Service Performance of S.E.M.T. Pielstick Engines

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The latest development in the range of S.E.M.T. Pielstick engines is the PC2 engine, constructional details of which are described together with performance characteristics. A high output has been obtained without exceeding the thermal and mechanical loads which are successfully carried by earlier lower rated engines. The specific fuel consumption is low over a wide range of operation and geared installations can be used advantageously to extend the range even further. Several different installations of Pielstick engines are described, suitable for a variety of applications.

The cargo vessel *Galatee* with two eight-cylinder engines was measured for wear of several parts at the last overhaul when 28,000 hours of service had been completed. Liner wear at the top of the top ring travel had occurred at a rate of only 0·0004in./1,000 hr., and the wear on other parts was correspondingly low.

Tests have been carried out using heavy fuel and a number of owners are operating Pielstick engines on heavy or intermediate fuels. Four engines in one vessel have now completed 13,200 hours' running. There has been no sign of crankcase oil contamination, but difficulty has been experienced with deterioration of exhaust valves as a result of the high vanadium and sodium content of certain fuels. Wear of the top piston ring groove occurred at first and was cured by using a cast-in ring bearer. Rates of wear of all other parts have been satisfactorily low.

The paper concludes with a description of the running of Pielstick engines as dual-fuel engines operating on natural gas. This is of interest in the propulsion of methane tankers.

S.E.M.T. PIELSTICK ENGINES OF THE TYPE PC

There are currently two types of PC engines, the PC1, which is already well known, and the PC2.

This latter, developed from the PC1 and following the same overall conception, has the following characteristics:

Bore	400 mm.	15·75in.
Stroke	460 mm.	18·11in.
Maximum continuous power	465 h.p. per cylinder	
Corresponding speed of rotation	465-500 r.p.m.	
Mean piston speed	7·1—7·6 m./sec.	1,400—1,500 f.p.m.
Mean effective pressure	15·6—14·5 kg./sq. cm.	222—206lb./sq. in.
No. of cylinders	6, 8 and 9 in-line 8, 10, 12, 14, 16 and 18 in Vee form.	

It can be seen from these characteristics that these are medium speed high performance engines which, in the various numbers of cylinders, cover a range extending to quite high powers.

The highest powered engine of the range, the 18-cylinder, gives over 8,000 h.p. for a weight of only 90 tons and in a space that is quite small as shown in Fig. 1, which gives the principal dimensions and weights of several PC engines.

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Fig. 2 is a cross-sectional drawing of the PC2 engine. This section shows the salient features of the PC engine which can be summarized as follows:

The frame is fabricated by electric welding from steel plate with forged steel and cast steel components.

The crankshaft is a single piece forging with notches in the webs for fastening the balance weights, two on each crank, one per web.

The main bearing housings consist of a cap and a bearing body. The latter is situated underneath the crankshaft and is fastened to the frame by two long through bolts. The bearings are thin steel shells lined with copper lead with a lead-tin bearing surface.

Cylinders are of a special cast iron, each with a separate cast iron water jacket designed to avoid any possibility of the cooling water coming into contact with the steel frame.

Pistons are made of aluminium-silicon alloy and are cooled by circulating oil under pressure from the lubricating oil system of the engine.

The body of the connecting rod is of H form and the bottom end bearings are steel shells lined with copper lead and having a lead-tin bearing surface.

Cylinder heads are of cast iron and have four valves, two inlet and two exhaust. The exhaust valves can be withdrawn without removing the cylinder head.

Reversing of these engines is performed by moving the camshaft axially, each cam having two symmetrical faces.

It is necessary to go into certain details to show how the relatively high performance of the PC2 engine (about 213 lb./sq. in., b.m.e.p.) has been achieved without exceeding the limits of thermal and mechanical loading accepted today and how at the same time the engine life has been extended.

Thermal Loads

To maintain the temperature of the top ring groove at

Type of engine	Dimensions					Approximate weight, tons
	A ft. in.	B ft. in.	B_1 ft. in.	C ft. in.	D ft. in.	
6PC2L	11–8	21–7	—	5–10	10–0	34·2
8PC2L	23–6	25–8	28–11	5–8	10–4	44·4
9PC2L	26–0	28–2	31–5	5–8	10–2	49·0
8PC2V	13–11	16–1	—	10–10	9–0	37·2
10PC2V	16–4	19–2	—	11–6	8–3	45·2
12PC2V	18–9	21–7	—	11–6	8–3	52·4
14PC2V	21–2	24–0	—	11–6	8–3	59·5
16PC2V	23–7	25–10	29–2	10–10	9–1	67·1
18PC2V	26–2	28–3	31–7	10–10	9–1	74·6

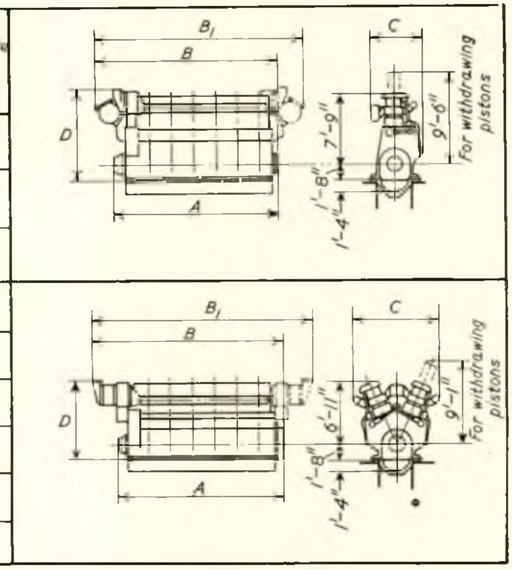


FIG. 1—*Overall dimensions of S.E.M.T. Pielstick marine engines Type PC2 400*

about 180 deg. C. (356 deg. F.) cooling by circulating oil under pressure has been used (Fig. 3). The oil passes up the connecting rod and through the piston pin and circulates in a steel coil cast into the piston. The return of the cooling oil to the bottom of the rod is by similar passages.

With regard to the valves the scavenge air flowing directly from the inlet valves to the exhaust valves during the overlap period keeps the temperature of the latter down to the level commonly encountered on less highly developed engines.

Mechanical Loads

The maximum combustion pressure at full load is between 90 and 95 kg./sq. cm. (1,280 and 1,350lb./sq. in.) which is only 70 or 80lb./sq. in. above that of the former PC1 engines where the power per cylinder was lower by about 40 per cent.

To obtain these results the injection timing and the compression ratio have been adapted to suit the higher performance and the system of pressure charging has been modified.

A careful study of the location of the turbochargers and

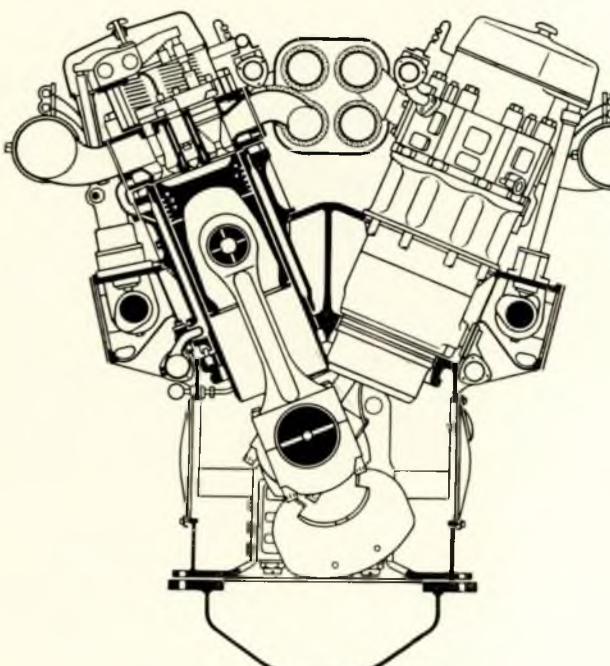


FIG. 2—*Cross-section of PC2V 400 engine*

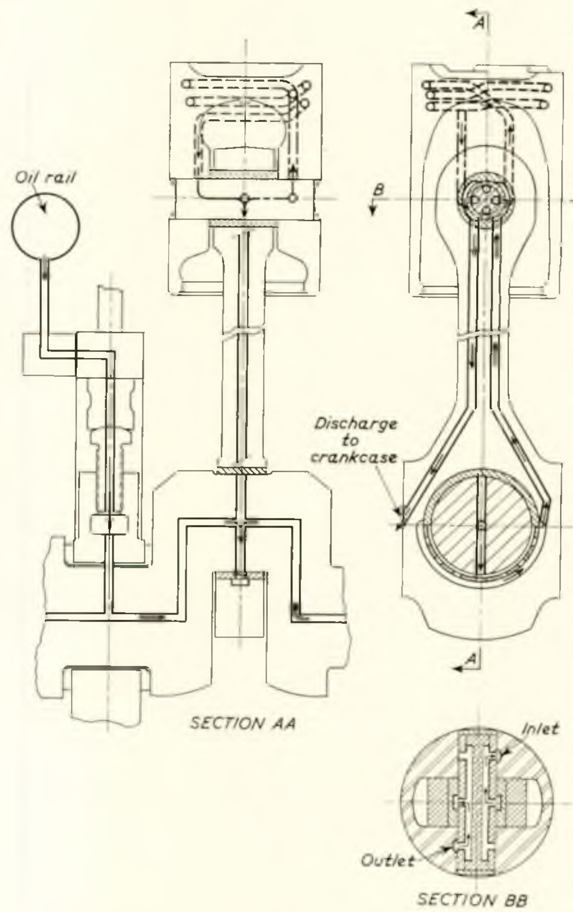


FIG. 3—*Piston cooling oil passages*

of the form and the length of the manifold has been carried out in order to make the best use of the pressure wave, that occurs on opening of the exhaust valves, and to obtain more effective scavenging.

The cam form and timing have been determined by a series of tests made with the object of achieving rapid emptying of the cylinder on opening the exhaust valves and so reducing as far as possible the power absorbed by the pistons during the exhaust stroke.

In order to compensate for this increase in combustion pressure of 80lb./sq. in. or so above that of the PC1 engine the diameter of the crankpins has been increased from 260 to 285 mm. The bearing pressure is actually lower than on the former engine, 4,835lb./sq. in. instead of 4,980. Similar modifications have been carried out on the main journals where the diameter has been increased from 270 to 285 mm.

Specific Consumption and Range of Operation

The specific consumption of PC engines in relation to the power per cylinder and the revolutions is shown in Fig. 4.

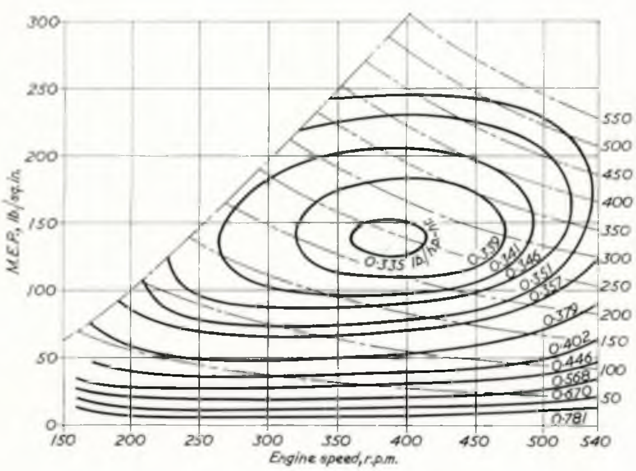


FIG. 4—*Fuel iso-consumption lines*

The consumption at full load is about 155 g./h.p.-hr. (0·346lb./b.h.p.-hr.) when the oil pump only is driven by the engine; it reaches 159 g./h.p.-hr. (0·355lb./b.h.p.-hr.) when all pumps (2 water pumps and 1 oil pump) are engine driven. The minimum consumption is in the neighbourhood of 150 g./h.p.-hr. (0·335lb./b.h.p.-hr.) and the speed range that can be covered in full fuel notch position extends from full speed down to about 60 per cent of full speed. This can be quite an advantage in the case of ships propelled by two engines coupled on to one shaft as it is possible, by running on one engine, to keep the specific consumption down to about the same as that under full load whilst slowing the speed of the ship to less than half speed as shown in Fig. 5.

This same figure shows the operating range in which one engine can be allowed to run, the ship's speed being reduced to only 70 or 75 per cent of full speed. With a variable pitch propeller even the regular service speed on one engine alone is about 80 per cent of the speed obtained on two engines.

PC ENGINES IN MARINE PROPULSION

Their advantages of weight, bulk, accessibility and ease of upkeep, together with their robust construction, render PC engines particularly well suited to marine propulsion. These engines can drive all their auxiliaries and have forced lubrication to all points requiring oil. A central control station where all controls and instruments are grouped may thus be installed in the engine room. This cabin can easily and with advantage be sound-proofed and so add to the wellbeing of the crew. All these possibilities show that PC engines will lend themselves especially well to automation.

It was considerations of this kind that decided the shipowning firm, Marlin Tanker Corporation, to choose for its new 50,000-ton tanker, a propulsion arrangement consisting of four PC engines geared to two shafts having variable pitch propellers. In fact the propulsion machinery in this case is controlled directly from the bridge by the master. The engine room watch is thus reduced to the strict minimum.

Actual Examples

Owing to the wide range of power covered by PC engines and to the possibility of installing one, two or four engines in one engine room, there are many different possible layouts, and it is always possible to find the one best adapted to the needs of shipowner and shipbuilder. This enables a shipping company to equip its entire fleet with engines of the same type, with the obvious advantage that this gives in the matter of spares, stores and particularly the getting together of engine room crews and officers, all trained in the running and maintenance of the one type of engine.

The figures which follow show several versions which have been adopted so far.

One Engine—One Reduction Gear and One Shaft

This is the arrangement in the cargo vessels *Labrador, Bruly* and *Bongo*. The engine is coupled to the shaft through a reduction gear. Between the engine and reduction gear there is a flexible coupling the purpose of which is both to absorb the torsional vibrations of the whole set and to protect the gear

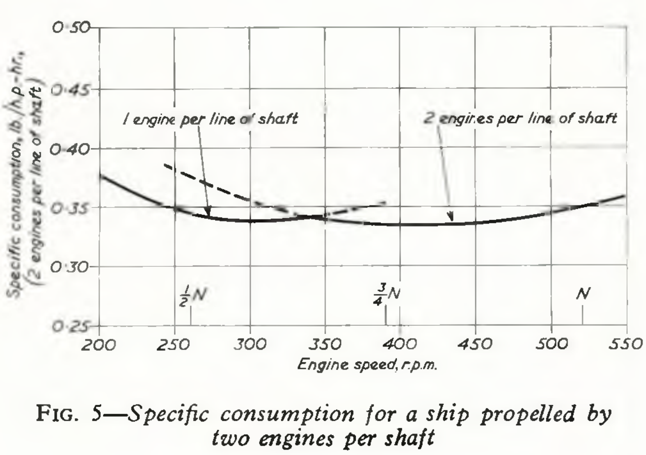


FIG. 5—*Specific consumption for a ship propelled by two engines per shaft*

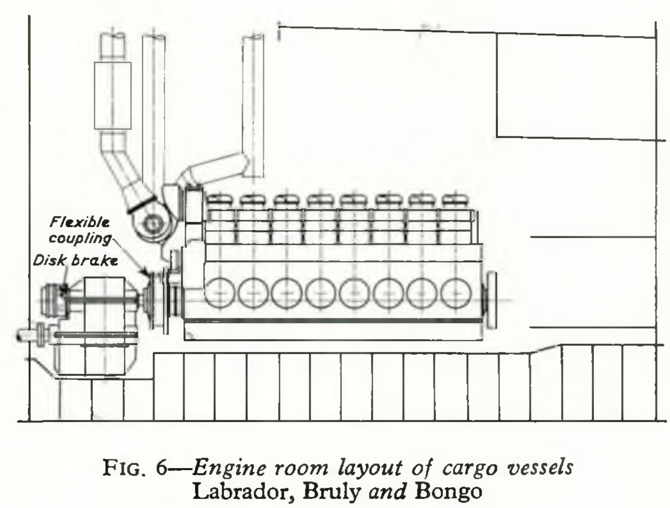


FIG. 6—*Engine room layout of cargo vessels* Labrador, Bruly *and* Bongo

teeth against irregularities in engine torque. Manœuvres are carried out using a main shaft brake mounted aft of the reduction gear. This brake is controlled from the central control station and works automatically by compressed air as soon as the stopping lever of the engine is brought to "stop". It is automatically released in the same manner when the engine is restarted.

A similar arrangement has served in converting engine rooms of several Liberty ships. The conventional reduction gear is replaced by a coaxial reduction gear and flexible coupling in some instances by a hydraulic clutch. The shaft brake is no longer necessary, manœuvring being carried out through the clutch or, in the case of a flexible coupling, by braking one rim of the coaxial reduction gear. The arrangement in the engine rooms of Liberty ships being what it was, the PC engine was one of the few engines with which this conversion could be made while retaining the main shaft and without altering the double bottom.

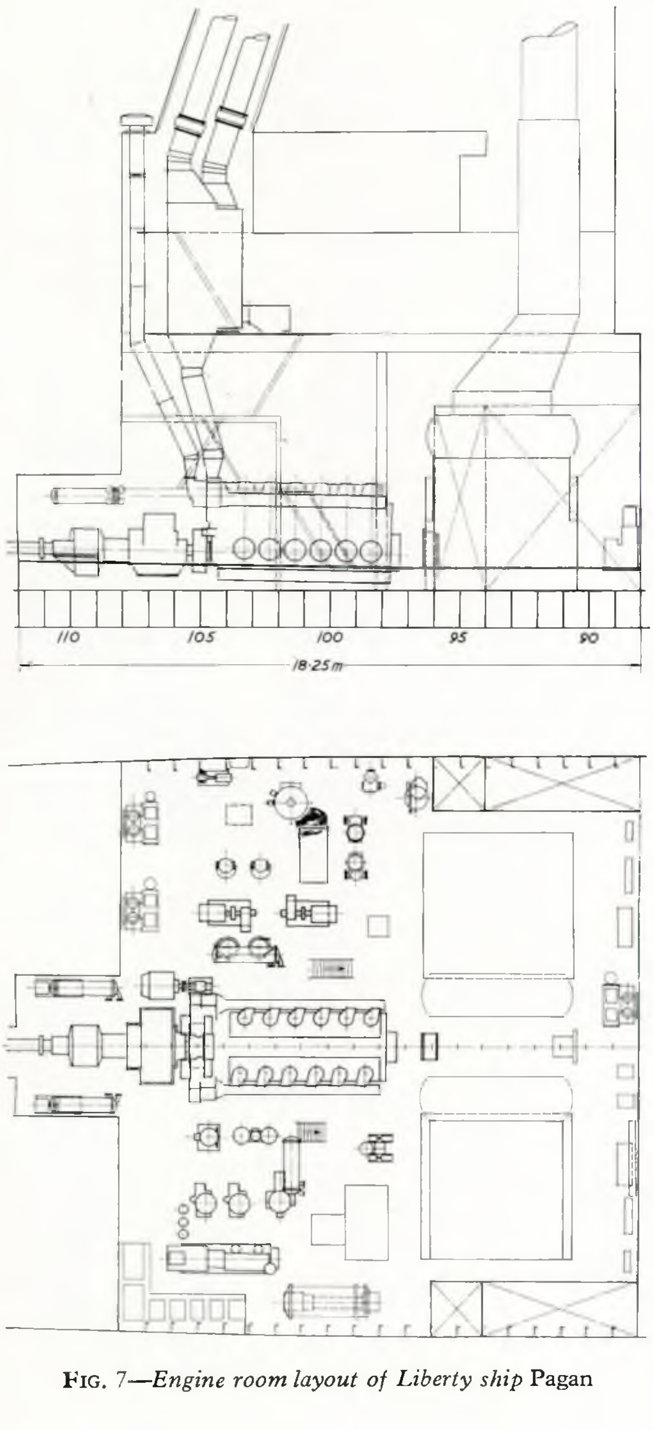


FIG. 7—*Engine room layout of Liberty ship* Pagan

One Engine for Each Shaft, Two Shafts, Variable Pitch Propellers

With this solution which has been applied to several cross-channel ferries and similar ships, the shafts and engines turn at a speed of about 350 r.p.m. and all manœuvring is carried out directly from the bridge by regulating the pitch of the propellers, engine speed being automatically adjusted to the propeller pitch. This kind of installation is of special interest in the case of ferries where, thanks to the small height of the engines, it is possible to have an extra deck.

Two Engines, Two Clutches, One Reduction Gear taking Two Input Shafts and One Output Shaft, One Propeller Shaft

This is the installation most often used. The couplings are either hydraulic or electro-magnetic. As in the case of the flexible coupling to one engine on one main shaft, these couplings protect the gear teeth and give better conditions regarding the torsional vibration inherent with Diesel engines.

This arrangement increases safety in operation because, with only one engine, a proportion of power is still available and the vessel can proceed at a speed reduced to only 70 or 75 per cent of full speed, or to only 80 per cent with variable pitch propellers.

The use of clutch couplings and two engines turning in contrary directions gives an elegant solution in manœuvring when entering or leaving port. The propeller may be coupled to one or the other of the engines, one of which has been started ahead and the other astern, there is thus no need for compressed air starts and repeated shifting of the camshafts.

Four Engines with Clutches and Reduction Gears, and One or Two Main Shafts

This installation is similar to the last but it provides for sets of engines developing over 20,000 b.h.p. on the shaft or shafts.

Seven ships of the Johnson Line, seven Dispatch-escort ships of the French Navy and one lighthouse tender are equipped in this way. The 50,000-ton tanker of the Marlin Tanker Corporation is also, but she has variable pitch twin screws as well.

Auxiliary Generating Equipment

In many cases it pays to fit generators of considerable power to the free ends of the engines or to the high speed shafts of reduction gears.

These generators can be switched into the ship's circuit while on cruise voyages. If the engines have clutch couplings the generators can be used to provide current for cargo or supply pumps or other machinery requiring a high power on occasion. In this case the arrangement can reduce significantly the installed power of the ship's generating sets and so the installation, maintenance and running costs and the engine room size.

Such generators are fitted at the free ends of the two after engines of the ships of the Johnson Line. They are in continuous use at sea, together with the ships' generator sets. In the case of the 50,000-ton tanker, similar generators only work in port when discharging or taking on board the oil cargo.

In the case of the new liner currently building for the Compagnie Générale Transatlantique, the propelling machinery will be composed of four in-line 8PC engines and the generators will be replaced by alternators. To allow these alternators to be permanently connected to the distributing system on board, variable pitch propellers will be used and the speed of the engines kept constant. It will also be possible by disconnecting one of the engines from the propeller shaft to use this engine to supply the current on board when the ship is alongside the

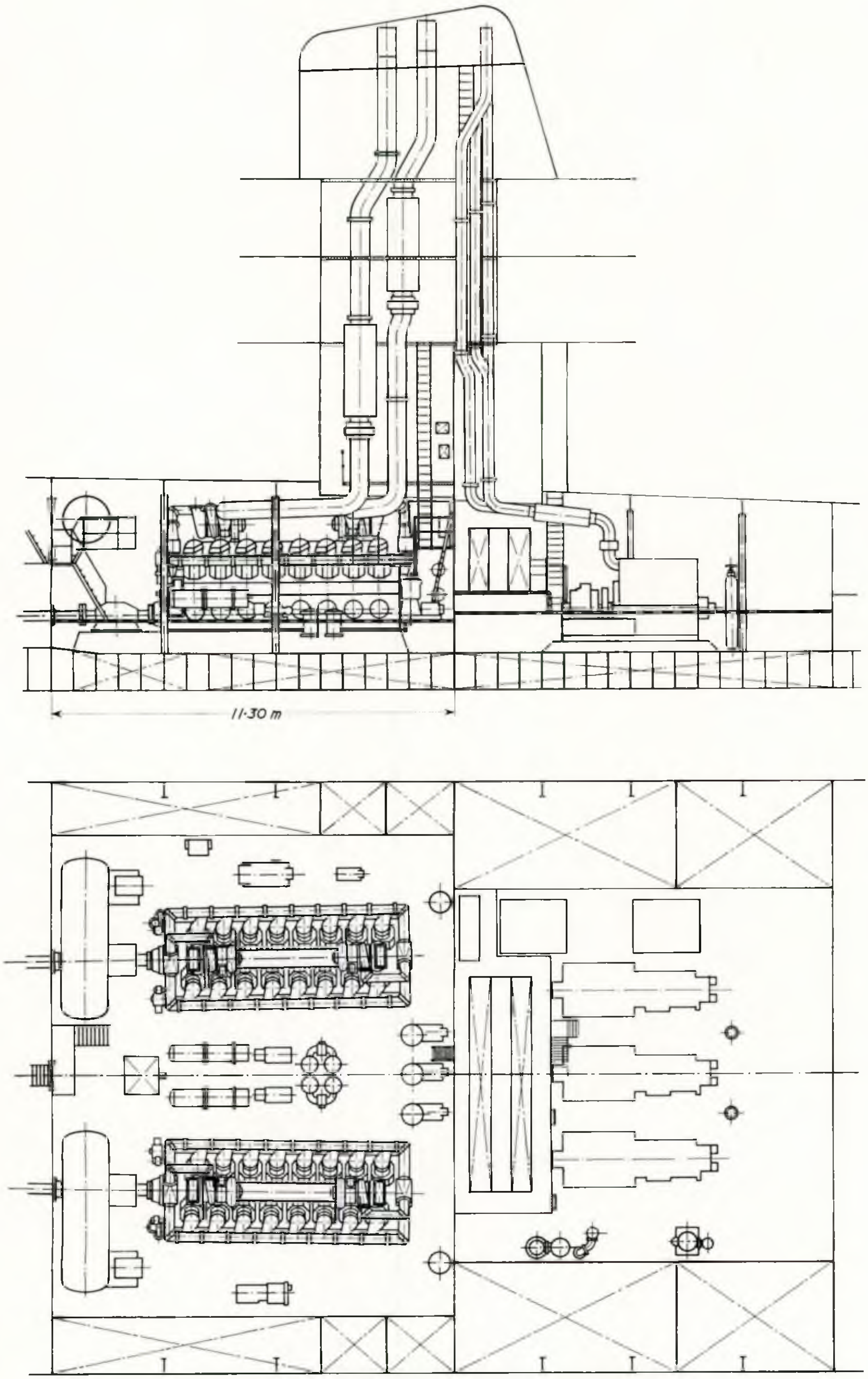


FIG. 8—*Engine room layout of the car ferry* Compiegne

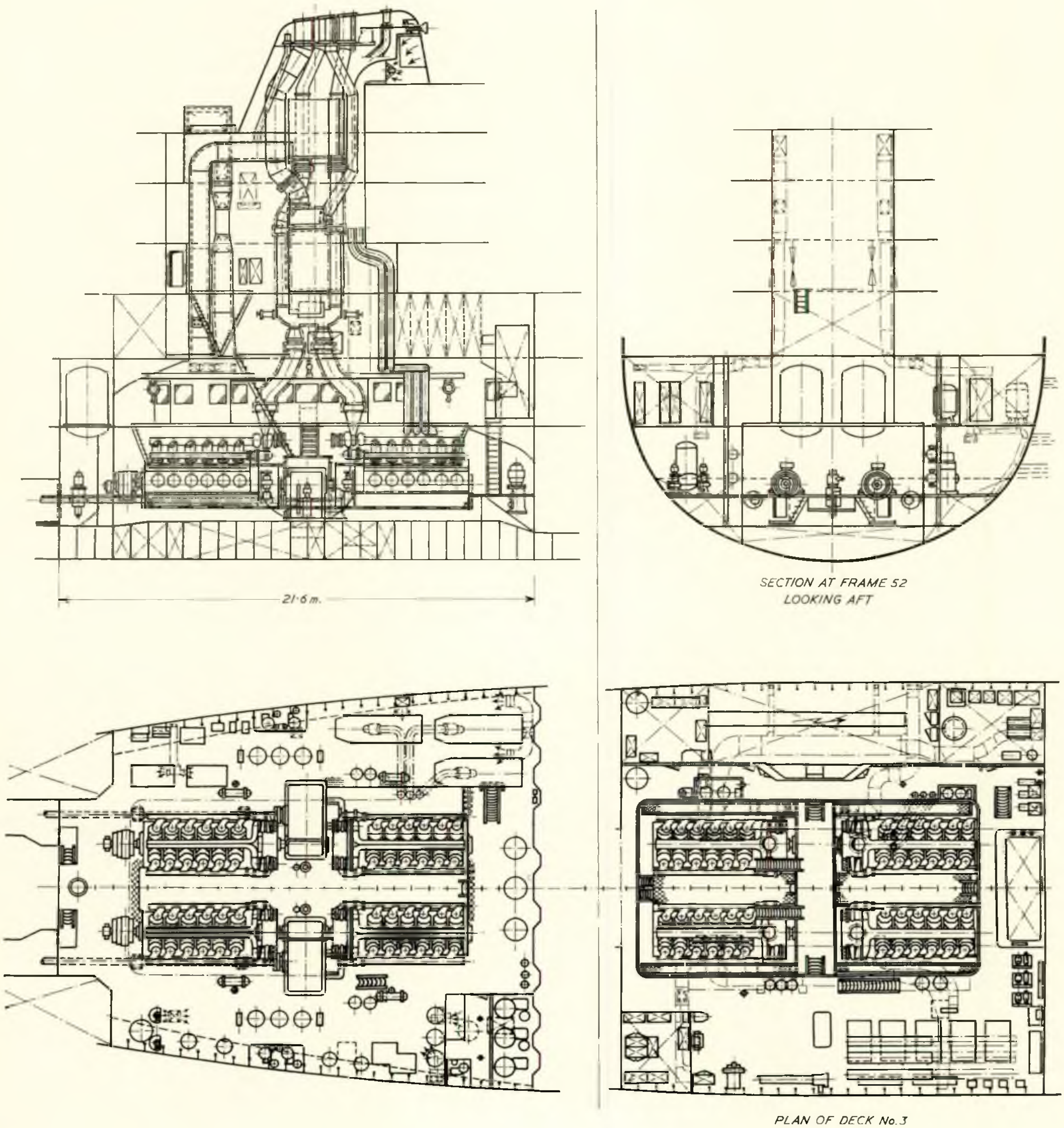


FIG. 9—*Engine room layout of the* Rio *Class ships*

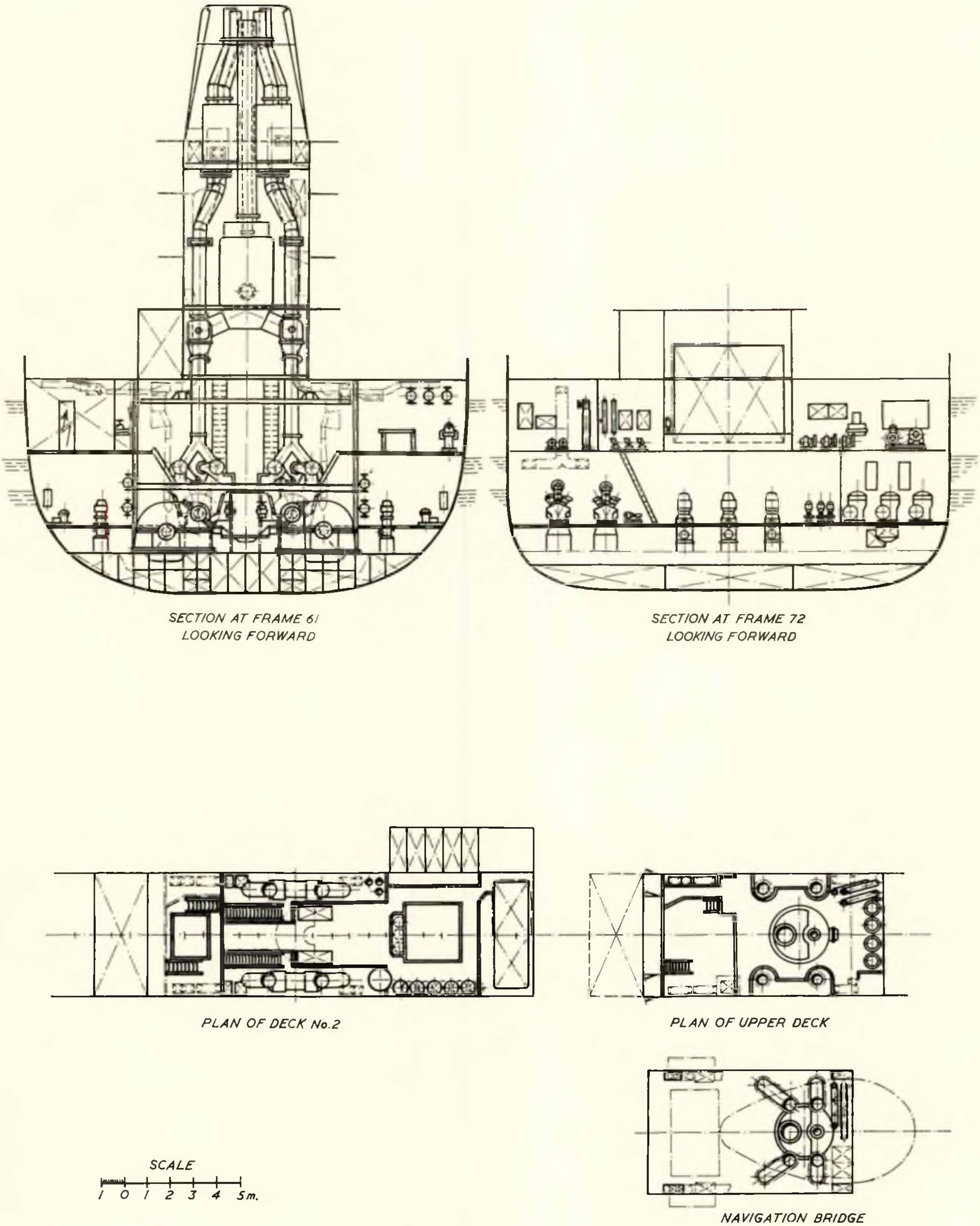


FIG. 9—*Engine room layout of the* Rio *Class ships*

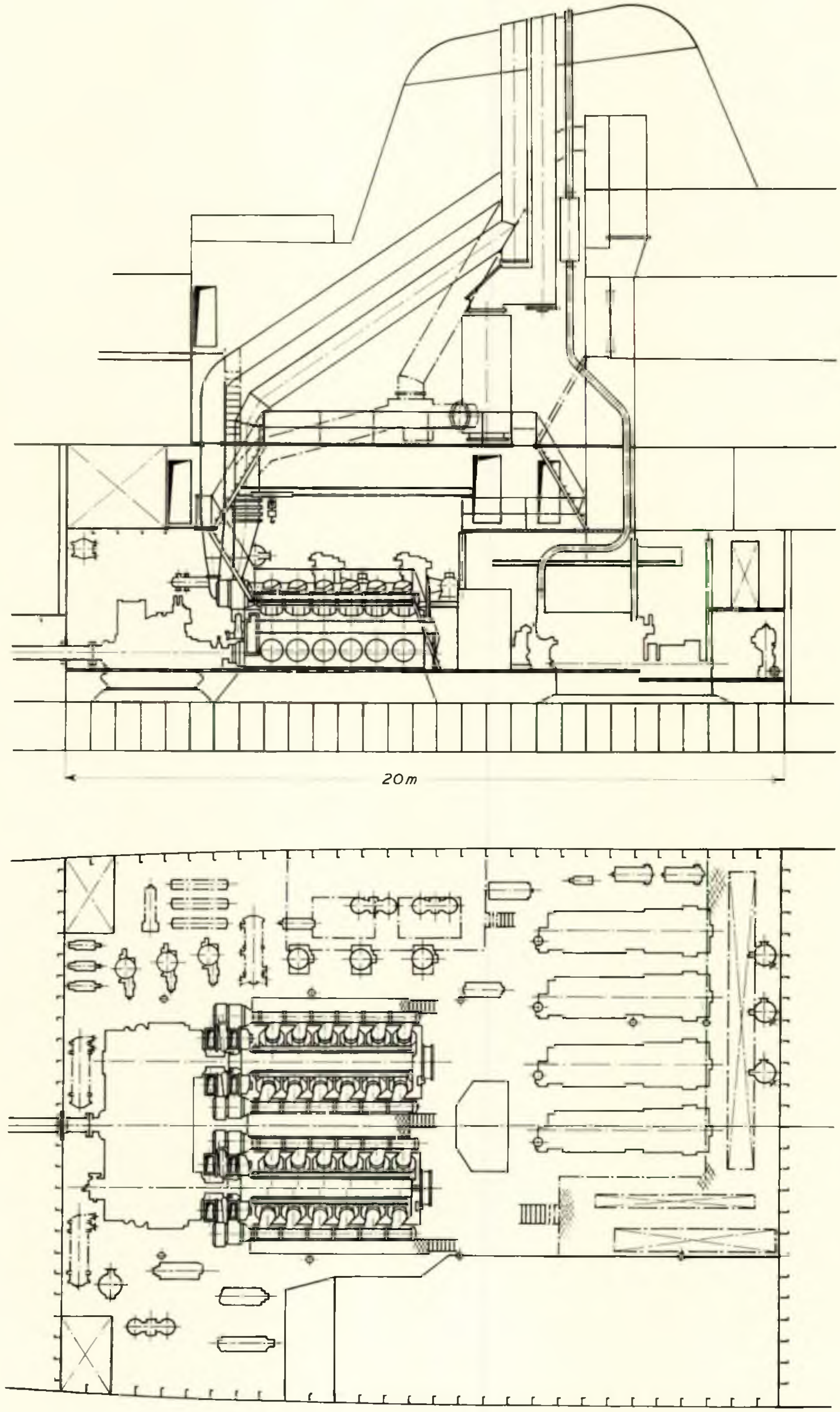


FIG. 10—*Engine room layout of the banana boat* Tarpon

quay. In this way the generating equipment aboard can be reduced to the strict minimum, i.e. to one auxiliary set.

Special Case of Double-duty Engines (Trawlers)

Another kind of installation was recently worked out for a trawler-factory which required large amounts of electric current. There is one engine which is coupled through an hydraulic coupling and a reduction gear to a single variable pitch propeller. In addition this engine drives an alternator connected in parallel with the ship's a.c. generators, thus reducing their number and size. When high power is required for propulsion the a.c. generator sets can in their turn feed the alternator and thus increase the power to the propeller.

The propeller pitch is adjusted so that when running free or during trawling the engine speed is always 428 r.p.m. so that the alternator can be connected to the other generators.

Diesel Electric Propulsion

In this case Diesel generating sets can be installed in selected locations where they do not detract at all from the useful cargo space. The light weight of the engines will even allow of their being on two levels at the after end of the ship.

This method of propulsion has been adopted by the Government of the U.S.S.R. for a dredger of 2,800 cu. m. (9,888·1 cu. ft.). On this dredger, four six-cylinder engines each drive constant speed generators, the current from which is transmitted to two electric motors during propulsion or to motors driving pumps during dredging.

Diesel electric propulsion has also been chosen for three refrigerating trawlers for the U.S.S.R. Each trawler has a set of propelling machinery consisting of three 6PC2L engines, each of which drives a generator. Each one of these generators can supply current both to the propelling motor and to the electrical system on board.

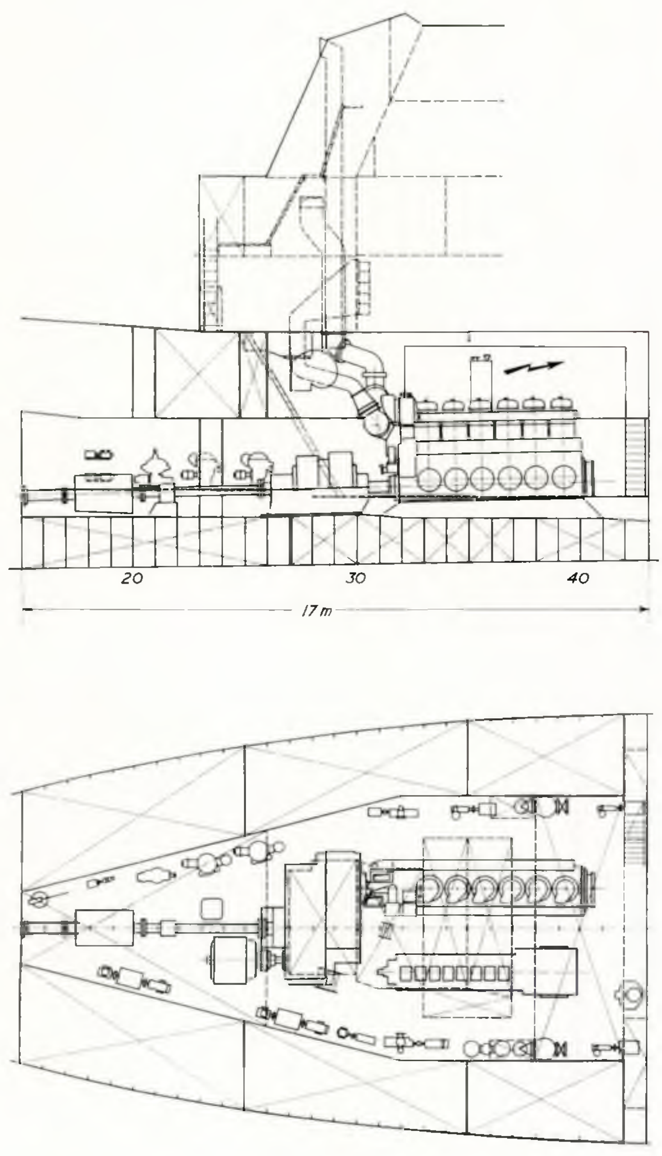


FIG. 11—*Engine room layout of the trawler-factory* Colonel Pleven II

OPERATING RESULTS.

Maintenance of Engines in Service

Maintenance of PC engines is greatly facilitated by the light weight of most of the parts and the ease with which they can be removed. The heaviest piece—the complete cylinder head, together with valves, housing and valve rocker mechanism—weighs 460 kg. (9·2 cwt.). A complete assembly of piston, piston pin and rings connecting rod and bolts, weighs 330 kg. (6·5 cwt.). The exhaust valves, which may be removed without taking down the cylinder head, can be taken out, by hand, by one man.

This gives an idea of the relative ease of taking down, repairing and reassembling.

Maintenance Schedule

The main upkeep operations which recur at regular intervals of time are as follows:

a) Every 1,250 hours replace injectors (one man half an hour per cylinder), the injectors removed being reconditioned afterwards.

b) Every 5,000 or 6,000 hours remove one or two pistons in order to see the state of the piston rings and of the liners (five to six hours per cylinder for two men). This is to make sure that the lubricating oil remains in good condition and that the next period of 5,000 or 6,000 hours may be undertaken.
Replace exhaust valves of all cylinders (1½ hours per cylinder for two men), the valves removed being reconditioned afterwards.

c) Every 10,000 or 12,000 hours remove pistons and renew the top piston ring (six to seven hours per cylinder for two men).

d) Every 15,000 to 20,000 hours check one or two main bearings (four hours for two men).

In the case of propulsion installations comprising several engines, it is possible to carry out all this maintenance without stopping the working of the ship, providing that it can be done during a period when full power does not have to be used.

Furthermore, the time an engine is out of use is reduced to a minimum, if the maintenance is carried out piston by piston, the work being spread over several trips, so as not to hinder the movement of the vessel. A stretch of 1,000 to 2,000 running hours between inspection of the first piston and of the last is currently observed.

Finally, very often, following good results obtained, the foregoing schedule is not strictly followed, particularly for exhaust valves which are generally only inspected when the pistons are taken out. Again, regarding the latter, one shipowner now intends to take the pistons out only every three years, that is after 18,000 to 20,000 hours.

Wear

Wear figures recorded on PC engines refer now to running times of the order of 30,000 hours and therefore can be extrapolated with confidence.

Fig. 13 gives the wear which has just been measured on the two 8PC engines on the cargo vessel *Galatee* after 28,000 hours' service. The values are particularly interesting because they are, amongst those known, those which refer to the largest

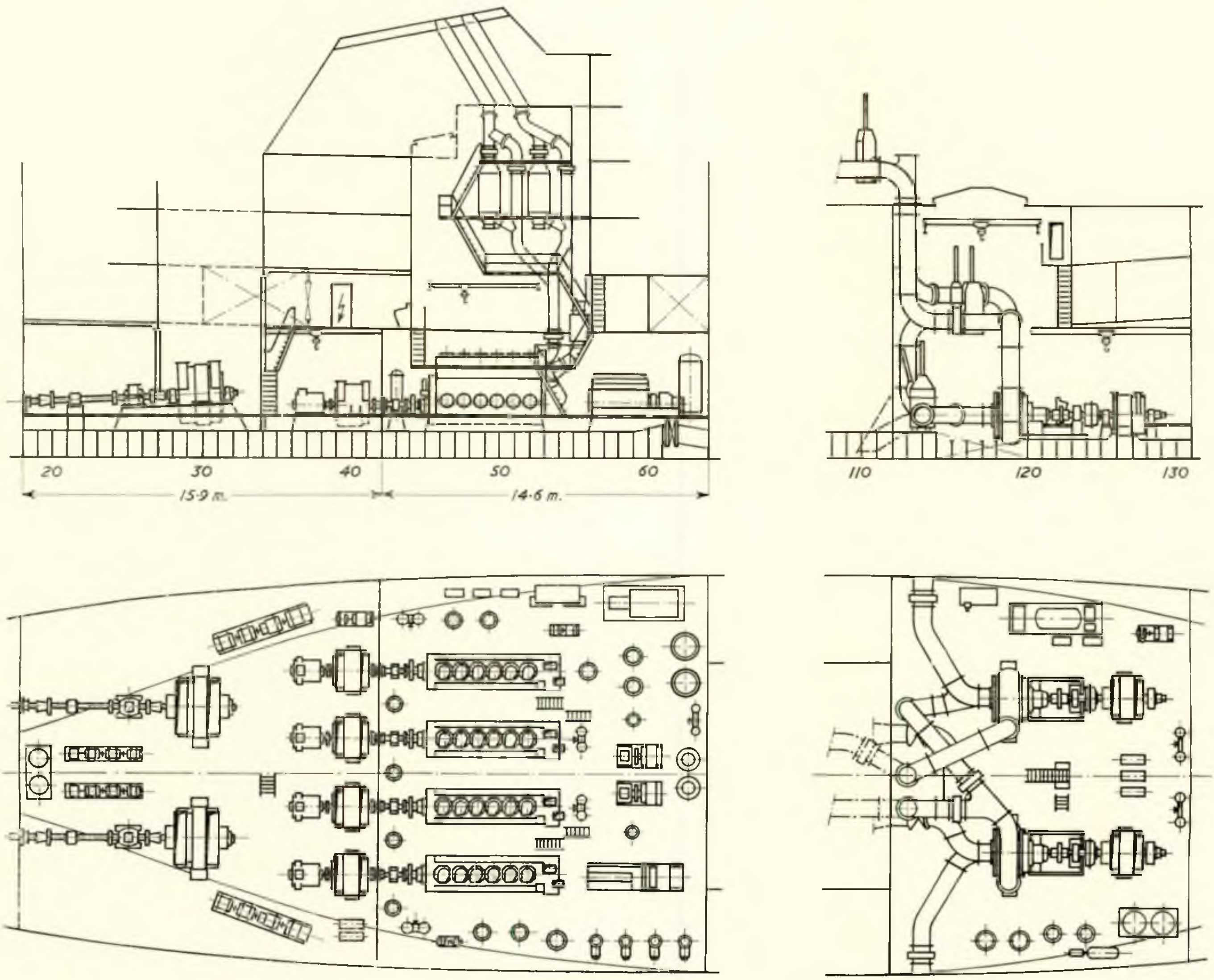


FIG. 12—*Engine room layout of a Diesel electric vessel*

number of running hours. Moreover, they fit in well with the measurements made on other PC engines.

The diametral wear rate on the liners at the top dead centre position of the top ring is usually of the order of 0·0008in./1,000 hr. In the case of the *Galatee,* the maximum wear rate is 0·0004in./1,000 hr. and the minimum wear rate only 0·0002in./1,000 hr. No PC engine liner has yet been replaced because of wear and it is therefore difficult to say what the actual life is but probably more than 100,000 hours of running.

On main bearings and connecting rod bottom ends, wear remains in the region of zero, the protecting flashing of anti-friction metal (0·0015 to 0·002in. thick) still being visible over almost the whole of the surface.

As regards the crankpins the figures recorded show a decrease of diameter of the crankpins or journals reaching a maximum of 0·0016in. and an ovalization of 0·0008in. after a running time of approximately 28,000 hours.

The wear rate of the top ring groove is less than 0·002in./1,000 hr. The first re-cutting of this groove is usually carried out after 10,000 hours. As the groove can be increased successively from 10 to 10·5 mm. then to 11 and 11·5 mm. the life of the pistons is of the order of 40,000 hours. In the case of the engines on the *Galatee,* the wear appears greater than that just quoted, but this arises from the fact that when the two successive remachining operations were carried out on the top groove more metal was removed than was necessary.

In a case where the rate of wear is considered too high, it is still possible to reduce it by using a ring bearer cast in the piston (this device is adopted for running on heavy fuel and is referred to later in this paper). It was adopted by way of experiment on three pistons on the *Galatee.*

These three pistons which have now run more than 15,000 hours, each have a wear of the top groove of only 0·004in.

The average wear rate of the piston pins is 0·00003in./1,000 hr. which gives for these parts, where the permissible wear is of the order of 0·004in., a life longer than 60,000 hours.

The wear rate of the bronze bushes in the small ends is of the order of 0·0002in./1,000 hr., 11 bushes out of 16 are still within the machining tolerances on the 8PCL engines in the *Galatee.*

The small increase in clearance between the timing gear teeth, 0 to 0·008in. in 28,000 hours, shows that the wear of these parts is low.

Finally, the wear of the valve spindles and their guides shows that the life of these parts is far longer than the present 28,000 running hours.

RUNNING ON HEAVY FUEL

Among most shipowners, shipbuilders and engine builders the view is prevalent that it is impossible to burn heavy fuel oil in a four-stroke engine with trunk pistons. This assumption is based on the following opinion: because the engine housing is not positively separated from the combustion gases, the lubricating oil must quickly become contaminated by contact with the acid products of combustion passing the piston rings and thus contain, in more or less high concentration, substances capable of seriously corroding both the copper lead main bearings and bottom ends as well as other moving parts of the engine.

Early tests made at S.E.M.T. and also later tests on engines in service, including some which have put in about 9,000 hours' running, involved frequent cooling off and reheating of the oil and gas contents of the crankcase, yet in spite of this no incident whatever has occurred which could in any way be traced to deterioration in the quality of the lubricating oil. It is also of interest to note that in one installation where the sequence of operation required starting-up and stopping the engine once a day, with the maximum risk of condensation of acid products in the crankcase upon cooling off, not the slightest sign of corrosion could be found.

Perhaps this can be explained by the excellent combustion in the PC engine which results from a favourable combustion chamber shape combined with a large amount of excess air. This tends to reduce the causes of piston and valve fouling, and this in turn reduces the work of dispersal and oxidation-inhibiting that the lubricating oil has to do, so allowing it to preserve its additives in an active condition for a long time. An additional explanation may be that piston temperature, even at full load, is low, enabling the piston rings to make a very good seal, thus only relatively small quantities of gas can reach the crankcase and therefore the amount of acid substances to be neutralized by the lubricating oil is sufficiently low to be accepted by the oil without deterioration.

Finally, it is true that progress made by refineries in the development of anti-oxidizing and dispersive additives has helped considerably by making it possible to have a very high concentration of these products in the lubricating oil.

Regarding the problems met with in the injection of very viscous fuel oils, these have been solved completely by the conventional methods of heating and centrifuging the fuel oil.

Heating of the fuel is carried out to a temperature such that the viscosity of the oil on arrival at the injection pumps is of the order of 115 sec. Redwood No. 1. This is done in double piping, either using steam if the installation can provide it, or using exhaust heat boilers. In this latter case heating is by hot water and if it is necessary to exceed 100 deg. C. (212 deg. F.) the system is pressurized by compressed air. It is usually an advantage to provide a viscosity control system which automatically regulates the temperature and flow of water or steam so that viscosity in the pumps will be maintained at the desired figure.

All piping which is not heated is suitably heat insulated.

The purifying system consists of one or two centrifuges working as clarifier and separator and one double filter, which is also heated. Tests are currently being made at S.E.M.T. using only one or two filters designed for the purpose and suitably heated to see whether such a less complicated installation would serve equally well.

Engines in Service

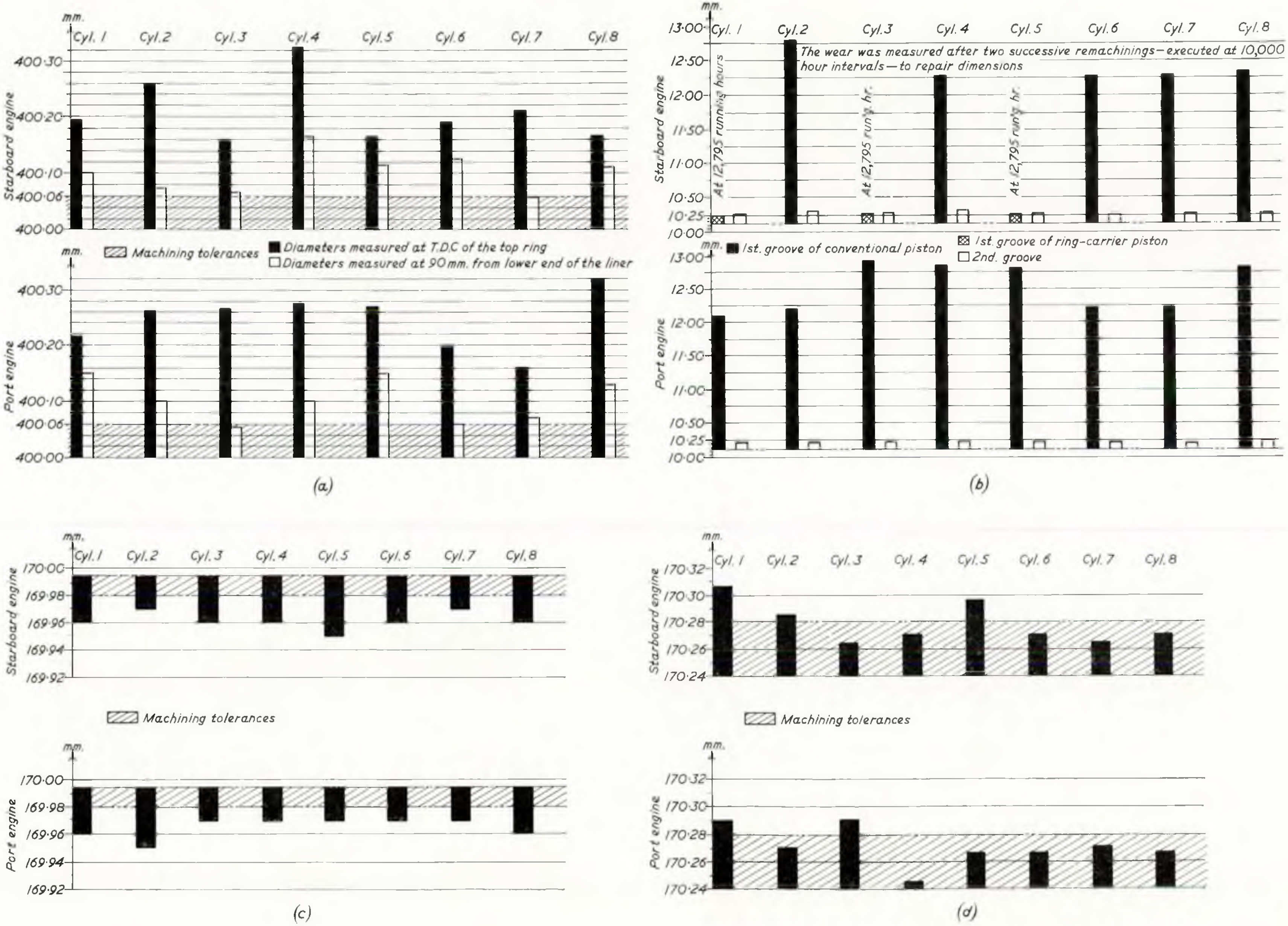
Nine ships are at present fitted with PC engines burning heavy fuels:

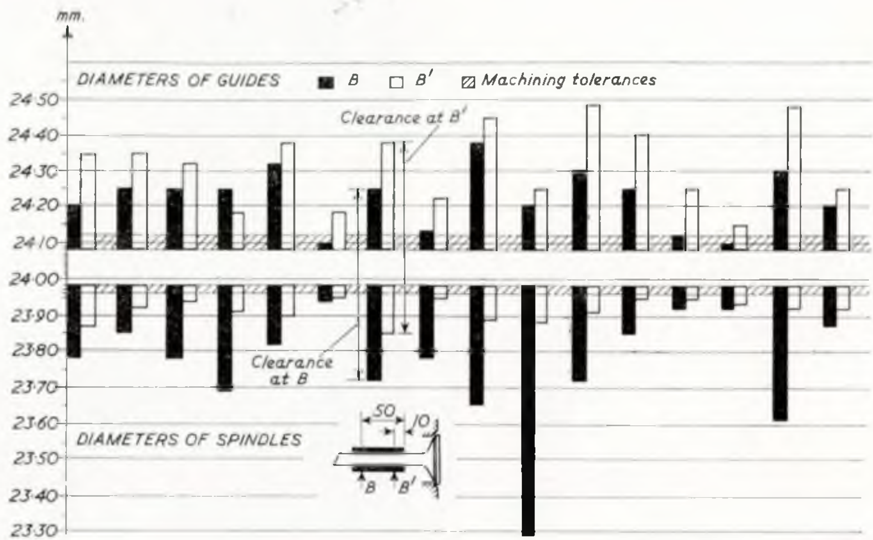
- three ships each of the Johnson Line with four 12-cylinder engines;
- one Zim ship, with two nine-cylinder engines;
- four Zim cargo vessels, each with one 12-cylinder engine;
- one l'Union Navale cargo vessel with two 12-cylinder engines.

Table I summarizes the number of running hours of the

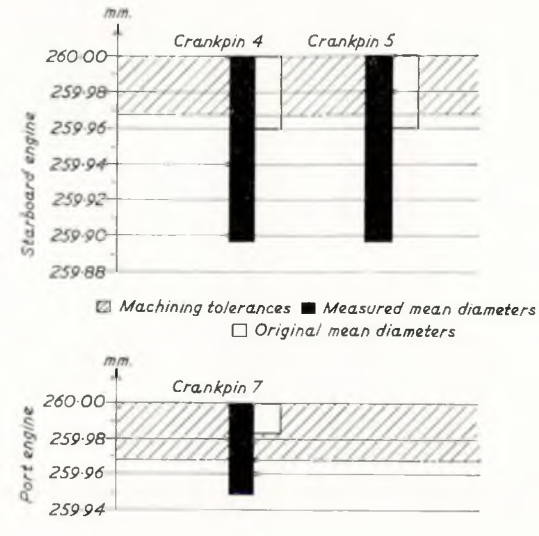
TABLE I—PC ENGINES. RUNNING ON HEAVY FUEL. (1—9—1963).

Name of ship	Number and type of engine	Number of hours to 1-9-1963		Heavy fuel oil					Lubricating oil		Wear in 1/1000 in. per 1,000 hr.
		Total	Heavy fuel	Viscosity sec. Redw. No. 1 at 100 deg. F.	Sulphur, per cent.	Ash, per cent.	Vanadium p.p.m.	Sodium, p.p.m.	Brand	Grade	Cylinder liner
Montevideo	4×12PC1V.	19,300	13,200	200–600	1·9–2·6	0·04	160–200		Mobil	Mobilgard 493	0·449
Moledet	2× 9PC1L.	12,000	10,000	700	0·7	0·02	15–45	25–45	Mobil	Mobilgard 493	0·204
Pagan	1×12PC1V.								Shell	Rimula 40	
Pan	1×12PC1V.								Shell	Rimula 40	
Velos	1×12PC1V.								Shell	Rimula 40	
Fenice	1×12PC1V.	7,050	3,000	550–1,500	2·6	0·025–0·09	60–160	30–45	Shell	Rimula 40	0·163
Buenos Aires	4×12PC1V.	23,950	2,500	200					Mobil	Mobilgard 493	
Rio-de-Janeiro	4×12PC1V.	23,800	3,950	200					Mobil	Mobilgard 493	
Carolia	2×12PC1V.	3,000	1,000	1,100	2·8	0·02	72	25	Shell	S.6524	
Test engine	1× 2PC2		3,700	3,400	3·2	0·09	195	90	Shell	S.5819A	1·225

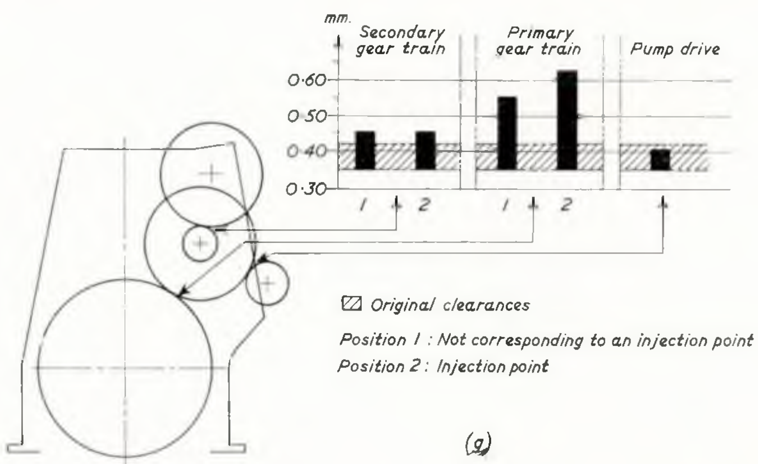

mm.
Cyl. 1 Cyl. 2 Cyl. 3 Cyl. 4 Cyl. 5 Cyl. 6 Cyl. 7 Cyl. 8
Starboard engine
Port engine
400·30
400·20
400·10
400·06
400·00
Machining tolerances
Diameters measured at T.D.C of the top ring
Diameters measured at 90 mm. from lower end of the liner
(a)
13·00
12·50
12·00
11·50
11·00
10·50
10·25
10·00
At 12,795 running hours
At 12,795 run'g. hr.
The wear was measured after two successive remachinings—executed at 10,000 hour intervals—to repair dimensions
1st. groove of conventional piston
1st. groove of ring-carrier piston
2nd. groove
(b)
170·00
169·98
169·96
169·94
169·92
(c)
170·32
170·30
170·28
170·26
170·24
(d)



(e)



(f)



(g)

a) Cylinder liners
b) Piston ring grooves
c) Piston pins
d) Top end bushes
e) Exhaust valve spindles and guides
f) Crankpins
g) Timing gear clearances

FIG. 13—*Wear measurements of* Galatee *8 PC1L engines*

engines on these ships since they commenced operating on heavy fuel, the maximum number of hours being 13,200 on the four engines of the cargo vessel *Montevideo* and 10,000 on the two engines on the vessel *Moledet*. Results in service depend appreciably on the composition of the fuel and the ash content. The latter varies very little with viscosity since a fuel of 3,500 sec. Redwood can be converted to 600 sec. Redwood by adding only 25 per cent Diesel oil; the ash content is chiefly a function of the place of origin.

Results in Service

Exhaust Valves

The only difference of condition observed on the PC engines when running on heavy fuel in relation to engines using high grade fuels is in the condition of the exhaust valves. Whilst these are taken down after only 5,000 to 10,000 hours when the fuel is Diesel oil, it is at present necessary to check them usually every 1,500 to 2,000 hours when the fuel used is a heavy fuel.

The problem involved is not one special to PC engines but is a general one.

The inspections are in fact necessary because of risk of attack of the seatings by corrosive salts with a vanadium and sodium base. If these salts, formed during combustion, are carried at a temperature greater than 550-600 deg. C. (1,022-1,120 deg. F.) they remain in a liquid state and in depositing themselves on the heads of the valves can attack and corrode the seats.

S.E.M.T. has thus been led as a first stage of development to:

a) employing different facing metal for the seats and,
b) to asking users to check the condition of the valves every 1,500 to 2,000 hours.

This position has not, however, been adopted uniformly by shipowners, as the example of the two installations having the greatest number of running hours shows.

Whilst the Johnson Line dismantle their valves systematically after running hours of the order of 1,250 to 1,800, the chief engineer on the vessel *Moledet* prefers to replace the valves only when they have been burnt. In this latter case maintenance work is reduced to a strict minimum. Certain operating personnel have as their only task to watch the individual exhaust temperatures on each cylinder, and as soon as one of these temperatures increases appreciably above the average value of the others he has to point this out to the chief engineer. The latter can then either continue to the next port where the vessel must put in and change the two valves in the cylinder at fault (the allowable running time from the moment when the exhaust temperature begins to increase to the moment when it attains an excessive value, is of the order of 40-50 hours), or, if the next port is a long way away, decrease the rack setting of the fuel pump of the cylinder having a higher exhaust temperature and then carry on under these conditions. Sometimes, if the schedule of the vessel allows it, the " Chief " stops the engine on which the valves are to be replaced and the vessel continues its journey on one engine alone during the period required for fitting the two new valves.

However, in the case of the *Moledet* one of the two valves taken down, after an average running time of 3,000 hours, is always found burnt and must be remetalled, but on the *Montevideo* burning is exceptional and at least 90 per cent of the valves can be reconditioned by light grinding on their seats.

These results, however, are not regarded as completely satisfactory, and studies and research are being carried out on a test engine, to increase the time between inspections from 2,000 to 5,000 hours. To this end an area has been fitted out with the special equipment (for heating, centrifuging, etc.) required for heavy fuel operation. The tests, which are directed to lowering the valve temperature and improving the valve seat facings, are at present being carried out with a specially refined fuel having vanadium and sodium contents representative of the worst commercial fuels. These contents are:

190 p.p.m. of vanadium—90 p.p.m. of sodium.

All these tests are being run at a power of 465 metric h.p. per cylinder at 450 and 500 r.p.m. (b.m.e.p. = 15·5 and 14·5 kg./sq. cm., 222 and 206lb./sq. in.).

Other Parts

Results obtained in service on all other engine components are quite remarkable.

The bottom end bearings, main bearings, top end bushes etc., show the same very slight wear that is found on engines running on Diesel oil.

Cylinder liner wear does not appear to be generally any higher than when running on gas oil. Using commercially available lubricating oils of the type offered specially for the cylinders of two-stroke engines burning heavy fuel, but of class S.A.E. 40 or S.A.E. 30, the maximum wear rate of the liner at the top of the top ring travel is still less than 0·0008in./1,000 hr.

On the four engines on the cargo vessel *Montevideo,* this wear is on an average only 0·00045in./1,000 hr. after 9,000 hours' running on heavy fuel (the latest set of wear measurements having been taken at that time).

On the *Moledet,* after 8,000 hours' running on heavy fuel, it is only 0·0002in./1,000 hr.

These very interesting results were obtained using normal liners, standard for PC engines and having no special protection, surface finish, or other treatment of the bore.

As regards the rings, no abnormal wear has been found of the chromium-plated surfaces which bear on the liners and the life of the top rings remains the same as when the fuel is of the Diesel oil class.

The first test bed trials on heavy fuel showed abnormal wear in the top ring groove. The increase in axial clearance was of the order of 0·004in./1,000 hr. and therefore hardly acceptable. The solution adopted by S.E.M.T. to overcome this difficulty was to replace the standard pistons by pistons with ring bearers inserted at the time of casting. With such pistons, the wear is reduced to an extremely low rate: 0·0004-0·0006in./1,000 hr. This is an excellent result and a piston life of well over 40,000 hours can now be expected.

Another solution, mentioned earlier, is also in course of being tried out; the piston, still of aluminium alloy, contains 18 per cent silicon instead of the normal 12 per cent. The hardness is therefore greater. Cooling, on the other hand, is of the shaker type i.e. of the same type as on the PA4 engine. The drop in temperature of the top groove, linked with the increase in hardness of the alloy, enables wear to be limited to 0·002in./1,000 hr., which is acceptable.

The clean condition of the pistons on removal is also another important feature which should be mentioned. The rings were perfectly free in their grooves and the skirts had very

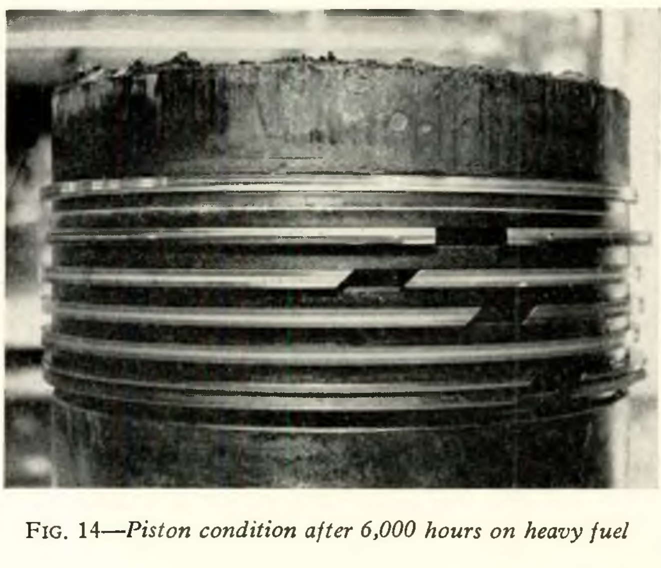


FIG. 14—*Piston condition after 6,000 hours on heavy fuel*

little carbon deposit as shown in Fig. 14, which shows a piston removed after 6,000 hours' running on heavy fuel.

The first pistons removed from engines in service after 5,000 running hours or so have been so surprisingly clean that owners have usually been able to decide to defer removal of other pistons until after 8,000 or 10,000 running hours.

No wear or burning of the piston crown is apparent.

These results, though due to a large extent to the particular design of PC engines, have been helped considerably, as already stated, by the use of oil with a high additive content and therefore more expensive than conventional detergent oils (by about 20 per cent). It is therefore necessary to ensure that savings from burning heavy fuel should not be cancelled out even in part by more frequent oil changes.

Actual results have shown that with these new oils, after a running time of more than 13,000 hours it was not necessary to change the lubricating oil. Taking into account the lubricating oil consumption of PC engines in service, which is under 0·004lb./h.p.-hr. and the small capacity of the crankcase (2·2lb./h.p., of oil) these results indicate that the use of heavy fuel in Pielstick engines is just as economically interesting as in crosshead engines.

system which is well suited to this duty by reason of having cooled sprayers. The only special feature is the helix angle of the pump plunger groove.

Engine Performance

Engine performance depends of course on the nature of the gas used. The natural gas available at the test station of Gaz de France at Alfortville has the following composition:

CH^4 : 93·4 per cent
C^2H^6 : 4·2 per cent
C^3H^8 : 1·3 per cent
C^4H^{10} : 1·1 per cent

Using this gas, a mean effective pressure of 185lb./sq. in. can be achieved without detonation and without special cooling of the charge air—temperature of the water feeding the air cooler is 30 deg. C. (86 deg. F.). The power per cylinder of the PC engine when running on this gas is:

340 h.p. at 428 r.p.m.,
380 h.p. at 500 r.p.m.,

which makes possible the building of gas engines in powers up to more than 6,000 h.p. If the gas contains a greater proportion of heavy hydrocarbons the power or volumetric compression ratio may have to be modified to avoid risk of detonation.

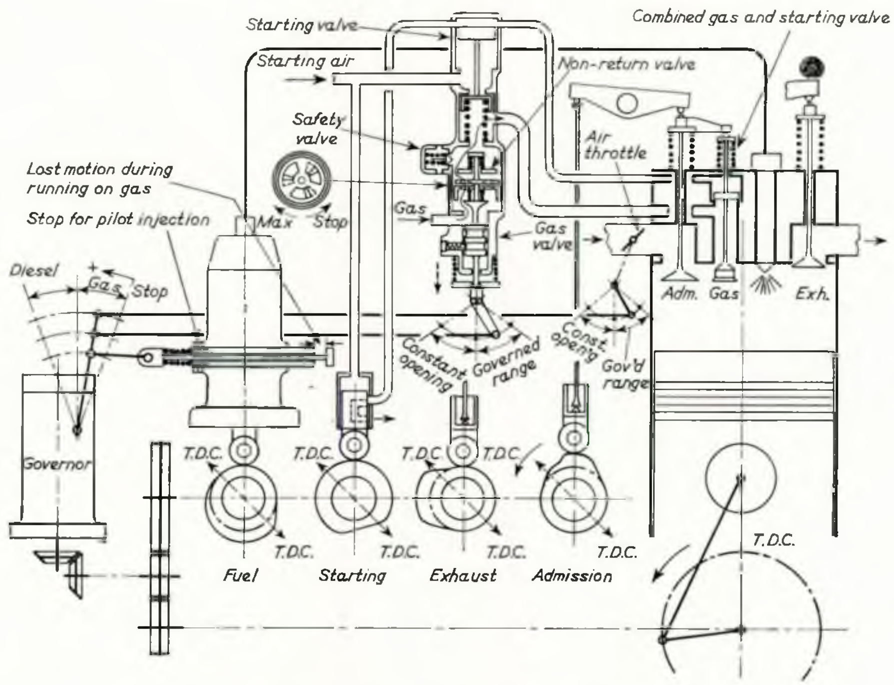


FIG. 15—*Diagrammatic arrangement for operation on gas*

RUNNING ON NATURAL GAS

Principles

A gas valve is fitted in each cylinder head. It has a constant opening and is operated by the inlet valve rocker gear. The profile of the inlet cam has been modified in such a way that the gas can only enter the cylinder when the exhaust valve is closed and yet the effective area offered by the valves remains sufficient for effective scavenging during the overlap period.

The quantity of gas entering the cylinder during each cycle is controlled by regulating a throttle valve at the entrance to a chamber situated immediately before the gas valve.

As the gas feed is at constant pressure, the quantity of gas admitted during each cycle is a function of the pressure in the chamber at the time.

The quantity of air is regulated by a butterfly valve. One butterfly valve is provided for each cylinder.

Pilot injection is carried out by the standard injection

In the case of electric power or compressor plant, the pilot injection quantity is constant at 5 per cent of the total heat input at full load.

Trials have also been carried out simulating the use of PC engines as propulsion units for methane carrying vessels which are at present being planned in several countries. In this case, the engines have to burn gases evaporated from the gas tanks.

Present information suggests that the amount of gas available is about 70 per cent of the fuel necessary for propulsion so pilot injection must provide the remainder.

The principle of governing, which has been adopted, deals with this situation very well as the percentage of liquid fuel is automatically increased or decreased to the right value as the supply of gas decreases or increases. No intervention by an attendant is required and all the evaporated gas can be burnt in the engine, provided of course that this quantity does not exceed the demand of the engine at full power. A further point of interest is the fact that during voyages with empty gas tanks, the propulsion engines can be run on heavy fuel.

In view of the low initial cost and the ability to burn natural gas or heavy fuel equally well, together with low maintenance costs, it is difficult to imagine a more economic solution than this to the problem of methane tanker propulsion.

ACKNOWLEDGEMENTS

The authors wish to acknowledge the help and co-operation of shipowners in providing information for this paper and to thank Société d'Études de Machines Thermiques and Crossley Brothers Limited, for permission to publish it.

Discussion

MR. K. H. TUSON said that, being personally most concerned with engines for power generation, he trusted he would be permitted to deal with the paper from that aspect even if it was somewhat outside the sphere of activity of the Institute.

He welcomed the advent of a new medium speed engine capable of developing up to 8,000 b.h.p. Many electricity undertakings overseas were faced with rapidly increasing loads and were in an awkward position if they were not large enough to use steam turbines which, in small sizes, were uneconomic, or for other reasons did not wish to do so. The remaining choice had hitherto been between a large number of small units, and slow speed engines running at 200 r.p.m. or less. Large numbers of units required considerable maintenance and supervision, whereas slow speed engines were extremely expensive and required high and costly buildings and sometimes difficult foundations. If medium speed generating sets developing 5 or 6 MW, having the necessary degree of availability and reliability, became available, they would fill an awkward gap.

He saw that the big end and main bearings were of the type commonly referred to as trimetal, which he had used with success to overcome difficult bearing problems. Had the authors found these readily available? Bearing specialists told him they were difficult to make and he had found delivery very prolonged.

Running on residual fuel was a fashionable topic at the moment, but some of the pros and cons were not always taken into account. If the use of Class G fuel reduced availability, the entire advantage in cost might be lost. At the top of page 458 the authors suggested adding 25 sec. oil to Class G fuel. The cost of the mixture became 70 per cent as against 63 per cent of that of Class A oil, and he suggested it would be easier and better to use 200 sec. Class E oil, the cost of which was 75 per cent of Class A.

He congratulated the authors on their endeavours to overcome the exhaust problem. The expedient of continuously watching temperatures as practised on the *Moledet* was somewhat primitive and would be impracticable in a large power station containing 100 cylinders or more. Monitoring pyrometers were now available which would continuously watch these temperatures and sound an alarm if a pre-set figure was exceeded.

Table I related mainly to PC1 type engines. Did the authors expect equally good results with the PC2 design? Liner wear in the test engine was much faster than in PC1 engines. He appreciated that a more difficult fuel was used, but felt that much more experience was required before residual fuel could be recommended with confidence for this design.

The necessary viscosity at the injection pumps, according to the authors, was of the order of 115 sec. Redwood No. 1, but Bailey and Cowdcray in a recent paper* to the Diesel Engineers and Users Association, considered it should not exceed 85 sec. Perhaps the authors would please comment.

MR. H. D. CARTER (Member) said that there was nothing new in geared Diesels for ship propulsion, and equally the direct-coupled slow speed engine had earned its place. He put forward the view that the modern trend, however, might well be along these lines. As the specific output of the geared Diesel engine improved by virtue of either increase of b.m.e.p. or speed, so it became a more formidable competitor. In the case of the Pielstick engine, it still operated at medium speeds, but the b.m.e.p. at 200 to 240lb./sq. in., according to speed and purposes, was a notable figure. It seemed to him that at some point, as the specific output of the geared Diesel engine increased, it must show a decisive advantage over the large direct-coupled job. Whether that point had now been reached was a matter of opinion.

He was, as might be gathered, in the same stable as one of the authors (Mr. Henshall). They felt that that stage had been reached and further that the engine had by no means reached the limit of its development. The attraction of the high specific output engine, geared, would, it was felt, increase in the coming years. In a nutshell, the case for the engine, therefore, was that of enabling a smaller ship to be designed to do the same work, or, alternatively, a ship of the same size to do more work—carry more cargo or more cars (if it were a vehicular ferry). There was a very special case for this engine for ferries; because of the low headroom the decks could be carried over the top of the engine room. With this geared Diesel engine there was a multiplicity of parts compared with the direct or slow speed unit; they were smaller parts and therefore each piece was more easily handled. However, there must be a very high degree of reliability before this type of installation could become a proposition and, to obtain that end, it was felt that the criteria were medium speed and high b.m.e.p. and perfected development.

One of the things that the Diesel engineer looked for was piston condition. The piston condition of these engines was something that never ceased to be a wonder to him. When the pistons came out of the engine after full load running they looked just as if they had been running on light load. This was frightfully important if one accepted that the piston was the heart of the Diesel engine. The good piston condition, as Mr. Henshall explained, was due in part to the system of pressure charging, whereby there was a very large cooling and scavenging effect during the overlap period at the top of the exhaust stroke, and to the excellent combustion. At the most there was a shimmer of hot air out of the exhaust pipe on the heaviest loads. This matter of running trunk piston engines on heavy fuel would be difficult to put over to marine engineers because of past history on quite different engines, big slow speed engines with cool crankcases. There had been severe troubles with corrosion of crankshafts and other parts due to condensed acidic products in the crankcase. However, with modern techniques the medium speed, high output engine was a different proposition. For one thing the crankcase was a good deal hotter than on the big slow speed engine of low rating in the early days, which gave trouble. The jacket water temperature was much higher than was acceptable in those early days. The lubricating oil was heavily alkaline and could be checked from time to time to see that it was still in good condition. The combustion was excellent and the products of combustion were swept right out of the combustion chamber

* Bailey, C. L. and Cowderay, J. A. 1964. "The Use of Heavy Fuel Oils in Medium Speed Engines". D.E.U.A. 295.

by the excess air. The life of the exhaust valves was understood. It was known at exactly what temperatures one could safely operate in order to avoid trouble with vanadium and sodium, and one knew, on the operating temperature accepted, what the life of the valve would be, so that the whole thing was under control. These were all felt to be very sound, substantial reasons which could be proved, so that it was not just a case of trying to do over again, what was done in the past. What was being done now was something quite different, and with good reason, if the fundamental causes were understood as to why the trunk piston engine should run successfully with heavier oil. The highest rating for these engines was the naval rating, where, in 18 cylinders, 9,000 b.h.p. was obtained—9,000 metric horsepower—so that with four engines it was possible to go up to 36,000 h.p.

Mr. S. Archer, M.Sc. (Member) said that the paper would be welcomed by many marine engineers who would be familiar with the name Pielstick but perhaps had had no opportunity of examining actual service results. From these, as now reported by the authors, covering up to 24,000 hours of service, it was clear that this medium speed, high duty, four-stroke engine was capable of giving good economy and reliability with a specific weight ranging as low as 25lb./b.h.p. It also appeared that wear rates, even on high sulphur, heavy fuels, were not excessive. As with all highly rated engines, some early difficulties, as the authors had stated, had been encountered, for example, excessive top ring groove wear (successfully overcome by cast iron inserts) and low exhaust-valve life when burning some grades of heavy fuel. It was a matter for the user to weigh such drawbacks against the economic gains and other advantages of this type of prime mover.

As at December 1963 there were nine ships fitted with PC1 Pielstick main engines in class with Lloyd's Register, with a total of 27 such engines, and from the point of view of repairs and maintenance on a cost basis their record would be judged to be competitive with other classed main Diesel engines.

He had a few questions to put to the authors. Firstly, with regard to Fig. 2, the drawing did not show very clearly what access there was to the underslung main bearings. It would be useful if in their reply they could describe this in greater detail. It seemed to be a rather important point.

Secondly, would the authors agree that liner wear was more a function of sulphur content than Conradson carbon and ash content? Recent work in Britain appeared to suggest that this was so.

Thirdly, reference was made to noise levels. Would the authors indicate what kind of noise levels a multi-engine installation might produce? In their opinion were soundproofed control rooms essential from a crew comfort aspect?

Turning to the last section, covering dual-fuel engines, to date Lloyd's Register had no experience of dual-fuel engines being used for the propulsion of methane carriers, and were very interested in these developments, especially in the safety aspects. The Society had recently issued some requirements of a general nature covering methane as a source of main engine power. Would the authors elaborate on this final section of their paper and state: firstly, what pressure and temperature limitations for the methane gas supplied to the engine were necessary for stable operation, having regard to the fact that the methane boil-off from the cargo tanks would be at a pressure of about 1½lb./sq. in. and at a temperature of some −157 deg. C. (−250 deg. F.) and would require to be heated and compressed? Secondly, what was the lowest methane gas to fuel oil ratio which could be burned satisfactorily in service? Thirdly, as these engines were of the trunk piston type, it could be expected that the normal crankcase atmosphere would contain an explosive mixture and that the flash point of the lubricating oil would be lowered by methane leakage resulting from piston blow-by. For these conditions what safeguards would the authors provide?

Fig. 15, showing diagrammatically the gas and starting air controls and combined gas and starting air valves, was not very clear. Could a detailed diagram be included showing precisely how the gas and starting air systems were separated?

Presumably, the indicator card representative of methane burning would be appreciably different from that when burning fuel oil alone. It would be of interest if the authors would comment on the differences in firing pressure and rate of pressure rise as these affected the scantlings of the crankshaft and possibly also the harmonic coefficients used for evaluating the torsional vibration stresses.

Mr. B. J. Mears (Associate Member) asked whether the authors would develop one aspect of their paper, briefly referred to on page 447, the provision of a sound-proofed control room for the well-being of the crew. More medical evidence was becoming available indicating the maximum levels of noise that could be tolerated by the ear without loss of hearing. He therefore considered that machinery installations should be designed so that operating staff were not exposed to harmful noise levels.

In the experience of his company, more shipowners were showing an interest in the reduction of noise levels. One owner had requested that noise levels to which staff were continuously exposed should not exceed the values recommended by the Medical Research Council.

These values were, for a frequency band of 300-1,200 c/s, a maximum sound pressure level of 85 decibels; for a frequency band of 1,200-4,800 c/s, a maximum sound pressure level of 80 decibels. In order to meet this requirement it would be necessary to provide a sound-proof control room.

It would be interesting to know whether the authors had any records of noise levels in the machinery spaces of the vessels referred to in the paper.

Were there any records available of the noise levels inside and outside the control room on the *Rio* class ships?

The rate of pressure rise during the combustion portion of the cycle influenced to a large extent the noise produced by an engine. Would the authors please comment on how far they considered efforts should be taken to smooth out the pressure rise? Would a figure for pressure rise of 30lb./sq. in./degree of crank angle be a desirable one to aim for?

Finally, with regard to the arrangements made for the supply of combustion air, Figs. 6, 7 and 11 appeared to indicate that combustion air was drawn from outside the machinery space. Was this correct? If so, did the arrangement have any useful effect in reducing noise in the machinery space?

Mr. V. H. F. Hopkins said that one of the most striking points in the paper, in his view, was the remarkably low maximum cylinder pressure achieved in the PC2, given as 1,280 to 1,350lb./sq. in. in association with b.m.e.p. between 206 and 222lb./sq. in.

He thought that the rise in maximum pressure of only 70-80lb./sq. in. above that of the PC1 was significant only to underline the fact that the maximum pressure in the PC1 was excessively high at over 1,200lb./sq. in. for 130lb./sq. in. b.m.e.p. or thereabouts. This was about 200lb./sq. in. higher than a number of other engines in the same class. Perhaps the comparison given for a 40 per cent lower power per cylinder, i.e. 130 b.m.e.p., at the same speed was not in fact meant.

However, peak pressure around 1,300lb./sq. in. for 200 b.m.e.p. and above was extraordinarily low and worthy of study. It was evident from the authors that considerable development had been carried out with all the variables relating to fuel injection, turbocharging and combustion, and undoubtedly much of the ability to run with a low peak pressure was due to the excellence of this work. Their own optimization study with computer aid, for an engine similar in size and performance to the PC2, showed that with optimum choice of compression ratio and blower pressure ratio, a lower fuel consumption would be obtained with 1,600lb./sq. in. peak pressure than with 1,200lb./sq. in. to the extent of five or six per cent, and slightly better still at 1,800lb./sq. in.—a law of diminishing return. The criteria in the design of this type of engine was set by fuel consumption on the one hand, to be economically competitive in performance, and, on the other hand, to be competitive in capital cost, i.e. cost per h.p. The higher the pressure, for which the engine was designed, the greater the specific cost, but the lower was the running cost, and an optimum could be chosen. An overriding factor, however, was imposed by the growing necessity to provide an economic unit for heavy fuel running, and it might be fair to say for this, the lower the peak pressure the better, consonant with the best combustion, i.e., fuel consumption, possible.

Had the authors' choice of peak pressure vis-à-vis fuel consumption been based upon these considerations? It would be of interest also to know what compression ratio and blower pressure ratio were contained in the PC2.

From the information on page 447, when driving its own three pumps, the fuel consumption in the working speed range for full load was 0·355lb./b.h.p.-hr., a rather high figure. He would hazard a guess that this could be 0·335lb./b.h.p.-hr. at the expense of somewhat higher peak pressure. These figures were, of course, on distillate fuels.

The figures given for ships running with heavy fuels, ranging from 200 to 1,500 sec. Redwood No. 1 were very encouraging. He took it they were all or mostly with b.m.e.p. for full load in the 130lb./sq. in. vicinity; perhaps some were with lower ratings and it would be helpful if the authors could give these. The wear figures seemed to confirm the experience of others with such fuels in engines of similar class. The PC2 test engine run of 3,700 hours on 3,400 sec. fuel having high sulphur and vanadium/sodium combination, was particularly interesting. Dare one ask what the exhaust valve consumption had been?

He agreed with the authors on the need for improved exhaust valve life in engines of the type concerned. The target of 5,000 hours with b.m.e.p. over 200lb./sq. in., if achieved, would be a real breakthrough on this problem; even half that time with such fuel and load would be remarkable.

Another feature worthy of comment was the piston design for b.m.e.p. into the 220lb./sq. in. region. There was a school of thought that for this class of engine a two-piece piston would be better, i.e. with an iron or steel crown and alloy skirt, thus providing for better cooling channels and longer groove life, etc. The single piece cast-in cooling coil design was simpler and "half the price", but he was surprised that a return oil drilling down the connecting rod was necessary. He noticed that the PC2 piston also had one less compression ring than the PC1. Was this change made to reduce friction or to strengthen the piston section beneath the ring grooves?

The references to running on natural gas were very interesting. Was it to be understood that the limit of 185lb./sq. in. b.m.e.p. was imposed by detonation above that, without lowering the compression ratio, or was there another limitation like piston temperature?

Finally, he wished to say how well the paper stated the case for the multi-engined marine installation. With the satisfactory performance now given by reduction gears and clutches, the recovery of additional cargo space should be very attractive to the shipowner. The case for Diesel electric propulsion was noteworthy where a very compact machinery layout on two or more levels at the aft end was possible, as referred to by the authors.

Mr. A. Austin said that his contribution did not really come under the heading of this paper, in as much as it was not related to service performance, but to the test performance of the first engines which had been built in this country.

So far they had built three Pielstick engines of the PC2 type in this country, one 12-cylinder unit and two ten-cylinder engines, all of the vee-form. The 12-cylinder unit was tested at a rating of 4,680 b.h.p. at 428 r.p.m., which gave a b.m.e.p. of 204·5lb./sq. in. The ten-cylinder units were tested at 3,950 h.p., 428 r.p.m. and b.m.e.p. of 207lb./sq. in. All the engines were tested on Class A gas oil. The test arrangement was such that each engine was driving its own lubricating oil, fuel booster and fresh water circulating pumps, and the test figures obtained for the 12PC2V, were a specific consumption of 0·347lb./b.h.p.-hr., with a 393 deg. C. (740 deg. F.) mean exhaust temperature. The 10PC2V units had a consumption of 0·346lb./b.h.p.-hr., with a mean exhaust temperature of 404 deg. C. (760 deg. F.) and in both cases the cylinder pressure was limited to a maximum of 1,250lb./sq. in.

During the tests of these engines, all of which were tested under consultant supervision, there were no attempts to tune the engines specially; in fact, the only adjustment during the test run was the standard procedure of checking the injectors for braking pressure and nozzle clearance, just before going on to full load for the first time. On all three engines the exhaust gases were practically invisible, from quarter load upwards, and the remarkable thing which struck them and the consultants was that on each engine the exhaust shade was better at overload than at full load, which seemed to indicate that they were offering these engines at powers well below their ultimate, which was a good thing at the moment, because they would be able to offer greater reliability in that condition.

He endorsed some of the remarks made by other speakers, in that on stripping the engines after test the piston conditions had been really good. The carbon deposit on the top of the piston had been very small, evenly distributed, and there had been no sign of any excessive heating at any part of the crown. All the rings had been free and the lubricating oil distribution on the crown and on the skirt had been quite normal.

All these three engines so far tested were, however, for stationary application in West Africa, none as yet unfortunately being for marine purposes.

Mr. J. Jones (Member) said that high rated engines of this cylinder diameter and upwards had been reported on previously, but no wear rates had been available as a guide to what the future would be in regard to maintenance. The figures given in the paper would, therefore, be of very great use to users of Diesel engines.

The authors had mentioned the prejudice of shipowners against trunk piston engines using heavy oil fuel as they were afraid of the lubricating oil becoming contaminated with acid products.

He agreed with the authors that four-stroke engines of the type discussed in the paper did not seem to suffer. He had had experience of a similar geared Diesel installation comprising two sets of engines, each set comprising one gearbox, two engines to each propeller. These engines had 17-in. diameter cylinders. After eight years there was no sign of corrosion of the parts in the crankcase. Was this because smaller engines (compared with the large direct propulsion units) used higher grade steels? Would the authors state the steel specification of the crankshaft material?

It was well known that low carbon steel was very subject to corrosion, whereas higher grade steels were not as subject, even if only higher carbon steels.

It had been said that evening that the crankcase dilution came from the products coming from the engine cylinder, and it rather indicated that sulphur content was not the cause of the trouble. He believed it was established that sulphur in the lubricating oil was the main cause of any crankcase corrosion.

A lot of research work had been done on Diesel engines, especially running on heavy oil with $2\frac{1}{2}$ to $3\frac{1}{2}$ per cent sulphur content. The wear rate had been cut in half by increasing the water inlet temperature to 77 deg. C. (170 deg. F.). This meant that the rate of water flow through the engine had to be high enough to permit a small rise in temperature, say 11 deg. C. (20 deg. F.).

He put it to the authors that it was the temperature of the crankcase end of the liner, on the one hand, and perhaps the quality of the steel on the other, which enabled the trunk piston to come out without any corrosion in crankcases.

Would the authors confirm that Fig. 4 (the fuel consumption figures) referred to fuel oil of 19,300 B.t.u./lb. gross calorific value, as heavier fuel was approximately 5 per cent lower value?

He had had considerable experience of exhaust valves on big engines and had tried various facings such as Stellite No. 6. Unfortunately, they tended to crack due to the hardness, whereas with other facings, being softer, there was a tendency to get the foreign matter embedded on the valve face. A compromise was made with Stellite No. 10.

On page 459 in the third paragraph, a figure was given for lubricating oil consumption of 0·4lb./h.p.-hr. Perhaps this should read 0·004lb./h.p.-hr. Even so it would appear to be high. In these days one would expect to get a lubricating oil consumption of at least half of that given in the paper. Many oil engines were now run at something like 10,000 rated h.p.-hr./gallon which was about a quarter of this.

The deposit shown on pistons on the screen was an accumulation of carbon from the lubricating oil, and not from the combustion of the fuel oil.

One of the speakers suggested there could be leakage from the combustion chamber to the crankcase when using methane gas. He had had experience of running gas engines; he had been in the business 50 years, and crankcase explosions never occurred except on engines running on gas which was about 50 per cent hydrogen. The remarkable thing was that many of them always occurred when the governor valves were closed. That was the remarkable thing. It was proved that the explosion occurred with worn cylinder liners permitting gas to escape, on the compression stroke, into the crankcase, the ignition coming from the combustion chamber via the clearance between the piston and liner. A case investigated at a coal mine occurred in this manner, and, as was well known, safety regulations called for wide surface joints on all electrical switchboxes to prevent ignition being supplied, from inside a switchbox, to any gas and air that might be outside the box. It was, therefore, asked how could ignition come from the combustion chamber?

However, it was proved that ignition did come from the combustion chamber, due to the attendants shutting down the engine by closing a gas control valve, which was not for that purpose as it was not gastight. The amount of leakage was not sufficient to keep the engine running, but resulted in a final explosion in the combustion chamber as the engine came to rest, with the crankpin on top dead centre, and a crankcase explosion occurred every time. Changing the method of shutting the engine down, by closing a gas valve, which was gastight when closed, prevented any recurrence and confirmed the theory.

It did, however, show the necessity for crankcase ventilation, drawing air through each crankbay with a lubricating oil separator in the outlet air pipe. Test samples of crankcase air on old engines, when analysed for combustible gases showed that the mixture was outside the combustible ignition range.

The proposal put forward for running on natural gas was a system used by all vertical gas engine manufacturers, only varying as to whether a separate inlet valve was used for gas, or whether the gas and air inlet valves were combined, only mixing as air and gas was drawn into the cylinder. His opinion was that for large gas engines the separate gas inlet valve was preferable because it gave a much safer engine, in that there was no connexion with the air passage, only through the combustion chamber.

The authors had forecast a rating which was approximately 175lb./sq. in. b.m.e.p., and in his opinion, this was on the optimistic side, though it might be possible on the composition of the gas which the authors had shown. It might be possible with these ships having 90 per cent methane, but if they went into the industrial market they would very quickly find that the variety of gases was very considerable—something like Heinz' 57. That was one of the drawbacks of manufacturing a gas engine, that often one had to tune up the engine on site, especially if running on a system like this, where there was quantity governing, so that constant mixture strength continued right through the range. The highest rate so far achieved in the industrial world on gas engines, including dual-fuel engines, was 154lb./sq. in. gauge, b.m.e.p., installed in a generating station of 44,000 h.p., in Canada (with American engines, unfortunately), but it had to be admitted that the amount of natural gas in America and Canada, and the various other sources, meant that they had more experience than their counterparts in Britain. This rating had only been achieved by using a turboblower with the expander turboblower to bring down the air temperature to a very low figure to retard the tendency to detonation.

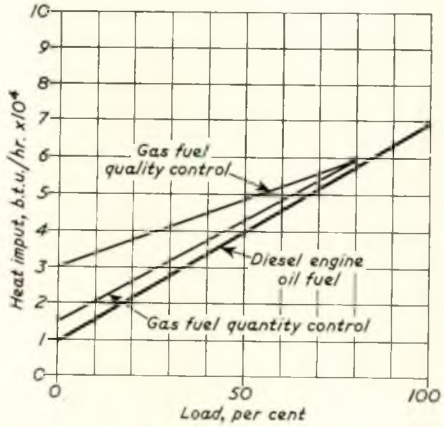


FIG. 16

In the first dual-fuel engines made, it was thought that the high pressure ratios would permit using quality governing, that was, using only gas governor valves.

While the engine ran satisfactorily at 75-100 per cent of rating, the consumption down to zero load was unsatisfactory. Fig. 16 showed a typical Diesel engine consumption in B.t.u./

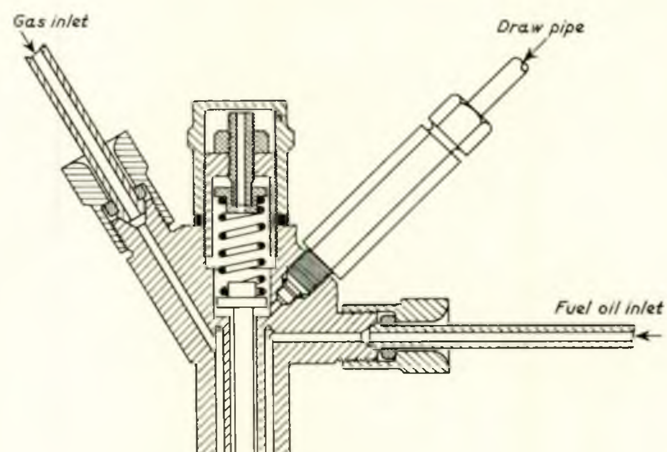


FIG. 17—*Hydraulically operated gas injection valve for internal combustion engines*

hr. from 100 per cent to zero load. A similar line showing the consumption of the gas engine with quantity governing control ran very closely to the oil engine line. When quality governing was used the line rose steeply at zero load so that twice as much gas would go through the cylinders as was required to drive the engine, the surplus gas passing through unburnt.

It occurred to him that it was time that a different view of gas engines was taken, and if gas did not require such a high temperature to ignite it, he was sure they would all be using the Diesel cycle by injecting the gas into the cylinder and obtaining self-ignition. It should be easy to build an engine and inject the gas into the cylinder, in the same way as oil, and burn it from a nozzle; then there would be no need to be worried about mixture strength or detonation. Fig. 17 showed a sectional view of such a valve. It seemed strange to put a compressor on an engine to pump air into it, and then put a governor valve on to prevent the air getting into the cylinder. He suggested that an engine built on these lines would allow one to get the same output as on a Diesel with the same economy at all loads. Ignition would still have to be either by pilot oil or electric spark ignition as used at the present time. Gas pressure would have to be raised to at least 500lb./sq. in. gauge above cylinder combustion pressure.

Mr. C. W. Herbert (Member) said that on page 453 some figures were given covering maintenance. It was noted in particular that the top ring had to be changed every 10,000 hours. What, if any, running-in procedure was involved after fitting the new ring?

In Table I the liner wear rates quoted for three ships fitted with PC1 engines were related to light residual fuels with low ash content. In these engines the wear rates ranged from 14 to 37 per cent of those quoted for the PC2 on heavier fuel with high ash and sulphur. Was the difference between the wear rates due to the differences in fuel quality, the fact that the ships concerned could possibly run at less than full load in service, or to some inherent difference between the PC1 and PC2 types? It would seem logical to think that the stated need to change the top ring every 10,000 hours was based on experience with the earlier PC1 engines, and equally logical to assume that the top ring wear would increase in proportion to the liner wear rate. This would presumably result in the need to renew the top ring much more frequently than the interval of 10,000 hours quoted in the paper. Could one reasonably infer that it might be necessary to examine the top ring every 3,000 to 4,000 hours, when operating under the more severe conditions quoted? On a relatively small installation with two eight-cylinder engines this would mean that one liner would have to be dealt with every 200 to 250 hours of actual engine operating time, and would seem to present a formidable maintenance problem. Would the authors care to comment on this?

In making these remarks he was primarily concerned with relating the performance of these engines to the currently popular slow-speed Diesels, and in no way wished to minimize the very real advance in medium speed engine design which he believed the Pielstick engine to represent. There would undoubtedly be an increasing market for this type of machine, but it was always necessary to bear in mind how it compared with accepted standards in every respect. He had had the privilege of listening to the PC2 running on overload and could endorse the authors' statement regarding noise. The actual engine noise was indeed quite tolerable, though there seemed to be urgent need for attention to the high frequency noises which emanated from the turboblowers. In his own opinion these high frequency noises were more fatiguing than the normal Diesel engine noises to which marine engineers were accustomed.

Mr. J. W. Bond commented on the freedom with which the authors discussed their figures on wear rates. Normally these matters were not dealt with in lectures, and they had been very courageous.

He had noticed in publications, the previous year, that the ten-cylinder engine was advertised for an operating speed of 400 r.p.m. Was there any point in this? He agreed that it had been stated that night that the author's company were running engines at 428 r.p.m. Was there a restriction on speed with this engine or was it just a matter of a misprint in the book in question? He believed that the air starting pressure on these engines was about 300lb./sq. in. Could they be made to operate on the more popular 450lb./sq. in. for both starting and reversing?

With regard to noise levels, he believed that, in France, deafness due to employment was considered a matter for state assistance. He was not quite certain whether this was so, but suggested that if it were so, probably the designers in France might have had this in their minds.

Mr. S. G. Christensen (Associate Member of Council) thought it a pity that more details of the design of the component parts of the engine had not been given. Any engine was only as good as the excellence of the design of the details and the balance that was obtained where compromise had to be made in design features.

Whilst he knew that new things must not be considered in the light of old, he felt that, even with main engines as light as this, it was poor design if an engine had to be lifted off its chocks in order to remove the crankshaft. That was what one must think in looking at the cross-section of the vee-engine shown in Fig. 2. The flange of the crankcase pan should be such that the chocks could remain undisturbed if the entablature, upper half of crankcase and other gear were lifted to remove a crankshaft. It was, of course, fully realized that the cost for chocking and aligning the crankshaft of this engine would not be of the same order as for the large slow speed direct drive engine.

Again referring to crankshaft removal, it must be assumed that after lifting the entablature and upper half crankcase, the main bearing housings must be removed by withdrawing the through bolts in order to remove the crankshaft. Whilst this might be a novel form of construction, he felt that it was one which could lead to eventual difficulty with maintaining crankshaft alignment during the life of the engine.

As the superintendent of a shipping company he often made voyages in the company's ships, and engaged in the various duties as would a normal seagoing engineer. It was when carrying out such duties that one saw faults in the work of the design engineer. He now referred to the round crankcase doors shown in Fig. 6. If the frame spacing of the ship were 27in. then it could be seen that the outside diameter of the door was somewhat less than this; the flange width of the door, when taken into account, made the working opening again less. Had the authors of this paper gone down into the works and done any dismantling work on their engine? If they had, he felt sure it must be in their minds to make these doors larger.

The thickness of the sides of the crankcase suggested that the longitudinal strength of the engine was in the entablature. If this were the case, why not make the door extend the full length of the crankcase and fit smaller inspection doors to each individual crankbay? This door could possibly be made in some transparent plastic material, which should be light and strong, and at the same time give full visibility to the internal parts of the engine when it was stopped. The cost would no doubt be higher, but the savings in overhauling costs would soon cover the extra cost of this arrangement.

On the design of the crankshafts for these engines, it appeared that full consideration was given to the high rise that occurred in the mean effective pressure when operating only one engine of a twin-geared set on a single screw. From the diagram shown in Fig. 4, fuel iso-consumption lines, and taking the cube law for the propeller into account, it would appear that the mean effective pressure rose to 275lb./sq. in. approximately. What was the specific consumption for this m.e.p. at the speed of 360 r.p.m.?

With the specific consumption figures put forward one must assume that the figures referred to the output of power from the engine crankshaft into the gearbox. Would the authors say something of the expected efficiencies for gearboxes, couplings, etc., so that the fuel consumption could be calculated on a shaft horsepower basis? We would then have a fuel rate of the specific consumption of this engine for comparison with the conventional slow speed direct drive engines.

Reference was now made to that section of the paper dealing with results in service. Two different procedures for exhaust valve maintenance were put forward. Which of these was the most commercially attractive when all the relevant factors were taken into account?

With regard to Fig. 13, looking at the gear drive for the camshaft it could be seen that there was a very small wheel incorporated in the train. The teeth on that wheel would do more work than other gear teeth in the system, and he felt that if that gear train drive came from a point along the crankshaft where there was an anti-node, it would not be long before there would be trouble with tooth fatigue in the small wheel.

Looking at some of the engine room arrangements shown he felt that any major repair work on the main engines would become costly due to the removals which must be made to obtain the access space necessary to lift the main engine parts. A lot more imagination could be used in this direction. Fig. 10 showed an engine room arrangement for the banana boat *Tarpon*. This arrangement had the main engine exhaust pipes crossing from the after end of the engine room to the forward end, whereby they crossed the engine room trunk space. This, he felt, was bad design. Why not make alternative arrangements for fitting the turboblowers either at the forward or after end of the engine, even though some obvious difficulties would come about with the characteristics of the exhaust pipe system? If, in this vessel, that had been the case, a much better engine room arrangement could have been drawn up, with the main engine exhaust pipes rising vertically.

The various forms of drive put forward by the authors for Pielstick engines were all very interesting. The Diesel electric system, with the use of alternating current and variable pitch propeller, was a very attractive one. It simplified the electrical requirements necessary for speed control with alternating current, allowed the engine to run at constant speed, although some difficulties might be encountered with engine speeds in heavy weather. This would certainly be the system if, and when, they thought of fully automated engine rooms. The first cost of such a system would obviously be very high, but with the degree of reliability it would give, some compensation was obtained from the high cost.

With such a form of machinery arrangement the difficulties of fast turn-rounds, planned maintenance and continuous survey work would be largely overcome.

With regard to the authors' comments on having the same unit sizes of engine throughout a fleet, that was a good point until the time some technical advancement was reached; then it might be found that it was not possible to advance with new ideas because of detail design changes, which prevented interchange of spare parts. This in turn would prevent taking full advantage of standardization.

Mr. L. R. C. Lilly said that clearly the authors were to be congratulated in that they had an engine which, when arranged in a group of four around a central gearbox or boxes, would give the same total power as a single large marine engine of some 20,000 b.h.p., such as the Doxford nine-cylinder J type engine. The weight was, however not more than half, and the volume space occupied less than half that of the single large engine. The floor area required appeared, however, rather more, in fact nearly double that of the single large engine.

The rate of liner wear on distillate fuel in the PC2 engines was encouragingly low and compared favourably with the large engines. When using residual fuel the wear was still surprisingly low, although his company thought it probable that those fuels with sulphur content above 3 per cent and giving high Conradson and ash values would cause augmented wear.

It had occurred to him that possibly the poor exhaust valve behaviour could be improved by the use of valve rotators, as had been the case with the smaller engines running on residual fuel.

It was noted that S.E.M.T. were carrying out tests to see if the generally accepted filtration system for residual fuel, of two centrifuges working as purifier and clarifier could be simplified. His company was of the view that one stage of centrifuging was quite adequate both on the score of cylinder bore wear and piston ring wear, and also in regard to wear in the injection system. Whether filters alone would do the job of a single centrifuge remained to be proved.

Turning now to the operation of the engines on natural gas, as might be required for methane-carrying vessels, the authors pointed out that when the gas tanks were empty the engine could run on residual fuel. Did this mean that, during operation on gas as a dual-fuel engine, the pilot charge could be residual fuel instead of a distillate fuel?

An interesting point in the design of these engines was the method of returning the piston cooling oil down the rod to be released at the big end level, clear of the moving parts, where it could not be splashed up again to increase the oil consumption, because one of the difficulties, with oil-cooled pistons which ejected oil that could fall onto the moving parts, was the increase in consumption that resulted.

Were the piston rings, shown in the piston drawing, normally given a barrel shape? They appeared to have a barrel shape in the photograph.

Mr. A. Norris (Member) said that from the information given in this interesting paper it would seem that the ships equipped with earlier PC engines had given good results in service, and it was hoped that similar results would be given with the new PC2 design. There was always the difficulty that people were suspicious of a new engine.

It was relevant to comment that the information given in Table I did not show a great deal of operating experience on high viscosity fuel. The ships named in the table had operated for long periods on other fuels, and it must be remembered that there were many other Diesel engines of similar type which would give extremely good service on good fuel.

This type of engine was obviously well suited for providing a relatively large power in a small engine room and, as such, was indicated for ships when maximum advantage could be taken of this and larger cargo hold spaces obtained. This was for cargo liners which often loaded to full capacity rather than ships of the tanker type which often loaded to full displacement.

The notable exception to this seemed to be the 50,000-d.w.t. tanker mentioned on page 447. An attractive feature of the ship, which was mentioned on the following page, was the provision of main engine driven generators to supply power for cargo pumping. It was surprising to find that this ship was not listed in Table I, as it seemed unlikely that the engines were operating on Diesel fuel. He believed that the installed power was nearly 18,000 b.h.p. and, assuming a service power of, say, 16,000 b.h.p., on published differential fuel costs the extra daily cost of burning Diesel instead of high viscosity fuel would be £235. As this would amount to more than £60,000 per annum it would be of interest if the authors could confirm that high viscosity fuel was being used, or alternatively give some information on the special factors which justified operation on Diesel fuel.

Looking at this type of engine from the point of view of tanker installations for the smaller powers, there was a case for consideration. There was a weight saving of the order of 160 tons for about 10,000 h.p., but, to take full advantage of the engine, the cheaper residual fuels must be used.

Mr. D. Royle, B.Sc. (Eng.) (Member) confined his remarks to the use of heavy fuel in S.E.M.T. Pielstick engines for marine propulsion. For industrial engines, special supply arrangements could be made to use fuels conforming to specified characteristics. The same was not true for ships. They had to bunker at ports where their trade took them. It was worth noting that Venezuelan fuels were still used extensively in the bunker trade, they were normally available at many Western Hemisphere ports and were also imported into Europe in sizeable quantities. They contained amounts of vanadium in excess of those mentioned in the paper. The fuels usually used by the m.v. *Montevideo* had been made from such a Venezuelan bunker fuel, to which had been added a distillate marine Diesel fuel in order to reduce the viscosity to the required level. Similar fuels had also been used by the m.v. *Buenos Aires* and *Rio de Janeiro* and, consequently, values for sulphur, ash and vanadium, as shown for the m.v. *Montevideo,* would also apply to these other two vessels.

It would be interesting to know whether any of the ships in Table I were water-ballasting their fuel tanks.

With regard to the test fuel that was in current use at S.E.M.T., the impression was given that a specially refined fuel had been made. It was suggested that while a normal bunker fuel, of Venezuelan origin, would probably have a sulphur content lower than that quoted, the vanadium content would be considerably in excess of the 195 p.p.m. quoted in the paper, being in the order of 250-300 p.p.m.

It was suggested that if a fuel of this quality was used as a test fuel and the PC engine was made to run satisfactorily on it, owners of Pielstick engined vessels would be able to use fuels with viscosities up to at least 1,500 seconds Redwood No. 1 at 100 deg. F. without having to impose any other restrictive quality features, as was the custom of most operators using heavy fuel in slow speed marine Diesel engines.

Correspondence

MR. R. BEATTIE (Member) wrote that the heading "Service Performance of S.E.M.T. Pielstick Engines" limited, to some extent, the scope of discussion as it dealt mainly with operating data collected from various ships fitted with main propulsion units of this type and the various layouts in operation.

There was no doubt that the design of the Pielstick engine incorporated many unusual features, the pistons in particular were completely different from any other design known to him. The method of cooling must be very effective, since no pistons had so far been found burnt or cracked, nor had shown signs of oxidization. He asked the authors what method could be adopted to clean the steel cooling coils in the event of sludge deposit adhering to the walls of the tube, and if any trouble had been encountered, under service conditions, from this source. Was lack of trouble of this nature due to the use of highly inhibited detergent oils combined with a high lubricating oil pressure, which would have a scouring action, also, what form of lubricating oil filtration was used?

One would expect a certain amount of contamination in the lubricating oil from a trunk piston engine and he asked the authors if there was a special technique in machining the piston and how distortion was overcome in an alloy piston of this nature, to obtain effective sealing from the piston rings, apart from the perfect combustion and allowing for the fact that a ring bearer was cast into the piston for the use of heavy fuel.

The figures quoted for liner and piston wear, especially in the engine of the *Galatee,* where a maximum wear rate in the liners of 0·0004in./1,000 hr. had been recorded, were interesting. Was this engine uni-directional or direct reversing? Had the authors any comparisons for liner wear on the two types of transmission? Normally one would look for a slightly higher rate of wear on a direct reversing engine although this had not proved so with certain medium speed two-cycle turbocharged engines where an even lower liner wear rate had been recorded.

The method employed for cylinder lubrication must be highly efficient to obtain the results given in the paper and he wondered if the authors could give details of this. He would also like to know what the piston ring wear rate was and at what stage renewal was recommended.

Reference was made to one lubricating oil pump. Did this indicate that the oil return to the service tank was by gravity and that the lubricating oil pump was purely a pressure pump?

What would the position be in the case of a shallow draught vessel where a gravity scavenge system for the lubricating oil would not be practicable and could the engine be run on the wet pump principle?

Mr. Beattie thanked the authors for the long awaited information they had given. There was no doubt, from the performance data given in the paper, that the Pielstick engines, under normal seagoing conditions clearly demonstrated the weight to power ratio advantage and reliability which could be obtained from a medium speed engine, as opposed to the slow running, heavy duty Diesel. The other noticeable feature was the compactness of the unit, overall height and length for the power output, which in itself was a recommendation for remote control.

He had no doubt that a great deal more would be heard about this engine in future.

MR. R. YATES (Member) commented that this was an extremely interesting and informative paper. Wear rate figures from engines in service, burning intermediate fuels, were of particular interest.

Liner wear rates for *Galatee* were comparable with other makes of supercharged four-stroke engines, which, when correctly lubricated, operated and maintained, would return figures of the order of 4,000 hours for 1/1,000-in. liner wear, burning marine Diesel fuel. This allowed 16,000 hours between piston overhauls, when the top ring groove would require an oversize piston ring. Referring to Table I, on page 455, liner wear rates for *Montevideo* after 13,000 hours, burning intermediate 200-600 sec. fuels, was a remarkably good result. *Fenice,* however, would appear to have rather less wear than *Galatee* had, burning gas oil. The test engine, admittedly burning a heavy fuel, returned about 800 hours for 1/1,000-in. wear. This latter figure would seem to be more realistic, averaging about double the liner wear compared with an engine operating entirely on gas oil, and was very similar to results obtained from the crosshead type engine. The photograph (Fig. 14) showing a piston after 6,000 hours on heavy fuel was typical of a piston lubricated with an improved alkaline oil, i.e. very clean condition below the top ring, all rings free in the grooves and no measurable wear, except in the top ring groove area. Above the top ring and on the piston crown, there appeared the rather heavy ash deposits found on an engine burning intermediate fuels, compared with a complete absence of such deposits in an engine burning Grade A or B fuels. It would be of interest to hear from the authors if a check had been made on the weight of ash collected from a cylinder over a specific number of hours and how this ash weight might vary for different hours in service. Ash formed on the combustion of heavy fuels seemed to be largely responsible for the short life of exhaust valves. The method described for keeping the valve temperature as low as possible, together with improved valve facing materials, would seem to be the best contribution to increased service life until such times as we had a completely different design of cylinder cover to improve the cooling of valves. Was the Pielstick engine fitted with valve rotators? Were the valve seats removable, and what material had been found to give best results for the valve cones?

Did the authors recommend that the engine should be changed over to gas oil for manœuvring and for running at reduced revolutions over long periods? There was no mention of jacket cooling water temperatures; would the authors say what was considered the optimum and what temperature rise across the jackets was permissible?

Referring to Fig. 3, the flow and return passages for the lubricating oil to the gudgeon pin and piston cooling coil, the drillings would appear to change direction at the same cross-section of the connecting rod. Was this unavoidable? The method employed for cooling the top ring area of the piston, together with cast iron inserts in the top ring grooves, had no doubt contributed to the much reduced wear rates quoted, but it would be interesting to hear if there had been any falling off in service of heat transfer, due to oil carbonizing in the coil, which might readily occur, particularly with engine-driven lubricating oil pumps.

Mr. W. H. Brooks, in a written contribution, referred first to the paragraph under "Thermal Loads", commencing on page 445, and commented that it would be interesting to know what was the normal operating pressure of the cooling oil system—it was noted that the discharge was into the crank-case. Were there any means of purifying the oil on return and what would be the interval between checks taken of the oil?

If the oil was sprayed back into the crankcase, what precautions had been necessary to prevent the catastrophic effect of an explosion of the vapour gases, in the event of a "hot bearing", resulting from a possible spark ignition caused by the function of such a dry bearing?

He further wrote that, on page 453, under "Operating Results—Maintenance Schedule (a)", it was stated that it was necessary to replace the injectors every 1,250 hours. Could these injectors be reconditioned on the vessel or would it be necessary to return them to a shore base? How many spare units would be necessary?

Reference was made, in the sixth paragraph on page 455, to preheating of the fuel, by means of hot water and pressurizing by compressed air. What would be the resultant maximum temperature from such an exhaust heating system?

Could the authors give any reason for the difference in performance of the two sets of valves in m.v. *Moledet* and m.v. *Montevideo,* as indicated on page 458.

In conclusion, Mr. Brooks was of the opinion that the paper made a valuable contribution to engine design.

Mr. R. M. Dunshea (Member) commented that it was always interesting to have particulars of propulsion machinery which was not of the massive proportions normally installed for such purposes.

In reading the paper, however, various points emerged which he felt should be examined.

The piston cooling arrangement was doubtless efficient when clean. There was always, however, a tendency for oil to "bake" in pistons so cooled. Should this occur, removal of carbon appeared difficult. There did not appear to be any provisions made for a check on the temperature and flow of piston cooling oil returns. It was considered that such an arrangement would be most desirable.

Exhaust valves had apparently given a certain amount of trouble when heavy fuel was burned. It was felt that cooling water could with advantage be circulated closer to the valve seats. Had any consideration been given to this possible solution?

The authors' statement of experience gained, burning heavy fuel, was interesting, but, from the particulars given, the viscosity of the fuel, had with one exception, been low and the sulphur content not particularly high. Generally, past experience, in engines where the piston was in communication with the crankcase, had not been extremely satisfactory. H_2O was a resultant product of combustion and it was thought that, as rings and liners wore, crankshaft corrosion could result. Only time would tell if these fears were groundless.

The proposal to use the main engine to drive alternators was interesting. Could particulars be given of the governors used and the method of speed adjustment used for synchronizing purposes?

The consumption of lubricating oil of PC engines was given as under 0·4lb./h.p.-hr. Should the consumption be as high as this figure, it would be most excessive, possibly this figure was given in error. Could the authors' comments be given in this connexion?

Mr. R. C. Cree (Member), in his contribution, wrote that a paper of this description gave considerable interest to those who through pressure of work did not keep up to date, as much as they should, with the recent developments of marine Diesel engines.

It was not completely clear what the authors meant by saying that "the speed range that can be covered in full fuel notch position extends from full speed down to about 60 per cent of full speed", as, in the case where the propellers were designed to absorb the full b.h.p. at maximum engine r.p.m. in trial conditions (as was often the case), if full fuel notch was maintained in a twin-engine installation with only one engine running, the engine would be overloaded. This could be seen from the graph of a twin-engine installation recently tank tested (Fig. 18) and it would be noted that, assuming constant

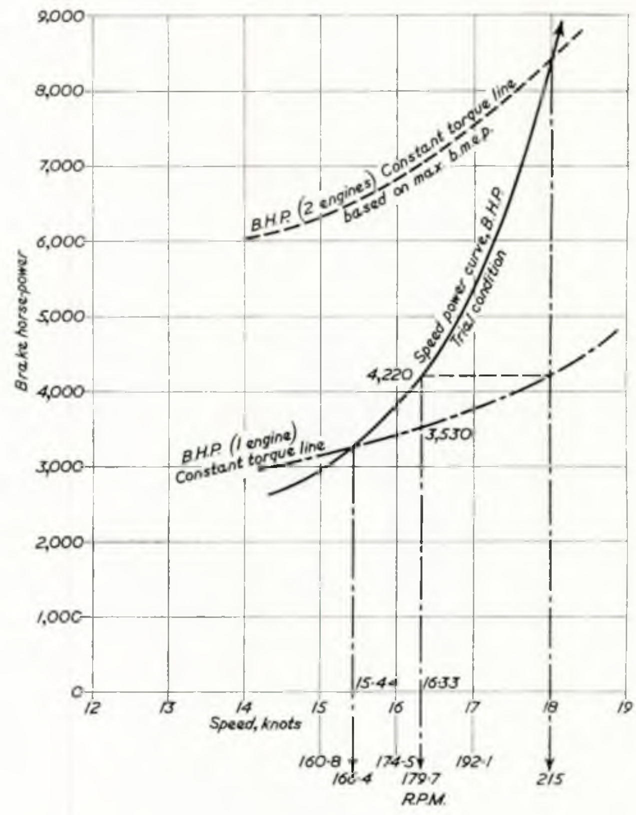


Fig. 18

maximum b.m.e.p., the engine speed that could be obtained by one engine without overload, was 166·4 r.p.m., giving a ship speed of 15·44 knots, whereas with full notch approximately 179·7 r.p.m. could be achieved giving a speed of 16·33 knots. However, to do this the engine would require to be on overload to the extent of about 20 per cent. From this it would be seen that it would be necessary in this condition to pull the engine fuel lever back to give an engine speed of 166·4 r.p.m. An interesting point would be seen from the diagram that, with the installation concerned, one engine would be capable of propelling the ship at 86 per cent of the ship speed obtained with two engines. If it was intended, on a twin-engine installation coupled to one screw shaft, to run regularly on the one engine, then the fitting of a controllable pitch propeller would have a definite advantage.

It was interesting to note from the paper that the PC engine lent itself to automation. Did this mean that the engine builders had designed their control gear so that it was easily adaptable to local or bridge control? Bridge control of main machinery was undoubtedly here to stay, but it had been the writer's experience that this important aspect of engine control gear had been for the most part an adaptation of the usual local controls to suit a remote controlled installation, the mechanics of this often being left to the sub-contractors who provided the linkage (be it mechanical, pneumatic, electric, or hydraulic) from engine to bridge. Surely the time had come when the engine builder could offer a guaranteed system of bridge control designed to suit his engine, rather than have the usual practice of considering bridge control as an extra and employing a sub-contractor to fit this to the standard engine, often after the engine had been installed in the ship. The authors had said that a ship was now under construction where alternators would be coupled to the main propulsion engines and that constant engine speed would be maintained so that these alternators could be connected to the distribution system. This system would appear at first sight to be practical and no doubt the authors were fully aware of the sensitivity of the engine governors required with such an installation, to ensure balanced loading of the alternators when two or more were synchronized. It would be interesting to hear in due course whether this arrangement proved satisfactory.

It would appear from the maintenance schedule that the authors had experienced an extension of the usual time expected between the overhaul of engine components. This, of course, was good news for all marine engineers who were so often pressed, by commercial demands, to extend the overhauling periods beyond that time which they had found to be necessary. A definite schedule of marine maintenance of course was now essential on almost every commercial service and it was imperative that breakdowns of these schedule programmes were kept to a minimum.

Generally speaking, the wear rate figures given were good, and it was gratifying to know that high performance engines such as the Pielstick PC2 engine were now being produced for installations of up to 20,000 b.h.p. and that wear rates were comparable with the low speed engine which was an automatic choice, a few years ago, for any installation over 6,000 b.h.p. The saving in space in fitting high performance engines was considerable and providing that there were no disadvantages in maintenance and initial cost, then the extra cargo-carrying capacity available, by fitting these units, would undoubtedly weigh in their favour.

Mr. G. T. Adams (Member) made several observations in a written contribution.

Would the authors kindly say if there had been cases of fractured cylinder heads? By reason of the number of valve pockets, the design must be complicated and uniform cooling difficult.

When the maximum wear of some liners was double that of other liners in the same engine, at the same r.p.m., on the same service, using the same lubricating oil, with the same total number of running hours, one was tempted to ask why.

The minimum fuel consumption was obtained at 75 per cent of the present full power. It would appear to be necessary to attain improved fuel consumption, at the higher ratings, if bigger outputs per cylinder were contemplated.

Makers of flexible couplings were prone to emphasize the degree of misalignment which their couplings would accept, but experience indicated that correct alignment of engines and gearboxes was very necessary, even though a flexible coupling was interposed.

Would engine builders please shop test their engines with an exhaust system of back pressure equivalent to the shipboard installation? In subsequent service the ship's engineer had to rely upon his exhaust temperature readings for performance guidance.

Mr. S. Archer, M.Sc. (Member), in a further contribution, wrote that a final point was the question of the small-bore, cast-in cooling coils in the piston heads. It was difficult to believe that such relatively constricted passages would not in time suffer from restriction due to carbonization of overheated cooling oil, with possibilities of piston seizure and similar ills. Surely the "cocktail shaker" design was much to be preferred from the aspect of service reliability.

Authors' Reply

The authors had found the large response and high quality of the discussion and questions to be very gratifying and stimulating.

As many of the comments and questions overlapped it was proposed to deal with them under the following five headings:

1) Engine Design and Installation.
2) Engine Cycles, Thermal Efficiency and Performance.
3) Running on Heavy Fuel.
4) Noise.
5) Dual Fuel.

Design and Installation

Mr. Tuson in opening the discussion welcomed the advent of a medium speed engine developing 8,000 horsepower. The Pielstick engine had been designed and was being developed with an eye to the market demand and it was felt that not only was it welcome in the power generation field, but that it suited very well the most popular range of marine applications when used as a geared installation. Mr. Carter had drawn attention to this and so had other speakers including Messrs. Hopkins, Lilly, Royle and Cree. The authors were particularly encouraged by Mr. Archer's comment that the Pielstick main engines in class with Lloyd's Register would be judged to be competitive with other classed main Diesel engines.

The method of supporting the crankshaft was unusual and had brought comment from Mr. Christensen. Mr. Archer had also asked that a fuller explanation should be given. Fig. 19 showed the main bearing assembly, the bearing housing was securely attached to the frame, the long studs taking the vertical component of the gas loads and the horizontal components being taken care of by the frame itself which was secured to the bearing housing by the short horizontal setscrews. The thin wall shells were held in place by a cap axially. Transverse location was provided by a register in the frame. This resulted in a design having all the best features of an underslung crankshaft; firing loads being carried directly through the frame to the bearings without passing through a joint into the underbase, whilst providing accessibility to the bearings equally as

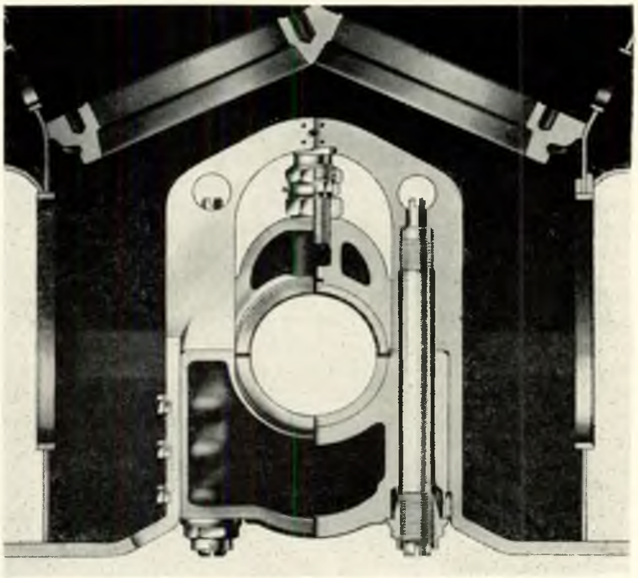


Fig. 19

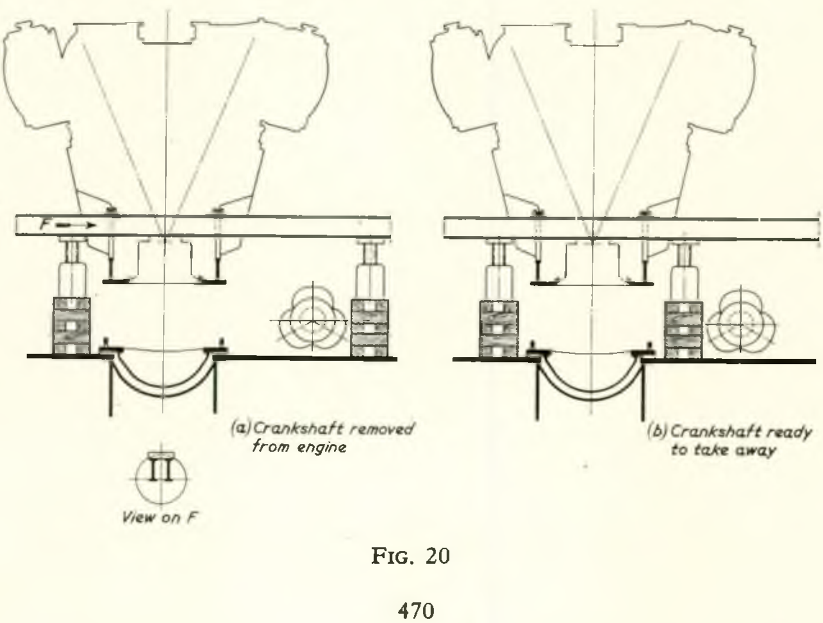


Fig. 20

good as with a conventional construction. The jacking screw was easily removed, the cap lifted and the bearing shells removed, the lower half being turned out in exactly the same way as with a crankshaft supported in a base.

Of course if the crankshaft had to be removed the frame had to be jacked up to do it, but then this applied to conventionally designed engines also. Removal of a crankshaft was a major undertaking whatever the design of engine. It could certainly be claimed, however, that it was a smaller job on a medium speed engine forming one of a number geared together than on a large direct-coupled engine. Fig. 20 showed the method of dismantling. As the main bearing shells were thin wall precision made bearings, inserted in an accurately bored frame, there was no fitting required when replacing them, the question of crankshaft alignment did not arise in the same way as with white metal bearings which had to be scraped. This point was raised by Mr. Christensen who also stated a preference for leaving the chocks in position when this sort of work was done. As could be seen from Fig. 20 the under-base was left in position with the chocks and the holding down bolts.

Mr. Christensen also suggested that the crankcase doors could be enlarged. Fig. 21 showed the welded frame which was all in one piece and formed a box girder which provided the main flexural strength of the engine. The crankcase and portions surrounding the cylinders was one piece and if a long crankcase door were substituted for the individual doors it would detract from the strength considerably. However, the circular doors provided reasonable access and it was possible even to withdraw a balance weight from the crankshaft without disturbing the bottom ends. Main bearings or bottom end bearings could of course be taken out quite easily.

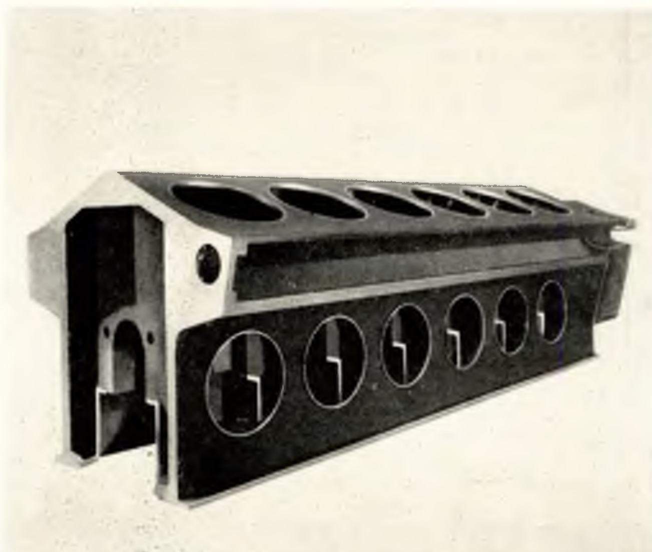


FIG. 21

When designing a gear drive one naturally based the design on the hardest worked gear, tooth size was made adequate from considerations both of fatigue strength and of wear, and this was the case with the small gear to which Mr. Christensen referred.

Mr. Christensen had suggested transparent doors and this was quite an interesting thought. The authors were familiar with transparent doors fitted with windscreen wipers, or clear view screens with lights inside the crankcase as experimental tools, but they had yet to find a user willing to pay for such refinements in service.

The engine room arrangement in Fig. 10 gave adequate access to the engine. The exhaust pipes crossing from the after end of the engine room to the forward end had been arranged by the installing engineers and not by the engine designer. Engines with 14 cylinders or above had four turbochargers, two at each end, and engines with 12 cylinders or below had two turbochargers at one end. Either end could be used with equal facility as the engine cylinder arrangement was symmetrical.

Economic pressures demanded development without interfering seriously with replacement of parts on existing engines. This was a problem with which all engine designers had to cope.

The system of piston cooling employed on these engines had brought questions, especially concerning fouling and cleaning, from Messrs. Hopkins, Lilly, Beattie, Yates, Dunshea, Brooks and from Mr. Archer in his written contribution. It was not claimed that this system of piston cooling was more efficaceous than any other, simply that it was adequate for the Pielstick engine and resulted in a top ring temperature of about 180 deg. C. (356 deg. F.) which was cool by recognized standards. If further cooling proved necessary as development progressed other methods were available. Cocktail shaker designs had been used experimentally and had given even lower metal temperatures. Considerable experience of this type of piston had been accumulated on S.E.M.T. Pielstick engines size PA of 185 mm. bore. A great number of these engines was in use on locomotives where the load cycle was such that they were on full load for a period then followed by idling, and then full load again with frequent shut-downs. As the lubricating oil pumps were engine driven the lubricating oil flow ceased when the engine was shut down and the conditions were such that if there was any tendency of the cooling coils to block up by baking of the oil then it would most certainly occur. From time to time pistons had been taken from these engines and cut up to observe the state of the cooling coils, these in every case had been completely clean. Measurements of the metal temperatures of these pistons showed the light alloy in contact with the cooling coil to reach a temperature of 140 deg. C. (284 deg. F.) when the engine was on full load. The coil itself must be still cooler and this was certainly not hot enough to cause decomposition of the oil.

The oil drilling was returned to the foot of the rod, this was quite convenient to do and saved a lot of splash at the lower end of the liner which would require heavier scraper rings. The oil emerged as a jet rather than as a spray as suggested by Mr. Brooks and the risk of crankcase explosion from this source, was probably less than from the oil thrown from the bearings in any engine. It was difficult to avoid the change of direction in both passages being at the same plane without adding to the mass of the rod undesirably. However, no troub'e had occurred as a result of this feature. A means of checking on the temperature and flow of piston cooling oil would be a refinement, arrangements for doing this could be complicated in a vee-engine and therefore costly.

Mr. Lilly and Mr. Hopkins had commented on the piston rings. The top ring was chromium plated and the others were fitted with a copper insert round the periphery. It was this copper insert which in the photograph gave a misleading impression that the piston rings were barrel shaped. The pistons were barrel shaped when cold, the dimension being chosen very carefully to result in parallel clearance with good sealing when at normal operating temperature. As Mr. Beattie rightly supposed the ring carrier was an added complication and successful sealing was the result of careful development work.

Mr. Beattie also raised the question of liner wear. The engines in the *Galatee* were direct reversing, but manœuvring was usually carried out by putting one engine ahead and the other astern and engaging with the propeller as desired. The direct reversing engines, of which Mr. Beattie spoke, were well known to one of the authors as they also were manufactured by his company, and in their experience direct reversing did not promote wear in a carefully designed two-stroke cycle engine.

Mr. Adams asked if there had been any difficulty with cracked cylinder heads. There had been one or two early cases in service, which were traced directly to casting difficulties being experienced at that time.

It was not essential to have a gravity drain for the lubricating oil system. Two pumps could be used, one scavenging the engine and one feeding it, or alternatively the engine could be run with a wet sump. Two engines in service in a shallow draught vessel did in fact use a wet sump system, they had an extra deep under-base to hold the required quantity of oil.

Mr. Bond called attention to the limit on speed shown in the published literature for the 10 cylinders. This was a question of the mechanical balance of the five throw, four-stroke cycle crankshaft, increasing speed leading to greater out of balance with which it was difficult to deal without incurring penalties of complication and cost. The air-starting pressure chosen had been adequate so far for the Pielstick engine. There was no reason in principle why it should not be raised to 450lb./sq. in. as this became more popular, but some detailed strengthening of components would probably be necessary.

In reply to Mr. Brook's question about injectors, these were conventional in design, although water cooled and could be serviced by engineers familiar with the conventional design.

Both Mr. Cree and Mr. Christensen had mentioned automation and Mr. Cree would be interested to know that the Pielstick engine did not have a local control station. It was designed from the first with the thought that controls would be grouped together at a console. Mr. Dunshea also had a question to ask about the particulars of the governors which had to be both sensitive and capable of rapid response.

In common with most engine builders the authors regarded the use of flexible couplings as a means of obtaining a "soft" equivalent shaft length, a very useful tool in dealing with torsional vibration problems and they agreed with Mr. Adams that correct alignment of engines and gearboxes was very necessary, even though a flexible coupling was interposed.

Mr. Norris, Mr. Christensen and Mr. Dunshea had mentioned electric transmission and Mr. Christensen thought it offered the best solution. Although the authors considered this form of transmission very useful, especially when applied to certain types of vessel they were of the opinion that the cost of initial installation and cost of running due to reduced efficiency, would prevent its universal adoption.

Cycles, Thermal Efficiency and Performance

Mr. Hopkins had made some very pertinent comments on the question of maximum firing pressure and b.m.e.p. He was right in saying that care had to be taken with every aspect of the engine and turbocharger match to ensure a high b.m.e.p. with a low maximum firing pressure. One of the important factors in the change from the PC1 to the PC2 had been in attention to the turbocharger cycle and exhaust manifolding efficiency. This had ensured a plentiful supply of cool air to the cylinders permitting high b.m.e.p. without raising unduly the compression or the firing pressures and providing conditions favourable to good combustion.

Information on compression ratio, compression pressure and boost level was given by Messrs. Kühn and Gallois in a paper to C.I.M.A.C. 1962*.

Careful matching resulted in good fuel consumption. Fig. 2 in the paper was taken from tests made on a two-cylinder experimental engine which (owing to its special balancing arrangements) did not have a high mechanical efficiency. Production engines were usually better than this and indeed Mr. Austin's contribution showed that those recently built in Britain were giving much better performance figures. The low fuel consumption extended to maximum load which should help to allay some of Mr. Adams' fears.

Both Mr. Jones and Mr. Dunshea had called attention to the lubricating oil consumption. The correct figure for the engine was 0.004lb./b.h.p.-hr. This was not so good as some engines, but it was certainly not a bad figure. Current development work showed promise of lower figures. Both Mr. Cree and Mr. Christensen spoke of increase in b.m.e.p. when running on one engine of a pair. The authors thought they must be under some misapprehension. The fuel injection system of any Diesel engine included a metering device, in the case of the Pielstick engine this was part of the fuel pump (as it was on nearly all medium speed engines which used a jerk type pump). When the fuel control was in the full notch position the metering device metered the full quanity of fuel every stroke and this corresponded (as a first order approximation) to full b.m.e.p. and therefore full torque of the engine. For the engine to develop full b.h.p. as speed was reduced, as shown by the horizontal dotted line at 4,220 on Mr. Cree's graph (Fig. 18), the meter would have to increase the fuel pumped per stroke as the number of strokes per minute decreased, clearly this was not possible. The engine therefore followed the constant torque line and was not overloaded. By using a variable pitch propeller the engine r.p.m. could be kept up on one engine and the ship's speed could be about 80 per cent of that obtained with two engines, not 86 per cent as Mr. Cree had suggested.

Mr. Chistensen mentioned gearbox efficiency and of course this very properly had to be taken into account when calculating the shaft horsepower and the fuel consumption in terms of shaft horsepower, but the geared Diesel did not necessarily suffer on this account as the use of a gearbox could frequently enable a more advantageous propeller speed to be used which by increased propeller efficiency could more than offset any gearbox losses.

Mr. Adams pleaded for engine builders to test their engines with exhaust systems such as would be used in the ship. This was not difficult to arrange if the proposed system for the ship was longer than the system required to conduct the exhaust out of the test shop (which was not always the case) and if the shipbuilder could tell the engine builder what back pressure he was going to impose on the engine.

Heavy Fuel

Nine speakers had referred to the question of liner wear on the PC2 engine when using heavy fuel. As could be seen in Table I, the experimental PC2 engine had higher liner wear than any of the PC1 engines in service, but the rate of wear was still very encouraging. The running on heavy fuel of this engine had been carried out at 465 h.p./cylinder, which was 25 per cent more than the recommended service rating for PC2 engines on heavy fuel and of course was considerably higher than the load carried by any PC1 engines. It had also been running on the worst possible fuels. Mr. Archer had asked for the authors' opinion concerning the cause of cylinder liner wear and they confirmed that they thought it was due to the sulphur rather than the Conradson carbon and ash content of the fuel, although there was no doubt that the action was a conjoint one of abrasion and corrosion and a fuel that formed an abrasive ash would wear liners a little faster given the same sulphur content. They certainly agreed with the suggestion by Mr. Jones and others that to use jacket water temperatures which kept the inner surface of the liner above the dew-point was the right way to combat corrosive wear.

In reply to Mr. Tuson they would say that 115 sec. Redwood No. 1 was the highest viscosity permissible at the injectors. They were aware that large two-stroke engines usually required a lower viscosity, and although there might be some benefit in a lower viscosity on the Pielstick engine it was certainly not essential. Several contributors had commented on piston condition and with reference to the photograph in Fig. 14 Mr. Jones had suggested that the deposit on the top of the piston was lubricating oil and Mr. Yates suggested it was ash. It was in fact a thick deposit of grease which had been spread around the top of the liner bore before withdrawing the piston in order to ease it over the inevitable slight deposit of carbon at the top of the liner. It was not a result of running on heavy fuel and could with complete honesty have been removed before the photograph was taken. It was left for fear of marking any deposits which were legitimate results of running on heavy fuel.

Eight speakers had commented on exhaust valve life and steady progress was being made in extending this life. In

* Kühn, K. and Gallois, J. 1962. "Moteurs Diesel a Quatre Temps et Haute Suralimentation". C.I.M.A.C. Conference, Copenhagen.

reply to Mr. Hopkins, 2,500 hours was already within reach and with the use of valve rotators, which had been suggested by a number of speakers, was certain of accomplishment. Such valve rotators were fitted to Pielstick engines being produced now. One or two other methods of keeping the valve heads cool were under development and included cages in which the seats were water cooled. This, in common with other developments would result in more costly components so that the first cost had to be balanced against the economy in running costs. This development work was continuing at a rapid rate and an exhaust valve life of 5,000 hours on heavy fuel between overhauls was not very far away.

The difference in performance of the valves in m.v. *Moledet* and m.v. *Montevideo* to which Mr. Brooks referred was apparent rather than real and resulted from the different techniques practised by the respective chief engineers.

Mr. Tuson suggested that procedures for keeping a check on the exhaust valves on the *Moledet* could have been eased by using automatic temperature monitoring systems. A more recent vessel, the *Fenice,* did in fact use a system of data logging. There was no need for alarms as the change in exhaust gas temperature occurred very slowly a change of about 40 deg. C. (104 deg. F.) requiring approximately 20 hours. This was easy to observe by regular watch on either the exhaust thermometers or the logged temperature.

Mr. Jones and Mr. Dunshea had commented on crankcase contamination. Mr. Jones put forward some interesting thoughts in suggesting that the high grade materials played a part. This might well be so, but in the authors' opinion the chief reasons were good combustion, good piston sealing, high overall temperature so avoiding condensation, water jackets completely separated from the crankcase so that there was no possibility of water entering and modern lubricating oil with adequate checks to see that it was maintained in an alkaline condition. They were indebted to Mr. Carter's contribution for underlining these conditions. Tests on the experimental engine had included a cyclic loading with substantial periods of time at light load in order to promote conditions under which the crankcase oil could well have become contaminated. Checks on the T.B.N. of the oil gave the following readings:

29 at the start of the test,
27 after 50 hours
27·6 after 200 hours
25·8 after 250 hours.

There was quite clearly no danger of corrosive conditions here.

Mr. Herbert had expressed concern about ring changes. The figures given in the paper covered a schedule which the authors regarded as adequate to maintain engines in good condition throughout their life. A reduction in these periods during the life of the engine was not anticipated.

With regard to running-in procedure, if only one ring was changed in a complete engine set the engine was usually put straight on full load. If all the top rings were changed then most users ran-in for a period of about ten hours increasing engine r.p.m. from about three-quarters to full over this period, i.e. increasing b.m.e.p. from half to full over the same period.

In reply to Mr. Beattie the method employed for cylinder lubrication was the usual one for four-stroke cycle engines of using oil control rings below the compression rings, one just above the piston pin and one at the bottom of the skirt. These were carefully designed to permit oil to reach the compression rings in the right quantity. Mr. Norris and Mr. Royle had referred to the nature of fuel and the authors agreed with their remarks. Pielstick engines were running on a wide variety of fuels bunkered at such places as Las Palmas, Borneo, Haifa, Marseilles and Agusta. With some fuels water washing and centrifuging was most advisable, with others filtration alone was sufficient.

In reply to Mr. Yates, manœuvring could be carried out on the heavy fuel as long as its viscosity was sufficiently low when it reached the injectors. As the injector cooling circuit was separate from the engine jacket circuit it was possible to circulate hot water through it for starting and change over to cold water as the engine warmed up. Such a system was in use on the experimental two-cylinder engine which regularly started up on heavy fuel. Mr. Yates, Mr. Brooks and Mr. Jones had all called attention to the importance of jacket temperatures and again Mr. Carter helped them by giving their views. When the Pielstick engine was first designed the jacket temperature was regarded as high by most engine builders, but with the spread of knowledge the importance of keeping the temperature of the inner wall of the liner high enough to avoid corrosive conditions was now recognized and even higher jacket water temperatures were being advocated. In principle they were in agreement with this suggestion but exercised caution in view of the many details (e.g. clearances) which might have to be modified if a higher water jacket temperature was adopted.

Noise

Mr. Archer, Mr. Mears and Mr. Herbert had all commented on noise levels. As Mr. Herbert had stated from his experience the turbocharger noise which was at about 3,000

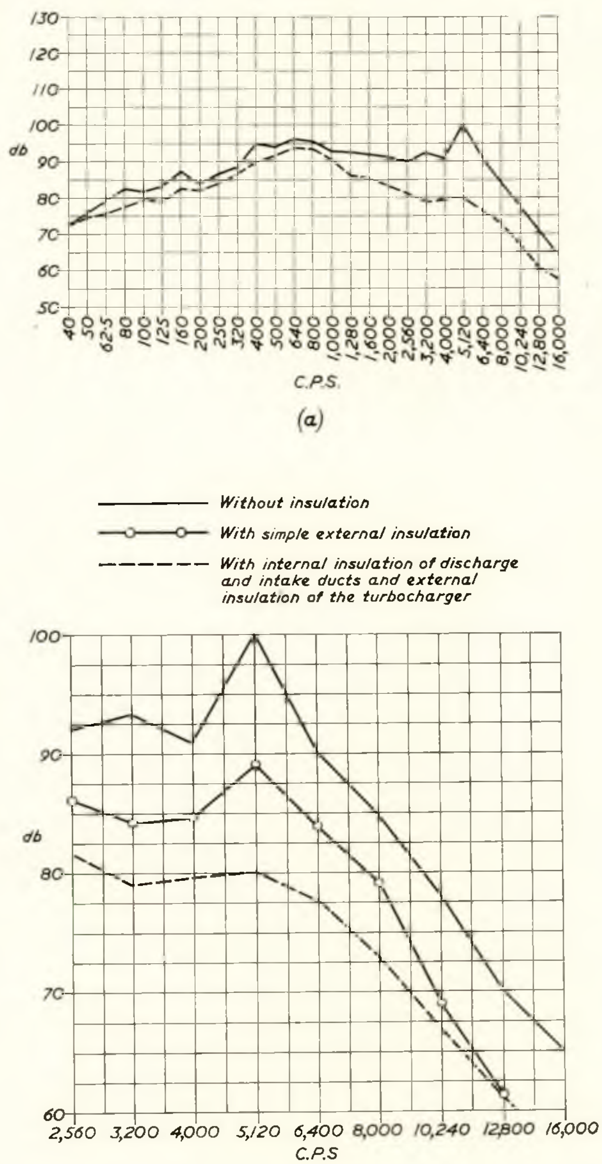


Fig. 22

c/s frequency was much more disturbing although the noise level was not higher than the engine noise at about 500 c/s. The Chairman had given some words of advice concerning initiative to be taken in dealing with the noise problem, the authors were glad to tell him that better air intake with damping and insulating of ducts between the turbochargers and inter-coolers had already resulted in a much more tolerable noise level. Tests had recently been made on a 12-cylinder PC2V engine and the results were shown in Fig. 22. It would be observed that there was a considerable reduction in the turbo-charger frequencies.

Fig. 22(a) showed a third octave band analysis of the spectrum measured at a representative point near to the turbo-charger end of a 12 cylinder engine. The peak at 5,120 c/s emanated from the turbocharger and Fig. 22(b) showed on a larger scale this part of the spectrum.

Many tests with different kinds of insulation were being carried out and Fig. 22(b) showed the results of two of these. The internal insulation of the discharge receiver and suction pipes, together with external insulation of the turbocharger, brought about a decrease of 20 dB in that frequency range. A decrease in noise amounting to half this amount could be obtained by simple external insulation.

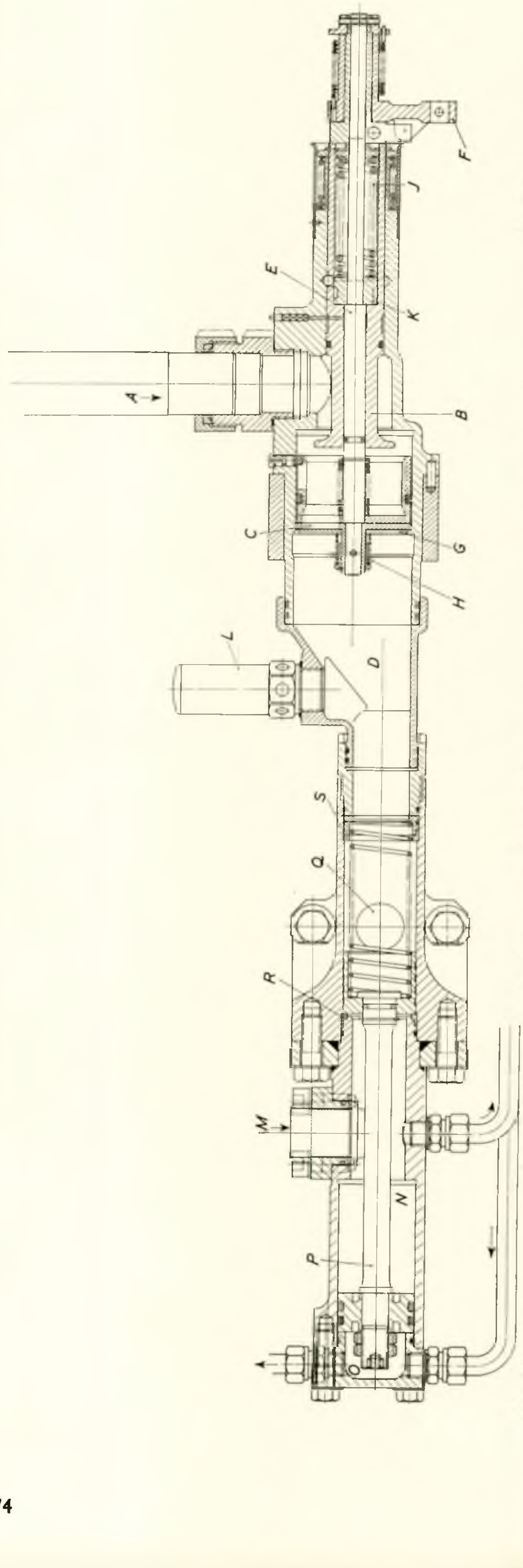
When PC engines were first introduced there were very few marine propulsion engines fitted with turbochargers and on this account the PC engine might have appeared noisy. As the development of larger engines had proceeded they also had come to use turbochargers and had thereby inevitably increased the noise they made. The authors had no doubt that the noise level from such engines was as high as that from the PC engines and that the development work in silencing that was now being done would be applied in turn to them also.

With a geared installation it was convenient to group controls of all engines at one point and the provision of a sound proof control room was no great difficulty. Although noise level in the engine room itself was not damagingly high a sound proof control room to which they could retire provided a welcome degree of comfort for the crew. As the most intolerable noises were those of high frequency a surprisingly large measure of comfort could be obtained by having the control console in a screened area, not necessarily a complete room. Some owners might find this more desirable.

Mr. Mears and Mr. Bond had raised questions concerning permissible levels of noise both in England and in France. Most enlightened countries were aware of the desirability of limiting noise levels and were introducing legislation progressively to see that permanent damage was not caused to men who had to work with machinery. The PC engine was not in the class of very noisy machines and the development work to make it quieter was not the result of it having infringed any regulations. A rate of pressure rise of 30lb./sq. in./degree as given by Mr. Mears the authors regarded as low and did not think it was necessary to reduce rates of pressure rise to this region. Again in reply to a question by Mr. Mears the ducting of the combustion air supply could often have a beneficial effect on reducing noise levels in the engine room, but care had to be taken that the ducting was not resonant at any part of its length as otherwise it could be worse than useless and could be most annoying if it passed through passenger quarters.

Dual Fuel

Mr. Archer had asked about limits of pressure and temperature of the methane supply, these were 2·5 kg./sq. cm. and 25 deg. C. (77 deg. F.), it was of course necessary to heat the gas. Tests had actually been conducted with gas temperatures ranging between 18 and 27 deg. C. (64 and 80 deg. F.). The normal minimum quantity of pilot oil was 6 per cent and any ratio of oil to gas could be used between this figure and 100 per cent oil. Tests made on the crankcase atmosphere revealed that when no scavenging of the crankcase was in operation there was a four per cent content of methane. When scavenged by 100 litres per minute of air (for a six-cylinder engine) this fell to 2 per cent methane. Scavenging could be conveniently arranged by bleeding air from the turboblower delivery at the



A Gas inlet
B Gas shut down valve
C Gas regulation ports
D Gas supply passage
E Regulating spindle
F Regulating lever
G Plate valve
H Spring for plate valve
J Spring for gas valve
K Trip mechanism
L Safety relief valve
M Compressed starting air supply
N Compressed air chamber
O Compressed air operating chamber
P Piston assembly
Q Outlet port
R Sealing ring compressed air
S Sealing ring gas

FIG. 23—*Gas and compressed air control valve*

rate of one per cent and a methane detector could be used to keep a continuous check on the crankcase contents. Mr. Jones's experience was very interesting in this connexion. Fig. 23 showed the controls of air and gas in more detail than in the original figure.

The gas supply was introduced at A and passed via the gas shut-off valve B through the regulation ports C into the chamber D this would cause the plate G to seat on the regulator opening of the ports C by rotation of the spindle E via the lever F from the governor. In order to enter chamber D it had to raise the plate G against the light spring H. If as a result of any accident there should be an excessive pressure in the chamber D this would cause the plate G to seat on the regulator port assembly and to move the whole assembly against the pressure of the spring J, thus shutting off the gas supply by means of the valve B. The trip mechanism K ensured that the gas supply could only be restored by resetting the mechanism by hand. There was also a safety relief valve at L.

The starting air supply entered at M and pressurized chamber N, it passed from this chamber to the timing valve operated by the starter cam for each cylinder and returned to the cylinder passing through chamber O. When the timing

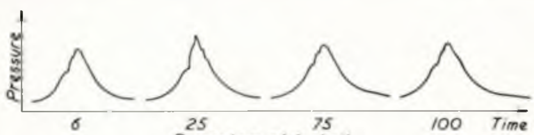


FIG. 24—*Pressure time indicator diagrams for different percentages of fuel oil. PC engine at 168lb./sq. in. b.m.e.p.*

valve was operated the pressure in chamber O caused the piston assembly P to move to the left closing off the gas port Q by which the gas left the chamber D for the cylinders. Sealing between the gas and air chambers was ensured by the piston resting against sealing ring R when starting air was not being supplied to a cylinder and against sealing ring S when starting air was being supplied.

Mr. Archer had raised an interesting point in connexion with indicator diagrams which were shown in Fig. 24. It was interesting to observe in Fig. 25 the change in maximum pressure with gas to fuel ratio and also the rate of pressure rise, which reached a maximum of 12 kg./sq. cm./degree when the oil was 23 per cent of the total heat intake. When running with pilot oil only or with complete oil or almost complete oil, which were the most likely zones of operation, the rate of pressure rise was less than half this amount.

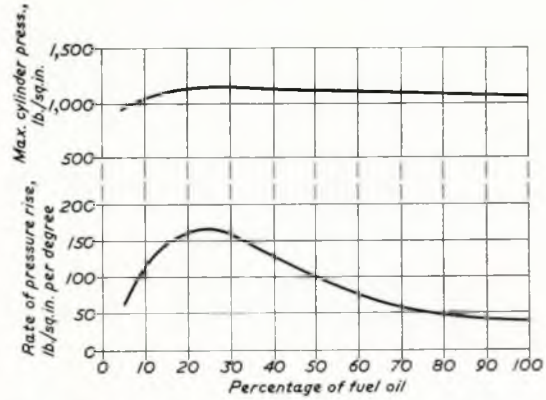


FIG. 25—*Maximum cylinder pressure and rate of pressure rise during combustion as functions of the percentage of fuel oil for a b.m.e.p. of 168lb./sq. in.*

Like Mr. Jones, the authors believed separate gas and air valves to be the safest means of introducing gas into the cylinders and they were extremely interested in his proposals for gas injection which they thought might well be followed in future.

In reply to Mr. Hopkins the limiting b.m.e.p. of the dual-fuel engine was due to detonation and not any other reason. In answer to Mr. Lilly's question the pilot charge was one of residual fuel as this was regarded as essential for the same reasons which he put forward. No difficulty with it had been experienced.

The authors thanked all the contributors for a most stimulating discussion.

Correspondence on "*Northern Star:* Evolution and Operation"

A further contribution to the discussion on the paper by Mr. G. S. Jackson and Mr. C. Winyard, published in the August 1964 issue of the TRANSACTIONS (*Volume 76, page 229*), *has been received and is given below, together with a brief reply by the authors.*

COMMANDER E. C. H. FEATHERSTONE, R.N. (Member) wrote to comment on the H.P. thrust failures and superheater slagging.

H.P. Thrust Failures

Provided a Michell type thrust bearing was properly designed and fed with clean oil, the collar and pads should never come in contact with each other, being always separated by a film of oil. It had always been taught that the white metal of the pads would extrude by high pressure before failure by contact with the thrust collar. He suggested that the failures had two more simple explanations:

i) Initial dirt in lubricating oil system from scale in pipes, residue sand in castings, dirt in drain tanks, etc . In new Admiralty vessels there was a very extensive programme to ensure that the lubricating oil system was free from foreign matter before starting trials. This included internal descaling of pipes, use of flushing oil to loosen sand and other foreign matter, pumping around with gauzes on all bearings, thorough clean out of lubricating oil drain tanks, lubricating oil reserve tanks, etc.

He gathered that the authors of the paper seemed to be of this opinion.

ii) Failure to keep the H.P. thrust block fully supplied with lubricating oil. This might be due to the layout of the pipelines, causing local starvation. It was of course essential that the lubricating oil gravity tank be always kept full and overflowing, though he had known this principle not always to be fully understood. Again, reverting to Admiralty installations, where an H.P. thrust failure was extremely rare, there was no gravity tank and the whole of the forced lubricating system was fed direct from the forced lubrication pumps.

Also the H.P. thrust was probably running at an initially higher temperature than many of the other bearings due to conduction from the rotor and was therefore more liable suddenly to attain dangerously high local temperatures (possibly not shown on the bearing thermometer), due to the causes just mentioned.

Slagging and Choking of Superheaters

This was always a hazard when burning Venezuelan residue, but steam atomization for some reason cut this down considerably. He had had two tankers, one with mechanical atomization and the other with steam atomization. Burning ostensibly the same type of fuel, there was considerable slagging and "birds-nesting" with mechanical atomization, but little or none with steam atomization. He could not explain the reason for this except that it was borne out in practice. Possibly steam atomization gave a better spray, though, to the eye, the two sprays were identical.

The authors acknowledged and thanked Commander Featherstone for his contribution to the discussion and made the following replies to the items raised:

H.P. Thrust Failures

i) They had no fault or issue to make with the recommended cleansing procedure advocated relative to the forced lubricating oil system, before any steaming of the turbines was carried out. In general, all the actions described were universal standard practice and were carried out in *Northern Star* for an extended period previous to any running of the turbines. This was made quite clear in the paper. The presence, however, of some foreign bodies in the oil stream was accepted as one of the contributory causes of the thrust failures.

ii) The supply and the running condition of the thrust as far as the lubricating oil was concerned, had been subjected to severe scrutiny by the experts, but no reason could be found to give grounds for serious consideration as a cause and this also applied to the conduction of heat along the rotor shaft to the journal. The lubricating oil system in *Northern Star* was in effect, the same as described by Commander Featherstone as Admiralty practice, even though a gravity tank was included. The purpose of the tanks was to provide a buffer for the bearings during the run-down period of the turbines, in the event of a failure of oil pressure due to the pump stopping or some such reason. The lubricating oil pumps discharged direct to the bearings and on to the top of a reflux valve, through which a $\frac{3}{8}$-in. bore hole was drilled and by which the gravity tank was kept overflowing.. The tank provided approximately a ten-minute supply of oil to the bearings.

Slagging and Choking of Superheaters

Steam atomization, applied to oil burning equipment, undoubtedly improved combustion, but it was by no means a panacea for slagging troubles. Slagging was a much more complex problem, as clearly illustrated by the discussion, being as it was, directly affected by fuel constituents, boiler design, relative to gas velocities, metal temperatures and ease of access for cleaning and, last but not least, consistently good combustion.

The authors took the opportunity of the publication of a reply to Commander Featherstone, to acknowledge a letter from Mr. Fowle, bringing to their notice the following statement in the reply to the discussion. This read, "chlorine bearing E.P. oil was invariably present in a machining failure". The word "invariably" was used in the context, there were exceptions to the general rule, and not in the dictionary sense. They trusted that Mr. Fowle would accept their apologies and substitute "generally" for invariably.

Erratum

On page 259 of the August 1964 issue of the TRANSACTIONS, in the sixth paragraph of a written contribution by Mr. B. Hildrew, M.Sc., D.I.C. (Member of Council), a temperature difference of 11 deg. F. (20 deg. C.) was given. This should have read 36 deg. F. (20 deg. C.), as intended by Mr. Hildrew, to whom we offer our apologies.

INSTITUTE ACTIVITIES

Minutes of Proceedings of the Ordinary Meeting Held at the Memorial Building on Tuesday, 11th February 1964

An Ordinary Meeting was held by the Institute on Tuesday, 11th February 1964, when a paper entitled "Service Performance of S.E.M.T. Pielstick Engines" by S. H. Henshall, B.Sc. (Member) and J. Gallois, was presented by the authors and discussed.

Commander F. M. Paskins, O.B.E., R.D., R.N.R. (Chairman of Council) was in the Chair and one hundred and thirty members and guests were present.

Thirteen speakers took part in the discussion which followed.

The Chairman proposed a vote of thanks to the authors which received enthusiastic acclaim.

The meeting ended at 7.55 p.m.

Section Meetings

Kingston upon Hull and Humber Area

Annual Dinner

The Fourteenth Annual Dinner of the Section was held on Friday, 6th November 1964, at the Guildhall, Kingston upon Hull, at 7.00 p.m.

Mr. G. W. Hill, M.B.E. (Chairman of the Section) presided over some 223 members and guests who had gathered in

Photograph by courtesy of the Hull Daily Mail and Yorkshire Times, Jameson Street, Hull.

Principal guests, with officers of the Kingston upon Hull and Humber Area Section, at the Fourteenth Annual Dinner held at the Guildhall, Kingston upon Hull, on Friday, 6th November 1964. Standing (from left to right): Mr. J. C. Proudfoot, Past Chairman of the North Midlands Section, Mr. J. A. Lacey, Chief Docks Manager, Hull Docks, Mr. H. F. Hesketh, Local Chairman of the Society of Consulting Marine Engineers and Ship Surveyors, Mr. A. C. Coward, M.B.E. (Member of Section Committee), Mr. R. R. M. Barr, General Manager of Associated Humber Lines Ltd., representing the Shipping Companies of the City, Mr. R. J. Shepherd, President of the Hull Association of Engineers, Mr. A. E. Franklin (Assistant Secretary of the Institute), Mr. S. Mathews (Member of Council), Mr. C. W. Fisher, President of the Hull Chamber of Commerce and Shipping, Mr. F. N. Sutcliffe (Member of Section Committee), representing Lloyd's Register of Shipping, Mr. G. A. Skelton, M.B.E. (Vice-Chairman of the Section), Mr. F. Atkinson, Chairman of the Humber Conservancy Board, Mr. Emlyn Jones, Principal of the Hull College of Technology, Mr. F. C. M. Heath (Vice-President), Mr. H. N. Pemberton (Vice-Chairman of Council), Mr. G. H. M. Hutchinson (Honorary Life Member and Founder Chairman of the Section), Mr. C. J. Potter (Honorary Treasurer) and Mr. D. A. Taylor (Honorary Secretary). Seated (from left to right): Alderman F. Hammond, Lord Mayor of Kingston upon Hull, Mr. G. W. Hill, M.B.E. (Chairman of the Section) and Mr. E. Chapman, Sheriff of Kingston upon Hull

Photograph by courtesy of the Hull Daily Mail and Yorkshire Times, Jameson Street, Hull.

Kingston upon Hull and Humber Area
At the Fourteenth Annual Dinner: The Vice-Chairman of Council, Mr. H. N. Pemberton (centre) in conversation with (from left to right) Mr. F. C. M. Heath (Vice-President), Alderman F. Hammond, Lord Mayor of Kingston upon Hull, Mr. E. Chapman, Sheriff of Kingston upon Hull, and Mr. G. W. Hill, M.B.E. (Chairman of the Kingston upon Hull and Humber Area Section)

the beautiful setting of the reception room. After a meal which reached new, high standards, Mr. S. Mathews (Member of Council), Principal Officer of the Ministry of Transport in Hull, proposed the toast "The City and County of Kingston upon Hull", in which he commented on the long traditions and dignity of the City. Large sums were being spent to increase the efficiency of the Hull Docks and although this was pleasing to see, Mr. Mathews felt that the communications to and from the city were still inadequate.

In replying to the toast, the Lord Mayor of Kingston upon Hull, Alderman F. Hammond, thanked the Section for its interest in the city and said how pleasing it was to give civic support to the function.

Mr. H. N. Pemberton (Vice-Chairman of Council), then proposed the toast "The Kingston upon Hull and Humber Area Section" and delivered a speech which showed to the members the dignity and sense of purpose which the Institute had at that time. Mr. Pemberton said that the strength and prestige of the Institute depended in the main on the standard of its technical discussions, its dissemination of knowledge and its character as a learned society. Because of its reputation, the Institute was able to play a leading part in the establishment of the Engineering Institutions Joint Council which, when the Royal Charter was granted early next year, would ensure by law the proper recognition of the title of the chartered engineer in the field of engineering technology. Far too long had engineering been regarded as subordinate to other sciences and arts in the academic world. Mr. Pemberton recently had heard of a headmaster's report on a boy in the sixth form which said "your son is not very bright but would probably get an honours degree in engineering". The object of the Joint Council and the Institute was to change all that.

The Chairman, in reply to the toast first welcomed the distinguished guests and in particular thanked Mr. Pemberton for his speech.

Turning to more serious matters Mr. Hill wondered how a first class degree would rank in twenty years time and suggested that it might have about the same value as the Higher School Certificate had about thirty years ago. He did not infer that the standard would be downgraded but that so many degrees would be obtained that employers would have to upgrade their minimum requirements. Mr. Hill went on to say that he had no doubts that the marine engineer, being a very adaptable person, would accept the challenge.

In the present day society, so enlarged in scope and so competitive in its demands on the available talent, engineers must fight to ensure their succession. Unless the industry was made as attractive as they knew it could be, they would be failing. Two elementary facts emerged from the "hot gospelling" of the General Election viz., modernization of education and industry must be closely allied to a better understanding of psychology and there must be an equal understanding of the forces which control the nation's destiny, in other words, more goodwill and co-operation.

So ended a most successful dinner.

Merseyside and North Western

A general meeting of the Section was held on Monday, 2nd November 1964, in the Conference Room of the Mersey Docks and Harbour Board, Dock Board Building, Pier Head, Liverpool 3, at 6.00 p.m., when a paper entitled "The application of Computers to Marine Engineering and Shipbuilding Problems" by Commander E. B. Good, R.N. (Member), was presented by the author.

The meeting was well attended by some eighty persons, including visitors from local shipyards and computer manufacturers. The paper was followed by a discussion and question period, during which contributions from the floor were made by seven members and visitors.

The standard of the discussion and contributions was particularly high and it was to be regretted that it was not possible to report them as they made an interesting adjunct to the original lecture.

A summary of Commander Good's paper follows.

Application of Computers to Marine Engineering and Shipbuilding

The paper presents a simple introduction to the part which computers can play in marine engineering and shipbuilding. It is written from the standpoint of an employer of computers in their broadest sense rather than a specialist computer programmer.

The composition of the paper covers 13 sections which can be grouped roughly into four main areas, namely, the opening sections which give an introduction to the various types of computers; and their broad method of operation; four specific examples involving the use of a digital computer; two specific examples involving the use of an analogue computer and, lastly, a general section and summary. The specific examples chosen are naval steam pipework design, thermodynamic performance estimates, main shafting analysis, critical path analysis, pressure variations in a boiler and main propulsion machinery and ship dynamic simulation.

Although the majority of the specific examples described have a strong research and design bias, it is hoped that the paper will help marine engineering and shipbuilding managements to a clearer insight into the operation of computers, the extent of their applicability and the advantages to be gained by employing them.

North East Coast

A junior lecture of the Section was held on Monday, 9th November 1964, at The Marine and Technical College, South Shields, at 3.45 p.m., when a lecture entitled "Marine Electrical Engineering" by J. W. Knights (Associate) was given by the author.

The attendance at the meeting numbered 540 of which approximately 190 were junior members of the Institute.

The lecture was attentively followed and highly appreciated although no one took advantage of the question period offered.

It was thought that with large meetings of that nature, the starting time should be advanced to 3.15 p.m. or earlier.

Scottish

General Meeting

A general meeting of the Section was held on Wednesday, 28th October 1964, in the Weir Hall of the Institution of Engineers and Shipbuilders in Scotland, 39 Elmbank Crescent, Glasgow, C.2, at 6.15 p.m.

Commander A. J. H. Goodwin, O.B.E., R.N. (Chairman of the Section), presided and, after extending a welcome to the thirty-eight members and visitors present, invited Mr. C. C. Herbert, B.Sc., and Mr. R. E. H. Orr to present their paper entitled "Computers for Marine Use".

The authors described the two types of computer—digital and analogue—from the point of view of computer users, not computer operators.

They took each computer in turn and spoke of the operation during the stages from suitable problems, through programming to the final result. Slides were shown of the equipment and sketches indicated the processes.

In early days the computer users were entirely separate from the programmers and the computer operators, and they tended to work in separate departments but usage has improved the relationship and the ideal position would be for the engineers to be programmers and in close relationship with the operators. A great deal depended upon the attitude of the computer staff. It was possible for a computer section to become an end in itself, a closed circuit with neither input nor output, and there was also a danger in managements employing their newly acquired computer to do "tricks" rather than to aid in day to day work of the establishment.

Real confidence only arose from actual use of the machine and it was probably necessary to encourage the engineering side (the user) to employ the computer on some un-productive or uneconomic problems in the first place, in order to establish confidence and in order to give some initial "feel" as to the sort of problem that was suitable for computer solution.

A very interesting discussion followed and much interest was shown in the contribution made by Mr. P. Jackson, M.Sc. (Member), who expressed the opinion that while the computer was better than the slide rule for torsional vibration characteristic calculations there were cases such as the best firing order for rotary, primary and secondary balance in a nine-cylinder engine where the calculation was done in under twenty-five hours by slide rule but might require many weeks to work out a programme.

The discussion was on a very high level and it was with regret that the Chairman had to close the meeting.

A vote of thanks to the speakers was ably proposed by Mr. D. S. Macfarlane (Member) and carried with loud applause.

The meeting closed at 7.50 p.m.

General Meeting

A general meeting of the Section was held on Wednesday, 11th November 1964, in the Weir Hall of the Institution of Engineers and Shipbuilders in Scotland, 39 Elmbank Crescent, Glasgow, C.2, at 6.15 p.m. when a paper entitled "Marine Auxiliaries" by J. R. Herd and D. M. Campbell, B.Sc., was presented by the authors.

Commander A. J. H. Goodwin, O.B.E., R.N. (Chairman of the Section) presided at the meeting and after extending a welcome to the fifty-two members and visitors present introduced the speakers.

Mr. Herd opened by pointing out that it was beyond the scope of a single paper to cover all the items of machinery which came under the heading of "Marine Auxiliaries", and he and Mr. Campbell had decided to deal with selected pieces of equipment, with one exception, taken from the machinery closely associated with the prime mover of a steam turbine driven vessel.

The authors proceeded to detail the present day design and capabilities of evaporators, including the Diesel waste heat evaporator, de-aerators, boiler feed pumps with particular reference to water lubricated bearings and feed water heaters, giving a brief history of development up to present "packaged" units.

It was interesting to note that it was some seventy years since the first evaporating plant was fitted into a ship and that, at that time, heating steam was generally in the range of 50 to 150lb./sq. in. gauge, generating vapour at a pressure slightly above atmospheric.

The interesting discussion that followed was handled by the authors in a very thorough and confident manner.

Glasgow had been isolated by dense fog for two days and the interest that the paper inspired was proved by the audience of fifty-two who attended.

The vote of thanks was in the capable hands of Mr. I. B. Smail, B.Sc.. (Eng.) (Associate Member) and was confirmed with enthusiasm by an appreciative audience.

The meeting closed at 7.48 p.m.

South East England

Junior Meeting

A junior meeting was held on Friday, 9th October 1964, at the Dockyard Technical College, H.M. Dockyard, Chatham, at 2.30 p.m.

The meeting which was arranged with the co-operation of Mr. J. A. Bess, B.Sc., Wh.Sch., Principal of the College, was attended by fifty students and six masters and members.

Mr. G. F. Forsdike (Chairman of the Section) was in the

Chair and introduced Mr. R. S. Hogg, who presented his paper entitled "Launching of Ships". Mr. Hogg displayed a complete command of his subject and gave a step by step verbal and pictorial description of the techniques necessary to ensure a safe and successful launching.

Subsequently the meeting was opened to questions and Mr. Hogg was able to answer these and elaborate in each instance, and thus in some measure satisfy the technical appetite of those assembled.

A vote of thanks to the speaker was proposed by Mr. Forsdike and was acclaimed by the meeting.

The meeting closed at 4.20 p.m.

Annual Dinner and Dance

The Second Annual Dinner and Dance of the Section was held on Friday, 16th October 1964, at the Gravesend Masonic Hall, at 7.30 p.m.

The Chairman of the Section, Mr. G. F. Forsdike, presided at the dinner and the principal guests were Mr. W. Young, C.B.E. (Chairman of Council), and Mrs. Young, and Mr. A. H. Wilson (Member of Council), and Mrs. Wilson. Bouquets were presented to the ladies on behalf of the Section, by Mrs. R. H. Cadle.

An excellent dinner, preceded by aperitifs, was enjoyed by 145 members, ladies and guests.

The Chairman proposed the Loyal Toast, followed by Mr. Young who gave the toast "The South East England Section". Mr. Young congratulated the Section on the decision to make the Dinner and Dance an annual event with a Ladies Night, and on the fact that it had been necessary to turn members away which amply justified that decision. He also stressed the importance of the Sections and the part they played in spreading the aims of the Institute.

In reply, Mr. Forsdike was emphatic in asserting that the Section had every intention of progressing and he looked forward to seeing all those present again on the occasion of the 1965 Dinner Dance.

The toast to the "Ladies and Guests" was proposed by the Vice-Chairman of the Section, Mr. A. H. Stobbs (Honorary Treasurer) who welcomed all guests and on behalf of the Section wished Mr. and Mrs. Young a successful and safe voyage on their tour of the Far East Sections.

Mr. Wilson responded on behalf of the guests and added his good wishes to the Chairman of Council.

Music during dinner was provided by Reg Simpson and Partner and dance music by the New Viceroys Group, under the guidance of Mr. R. Brown (Member) who officiated as Toastmaster and Master of Ceremonies throughout the evening.

During the supper interval, an excellent performance of topical Calypso and tropical Hawaiian songs and music was given by "Maxina y los Muchachos" and the Tradewinds, under the direction of their leader Dave Grew.

The evening's festivities closed at 1.00 a.m. after those present had joined in "Auld Lang Syne" in the customary manner.

South East England Section
Principal guests at the Second Annual Dinner and Dance at the Gravesend Masonic Hall. From left to right: The Chairman of Council, Mr. W. Young, C.B.E., and Mrs. Young, Mr. G. F. Forsdike (Chairman of the Section), and Mrs. Forsdike, Mrs. A. H. Wilson, and Mr. A. H. Wilson (Member of Council)

General Meeting

A general meeting of the Section was held on Tuesday, 20th October 1964, at the Clarendon Royal Hotel, Gravesend, at 7.30 p.m., when a paper entitled "*Northern Star*: Evolution and Operation" by G. S. Jackson (Member of Council) and C. Winyard (Member) was presented by the authors.

Mr. G. F. Forsdike (Chairman of the Section) presided at the meeting which was attended by forty members and guests.

The paper, supported by various slides which served to illustrate the many difficulties and novel features of the ship, was received with great attention. Numerous questions relating to the design and the troubles that befell the ship on her maiden voyage were answered by Mr. Jackson and Mr. Winyard. Ultimately the Chairman had to call a halt to the questions. He proposed a vote of thanks to the authors for their paper and this was seconded by all present.

The meeting closed at 10.00 p.m.

Technical Visit

A technical visit to the Arthur Guinness, Son and Co. Ltd., Park Royal Brewery, London, was made by twenty members of the Section on Saturday, 31st October 1964, at 2.30 p.m.

The party spent a pleasant and interesting afternoon and were privileged in having the Superintendent Engineer, Mr. Scott, himself an ex marine engineer, to accompany the party and answer the many technical questions put by the members.

There were many examples of the craftsmen of twenty-eight years ago, the "copper boilers" with their riveted seams, so perfectly formed; copper "bends" and "tee" pieces, peened, planished and formed by the artistic hands of a true craftsman of that era.

The extensive use of stainless steel was now evident throughout, only two of the original wooden vats installed when the brewery was built were still in use being replaced as they become obsolete.

At the end of the tour members were taken over the power station where the resemblance to a ship's engine room did not stop at the machinery, all the five shift engineers were all ex marine engineers and the shift engineer on duty a member of the Institute.

The tour ended with the party being entertained in the true Guinness fashion. Thus fortified for the journey home, it was left to the Honorary Secretary, Mr. R. H. Cadle, to propose a vote of thanks to the hosts and guide which was seconded by all present.

South Wales

Junior Meeting

A junior meeting of the Section was held on Friday, 23rd October 1964, at Swansea Technical College, at 7.00 p.m., when a paper entitled "Ships and Fire" by C. N. Bidgood, O.B.E., M.I.F.E., was presented by the author, Chief Fire Officer, City of Cardiff Fire Service.

Mr. T. C. Bishop (Chairman of the Section) presided at the meeting and the audience of fifty-five comprised of members from Cardiff, Swansea and the surrounding districts, students from the Llandarcy Oil Refinery and members of the local fire services.

An excellent address was followed by a lively discussion and more than a dozen questions were ably dealt with by Mr. Bidgood in an efficient manner.

Mr. R. A. Simpson (Corresponding Member, Swansea), expressed the appreciation of those present in his vote of thanks to the speaker and a successful evening was concluded by a vote of thanks to the Chairman.

Annual Dinner

The Annual Dinner of the Section was held on Friday, 6th November 1964, at the Royal Hotel, Cardiff.

Mr. T. C. Bishop (Chairman of the Section) presided at the Dinner at which 160 members and guests were present. Among the guests were the Honorary Treasurer of the Institute, Mr. J. Calderwood, M.Sc. (Honorary Vice-President), and representatives of many of the major shipping companies.

After the Loyal Toast the toast "The Shipping Industry" was proposed by Mr. D. M. Morgan Rees, M.A. (Cantab.), LL.B., who made lighthearted references to his varied experiences whilst abroad. The response to the toast was made by Sir Charles G. Alexander, Bt., M.A. (Associate), in a speech which so ably embraced the present operational and financial problems of shipowners as well as those that were encountered by local members engaged with the repair and maintenance of vessels.

Mr. Calderwood, in proposing the toast "The South Wales Section" referred to the obvious virility of the Section and expressed regret that other Members of Council had been unable to attend the Dinner owing to prior engagements.

The Chairman in his reply, paid tribute to the ready help always given from Headquarters and thanked both Members of Committee and the members of the Section for their excellent support. He made particular reference to the work of Mr. D. Skae (Vice-President) and of Mr. R. A. Simpson (Corresponding Member, Swansea), which had so largely contributed to the success of the Section. In conclusion he welcomed the visitors to the Dinner.

Mr. W. D. A. Rowlands, M.A., replied on behalf of the visitors and thanked the Chairman and members for their generous hospitality.

West of England

A general meeting of the Section was held on Monday, 9th November 1964, in the Small Engineering Lecture Theatre, Queens Buildings, University of Bristol, at 7.30 p.m., when a paper entitled "Some Factors Influencing the Life of Marine Crankshafts" by S. Archer, M.Sc. (Member of Council) was presented by the author.

Mr. J. P. Vickery (Vice-Chairman of the Section) was in the Chair and the audience, which included Mr. F. C. Tottle, M.B.E. (Local Vice-President) numbered twenty-four.

Mr. Archer, who illustrated his lecture with slides, said that the concept of the crank had been understood for hundreds of years and that an Italian manuscript of 1335 had shown a crankshaft which was used in a crank-propelled paddle boat. It was in the year 1800 that serious development of the crankshaft took place and in 1840, the material used was changed from wrought iron to forged steel. Failures occurred in those days due to overheated bearings and bad forgings. Torsional vibrations also exacted a heavy toll. However, due to improvements in foundry technology, non-destructive testing of metals, and better design, a high order of reliability had been achieved.

Mr. Archer gave a very interesting report of crankshaft defects over a ten-year period from 1953, and drew certain conclusions as to the relative reliability of the different types.

The paper was enthusiastically received with eight members asking questions.

In conclusion, Mr. Vickery thanked the speaker and the meeting ended at 9.35 p.m.

Overseas

Auckland

The Annual Dinner and Dance of the Section was held on Saturday, 10th October 1964, in the Berkeley Lounge, Mission Bay, Auckland.

Seventy-six members and guests attended.

After the dinner, dancing to Gordon Kerr's Debonair Band continued throughout the evening.

The function ended at midnight and all those present appeared to have thoroughly enjoyed themselves.

Election of Members

Elected on 16th November 1964

MEMBERS

William Louis Herbert Clayton, Lt. Cdr., R.N.
Stanley Bradley Cusworth
Edward Ives Forster
Andrew Gibb
Derrick Griffiths
John William Hickey, M.B.E.
David McCallum
Robert Brown McKendrick
William Lewis Mitchell, Eng. Lt. Cdr., R.N.
Robert Henry Mocatta, Lt. Cdr., R.N.
Douglas Thomas John Nicholson
Arnold Joseph Rodricks
Leslie Richard Sharp
Mohammad Aslam Toosy, Capt. (E), P.N.
Eric Wishart
Moritada Yamamoto

ASSOCIATE MEMBERS

Muhammad Riaz Ahmad
Beji Hormusji Balsara
Alfred George Botley
Desmond Edward Byrne
Gordon Campbell

George Chapman
Joe Craggs
Ronald Cunningham
Leslie Douglas Dean
Charles Donaldson
John Thomas Alban Elliott
Gordon Farquharson
David Neil Flather
Fred Albert Forgan
Thomas A. Gahan
Ronald Esmond Gardiner
Brian Hayter
Anthony Highley, B.Sc. (Mech.Eng.), Cardiff
Warren David Hopkins
Lloyd Anthony Houghton
Malcolm MacGregor Wilson Howie
Charles Albert Hunter
Ratan Lal Jain, B.Sc. (Agra)
Kenneth Charles Jones
Vedat Kanoglu
Philip Kernahan
John Ralph Leggate
Charles David Low
John Mannis
Jaybrata Mukhopadhyay, B.Sc. (Calcutta)
Rex John Neate
William James Nelson
Louis Maurice O'Sullivan
Peter Otway
Goronwy Owens
David Albert Park
Richard Allen Plummer, Eng. Lieut., R.N.
Kenneth Harold Reah
John Bertram Rimmer
Keith Roberts
Alexander Pratt Robertson
Kevin Routledge
James Robert Seaward
Michael Rodney Smith
Clifford Russell Speed
Gerald Twigge
William Stewart Watson, Eng. Lieut., R.N.Z.N.
Robert Paul Williams

ASSOCIATES
Khwaja Rafi Ahmad
Peter Arthur Briggs
Edmund Wilhelm Brillant
Sydney Burgess
Archibald McLean Galbraith
Md. Sahabuddin Khan
George Henry Adam Lee
Donald James Macintosh
Edward Charles Peckham
Patrick Joseph Ritchie
Alexander Renfrew Young

GRADUATES
David Bell
Murray Cowie
Parvataneni Ramesh Chandra Datt
Peter Dodson Hindle
James Bruce Alexander Merry, Lieut., R.N.
Brian Christopher Patrick Seignott
Brian Rex Sheppard
Martin Van Der Merwe
David Gordon Watkins
Rodney Ian Michael Wood, Lieut., R.N.

STUDENTS
John Graham Balderstone
Dennis Brooks
David Kynd Brown
Jeremy Drew
Ee Lee Yah
Colin Ernest Fletcher
Geoffrey McLelland
Jeffrey Moylan
Anthony Douglas Muncer
David Malcolm Pink
Geoffrey Stone
Kenneth Thompson
Geoffrey Holland Youngs

PROBATIONER STUDENTS
Nigel Geoffrey Russell Bateman
John Charles Baylem
Roger Alan Burton
Roy Charnock
R. M. Cholawo
Peter Terence Concannon
David John Croston
John Stanley Dobson
John Edwin Gregory
Raymond Housley
Ian Philip Hulme
Roger Ison
Denis Wynne Jones
Stuart Alexander Lord
Colin Collingwood McKenzie
Brian Wilson Maxwell
Timothy Russell Noble
Thomas Hubert Powell
Robert James Pritchard
Adrian Jacques Sadler

TRANSFERRED FROM ASSOCIATE MEMBER TO MEMBER
Barrie James Austin, Lt. Cdr., R.N.
John Beadle
Frederick Charles Bown
Clelland Campbell
William Vance Crockett
Andrew Cushnaghan
Raymond Patrick Duell
Peter Michael Fitzpatrick
James Melville Fulton
Owen Thomas Griffith
Eric Coleman Knowles
Harold Douglas Makinson
Brian John Mears
James McArthur Merrilees
Anthony John Monk, Cdr., R.N., B.Sc. (Eng.), London
Colin Alexander Osborne
Alan Robert Webb

TRANSFERRED FROM ASSOCIATE TO MEMBER
Alfred William Edward Jury
Leo B. McGuinness
Edward Francis Lucas Pimlott
James Leslie Scott
Gilbert Smith
John Lindsay Stewart
Richard Wilson

TRANSFERRED FROM GRADUATE TO MEMBER
Kees Watts

TRANSFERRED FROM ASSOCIATE TO ASSOCIATE MEMBER
Robert Joseph Tozer

TRANSFERRED FROM GRADUATE TO ASSOCIATE MEMBER
Harold James Arnold
John Charles Beland
Geoffrey McKinnon Billings
John Oliver Brinkley

Brian John Down
Richard Arthur Jones
Robert Cutter Jordan
Stanley Keith Metcalf
Gordon Grant Simpson
Paul Charles Keats West
Alan Wilson

TRANSFERRED FROM STUDENT TO ASSOCIATE MEMBER

Brian Michael Egan
David Reid
Michael David Seaman
Michael Ray Thornton

TRANSFERRED FROM PROBATIONER STUDENT TO ASSOCIATE MEMBER

Christopher Daws
Brian Clifford Jackman

TRANSFERRED FROM GRADUATE TO ASSOCIATE

William Charles Classon
James Coutts Dale
James Wallace Downie
Stanley Maxwell Pilling

TRANSFERRED FROM STUDENT TO GRADUATE

Sheridan Irwin Lee
Kenneth Charles Williamson

OBITUARY

CAPTAIN HERBERT FARQUHAR ATKINS, D.S.O., D.S.C., R.N. (Member 14645) died on 4th August 1964, in his sixtieth year.

Captain Atkins was trained at the Royal Naval Colleges at Osborne and Dartmouth, at the Royal Naval Engineering College, Keyham, and at H.M. Dockyard, Devonport. He underwent sea training in H.M.S. *Thunderer*.

In 1926, he joined H.M.S. *Hood* as Lieutenant (E) on watchkeeping duties. During the next twelve years he served at sea as Lieutenant (E) in H.M.S. *Queen Elizabeth*, from 1929 to 1931, and as Lieutenant (E) and Lieutenant-Commander (E) in H.M.S. *Cornwall*, from 1933 to 1935. His tours of shore duty during this same period consisted of an advanced course in marine engineering at the Royal Naval College, Greenwich, from 1927 to 1929, and appointments as Assistant Admiralty Overseer at Sheffield, from 1931 to 1933, and as Assistant Professor of Marine Engineering, R.N.C., Greenwich, from 1935 to 1938. In the latter year he was promoted Commander (E).

During the Second World War, he served in H.M.S. *Emerald* and *Orion*. He also held an appointment as First Assistant to the Engineer Manager at H.M. Dockyard, Portsmouth, and later served in the Department of the Engineer-in-Chief of the Fleet.

In 1945, he became Engineer Officer in H.M.S. *Formidable* and, two years later, was appointed Senior Instructional Officer, Naval Damage Control School, H.M.S. *Phoenix*. He was promoted Captain (E) in 1949 and in the same year received an appointment as Assisting Naval Assistant to the Second Sea Lord at Admiralty. He became Superintendent of the Naval Wing at the National Gas Turbine Establishment in 1951.

On his retirement from the Royal Navy in 1957, he was appointed Secretary to the Executive Committee of Vickers Nuclear Engineering, and assistant to Rear-Admiral Sir Edward Rebbeck, K.B.E. He held this appointment until 1963.

Captain Atkins was elected a Member of the Institute on 11th November 1953; he was also a Member of the Institution of Mechanical Engineers.

DANIEL MACKAY BAIN (Member 8588) was born on 25th August 1908. After serving an apprenticeship with John Brown and Co. Ltd., from 1918 to 1923, he went to sea as an engineer with the Orient Steam Navigation Co. Ltd.; from 1928-1937 he served in Royal Fleet Auxiliary vessels. During his sea service he gained a First Class Board of Trade Certificate. In 1937 he became a shift engineer at the Ford Motor Co. power station at Dagenham and later was made assistant foremen there. He accepted an appointment as an engineer surveyor with the Cornhill Insurance Co. Ltd., in 1940, and remained in the service of that company until the time of his death, on 12th August 1964. He served in the Newcastle upon Tyne area, also in Northern Ireland, and, since 1947, had been working in Southern England. He was well esteemed by his business colleagues and all others who knew him.

Mr. Bain was elected a Member of the Institute on 7th February 1938. He leaves a widow.

FREDERICK BERTRAM BLAKEY (Member 8742), a Member of this Institute since 10th October 1938, died suddenly on 2nd July 1964, following an operation for the removal of a gastric ulcer.

Mr. Blakey was born on 7th February 1898, the son of Frederick William Blakey, an engineer with Broken Hill Pty. Ltd., who had retired from sea service as chief engineer of the s.s. *Aorangi*. The family later settled in Auckland, New Zealand, where Mr. Blakey Senior went into business on his own account as a marine surveyor.

Mr. F. B. Blakey was educated at the Auckland Technical College at Mt. Eden, prior to serving his apprenticeship with Geo. Fraser and Sons Ltd. He went to sea as a junior engineer with the Port Line and served with that company until his marriage to his first wife, Florence, in 1927. During his service with the Port Line he gained a First Class Board of Trade Certificate. Following his marriage, he spent about two years with the Vacuum Oil Company, N.Z. Ltd. at Wellington, as supervisor. In 1930 he joined S. T. Williamson Shipping Co. (Hong Kong) and took one of their ships out to Hong Kong, where he remained as chief engineer with the line until 1938, in which year he rejoined the Standard-Vacuum Oil Company of New York, in Hong Kong, in a shore appointment. He trained as a Terminal Superintendent at the Lai Chi Kok installation and, on completion of his training course, was transferred to Penang, Malaya, as Terminal Superintendent at Butterworth.

Mr. Blakey was in New Zealand on furlough when Singapore fell to the Japanese forces and he rejoined the Standard-Vaccum Oil Company in Wellington for the duration of the war.

At the end of hostilities with Japan, he was ordered to join the Petroleum Board of South East Asia Command, at Kandy, Ceylon, with the rank of Temporary Major. From Kandy he was sent to Hong Kong to assist in the rehabilitation of the oil installations. Later he was transferred back to Penang and remained there until S.E.A.C. was disbanded, when he assumed duty as Operations Superintendent, North Malaya, for the Standard-Vaccum Oil Company, a post which he held until the beginning of 1953.

From the latter year until his retirement in 1956, he was Operations Executive, Malaya, with the same company, stationed in Singapore. He retired to the Crown Colony of Fiji where the climate suited him after so many years in the Far East.

Mr. Blakey is survived by his second wife, Beatrice, whom he married in 1960, his first wife having died in 1951. There is also the daughter of his first marriage.

CAMERON GIBB (Associate 10679), who was born on 23rd January 1903, served his apprenticeship with Barclay Curle and Co. Ltd., for five years. He afterward served from junior to second engineer with the British India Steam Navigation Co. Ltd. After leaving the sea, he became charge engineer with Messrs. R. and J. Garroway and, in 1938, took up an appointment as chief engineer to Burrell and Co. Ltd. He held that position at the time of his death.

Mr. Gibb was elected an Associate of the Institute on 5th February 1946 and took a keen interest in its progress, attending many of the technical meetings. His death occurred on 13th July 1964.