional vibration calculations.

The computer in use is an ALWAC III E. It is a medium size computer equipped with both general storage and working

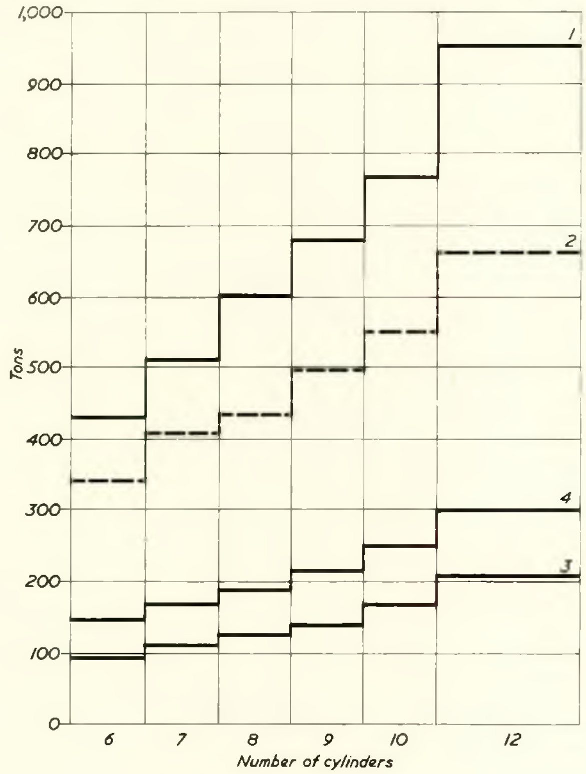


FIG. 23—*Engine type 850/1700*

Preliminary weights: 1) Complete engine, turbocharged; 2) Turbocharged engine type 760/1500 (for comparison); 3) Crankshaft; 4) Crankshaft and bedplate

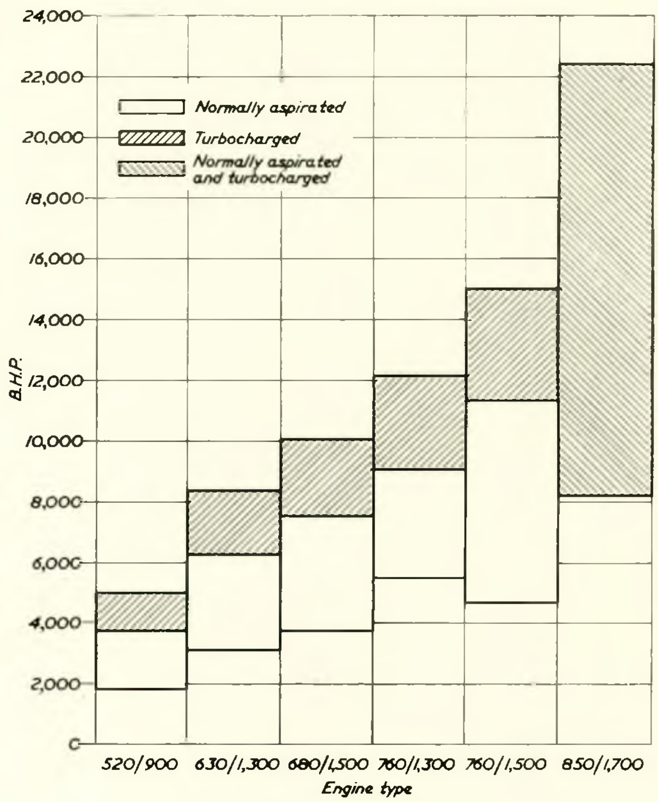


FIG. 24—*GV main engines: outputs*

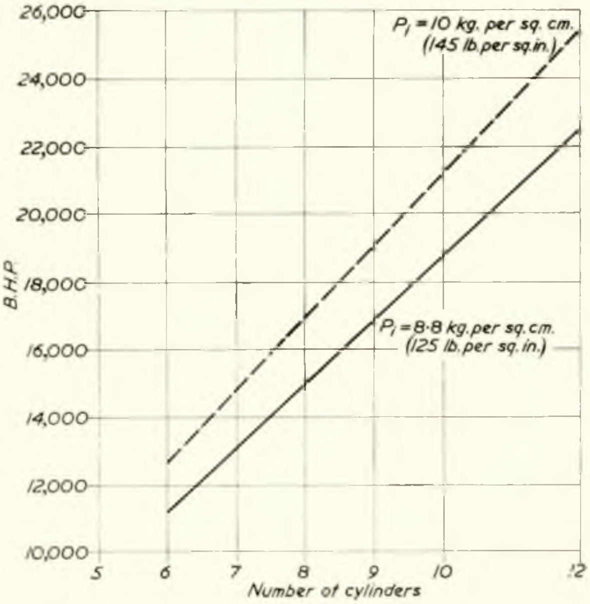


FIG. 25—*Engine type 850/1700 VGSU*

Cylinder bore, 850 mm. (33·5in.); revolutions, 115 r.p.m.; stroke, 1,700 mm. (67in.)

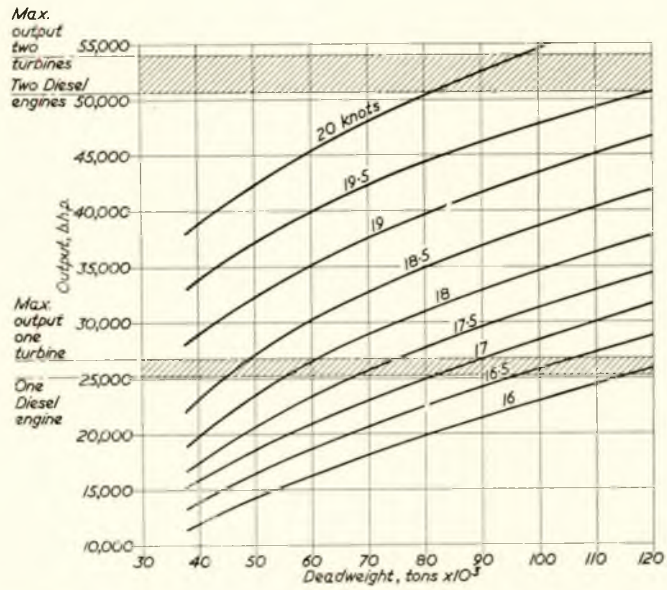


FIG. 26—*Engine power estimated for different speeds and sizes of ships*

storage. These memory circuits consist of rotating magnetic drums with a total capacity for data and instructions of 8,320 words, which corresponds to approximately 75,000 decimal digits. The computer is a binary machine. Input and output of data is possible by either flexowriter, punched tapes or Hollerith cards.

Besides using this electronic computer in torsional vibration calculations it is also being used for determining the natural frequency of the longitudinal vibrations in crankshafts and intermediate shafts. These calculations are very competently made by digit computers considering the great number of vibrating masses.

Because of the great possibilities of making extensive calculations with computers, a method of determining the natural frequencies of the rocking vibration of the engines is now being examined.

With the increased cylinder bore and stroke the piston will be subjected to a considerable increase in mechanical and temperature stresses. Bearing surfaces and other components in the reciprocating parts can be calculated and dimensioned with a fair degree of accuracy so that, in spite of the higher outputs, the specific stresses are kept well within the experienced values. The piston, however, is a complicated element, and even if it is feasible to calculate the particular mechanical stresses as well as heat flow, etc., there are too many factors involved to make it possible to reach an immediate and definite solution. The time factor especially is of great importance.

The piston material is an important factor in this connexion. Extremely satisfactory results have been attained with forged pistons of low alloy (wrought) chromium molybdenum steel. Fatigue cracks in this material, which is used for oil cooled pistons, have proved to be very rare. However, even after the selection of this material, alterations in the design may enable the piston to withstand stresses better. Different executions of the design are thus being studied.

When running on heavy oil the sealing arrangements between the scavenging air belt and the crankcase are extremely important as they must prevent acid combustion products from entering into the crankcase. The piston rod stuffing boxes are equipped with efficient scraper ring sets, each consisting of an upper set of scraper rings in the scavenging air belt with a drain pipe to the sludge tank, and a lower set with drainage out from the engine; these sets prevent oil from leaking between the scavenging air belt and the crankcase. The sets of scraper rings are separated by sealing rings which act as seals against leakage of air from the scavenging air belt. The pull rods for the valve mechanism pass through the scavenging air belt completely encased in tubes.

The further development of the type of Götaverken engine described here will naturally bring along still higher outputs (Fig. 24). Soon, however, a limit will be reached where—at the present numbers of revolutions—the propeller will be unable to absorb higher outputs without a considerable reduction in propulsive efficiency. This limit should be at 25,000—27,000 b.h.p.

For type 850/1700 VGSU, which is now under construction, the mean indicated pressure has so far been limited to 8·8 kg./cm.^{2} (125lb./sq. in.), giving an output of just over 22,000 b.h.p. on a twelve-cylinder engine. The engine, however, is dimensioned for a mean indicated pressure of over 10 kg./cm.^{2} (142lb./sq. in.) which means that later on, when sufficient operating experience has been obtained, the mean indicated pressure may be brought up to this level by increasing the amount of turbocharge. This should mean that an increase corresponding to 25,000—26,000 b.h.p. (see Fig. 25) would be obtained. The company's opinion is that the Götaverken engine with its uniflow scavenging design and the constant pressure turbocharge system offers great opportunities for such a development.

With the estimated tendencies in tanker development the propeller performances required should be obtained in accordance with the diagram in Fig. 26. From this diagram it can be seen that Diesel engine propulsion should be applicable for practically all tanker sizes.

Under the present conditions, when fuel cost is of more significance than it used to be, Diesel engine propulsion has shown its economic advantage. It is probable that with a return to a more normal shipping market situation the competition will remain keen, and Diesel engine propulsion will be of greater importance than it has been in the past.

Discussion

Rear Admiral W. G. Cowland, C.B., (ret.) (Member) said that some three years ago the Institute had held a symposium on "Recent Developments in Marine Diesels"* at which papers by the technical directors or chief engineers of Burmeister and Wain, Doxford, Sulzer and Werkspoor were read.

Since then, papers discussing foreign designed Diesel engines had been read by Dr. Pieri† on the Fiat engine and by van der Zijden‡ of Holland, the latter in connexion with wear in slow speed Diesel engine liners. He mentioned this because he wanted it to be noted that the Institute was largely international; unlike the American societies they did not call themselves a national society. They were proud of the large number of foreign members on the members' list and therefore welcomed the present contribution from Mr. Linden of Sweden.

He had a feeling, very broadly speaking, that engineers in this country were not until recently well informed on engineering in Sweden. He had been visiting Sweden for about eight years and was always impressed by whatever field of engineering he encountered there. His own personal experience was that whenever he went to Sweden he found a desire to do business and get down to technical brass tacks in a spirit of give and take based on mutual confidence. He did not suggest that the Swedes were easy; the British would not like it if they were, but they, like us, wanted to get on with the job without allowing irrelevancies to creep in.

Götaverken were progressive. They had been responsible for several advanced designs, in particular the gasifier and gas turbine mentioned in the paper. They also designed and built for the Swedish Navy some high speed, very light weight turbocharged opposed piston Diesel engines of over 3,000 b.h.p. In his view that was a very considerable achievement for a predominantly marine engine firm. He said that because the organization and outlook as regards design and development of those engines, especially as regards the expenditure required, must, in his opinion, approach more nearly that of an aircraft engine than a marine engine firm.

He much appreciated the honour of being asked to open the discussion on Mr. Linden's paper, especially as he felt that those engines should be as well known in this country as were those of most other Continental designs. The author had just told him that two years ago there was not a single British ship fitted with a Götaverken engine; now there were in service about 20. So therefore the engine deserved to be better known in this country. They were a very interesting range of engines, giving powers for the 520-mm. bore engine at 160 r.p.m. of 1,850 b.h.p. to 15,000 b.h.p. in the twelve-cylinder version of the 760-bore engine. He also noted that they were proposing up to 22,000 b.h.p. in 12 cylinders of their new 850-bore engine at 115 r.p.m.

The detail design had many interesting features, some of which he would suggest were not too easy to follow from the paper, e.g. cylinder covers, crosshead design and fuel pump operation in particular. No doubt subsequent speakers would elaborate that point and ask for further details on other points.

He had always felt that one of the best guides on the thermal stressing of an engine was a knowledge of the piston temperature, and he was very interested in the curves given in Fig. 12 and would ask the author to elaborate on them.

The turbocharged versions of those engines were rated, he believed, at 8·8 kg./cm.^{2} and the unblown versions at 6·75 kg./cm.^{2}—a 33 per cent increase approximately in power. He would ask the author to give relevant temperatures at points 1 to 5 for pistons working with blown and unblown engines at their normal rated full power.

He noted that they had adopted the constant pressure system of turbocharging. The engine was through scavenge and would appear to lend itself to the pulse system with only minor modifications.

The author had said that the high wages in Sweden had forced them to reduce the man hours in manufacture and erection and he was therefore somewhat surprised that they had not adopted the pulse system and abolished the scavenge pumps with the consequent saving in cost and complication—at least on six, eight, nine and twelve-cylinder engines. He hoped the author would comment.

At one time he had thought the valve gear was unduly noisy, but he believed by using multi-springs that had been very considerably reduced, and possibly Mr. Linden would mention this point in his reply.

Mr. H. B. Siggers (Member) was glad to have had the opportunity of joining in the discussion because, not only was the author an old friend, but in the course of his duties as a surveyor to Lloyd's Register at Gothenburg from 1939 to 1945 he spent a great deal of time in the Götaverken shipyard and engine shops—an experience of which he had the most pleasant memories.

The first Götaverken two-stroke slow speed engines were fitted in two ships completed in 1939—one Norwegian and one Swedish. Both of these were still going strong, the latter having completed 20 years' service and the former rather less, having been laid up at Gothenburg for several years during the war after an unsuccessful attempt to break the German blockade and cross over to the United Kingdom.

The Swedish ship was classed with Lloyd's Register and as a matter of interest he had looked through all their reports on her machinery over these 20 years and found that, apart from the occasional renewal of worn liners and pistons, nothing of note appeared to have occurred—quite a good record for an engine of that age.

That, of course, was only one Götaverken engine out of approximately 180 in ships classed with Lloyd's Register, and therefore he pursued his researches by reference to their punched card system for recording defects covering the last seven years, and found that there had been surprisingly few troubles in this period, apart from the fractures in welded entablatures which were mentioned by the author. There were very few cracked liners, covers or pistons and he would think

* "Recent Developments in Marine Diesels". 1956. Trans. I.Mar.E., Vol. 68, p. 365.

† De Pieri, R. 1959. "Recent Developments in Italian Marine Diesel Engines". Trans.I.Mar.E., Vol. 71, p.1.

‡ van der Zijden, M. J., and Kelly, A. A. 1956. "Combating Cylinder Wear and Fouling in Large Low-speed Diesel Engines". Trans.I.Mar.E., Vol. 68, p. 272.

that the Götaverken engine would bear comparison with any other in regard to reliability and cost of maintenance.

The author mentioned that the early engines had a reciprocating scavenge pump at the forward end driven by the crankshaft and later this was abandoned in favour of using the underside of the working pistons as scavenge pumps, which in turn gave way to the present system of a separate pump for each cylinder driven direct from the crossheads. He had the impression that there was a fourth arrangement using the underside of the pistons, together with small auxiliary pumps attached to the crossheads.

The elimination of the large pump driven by the crankshaft obviously reduced the length of the engine by an appreciable amount, and he would like to ask the author whether, in addition to this advantage, the later systems gave better scavenging and fuel consumption as well.

He had read papers by the protagonists of loop scavenging which had proved, at least to their authors' satisfaction, that this was the best system, but it had always seemed to him that the uniflow system as used by Götaverken must be more efficient, and at the same time result in smaller temperature stresses at the lower part of the liner. Could the author say whether, in the early days of experiment, loop scavenging was considered for this engine and discarded in favour of the uniflow system?

He believed Götaverken were alone in employing cams bolted to the crankwebs to operate the exhaust valves through pull rods, a simple and efficient method which, so far as their records showed, had given practically no trouble. A few years ago he saw an engine running in the firm's experimental shop where four springs instead of two were used for closing the exhaust valves. They were fitted at the four corners of a rectangle and were understood to make for quieter and more efficient valve operation. Would the author tell them whether this was indeed the case and if this system was being adopted?

The first welded bedplate was made in 1940 for a ship called *Remmaren*—he remembered this because he had the pleasure of surveying it—but unfortunately she soon became a war loss, so little service experience was gained from her, but this type of bedplate had since proved itself in many other ships and he had never heard of one fracturing. He believed that two points had largely contributed to this success; first, the use of steel castings for the bearing pockets with webs so arranged as to permit of full strength butt welds between the pocket and the vertical web of the transverse girder and between the pocket and the radial stiffeners; and second, the fact that from the beginning all transverse girders had been stress relieved after welding.

As Mr. Lindén indicated in the paper, the story of the welded entablature had not been so happy, and many fractures were recorded in the early years, mainly at such points as attachment of guide plates, scavenge air openings and shelves for fuel pumps. Since the introduction of higher quality material and stress relief, which he believed was in 1954, only one case of fracture had been reported to Lloyd's Register.

He wondered, however, whether these improvements had really provided the complete solution to the problem. It was true that the last 5 years had been trouble free, but the life of a ship was at least 20 years, and he felt that the discontinuities and changes of section that could not be eliminated in a component of this kind, combined with the inherent flexibility in tension of a steel structure, might yet give rise to further fatigue cracks and that it would have been preferable to retain the through bolts of the original cast iron design and thus tie the cylinder, entablature and bedplate together as a rigid entity.

Götaverken were of course not the only makers to have had fractures in their welded entablatures, but it was significant that where tie bolts had been used there had been very little trouble.

He was told by an owners' superintendent engineer recently that Götaverken were now offering their engines with either welded or cast iron entablature. Would the author confirm this, and, if it were the case, tell them what were the views of owners regarding the merits of the two versions?

The design of cast iron cylinder cover used, held in position by what might be called a heavy cast steel "keep" carrying the exhaust valve, gave a simple casting free from excessive temperature stresses in service, and, as would be expected, cracked covers did not figure very much in their records.

In reading the section of the paper dealing with the latest version of the Götaverken engine, he was cheered to note that cast iron entablature with through bolts was being employed, but rather cast down on finding that this was only because, at the time when the engine was being designed, the classification societies' proposed requirements for welded entablatures were not known, and that an all-welded version would follow. If this meant the abandonment of the through bolts it would be a mistake in his opinion.

From Figs. 21 and 22 it appeared that the twelve-cylinder engine would weigh about 1,000 tons, have a length of about 90ft. and height of nearly 40ft., with a crankshaft weighing over 200 tons. Would the author tell them if his firm had made any investigations into the longitudinal deflexion of the bedplates of very long and heavy engines, in heavy weather? It would seem that this might be an important factor when designing the engine seating, since, if bedplate deflexions were excessive, trouble with the crankshaft could be expected sooner or later. Another point—with an engine of this size fitted aft, would the width of the tank top at this position be sufficient to give adequate athwartship stiffness to the seating or was it considered that it might be desirable to stay the engine to the ship's side to prevent it rocking transversely?

Mr. P. Jackson, M.Sc.(Eng.) (Member) said he had read the paper with a great deal of interest and he agreed with Admiral Cowland that it formed a very useful addition to the papers which had already been written on various types of heavy marine engines. There had been the paper on the Harland and Wolff engine by Mr. C. C. Pounder*, the paper by Dr. De Pieri on the Fiat engine, and now the present paper on the Götaverken engine, and he hoped that there would be one on the new Doxford engine next year.

This was a very interesting paper and in the early part Mr. Lindén had outlined the ideas behind the introduction of the Götaverken engine. Through scavenge and air swirl were regarded as essentials and a large exhaust valve was necessary to permit through scavenge without undue resistance. Such features on a single-piston engine gave a very compact design with good combustion and one capable of development to higher powers, and, as Mr. Lindén had said, the mean pressures developed by turbocharged engines would doubtless increase in the future. Mr. Jackson stated that he had had the advantage of visiting the Götaverken works and seeing the manufacture of the engine, and he believed that the aims which were envisaged in the early days, of an engine capable of efficient manufacture, had been realized.

However, he would like to put certain questions to the author. In the first place, one that had been partially mentioned by a previous speaker: why, when there was a camshaft on the engine for operating the fuel pumps and lubricators, had Götaverken adopted a design of a camtrack on the periphery of the balance weights for operating the exhaust valve? The roller running on this camtrack would have to run at a very high peripheral speed, and during the portion of the revolution when it was not on the face of the cam it would lose speed and then be accelerated again very rapidly when struck by the cam. He would have expected such operation to lead to scoring and uneven wear of the cam and camtrack, even if they were made of the very best nickel case hardening steel and not only when the rollers were incorrectly hardened, as the author had mentioned. In Mr. Jackson's opinion, it

* Pounder, C. C. 1957. "The Harland and Wolff Pressure Charged Two-stroke Single-acting Engine". Trans.I.Mar.E., Vol. 69, p. 161.

would have paid to operate the exhaust valve from the camshaft.

He also supported Mr. Siggers in suggesting that through bolts were an essential with welded framing on a large engine, in order to relieve the framing and the welding of the task of carrying the combustion loads. This did not occur on an opposed piston engine and Doxford had never had any trouble with entablatures, though they had encountered some trouble on large engines with the welding of the bedplates. He would like to know what the author meant by his statement that they had adopted "high tensile steel" for the plates of their welded entablatures; was it a steel similar to P.5 or P.6? that is, boiler quality welding steel. He had always understood that high tensile steel, particularly any having a high manganese content, was not suitable for welding. With regard to welding defects in bedplates, he believed that this could be associated with the remarks made by the previous speaker, in that such troubles were caused by the different deflexions of the ship between the loaded and unloaded condition. Such deflexions could cause quite heavy stresses and Doxford and other large engine builders were investigating these deflexions and methods of securing true alignment of the engines at the present time.

The investigations which the author had made into piston head temperatures were very interesting and, he noted that Götaverken had adopted a chromium molybdenum steel, which was presumably a semi-heat resisting steel for piston heads. Doxford had progressed rather the other way, in adopting a very ductile steel that would yield to expansion under high temperatures and they had not had any difficulty with piston heads for several years; and they had far more engines in service than had been mentioned by Mr. Lindén.

The crosshead construction adopted by Götaverken was very interesting. He himself would not have thought that oil grooves in the crosshead pins would have been any better than grooves in the bearing. However, the proof of the pudding was in the eating and the author had given figures to support his case. It would be of interest to know, however, the ideas behind the evolution of oil grooves in the crosshead pin.

He had studied the design of the scavenge pump outlined by the author and considered that it would not be a very efficient pump with the suction and delivery valves mounted on the back as described, since the clearance volumes at the end of the pump stroke were very large. Perhaps Mr. Lindén could state the volumetric efficiency of two such pumps, though he had said that on the new engine there was to be one pump only. On their turbocharged engine the Götaverken company had adopted the constant pressure system of turbocharging. Would the author outline why this had been done, because the fuel consumption with such an arrangement could not be so good as with the impulse turbocharging system, and it would always be necessary to employ scavenge pumps which absorbed power and which were an extra complication. The exhaust temperatures given by Mr. Lindén for their supercharged engines were very low and he would like to know whether they had encountered any difficulties with regard to the raising of steam from the exhaust gases at such a low temperature.

He would like to ask a question concerning the starting air distributor of the Götaverken engine. There was originally a rotary air distributor driven from the camshaft very similar to the one used by Doxford, but when he visited Götaverken some three years ago they were trying out a valve type of distributor as an alternative to the rotary distributor and it would be interesting to know which of the two designs had ultimately been selected.

Finally, in the section that the author had shown of the new large engine, there was a cooling coil in the air space around the cylinder barrel or along the side of the entablature and he wondered what was the purpose of this coil.

Mr. S. Archer, M.Sc. (Member) said that as one who had sailed in a Götaverken ship he had formed a high regard for their technical excellence with regard to marine engineering. It was as long ago as 1916, so it was particularly interesting to see the tremendous progress which had been made since those early days. Perhaps he should make a slight correction there—the engine was *built* in 1916!

At any rate he had been particularly charmed by Mr. Lindén's engaging frankness and objectivity in presenting the paper. It was not always that one encountered such outspoken honesty in technical discussions. He wished there could be more of it, because it was only by exchanging mutual experiences that progress would be achieved.

There were just one or two small points upon which he would like further information. First of all with regard to the specific fuel consumption of the power gasifier-turbine propulsion installations, it would be of interest to compare that with more recent developments in that field. He would also like to know what the exhaust temperature would be, and the material of the gas ducting.

Fig. 7 appealed to him intensely as being a most instructive diagram. It was called an exploded view of the entablature in position for welding, and he would like to know how many casualties there were after the explosion! He hoped Mr. Lindén would forgive the joke, but it had occurred to him that it was rather an unusual diagram.

In these days a lot was heard about scavenge fires. He did not know whether, with the advent of boiler fuel, they had become more prevalent than in the old days, but he had recently had occasion to hear a number of differing points of view on the subject and he wondered whether Mr. Lindén, in his experience, had ever heard of a red hot crankcase top. It had been suggested that it could occur, and if so, it would be a very likely cause of a crankcase explosion. He hoped it was not true.

The design of the crosshead pins was indeed interesting. He noted that the bending was eliminated by having the pin continuously supported over its full length, which perhaps rather tempted one to go for materials which were not quite so strong in tension. He wondered whether Götaverken had carried out any experiments in that respect. He thought they had also had some experience with chromium plating of the crosshead pins and would like to know if they would recommend this method of treating the pins.

He noted that at the present time Götaverken were offering their latest engine in cast iron construction, but that they were prepared to offer it as an all-welded design also. If so, what would be the differential in weight per brake horse power? Figures on that would be of interest. He observed that Götaverken favoured the use of steel castings in their design and he thought that must be a challenge to foundrymen because it was not everywhere in the world that one could rely on sound steel castings, but perhaps Sweden was more fortunate than some countries in that respect. Could the author give some idea of the composition and physical properties of the steel castings used for these parts?

With regard to Fig. 12, he noticed that a great point was made of the improved temperature conditions after fitting the insert in the piston and the author had compared the temperature distribution between those two types of pistons, but in the original one the engine suffered from the disadvantage that the cooling oil pump was apparently cut off after one minute, but with the others, and certainly in (d) it was stopped only after 30 minutes. Was that a fair comparison? For a reasonable comparison one should surely stop the pump at the same time in both cases.

He could not refrain from a comment on the preceding speaker's remarks concerning the strengthening of ships by engine bedplates. He felt that was primarily a challenge to the naval architect, particularly with the heavier engines which were coming along. He wondered, in the same way that it was not possible to scale up an animal because it would then be too heavy to stand on its feet, whether something similar might not apply to those mammoth engines and that therefore there should be relatively a very much stiffer foundation on which to secure such engines. He thought that in those engines where the engine framing did not have to support the combustion forces, the temptation would be to make the bedplate somewhat lighter. Therefore if the seating was not correspondingly increased in stiffness, the engine bedplate would pay the penalty. This was a matter of much concern to all owners of such ships and the author's views would be interesting.

Mr. G. Yellowley (Member) said Mr. Lindén's paper added yet another valuable contribution to the series of papers in the Transactions describing the principal prime movers available to shipowners at the present time.

All the authors, including Mr. Lindén, had stated the guiding principles in designing their engines, and as was to be expected they were in agreement on the basic aims, namely low first cost, minimum weight and space, reliability, low maintenance charges and economy, but as Mr. Pounder had so clearly stated in his excellent paper before the Institute, the best all-round compromise had to be sought, and judging by the present paper Götaverken could be congratulated upon developing an engine in which that aim had been achieved.

He felt he should explain that he spoke with special knowledge of the Götaverken engine as he had visited their workshops and witnessed trials prior to the firm with which he was associated being granted an exclusive licence for building the engine in this country.

It would be noted that the author placed reliability in service as a prime consideration, and that, it would appear, had influenced the design of the turbocharged version of the Götaverken engine in that the scavenge pumps had been retained in conjunction with the constant pressure system of turbocharging with a means of quickly changing over to normal aspiration should a turboblower fail.

The penalty for that was a slight increase in fuel consumption which he understood was only of the order of 1 per cent, and possibly the author could corroborate that figure.

The advantages of increased reliability, simplicity of exhaust arrangements and the ability to limit the number of turboblowers to only two in engines of up to 15,000 b.h.p. outweighed that slight deficiency.

It was noted that Götaverken at one stage used the underside of the working piston as a scavenging pump, and it would be interesting to know why that method was not used in the turbocharged engine.

As the number of turbocharged engines now in service was considerable he would like to ask the author if the reliability of the turbocharger had been proved, and thus enable consideration to be given to the elimination of the exhaust turbine bypass valve, and thus further simplify the exhaust pipe arrangement.

As the description given by Mr. Lindén showed, the engine was a straightforward single piston engine with uniflow scavenge, built on short engine centres, thus giving a stiff and robust crankshaft. The bedplate and box type entablatures led to ease of installation and gave an engine structure of great strength.

Cylinders, liners and pistons were important items in regard to low maintenance costs, and it was noted that a very simple liner was employed, with a single row of fully machined scavenge ports.

Piston design had always been subject to modification and all engine builders had tried, and were still trying, to produce an optimum design that would satisfy the steadily increasing engine ratings.

The firm with which he was associated had also given some thought to piston design and could claim some success with a simple form of piston made wholly of Lanz Perlit cast iron and fitted in opposed piston two-cycle engines.

It was noted that piston temperatures had been measured under running conditions to an accuracy of about 2 deg. C., which was very good, and he was glad to have the results of tests carried out on an experimental engine. In order to compare with other published data on that subject he would like to ask if the author had recorded the temperatures near the top piston ring groove and also if he had any data regarding cylinder liner temperatures.

The subject of connecting rod top end bearings was always to the fore when engineers foregathered, and it was pleasing to note the success claimed by the author's company by employing oil grooves in the crosshead pin instead of the white metal. It would be of interest if Mr. Lindén could indicate the manner in which the 50 per cent reduction in bearing surface was obtained.

With regard to future developments it was noted that the investigation into the various machinery installations to give higher outputs resulted in favour of the slow running direct coupled engine, and that Götaverken were now developing an 850-mm. bore turbocharged engine which would ultimately be capable of developing 25,000 b.h.p.

Mr. G. Victory (Member) said that as he had read through the paper a number of queries had presented themselves: some of them were explained away at later stages of the paper and some remained. On the latter he wished to comment, mainly from the point of view of the operating engineer.

On page 197 Mr. Lindén said that his company's four-stroke auxiliary engines were still included as standard practice in their ships. He would suggest that as the mean effective pressure was about 70 on natural aspiration and 105 after supercharging, combined with a piston speed of 1,250ft. per min., it was time Mr. Lindén turned his attention to those engines and redesigned them. Incidentally, the product of piston speed and pressure was a very useful one for comparing ratings of engines, and he thought it could be more widely used. For modern two-cycle engines this product should be about 90,000 with natural aspiration and could be as high as 350,000 with pressure charged engines. For example, the naval engine referred to on page 210 had a factor of about 250,000, so that was quite highly developed. With the four-stroke engine this numeral should range from 170,000 for natural aspiration to 700,000 and over in very high speed highly pressure charged engines. Mr. Lindén's generator had a factor of about 113,000 supercharged, which could be allowed to speak for itself.

On page 204 the author stated that in measuring the temperatures in the pistons he placed thermocouples in the part of the piston where experience had shown that burning of the surface generally took place. That burning generally took place seemed rather a strange admission. Had any attempt been made to have a non-uniform distribution of fuel injection from the two sprayers, or were they merely of the conical injection type?

In scavenging arrangements the Götaverken engine seemed to have run the whole gamut from two large individual pumps to under-piston supercharging, which Mr. Lindén had said led to the present system but which he felt might be more accurately stated to have reverted to the present system, whereby each cylinder had an individual pump. There would appear to be a duplication of parts, and in view of the number of scavenge valves there must be a lot of maintenance work on them. Although the scavenge valves were said to be accessible he took that with a pinch of salt, knowing what a very dirty and awkward job it could be when they had to be cleaned, particularly when there were so many of them. He would suggest that in the interest of marine engineers in general Mr. Lindén might go one step further and throw the whole lot away. Perhaps he could follow it up with the camshaft, for when he went to the trouble of operating the exhaust valves from the crankshaft it was difficult to see why he threw in a camshaft for good measure.

Referring to the exhaust valves, as near as he could get by scaling, and allowing for a constant ratio of cylinder to valve areas, it would appear that in the 850-bore engine it must be about 20in. diameter, with an overall length of about 6ft. They would not assist the smaller marine engineer to use his intelligence in the place of brawn and muscular over-development! Little care seemed to have been taken to mask the valve stem, and an appreciable length of it was uncooled. Even allowing for the lower gas temperature of a two-cycle engine he wondered just how well the valves were kept gas tight. He would have liked to have seen enlarged views of the valve and operating mechanism. Presumably there were springs in the pots, although they were not mentioned in the text. They would have to be fairly powerful if they had to support a valve of that weight and produce the necessary accelerations to close it.

With regard to piston cooling, it appeared that the inlet oil was brought up through the centre of the honeycomb on to the crown, allowed to disperse over the edges and come back down the rod, but from what he could see of the slide shown of the new engine it seemed that the oil came out of the two sides and then came back down to holes around the rod insert. How was the centre of the crown adequately dealt with?

The question of engine bedplates had been raised by one or two speakers. He felt that particularly with aft engine jobs the engine shaft, propeller, stern frame, and the usual stern end accessories should be considered as a unit, and if the naval architects could not design a ship which would not cause undue deflexions in that unit, then he thought there was something to be said for the Mississippi pusher idea of sticking on a propulsion unit which was not even rigidly attached to the ship! There was no reason why the cargo carrying part of the ship should cause undue deflexions in the engine propelling section.

Finally, on page 204, the crankshaft was said to have an ample radius between each crankpin and web to reduce the concentration of stresses in the material, and in order not to reduce the bearing length of the big end bearing, those radii were located in the webs. This was not new, but another well known engine that adopted this arrangement was understood to have had some crankshaft failures in which the recess might have been a contributory factor. It would certainly appear to introduce a weak section to oppose the "winkling out" effect of the crankpin should the crankweb deflexions be excessive. However, Mr. Lindén was probably very fortunate in having a very stiff crankshaft, but it would also appear that there was excess metal and hence excess weight in the other thicker parts of the web.

The CHAIRMAN, before asking Mr. Lindén to reply to the discussion, said he would like to ask one or two questions of his own. The first concerned the use of oil cooling for pistons. In very large bore engines Mr. Lindén was apparently continuing to use oil cooling. He understood that one well known manufacturer of marine engines had recently decided to revert to water cooling with engines of that size. In view of the well known disadvantages attending the use of water cooling, presumably the reasons were very compelling, and he would like to ask the author whether his firm had found it necessary to consider this point.

Secondly, he wished to reinforce a question asked by a previous speaker regarding the life of the exhaust valves. From the diagram shown they looked very formidable, and it was well known that the larger the valve the hotter it tended to run. Going back more years than he cared to remember, he had very vivid recollections of having to spend many weary hours grinding in exhaust valves, so he found himself wondering what the life of those valves was between overhauls, particularly when using residual fuels.

Finally, he wondered what arrangements Götaverken had made with a view to avoiding the possibility of crankcase explosions.

Correspondence

MR. W. MCCLIMONT, B.Sc. (Member) congratulated the author not only on the subject matter of his paper but also on the creditable presentation of it in English. It would be hypocritical, however, to omit two general observations of faults; these observations were certainly not made in any spirit of malice. The first criticism was that the English missed the appropriate idiom at places and, as a result, the ideas being developed were either obscured or rendered inaccurate; the second criticism was that the paper was somewhat disjointed and at times repetitive. Perhaps it would have been better to have concentrated more on the philosophy underlying the more unusual features of the engine design and at the same time to have dispensed with those parts of the paper which read rather like extracts from a brochure.

It was interesting to note that cracks were experienced on certain welded entablatures and that annealing had been introduced to deal with this problem. The statement, however, that a high tensile steel had been used was rather surprising, particularly since this was said to have been done because it was suspected that brittle material had been used. Could the author specify what he meant here by high tensile steel, and indicate what properties he considered had been improved.

Long studs fitted to the steel castings at the top of each entablature were used to hold down the cylinder covers. Marine engineers in this country, unlike their Continental counterparts, had an inherent dislike of studs for dynamically loaded parts; had any difficulties been encountered either with fatigue failure or loosening of these studs? Being screwed into steel castings, one assumed that a relatively high tensile steel was used for the studs but some details of the design of the stud ends would be welcome. For instance, were the studs driven down to wedge on the thread run-out or did they have a collar? Was any form of torque control used during fitting? He would suspect it would be necessary.

The cylinder cover was in two parts and he wondered what were the advantages of the two-piece construction. The arrangement looked to him like a cast iron cover, with the advantages of that material in that location, to which a cast steel backing plate had been added for strength, or, more accurately, rigidity.

It seemed to him that the replaceable valve seat introduced an extra sealing face. This seat was not itself watercooled and, with doubtful conduction at the interfaces, the valve seat might run quite hot. Looking at the whole cover design one wondered whether the lower part of the water cooling space in the valve housing might be redundant. Could the author be more specific about the disc of heat resisting material used in the valve and how it was attached to the mild steel stem?

He was afraid that he had read and reread page 205 and studied Fig. 12 but he still did not know what it all meant. He would like to know precisely what a cooling insert was. The statement regarding the temperature at measuring point 5 in Fig. 12(c) being influenced by the "fact that the cooling device cannot press the oil up into the moulding of the piston, so this remains dry" appeared to be at variance with an earlier statement that "the element 5 has been placed in that part of the piston where coke formation generally occurs". Incidentally, might he observe that the use of different scales in Figs. 12(b), 12(c) and 12(d) was most misleading.

The piston in Fig. 12(a) was different from the one in

Fig. 11. Would the author care to comment on that? At the same time some information regarding the material of the piston skirt would be welcome.

The section of the paper concerning the crosshead pins with the oil grooves in the pins instead of in the white metal was intriguing. What was even more intriguing, however, was by what criterion this arrangement had been proved so superior to the standard bearing that a shipowner had decided to fit bearings of the new design as soon as a change of bearings was required.

When discussing the new engine type 850/1700 VGSU, the author said that the bore would be limited to 850 mm. "in order to save longitudinal space in the engine room". Surely the reverse was true; the rest of the paragraph concerned seemed to argue so.

Did the statement regarding the new 850/1700 type engine that "each working cylinder is thus combined with a double-acting scavenging pump of piston type, which is actuated from an arm on the crosshead" mean a reduction to one pump per cylinder instead of two?

The references to the use of electronic computers were very interesting. His organization had also been looking into their use, and he thought they would agree with the author about their potential value for torsional vibration calculations. He was not so sure about longitudinal vibrations in crankshafts and intermediate shafts; whilst he doubted if they could face the solution of these problems without a digital computer, he doubted very much also whether the information necessary for these calculations was yet available. It was gratifying, however, to find that the problem was being tackled.

Determination of the natural frequencies of the rocking vibration of engines by computer calculations was perhaps even further away. In this field, data on which to base calculations was even more sparse.

One point which appeared rather vague in the description of the new high power engine was whether the pistons were to be water or oil cooled. Reference had been made to fatigue cracks in pistons of low alloy chromium molybdenum steel; would he be correct in thinking that these had been in the side wall in line with the top of the skirt?

Finally, he noted an absence of any comment on cylinder lubrication.

Mr. M. MacDermott (Member) noted that the Götaverken descriptive brochure in his possession referred to engine types DM, whereas Mr. Lindén had written in his paper of engine types VGSU. Were these quite separate or could the descriptions be reconciled?

During the construction of Götaverken engines at Le Havre, for the company with which he was associated, they had experienced an abnormal number of minor fractures at the upper surfaces of the main bedplate cast steel saddles and the cast steel entablatures. A Götaverken representative at Le Havre stated that similar fractures were not uncommon experiences during construction at Götaverken and little significance was attached to them there. He would be most interested to learn whether Mr. Lindén would confirm the comment, and if so would he indicate the reasons for the fractures and the reasons for regarding them as unimportant.

He observed it was stated that the cylinder entablatures were linked together by fitted bolts to form a longitudinal girder. Did he understand from this that all bolts were fitted, and, if this were the case, was there any record of fractured bolts or other difficulties associated with this form of tying together highly stressed units of the engine?

Concerning the interesting observations about the accurate measurements of temperatures at various points within the piston heads, and appreciating the progress that had developed from this knowledge, he would be very interested to learn what was the specification for the lubricating oil used for piston cooling, the specification of the lubricating oil for the cylinder liners, and the specification of the piston rings. The avoidance of piston ring failures was largely due, he was convinced, to efficient piston cooling and, of course, satisfactory lubrication, and he would find it most interesting to learn whether any new data were available at Götaverken to connect piston ring life expectation in terms of running hours.

In common with the great majority of marine engineers, he found the Götaverken practice of oil grooves within the crosshead pins a fascinating departure, and the reduction in bearing surfaces stated to be as much as 50 per cent was surprising. Had Götaverken now incorporated this as a standard arrangement, and did they maintain the bearing surface reduction to the same degree? Having achieved such considerable success with their crosshead pins, had they experimented on the same lines with other bearing points within the engine?

During his talk, Mr. Lindén referred to cast iron entablatures. It was his (Mr. MacDermott's) impression that the Götaverken practice was to use cast steel for these members. Was that so?

Referring to Fig. 23, he was not clear concerning the difference between the descriptive phrases, "turbocharged" and "normally aspirated and turbocharged", since he had understood that with Götaverken these were one and the same thing by virtue of attached mechanically driven scavenge pumps.

Rear Admiral A. L. P. Mark-Wardlaw (ret.) (Member) had noted with particular interest the geared installations in Fig. 20. He himself had always been a protagonist of multiple geared units and preferred twin-screw propulsion. Perhaps it was because of the owners' and operators' responsibility that he viewed the problem this way, remembering the views expressed by their distinguished Past President, Lord Howard de Walden, who, in his Presidential Address, referred to the optimism of those who relied on one shaft. Noting the high wages referred to by the author, he felt that whereas for bigger units fewer machined parts had been provided, to make the smaller multiple units economical for the manufacturer standardization with line production would be required. Output in increased numbers might result if world trade increased on transfer from armaments to merchandise.

Looking then to the multiple engines and shafts, apart from weight and space, they must consider reliability, propeller loading, and, in the author's final words, economy. On the latter point, appreciating the importance of Diesel engines and their better economy, they must consistently improve within the type, bearing in mind improvement outside it. With these thoughts in mind he had looked over such figures as were published by various shipowners and engine builders the world over.

Noting Fig. 19(b) and the excellent fuel consumption results reported for the Götaverken engine, he had tried to find a lower figure from Diesel engines in the coaster trade and was rewarded by an engine made in Groningen and fitted, he believed, to some 2,500 coasters since 1906. It was the Brons GV engine with the pulse system and consumption of 156 grammes as compared with the Götaverken 162 grammes; the respective exhaust temperatures appeared to be Götaverken 430 deg. C. and Brons 405 deg. C.

As regards comparative reliability, on which, perhaps understandably, little was published, he wondered whether the pulse or constant pressure systems had the advantage on this score and whether the lower fuel consumption might bring less reliability. He noted however that with the constant pressure system the loss of one turbocharger might reduce speed to two-thirds of normal; perhaps this risk was avoided in the pulse system.

If marine engine builders were to enjoy in the future a vast turnover similar to the motor car manufacturers', perhaps those concerned with engines for the ocean trade and coasters could with advantage get together to mass produce a standardized geared twin-screw Diesel unit that would keep at bay competitors in other types.

Author's Reply

Mr. Lindén, replying to the discussion, remarked that there had been many questions and the points raised were very interesting both to himself and his firm. He would try to answer as fully as possible, but he was not a specialist on all the details; he hoped, however, that he would be able to cover most of the points discussed.

Dealing first with Fig. 12, a clarification of the piston construction shown, as well as a more detailed explanation of the temperature measuring, had been asked for by several speakers.

Before the cooling insert was introduced for the pistons there were only two pipes placed in the cooling chamber, one pipe for the inlet and one for the outlet of the cooling oil. By measuring it was soon found that after the engine had been stopped the oil level in the piston dropped to the upper edge of the outlet pipe and the inside of the crown was freed from oil. This meant that overheating of the inside of the piston occurred if the engine was stopped after having been subjected to high load. This overheating was confirmed by drilling melt plugs into the inside surface of the crown. It was impossible to determine by this method, however, how quickly the temperature was rising. After mounting thermo-elements in the piston top it was possible to measure the temperature variations when the engine was stopped. In order to reproduce the same cooling condition as before the introduction of the cooling insert, the pump was stopped so that the inside of the piston was freed from the cooling oil. The results showed about the same temperatures as those which had been established earlier by melt plugs. If the cooling insert were eliminated and the same tests had been carried out, but with the pump in service, practically the same result would have been obtained, as the cooling oil in this case would only have swept over the relatively cool lower parts of the piston and this would not have made any difference in the heat reduction from the inside surface of the piston crown. Only measuring point 5 had possibly been affected with a somewhat quicker heat reduction from the mantle surface of the piston.

The temperatures at the measuring points 1, 2, 3, 4, and 5 in the piston were about 155, 200, 245, 290, and 225 deg. C. respectively. These values referred to a standard engine at normal load. For the turbocharged engine, about the same values could be obtained at an effective mean pressure of about 7 kg./cm.^{2}. When the effective mean pressure was raised to 7·5 kg./cm.^{2}, these temperatures increased by about 10 to 15 degrees and the temperature drop at 1 to 4 increased from about 145 deg. C. to about 150 deg. C. A certain decrease of these temperatures could be obtained by lowering the compression or decreasing the ignition pressure. However, as a rule this meant an increase in the fuel consumption.

The temperature near the topmost ring on the piston had only been measured by means of melt plugs and for a standard engine at normal load it was about 150 deg. C. to 170 deg. C. The temperature of the inside surfaces of the cylinder liner at the topmost piston ring's top dead centre was about 175 deg. C. at an effective mean pressure of 7·4 kg./cm.^{2} and a water temperature of about 65 deg. C.

The fuel was injected through two fuel injection valves, each with four holes in the atomizer. This meant that the jets must be directed obliquely downwards towards the piston in order to prevent overheating of the exhaust valve, and also to get the best combustion possible. The air turbulence in the cylinder distributed the fuel but it was not possible to avoid the exposure of certain parts of the piston to more heat than others. Before any noticeable burning occurred the piston had as a rule been in service for a number of years. In cases where the atomizer was defective, the burning could occur more quickly. In order to record the highest temperatures in service the thermo-elements had been placed at these spots.

The merits of the constant pressure turbocharging system with its scavenging pumps, as compared with the pulse system, and certain relevant problems, had been raised by Admiral Cowland, and Messrs. Siggers, Jackson and Yellowley.

Regarding the total cost of installation on a given engine, the constant pressure system allowed the adoption of blowers working at their peak capacity, which was higher than was generally the case with pulse supercharging. As blowers were the most expensive items, any reduction in their number from, for instance, 3 to 2 would be considerable, especially as the number of blower foundations and connecting pipes required would also be reduced. Scavenging pumps and liners were simple items and their cost did not exceed that of an auxiliary blower for emergency use, together with its motor and piping.

The question as to a possible difference in fuel consumption between the pulse and constant pressure systems could be answered basically by the timing of the exhaust valves. On a pulse type engine they must open earlier than on a constant pressure engine, enabling a fraction more of the useful expansion work available in the cylinder to be transferred to the blower. The constant pressure GV engines retained the scavenging pumps which absorbed some power. Depending upon the chosen pressure and temperature levels throughout the engines, either system could be the better, but must be judged against the safety margins. Götaverken was of the opinion that the margins included in the system they had adopted enabled them to offer a very reliable engine.

Even when the exhaust temperatures in the GV engine were moderate, there had been no difficulty in raising steam in exhaust gas boilers.

Regarding the scavenging system, loop scavenging had never been considered in GV engines. The present arrangement of two scavenging pumps per cylinder gave better scavenging efficiency than one large pump, as they gave a better distribution of air to the cylinders and less pressure fluctuation in the scavenging air receiver. Underside piston scavenging combined with small pumps attached to the crossheads was modified to the present arrangement because of the maintenance difficulties experienced with the circumferential ring, actuated by the exhaust valve mechanism, which covered the scavenging ports during most of the stroke. The present pump system was much easier to deal with and gave a volumetric efficiency of 95 per cent. On the new engine the two double-acting pumps would be reduced to a single one. Due to the elimination of the large single scavenging pump driving mechanism, the total mechanical losses within the engine were reduced and the mechanical efficiency thereby improved.

Messrs. Siggers, Jackson and McClimont had discussed the construction of the entablatures and the material used for this equipment. In dealing with this question, a few words about the design principle without through-going stay bolts

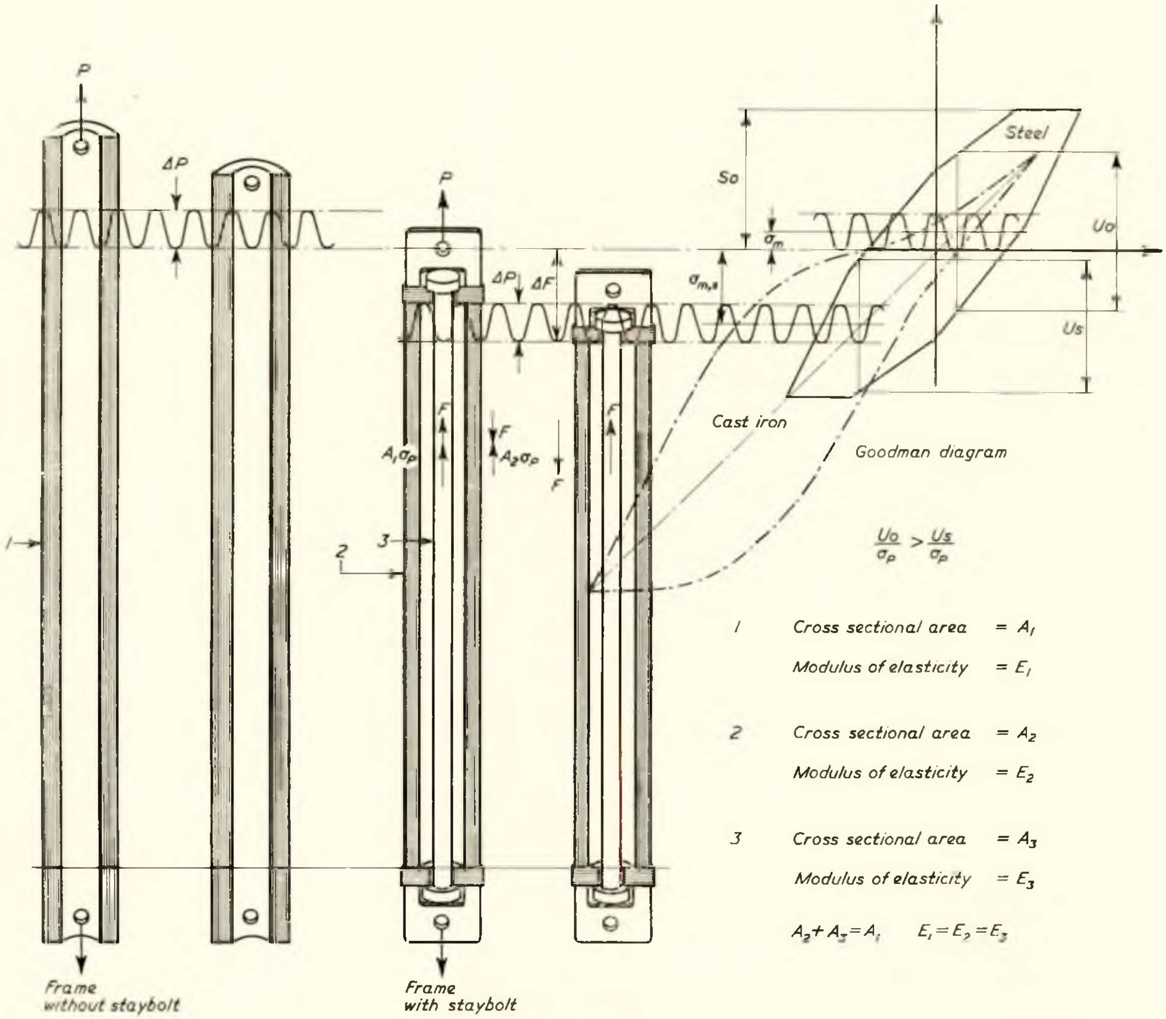


FIG. 27—*Forces and stresses in simplified engine frames, with and without stay bolts*

or tie bolts might be appropriate. The first Diesel engines looked like steam engines. Both engines had the same type of open cast iron frames, stressed in tension by the cylinder pressure, which resulted in severe damage in those early Diesel engines. The steam engines worked with a low cylinder pressure of about 15lb./sq. in., whereas the maximum cylinder pressure in the Diesel engines was in those days about 700lb./sq. in. Cast iron, which did not stand up to tensile stresses very well, withstood compressive stresses much better. Therefore the design principle with through-going stay bolts had to be adopted, with the result that the frames were relieved of the tensile stresses and subjected instead to compressive stresses. After turning over to welded construction, some manufacturers retained the stay bolt principle but others dispensed with it.

There had been much discussion as to whether stay bolts should or should not be used in welded engines. The forces in frames with and without stay bolts were schematically compared in Fig. 27. The frame with stay bolts had the cross sectional area $A_2 + A_3 = A_1$. The frame without stay bolts had the same cross sectional area A_1. This relation was chosen because frames of the same weight and with the same price had to be compared. In accordance with Hooke's law both frames must elongate to the same extent under equal stresses. Hence the same variable stresses due to the cylinder pressure occurred in frames with and without stay bolts. However, the frame with stay bolts was compressed when the stay bolts were tightened. In practice the cross sectional area of the frame varied with the height of the frame. There was a minimum value of necessary prestress if leakage was to be avoided and the static stresses caused by the prestress could be severe in the smallest section of the frame. For steel, the fatigue diagram was equal for tensile stresses and stresses in compression. The necessary prestress caused a decrease of the factor of safety in the frame with stay bolts and this was especially accentuated at the cross section with the smallest scantlings. Mr. Siggers's very great experience of marine engineering was appreciated, but a reply could be given to his statement that "where tie bolts had been used there had been very little trouble". It was known that frames with tie bolts had cracked in the same way as Götaverken's and it was also known that the bedplates attached to frames with tie bolts had cracked seriously. GV opinion was that these cracks were caused by the concentration of stresses due to the tie bolts and in some instances to poor design. The welded bedplate with tie bolts for the new engine under construction had therefore been carefully designed to avoid trouble due to stress concentrations caused by the tie bolts.

The welded GV engines were still in accordance with the original design but minor alterations had been made in order to avoid the occurrence of stress concentrations in the entablatures. Thus the points mentioned by Mr. Siggers had been answered. Calculations and measurements had shown that the stresses had been diminished at these points. Previously the cracks usually occurred within the first year of operation. Nowadays the results were better. If the better results were due to the use of better material and annealing, there must be an increased probability of even better results after the modifications to the design had been put into effect.

Referring to the material used, as the cracks that had occurred grew very slowly, the mode of failure was characteristic of fatigue. Because the first entablatures were made from ordinary ship plate the quality varied, especially the carbon content. If the carbon content was high the weldability was not so good and small cracks occurred in the brittle heat affected zone, causing severe stress concentration and promoting fatigue cracks.

To obtain good weldability, the sum of the carbon content plus one-sixth of the manganese should not exceed 0·40 per cent, but this was not always the case with the steel used.

Furthermore, several fatigue tests had shown that the endurance limit at two million stress cycles was equal to the yield point. Thus a material with a high yield point value and therefore a high U.T.S. value would be better able to endure fatigue conditions, but the steel must still have good weldability.

The new material now in use had a carbon content of 0·15 per cent maximum and a manganese content of 0·9 to 1·5 per cent; the steel was killed and fine grain treated. The sum of the carbon content and one-sixth of the manganese content was 0·40 per cent maximum. Hence the steel had good weldability. Further, the yield point of this steel should not be below 19 tons/sq. in. and the U.T.S. between 30 and 37 tons/sq. in. An endurance limit at 24 tons/sq. in. when cycled 2×10^6 times had been obtained. This steel was in accordance with the requirements for Lloyd's Register's P5 except the rule saying that the U.T.S. was to be between 26 and 30 tons/sq. in.

The deflexion problem, referred to by Messrs. Siggers, Jackson and Archer, had been studied very thoroughly by the author's firm.

There had been much discussion regarding the influence of a ship's movements in heavy seas and at different load distributions. If the load of the ship was distributed so that most of the weight occurred at the ends, there would be hogging deformation and stresses, while loading with the weight amidships would cause sagging deformation and stresses. In a vessel with the machinery amidships, the engine room double bottom could undoubtedly be deflected due to hydrostatic pressure. If these deformations were to influence the strength of the engine frames, more cracks should have been recorded from ships with the engines amidships. This has not been observed, however, and as the engines had always been situated near the neutral line, the influence of the ship's movements from hogging to sagging must have been very slight. In any case, as the moment of inertia of a transverse section of the bedplate was only a fraction of that of the double bottom it must obviously have been impossible to stiffen up the double bottom or the ship by increasing the bedplate scantlings.

In order to estimate the influence of ship movements, measurements had been taken on board a bulk ore carrier. These measurements were made when loading and unloading and heeling the ship, the draught variation was about 7ft. and a list of 15 degrees was applied. Strain gauges were placed on the upper and lower flanges of one of the longitudinal bedplate girders, and at different places on the entablatures. Only very small values were obtained. It was possible to calculate the deformation of the bedplate.

By using a displacement diagram the peak movement could be estimated and the stresses in the bedplate and entablatures due to the ship movements could be calculated. The stresses were very low and could not possibly cause any trouble. If the length of the engines was increased the deformation would increase, but the deformation/length ratio was supposed to be constant or diminishing and thus the stresses were still the same. Hence GV could not see any likelihood of trouble.

In only a few cases so far had there been occasion to stay the engine to the ship's side on account of rocking, and these were associated with a turbocharged ten-cylinder engine. This rocking motion had a coupled vibration of the engine body—double bottom with the excitation forces acting *via* the guides. The beam of the ship where the engine room was situated naturally had some influence also, as the natural vibration frequency was dependent thereon. The most vital influences, however, were the firing order and the number of engine cylinders, these being the predominant factors affecting the summation of the impulses from the different cylinders. As an example of this, it had not been necessary to stay to the ship's side a twelve-cylinder engine of the same type installed aft in a tanker.

The design of the exhaust valve springs discussed by Admiral Cowland and Mr. Siggers, had an alternative version, now standard, which incorporated four springs instead of one. The main advantage was that the spring noise due to contact between the coils had been reduced.

In reply to Mr. Jackson, the valve motion from the crankshaft had shown very good results in service, and, combined with the camshaft for the fuel injection pump mechanism, a simple and speedy reversing mechanism had been obtained.

Mr. Jackson and Mr. Archer had been interested in the lubrication grooves in the crosshead pins. The first pins of this design were fitted in an engine where troubles with crosshead pins had occurred repeatedly. The first of these pins had been in service for 34,000 hr. and the bearings were in the same condition as when first fitted.

The bearing area had only been reduced for experimental purposes in order to examine the safety factor. This bearing had now been in service for about five years and was still perfect. There was no intention of reducing the bearing area of the engines GV had in production. Experiments with other bearings in the engine had not been carried out.

The crosshead pins were normally executed with flame hardened surfaces but chromium plated pins had also been in service for a long time with good results.

In reply to Mr. Jackson's questions about the starting air distributor, the former type, with valves and camshaft, was still used for cast engines while the type with rotating slide was adopted for welded engines.

There were no cooling coils around the cylinder.

Replying to Mr. Archer, comparative values for the fuel consumption of gasifiers of the 520/900 size were not available, but experience with other installations had shown that the fuel consumption for gasifiers of this size should be at least 5 per cent less than for the minelayer mentioned.

The exhaust temperature after the turbine was estimated to be about 185 deg. C. (365 deg. F.).

The gas ducting material had the following analysis: C 0·20 per cent, Si 0·40 per cent, Cr 0·3 per cent, Cu 0·4 per cent, P 0·05 per cent and S 0·05 per cent.

No crankcase explosions due to fire in the scavenging air receiver and overheating of the crankcase roof had been brought to the author's notice.

As a welded version of the 850/1700 engine had not so far been constructed, comparative weights could not be given.

The steel castings were in accordance with the requirements of the various classification societies. However, to obtain as good a quality as possible, a control arrangement had been established so that inspectors from GV paid regular visits to steel works both in Sweden and on the Continent, where material had been ordered.

Mr. Yellowley had suggested that consideration should be given to the elimination of the bypass valve. This valve had been modified to ease maintenance work and to reduce its dimensions, but as circumstances could arise in which it would be desirable to shut down the blowers when the engine was running, for other reasons than a blower breakdown it had been retained.

The Chairman and Mr. Victory had asked for more information about the exhaust valves. The exhaust valve life depended *inter alia* upon the setting of the valve mechanism, the fuel injection pump and valve, the fuel oil quality, etc. Even on a Diesel engine burning heavy fuel, grinding should not be required more frequently than about every sixth month.

In reply to Mr. McClimont's question, the material from which the exhaust valve discs were made was a chromium steel with the analysis: C 0·40 per cent, Si 2·5 per cent, Cr 12 per cent, Mo 0·8 per cent. The valve discs were shrunk on to the valve stems, the ends of which were protected at the combustion chamber end by a threaded plug of chromium steel. This plug served at the same time as an additional axial locking device for the valve stem.

Mr. Victory had questioned the arrangements for piston cooling. The cooling of the working pistons for the 850/1700 engine was arranged in accordance with the principle illustrated in Fig. 11. The cooling oil discharged helically from the centre around the inside of the piston crown and led out through a central pipe in the piston rod.

In reply to the Chairman's other questions:

The cooling of the piston with lubricating oil had given such satisfactory results that water cooling with its inconveniences had not been considered.

Crankcase explosions had been *inter alia* prevented by:

1) Efficient sealing between scavenging air receiver and crankcase.
2) Complete isolation between the cylinder proper and the crankcase (crosshead engine with short piston).
3) Generously dimensioned white metal lined main bearings, bottom end and crosshead bearings.
4) Generously dimensioned safety valves on the crankcase.

Replying to Mr. McClimont's further comments: The cylinder cover studs reached down to the bottom of the tapped holes in the steel castings and were made with an undercut thread run out at the end of the material. The body of the studs was turned down to a dimension which was less than the core diameter of the thread (Fig. 11 showed the same type of stud attachment). Fatigue failure had not occurred in these studs.

The cylinder cover was divided into two parts for the following reasons. The lower part was made of cast iron in order to get a material resistant to the hot and perhaps sometimes corrosive gases in the combustion chamber. However, as cast iron did not compare with steel under the bending stresses present in the upper part of the cover, due to the prestressing of the long studs mentioned above and the gas forces, cast steel was used.

In describing the new engine, the wording, "The bore will be limited to 850 mm. (33·5in.) in order to save longitudinal space in the engine room" was misleading and had been corrected to read, "Limitation of the bore to 850 mm. (33·5in.) will result in the saving of longitudinal space in the engine room".

The number of scavenging pumps in the 850/1700 type engine had been reduced from two per cylinder to one.

The author had thought it necessary to give rather detailed descriptions of those parts of the engine under consideration which had special features, even at the risk of giving the impression that these were "extracts from a brochure". As the method of cylinder lubrication employed was quite a usual, ordinary arrangement, reference to it in the paper had not been considered necessary.

Mr. MacDermott had asked about the letters DM preceding the engine type number in one of the Götaverken brochures; this number only indicated that the engine was a Diesel engine.

The steel castings for the bedplate and top pieces for entablatures were examined very thoroughly by magnaflux and cracks, if observed, were as a rule eliminated by machining.

The bolts which connected the entablatures at the upper parts were fitted bolts which distributed the ignition forces between two adjacent entablatures. It was important that these bolts should be well fitted and tightened, otherwise there was a risk that they would be "chewed up", resulting in an unfavourable load distribution.

The lubrication oil used for piston cooling could be of any reputable make with the same specification as the circulating oil in the system.

The cylinder oils used were the ordinary types from the various oil suppliers. With the newest cylinder lubricants for engines operating on heavy fuel oils the following cylinder liner wear values had been obtained:

	Maximum, hr.	*Minimum, hr.*	*Mean cylinder wear, mm./1,000 hr.*
Diesel	8,218	3,166	0·083
Fuel	7,982	2,475	0·105
Supercharge	5,500	2,540	0·158

The specification for piston rings mostly used in GV engines was as follows: C 3·1—3·4 per cent, Si 1·3 per cent, Mn 0·6 per cent, P 0·35 per cent, S 0·05 per cent; Hardness 180; 210 HB.

The avoidance of piston ring failure was largely due to efficient piston cooling, and it had been found that the amount of cylinder oil used should not be more than 0·25 grams/i.h.p./hr. to get minimum cylinder liner wear.

A reduction of bearing surfaces had only been made for research purposes. The crosshead pins with grooves were not yet introduced as standard. The tests only referred to crosshead bearings.

It was Götaverken practice to use cast iron entablatures when they were not of welded construction.

Fig. 24 showed the output ranges for the various engine types. The sectioned areas for engine types 520/900 to 760/1500 showed the output increase obtained by providing them with turbocharging. For engine type 850/1700 the output range had been indicated by a completely sectioned area, this range including engines with and without turbocharging.

INSTITUTE ACTIVITIES

Minutes of Proceedings of the Ordinary Meeting Held at the Memorial Building on Tuesday, 8th December 1959

An Ordinary Meeting was held by the Institute on Tuesday, 8th December 1959 at 5.30 p.m., when a paper entitled "The Development of the Götaverken Diesel Engine" by R. A. Lindén, M.R.I.N.A. (Member) was presented and discussed. Mr. R. Cook, M.Sc. (Chairman of Council) was in the Chair and ninety-three members and visitors were present. Seven speakers took part in the discussion that followed.

A vote of thanks to the author, proposed by the Chairman, was accorded by acclamation. The meeting ended at 7.55 p.m.

Section Meetings

Kingston upon Hull and East Midlands

A meeting of the Section was held at the Royal Station Hotel, Kingston upon Hull, at 7.30 p.m. on Thursday, 17th March 1960 which took the form of a lecture given by Mr. M. E. O'Keeffe Trowbridge, B.Sc., A.C.G.I., on the "Centrifugal Purification of Oils for Marine Service". The lecturer traced the development of the centrifuge, not only for its marine application, but in industry in general. In the latter case he explained how certain industrial processes had been revolutionized by the introduction of the centrifuge.

Mr. Trowbridge then described the various types of centrifuge used in marine engineering, illustrating these with lantern slides. In conclusion, he discussed the latest trends and possible future developments in the marine centrifuge. A short but interesting discussion then followed which centred mainly on the shipboard use of the centrifuge and on the difficulties experienced by ships' personnel in fixing the rate of "through-put".

A vote of thanks was proposed by Mr. A. R. Eaton and seconded by Mr. C. A. Potter.

The Chairman closed the meeting at 9.30 p.m.

A further meeting of the Section was held at the Great Northern Hotel, Leeds, on Thursday, 31st March 1960 at 7.30 p.m., when Mr. Bryan Taylor, B.Sc.(Eng.) (Member) gave his lecture entitled "The Holmes-Werkspoor Engine: Building, Testing and in Service". He described the many unique design features of the engine and illustrated his lecture with coloured lantern slides of a very high standard. Mr. Taylor then described the special test bed and testing arrangements which were constructed for this engine, and in conclusion gave the latest reports from engines in service.

An interesting discussion then followed, members from all areas in the Section taking part.

Mr. J. Whitaker of Leeds then proposed a vote of thanks which was seconded by Mr. R. Rawlings of Lincoln.

The Chairman, Mr. G. W. Hill, closed the meeting at 9.30 p.m.

Sydney

A meeting of the Sydney Section was held at Science House, Gloucester Street, Sydney, on 23rd March 1960. Captain G. I. D. Hutcheson, R.A.N. (Local Vice-President) was in the Chair and there were seventy-seven members and guests present.

The Honorary Secretary announced the names of the office bearers for 1960 as follows:

Chairman: Captain G. I. D. Hutcheson, R.A.N.
Committee: W. G. C. Butcher
W. F. Ellis
J. Munro
Captain(E) R. G. Parker, O.B.E., R.A.N.
W. T. Mathieson
F. J. Ward
Honorary Secretary: N. A. Grieves
Honorary Treasurer: J. W. Lamb

The Honorary Secretary also announced that Captain Hutcheson had been nominated President-elect of the Institution of Engineers, Australia.

Mr. R. W. Joselin then delivered a paper entitled "Problems Associated with Welding in Ship Construction and Repairs". Messrs. Barnes, Hilton, Gillies, Grieves, Large and Williams contributed to the discussion. A vote of thanks to the author was proposed by Mr. Buls, seconded by Mr. Ward, and carried with acclamation.

After the meeting, supper and the usual refreshments were served.

West Midlands

At a meeting held at the Science Museum, Newhall Street, Birmingham, at 6.30 p.m. on Thursday, 31st March 1960, Mr. N. W. Bertenshaw, B.Sc.Tech., Curator of the Museum, presented an illustrated lecture entitled "Development of the Steam Engine". Mr. J. R. Cotterill, J.P. (Chairman of the Section) was in the Chair and the meeting was attended by thirty-eight members and guests.

With the aid of slides Mr. Bertenshaw described how the early atmospheric engine, as used for pumping the water out of mines, had been developed into an efficient steam engine by Newcomen and Watt. Illustrations were shown of Watt's early condenser and air pump and the famous parallel motion mechanism which finally enabled the engine to exert a pushing as well as a pulling force.

The lecturer concluded by describing the engines and boilers fitted into Brunel's *Great Eastern* of 1895.

Mr. Bertenshaw answered the five questions that were asked, after which the Chairman thanked him on behalf of everyone present for a most interesting lecture.

The audience then adjourned to the heavy engineering section of the Museum, where a number of old engines, including an "Amos Bean" engine (1864) and a "Galloway Uniflow" engine (1922) were actually run on steam.

The last meeting of the 1956/60 session was held at the Engineering Centre, Birmingham, on Thursday, 28th April 1960 when an illustrated lecture entitled "Epicyclic Gearing" was presented to fifty members and guests by Mr. H. Norman G. Allen, M.A. (Member) and Mr. T. P. Jones (Member). The Chair was taken by the Chairman of the Section, Mr. J. R. Cotterill, J.P.

The lecture was introduced by Mr. Allen and the paper itself read by Mr. Jones. In addition to numerous slides shown during the lecture there was also a sound film on the manufacture of parallel and epicyclic gears.

After the film a discussion took place when all questions were very ably answered by the lecturers.

Mr. R. S. Robinson, B.Sc. (Member) proposed a vote of thanks to the authors for a very interesting paper.

The meeting closed at 9.0 p.m.

West of England

A meeting of the West of England Section was held in the Small Engineering Lecture Theatre, University of Bristol, on Monday, 25th April 1960 at 7.30 p.m., when the following films were shown:

"An Introduction to the Heat Engine"
"British Adventure"
"Journey from the East"

There were twenty-two members and visitors present; the meeting ended at 9.15 p.m.

Election of Members

Elected 16th May 1960

MEMBERS

Albert Paterson Alexander
Robert Baird
William John Davies, B.Sc.(Wales)
Charles Westcott Edgar
Charles William Ernest Cyril Evans
Roy Davidson Foster
Thomas Kinnear Glenday
Thomas Dartnell Green
Thomas Frank Heslop
Henry John Hodder
Robert Humphrey, B.Sc.(Durham)
Vincent Iddon, B.Sc.(Eng.) London
Richard William Cummings Jeffrey
Hans Ludwig Langgut
John Pettigrew Naismith
Ronald McInnes Overell
Alec Russell Ryan
Bertil Sandlien
Ghanshyam Babaji Satam
Charles Webster Tuson
Bruce Edward Wicken
Ernest Cuthbert Wilson

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Neville Arthur Bailey
James Bennett
John Reginald Blake
Anthony Bolton
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Andrew Shaw Cleveland
Eric Joseph Clifford
Kenneth Gibson Collinson
Norman Keith Croft
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Robert Harris Davison
James Bernard Dockree
William Douglas
Julian Henry Edlund
John Fitzpatrick
Douglas Neil Fleck
Derek Henry Hollingsworth
Keith Hughes, B.Sc.(Manch.)
Raymond Francis Isaac
Robert Dickson Johnston
Geoffrey Parker Jordon
William George Judd
Robert McAleese
Michael Gerard McGuinness
John Joseph Mallard
Brian Thomas Murray
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Peter George Eckford Watson
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William Lewis Hapgood-Strickland
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Donald Mackintosh
John Harris Martin
Roy Augustus Nottingham
Alan David Charles Owen
Douglas Gordon Saunders

GRADUATES

Charles C. Aderiye
John Barrow
Kenneth Beddoe
John Peter Currivan
William Howard Jones
Edward Frank Kirton
Trevor John Smith
Jeremy Bryan Strugnell
Ronald Ian Wakeman
James Roy Laird Webster
Robert Edward Wilson

STUDENTS

Sutikshan Kumar Aggarwal
Jamshed V. Appoo
David Alan Bolger
Patrick John Booth
Peter Jeffery Brockington
Rodney Michael Bryenton
Derek Ralph Chamberlain
Peter Sherwin Cole
Dipankar Das Gupta
Michael Drake
Dilip Kumar Ghose
Peter John Goodwin
Richard Charles Green
Hugh Leslie Higgins
David Martin Howe
Shamnawaz Husain
Abdul Hamid Khan
David William Litson
Trevor John Lloyd
Derek Arthur William Nicholls
Man Mohan Piplani
Amal Kumar Ray
Edward William Rowland
John Joseph Sullivan
Norman Swaffield
Peter James Tart
Richard Thomas Weston
Arun Tewari
John Stanley Widdowson
Paul Churchill Williams

PROBATIONER STUDENTS

John Reginald Boughey
David Charles Dale
John Lars Hallgren
George Kevin Hepworth
Colin Lithgow Herbertson
Ronald Keith Jacobs
Robert Clive Jarvis
Christopher John Kenrick
David John Nightingale
Robert Andrew Sharpe
Michael John Heatherington Weddle

TRANSFER FROM ASSOCIATE MEMBER TO MEMBER

John Hugh Archer
Roelof Boorsma
Cyril Douglas
Desmond John Haley
James Robert Lang
Eric Clague Lewin

William Henry James Moore
Harold Platt

TRANSFER FROM ASSOCIATE TO MEMBER

George Muir
William James Stewart
Leonard Teasdale
Godfrey Hunt Thomas
Gilbert Brian Yates

TRANSFER FROM ASSOCIATE TO ASSOCIATE MEMBER

Charles Frederick Giorla
Frank Menzies

TRANSFER FROM GRADUATE TO ASSOCIATE MEMBER

Rollo Charles Wheldon Aisbitt
William Brian Bambra
Michael John Blake
Michael John Fitzpatrick
George Stanley Mole

TRANSFER FROM STUDENT TO ASSOCIATE MEMBER

Nitindra Nath Bose

TRANSFER FROM STUDENT TO GRADUATE

Eric Cyril Avery
Gerrard Allan Hart

TRANSFER FROM PROBATIONER STUDENT TO GRADUATE

Colin Cummins

TRANSFER FROM PROBATIONER STUDENT TO STUDENT

Michael John Maurice Martin

OBITUARY

ALEXANDER ANDERSON BOOTH (Member 8287) served an apprenticeship with Messrs. Allan Brothers of Aberdeen and was then employed for two years by Messrs. McKinnon and Company. He went to sea in 1917 and served with various companies, obtaining a First Class Board of Trade Certificate. In 1933 he joined the Khedivial Mail Line as chief engineer and served them in this capacity until 1953. The last five years of this period were spent in the s.s. *Mohamed Ali El-Kebir* on regular passenger service between Alexandria and New York. Mr. Booth returned to Scotland in 1953 on sick leave from the Khedivial Mail Line and had hoped to return to his work there; however, for a considerable time he was not well enough for duty and the Suez crisis in 1956 ended any hope of his return. He died, aged sixty-nine, on 25th March 1960.

Mr. Booth had been a Member of the Institute since 1936.

THOMAS H. G. BRAYFIELD (Member 2627) died in Hong Kong on 19th April 1960, at the age of eighty-two. He served an apprenticeship with Cammell Laird and Co. Ltd. of Birkenhead from 1895/1900 and then spent seven years seagoing, as fourth to chief engineer. He obtained an Extra First Class Board of Trade Certificate. He joined the Hong Kong ship surveying firm of Messrs. Carmichael and Clarke and remained with them until his retirement from active business a few years ago. During the second world war he was interned in Hong Kong.

Mr. Brayfield was a fellow of the Society of Consulting Marine Engineers and Ship Surveyors and a member of the Royal Institution of Naval Architects; he was elected to Membership of the Institute of Marine Engineers in 1912.

STANLEY DAWSON CASEBOURNE (Honorary Life Member 2060) died suddenly on 22nd March 1960, aged seventy-nine years. He was senior partner of Messrs. Casebourne and Turner of Liverpool, consulting marine engineers and ship surveyors.

He was apprenticed to the Central Marine Engine Works, West Hartlepool, from 1897/1902 and subsequently served as a seagoing engineer with the Strick and Shaw Savill Lines. In 1907 he started a business as a consulting engineer in London with his brother, Mr. C. B. Casebourne, and the following year he moved to Liverpool where he went into partnership with the late Mr. J. Armour as a consulting marine engineer and ship surveyor. In 1929 he formed a partnership with the late Mr. G. F. R. Turner, an association which continued until Mr. Turner retired about ten years ago.

Mr. Casebourne was a founder member of the Society of Consulting Marine Engineers and Ship Surveyors; he served on the first council in 1920, again in 1929/31, 1952/56, and from 1958 until his death, representing the Liverpool district.

Mr. Casebourne was elected a Member of the Institute of Marine Engineers in 1908 and was made an Honorary Life Member by the Council in 1959 in recognition of his long membership. The parchment recording this appointment was presented to him at a ceremony in Liverpool by Mr. E. L. Denny, who was President of the Institute at the time.

JAMES FRANCIS DOUGLAS (Member 12700) was apprenticed to J. G. Kincaid and Co. Ltd. of Greenock from 1929/34 and then spent four years as a seagoing engineer, obtaining a Second Class Board of Trade Steam Certificate. He then joined Cammell Laird and Co. Ltd., as assistant repair manager for four years, then as plant manager until 1948, and as repair manager until 1956, when he received his final promotion to general manager. Owing to ill health he resigned this position in May 1959 and after a period of recuperation he took up a managerial appointment with Dawnay and Co. Ltd., constructional engineers at Swansea. It was while he was returning to Swansea from a weekend visit to his home in West Kirby that he met his death in a motoring accident on 14th March 1960.

Mr. Douglas was elected a Member of the Institute in 1950 and served on the Committee of the Merseyside and North Western Section in 1956 and 1957.

JOHN WILLIAM FEARMAN (Probationer Student 15737) died on 4th September 1959, aged twenty-one, after a severe operation and many months of illness.

He had been educated at the Brighton, Hove and Sussex Grammar School from 1949/54 and passed the General Certificate of Education at Ordinary Level in Physics, Mathematics, Mechanical Drawing and English Language. He was then apprenticed to the British Tanker Co. Ltd. and attended the South East London Technical College, but owing to ill health he had to leave the company just as he was about to go to sea.

As his one ambition in life was to become a marine engineer this was a terrible blow to him, so he continued to study engineering at the Brighton Technical College, with one object in view, that when his health improved he could go to sea. This was not to be, however. After many periods in hospital, during which he never lost hope or his sense of humour, it was found that he had a tumour on the brain which proved to be inoperable. Deep radium treatment gave him a few months of reasonable life at home but gradually he became completely paralysed and died in his parents' home at Rottingdean, Sussex.

COMMANDER WILLIAM JOHN HENTON-JONES, D.S.O., R.D., R.N.R.(ret.) (Member) died on 7th June 1959. He was educated at Bridgend County Grammar School and received his technical training at Cardiff Technical College. He served an apprenticeship with the Hills Dry Dock and Engineering Company, Cardiff.

During the first world war he served as second lieutenant in the Royal Flying Corps and later became a pilot officer in the Royal Air Force. In the latter appointment he served in non-rigid airships.

From 1919/26 he served in the mercantile marine with various shipping companies and qualified for a First Class Board of Trade Steam Certificate. In 1926 he joined the Royal Naval Reserve and served as Lieutenant(E) full time for two years. Then in 1928 he joined the Powell Duffryn Coal Company, Bargoed, as a shift charge engineer. From 1929 to 1938 he served at the Lancashire Electric Power Company's Radcliffe Power Station, at first as shift charge engineer and later as deputy resident engineer. In 1938 he became a shift charge engineer at the company's Kearsley Power Station, but on the outbreak of the second world war he was mobilized as Lieutenant Commander(E) in the Royal Naval Reserve.

During the war Commander Henton-Jones served as engineer officer in many ships. He started in 1939 with the Fleet Tenders "A" and "B", the dummy battleships, then in 1942 he served on the destroyer H.M.S. *Broke,* which was then defending the North Atlantic and the North Russian convoys. He was mentioned in despatches in April 1941, and in June 1942 was awarded the D.S.O. for personal bravery and devotion to duty in a battle for the relief of Malta. On 12th August 1942, while escorting one of the Malta convoys, his ship, H.M.S. *Cairo,* was torpedoed and had to be abandoned. Fortunately, most of the crew were rescued by the battleship *Rodney.* After spending some time in H.M.S. *Diomede,* Commander Henton-Jones was sent to Northern Ireland as engineer overseer at Harland and Wolff Ltd., Belfast, with special responsibilities regarding the manufacture and inspection of main turbines, boilers and auxiliary plant, flight deck machinery and petrol services for the first of the light fleet carriers, H.M.S. *Glory.* After the trials, he sailed to the Far East, where the surrender of the South East Asia Command at Rabaul, New Guinea, was accepted by General Sturdee on board his ship. He retired from the Royal Naval Reserve in 1948 with the rank of Commander(E).

In 1948 Commander Henton-Jones became deputy station superintendent at Kearsley Generating Station, and in 1954 station superintendent of the C.E.G.B.'s Agecroft "B" Generating Station, where he served until his sudden death last year.

Commander Henton-Jones had been a Member of the Institute since 1938 and he was also an Associate Member of the Institution of Mechanical Engineers.

William Whaley Tennent (Member 5447) died, aged seventy-five, on board the *Himalaya* on 26th March 1960, between Sydney and Vancouver, and was buried at sea. He was on a round-the-world trip, largely for reasons of health, and had already spent some time in India, Australia and New Zealand.

Mr. Tennent was educated at Glasgow High School and served an apprenticeship with J. Brown and Co. (Clydebank) Ltd. After several years at sea, and obtaining a First Class Board of Trade Certificate, he spent five years as an engineer surveyor with the British Corporation Register, and then joined Bull's Metal and Marine Ltd., serving on the board of directors for many years. He was still associated with the firm at the time of his death.

Mr. Tennent was a considerable athlete in his younger days, having played for Queen's Park and Partick Thistle Football Clubs and for West of Scotland rugby and cricket clubs.

Mr. Tennent was a Member of the Institution of Engineers and Shipbuilders in Scotland and was elected to Membership of the Institute of Marine Engineers in 1926.