Service Performance with the Fiat Marine Diesel Engine Type 900S

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Presented by ***Dott. Ing. A. Gregoretti.***

This paper starts with a short description of the evolution in the marine Diesel engine field, towards high outputs, with particular reference to the circumstances which led Fiat to design and build the 900S type engine.

Afterwards, the structural characteristics of this engine, based on 900 mm. bore cylinders and pistons with 1,600 mm. stroke developing an output of 2,300 b.h.p./cylinder, are set forth, as well as the most interesting constructional characteristics of these machines.

The paper includes the results of a series of tests carried out with a proper experimental installation on a two-cylinder set. Some technical results obtained in the same tests especially concerning the behaviour of liners and piston rings, are also included.

The paper also deals with the main operating data of 14 engines and gives some information concerning the behaviour of these engines in actual service. A report is made on some improvements carried out following practical experience.

INTRODUCTION

The evolution of shipbuilding towards ships of ever increasing dimensions—a tendency as old as navigation itself and formerly developed according to a law with linear rate—has gained momentum during the last ten years.

The output necessary for the propulsion of ever larger and faster vessels has obviously increased and the requirements for higher powered machinery compelled the builders to face new problems.

It has been possible for the Diesel engine builders to meet the owners' requirements, as in the last ten years, contemporaneously with the new needs, a considerable technical evolution has been apparent. The adoption of supercharging and other advances, both in materials and lubrication, have allowed the construction of machines of which the power exceeds to a sufficient extent the maximum which could be considered attainable ten years ago.

This evolution enabled all the builders of large engines to build large-bore engines; this means that from the almost universal standard bore of between 720 mm. and 780 mm., 840 mm. and 930 mm. bores have been attained and some builders are already thinking of adopting larger bores.

It is known that large-bore cylinders, with dimensions in the foregoing range, are not a recent technical attainment as since 1914, engines with 750 and 900 mm. bore have been built. These endeavours were not continued as the oldest machines were based on programmes too advanced with respect to the actual state of the technique at that time. Other following constructions, though having acceptable performance, did not give better results than those obtainable with smaller cylinders, sufficient to cover, at that time, the required power range.

Fiat began to consider this problem in 1958, with the design of cylinders having 900 mm. bore for which was foreseen a possible output of about 1,900 b.h.p./cylinder.

The technical design of this engine was not very different from the 750 mm. bore engines built up to that time; it was based on the experience acquired through the operation of very many engines of this type, and on the performance of many experimental parts of more advanced design, already in service for some years on existing engines.

On the other hand, due to the remarkable extrapolation with respect to the existing dimensions (20 per cent increase on the piston diameter involves an increase at least of 50 per cent on the forces involved, due to the increase of the service pressures), it was deemed advisable to build, initially, an experimental two-cylinder unit which began operation in 1959. At the same time, an order was accepted, for a nine-cylinder engine which was completed in December 1960. Soon, test results allowed raising the power to 2,100 b.h.p./cylinder.

A number of engines with cylinders of the same type have been successively ordered to date; up to 31st April 1964, Fiat and their Italian licencees, Ansaldo and Cantieri Riuniti dell'Adriatico, built nine eight-cylinder engines, four nine-cylinder engines and a 12-cylinder engine. Another six nine-cylinder engines and seven ten-cylinder engines, a total of 13 engines and 124 cylinders are on order or under construction. Fourteen engines are currently operating regularly

The results of test and workshop research and the operating results of these engines are the principal object of this paper. However, it is considered convenient beforehand to give a short description and to make known some ideas which were followed in the design of these engines.

DESCRIPTION OF THE 900S ENGINE

In the main, the informative criteria and the structural description of the Fiat engines were set forth in a paper which the author presented at this Institute[1]. Those interested in going more deeply into the matter may refer to this paper. In the present paper, only those features which can be considered as specific to Fiat engines and, in particular, type 900S, will be discussed.

The construction of the Fiat type 900S engine is shown in Figs. 1 and 2, which show the cross-section and longitudinal section respectively. The external appearance and some parts from eight, nine and 12-cylinder engines are shown in Figs. 3, 4, 5 and 6, chosen from the most interesting engines.

The 900S type engine has 900 mm. bore cylinders, with 1,600 mm. piston stroke. The output for which the engine

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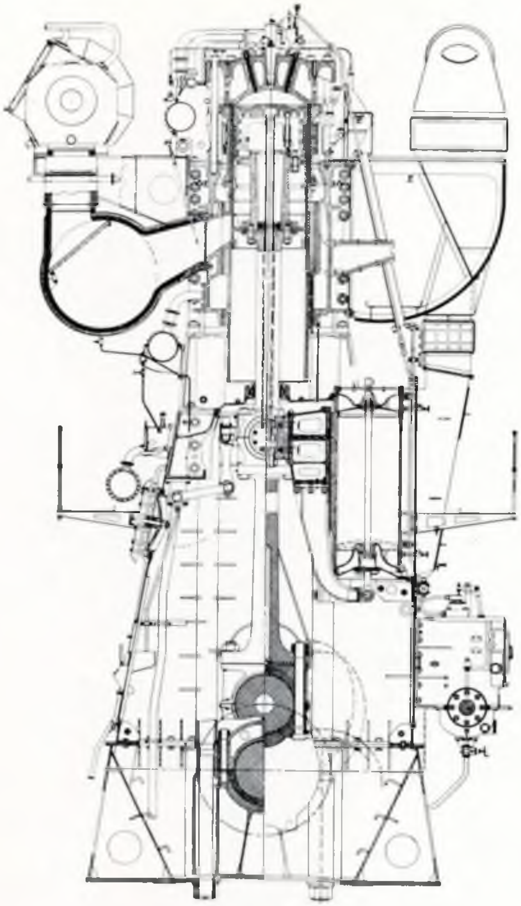


FIG. 1—*Fiat engine type 900S—Cross-section*

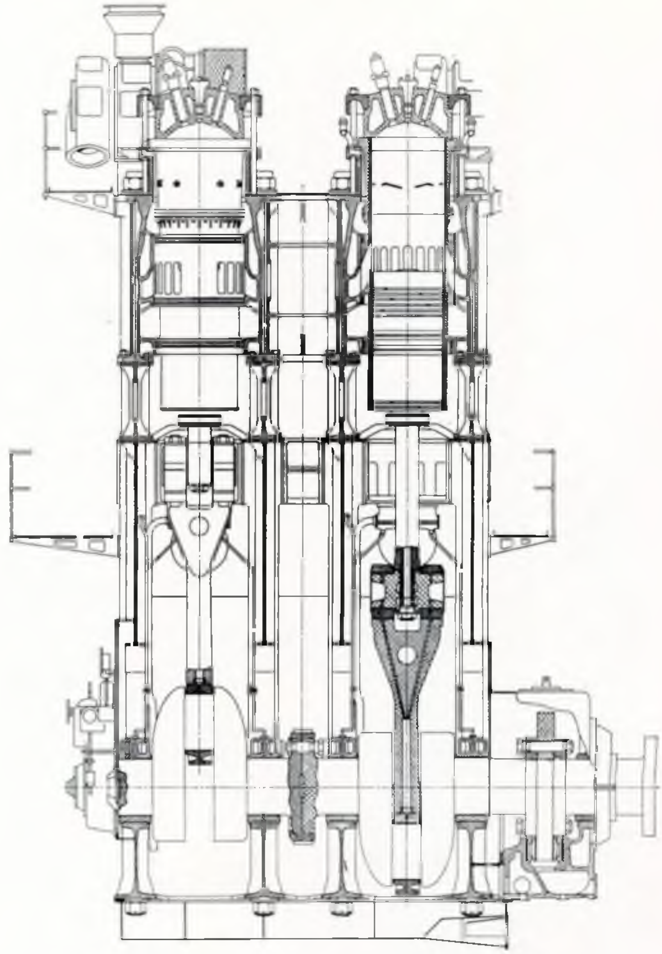


FIG. 2—*Fiat engine type 900S—Longitudinal section*

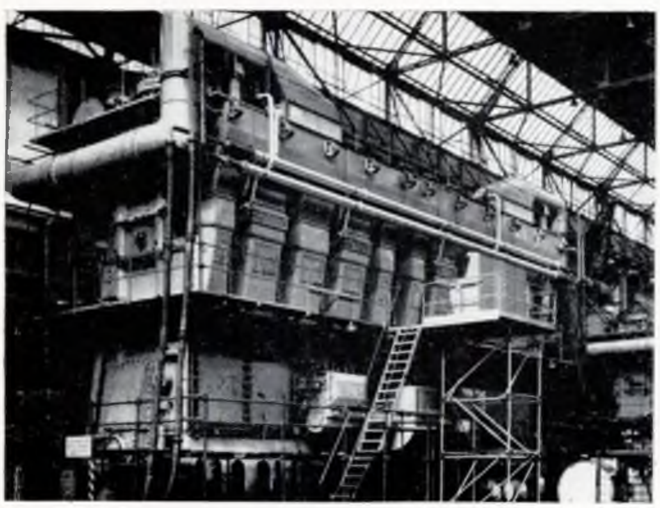


FIG. 3—*The Fiat 909S engine on the test bed—It may be seen that the arrangement of the manœuvre console is separate from the engine, placed on an intermediate grating*

FIG. 4—*The Fiat engine type 9012S on the test bed—In this case also the manœuvre console is separate from the engine*

FIG. 5—*Fiat engine type 9012S—View of cylinder heads photographed in the engine room of m.t.* Carlo Cameli

FIG. 6—*Bedplate and crankshaft of Fiat engine type 9012S during assembly*

was supplied up to the end of 1963 is 2,100 b.h.p./cylinder at 122 r.p.m. corresponding to 7·61 kg./sq. cm. m.e.p. and 6·51 m./sec. piston speed; basede on the experience obtained through the shop trials and operation at sea, the output was raised to 2,300 b.h.p./cylinder (m.e.p. = 8·33), maintaining the same revolutions with a possible ten per cent overload margin.

As with all single-acting Fiat engines, this engine has cross-scavenging, i.e. with air intake and exhaust only through ports and is supercharged with constant-pressure exhaust gas turboblowers, operating in series with scavenging pumps driven by the main pistons.

The framework of the engine consists of a welded bedplate structure, a group of welded A-frames and a block of cast iron cylinder bodies.

The longitudinal connexions are dimensioned so as to assure the necessary longitudinal stiffness, and bedplate, A-frames and cylinders are tightened vertically by steel tie-rods which connect the upper part of the cylinders with the bedplate bottom.

The principal feature which characterizes the Fiat engine in comparison with the others on the market, is the crankcase completely separated from the main cylinders, which are open at the bottom to the engine room (Fig. 7).

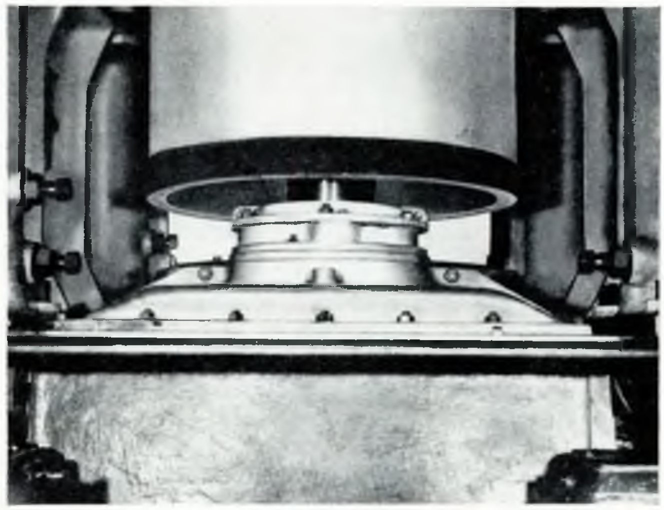


FIG. 7—*Separating diaphragm between cylinder and crankcase, open and inspected from outside*

This type of design does not permit the use of the region under the cylinders as a reservoir or additional scavenging pump, it is more expensive and means a higher engine. However, it gives the following advantages:

a) It is possible to check piston behaviour continuously during operation and it is also possible to check the quality and quantity of fuel oil residue passing around the piston.

b) The seal arrangement around the piston rod, where the latter passes through the diaphragm closing off the crankcase, is completely accessible and works without any pressure differential between the inside and outside. There is thus no pressure action to drive lubricating oil and combustion residues through the seal, which consequently is always clean and efficient.

c) Due to the efficiency of the seal, it is impossible for combustion residues to contaminate the lubricating oil.

d) The combustion residues cannot accumulate in the region under the pistons, because of the provision of suitable discharge arrangements and because easy continuous cleaning of the whole space is possible.

e) Any possibility of fires in the region below the main cylinders is completely precluded. Such fires have in many cases caused explosions and very serious damage in other engines where the same space is enclosed.

f) The efficiency of the sealing arrangement in the oil scraper elements on the piston rod, which is assured for the previously stated reasons, allows total and permanent checks on the oil quantity that the rod may draw out of the crankcase. In this way, the general lubricating oil consumption is appreciably reduced and these conditions are maintained throughout the life of the engine.

The crankshaft is of the usual semi-built type, with cast steel parts; it has been dimensioned with a 700 mm. diameter for the main journals and the crankpins. According to the classification societies, these dimensions are also suitable for the 12-cylinder engine. A weight limit of 70 tons has been accepted for the crankshaft, allowing a five-crank section. Consequently, the crankshafts of engines up to ten cylinders have two shaft sections, the 12-cylinder engines have three shaft sections.

The cylinder body consists of two parts, the lower is part of the main structure of the engine, whilst the upper is separate, without lateral connexions.

The cylinder liner is of the usual Fiat construction with a steel upper part, a wear resistant inner cast iron bush, and cast iron lower part.

Unlike the earlier engines, in this case the cylinder head is totally external to the liner; this construction is somewhat lighter, reduces the engine height and the dismantling headroom; for this reason it was preferred for this large machine. To reduce the temperature of the upper liner flange which, again as distinct from the earlier engines, is not insulated against heat from the lower head part, suitable water circulation has been adopted for the outer part of the flange.

The piston of this engine is of a thin-walled type, resting and centred on (by means of ribs) an internal portion directly attached to the piston rod. With this construction, the pressure exerted on the bottom is directly transmitted through the ribs to the piston rod. The cylindrical part of the skirt does not transmit any mechanical stress and has only to resist the external gas pressure. The various internal ribs are made so as to form channels for the cooling oil to circulate with a positive flow and known speed, and without any danger of dead spaces where deposits and carbon incrustations may collect.

This design has allowed the retention of oil as the means of piston cooling, and a reduction in the maximum temperature of the piston head by about 100 deg. C. (180 deg. F.) in comparison with the old design, thus bringing this temperature into a range where the material can resist the combustion oxidizing action indefinitely.

For all large engines, Fiat has adopted cross-scavenging and constant pressure supercharging. It was considered that several advantages would compensate for the presence of the additional reciprocating scavenge pump to be installed in the supercharging system, i.e.:

i) no large exhaust valves on the cylinder head;
ii) the elimination of the valve camshaft, drive and reversing devices;
iii) constant pressure and flow through the supercharging turbines, without inducing in them dynamic actions due to discontinuous gas flow; it is known that these dynamic actions give rise to reduced life and also to the possibility of failure of the turbine blades and rotor bearings.

The cost of the pumps (incorporated in the shoe guides) and that of the related pistons, is no higher than that of the exhaust valves and of their control in a longitudinal scavenging engine. As far as maintenance is concerned, experience has proved that the adoption of scavenging pumps implies no additional charge, which cannot be said for the exhaust valves and their controls.

From the thermodynamic standpoint, the utilization of constant pressure allows a greater expansion in the main cylinder. It is not necessary to advance, by almost a half-stroke of the piston, the opening of the exhaust to operate the turbines of the turboblower, as is the case when the latter are working on the pulse system. The higher indicated power, obtainable with respect to the engines working with advanced exhaust, is used to drive the scavenge pumps, keeping the whole efficiency of the supercharging cycle practically unchanged in both cases.

The other structural components (fuel pumps, injection and starting devices, lubrication, manœuvring and reversing arrangements) are fundamentally the same as successfully used in other Fiat engines.

TESTS CARRIED OUT ON THE TWO-CYLINDER ENGINE

As stated in the foregoing paragraphs, the preliminary operational experience was obtained on a two-cylinder engine installed in the factory test shop. At the end of the first cycle of operation, this engine was moved to a new test shop suitably built, equipped with all the most modern research and control instruments (see Figs. 8 and 9). This new installation has been allocated for a closer study of all the problems which may arise following practical operation in ships, and for the study of all those steps which can improve the performance and behaviour of large engines.

The first series of tests, conducted in 1959-1960, was devoted to the general tuning-up of the engine and to the testing of the general design of the various components. As it was impossible to carry out tests over long periods, the author's company continued the practice, followed, from the beginning, by Fiat in the construction of new engines, of bringing the machinery up to its highest overload. This policy makes the research into, and elimination of eventual weak points, easier.

FIG. 8—*External view of the new experimental test room of Fiat—Grandi Motori Works*

After a cycle, of reseach and control, at successively increasing outputs, the engine was, in February 1960, brought up to 3,000 b.h.p./cylinder, in the presence of the representatives of the main Registers and Classification Societies, thus attaining what was, at that time, a world record for Diesel engines.

The results of these early tests have made it possible to raise the output of the production engines from 1,900 to 2,100 b.h.p./cylinder. The remarkable possibilities offered by the results from the experimental engine would have allowed an increase in the output to a higher level; it was, however, preferred to keep the rating within sensible limits and to await the results of further shop trials and to ascertain the behaviour in operation of the first engines before going for a further increase in output.

In 1962, the engine was installed in the new experimental plant with a view to operating without the restrictions imposed by production requirements. A programme of long tests at constant high load was carried out, analysing at the same time,

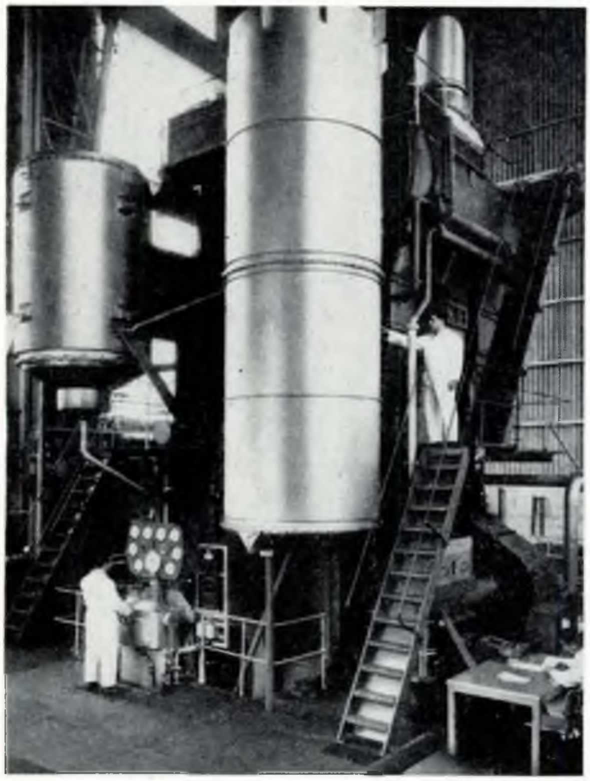


FIG. 9—*Two-cylinder experimental engine type 902S in the new test room. The two additional manifolds used in the 2-cylinder engine to avoid excessive pressure oscillations of exhaust gas and supercharging air may be seen in the foreground*

the behaviour of those elements responsible for the engine's resistance to high loads, using advanced research methods.

Among the most significant tests, the following must be mentioned:

a) a test of 500 continuous operating hours at 2,200 b.h.p./cylinder at 122 r.p.m. (m.e.p. = 8 kg./sq. cm.) on medium quality fuel (light marine fuel of 650 sec. Redwood viscosity);

b) a test of 1,000 continuous operating hours conducted at the same rating on Bunker " C " type boiler oil of 1,500 sec. Redwood viscosity (this was the worst quality boiler oil found at that time on the Italian market);

c) a 125-hour test, also on the same boiler oil, at 2,550 b.h.p./cylinder, 122 r.p.m. (m.e.p. = 9·2 kg./sq. cm.);

d) another test of the same duration at 2,750 b.h.p. output, 122 r.p.m. using the same boiler oil (m.e.p. = 10 kg./sq. cm.).

All tests were carried out without any stops and the condition of the engine was found to be satisfactory, in all its components.

Several other tests of shorter duration, were conducted to check the behaviour of several changes introduced in the design of some component parts. As already reported, the original design of the engine is the result of those things that practical experience, achieved in the past, had proved to be most satisfactory; the availability of an experimental engine has however, allowed different solutions to be tried and something better to be sought. Different designs for the upper section of the main cylinder liner have thus been tested, as well as different proportions of the piston seal ring and minor changes in the internal profile of the piston cooling channels.

Similar research based on traditional test shop methods would have given no appreciable results; in this case it is considered to have succeeded, with the help of the most advanced technical means, in recording a remarkable number of data, from which it has been possible to draw quite reliable conclusions. For example, it was possible to measure the continuous temperature at 72 different points on the cylinder liner at about 1 mm. under the surface exposed to heat, and to make the same measurements at 24 points on the piston heads. This permitted an efficient study of different solutions, among which that giving the lowest temperature, in the parts involved, was considered to be the best.

It is not the aim of this paper to go into details, as far as the results of these tests are concerned. Some parts, of new design, are already being tested in engines in ships, and may be introduced into standard construction after more running experience. In any case, the comparative results between the different constructional solutions applied and tested, and the results of continuous recording during lengthy operation permitted some idea of the internal life of the engine to be obtained. As a matter of interest, it is pointed out that continuous temperature measurements were taken from a total of 500 meters tape enabling a check to be made, day by day, week by week, of combustion behaviour, the piston ring sealing, etc.

It was found that the temperature distribution on a given level of the liner is not uniform, but has a " cam-shaped " profile, as the area of higher temperature shifts, in the course of time, from one position to another. It was noticed that the temperature of a given point of the liner may show a remarkable variation, that the stability periods may be followed by shorter or longer variations which, in opposition to the preceding situation, can be defined as of dynamic character. Probably, this behaviour may be ascribed to the piston rings which can rotate freely in their seats. It is thought that the hottest zone of the liner is that corresponding to the piston ring gaps.

Some recordings, from which can be seen the values and the variations of temperature in the upper part of the liner and on the piston crown are shown in Figs. 10, 11, 12, 13 and 14. It is worth repeating that the temperatures were measured at 1 mm. under the surface exposed to the flame and therefore represent a value very close to the actual temperature of the surface. In many other articles appearing in technical publications, similar temperature values have been reported, referring however to positions quite remote from the surface and, consequently, the comparisons cannot be considered so reliable.

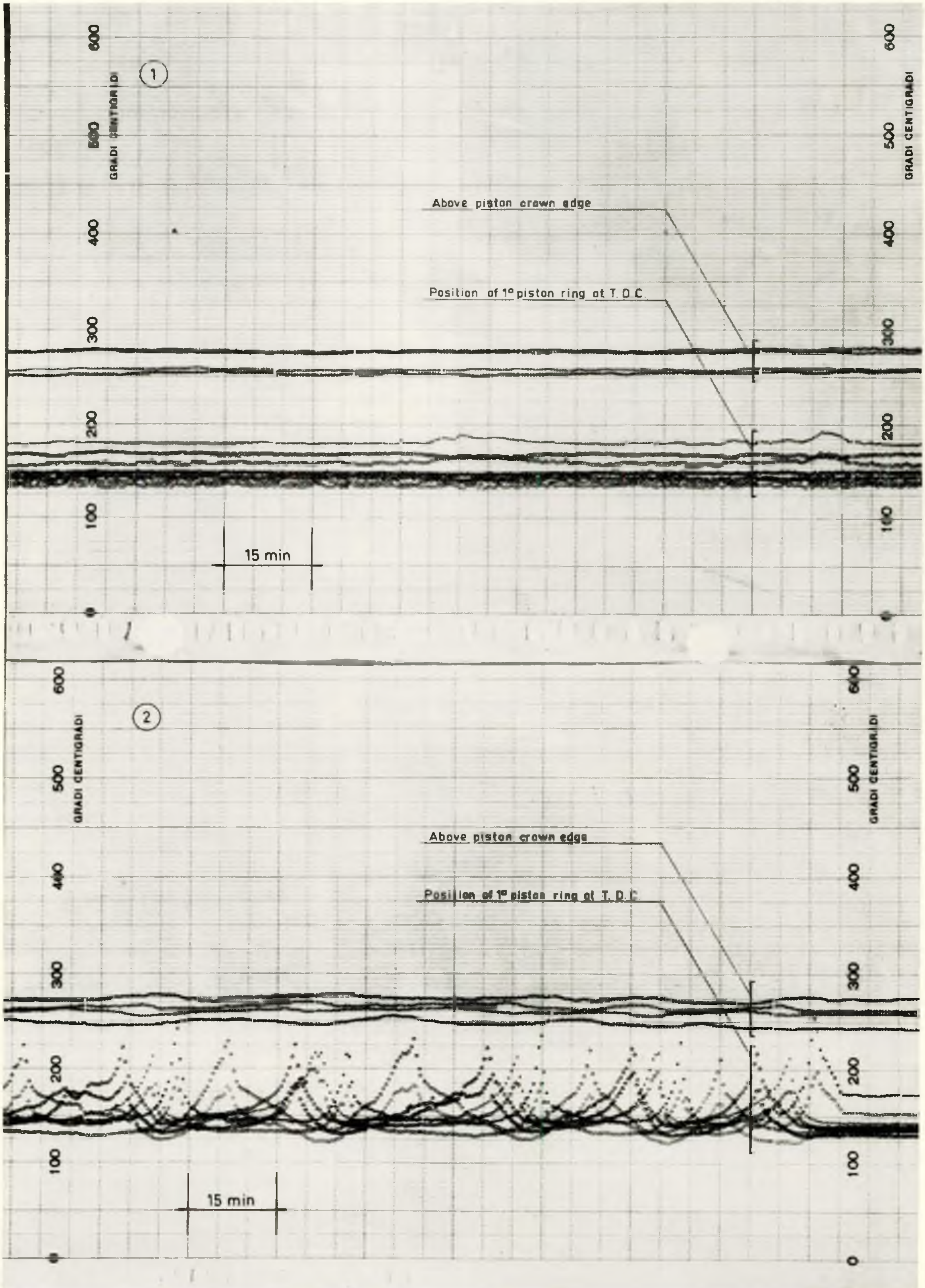
Many other engine parts have been kept under survey during the tests, such as, for instance, the main and big end bearings so as to check eventual variations of temperature.

This research confirmed that the original design of the engine was to be considered satisfactory inasmuch as, for high rating operation only, a slight reduction of the cylinder liner thickness, with increased water velocity in the cooling jacket, was found convenient. Also from this engine, it was found, like other engines, that the nodular cast iron piston rings, formerly used, do not give good results when subjected to high engine ratings.

Following the results of the foregoing tests, supported by the behaviour of the engines fitted in some ships and kept under survey, the author's company was able to decide, on the basis of sufficient knowledge, to raise the output of the production engines from 2,100 to 2,300 b.h.p./cylinder.

During the shop trials of the first 12-cylinder engine built, which was formerly supplied with a total output of 25,200 b.h.p. (2,100 b.h.p./cylinder) a maximum output of more than 32,000 b.h.p. was reached corresponding to about 2,700 b.h.p./ cylinder. As far as is known, this output is still the highest so far developed by a Diesel engine.

In the course of the tests, no weak points were found in the engine, neither in the normal nor overload conditions, as all the component parts involved proved satisfactory. The final results will be obtained from actual operation at sea.



Photographs from original records, representing two characteristic conditions:

1) Small fluctuation (probably corresponding to small movements of the first piston ring around a mean position)
2) Steady motion of the point of maximum temperature along the circumference (probably corresponding to a steady rotation of the first piston ring)

FIG. 10—*Fluctuations of recorded temperature on cylinder liner surface with steady load*

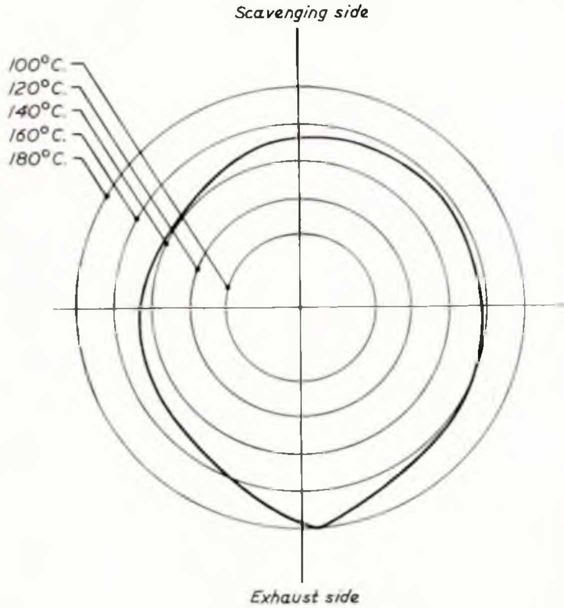
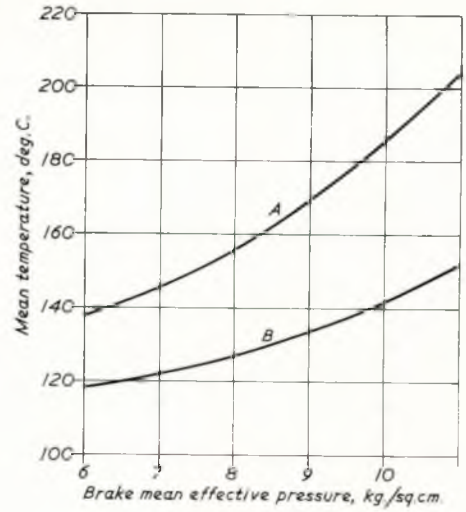
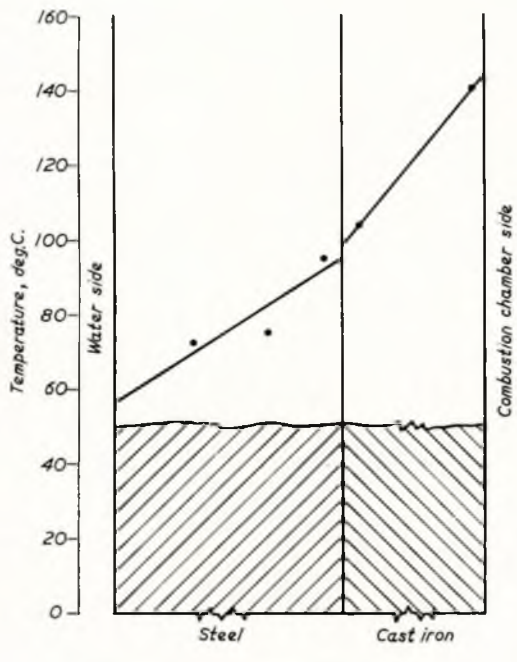


FIG. 11—*Recorded temperature of cylinder liner surface along a circumference. The curve refers to temperatures measured simultaneously. The point of maximum temperature corresponds probably to the gap of the first piston ring*



122 r.p.m.
Fuel: residual oil
A—At 1st piston ring position at t.d.c.
B—At 4th piston ring position at t.d.c.

FIG. 13—*Recorded mean temperature of cylinder liner surface versus mean effective pressure. Mean values on the cirumference obtained from diagrams as in Fig. 11*



122 r.p.m.—b.m.e.p.—8 kg/sq. cm.
Fuel: residual oil
Position of 2nd piston ring at t.d.c.—scavenging side

FIG. 12—*Recorded temperature of cylinder liner at different depths—The different temperature gradient in the steel and cast iron parts is due to the different thermal conductivity—The close contact between steel and cast iron is confirmed by the small temperature difference at the contact*

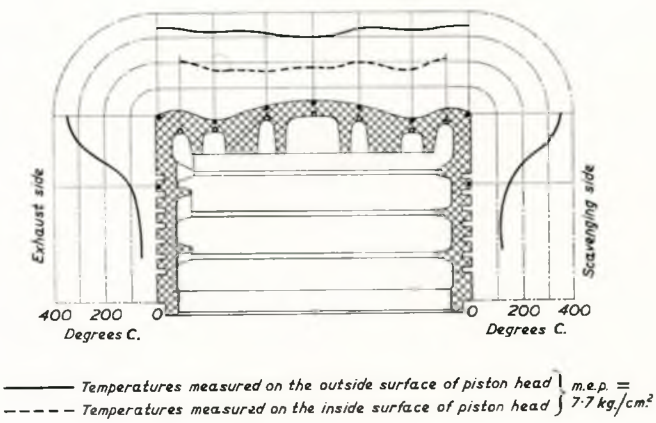


FIG. 14—*Temperatures measured in different points of the piston during operation at 7·7 m.e.p.*

SERVICE PERFORMANCE OF THE 900S ENGINE

The engines so far built by Fiat and their Italian licencees, have been installed in tankers or ore and coal carriers.

Up to 31st April 1964, fourteen ships were in service, of which the characteristics and main operating data are given in Table I.

The service, to which these ships are assigned, is perhaps the heaviest where continuous operation, very short stops and the necessity of having the engine always ready even during the stops, are concerned. The bulk carriers are generally employed on a triangular route from Italy to Liberia or Venezuela where they load iron ore for North America (one day for loading and one day for unloading); they are immediately re-loaded with coal (one day) to be carried to Italy (two days for unloading).

Two years from the date of the first engine entering service, a positive evaluation can be given of the behaviour of these engines, as the initial troubles which, for various reasons, occur

TABLE I—CHARACTERISTICS AND MAIN OPERATING DATA OF VESSELS WITH FIAT 900S ENGINES

Name of vessel	Engine		Ship		Speed on test mp.h.	Date of entering into operation	Total operating hours up to 31st May 1964
	Type	Builder	Type	D.w.t. capacity			
R. Cafiero	909S	Fiat	Tanker	38,000	18·59	22nd March 1962	16,000
Edera	908S	Ansaldo	Bulk carrier	35,000	17·65	22nd January 1962	14,500
Centauro	908S	Fiat	Bulk carrier	35,000	18·08	18th March 1962	12,500
Ma. Lolli Ghetti	908S	Fiat	Bulk carrier	35,000	17·79	6th September 1962	11,200
Gemini	908S	Ansaldo	Bulk carrier	35,000	18·13	7th September 1962	10,000
Mario "Z"	908S	C.R.D.A.	Bulk carrier	35,000	18·15	29th August 1962	11,000
Galassia	908S	Fiat	Bulk carrier	35,000	18·15	25th March 1963	7,200
Sagittarius	908S	C.R.D.A.	Bulk carrier	35,000	17·58	13th April 1963	6,500
Sandalion	908S	C.R.D.A.	Bulk carrier	35,000	18·10	4th September 1963	5,000
Leonardo da Vinci	909S	Fiat	Tanker	48,000	17·75	25th December 1963	3,000
Poseidon	908S	C.R.D.A.	Bulk carrier	35,000	18·18	18th March 1964	1,200
Agip Trieste	909S	C.R.D.A.	Tanker	48,200	17·60	1st April 1964	1,200
Fedor Poletaev	909S	Fiat	Tanker	48,000	17·50	10th April 1964	1,000
Carlo Cameli	9012S	Fiat	Tanker	91,600	17·95	3rd March 1964	—

in all newly-built machines and that are only pointed out by actual operation, have been overcome.

All the problems deriving from dynamic action inside the engine (vibration in the engine structure, torsional and axial vibration, hull vibration) were solved satisfactorily.

Beginning in 1961, a check was made of the crank arrangements of all engines by means of electronic computers, attaining results better than those previously found.

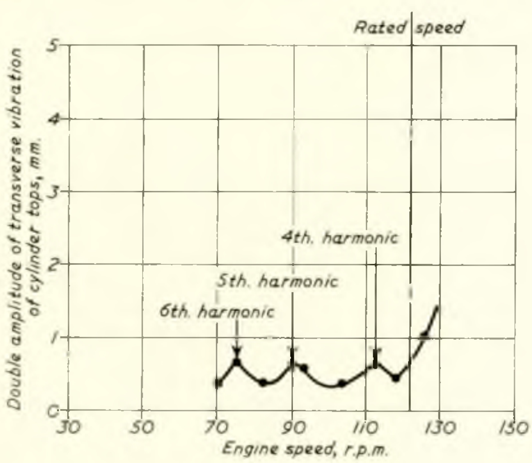
The actual recordings of engine vibrations made in the test shop and those concerning torsional and hull vibrations, carried out in the ships, comply with those expected and are shown in Figs. 15, 16 and 17.

It is worth reporting that further engine vibration recordings effected in the ships gave equal or perhaps better results than those obtained in the workshop, probably due to the improved stiffness of the ship foundations.

In accordance with the requirements of charter-parties and the owners, the engines have been employed at different outputs in the range 1,600-1,900 b.h.p./cylinder. They all have been operating on boiler oil with viscosity varying from 1,200 to 1,500 sec. Redwood.

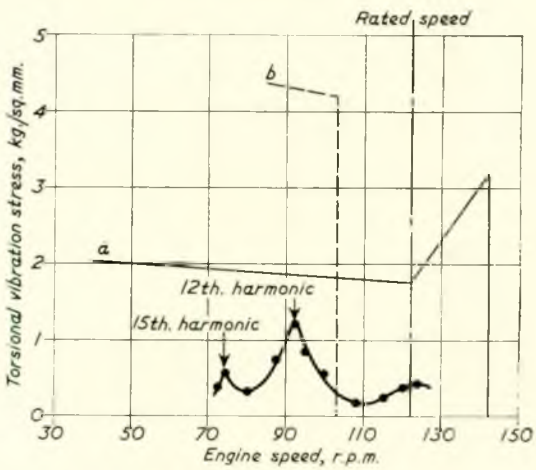
The ships never had to be stopped for engine faults; all maintenance was carried out during the normal stops. The actual heavy boiler oil consumption is evaluated at about 165 g./b.h.p.-hr., in close agreement to the consumption measured in the test shop.

The total lubricating oil consumption is 0·8 g./b.h.p.-hr., on average. As these engines are of a new type and engineers are not thoroughly conversant with them, this value includes a



Skew mode frequency = 450 cycles/min.

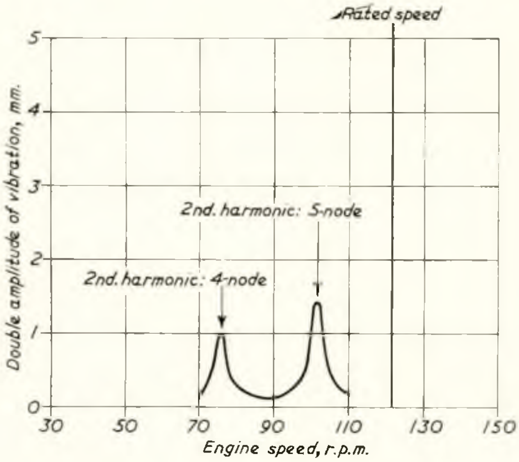
FIG. 15—*Engine 909S m.t.* Leonardo da Vinci—*Transverse engine vibrations on test bed*



a) Stress limit for continuous operation according to Lloyd's Register
b) Stress limit for transient operation according to Lloyd's Register

Two-node frequency = 1,104 cycles/min.

FIG. 16—*Fiat engine 909S m.t.* Leonardo da Vinci—*Recorded torsional stresses in crankshaft on board*



Four-node frequency = 152 cycles/min.
Five-node frequency = 203 cycles/min.

FIG. 17—*Fiat engine 909S m.t.* Leonardo da Vinci—*Vertical hull vibration, stern*

certain degree of higher than normal consumption, considered advisable by many engineers aboard the ships to assure the best engine operation. In the total value, the oil consumption for general lubrication and piston cooling is very low, about 0·1 g./b.h.p.-hr.

SERVICE PERFORMANCE OF MAIN COMPONENTS

During this operating period the following experience was gained with regard to the behaviour of the main components:

a) Based on earlier experience, the first engines built were fitted with nodular cast iron piston rings. This material was adopted for the type 750 engines over ten years ago, for all non-supercharged engines and for the first engines with medium supercharging and m.e.p. not higher than 6 kg./sq. cm., always with good results. The use of these piston rings allowed retaining the pistons in the cylinders for up to 6,000-8,000 hours, the rings and the liners being in good condition even after such a long period of operation. Beginning in 1962, on the two engines of the type 900S and on other type 750 engines, in which an increased load was required, both as mean effective pressure and piston speed, some strange irregularities were experienced, often localized in certain main cylinders in the form of abnormal wearing of the piston rings and scraping of the cylinder liner. At the same time a certain tendency to yielding of the material of the piston rings was noticed, especially in the proximity of the gap, with permanent deformation which might allow gas to escape. This abnormal behaviour was suddenly noticed for the first time on the engine type 908S fitted in the motor ship *Edera* and led to the replacement of two excessively worn liners. The replacement of the piston rings in this and other engines also equipped with them and whose behaviour was on the other hand fairly acceptable, put an end to this trouble. It was considered advisable to point out the foregoing due to its technical interest, though it is not to be ascribed to the engine design.

b) The use of grey cast iron piston rings already adopted for all engines, gave more satisfaction than expected. The quality of the material, the right proportioning of the rings with respect to the piston size and efficient cooling of the piston itself made it possible to get the life of the rings fitted in these engines equal to that required of the nodular cast iron rings. Several pistons have exceeded 6,000-7,000 operating hours without requiring dismantling and with the rings in good condition.

Wearing of the liners, when the engines operate at comparatively high loads (1,700-1,900 b.h.p./cylinder) resulted in about 0·1 mm. wear for 1,000 operating hours; it seems that a higher wear rate has been found when for any reason whatever, the ships had to run for a long period at reduced rating.

For this, it has been decided, whatever the load to which the engine is subjected, not to bring the cooling water temperature below 60 deg. C. (140 deg. F.).

c) The piston crown surface has always been found in very good condition without any signs of burning through overheating, thus confirming the test shop results, from which the piston cooling system had been considered efficient.

No carbon deposits were ever found inside the cooling channels, thus proving that the continuous and well distributed oil flow in the hot parts prevents the formation of deposits.

The systematic checks that the maintenance departments of the author's company carry out on the piston and cylinder heads, with the help of ultrasonic equipment, have indicated the beginning of failure in the upper peripheral cooling channel of the piston, in a region where the surface is rough due to casting, with discontinuity of material and probable shrinkage failure during cooling.

This trouble has been eliminated by machining this part of the piston so as to improve the material as far as fatigue resistance is concerned.

The breaking of some bolts connecting the external part of the piston with the inner bearing part, was considered to be due to excessive tightening at assembly. These bolts are subjected, in operation, to an exceptional increase of stress due to thermal deformation of the hot piston part and they must be only slightly tightened. Good results were obtained by systematically reducing the tightening torque of such bolts.

d) Two cases of connecting rod big end bearing failure were due to inaccurate finish of the corresponding crankpins. Further checks ascertained that the large dimensions and the weight of these crankshafts make it more difficult to obtain, using the same methods, the same level of accuracy obtainable in the smaller crankshafts. These troubles have quickly been put right by a better finish of the pins on site.

For new crankshafts, special specifications for finish and accurate checking to ensure that the crankpins are actually round with a perfectly straight generating line have been imposed on the crankshaft builders.

e) The small end bearings which, in all two-stroke engines, call for particular attention due to the difficulty involved in lubrication, showed behaviour considered to be satisfactory. As in the other engines, the pin surface has been induction-hardened and the finish of the surface is very accurate. This care, jointly with good oil circulation, assuring good flow for cooling and lubrication, resulted in good behaviour of these bearings, despite their large size.

In this connexion an interesting experience, may be quoted. Two crosshead pins of oversize diameter (480 mm. diameter instead of the normal pins of 420 mm. diameter) were tested on the experimental unit. Although the limited duration of the tests has not allowed conclusive results to be obtained, the behaviour of the smaller bearings was considered better than that of the larger, and all engines are actually fitted with the bearings of 420 mm. diameter, initially designed.

REFERENCE

1) DE PIERI, R. 1959. "Recent Developments in Italian Marine Diesel Engines." Trans.I.Mar.E., Vol. 71, p. 1.

Discussion

MR. J. CALDERWOOD, M.Sc. (Honorary Vice-President) congratulated Dr. Gregoretti on his presentation of the paper. It was a remarkable achievement, since English was not his first language.

He expressed the regret of all those present at the untimely death of the author of the paper, Dr. Roberto De Pieri. He had met him on perhaps half a dozen occasions and had a great respect for him.

The paper rather stressed the advantage of the open cylinder at the bottom. He could appreciate all the good points that were claimed for this but, surely, a considerable amount of the blow-by must get into the engine room, and with the open bottom cylinder the engine room conditions must be less pleasant for the staff than with the more usual design of most other engines. It would be interesting to know what was the experience in service.

The next points he had noted were in relation to the constant pressure system. It was mentioned that blade failures in the turbines of pulse system pressure charging were caused by the pulse. He did not think there was any real evidence for this and did not think that blade failures had been so common in the pulse system or, where they had occurred, that they had been in any way connected with the fact of the pulse. It had usually been a blade frequency which had corresponded with the number of nozzles or something else, but never with the actual pulse frequency which was far below any blade frequency.

It was also claimed that the constant pressure system gave the same overall efficiency due to the better use of the gas in the cylinder. It was true that there was a better use of the gas when it was in the cylinder with constant pressure, but something was lost elsewhere, and the figure quoted for consumption rather confirmed that, on the efficiency basis solely, the constant pressure system was rather less efficient. The figure of 0·36 was quoted, which was somewhat higher than most people were getting with a pulse system.

There appeared to be a misprint in the second line of the second column on page 28, where, talking about the constant pressure system, the author referred to an advanced exhaust. It was in fact surely the pulse system that had the advanced exhaust and not the constant pressure system. This may have been a mistranslation.

There was a reference under test and operating experience to 1,500 seconds fuel being the worst fuel obtainable in Italy. He wondered what was meant by "the worst fuel". Did the author mean that that was the highest viscosity fuel obtainable in Italy? It was his own experience that viscosity had very little relation to the performance of a heavy fuel in an engine; in fact, his own experience suggested that some of the comparatively light fuels, which were a mixture of distillate and heavy fuel, gave far more trouble than the very heavy fuels. He had known far more trouble with exhaust valves, on four-stroke engines, with the 200 seconds fuel, than he had ever known with the 3,000 seconds fuel. Some explanation of what was meant by "worst fuel" would therefore be of interest.

With regard to the operating experience, the author had stated that carbon had never been found in the oil passages in the piston. There were two points that he wished to make in this connexion. One was that the service of these ships was a "there and back" service with very little manœuvring and very little running except at full load. On oil cooled pistons his own experience had always suggested that the worst thing for the formation of carbon inside the pistons was frequent stopping and starting. What was Dr. Gregoretti's opinion on that? Would he have expected the conditions to be somewhat worse had these engines had a great deal of manœuvring to do?

One point of particular interest was the reference to the failure of nodular iron piston rings at these high ratings. His only experience of this was on a high-rated experimental engine; in this engine, behaviour of nodular iron rings was completely unpredictable. Ordinary grey iron was not necessarily more satisfactory, but it was more predictable, and nodular iron could be good in one piston and very bad in the next one.

With regard to the good performance of grey iron, was a very close eye kept on the micro-structure of the rings, because he had found that the ordinary grey iron piston rings supplied by most of the suppliers varied enormously from cast to cast, not from batch to batch. There could be one batch of rings delivered made obviously from several casts and some of them were excellent and some extremely bad.

Finally, he thanked Dr. Gregoretti for presenting this most valuable and interesting paper.

DR. ING. F. SCHMIDT said that those who had witnessed and observed the development of large Diesel engines over a period of more than 40 years could not but express their admiration and at the same time congratulate the Fiat Company on the successful development of their large Diesels.

The results presented proved that suitable design and proper choice of the compromise which each design necessarily involved could lead to a 100 per cent successful solution. They represented a strong argument against some publicity methods which claimed that certain design characteristics were absolutely superior and represented the only ones that were right. He also drew attention to the fact that this was already mentioned in the paper* on M.A.N. engines.

At the same time, however, allowance must also be made for the fact that transverse scavenging, as it had been retained by Fiat from the outset, was considered by many authorities as not being very suitable.

For supercharging their engines, Fiat had chosen series-operation with scavenge pumps attached to the crosshead slipper. This afforded the great advantage that the quantity of charge and scavenge air was not limited by the turbine output available. Part of the compression work could be assigned to the scavenge pumps, thus securing outstandingly large air quantities and low exhaust temperatures when compared with other engine designs. The more the degree of supercharging and the charge pressure were raised, the more the compression work of the scavenge pump increased—a fact which naturally affected fuel consumption rates. To be able to assess these factors it would be interesting to know whether there were any data on the percentage of scavenge pump work performed at different p_e and whether the fuel consumptions given in the paper and

* Sorensen, E. and Schmidt, F. 1964 "Recent Development of the M.A.N. Marine Diesel Engine". Trans.I.Mar.E., Vol. 76, p. 197.

in previous contributions had been referred to 10,000 WE/kg. At one point mention was made of the fact that in constant pressure operation the turbochargers were much more reliable than in pulse operation. His company had supplied many hundreds of engines turbocharged on the pulse system and approximately the same number of engines operating on the constant pressure system. They had never noticed anything to effect that service life and operational reliability of the turbine blading were less in pulse operation.

The remarks concerning the free piston undersides on page 27 prompted the question whether they actually remained entirely open during operation. In his experience the slight blow-by of sulphurous exhaust gases was very offensive to staff and necessitated the sealing-off of spaces bv covers. This, however, would imply that some of the arguments put forward lost much of their weight. Apart from this the efficiency of the stuffing box in the diaphragm (in both directions) could be checked quite easily if the respective control equipment were installed.

With regard to the temperature fluctuations on cylinder liners reported on page 29, in Augsburg such temperature measurements had been taken on a 780 mm. experimental cylinder as far back as 1955, placing 12 measuring points on the circumference. Severe periodic fluctuations occurred (periods of 20-60 min., and deviations of up to 100 deg. C. [212 deg. F.]). The assumption that such fluctuations were due to piston ring rotation or position of the ring gaps was fully confirmed. A piston withdrawn immediately after a particularly high temperature peak had occurred proved that the gaps of the two top piston rings had been *at this very spot*. The rotating movement of the top piston ring could be followed almost exactly by means of these 12 temperature measurements. It was interesting to note that the rings travelled to and fro up to approximately two-thirds of the circumference, but that they never executed a complete rotation. A paper* presented by Mr. Omotehara at the International Shipping Conference at Calcutta in February 1964, reported on temperature measurements taken on the piston of a UEC engine type 72/150. The continuous temperature measurements taken on the inside of the upper piston ring groove and on the land below (on the very outside) also showed temperature peaks of 60-90 deg. C. (140-194 deg. F.) with periods of approximately 60 min. The character of the temperature fluctuations at the two measuring points left no room for doubt that the passing of one or two ring gaps must be the cause.

In conclusion he wished to refer to the paper† of Mr. Schrakamp (C.I.M.A.C. Congress 1962) which reported very similar temperature fluctuations.

The semi-built crankshaft had a crankpin diameter of 700 mm. and a crank radius of 800 mm. If 705 mm. were assumed to be the smallest possible shrinking diameter, its distance to the crankpin was only 97·5 mm. Was not that somewhat risky?

In the shrinkage region the tangential stress was of approximately 24 kg./sq. mm. (15 tons./sq. in.) and this was a radial stress of 11-12 kg./sq. mm. (7·1-7·7 tons/sq. in.), and immediately above it, in the centre plane, there was already the fillet between crankpin and web where, as was well known, the highest alternating stress of the crankshaft arose. In his opinion the question as to the shortest distance permissible without producing a dangerous stress concentration presented an important problem which apparently had not yet been dealt with. The comments of Dr. Gregoretti would be interesting. He personally considered that a distance of 0·17 relative to the crankpin diameter was still reasonable, whereas the shaft just considered showed 0·14.

* Omotehara, I. 1964. "Development of the Mitsubishi UE Diesel Engines". Trans.I.Mar.E., Indian Division Supplement, No. 11, November, p. 51.

† Schrakamp, J. W. A. 1962. "Temperature Fluctuations of Cylinder Liner and Piston measured on the Main Engine of a Ship at Sea". Congrès International des Machines à Combustion (C.I.M.A.C.) 1962 Conference (Copenhagen), p. 1100.

Mr. P. Jackson (Member) expressed his regret at the death of the author, Dr. De Pieri, whom he had met at the various C.I.M.A.C. conferences.

The Fiat engine, in its principles and design, was almost entirely opposed to the engine he had sponsored during the past few years. It had loop scavenge, whereas he believed in through scavenge. He did not think that the loop scavenge engine could possibly have the same scavenging efficiency and the same clearing out of the cylinder as a through scavenge engine, and this accounted for the relatively low rating of the engine, as put forward. Even at 2,300 b.h.p./cylinder it only worked out at about 117 brake mean pressure, whereas his company were working some ten per cent higher than that. But he also noted that the engine was generally offered and installed at even lower ratings.

What was the actual rating? This was a very vexed question to both builders and users. His company's rating was the service rating, and ten per cent over that was given for sea trials and 15 per cent for one hour on shop test. These were definite statements, and so he wished to ask what Dr. Gregoretti's ratings really meant.

A loop scavenge engine required a separate scavenge pump or some assistance to the turboblower. All loop scavenge engines had some kind of assistance. Two Continental engines had under-piston assistance, but the Fiat engine had a separate scavenge pump, which was a necessity, as Dr. Gregoretti had said, because of the open-ended cylinder. Again, so far as he was aware, all constant pressure systems required some assistance to the turboblower, whether they were through scavenge or otherwise, so for two reasons the Fiat engine required a scavenge pump. This being a complication, one looked to find the advantages which Dr. Gregoretti had claimed.

The separate scavenge pump must have an adverse effect on fuel consumption. The fuel consumption could not possibly be as good as a through scavenge engine with impulse charging; nevertheless the figure given of 165 g. for 1,500 seconds fuel was quite good.

With regard to the question of using more of the cylinder by the greater use of the expansion stroke, he could see how that was achieved, but felt it was at the expense of the exhaust lead relative to the opening of the scavenge ports, and for that reason one required the valves controlling the scavenge ports. The type of flap valve was shown in Fig. 1. He (Mr. Jackson) had tried valves for this same purpose, because on turbocharged engines of practically all makes, the expansion ratio was less than that of the compression ratio. That was a disadvantage of the turbocharged engine. If only this could be remedied there were possibilities of some eight per cent improvement in fuel consumptions. But his experience, with valves controlling the scavenge ports, was that they became choked and clogged up with the sludge which was one of the products of combustion of boiler fuels, as the paper admitted. However, he had not used flap valves of the type shown in the paper. He had used assemblies of leaf springs.

Great stress was laid, in the paper, on the open-ended cylinder. This construction had been employed by Sulzer, B. and W., and others, in pre-war days. However, it had advantages, as was pointed out on page 27, though it resulted in a heavier and more costly engine and in an increase in engine height of some 4ft., which was quite considerable.

He supported Mr. Calderwood's remarks that the space underneath the piston became very messy, due to the sludge coming down past the piston and a certain amount of lubricating oil coming up. This was not due to any pressure difference. It happened as a result of defective or inefficient scraping arrangements in the diaphragm gland, but it was not regular; it was not even the same from cylinder to cylinder. It occured sometimes without any apparent explanation. The space would suffer from being messy with sludge coming down and oil coming up, and also any blow-by could get into the engine room, as Dr. Schmidt had pointed out.

He agreed that scavenge fires would be eliminated and the possibility of a crankcase explosion would be minimized; the valves controlling the scavenge ports would also tend to eliminate scavenge fires.

The piston was quite a heavy one. From Fig. 1 it looked as though the piston skirt did not cover the scavenge ports at the top of the stroke, in which case there would be blow-back of scavenge air into the cylinder and out into the atmosphere. This surely could not be so, but it did look like it from Fig. 1. This heavy piston would lead to big out-of-balance couples which could cause vibration. There had been considerable vibration difficulties on two-cycle single piston engines having six cylinders, though apparently Fiat had not built engines as low as six cylinders. His company had tried to keep the out-of-balance down to 200ft. tons, say 70 metre tons. In a paper* to the Institute some two years ago, cases were mentioned of vibration in ships caused by couples as low as 110ft. tons, which would be about 35 metre tons.

The cylinder liner construction was very interesting. He had questioned this with Dr. Gregoretti in Copenhagen two years previously. Mr. Jackson had never succeeded in putting an insert into a liner and getting satisfactory operation. One could hone the liner plus the insert together, but after even a few hours of operation at full load a step could be felt and heavy wear of the piston rings then took place. He did not understand how Fiat got over this, and he would have expected it to be particularly difficult on a liner where hot exhaust gases were taken out at one side and cold air was admitted at the other.

The lubricating oil consumption was very high, 2½ times what was claimed for other engines, and in that connexion he wished to ask Dr. Gregoretti about the lubricating points. He recalled that at Copenhagen Dr. Gregoretti had said he had lubricating points both at the top of the stroke on to the piston rings and at the bottom of the stroke, and this doubling up of the number of quills might be the reason for the high consumption.

The breaking of the studs between the piston head and the piston rod was something he also had experienced and he agreed with Dr. Gregoretti that it was caused by distortion and expansion of the piston head. To try to remedy this he had reduced the diameter of the bolt circle, but there was no advantage. He had also increased the length of the piston studs. He did not believe the breakages were due to over-tightening; he had in fact formed the opposite conclusion.

He was interested in what was said about hardening the crosshead pins. His company had tried chromium plating, but their experience with induction hardening with a 0·5 carbon steel had been disastrous. The surfaces of the pins had longitudinal cracks after induction hardening. He was interested to know what kind of steel had been used for induction hardening.

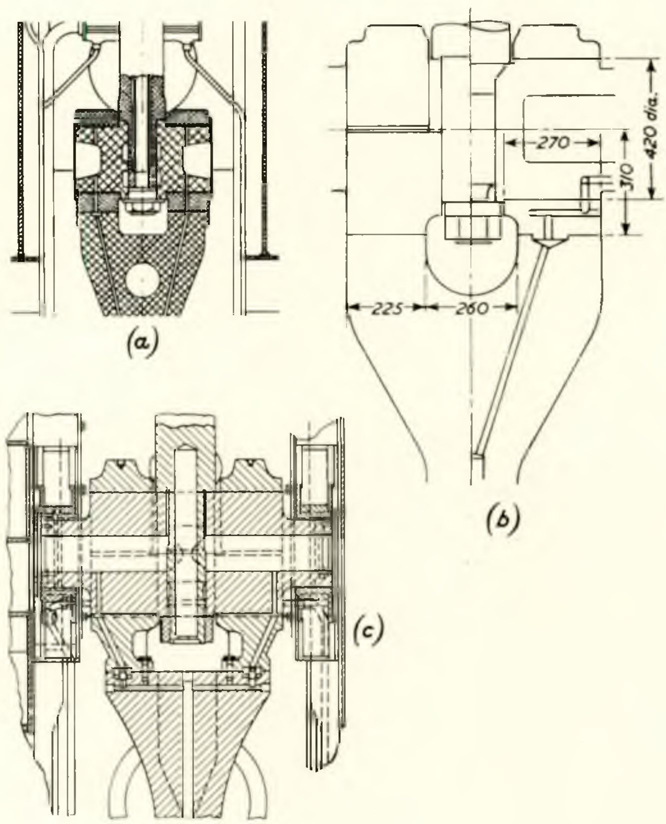
He supported the statement that small pins, provided they were not too heavily loaded, ran quite as well as big pins. He had made this same experiment, because the pins of his company's engines were considered small, and so, like Dr. Gregoretti, he had tried some larger pins, without any benefit.

In conclusion, he wished to ask for further explanation of the paragraph on page 32 "Beginning in 1961 a check was made of the crank arrangements of all engines by means of electronic computers, attaining results better than those previously found."

Mr. A. R. Hinson (Associate Member) said that on looking through the records of Lloyd's Register of Shipping he had formed the opinion that whereas the Fiat engine compared favourably with other engines as far as the performance of main bearing, bottom ends and running gear generally was concerned, there seemed to be room for improvement in the top ends.

He noticed in the last paragraph of the paper that Fiat had fitted, as an experiment, two crosshead pins of oversize diameter and had concluded that the smaller bearings were better than the larger. All engines were now fitted with bearings of 420 mm. diameter as initially designed. This he found interesting, since, for engines having cylinder bores between 750 mm. and 900 mm., 420 mm. seemed to be about the diameter chosen for crosshead pins. The diameter had been increased from 390 to 420 mm. on the Mitsubishi Nagasaki Diesel UE type engine and this increase no doubt contributed to the elimination of crosshead failures. Did the results of the Fiat experiments show that the better results with the 420 mm. diameter pins were due to the lower maximum rubbing speeds, or were there other factors?

Considering stiffness in top end bearing assemblies, he noticed that both the Fiat and Mitsubishi crosshead pins had relatively large recesses and both employed a forked connecting rod, as shown in Fig. 18.



a) Fiat
b) Mitsubishi
c) Sulzer

Fig. 18

The Sulzer engine employed a conically ended connecting rod to which the bearings were bolted and, according to advertisements in the technical press, this construction was beneficial from the standpoint of deflexion.

Mr. Hinson thought that the forces should be transferred from the piston rod to the connecting rod as directly as possible and that the pad fitted under the piston rod in the Doxford J engine did much to achieve this. The old LB Doxford had a relatively low incidence of top end failure, probably partly owing to the flexibility conferred by the spherical piston rod washer and spherical bottom end bearings which, however, were no longer required on the J engine.

Had the author's staff performed any experiments to determine the stresses and deflexions in their crosshead assembly? Details would be appreciated.

* Johnson, A. J. and McClimont, W. 1963. "Machinery Induced Vibrations". Trans.I.Mar.E., Vol. 75, p. 121.

Mr. S. G. Christensen (Member of Council) said that it was known that the maintenance work required to keep a Diesel motor in a good efficient working condition might be grouped under two headings:

a) Maintenance work to maintain proper combustion
b) Maintenance work to keep bearings in a good state of alignment and correct adjustment.

It was also known that the relative height of the scavenge and exhaust ports had a considerable bearing on the rate at which fouling occurred generally being much worse in the scavenge ports where the scavenge ports were higher than the exhaust ports.

Mention was made of the simplicity of the cross-scavenge engine and there being no need for exhaust valves. Was this engine completely valveless in respect of the scavenge and exhaust systems, or were scavenge valves fitted? It would appear from Fig. 1 that some form of non-return type scavenge valve was fitted in the scavenge trunk. If this were so, then it might be presumed that the height of the scavenge port was greater than the height of the exhaust port. With this in view could Dr. Gregoretti give some idea of the length of time, in terms of thousands of hours, that the engine was able to run, for example, say on 900 seconds fuel, before scavenge ports must be cleaned? Together with this information it would be interesting to hear also of the worst and best cases for the time periods between cleaning of ports.

As reciprocating scavenge pumps were also part of this engine it would be interesting if a set of curves could be given showing the "fall off" of mechanical efficiency which occurred as port fouling advanced and scavenge air pressure rose.

In view of the resistance offered by another major engine builder of a similar type of engine to fitting oil gutterways in cylinder liners, it was interesting to note that gutterways were fitted in the Fiat engine cylinder liner. With this observation in view, what recommendation did Dr. Gregoretti make regarding the viscosity of cylinder oils with reference to their SAE number?

Referring to cylinder unit overhaul, cleaning of piston ring grooves, etc., was the piston lowered into the crankcase for this operation, or lifted out of the cylinder as was usual? His reason for asking this question was due to what appeared, in Fig. 2, to be a collar on the lower end of the piston rod, where it entered the crosshead block. Whichever method was adopted it was obvious that the diaphragm and piston rod packing would have to be dismantled.

Dealing with matter which came under the heading "Maintenance work on bearings", the construction of the crosshead, little end, or top end bearing was of great interest, as experience went to show that this was the most vulnerable bearing of any large or highly rated Diesel motor. From details shown in Fig. 2, the size of lubricating oil lines leading to the main bearings indicated that the crankshaft was not drilled for oil passageways, and that the top end lubrication was an extension from the piston cooling supply, with a further extension for the lubrication of the bottom end via the connecting rod. If this were so, what provision was made for preventing damage to the bottom end bearing and crankpin in the event of breakage of white metal in the top end bearing, whereby the flow of lubricant to the bottom end bearing was cut off?

Perhaps Dr. Gregoretti could give the main features of the top end design shown, and explain the arrangement of oil passages in the crosshead pin and bearings, also their main function as to whether they were for lubrication or cooling purposes.

Reference was made to that part of the paper which dealt with crosshead pins and stated that the bearing surface of the crosshead pin was induction hardened. As deflexions obviously occurred in a crosshead pin in service, and hardening of the surface of a material produced a change in state of the material, where hardening had taken place, so that a boundary existed where the change of state occurred, could Dr. Gregoretti say if any case of exfoliation of the hardened surface material from the parent had occurred? What was the specification for the crosshead pin material, and what was the depth of the hardened surface?

As the wear rate of the bearing surface of a crosshead pin had a considerable effect on the life that might be expected from the crosshead bearing, it would be interesting if a figure could be given for the wear rate of the crosshead pin bearing surface. Again the wear rate on the crosshead pin was influenced by the circumferential angle of the bedding area of the bearing surface. What was this circumferential distance in terms of degrees? A common figure was 120 degrees, although some engine builders had considerab y reduced this figure with the possible result that there had been many unpublicized failures of top end pins and bearings.

In view of the influence of copper in tin antimony bearing alloys and its use to prevent segregation and improve hardness, did Dr. Gregoretti recommend copper as a constituent, especially in view of the strong possibility that copper lowered the resistance of white metal bearing alloys to fatigue? Perhaps Dr. Gregoretti could give an approximate analysis of the white metal used in the top end bearings of the Fiat engine.

Referring to other engines, another engine builder had shown in a survey of running costs for large Diesel engines, an allowance for top end pin and bearing renewal or reconditioning every 24,000 hours, with bearing examination every 3,000 hours. Could a longer period of useful life be expected with the Fiat design of top end bearing, and could the examination periods for other bearings be extended beyond 3,000 hours?

Reference was made in the paper, dealing with this item, to the fact that an oversize diameter top end pin did not give as satisfactory service as one of standard size when on test. Was this really surprising when one considered that the design factors for top end bearing dimensions must be influenced by the product of the rubbing speed of the bearing surfaces and the pressure on the surface under consideration. It was a pity that the connecting rod mechanism was such that maximum rubbing speed occurred at the time when the bearing loads were a maximum.

In view of the bottom end bearing failures that were mentioned, what surface finish was recommended for top end and bottom end pins in terms of microns or micro-inches, and what process did the Fiat motor company use to obtain this finish?

Anyone who had had to machine a thrust block collar with the crankshaft *in situ* in the engine would welcome the design of the thrust block shown in Fig. 2, whereby the thrust collar was easily removable from the engine. Were there any particular design features which had been adopted in this item to prevent cracking occurring at the coupling flange fillets or between the coupling bolt holes.

Referring to liner wear rates it was interesting to see that a minimum figure of 60 deg. C. (140 deg. F.) was given for the cooling water temperature outlets, as one engine builder gave a figure a little above this as the maximum. His experience went to show that a figure of the order of 71 deg. C. (160 deg. F.) was even more to be desired, and that the cylinder liners should be heated to the normal cooling water inlet temperature before a main engine was started.

Mr. S. Jambunathan (Associate Member) said that he had worked in a Fiat-engined ship six years previously. With regard to the open cylinder, he had always found that every time the engine was stopped (after each voyage) it required cleaning. On at least two occasions he had observed a spark coming out of the open end. The additive oil used for cylinder lubrication often used to separate because of the intense heat around the lubricators, which were located near the open ends of the cylinders. It was a problem to keep the oil going up.

With regard to the use of lubricating oil for piston cooling, he thought that this was not justified. Of late, pistons were being water-cooled. He had worked for about 15 months on a 750 Fiat engine; every third day it was necessary to clean the lubricating oil filters, otherwise there was as much as a 10lb. pressure drop. In other engines one never bothered about cleaning lubricating oil filters, especially if they had water-cooled pistons with external glands. Doxfords had introduced the external glands, so that there was absolutely no possibility of corrosion at all, because the water could not leak inside the crankcase if it were used as a cooling medium.

MR. J. F. ALCOCK said that of the many points of interest in this paper, he was particularly concerned in two. The first was the question of the possibility of scavenge fires, mentioned at the bottom of page 27. One could see that fires in the under-piston space could be prevented, since it was accessible for sludge cleaning, but what about fires in the scavenge belt itself? He had recently witnessed a fire in another make of engine in which the scavenge belt was connected to the under piston space. It had an observation window which gave one a view of the floor of that space, but one could not see the scavenge ports above. Through that window a large number of "sparks" could be seen, probably lumps of carbon about 1 to 2 mm. in diameter, being blown up and down by the piston displacement; there was very little sludge on the floor of the under-piston space and it did not ignite. He had watched it for some considerable time. On the other hand, somewhere up round the scavenge ports there was the source of this plentiful supply of sparks which he had watched for seven or eight minutes, and on the outside of the belt, at about port level, there was a patch of scorched paint. He saw similar scorched patches which had resulted from earlier fires from other cylinders. He was pretty sure that the heat contained in the sparks which he saw flying about could not have heated the pretty thick metal of the scavenge belt to anything like scorching point, and it seemed certain that up in the scavenge ports there was a "cosy corner" where an accumulation of sludge had lodged and was burning away merrily. This showed that even where there could not be a fire under the piston there could still be a scavenge fire.

The paper mentioned "suitable discharge arrangements" for sludge. It was not clear from Fig. 1 where that sludge went. He noticed there was a sloping floor, but was not clear whether the sludge was fluid enough to flow down that floor and into a drain or whether it had to be scraped off periodically.

He wished to refer next to the local temperature fluctuations, as shown in the bottom diagram in Fig. 10. These were very interesting and it was surprising to see that large magnitude, something like 100 deg. C. (180 deg. F.), but they were rather puzzling in some ways. If they were only due to the ring gap and if the ring rotation rate were more or less uniform one would expect short sharp peaks separated by long periods of roughly uniform temperature. Some of the traces were like that, but quite a number had a gradual rise and fall which was rather difficult to explain. One would not have expected the local effect of the ring gap to spread very far round the bore.

There was one possible explanation which had occurred to him, though he was not at all sure of it, and it was simply put up in order to be shot at. The rings travelled between the hot top end of the bore and cooler parts down below and they must pick up heat from the hot zone and reject it to the cooler zones by a process which might be called "mechanical convection". An analogy was the ordinary domestic flat iron, alternately heated on a hot plate and applied to cool fabric.

In some work by Hug at the E.T.H., Zurich, many years ago (about 1930) he measured temperature fluctuations in a gas ring and found a swing of somewhere about 20 deg. C. (36 deg. F.).

Making a rough "guestimate" of what that meant it looked as if something like a quarter of the heat blowing out through the liner at the top ring level might be taken by the rings down to the lower parts and spread by that method. With that form of heat transfer the main resistance would probably be in the oil film somewhere down the cylinder where the pressures were lower and it was cooler, so that the viscosity was greater. The thermal resistance there should therefore vary with the spring pressure, which in this region would be the main pressure and would be expected to affect the temperature. If that varied gradually round the circumference one would expect the liner temperature to do the same, which was rather what was in some of these diagrams in Fig. 10.

Some support for this argument was afforded by the following facts. Again taking a close look at the second graph of Fig. 10, it could be seen that there was a gradual slope and then a short sharp jump at the top that was probably due to the gap passing over the thermocouple, but the gradual rise and fall still had to be explained.

This "mechanical convection" would depend on the conductivity of the ring material, for if that was low then in a given time less heat would be sent into the surface of the ring, and therefore less would be carried down. A nodular iron had a lower conductivity than grey iron, something like one-third less as a rule, so that a nodular ring should carry away less heat than a grey iron ring, and this might explain the trouble found with nodular rings at high load.

Finally, with regard to the piston temperatures shown in Fig. 14, they were very satisfactorily low figures and remarkably uniform. One piece of useful information would be the conductivity of the metal. He was not certain whether it was cast iron or steel. It would also be useful to have the approximate temperature of the cooling oil; the exact figure did not matter, since the cooled surface was at about 200 deg. C. (360 deg. F.) and a few degrees either way did not make much difference. The oil velocity through these passages would also be of interest.

MR. A. A. J. COUCHMAN, B.Sc. (Associate Member) said that he was rather surprised at the ease with which Dr. Gregoretti dismissed his vibration problems, and in this context he had two particular questions to put to him.

With 12-cylinder engines one would expect the 0-node mode of axial vibration, which was the lowest mode, to be in the region of 500 or 600, being related mainly to the crankshaft. For an engine of this type running at about 115 to 120 shaft r.p.m. one would expect the crankshaft to be excited by propeller-exciting forces from perhaps four, five and six-bladed propellers. The shafting length had very little effect on this mode of vibration.

In eight-cylinder engines again the 0-node mode of axial vibration was likely to be in the region of 1,100 cycles/min., and this could be excited by engine forces. Had Dr. Gregoretti encountered these troubles or any associated torsional and axial vibration problems?

Turning to transverse engine vibration, in the case of parallel modes, the seating in the ship or the stiffness of the seating came very much into the problem of calculating resonant condition, but where the skew mode and 12-cylinder engines were concerned it was very much the stiffness of the engine entablature and the orientation of the forces relative to one another which really mattered, and he therefore suggested that the reason why similar frequency values were being obtained in the workshop and in the ship was not because of any difference in the stiffnesses of the foundation of the shop and in the ship, but because the actual engine characteristics remained constant.

Finally, had Dr. Gregoretti ever encountered a fore and aft rocking motion in 12-cylinder engines? If so, what remedial measures had he been able to use.

MR. C. C. J. FRENCH (Member) said that, like his colleague Mr. Alcock and other contributors, he was interested in the temperatures quoted in Figs. 10-12. These were the temperatures in the tests on the two-cylinder engines. The temperature swing which was attributed to ring rotation would appear from Fig. 10 to be somewhere about 100 deg. C. (180 deg. F.), but in Fig. 11 there were some temperatures measured round the circumference in which the difference was only about 40 deg. C. (72 deg. F.) Were these two readings at different load conditions or taken at different points along the length of the liner?

Further to this, in the discussion of service performance, there were some interesting details as to heavy liner wear and possibly increasing gas blow with nodular iron piston rings.

Were these high temperature swings on the two-cylinder engines obtained with nodular iron rings, and, if so, could this be a contributory factor to the magnitude of the temperature swings?

With regard to the cast iron insert to the liner and the temperatures given in Fig. 12, there was a difference between the gradients of the two lines of almost 2 to 1 as drawn. Assuming that the steel had a conductivity of some 0·12 c.g.s. units (the normal conductivity) this meant there was a very low conductivity for the cast iron if the gradients were really of this order. Was the cast iron of an austenitic or nodular type which would have as low a value as this? This was really the only way of checking on whether it was appropriate to draw the lines with these slopes. Would Dr. Gregoretti say what was the thickness of the steel and give the coolant temperature, in order that the heat flows and water side conductance could be calculated?

Reply to Discussion

by DOTT. ING. A. GREGORETTI

DR. GREGORETTI in reply to the discussion, expressed his appreciation of the kind words of sympathy concerning the death of the principal Director, Dr. De Pieri.

He was grateful, too, for the large number of questions, showing the interest in the paper.

With regard to Mr. Calderwood's questions, he said that Mr. Calderwood had mentioned the blow-by or gas escaping from the lower part of the liner and going into the engine room. This was the first time that Dr. Gregoretti had heard any complaint about this arrangement. He had personally only observed such a phenomenon in cases where there was something wrong in the liner, such as scraping or broken piston rings, but normally there was no gas blow-by through the lower part of the liners. For preventing this blow-by, two piston rings were installed in the lower part of the piston skirt.

On the other hand, gas escapes were minimized owing to the low exhaust gas pressure existing in the constant pressure operation of the supercharging turboblowers.

As far as the behaviour of the turboblower blades was concerned, he had had no experience of inconvenience in the pulse-system operation; however, as the pulse-system operation required larger gas passage sections in the turbine and consequently higher blades, he considered that a greater danger might be represented by the fatigue failures due to the lower vibration frequencies of the blades.

With regard to fuel consumption, the figure given in the paper related to the service fuel consumption, but he agreed that this might vary according to the quality of fuel, not only in regard to viscosity but also ash content, ash composition and so on.

The 1,500 seconds Redwood fuel used in these tests, was the cheapest one available at the moment on the market. He agreed with Mr. Calderwood that greater difficulties were likely to occur by employing more fluid fuel oils especially when these were obtained by mixing a distillate with heavy fuels of the worst quality.

As concerned the formation of carbon deposits in the piston, he agreed that this phenomenon might occur during manœuvring if the oil pump was directly driven by the engine. However, in the installations with large Fiat marine engines, the oil pumps were almost always independently driven by means of electric motors. When the oil pumps were directly driven, there were standby electrically-operated pumps.

If one was careful to keep the electrically-operated pumps in operation during manœuvring and for a certain time after the engine had been stopped, there was no possibility of carbon deposit formation.

He agreed with Mr. Calderwood that the behaviour of nodular cast iron piston rings was not constant and that there were some surprises.

This uncertain behaviour was probably due to the fact that the nodular cast iron might undergo remarkable structural transformations or hardening, consequently to accidental overheating which, for different reasons, might sometimes occur during engine operation. The grey cast iron rings were not affected by these phenomena and this was the reason for their better behaviour, especially at the highest loads.

Dr. Schmidt's contribution was very extensive and interesting.

The work absorbed by the scavenge air pump increased with mean effective pressure; however, due to the fact that, simultaneously, the work of the exhaust gas turboblower increased, the final result was a decrease of the work of the air pump, measured as percentage of the main piston work.

The fuel consumption mentioned in the paper was that actually measured during operation and no correction had been made to take into account the heat value of the fuel employed.

Dr. Schmidt had never met with any inconvenience in the turboblowers installed on engines operating on the pulse-system.

On this matter and concerning the eventual escape of gas from the cylinder liner lower part, Dr. Gregoretti referred to the reply already given to Mr. Calderwood.

With regard to the fluctuation of the temperatures of the cylinder liners, he agreed with Dr. Schmidt that possibly this was due to the movement of the ring, and that the high temperatures corresponded to the moment when the piston ring gap was passing the thermocouple. However, he had some doubts about this, because the behaviour of various types of engines varied very much, so that this problem was not completely clear to him.

The dimensioning of the 900 type engine crankshaft was very similar to that of the 750 type engine which had totalled some ten thousand operating hours without any inconvenience. In particular the distance between the main journals and the crankpins corresponded to 13·8 per cent of the pin diameter.

On the other hand, there were some 900 type engines, the crankshafts of which had already exceeded 100 million stress cycles with no inconvenience. This proved that the stresses in the shrinking zone between the main journal and the web did not reach dangerous values.

Dealing with Mr. Jackson's observations, he agreed that with a uniflow scavenge engine it was possible, for the same dimension of cylinder (same diameter and same stroke), to have some more power. Nevertheless, there were other aspects of the uniflow scavenge engines to be considered—for instance—the behaviour of the exhaust valves, when really very bad fuels with high viscosity and high ash content and vanadium content were used. His company had, in service, many four-stroke engines running on residual fuels, and the only serious trouble was from the bad behaviour of the valves, especially at high loads. The life of the valves could be improved with the use of special materials, but after some thousand hours of running the valves needed overhauling. When one extrapolated this experience to larger valves, no doubt the problem could be more difficult in large two-stroke engines.

With regard to rating, the actual rating was 2,300 h.p./cylinder for the 900 type engine. An overload of ten per cent was admitted.

The actual operating output was essentially a matter of policy for the shipowner. In fact, if the engine was continually operated at maximum output, the maintenance cost increased inevitably, for instance due to heavier wear of the cylinder liners and of the other parts subjected to wear, increased lubricating oil consumption, etc. Consequently, the shipowner must prepare an accurate budget of the expected operating cost of the engine at full output or at a lower output, for instance 90 per cent, and of the freight income with respect to the ship's speed.

He had many statistics on many marine engines, not only those of his own firm, but of other types as well, and never had he seen engines operated at full power. This was in his opinion a good policy and one that the owner must not forget.

He agreed with Mr. Jackson that, when the engine was new, the fuel consumption was a little worse when one had scavenge piston pumps, in comparison with the engine, for instance, the uniflow scavenge engine, which did not have pumps; but this difference was very small, and the figure in actual operation for both types of engine would be practically the same.

This better service operation of the engine equipped with scavenge pumps was due to the fact that, especially during manœuvring and in navigation at reduced load or when, for any reason whatever, the thermal operating conditions of the engine were no longer satisfactory, a great quantity of air was always available for good combustion in the main cylinder.

The behaviour of the scavenge valves at the inlet ducts to the cylinder had always proved satisfactory. They did not become excessively dirty and their maintenance did not require particular work, also by virtue of the fact that they had been designed to be dismounted and remounted for cleaning with minimum work.

The scavenge valves were arranged in a lower position with respect to that of the air inlet ports in the cylinder and consequently the piston skirt closed the ports completely even at top dead centre; Fig. 1, showing the valves in offset section, could give rise to misinterpretation of this part of the drawing.

Concerning the breakage of bolts connecting the outer part to the inner part of the pistons, this had been found to be due actually to an over-tightening of the bolts. A long series of experiments was made, heating the piston, circulating the oil inside and installing many strain gauges to measure the stresses, and this was the conclusion. In the hardening of the crossheads he agreed with Mr. Jackson that it was difficult to harden the pins, and this was not done until there was an induction hardening system available, because there had always been difficulties with flame hardening. After the hardening, the pins were stress-annealed at 280 deg. C. (536 deg. F.). The steel used for the induction hardened pins had a 0·4 carbon content (C=0·4; Mn=0·8; Si=0·35).

As regards the unbalanced secondary actions due to the piston masses, he considered that the engine with heavier pistons was not in a worse position with respect to that with lighter pistons. In fact, it being very difficult to get complete balancing of the forces, and supposing that the hull vibration frequencies unfortunately coincided with the double engine speed, namely there was a resonance phenomenon, the amplitudes could reach considerable values, even with small forces or exciting moments. More than a question of free actions intensity, the matter was one of avoiding resonance phenomena, through appropriate dimensioning of the hull structures.

As far as the construction of the cylinder liner in two parts was concerned, no difficulties were experienced, deriving from the formation of a step corresponding to the connexion zone of the two parts. To avoid the eventual formation of this step, the cylinder liners were ground with a slightly tapered surface in the coupling area.

Experiments were currently being conducted to reduce the number of the lubricating quills in the cylinder liner and from the tests carried out to date, this seemed a possible solution which could lead to a further reduction of the specific lubricating oil consumption with respect to the 0·5g./b.h.p.-hr. so far obtained.

As concerned the dimensions of the crosshead pins, experiments conducted on other engine types had also confirmed that variations of diameter did not greatly affect the behaviour of the bearing. Instead, it was much more important that the assembly formed by the upper part of the connecting rod, bearings and pins be dimensioned so as to minimize the load concentration on the bearings.

Moreover, it was necessary that the lubricating quills be placed in such a way as to ensure a positive oil replacement in the contacting surfaces and, above all, that the pin surface be finished with accuracy.

Turning to Mr. Hinson's contribution, Dr. Gregoretti agreed with him that when the holes in the crosshead pins were too large in diameter, some deformation could occur, and that when this happened the distribution of the pressure above the white metal became worse and could produce some breakages in the white metal. His company had experienced such troubles on one engine of medium size and this inconvenience had been eliminated by putting a steel cylinder into the hole, to avoid deformation of the pins.

The remark by Mr. Hinson that only slight differences existed between the dimensions of the 750 type engine connecting rod—as shown in the drawings in Dr. De Pieri's paper[1] of 1959—and that of the 900 type engine, was not correct. In fact, in the 900 type engine, as in the most recent 750 type engines, the connecting rod as well as the lower part of the crosshead bearings had been considerably strengthened, the pins stiffened, and the dimensions of the lightening holes reduced. Furthermore, a new arrangement had been investigated for the lubricating oil ducts.

The white metal was applied to the bearings by centrifugal-casting, immediately followed by quick cooling with water jets to get a perfect bond of anti-friction material to the support. This process was very important. In fact, some inconveniences were noticed in certain bearings where, due to accidental lack of water, the cooling of the external part—after casting—was not sufficiently effective.

As concerned Mr. Hinson's last remark, referring to page 32 of the paper, he confirmed that, obviously, the availability of electronic computers, though not allowing improvement of the results, made it possible to obtain, in short time, all the data useful in choosing the best compromise, taking into consideration all the elements of engine operation which were affected by the firing order.

He was most grateful to Mr. Christensen for his interesting contribution. Mr. Christensen had asked what was the interval between the cleaning of the scavenge ports. It was difficult to give an exact figure, for when the engine was running at very high power the ports remained clean.

Normally, the cleaning of the scavenge ports was carried out on occasion of the piston withdrawal for normal maintenance, that was about once a year. More frequent cleanings could be necessary if the engine operated for a long time at low load and if a lubricating oil of unsuitable characteristics was employed. In this case, the cleaning was effected by entering the exhaust pipe—of large dimensions—and cleaning the scavenge ports by means of long steel rods inserted through the exhaust ports. This work took about one hour.

With regard to the variations in operation of the mechanical or total efficiency of the engine, he really had no figure to give. In a long-run test, something like 1,000 hours and burning residual fuel, practically no variation in the efficiency of the engine was noticed. He imagined this was essentially due to the fact of having volumetric scavenging in series with the turboblowers.

The question concerning the arrangement of gutterways in the cylinder liners had often been considered by Fiat, with the conclusion that their presence was actually of some utility especially in the large-bore engines subjected to high thermal loads.

For the cylinder lubrication, type SAE 50 oil was recommended.

Mr. Christensen was right in saying that there were no holes in the crankcase. The oil to the main pin bearing was derived from the cooling system of the piston : the oil entered into the crosshead and passed through the connecting rod and arrived at the big end bearing. To avoid having broken pieces of white metal reach the main bearing of the connecting rod,

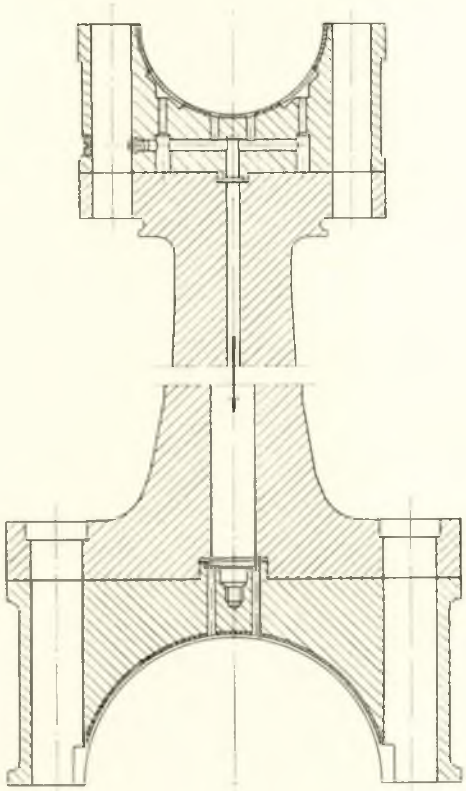


FIG. 19—*Big and small end bearings lubricating oil passages*

the crosshead bearing had been designed as shown in Fig. 19.

Furthermore, also in the upper part of the big end bearing, a small well had been provided for collecting crushings of white metal.

Actually these precautions were almost unnecessary considering the fact that in the engines not provided with this well, no serious inconvenience had ever occurred. On the other hand, with the improvements carried out over the last few years, to the design and technology of the crosshead bearings, it was improbable that white metal particles—of a certain size —would become detached.

With regard to the circumferential distance in terms of degrees of the travelling surface of the crosshead bearings, this was also practically 120 degrees. A reduction in this amplitude was tried, to improve the cooling of the bearings, but the results were not good.

The white metal layer on the crosshead bearings was 4mm. thick, and corresponded to the following analysis : tin 80 per cent, antimony 11 per cent, copper 9 per cent. Actually the copper content made the alloy more brittle, but—at the same time—it improved the hardness. However, the composition indicated seemed to be the best compromise confirmed by operating experience.

With the latest improvements to the crosshead bearings, it was actually possible to attain 24,000 operating hours before the bearings required re-conditioning.

However, it was advisable to check the condition of the babbitt metal of the bearings every 6,000 operating hours.

Wear of the crosshead pins was practically negligible; the pins were kept in service for ten years; after that period it would be necessary to grind them to restore the working surface.

Finishing of the crosshead pins was carried out—by Fiat— by means of a proper lapping machine as shown in Fig. 20. The maximum allowed roughness was 0·1 micron.

The construction of the thrust bearing with inserted thrust ring, was a practice adopted by Fiat for many years, and no trouble had ever occurred. Corresponding to the zone where the coupling bolts were located, the disc thickness was adequately increased.

The temperature of 60 deg. C. (140 deg. F.), recommended for the cooling water was not exceptional, inasmuch as other large-bore engines operated with this temperature.

Mr. Jambunathan said that he had met some difficulties with lubricating and cooling oil in a Fiat engine built many years ago. Probably this engine was still provided with old-type pistons, with free cooling oil circulation where carbon formation could deposit on the walls skimmed by the oil, should the standby oil pumps not be operated for a certain period of time after stopping the engine.

With the most recent, thin-walled type pistons, no carbon formation occurred in the piston and the oil remained clean.

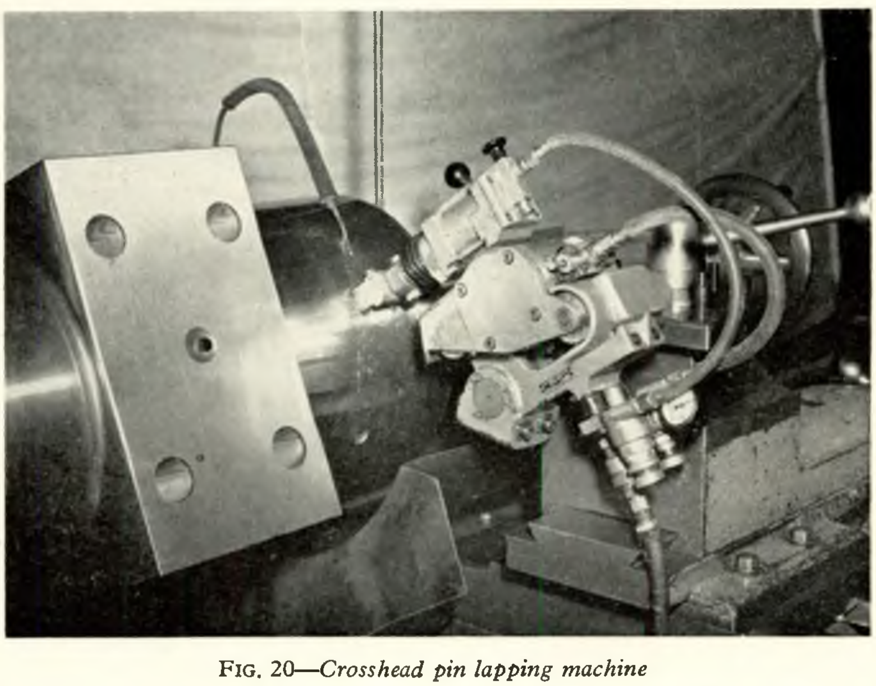


FIG. 20—*Crosshead pin lapping machine*

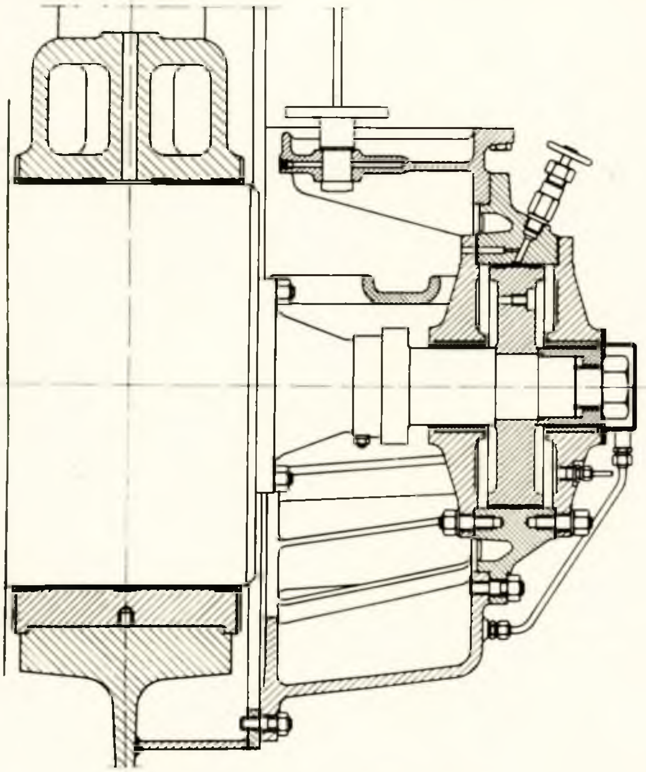


FIG. 21—*Axial vibration damper*

Cleaning of filters—which were of self-cleaning type—was performed about every six months; however, it was recommended that, three times a day, i.e. once each watch, the shaft which carried the cleaning combs of the filtering cartridges be rotated.

Mr. Alcock asked in which way the cleaning of the separating diaphragm open to the engine room was effected. Generally, if combustion was normal, and the cylinder lubricating oil of suitable quality, the deposits were scanty and of little substance; they slid along the sloping diaphragm wall and gathered—through relatively large pipes—into a drainage receiver. If, on the contrary, the deposits were of a certain consistance, they were removed manually by the personnel on board.

The open diaphragm allowed also the ascertainment of the nature and substance of the deposits and—indirectly—the condition of cylinder lubrication and of combustion.

Regarding the temperature fluctuations found in the cylinder liner internal wall, he considered that the supposition made by Mr. Alcock to explain them was rather hazardous. The hypothesis that they were due to gas passing through the piston ring gaps was more trustworthy, though this supposition —as already hinted—had not yet been completely confirmed.

The external part of the piston heads was of cast steel construction. The velocity of the oil within the cooling ducts was from 2 to 2·5m./sec.

Mr. Couchman made some remarks concerning the axial vibrations. This was actually a very important matter in large-bore engines. In the 12-cylinder engine, a frequency of 540 cycles/min. was measured for the first axial vibration mode. However, no dangerous vibration was experienced on board, the ship being fitted with a four-bladed propeller and therefore the resonance phenomena occurred above 115-122 r.p.m., which was the normal engine operating range. In the eight-cylinder engine, the frequency of the first axial vibration mode was 752 cycles/min. and, in order to avoid excessive vibrations due to resonance phenomena with the exciting actions of the engine, an axial vibration damper, as shown in Fig. 21, had been mounted.

A coincidence of frequencies between torsional and axial vibrations never occurred. However, it was noticed that intense torsional vibrations caused forced axial vibrations.

The differences between frequency and amplitude values recorded on the test bed and those obtained on board, were not only to be imputed to the elastic characteristics of the foundation, but also to the restraints existing aboard between the top of the engine and the hull.

These restraints might be caused simply by the gratings or by the connexions placed purposely to reduce the resonance amplitudes of the transversal vibrations.

In the 12-cylinder engine, no rocking movement was noticed. Vibrations of this kind occurred sometimes with very low amplitude and, anyway, not so high as to cause dangerous stresses in some parts of the engine or foundation.

Mr. French raised some questions relating to the temperature measurements made on the liners. Fig. 10 and Fig. 11 were quite different things. The recordings in Fig. 10 were given only in order to give an idea of the form of the temperature curves, and there was no relation between these and the curves in Fig. 11.

The cylinder liner bush on which the radial recording of the temperatures (see Fig. 12) was carried out, was of nodular cast iron, i.e. of a material having a lower thermal conductivity than the external steel part.

The thickness of the external part was 40mm. The cooling water temperature was 48 deg. C. (118 deg. F.).

INSTITUTE ACTIVITIES

Minutes of Proceedings of the Ordinary Meeting Held at the Memorial Building on Tuesday, 13th October 1964

An Ordinary Meeting was held by the Institute on Tuesday, 13th October 1964, at 5.30 p.m., when a paper entitled "Service Performance with the Fiat Marine Diesel Engine Type 900S" by the late Dott. Ing. R. De Pieri was presented by Dott. Ing. A. Gregoretti and discussed.

Mr. W. Young, C.B.E. (Chairman of Council) was in the Chair and eighty members and visitors attended the meeting.

Nine speakers took part in the discussion which followed.

The Chairman proposed a vote of thanks to Dott. Ing. Gregoretti which received an enthusiastic response.

The meeting closed at 7.55 p.m.

Annual Conversaziones 1964

Two Conversaziones were held at Grosvenor House, Park Lane, London, W.1, on Friday, 4th December, and Friday, 18th December 1964. The President, Mr. A. Logan, O.B.E., and Mrs. Logan, together with the Vice-Chairman of Council, Mr. H. N. Pemberton, and Mrs. Pemberton received the members and guests on 4th December, Mr. and Mrs. Logan, together with the Chairman of Council, Mr. W. Young, C.B.E., and Mrs. Young, received the members and guests on 18th December. The numbers present on these two occasions totalled over 1,900.

Music for dancing was played by The Sydney Jerome Dance Orchestra and the Russ Henderson Steel Trio, and among those who appeared in the cabaret were Joan Turner, Lance Percival, The Pan Yan Jue Trio, The Two Maxwells, The Frank and Peggy Spencer Formation Dancers. The floor show entitled "Vitality" included Alan Fenn.

On the evening of 18th December carols were sung after dinner by members of the choir of St. Margaret's, Westminster, S.W.1, and were greatly enjoyed by those present.

At the Annual Conversazione held at Grosvenor House, Park Lane, London, W.1, on Friday, 4th December 1964. From left to right: The President, Mr. A. Logan, O.B.E. (Member), Mrs. Logan, Mrs. H. N. Pemberton and Mr. H. N. Pemberton (Vice-Chairman of Council)

Section Meetings

Merseyside and North Western

General Meeting

A general meeting of the Section was held on Monday, 7th December 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 "Marine Training in France" was presented by Captain B. Sermier.

The lecture was based upon the paper presented at the Education Group Symposium, held at the Institute in London last May, but Captain Sermier had brought it up to date with the latest information on experience with the various trials under way.

The meeting was well attended by some 110 persons, including representatives from Liverpool Education offices, Birkenhead and Riversdale Technical Colleges, the Liverpool College of Technology, the Merchant Navy and Airline Officers' Association and the Liverpool Marine and Engineer Superintendents' Committees.

In the discussion which followed sixteen speakers took part.

Meeting in conjunction with "Ladsirlac"

A meeting in conjunction with "Ladsirlac", was held on Wednesday, 13th January 1965, in the "Brown" Library, William Brown Street, Liverpool, 3, at 7.30 p.m., when a most interesting and informative paper entitled "Safety in Nuclear Ships" was presented by the author, Mr. F. R. Farmer, B.A.

Sir Alan Tod, C.B.E., presided at the meeting which was well supported, about fifty people being present.

Sir Stewart MacTier, C.B.E., B.A. (Companion) opened the discussion and at the conclusion of the meeting a vote of thanks to the speaker was proposed by Mr. J. A. Smith, D.S.C., V.R.D., B.Sc. Mr. T. Kameen (Chairman of the Section) thanked Sir Alan Tod for taking the Chair, and the Liverpool City Authorities, through Dr. Chandler, were thanked for granting the necessary facilities and for making all arrangements.

Meeting by invitation of the Institution of Electrical Engineers

By kind invitation of the Institution of Electrical Engineers, members of the Section attended a meeting on Monday, 18th January 1965, in the Lecture Theatre, the Royal Institution, Colquitt Street, Liverpool, 1, at 6.30 p.m., when Mr. M. J. A. Bolton (A.I.Mar.E.) gave an informal lecture entitled "The Novelty and Anomaly of Electricity at Sea".

A warm welcome was extended to the Chairman of the Section, Mr. T. Kameen, who, with several other members of the Institute attended by special invitation.

Commencing with an outline of the development of electricity in ships over the years, the author reviewed and illustrated the types of electrical system appropriate to the various classes of ship from large passenger liners down to small general purpose craft.

By comparison with standard electrical systems ashore, marine installations were shown to be quite novel, with their wide choice of voltage and frequency and the choice of a.c. or d.c. insulated systems versus earthed systems.

The conservative outlook of the marine engineer in adhering to a familiar low or medium voltage system was shown to produce high normal load currents and potentially dangerous fault currents, but it was acknowledged that in spite of the apparent risk, ship installations have been remarkably trouble-free in practice and rule of thumb engineering appears to have been justified.

Reference was made to the acute shortage of qualified marine electrical engineers, both ashore and afloat, and it was thought to be inconsistent with the general high level of safety requirements that large modern vessels should still be permitted to put to sea without any obligation on the part of their owners to carry suitably qualified electrical staff.

The plea was made that the national associations responsible should not delay too long in remedying the situation.

In the discussion which followed the lecture, marine engineers, electrical and mechanical, were gently chided for a tendency to procrastination in keeping themselves abreast of electrical engineering practice ashore.

Considerable support was given to the view that electrical qualifications up to the standard of the Electrical Technician's Certificate should be introduced as soon as possible, and established as a minimum requirement for senior seagoing electrical staff.

Scottish

A general meeting of the Section was held on Wednesday, 13th January 1965, 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 at the meeting and after extending a welcome to the thirty-nine members and visitors present, introduced the speaker, Mr. I. M. Mackenzie, B.Sc., who presented his paper entitled "The Modern Manufacture of Steel for Marine Purposes".

The British film "Listen to Steel" was first shown by the speaker. The film was excellently prepared and showed the whole process of steel making from the charging platform through the Kaldo, the L.D., the V.L.N. and the Ajax processes of steel melting, vacuum processes, the continuous casting processes and the continuous hot strip mill. The shots taken during the process of forging a hollow drum under an hydraulic pressure were most impressive. The film showed the upsurge in technical processes employed in the manufacture of steel.

Mr. Mackenzie opened his address by saying that he wished to talk about what was being done by the steel industry to maintain and improve the competitive position of steel as a basic constructional material.

He referred to the film and enlarged upon details of steel making processes in Great Britain, pointing out that the basic open-hearth process had held a dominant position for many years, and referred to the types of furnaces and gave details of oxygen-blown processes at the same time pointing out that fast production rates had raised difficult control problems. It had been necessary to develop speedy methods of carrying out chemical analysis. Furnace capacities had increased considerably and a capacity of 250 tons had become common, but there were furnaces of 300 and up to 500 tons capacity in operation. A similar process had taken place in the case of electric furnaces which now range up to 200 tons capacity; the problem with a battery of very large electric furnaces was the peak electrical loading involved and special measures might have to be taken to ensure that all furnaces did not require maximum power at the same time.

Mr. Mackenzie stressed the point that much could and was being done to improve the quality of tonnage steels by improving control of the steel making processes. He said that in the past, the attitude of most engineers and steel producers was that while defective steel was to be deplored, as long as the incidence of defects was low, there was no need to take any special action about them since they would be looked after by the factors of safety. Increasing competition had forced both steel users and producers to revise radically their attitude towards control of quality.

The speaker had endeavoured to forecast future developments in the steel industry and anticipated that there would be further progress towards the use of higher strength steel. He thought that ships might also contain a wider range of special steel products such as clad steel plates, 9 per cent nickel steel and cold-formed sections.

Mr. Mackenzie closed by saying that there was plenty of scope for improvement of the existing manufacturing and inspection processes and much remained to be done to extend the field of application of the more advanced types of steel and steel production.

The discussion which followed was most interesting and the speaker dealt with the replies in a capable and confident manner.

The vote of thanks was in the hands of Mr. H. Brady

At the second Annual Conversazione held at Grosvenor House, on Friday, 18th December 1964. The President, Mr. A. Logan, O.B.E. (Member), is seen here with from left to right: Mrs. Logan, Mrs. W. Young and Mr. W. Young, C.B.E. (Chairman of Council)

(Vice-Chairman of the Section) who paid special tribute to Mr. Mackenzie for the high quality of his paper and the most interesting film.

The meeting closed at 8.00 p.m.

South East England

General Meeting

Following the Annual General Meeting, a general meeting of the Section was held on Tuesday, 19th January 1965, at the Clarendon Royal Hotel, Gravesend, at 7.45 p.m., when papers on Marine Salvage: "General Aspects" by Commander C. G. Forsberg, O.B.E., R.N., and "Typical Examples of Salvage by Re-floating" by P. F. Flett, O.B.E., were presented by the authors.

Mr. G. F. Forsdike (Chairman of the Section) was in the Chair and ninety-two members and guests were present.

After outlining all the different types of salvage, types of ship and the varied types of gear used in salvage work, together with the legal side of salvage work, Commander Forsberg went on to describe the immense task that had been left as a result of the last war and the time taken to clear some of the wrecks around the shores of Britain.

Mr. Flett, with the aid of a large number of slides, gave a series of typical examples of salvage by refloating, explaining each case and outlining the particular difficulties each one had posed.

At question time both the lecturers were able to elaborate on the many points raised by members, finally it was left to the Chairman to thank the two speakers for their excellent papers which received acclaim from those present.

Junior Meeting

A junior meeting of the Section was held on Wednesday, 20th January 1965, at Gravesend Technical College, at 7.30 p.m. when a paper entitled "The Steam Reciprocating Engine" by G. Yellowley (Member), was presented by the author. The Principal of the College, Mr. F. L. Fox, B.Sc. (Hons.) invited the Chairman of the Section, Mr. G. F. Forsdike, to take the Chair and introduced Mr. Yellowley to the thirty-two students present.

Mr. Yellowley's command of his subject held the attention of the audience throughout and the interest of the students was obvious from the many questions put forward by them at question time.

The meeting closed at 9.30 p.m.

South Wales

A junior meeting of the Section was held on Monday, 18th January 1965, at the South Wales Institute of Engineers, Park Place, Cardiff, at 6.00 p.m.

Mr. T. C. Bishop (Chairman of the Section) presided at the meeting at which more than fifty members and students were present.

The showing of the technical film "Fire Below" was introduced by Mr. C. N. Bidgood, O.B.E., M.I.F.E., Chief Officer, Cardiff City Fire Brigade, who ably dealt with the questions which followed.

Mr. H. S. W. Jones (Member) proposed a vote of thanks to Mr. Bidgood for the excellence of his presentation of the film and for his lucid explanation in answer to the many questions.

Following a vote of thanks to the Chairman proposed by Mr. F. F. Richardson (Member) the meeting terminated at 8.00 p.m.

West Midlands

A general meeting of the Section was held on Thursday, 21st January 1965, at the Engineering and Building Centre, Broad Street, Birmingham, at 7.00 p.m., when a paper entitled

"Power Station Oil Fired Boilers" by W. C. Carter (Member) was presented by the author.

Mr. H. E. Upton, O.B.E. (Vice-President) was in the Chair and ninety-five members and visitors attended.

Mr. Carter dealt with boiler design for both land and marine applications together with designs embodying forced circulation. Particular emphasis was made of present developments and design in the manufacture of large generating plant for present day power stations, and the fact that the experience gained in the different constructions of boilers were co-ordinated to provide the best possible compromise.

The developments and features were fully described with the aid of slides.

With the large attendance, it was understandable that the discussion reached an excellent level, all questions being ably dealt with by Mr. Carter.

The meeting closed at approximately 9.00 p.m.

West of England

The first joint meeting of the Section with the Royal Institution of Naval Architects was held on Wednesday, 20th January 1965, in the Lecture Theatre of the City of Bath Technical College, Bath, at 7.00 p.m., when a paper entitled "The Design of Screw Propellers With Reference to the Kort Nozzle" by T. E. Hannan (Associate Member, I. Mar.E.), was read by the author who was also an Associate Member, R.I.N.A.

Mr. J. P. Vickery (Chairman of the Section) was in the Chair and amongst the sixty-four persons who attended were Mr. Ivor E. King, C.B., C.B.E., Chairman of Council of the Royal Institution of Naval Architects, and Mr. F. C. Tottle, M.B.E. (Local Vice-President, I.Mar.E.).

Mr. Hannan gave a most interesting and fascinating lecture with the aid of slides and test charts, and it was obvious that his knowledge of his subject was both unique and complete.

A lively discussion period followed and all questions were answered by the speaker in a capable manner. A vote of thanks was given to both the author and the organizers of the meeting by Mr. King, and this was warmly applauded by those present.

The meeting ended at 9.20 p.m.

Election of Members

Elected on 11th January 1965

MEMBERS

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Frederick William Barnes
Robert Younger Bell, B.A. (Cantab)
Desmond Arthur Bennett
John Francis English Booty, B.Sc. (Dunelm)
Raymond Brennan
George Watt Muckle Chalmers
Harold Frederick Chambers, Cdr., R.N.
Thomas Chambers
Arthur Noel Stewart Cook
Gordon Coxon
Allen Russell Darnley
Edward John Engel
Arnold Henry Forster
Albert Gardner
John Gray Grant
James Albert Green, Cdr., O.B.E., R.N.
Antonio Gregoretti
William Wilson Hunter, B.Sc.
Albert Copeland Jobling
Arthur Wynn Jones
Gilbert William Robert Kenrick
Vincent Ernest Klaas
Richard Lamb
Ernest Levison
Archibald Logie
Gunter Mau
Alan Sharman Meadows
Thomas Cecil Daniel Mordecai
William Morgan
Eric Tage Petersen
Douglas Risk
Robert Edward Rollins
Reginald Henry Sheppard, Cdr., O.B.E., R.N.
Trevor Denison Short, Cdr., R.N.
Frederick Smurthwaite
Norman Stephenson
Robert Campbell Stuart
David Stewart Summers
Ernest Charles Wild
George Percival Williams
John Raymond Williams
Jack Worden
Gordon George Yule

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Samuel Hugh Allen
Brian Geoffrey Armitage
Robert Trevor Bell
William John Best
Eruch Framroze Billimoria
David Humphrey Blazey, Lieut., R.A.N.
John Trevor Bunney
George Malcolm Burnell
Paul Jose Cabral
David Maurice Campbell, B.Sc. (Glasgow)
James Adam Cartwright
James Patrick Chester
Barry Cowper-Foster
Hans Christian Doderlein
John Vincent Dodshon
James Fairweather
Donald Gair
Malik Arshed Gilani, Lieut., P.N.
John Goldie
Fabian Anthony Gonsalves
Ramon George Goodley
Henry Walter Lindsay Griffiths
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Berowald Macintosh Innes
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Samuel William Marsh
John Metcalf
Peter Garland Millett
John James Morrison
Edward Nicol
Parthasarathy Ram Gopal
John Garnet James Richardson
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Elam Liston Taylor
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Peter Henry Visser
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James Ramsay Wilson

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Allan Rex Bray

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Henry Headspith
David Morris
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Charles Eric Pepper
Allan Gilbert Stent
Richard Gordon Stokes

GRADUATES

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William Terence Carroll
Chan Cheuk Man
Michael Finlay Craig, B.Sc. (Durham)
Robert George Herkess
John Leonard Hobbs
Houshang Hosseinzadeh, Sub. Lieut., I.I.N.
Derek Leviston
Robert Harrison Lewis
Cedric Robert McGregor
George Irving Papworth
Linus Saldanha, Lieut., I.N., B.Sc. (Eng.) (Mysore)
Abdul Shakoor
George Donald Smith
Thomas Smith
Peter Ramsay Snadden
Robert Edward Wilkes

STUDENTS

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Colin Stuart Barry
David Anthony Brown
John Byer
Michael Eric Chamberlain
Oliver Cousins
Peter Reid Crawford
Daniel Baxter Easton
Alexander Kodzo Ekpe
John White Cathcart George
Raymond Frank Gill
Stanley Grant
Kevin Michael Hall
Peter Clarke Hann
Thomas Richard Harker
Stephen Harris
William Andrew Henderson
Robert Henry Hoult
John Elliot Hughes
Gerald Leslie Humphreys
William George Bailey Leitch
David John Leslie
Thomas Crawford McDermott
Duncan Forbes MacDonald
Alistair James McGregor
John Alexander McIntyre
Alasdair Campbell Morrison
Alan David Newberry
Richard James Newman
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James Rew Stevenson
Peter Martin Stockwell
Robert Henry Taylor
Peter Anthony Thomas
John Joseph Trawber
Roy Charles Tubbs
Andrew Robert Tully
Roger Wale
Graham Lorentz Walker
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Alex. George Wilson
Wong Chee Yuen
George Robertson Yeaman

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James Arthur Chappell
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Robert Ian Gatherer
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D. I. McCulloch
William John Sinclair Smith
David John Solomon
Ralph Tucker

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Dudley Bernard Abraham
Alfred Gladwin Bartholomeusz
Phillip Curtis
Peter Dobbie
Mohammad Mustafa Jaffari
Satya Prakash Kapoor
Peter Spencer McMillan
Norman Douglas Merry
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Walter Ross
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Anthony Simpson
Walter McDouall Stuart
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TRANSFERRED FROM ASSOCIATE TO MEMBER

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Marinus Hengstmengel
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John Wilfred Knights
Bernard O'Connor
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Brendan Aloysius McCann
John Richard Mathews
Ranjit Singh Mundi, Lieut., I.N.
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Alan Tout
Robert Graham Woodgate

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Richard Jeremy Daulby
William Nicol McIlvean
Keith Taylor

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TRANSFERRED FROM GRADUATE TO ASSOCIATE

Peter Edlington
John Meriton Stirling

TRANSFERRED FROM STUDENT TO GRADUATE
Derek Anthony Hancock
Robert Brian Millard

TRANSFERRED FROM PROBATIONER STUDENT TO GRADUATE
Charles Frederick William Dean
Peter John Mould

TRANSFERRED FROM PROBATIONER STUDENT TO STUDENT
David James Greene
Anthony George Richard Manser
Simon Charles Sutton
Rory John Lawrence Walker

Elected on 8th February, 1965

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Harold Cecil McIntyre

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Maurice Godfrey Lancaster, B.Sc.
Archibald Muir
Francis Mullen
Malcolm Frederick Musgrove
Robin Crawford Reid, B.Sc.
Harold Richardson, B.Sc. (Durham)
David George Sadler
Rajinder Sehgal, B.Sc. (Indian University)
Jaswant Singh
David Tindall

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John Frederick Cousins
John Neil Dryden
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Michael John Minnis
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Raymond Wild
Thomas Andrew Wilkin
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Rodger Lowcock
Christopher Trevor Norman
David Peter Orman
John Duncan Pinchin
Peter Bryan Pinner
Peter Joseph Reilly
Stewart Edward Richards
Michael Patrick Rickard
Alan Peter Sayers
Graham William Schofield
Paul Thomas Smith
Peter John Smith
David Alexander Stewart
Christopher Iain Stockley
Colin Storey
John Wilson Edwin Thomson
John William Watson

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Andrew Gordon Armstrong
Ralph Myers Brown
George Merion Dixon
Frank Vincent Edwards
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Thomas Jaffrey Henderson, Lt. Cdr., R.N.
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Krishan Kumar Malik, B.Sc. (Bombay)
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Eric Hewitt Oakes
Reginald James Oldreive
Leonard Eric Pendry
Sydney William Pierce
William Rutherford
David Wilson
Robert Edward Witton

TRANSFERRED FROM ASSOCIATE TO MEMBER
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John Yeoman Loveridge
John Porteous
Sidney Victor Rainey

TRANSFERRED FROM ASSOCIATE TO ASSOCIATE MEMBER
Krishna Sadashiv Durge, Lieut., I.N.
Joseph Anthony Justin Rodrigues, Lieut., I.N.

TRANSFERRED FROM GRADUATE TO ASSOCIATE MEMBER
Hassan Masood Ansari, Lieut., P.N.
Ian Grantley Baker
James Bowman
Patrick John Cafferty
Ian Robert Jamieson
Surendra Nath Jha, Lieut. (E), I.N.
Stephen Malcolm Hector Parks
Raghunath Patnaik, Lieut., I.N.

Ronald Allan Purvis
Frederick Philip Russell

TRANSFERRED FROM STUDENT TO ASSOCIATE MEMBER
Erik John Bannister
Ronald Thomas Fairclough
Brian Richard Grace Jones
Geoffrey Charles Rae
James Kenneth Rankin
Alan Stewart Whitaker

TRANSFERRED FROM PROBATIONER STUDENT TO ASSOCIATE MEMBER
Malcolm Reid
Michael Antony Pierpoint Westropp

TRANSFERRED FROM GRADUATE TO ASSOCIATE
Kenneth Church
Abdus Samad
Alan Thomas Tindall

TRANSFERRED FROM STUDENT TO GRADUATE
Michael Edwin Gordon Hadlow

TRANSFERRED FROM PROBATIONER STUDENT TO GRADUATE
Roger Thistlethwaite
Brian Turner

TRANSFERRED FROM PROBATIONER STUDENT TO STUDENT
David Thomas Brown
David Fraser Darrah

It is much regretted that, in the November 1964 issue of the TRANSACTIONS, there was published in error an obituary notice of Mr. David Wilson Boyd (Member), to whom we offer our most sincere apologies.

OBITUARY

The Right Honourable The Earl Alexander of Hillsborough, K.G., P.C., C.H. (Honorary Member), Leader of the Labour peers in the House of Lords until the 1964 general election, died on 11th January at the age of seventy-nine. He had held many ministerial offices and, when in the House of Commons, had been First Lord of the Admiralty during five years of the last war. He was created a viscount in 1950 and had previously sat in the House of Commons as Labour member for the Hillsborough division of Sheffield since 1922, with the exception of the period 1931 to 1935.

Albert Victor Alexander was born at Weston-super-Mare on 1st May 1885, and went to Barton Hill elementary school in Bristol. On leaving school he began work as a junior clerk in the offices of the Bristol Education Committee. Five years later he entered the service of the Somerset County Education Committee, with which he remained for seventeen years, rising to the post of chief clerk. He also acted for some time as secretary of the Somerset Branch of the National Association of Local Government Officers.

When war broke out in 1914, he joined the Artists' Rifles, attaining the rank of corporal and, later, a commission. He served throughout the war and was demobilized with the rank of captain.

A keen supporter of the Co-operative movement, he was, in 1920, appointed secretary to the Parliamentary Committee of the Co-operative Congress and it was as a Co-operator that he first entered Parliament in 1922, as Member for the Hillsborough division of Sheffield. Until 1924 he acted as whip to the Co-operative Party, having been re-elected in 1923 with an increased majority. He was appointed Parliamentary Secretary to the Board of Trade in the first Labour Government and was made First Lord of the Admiralty in the second Labour Government, becoming at once immersed in the negotiations of the London Naval Conference. He lost his seat in 1931, but returned to Parliament in 1935.

Although still in opposition when war broke out in 1939, he gave full support to Mr. Churchill's administration of the Admiralty and was kept fully informed on naval matters. Thus, when, in the Coalition Government of 1940, he was appointed to be First Lord for the second time, there was continuity of policy. His competence and extensive grasp of the problems of naval administration were of great value to the country throughout the war. Replaced at the Admiralty by Mr. Brendan Bracken for the duration of the "caretaker" Government, he returned there on the formation of the third Labour Government.

In 1946, after the Government changes in October, he was appointed Minister without Portfolio. At the same time, proposals were put forward for a Central Organization for Defence, which included plans for the appointment of a Minister of Defence. Mr. Alexander was to remain in the Cabinet and his name was to be submitted for that office. In December 1946, after the Ministry of Defence Act had reached the Statute Book, his appointment as Minister of Defence was announced.

In the New Year Honours List for 1950 he was elevated to the peerage and when, after the general election of that year, it was felt essential that the Minister of Defence should be in the Commons, he accepted the post of Chancellor of the Duchy of Lancaster. He continued to represent the Government on defence in the Lords and when later the Labour party went into Opposition, he made many contributions to defence debates. On the retirement, in November 1955, of Earl Jowitt as Leader of the Opposition in the Lords, Viscount Alexander, the Deputy Leader, accepted nomination and was elected unanimously.

He had been a Privy Councillor since 1929, and an Elder Brother of Trinity House since 1941. He was made a Companion of Honour in the latter year. In 1963 he was created an earl.

He married, in 1908, Esther Ellen, youngest daughter of the late George Chapple, of Tiverton, Devon. They had one daughter. There is no heir.

LIEUTENANT-COMMANDER JOHN DAVENPORT BUCKLEY (Member 22060) died in December 1963, leaving a widow.

He entered the Royal Navy, in 1939, after sitting the Civil Service Commissioners' Examination, and, from 1940 to 1943 completed a course at the Royal Naval Engineering College at Keyham. He went to sea, under training, in cruisers, in December 1943, becoming a watchkeeping officer in 1944. From 1946 he served as Engineer Officer, mainly in submarines. He was Staff Engineer Officer with the Naval Mission to Greece from November 1952. He joined the Reserve Fleet at Malta in 1953 and later served as Senior Engineer and Engineer Officer in cruisers and frigates in home waters and the Mediterranean. In November 1959, he became Senior Engineer in H.M.S. *Vanguard*.

He joined the Atomic Energy Research Authority at Harwell in early 1961, as an Operations Shift Manager, on the DIDO reactor. Before completion of the customary six months' training, internal re-organization required that he transfer to the BEPO reactor, where he completed training and took up his duties as Shift Manager at the end of 1961, in which appointment he remained.

He made his mark upon the BEPO organization as a man of sensitivity, integrity and courage, which, coupled with an equable temperament and a strong yet wholly admirable personality, allowed him to deploy a small technical team of reactor operators to best advantage. Those who worked with him found it a pleasure, augmented by his broad appreciation of culture and the arts, and his practical philosopher's tolerance of the imperfections of the world around him. He was firm but just, when the occasion demanded, but could always be relied upon for sympathetic real help for his colleagues in difficulties.

Lieutenant-Commander Buckley was elected a Member of the Institute on 3rd March 1960. He was also an Associate Member of the Institution of Mechanical Engineers.

PETER MARTIN, B.Sc. (Member 8996), died on 29th July 1964 at the age of forty-four, after a long illness. He was educated at Eltham College and Rochester University, N.Y., and his apprenticeship was served with Wm. Denny and Bros. Ltd. He went to Glasgow University, with a Denny Scholarship, to take a Bachelor of Science degree in mechanical engineering.

Joining the Army in 1940, he was commissioned in the Royal Engineers and served throughout the war, finally with the rank of Captain in an Armoured Assault Squadron, developing and using, in action, special equipment for the attack of heavily defended positions.

He was released in 1946 and joined the then newly-formed National Gas Turbine Establishment where he later became a Principal Scientific Officer. He made many contributions to the knowledge of mechanical and combustion design during the first post-war phase of gas turbine development and, more recently, was one of those responsible for the development of the ambitious facility at N.G.T.E. for testing engines under simulated flight conditions.

With a lifelong enthusiasm for sailing, he became a senior officer and an active and successful sailing member of several clubs devoted to dinghy and ocean racing, and small boat cruising. His interests extended also to competition driving. A qualified marine engineer, he was greatly interested in the technical aspects of sailing and only eighteen months prior to his death, had set up a successful boat-building enterprise, managed in his spare time.

Mr. Martin was elected to membership of the Institute, as a Student, in 1939, transferring to Graduate in 1940. He was elected to full membership on 3rd February 1954.

He leaves a widow and four children.

CAPTAIN CHARLES EDWARD SIMMS, D.S.O., R.N. (Member 12674), who died on 30th December 1963, had been a Member of the Institute since 5th December 1949. He was also an Associate Member of the Institution of Mechanical Engineers.

Born on 25th May 1900, he was trained in H.M.S. *Conway*, winning a Samuelson Prize for engineering. From 1915 to 1923, he was at the Royal Naval Colleges at Dartmouth and Greenwich and also the Royal Naval Engineering College at Keyham. He joined the Fleet in 1923, serving in H.M.S. *Marlborough*, and embarked upon an extensive and varied career with the Navy. He served at sea as Engineer Officer and Senior Engineer, both with the Royal Navy and the Royal Australian Navy, he was also at one time during the Second World War, Engineer Officer to the French Fleet at Devonport. Other appointments included two years as a lecturer at the R.N.C., Dartmouth, a period when he was engaged on Divisional regulating duties at Chatham and, from 1940, a year when his services were on loan to the Ministry of Mines (Petroleum Warfare). From 1941 to 1944, he was Superintendent of the Royal Canadian Navy Dockyard at St. John's, Newfoundland, from 1944 to 1945 he held an appointment at the Admiralty and, from 1945 to 1947, he was Captain (E) in command at H.M.S. *Imperieuse*, Stephens Training Establishment, Devonport. In 1947 he became Fleet Engineer Officer, Mediterranean Fleet, on the staff of the Commander-in-Chief, Mediterranean. He retired from the Navy in 1951.

On his retirement from the Service, Captain Simms worked with the idea of establishing his own business, and the firm of C. and A. Simms (Engineers) Ltd. was incorporated in March 1953, concerned mainly with the importation from Germany of wood-working and injection moulding machinery, much of which is unique and has no counterpart in the United Kingdom.

Apart from his business, Captain Simms' activities centred chiefly around the Conservative Association, of which he was Maidstone Divisional Vice-Chairman, and the village cricket club, in which he took a lively interest. His hobbies included a collection of pewter and a number of antique clocks; he was always a keen gardener.

WILLIAM OGG STEPHEN (Member 7231) died suddenly on 30th June 1964, in his sixty-third year.

He served an engineering apprenticeship at the Lilybank Engine Works and, in March 1924, joined the Union Castle Line as a junior engineer officer. He gained a First Class Ministry of Transport Certificate, with Motor Endorsement, whilst serving with the company.

During the Second World War, Mr. Stephen was serving as Intermediate Second Engineer Officer in m.v. *Dunvegan Castle* when that vessel was torpedoed in August 1940. He transferred, soon after this incident, to Admiralty service and served as a chief engineer until May 1946.

On his return from Admiralty service, he joined the *Carnarvon Castle* as Intermediate Second Engineer Officer and was promoted to Chief Engineer of the *Sandown Castle* in March 1947. He served in that vessel until May 1949, when he transferred, first into the *Riebeeck Castle*, for a short period, and then into the *Kenilworth Castle*, where he stayed until June 1952. In July 1952, he was promoted to the passenger service and was appointed to the *Llangibby Castle*. He remained in the passenger service as a Chief Engineer until July 1962 when, because of a reduction in the passenger fleet, he returned to the cargo service. His last vessel was the *Rowallan Castle*, which he joined in November 1962 and left, at Gothenburg in May 1964, for medical treatment.

Mr. Stephen was a bachelor and came from Broughty Ferry, near Dundee. He was elected a Member of the Institute on 5th December 1932.