The Development of Heavy Duty Marine Diesels During the Past Five Years

W. KILCHENMANN (Member)*

The paper begins by explaining that although it was intended to cover the development of heavy duty marine Diesel engines during the last five years, the author cannot confine his observations within such a limit as from the scope of developments as have been completed in the time in question they must, of course, have begun before Spring of 1956. Thus the paper can only touch upon developments, a greater part of which really belong to an earlier period, but which are still, today, being carried on and perfected in detail by various companies.

Furthermore, the paper is intended to cover as generally as possible, the problems to be solved, as they are of general application and will affect every engine builder.

The examples and solutions described in the paper are those experienced by Sulzer Brothers, and as a representative of the company the author has not attempted to comment on the technical solutions of other manufacturers.

INTRODUCTION

Ten years ago, another period of rapid development in Diesel engine design began, similar as regards intensity and importance to the main development periods of earlier times. The principal problems which have had to be solved in the last few years for the technical development of heavy duty marine Diesel engines are the following:

1) Welded design;
2) Combustion of heavy fuels;
3) Turbocharging of large two-stroke engines;
4) Further increase of output by adoption of large cylinder bores.

The demand for welded engines was made much earlier and has been satisfied by several manufacturers. In particular the welded frames of medium size American engines, such as General Motors or Fairbanks-Morse. The large engine for the merchant marine field has, however, quite a different aspect. In isolated cases, welded frames were realized long ago, e.g. in 1933 by the late Mr. Keller at Doxford's. It is of course true that this opposed piston engine is particularly favourable for welding, as the large combustion loads are taken off the frame by its moving side-rods. But the great change-over to the welded design only began within the last decade. Even if this period exceeds the last five years covered in this paper, the author would like to discuss this problem, as this change-over still has not led to a satisfactory solution in some places and thus is not yet completed.

WELDED DESIGN

The demand for welded frames in the first place raised problems of design and manufacture. It was, of course, not possible simply to replace the cast frame elements of the existing engines by similarly shaped weldments. Experiments in this direction were not particularly successful. Welded engines call for forms adapted to the possibilities of the welding process and the peculiarities of welded designs. Careful consideration of the flow of force and caution in the avoidance of stress concentrations, coupled with the shaping skill of the designer, are necessary for the design of bedplates and frames capable of withstanding high loads without the danger of cracks and yet simple and easy to manufacture. They should be made up of a minimum of cut-out plates and assembled by readily accessible welding seams. Thus the old ribbed shapes suitable for casting had to be changed to the box design of welded bedplates and frames, first introduced for the columns by Doxford before the war.

Here the question arises as to whether basically a welded design should be expected to withstand the great firing forces repeated at each stroke and reaching some five hundred tons for the largest bores, or whether tie-rods should be used. As previously with the cast engines, both principles are still applied today in welded designs. Cast iron marine engines made by the author's company had no tie-rods, but with the change-over to welding they were introduced in the conviction that they would considerably simplify the problem. By their use, the large pulsating tensile forces are guided in an absolutely straightforward way through sound forged material, and not through complicated shapes with obscure stress concentrations, which may also be weakened by imperfect welding seams. In this design, the tie-rods pass from the top of the cylinder block down below the main bearing in the bedplate and take the tensile stresses produced by combustion in the cylinder, thus relieving all welded parts of tensile stresses (Fig. 1). These tie-rods are of course pre-stressed at assembly in such a way as to keep the welded parts under compression at all times, even at the maximum combustion loads. In order to keep the bending stresses in the bearing saddle as low as possible, the tie-rods are positioned very close to the bearings. This requires a special arrangement for the fastening of the main bearing cap, which is pressed down from above by jack-bolts. This design has proved to be very satisfactory, not only here but also for many years in engines with strong upwardly directed forces, like four-stroke or double-acting two-stroke engines. Those who believe in cast iron engine parts often object that welded constructions are much more elastic and therefore will be subject to excessive deformations and particularly transverse vibrations. In reality exactly the opposite is the case. The modulus of elasticity of steel, being twice as high as that of cast iron, combined with the inherently stiffer

*Technical Director, Diesel Engine Department, Sulzer Bros., Winterthur, Switzerland.

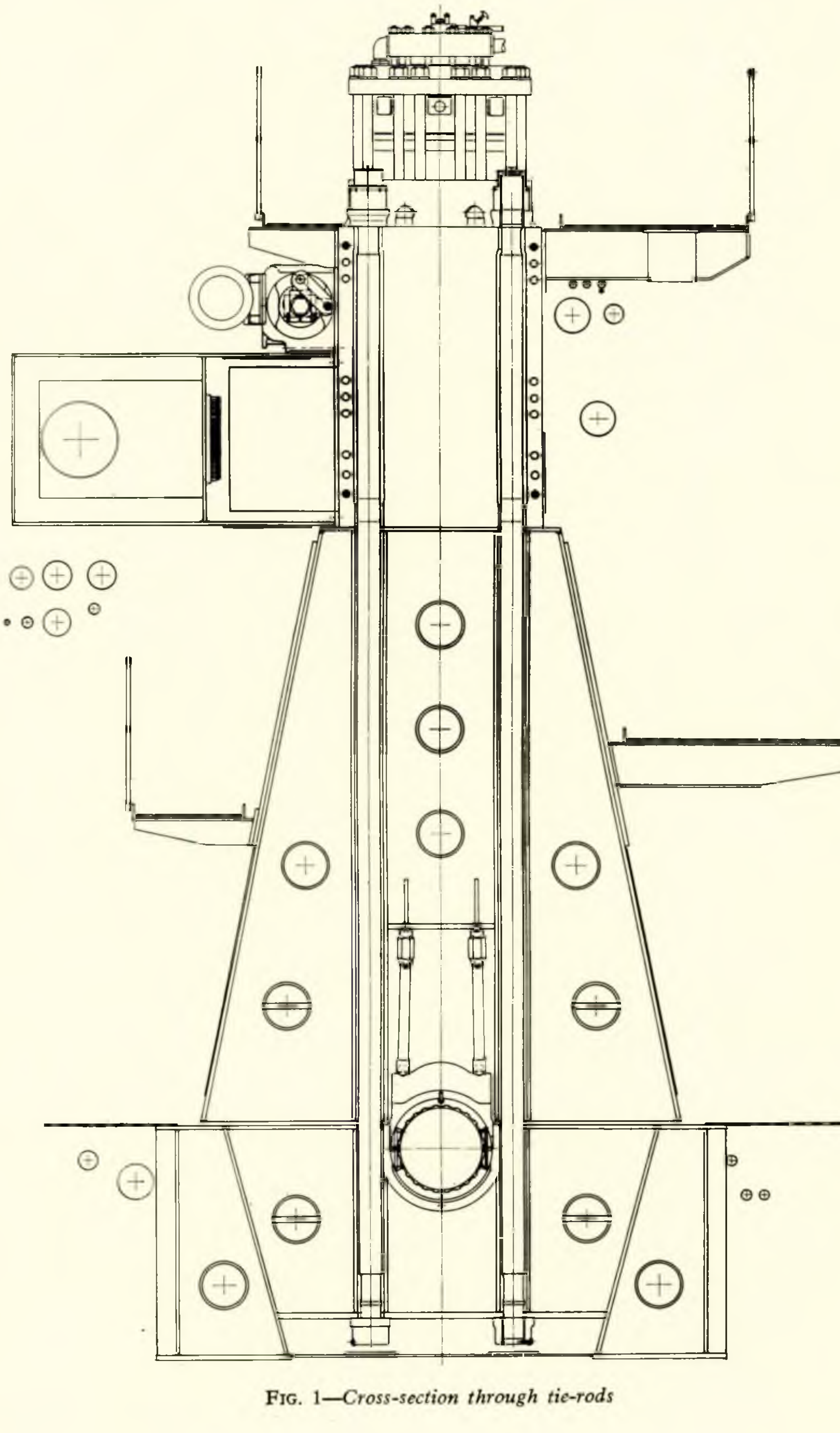


FIG. 1—*Cross-section through tie-rods*

FIG. 2—*Cross-member of bedplate*

box design makes correctly designed welded engines stiffer and less prone to transverse vibrations in spite of their lower weights. The following pictures will illustrate the manufacture of welded bedplates and frames.

Fig. 2 shows the finished cross member of the bedplate; in this form it is annealed to relieve the welding stresses. Around each tie-rod there is a square box which takes the force exerted by the tie-rod and transmits it as directly as possible to the frame, from which it is led to the cylinder block. In Fig. 3 the cross-members are assembled with the longitudinal girders to form the finished base. Fig. 4 shows a finished A-frame. Here also, the plates are arranged in a square box-shaped structure around the tie-rod, thus directly taking the compression force exterted by it.

BOILER FUEL UTILIZATION

The second point mentioned—that of boiler fuel utilization—was also settled in the main before the five-year period under review, but a few words could be added on the subject. Its problems were regarded at the beginning as being much more difficult than they were in reality, and far too much has been written and said on this subject. In the meantime, it has been found—at least for the large slow speed engines—that it is not primarily the engine which has to be developed for the combustion of boiler fuel, but that appropriate means have to be found for heating and purifying the heavy fuel properly.

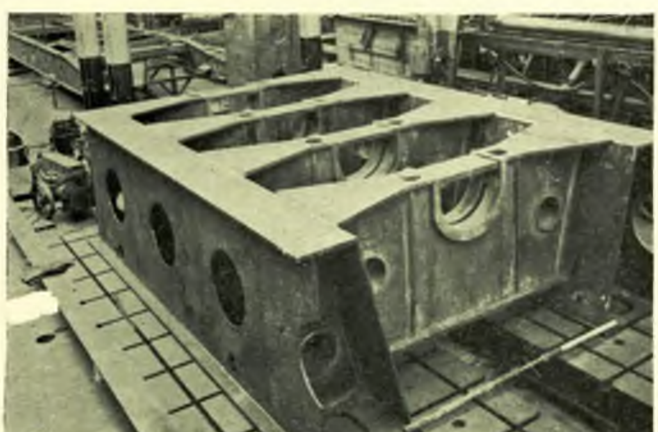


FIG. 3—*Fabricated bedplate*

FIG. 4—*A-Frame*

It is thus mainly a problem of fuel preparation. In some aspects, however, the engine itself was also involved.

In order to prevent the formation of carbon trumpets on the injector tip, the latter requires intensive cooling. For this reason, a special fuel nozzle has been developed by the author's company, in which the cooling medium is led down very near to the nozzle tip, thus controlling its temperature very closely. In the many years of its use, this design has proved to be completely free of the carbon trumpets, which have often been known to give trouble when burning boiler fuel. The results, shown in Fig. 5, of temperature measurements with the conventional type of injector and the new one utilized for boiler fuel show the great temperature difference at the critical spots. With cold gas oil (68 deg. F. = 20 deg. C.) the difference in the tip temperatures is not very important. The use of hot boiler fuel however raises this temperature by 150 deg. to 350 deg. F. (83 deg.; 175 deg. C.) for the new design, but by as much as 245 deg. to 580 deg. F. (135 deg.; 305 deg. C.) for the conventional type. In view of such high temperatures, it is easily understandable that the old type often suffered from trumpet formation.

Owing to the high temperatures of the boiler fuel—which has to be heated to lower its viscosity to less than 27 centistokes —it is necessary to increase the clearance somewhat between the fuel needle and its guide and between the plunger and bush in the fuel injectors and fuel pumps, compared to the values for light fuels, in order to avoid sticking of plungers and needles. This is particularly important during change-over from light to heavy fuel or *vice-versa,* i.e. when the temperatures of the fuel are changed. Such a change-over is actually not necessary for manœuvring, from the purely technical standpoint, as in this design a circulation system keeps the fuel pump and its piping always at the running temperature, and the high pressure lines are heated by a steam tracer pipe.

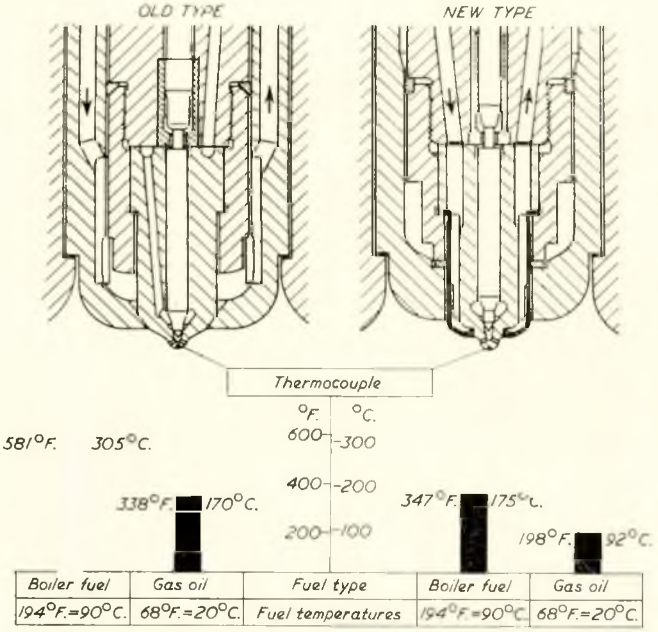


FIG. 5—*Fuel nozzle temperatures 1T48*

Bore = 480 mm. Stroke 700 mm. Speed = 250 r.p.m.
Full load b.m.e.p. = 71lb./in.^{2} = 5 Kg./cm.^{2}
Nozzle Cooling Water Temperature = 122 deg. F. = 50 deg. C.

However, in practical service this change-over is still sometimes used for various reasons.

With regard to the use of boiler fuel, in order to prevent the entrance of sulphurous and sometimes corrosive combustion residues where they might pollute the bearing oil and even cause corrosion, the crankcase is separated from the cylinder part of the engine by a diaphragm with stuffing box. With the use of boiler fuel in general somewhat greater fouling and increased wear of piston rings and cylinder liners previously had to be reckoned with. However, this inconvenience can nowadays be compensated by the use of special highly alkaline cylinder lubricants.

REALIZATION OF HIGHER OUTPUTS

The main development problems of all companies in the last five years have been, firstly, the introduction of turbocharging for large two-stroke engines with a view to increasing their outputs and reducing their weights and prices and, secondly, the attainment of still higher individual outputs than the Diesel engine has yielded up to the present for the purpose of entering the power range above 10,000 b.h.p.—a field so far generally monopolized by steam turbines—and of introducing there the advantage of lower fuel consumption. This power increase has been obtained on the one hand by the turbocharging of two-stroke engines of conventional diameters and on the other by the adoption of still larger cylinder bores. In both cases, the first engines were put on the market by Burmeister and Wain.

TURBOCHARGING

Regarding supercharging, it may well be asked why it took so much longer to apply this to the two-stroke than the four-stroke engine, in which it has been commonplace for many years, after having been pioneered by Dr. Alfred Büchi. For the four-stroke, the problem is much simpler; all that is needed to increase the weight of air available for combustion in the cylinder by raising the general pressure level, is in this case a free-running turbo group consisting of an exhaust turbine and a charging blower. Here advantages may also be gained by special measures, such as the adoption of the pulse system with combination of the exhaust ducts according to the firing order, and appropriate choice of valve overlap and intercooling of the air. In the two-stroke engine the problem is not quite as simple, as this type is basically not self-aspirating like the four-stroke engine, but requires—at least for starting and manœuvring—another source of energy in addition to the turbocharger for the supply of scavenge air. The same basic approach, namely, a general increase of the pressure level as with the four-stroke, may also be used here by arranging a turbo group ahead of the complete engine including its scavenge pump (see Fig. 6a). This solution—first proposed by Curtis—was applied many years ago and is still used frequently today. Here, the scavenge air is compressed in two stages—turboblower and conventional scavenge pump—connected in series and is usually intercooled between the stages. This method has the drawback of retaining the heavy and voluminous scavenge pump with all its losses. The engine is consequently expensive, while the fuel consumption is not reduced by comparison with the atmospheric version. A certain improvement in fuel consumption and expenditure may be obtained by arranging the two compressors not in series, but in parallel (see Fig. 6b). Each stage then compressess only part of the total volume to the full scavenge pressure, and the size of the parallel compressor is substantially reduced, as it only has to deliver a small additional volume of air. This compressor, driven by the engine itself or by any outside source of energy, may be a conventional scavenge pump with a material reduction in displacement volume, or else a rotary blower. The advantages mentioned above are offset to some extent by the not very satisfactory behaviour of this combination during starting and manœuvring. For this reason the straight parallel system is rarely utilized, preference being given to compromise solutions such as combined series—parallel arrangement or even a complete change-over to series arrangement during starting.

As the energy shortage in starting and idling is only small, it is possible to deliver it to the turbocharger itself by any kind of auxiliary drive. This may be done electrically or by an oil

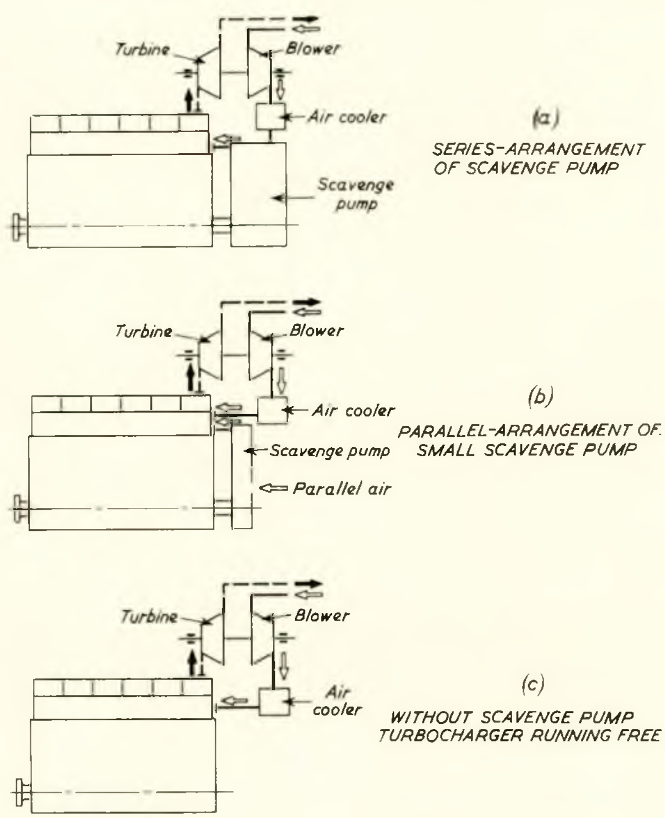


FIG. 6—*Basic arrangements for two-stroke turbocharging*

turbine supplied from the lubricating system, or also by an over-running clutch and step-up gear from the engine crankshaft; all solutions which have either been realized in practical service or at least demonstrated in tests. All of them, however, require undesirable additional elements.

The goal aimed at of course is the supercharging of the two-stroke engine by a free-running turbo group alone (Fig. 6c), which would avoid all additional blowers, driving elements, etc. At the present stage of development this can just about be accomplished with the turbochargers commercially available working under full-load conditions, but there remains the problem of starting and manœuvring. In a start the engine is accelerated from rest by compressed air, without the blower delivering any pressure. Thus it is necessary to accelerate the blower as fast as possible, or to have another supply of scavenge air during idling and acceleration. For uniflow engines the energy necessary for accelerating the turbocharger may be obtained from the engine by opening the exhaust valves a little earlier than normal, which, in combination with the relatively narrow exhaust ducts of the pulse system causes rapid acceleration of the turbo group.

For their cross-scavenge engines the author's firm has chosen a different method and has tried by a trick to retain this inherently simple scavenge system and still to obtain the same good starting and manœuvring conditions as in any unsupercharged engine by utilizing the piston underside as a scavenging aid. This particular solution, which does not require any additional structural elements is now described.

The Sulzer RD-turbocharging system

As Fig. 7 shows, the scavenge receiver is sub-divided. The outer part is common to the whole engine, the inner part, to-

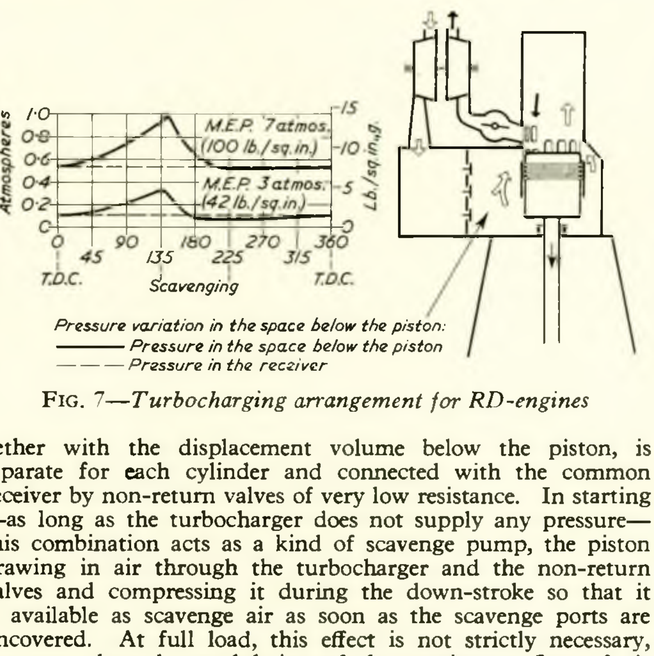
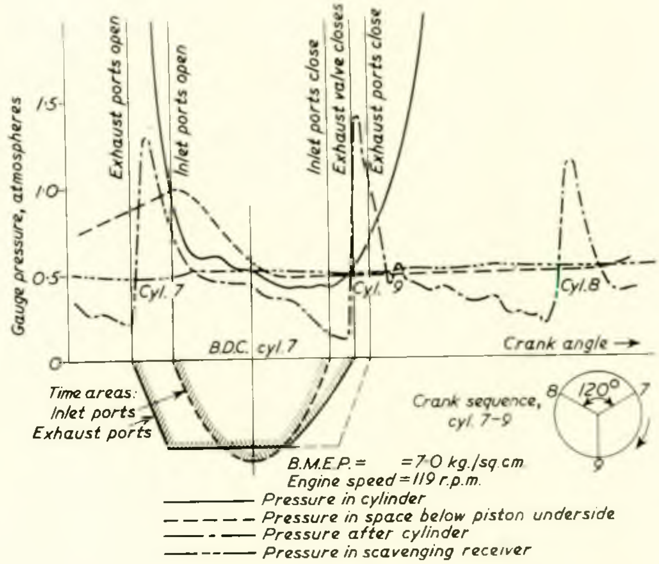


FIG. 7—*Turbocharging arrangement for RD-engines*

gether with the displacement volume below the piston, is separate for each cylinder and connected with the common receiver by non-return valves of very low resistance. In starting —as long as the turbocharger does not supply any pressure— this combination acts as a kind of scavenge pump, the piston drawing in air through the turbocharger and the non-return valves and compressing it during the down-stroke so that it is available as scavenge air as soon as the scavenge ports are uncovered. At full load, this effect is not strictly necessary, but even then the modulation of the continuous flow of air from the turbocharger by the superimposition of a periodical accumulation and compression by the piston underside offers certain advantages. Owing to the existence of a pressure peak in the air chamber at the moment when the scavenge ports are uncovered, scavenging begins with a strong momentum, which means that the scavenge air jets entering the cylinder have considerably higher penetration than in the case of scavenging with constant receiver pressure. Close to the bottom dead centre—after the piston has uncovered the scavenge ports completely—the pressure in this chamber has decreased to the receiver pressure, and in a second phase of scavenging the air flows directly from the common receiver through the valves to the ports. As soon as the piston has closed the



9RSAD 76

FIG. 8—*Pressure fluctuations in the gas-exchange system of cylinder 7*

scavenge ports on its up-stroke, new air is again accumulated in the scavenge air chamber.

Fig. 8 represents the pressure variations in cylinder, scavenge chamber and exhaust pipe during the scavenging process. It shows that RD engines have their blowdown period co-ordinated in such a way with the magnitude of the pressure peak in the scavenge chamber that at the moment when the piston uncovers the scavenge ports the pressure in the cylinder has just fallen to this value. This makes it possible to avoid any blowback of combustion gases through the scavenge ports into the air chamber. The scavenge ports and chamber therefore remain clean and the danger of fires is precluded, while no blowdown energy is lost to the turbine.

Owing to the pressure peak in the scavenge chamber, the blowdown area, i.e. the difference between exhaust port and scavenge port height, can be made considerably smaller than is usual with normal pulse turbocharging. For the engine this results in an extension of the expansion stroke, whereby the mechanical energy used for the compression of the air in the scavenge chamber is partly compensated. Furthermore, owing to the pressure peak in the scavenge chamber a greater air quantity flows through the cylinder, supplying additional energy to the turbine by a stronger impulse. The increased throughput of scavenge air lowers the thermal load of the engine still further. The effect of the piston underside is so considerable that even if all turbochargers should fail, a ship could still proceed at 70 per cent of her full speed. All these advantages have been obtained without any additional constructional elements worth mentioning.

Scavenging Tests

In view of the importance in turbocharged two-stroke engines of obtaining a scavenging process giving optimum purity with modest scavenge air expenditure, new efforts have been made to investigate systematically whether and how the scavenge system can be further improved. It is not easy to make any substantial improvement in a port arrangement already developed to a high degree of perfection and long proved in service on non-supercharged engines.

Scavenge tests on an engine are difficult to make and require a good deal of work if reliable results are to be obtained. To permit systematic procedure and quick progress, a scavenging model was built on a reduced scale (Fig. 9), in which the scavenge and the exhaust ports could be easily varied in shape, section and direction. The influence of these variables was

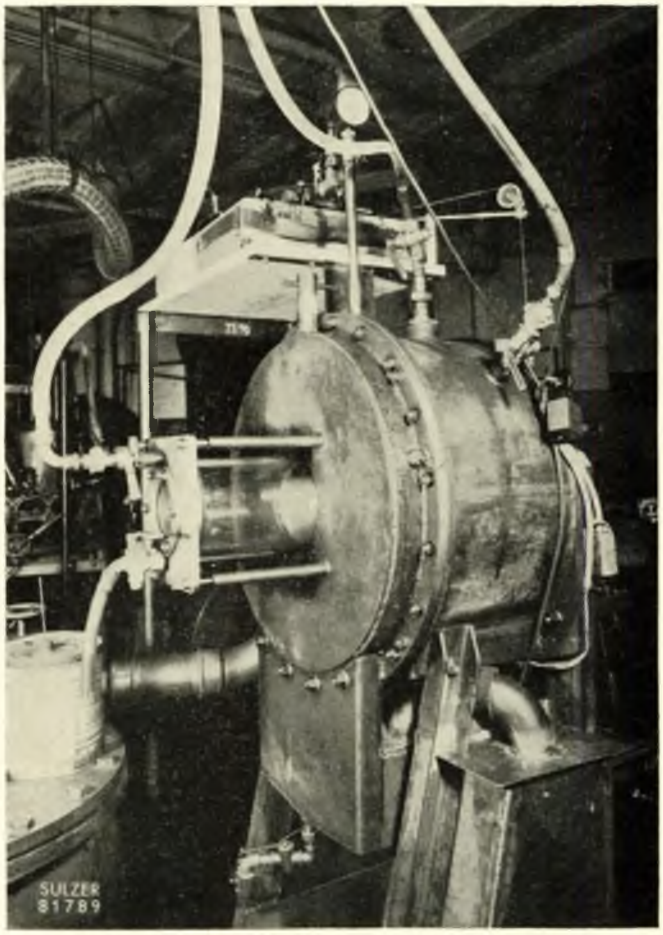


FIG. 9—*Model for cross-scavenging tests*

determined by measuring the purity factor of scavenging and the pressure loss through the ports. A simple mechanism allows this model to reproduce a single scavenging process by moving the piston according to its law of motion over the ports, thus opening and closing them, while the pressurized scavenging medium executes the scavenging process. Instead of scavenge air and combustion gas, the model employs two different liquids—trichlorethylene and alcohol—which give the same conditions with regard to specific gravity and cinematic viscosity as scavenge air and combustion gases during the scavenge process. This substitution furthermore, allows the process to be slowed down considerably, if the Reynolds number is regarded as the valid criterion for the model law. At these speeds the non-stationary scavenging process can either be evaluated visibly or by taking slow-motion moving pictures.

When a single scavenging process is carried out, the specific gravity of the cylinder contents is determined by an areometer, and the quantity and specific gravity of the mixture scavenged out from the cylinder is also measured. In this way an exact determination of the purity factor as a function of the scavenge air ratio is obtained. It is an opinion widely held on intuitive grounds that uniflow scavenging produces much better air purity factors than cross scavenging because the incoming cold scavenging air does not mix with the hot combustion gas in the cylinder. A sharply defined front between scavenging air and exhaust gas is thought to persist during the scavenging process and to displace the exhaust gases into the exhaust pipe. A glance at Fig. 10 will show, however, that a sharp separation between combustion gases and incoming scavenging air is obtained equally well in a good cross-scavenging system. This figure—representing a sequence of phases from a high speed film of a scavenging process—shows that the scavenge flow moves upwards along the cylinder wall without mixing with the remaining cylinder contents, that after reaching the cylinder head it performs a U-turn, and that it finally pushes the remaining exhaust gases out of the cylinder. On the basis of such model tests, the most promising port arrangements were chosen from some 40 configurations investigated and were tried out in engine tests. Fig. 11 shows

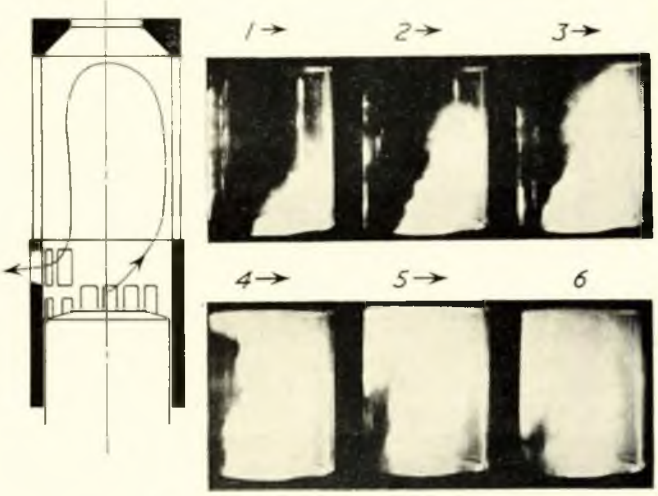


FIG. 10—*High-speed pictures of a scavenge process (model tests)*

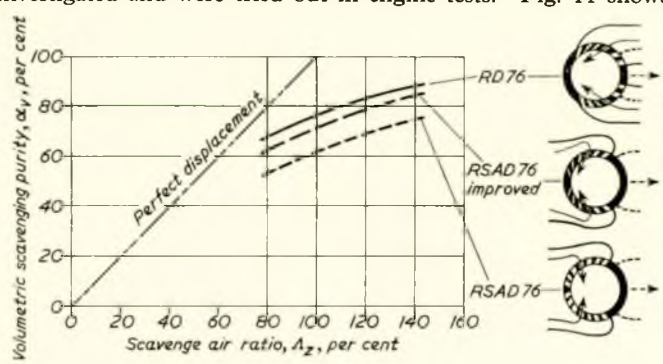


FIG. 11—*Improvement of scavenging port arrangement (model tests)*

at bottom right a port arrangement well proved in former non-supercharged engines, in the centre another version improved in the light of the scavenge model tests, and at the top the still further improved configuration of the new RD engines. The figure demonstrates what a beneficial influence this

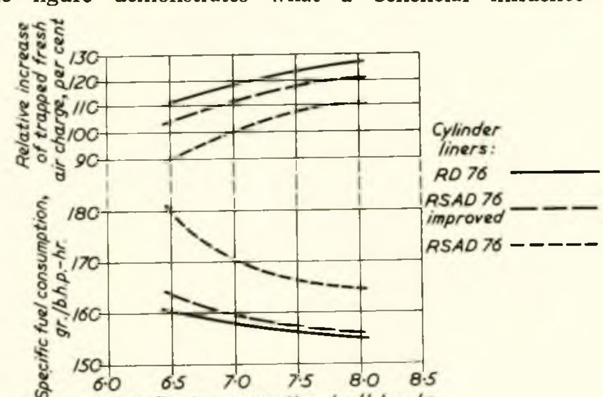


FIG. 12—*Improvement of scavenging port arrangement (engine tests)*

development has had on the air purity factor. Even if the figures determined in the model do not correspond in their absolute values with those obtained on a running engine, it is still possible to compare the results obtained with the various configurations. In any case, a port arrangement giving better results in the model has each time brought about an improvement on the engine. Fig. 12 shows the results obtained on a single cylinder test engine of 760 mm. bore with the same three arrangements as in the preceding figure. It can be seen how the fresh air charge trapped in the cylinder has been increased in the engine and also how the fuel consumption has thereby been lowered, particularly at low scavenging air ratios.

In order to compare the results obtained for cross scavenging with uniflow scavenging, the latter system was also investigated on a corresponding model. The curves in Fig. 13

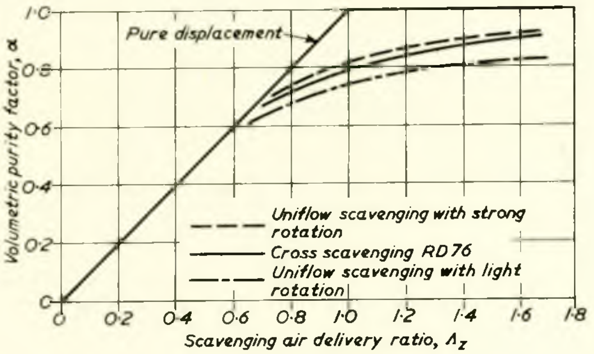


FIG. 13—*Comparative scavenging model tests*

correspond to the best port arrangements for both cross and uniflow scavenging. As will be seen, there is no great difference between the scavenging efficiency of good cross scavenging and good uniflow scavenging. The two curves for uniflow scavenging correspond to two arrangements with a very different swirl, i.e. with a large variation of the tangential component of the scavenging air entering the cylinder. Later tests on a uniflow scavenged engine with a single central fuel valve have shown that the port arrangement corresponding to the upper curve with a strong swirl gives poorer results with regard to combustion and fuel consumption than the arrangement with less swirl, as represented in the lower curve. Of course, the higher efficiency of uniflow scavenging with a strong swirl may be utilized in practice, but it requires special adaptation of the number of fuel valves and their arrangement to the intensity of the swirl.

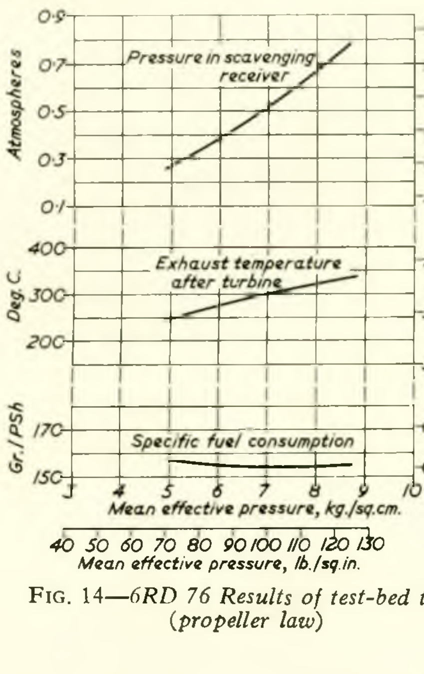


FIG. 14—*6RD 76 Results of test-bed trial (propeller law)*

What has been seen up to this point have in general been laboratory investigations only. It is now intended to show a few results obtained on real engines. Fig. 14 shows the figures for fuel consumption, scavenge pressure and exhaust temperature as obtained with this best port arrangement on a six-cylinder RD76 engine. The fuel curve rises only slightly at overload, indicating that the engine would carry much higher loads. Actually, such overload tests have been run and have proved the extraordinary overload capacity of the RD type, thus confirming in practice the results of the model tests.

The single cylinder RD76 engine has been run for about 600 hours with a b.m.e.p. of 9-10·5 kg./cm.^{2} (128-150lb./sq. in.). Still more convincing, however, are the results shown in

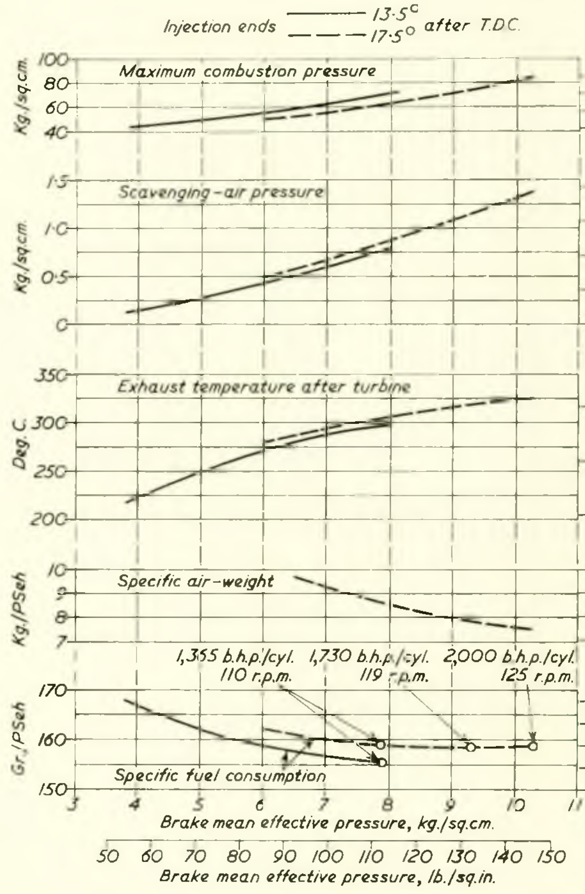


FIG. 15—*High output tests 6RD 76 (propeller-law)*

Fig. 15. These are test results obtained on a standard six-cylinder RD76 engine on which only the turbocharger and the fuel injection equipment had been somewhat adapted. The output of this otherwise standard production engine was raised in an overload test from a nominal 1,500 to 2,000 b.h.p. per cylinder at 125 r.p.m., corresponding to a b.m.e.p. of 10·3 kg./cm.^{2} (146lb./sq. in.). The two sets of curves represent two different injection timings. The one corresponds to the standard setting, whereas in the other—for higher outputs—the injection timing has been retarded in order to reduce the high firing pressures which would have resulted from the type of fuel pump used and the very large fuel quantities injected. If still higher maximum pressures had been admitted, the fuel consumption curve could have been reduced accordingly. In spite of this late injection timing, which increased the fuel consumption and thus the thermal load of the engine, it will be seen that the exhaust temperature is remarkably low. This

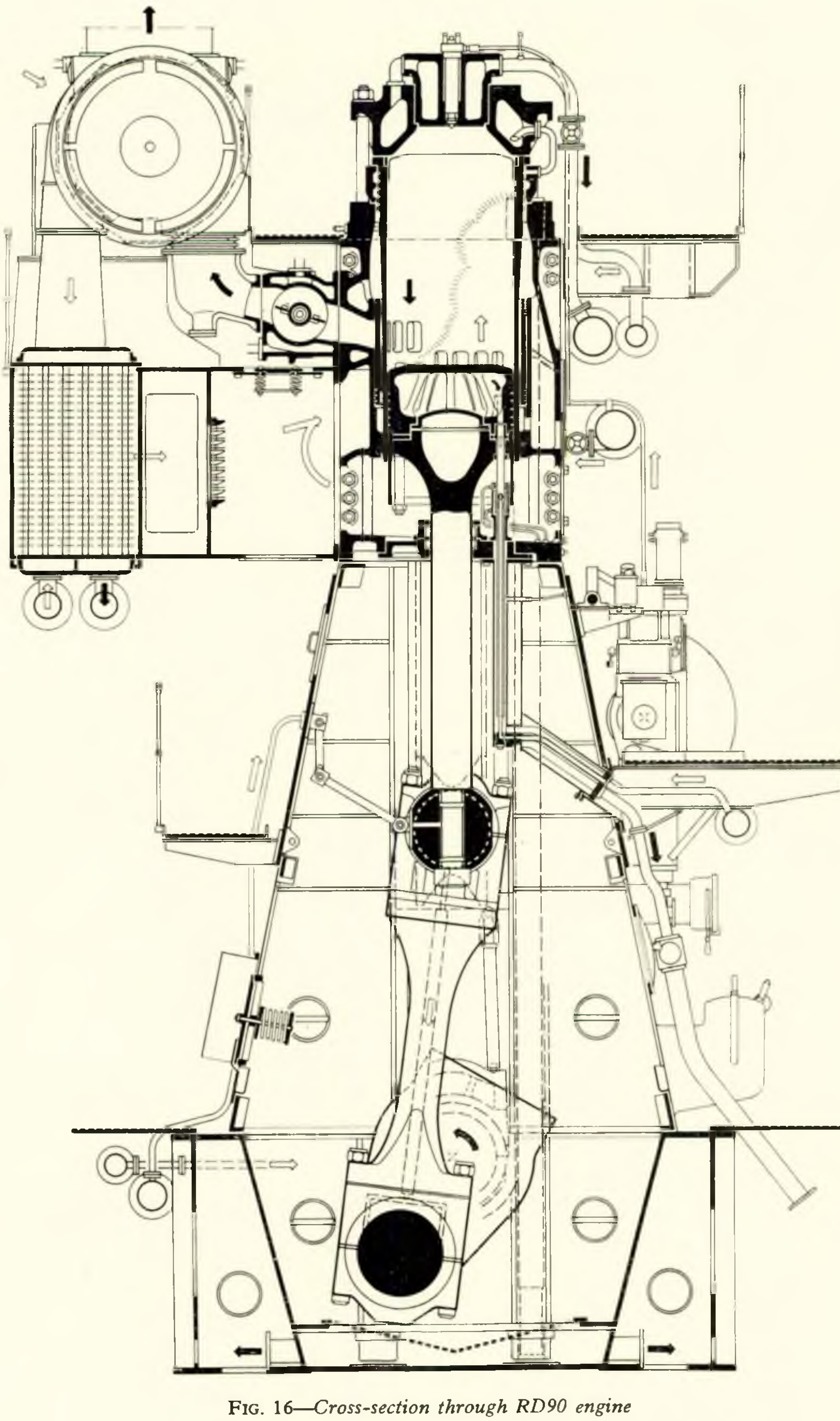


FIG. 16—*Cross-section through RD90 engine*

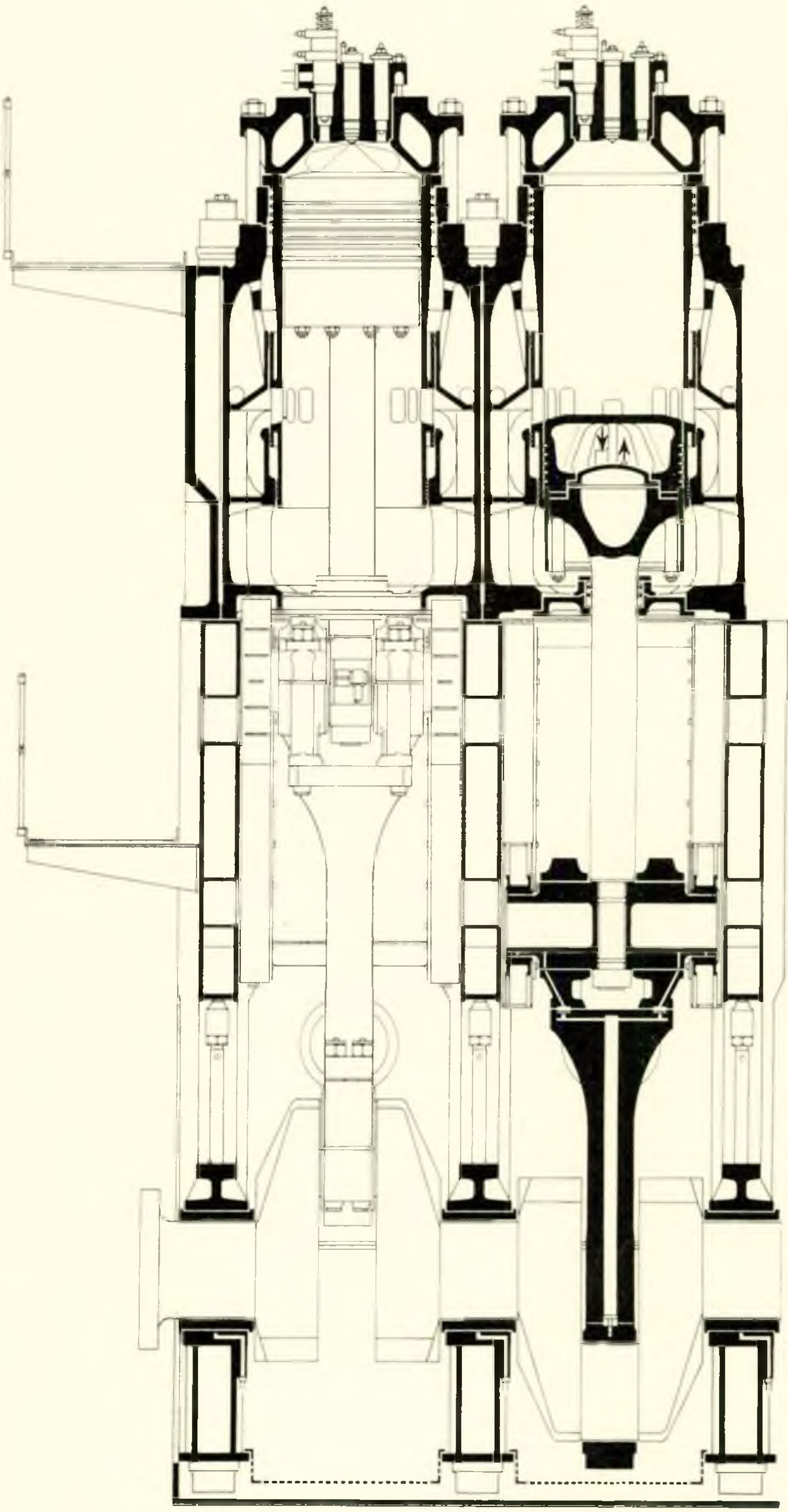


FIG. 17—*Longitudinal section through RD90 engine*

indicates that even at such high loads the engine gets plenty of air—thanks to its efficient turbocharging and scavenging systems—and is thus able to burn the large fuel quantity cleanly and smokelessly. Of course, it is not intended—at least in the near future—to sell engines for such high outputs, as these tests were run only for the purpose of demonstrating the generous overload capacity of the RD type.

As can be seen from Fig. 15, there is a further basic advantage in any supercharged engine. It is the auto-stabilization of the free-running turbocharger obtained without any additional regulating means. If more fuel is injected into the cylinder, the exhaust temperature rises, thus supplying more energy to the exhaust turbine. This in turn runs faster and delivers more air at a higher pressure. Thus a higher load is automatically accompanied by a higher supercharge. This is in contrast to a non-supercharged engine, which has a sharp load limit given by the constant weight of air in the cylinder.

An additional remark should be made regarding the choice of the scavenging system. Longitudinal and cross scavenging are not really two basically different processes, but just two different solutions to the same original problem. Both systems have been improved in the continuous process of mutual competition, thus benefiting the user, i.e. the shipowner, and both show their particular advantages in certain cases. However, personally, the author is convinced that for large marine engines cross scavenging is definitely to be preferred because of its greater simplicity; for—as has been seen from the results shown above—the cross-scavenged engine is certainly comparable on the same level to the much more complicated uniflow engine.

Description of RD Engines

The author described, as examples of real engines, the latest types and some of their components as developed by his company on the basis of the general considerations set forth above, backed by long running experience. The largest of these types to which the following pictures apply is an engine with 900 mm. bore and 1,550 mm. stroke developing 2,000 b.h.p. per cylinder: its cross and longitudinal sections are represented in Figs. 16 and 17. As mentioned at the beginning of this paper, bedplate and frames are welded and the cylinder block is made of cast iron. Scavenge air receiver and turbochargers are arranged in a very compact way, with the air going from the blower to the cylinder by the most direct path, to the exclusion of long ducts, passing on the way through an intercooler incorporated in the receiver. Downstream of the exhaust ports there is a rotary valve, which closes them when the very short piston is at top dead centre and thus prevents the escape of unused scavenge air through the exhaust ports. This arrangement allows the piston to be kept very short, thus reducing the engine height. A short piston will also have better running conditions and thus will require less lubricating oil. Furthermore, this valve throttles the exhaust ports before the scavenge ports are closed by the up-going piston, thus filling the cylinder with fresh air of the maximum possible density. The two-piece cylinder head will be described later in detail. As mentioned before, cylinder and crankcase are completely separated by a diaphragm, with the piston rod passing through it in a stuffing box, in order to prevent corrosive combustion residues from entering the crankcase, where they might cause trouble.

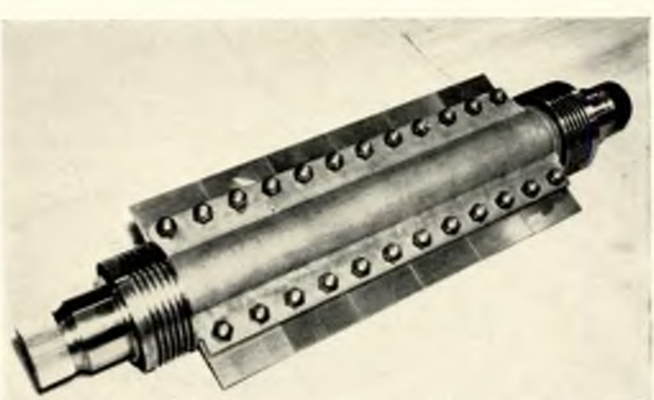


FIG. 18—*Rotating exhaust valve*

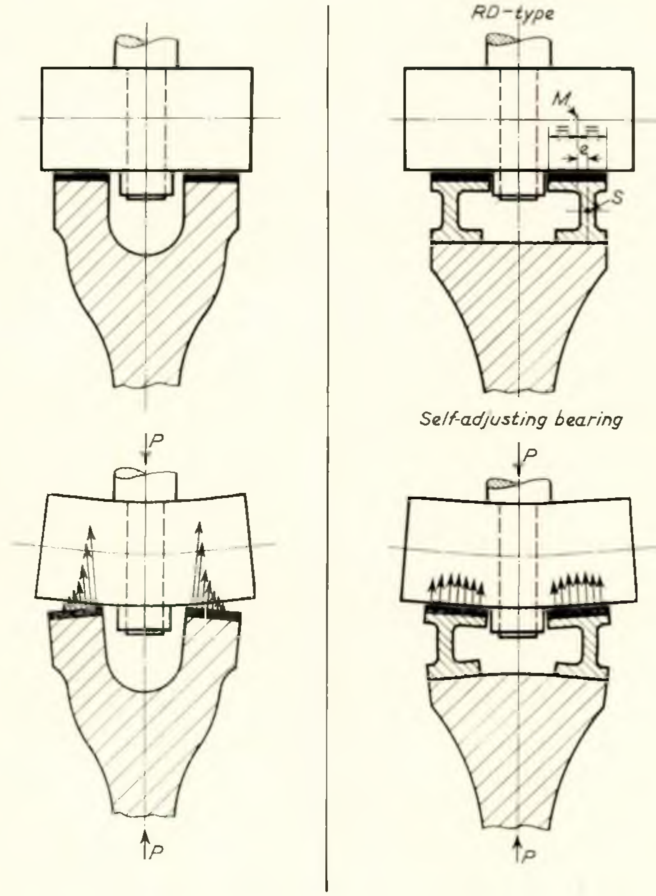


FIG. 19—*Crosshead-bearing*

The fuel pumps are concentrated in one or two blocks, according to the number of cylinders, at about half engine height. Their camshaft is driven by gears and actuates the exhaust valve shaft through a small chain. The exhaust valve itself is very simple and is represented in Fig. 18. As it has only narrow sheet steel edges to seal against its casing, it is very reliable, allowing carbon deposits no chance of accumulating on a wide cylindrical sealing surface, which might block its motion. These sealing edges are sub-divided in their length for the purpose of creating a safety expedient against any pieces of broken piston ring. In the event—which has not occurred yet—of a piece of a broken ring being jammed between casing and exhaust valve, one of these segments would simply bend and the operation of the engine would be impaired only very slightly.

It is widely known that crosshead bearings of large single-acting two-stroke engines have to be designed with particular care in order to ensure trouble-free running. The following pictures give an impression of how carefully this design has been studied by the author's company for their new engines.

Fig. 19 shows on the left side how a crosshead bearing on the usual forked connecting rod is deformed by the combustion load. The fork bends slightly outwards, the ends of the crosshead pin flex upwards, thus producing a load concentration

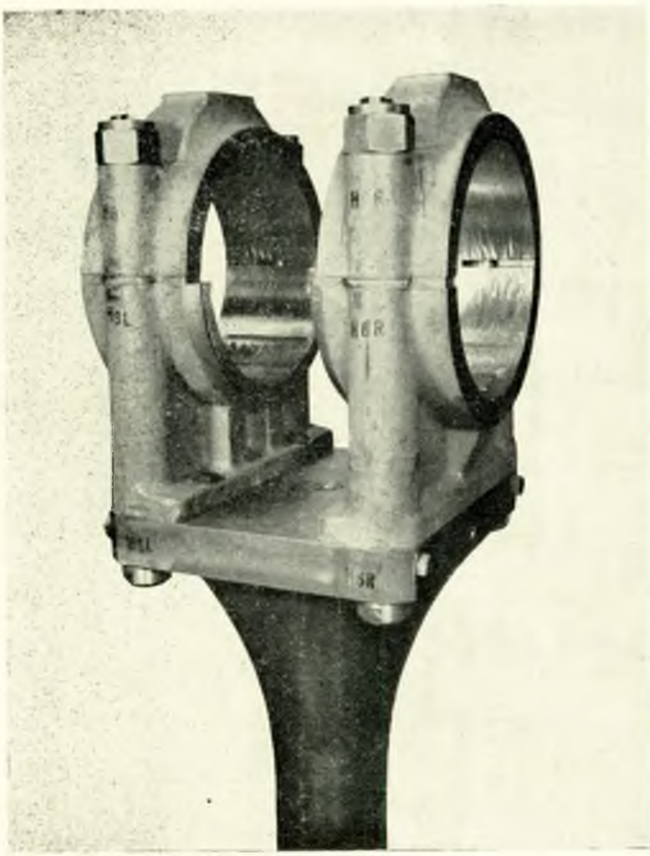


FIG. 20—*Crosshead bearing of RD-type engine*

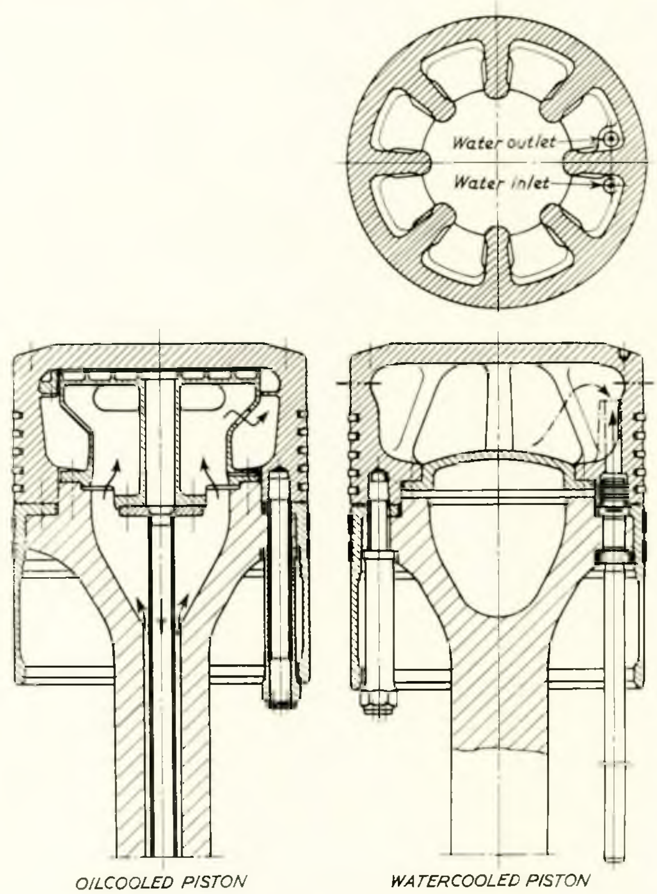


FIG. 21—*Sulzer RD-engine piston design*

on the inner edge of the bearing. This effect may be reduced by stiffening the rod and increasing the pin diameter, or it may be corrected for one particular load by scraping the bearing to make it slightly conical. There has been a radical change made in the design, however, by mounting two independent bearings on the rod. Their supports are displaced relative to the centre line of the bearing surface *M* by a distance *e*, and are so elastic that they bend slightly inwards under load, thus compensating for the upward flexing of the pin, as represented at bottom right of the figure. In this manner, a perfectly uniform load distribution over the bearing width is automatically obtained at all loads. These conditions have not only been theoretically investigated, but have been checked in careful experiments with strain gauges and optical deformation measurements, and the new design has been proved in prolonged service. Fig. 20 shows how simple such a crosshead bearing looks.

With regard to the higher outputs and—still more important at the moment—the larger bores, a few basic considerations had to be kept in view. Every endeavour was made to find really satisfactory solutions to the problems of the continually rising mean effective pressures and the adoption of a bore of 900 mm.

In this connexion a closer examination could be made of the piston, cylinder liner and cylinder head, for which interesting and it is believed convincing designs have been found. Rising mean effective pressures will boost the maximum combustion pressures, and this must be taken into consideration, especially with the large bores. At first sight, it would seem logical to compensate the higher stresses due to the high pressures and the large bore by increased wall thickness. Unfortunately, this problem cannot be solved as simply as that, as increasing wall thickness will raise the very important heat stresses. Means will have to be found of coping on the one hand with the gas loads due to the high firing pressures, even for the largest bore, and on the other hand of keeping the heat stresses within reasonable limits. It appears from incidental experience with various engine makes in the field that with the present highly rated supercharged engines oil cooling of the pistons seems to approach its limits, as under unfavourable conditions coke deposits on the inside and burning-away of the metal on the outside have occurred. Consequently, this experience and extensive studies and measure-

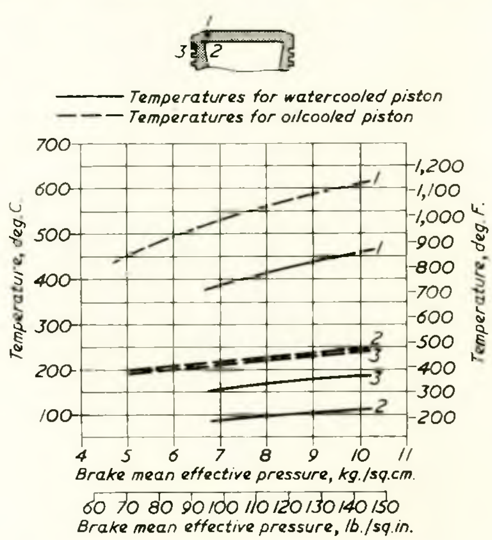


FIG. 22—*Piston temperatures measured on test engine 1 RD 76 —comparison between water and oil cooled piston*

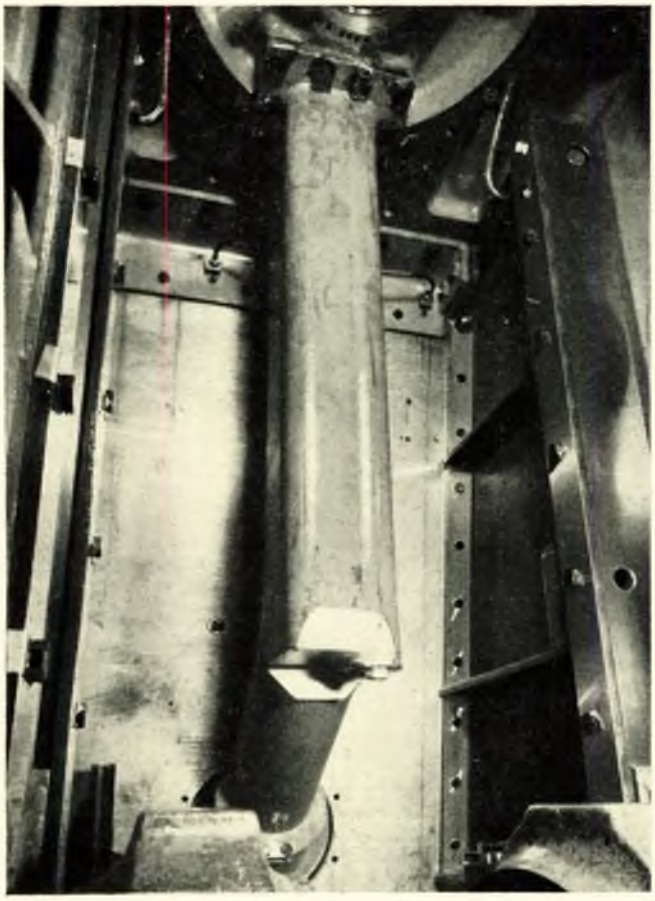


FIG. 23—*Watertight casing in crankcase for piston cooling tubes*

ments on the single cylinder RD76 engine have led to the conclusion that the pistons are better cooled with water. A piston shape (Fig. 21) with internal ribs supporting the crown allows for relatively thin walls, and thus for a reduction of the heat stresses. This shape, proved in service for twenty years in water cooled pistons, is not only particularly favourable with regard to heat stresses; the outside of the piston also remains relatively cool even at high loads, and the danger of burning-away of the metal is precluded. With water cooling there is of course no danger of coke formation in the piston cooling chamber either. Fig. 22 illustrates the great difference between water and oil cooled pistons: the maximum temperature of an oil cooled piston at a mean effective pressure corresponding to an unsupercharged engine is about the same as that of a water cooled piston at twice the load. The very important temperatures above the uppermost piston ring groove are also remarkably low.

However, water cooling with all its advantages could only be used after means had been found of leading the water to and from the piston without the danger of water leaking into the crankcase (Fig. 23). The water is now supplied and led away by telescopic tubes which have no contact with the crankcase whatsoever, as they pass through it in a special, completely watertight enclosure which will catch any leakage water and lead it off. The stuffing box has been designed to prevent effectively any upward passage of water into the air space and any entrance of air—which may sometimes carry traces of oil—into the water system. New wear-resistant sealing elements made of a nylon material are combined with pipes which drain off separately any air leaking from the scavenge space into the stuffing box and any water scraped off the telescopic tubes.

The design of the stuffing box is such that it may be dismantled from the engine with the piston in the top dead centre position without the telescopic tubes being removed. The telescopic tubes are not fixed in the piston but in the flange of the piston rod (Fig. 24). This arrangement has the great advantage that the tubes are automatically aligned, provided that the bores drilled for them in the piston-rod flange are parallel to the piston rod itself. The piston head may now be dismantled and even exchanged without breaking the connexion of the telescopic tubes with the piston rod. This feature will be very handy for overhauling work.

In connexion with the high loads and large bores, the problem of cylinder liner strength also had to be studied closely (Fig. 25). Here again, the liner thickness could not be simply increased according to the gas loads, as the heat stresses would have risen unduly. To increase its resistance to high firing pressures, the liner is restrained by the intermediate steel ring between cylinder jacket and head. This ring surrounds the cold liner with a small clearance and supports it very effectively as soon as the liner expands at the running temperature. Careful measurements on a single cylinder RD76 engine have shown that this support will reduce the cylinder liner stresses by roughly 50 per cent, as shown in Fig. 25; the heat stresses have also been reduced accordingly. In the next picture (Fig. 26) the actual design of this supported liner and its adjoining parts is represented. This illustration also shows the design of the cylinder head, which is extremely simple because—due to the cross-scavenging principle—it does not contain any exhaust valves. Its design was evolved before the last war with the aim of minimizing the heat stresses. It consists of two concentric parts with the joint at the place

FIG. 24—*Dismantling of water cooled piston head*

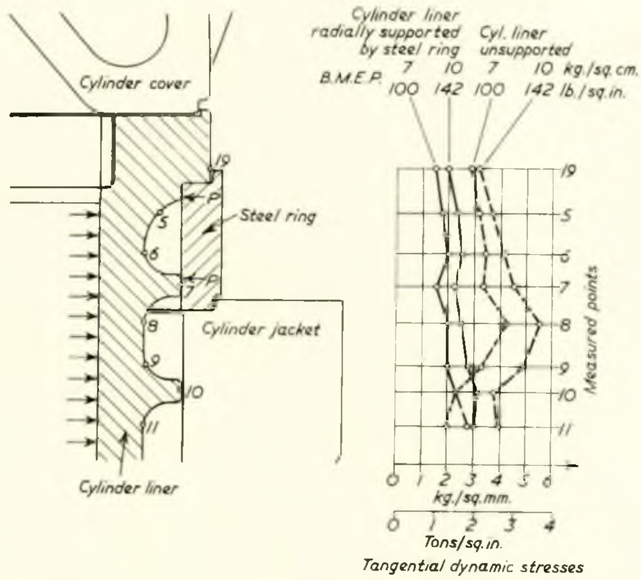


FIG. 25—*Tangential dynamic stresses in the cylinder liner of 1RD 76*

where the single piece design used to develop heat cracks, and in fact, with this new design, no cylinder-head cracks due to heat have been experienced in the last twenty years. The outer part—a steel casting—is symmetrical about the cylinder centre line and is extremely resistant to gas loads on account of its conical combustion chamber wall. Owing to this basically very strong shape, the wall thickness can be kept to a minimum, thus reducing heat stresses. Fuel, starting and safety valves are incorporated in a central insert which is small enough to withstand the gas pressures and heat stresses with ease.

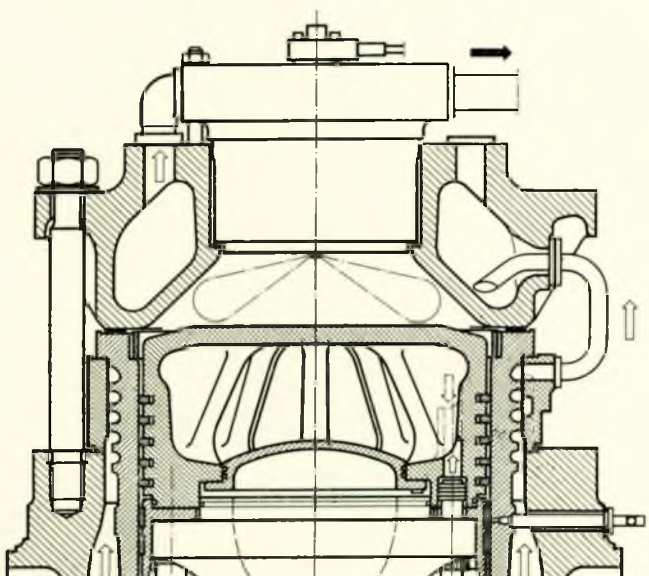


FIG. 26—*Combustion chamber for RD-type engine*

FIG. 27—*8RD 76 engine—exhaust side*

After this description of some details which have been developed in the last few years, Fig. 27 shows the general view of such an engine. In this case, it is an eight-cylinder unit of 760 mm. bore developing 12,000 b.h.p. at 119 r.p.m. The design of the newly developed elements and the basic investigations with regard to scavenging, strength, etc., have been carried out on such a general scale that the basic principles adopted hold good for various dimensions and may be used for engine types of different bores. Thus, a complete range designed on exactly the same principles has originated for outputs and specific weights as shown in Fig. 28.

The RD76, with 1,500 b.h.p. per cylinder, has a power range up to 18,000 b.h.p., and the RD90, with the conservative rating of 2,000 b.h.p. per cylinder, up to 24,000 b.h.p. The two smaller types are intended to complete this range down to 3,000 b.h.p. The lower half of this figure shows the weights per brake horsepower of these types, which range between 32 and 43 kg./b.h.p.

FACILITATION OF OVERHAUL WORK

The author wishes to refer now to a subject which is often neglected in papers on development work, e.g. measures and devices intended to facilitate overhaul work on the engines. The general design—as shown in Fig. 16—has already been simplified as far as possible with an eye to overhaul work. The

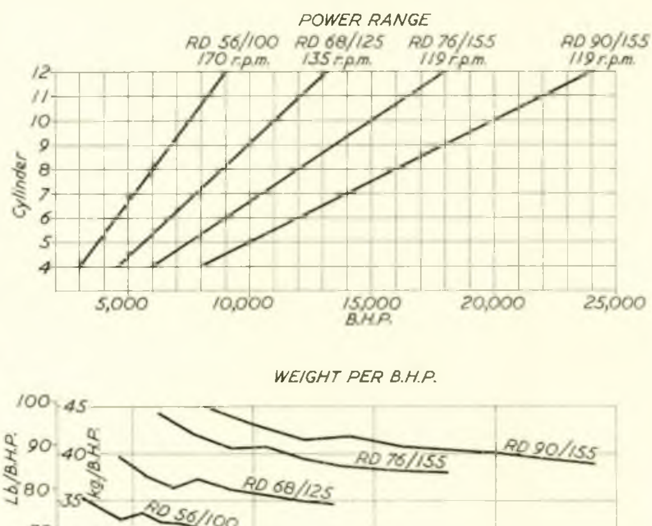


FIG. 28—*Power range and weights of Sulzer RD-type engines*

FIG. 29—*Inspection work inside crankcase*

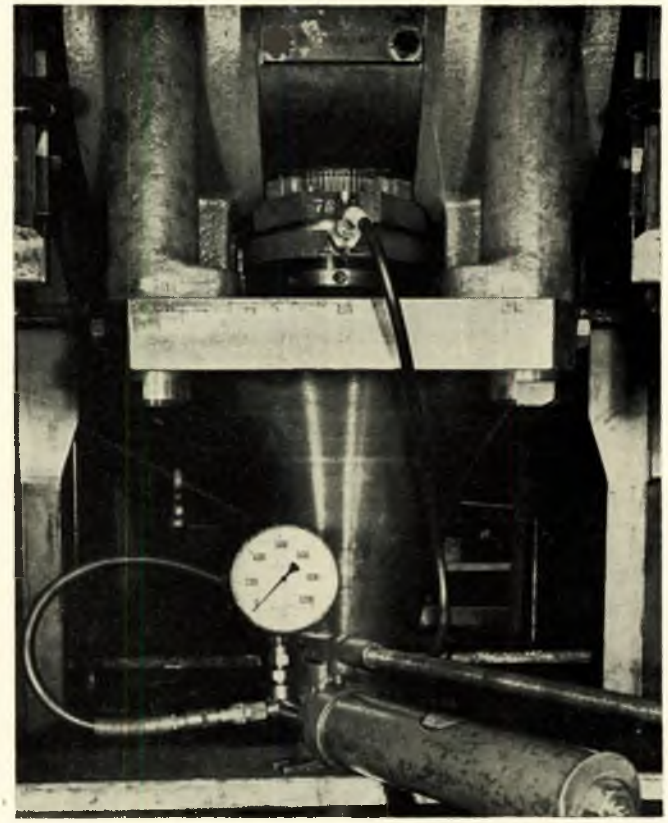


FIG. 30—*Hydraulic stressing of piston rod nut*

turbochargers are placed in a very accessible position on top of the scavenge air receiver. Doors of ample size give comfortable access to the spacious crankcase and the running gear (Fig. 29).

The larger the bore of the engine, the more difficult it becomes to handle various connexions which have to be loosened or tightened during overhauls. For this reason the author's company has designed special hydraulic devices for the most difficult of the connexions. The large tie-rods which have to be pre-tightened to 200 tons are stressed up to this force by a hydraulic pump, whereupon the nut can easily be run up by hand. The piston rod nut in the crosshead is served by a similar device (Fig. 30), which stretches the bolt hydraulically, whereupon the nut may be easily loosened or tightened by hand. The same hydraulic pump is used to tighten the thrust bolts of the main bearing cap (Fig. 31). Here, the hydraulic piston is already incorporated in the bolts as a permanent feature. The pump is equipped with a safety valve which ensures that the parts never can be overstressed even by careless or unskilled personnel. Up to now the tightening of the drive chains by fitters had to be done by "feel". For the drive chains of the rotary exhaust valves used on the new engines a hydraulic tensioning device is utilized, which allows even unskilled personnel to stress the chain correctly (Fig. 32).

DEVELOPMENTS OF ENGINE OUTPUTS AND WEIGHTS

It may now be worth while to stop and cast a short look back on the development of engine outputs in the last fifty years (Fig. 33). This figure shows how the low piston speed and the b.m.e.p. of the old engines restricted their outputs, in spite of large cylinder bores, to the 5,000-10,000 b.h.p. which were usual until quite recently. Even then a few exceptions exceeded these limits, as for instance the SDT76 of the Dutch passenger liner *Oranje* with its somewhat higher engine speed,

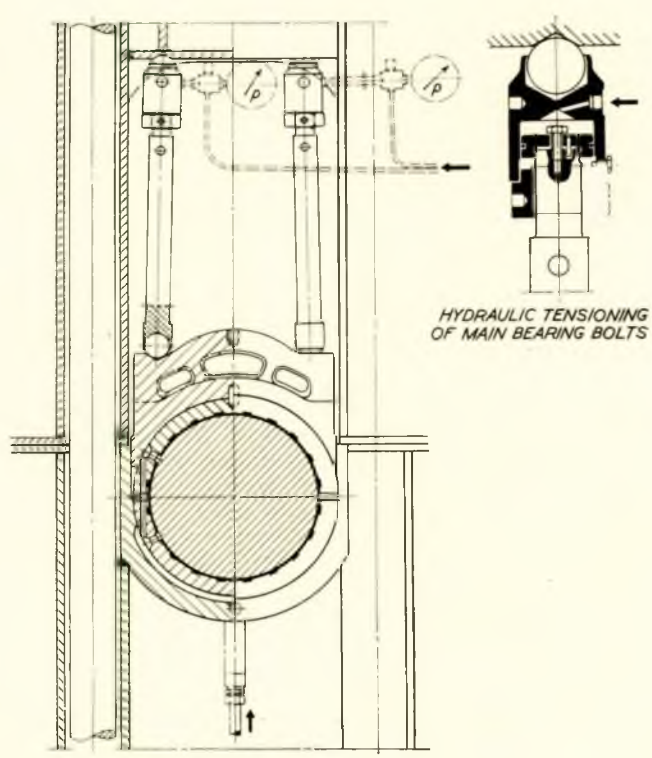


FIG. 31

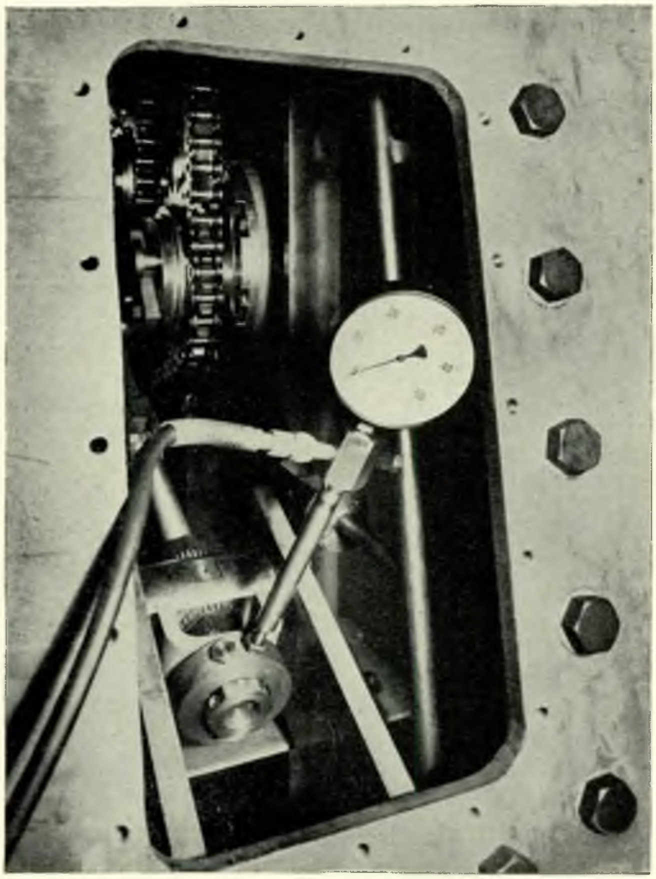


FIG. 32—*Hydraulic tensioning of exhaust valve drive chain*

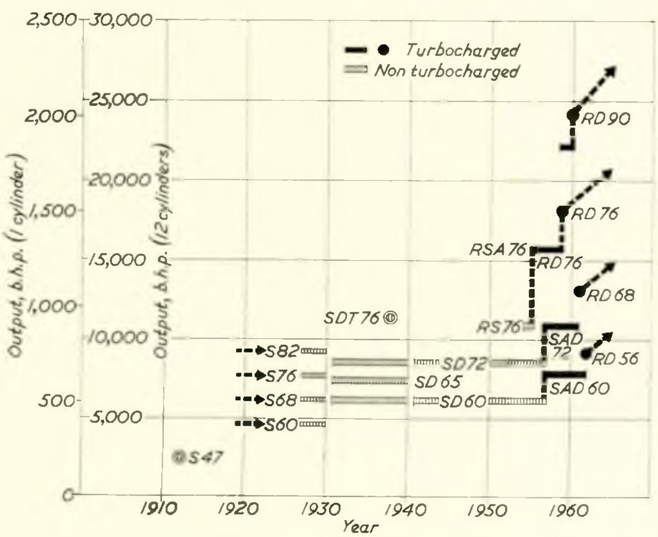


FIG. 33—*Sulzer marine Diesel engines—increase of available engine power*

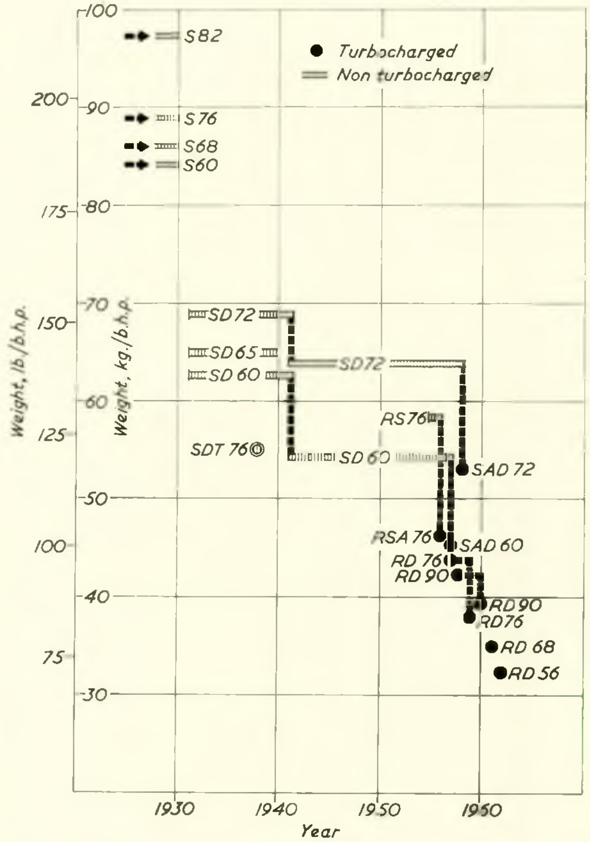


FIG. 34—*Sulzer marine Diesel engines—decrease of engine-weight per b.h.p.*

or the double-acting designs which are now extinct and are therefore not shown in the diagram. It was only the introduction of turbocharging for large two-stroke engines which allowed the output to be extended to 15,000 and later to 18,000 b.h.p., and thus gave access to the range of large tankers previously monopolized by steam turbines. The increase of bores to 900 mm. now extends the power range to 24,000 b.h.p., and no doubt even higher figures will be reached in the not too distant future.

This picture shows the development of the Sulzer engines with which of course the author is primarily familiar. Similar diagrams can be drawn up for the other manufacturers of large engines, as the development of Diesel engines has proceeded with all companies at approximately the same tempo and the outputs have been raised to approximately the same levels.

In conjunction with the power increase, the unit weight of the engines has been reduced (Fig. 34). It will be clearly seen how the weight of engine installations was radically reduced thirty years ago by the introduction of airless injection, which permitted the elimination of the bulky compressor, and has been further cut down more recently by the power increase due to turbocharging. Here again, the figures valid for Sulzer engines will correspond approximately to those of other makes.

Owing to the developments of these last few years the Diesel engine—regardless of its make—has now reached an output twice as high as that available only a few years ago. Now apart from few installations with extremely high outputs, such as those of fast passenger liners, the Diesel is capable of propelling any vessel on the seas.

Discussion in Montreal

WEDNESDAY, 15TH MARCH 1961

MR. T. N. ROSS (Member) noted that the excellent paper they had heard emphasized the phenomenal increase in b.h.p. per cylinder of modern engines. The consequent massiveness of the moving parts prompted him to ask whether any special arrangements were made for survey and overhaul.

The author's remarks under the heading "Welded Design" had been carefully read and, whilst the superiority of the all-welded entablature was neither admitted nor denied, it would be interesting to hear his views on this subject.

The fractures to be found in welded bedplates usually occurred in main cross-members in way of the main bearings and adjacent structure, the coverless type of engine being by no means immune in this respect. The provision of pre-stressed tie-rods relieved the welded bedplate and A-frame structure of all pulsating stresses, produced by the combustion of the fuel, provided the total load in the rods due to pre-stressing was always in excess of the total load on the cylinder head. Due to tension in the tie-rods, the bending moment in the main bearing cross-members was in the opposite direction to the fluctuating bending moment of engines without tie-rods and was constant, provided the rods were located close to the centre line of the crankshaft. Mr. Ross thought that this factor would be most important in relieving the stresses, due to combustion, imposed upon the A-frames and surrounding structure. It was his experience that fractures were seldom found in these parts and he would appreciate the author's views on this. The statement, that fabricated steel bedplates were stiffer, due to a higher modulus of elasticity in the steel compared to cast iron, was not understood. In regard to stiffness, while it was agreed that a properly designed steel box form could be stiffer than a solid cast iron selection, would not a properly designed cast iron box form be even stiffer? The author's opinion on the vulnerability of fabricated steel bedplates in the event of the ship grounding would also be appreciated.

Regarding the self-adjusting type of crosshead described, would not a spherical shell type of bearing serve the same purpose?

In concluding, the author was to be congratulated for a most interesting and informative paper.

MR. J. D'OTTAVIO (Member) found the paper extremely absorbing because of the pains taken by the author to include only that which was considered new. Since the innovations in design described by Mr. Kilchenmann seemed to be justified by success, it was difficult to take issue of any of them. There were however certain points about which he would appreciate additional information.

Reference had been made to annealing of the cross-members of the bedplate. Was this the only annealing that was carried out or were the frames and complete bedplate eventually annealed also? Perhaps the term anneal should have read "stress relieved". Also, would the author care to say whether the weld in the A-frame between the athwartship plate and the fore and aft toe plate was a full penetration weld.

It had been stated on page 107 that the engines could produce sufficient power without turbo-blowers to proceed at 70 per cent speed. This was about 35 per cent of the total power. Normally about 50 per cent of excess cylinder volume was required for good scavenging and the volume on the underside of the piston was only equal to the volume of the working cylinder. Would not such operation result in excessive soot accumulating in the ports and chambers? Furthermore was not the possiblity of fire in the scavenge chamber under the piston, with ensuing hot diaphragm and crankcase explosion, increased when burning heavy fuel oil? The unburnt products dripping into this space were not under constant observation, as would be the case if the underside of the cylinder were open.

Must he also conclude from the remarks on page 107 that the supporters of port scavenging engines had given up hope of ever relying on the turbo-chargers for the supply of scavenging air, under all operating conditions?

The hydraulic pump developed for the piston rod nut must have been of ingenious design considering the limited space available. Perhaps details of this equipment could be given.

Referring to piston design, Purday* had stated: "There seems to be no doubt that internal ribs as used in the past are conducive to cracking both of cooled and uncooled pistons. They are therefore omitted in most modern designs". In view of the author's claims to success, the diversity of opinion was most interesting. Would the author be prepared to comment on the reasons for success and also upon the means taken to force the water to sweep all around the piston before returning to the outlet connexion?

Because the temperature stresses were proportional to thickness and heat flux, it followed that a thinner wall was subject to lower stress from the same flux. With the extremely low temperature reached with water, however, the flux was increased and one wondered whether the end result justified the use of water. It would be of interest then to hear from the author on the subject of temperature cracks.

Temperature cracks were supposed to be due to the difference in temperature between the hot and cold side of the wall. The hot side was in compression and yet that was the side on which the crack appeared, became wider when the piston cooled down and almost disappeared when the piston heated up. These observations did not seem to agree with theory.

MR. R. BAIRD (Member) thought that the paper was very timely, considering the new mammoth Diesels now available to shipowners. It was notable that the author's company was among those successfully developing engines producing 24,000 b.h.p.

It was interesting to find that tie-rods were back again, because they had often been a source of trouble in earlier engines. This arrangement and the pre-stressing technique

* Purday, H. F. P. 1948. "Diesel Engine Design".

used with the rods had so far resulted in absence of cracks in welded frames and bedplates.

Would the author give some information regarding the crankshaft of the RD90 engines, presently rated at 24,000 b.h.p. From the cross-section of the engine on page 110, the diameter of these crankshafts would seem to be about 25in. What deflexion was allowed? Also, had it been necessary to incorporate any special design features, considering the power transmitted and that broken shafts had not been uncommon in the intermediate power range.

The rotary exhaust valve appeared to be of simple design. However, efficient lubrication of the bearings might present a problem. Considering that the valve was exposed to relatively high temperatures, did it require special lubricating arrangements?

The partly dismantled piston on page 114 indicated sound practical design from an overhauling point of view. It seemed that carrier rings or inserts were not used in the way of the grooves and this might mean subsequent machining after wear and also the use of oversize rings. Would the author care to elaborate on this arrangement and the material used in construction?

Reference on page 117 to the extention of the power range beyond 24,000 b.h.p. gave rise to speculation. Presumably there was still a reserve of power available. There was, however, a limit to the power which the propeller could absorb and still maintain a satisfactory propulsive efficiency, assuming that the revolutions were maintained at present figures. Would that be a limiting factor in further radical increases in the power range of large Diesel engines?

MR. S. G. WILKINSON (Member) complimented the author on his paper and suggested that the design of the top and main bearings gave rise to some interesting features.

The heavy inner vertical ribs were omitted in Fig. 19, and appeared rather indistinct in Fig. 20. Were they attached to the inner side of the crosshead bearing shell? Perhaps the author would give some figures as to the upward flexing of the pins and comment on service experience.

The lubricating oil system shown in Fig. 16 had two supply mains and he wondered if there was a separate pump for the crossheads. If not, what was the lubricating oil pressure at this point?

Lubricating oil entry to the bottom part of the main bearings indicated circular ports providing passage through vertical drilling to the top. Did this remove scraper action or interference from the distance pieces? Regarding the securing of the main bearing caps, he suggested the method of dowelling, to true the two halves, might be questioned.

Even though the crosshead bolts were assisted by locking screws at the inner flange, they would appear to be extremely long and might tend to become ill fitting in time.

MR. WALTER P. GRAHAM (Member) said that the author had presented a most informative paper, one on which it was difficult to comment, especially when it described improved design and development by an experienced engine builder. Consequently, he would confine his remarks to questions for the purpose of obtaining information.

Would Mr. Kilchenmann care to express an opinion regarding the advantages or disadvantages of using chrome-plated liners or piston rings?

In the case of untreated liners and piston rings, would the author recommend that the piston rings be either harder or softer than the liner material?

Did the author hold any particular views on piston rings made with non-ferrous metal insert?

MR. R. BOYCE (Member), whose paper was read by Mr. D. K. Nicholson (Associate Member), considered Mr. Kilchenmann's paper to be a valuable contribution to the knowledge of up-to-date design in the Diesel engine field, for which he was to be congratulated.

With the Sulzer arrangement of individual cylinder blocks for each unit, the top nuts of the tie-rods had to bear on two adjacent blocks as shown in Fig. 17. In very long engines of 12 cylinders and 900 mm. bore a relative amount of fore and aft displacement, due to longitudinal flexing under bad weather and unfavourable loading conditions, could not be avoided. This would have an adverse effect on the contact face between the nut and the block, tending to reduce the tension originally imposed in the rods. Would the author be kind enough to comment upon this aspect of the tie-rod arrangement?

The pre-stressing of the bolts holding down the main bearing keeps could be followed in Fig. 31. However, it would be appreciated if the procedure used in the tie-rods could be described. Information with regard to the material characteristics and fatigue endurance limits of the tie-rods would be welcomed.

No reference was made in the paper to utilization of the exhaust gases after leaving the turbo-charger. Was there a possibility of improving the overall thermal efficiency of the plant by installing an exhaust gas boiler?

The author had indicated that the rotary exhaust valve would remain operative if slight damage occurred due to pieces of broken piston rings passing through it. Such a mishap would no doubt impair the valve tightness and give exhaust gases from other cylinders access to the air chamber of the defective cylinder. Under adverse circumstances, such as mist in the scavenge air, this could lead to an explosion in the chamber and possibly in the common scavenge manifold. Would the author indicate any additional safeguards apart from the rotary exhaust valve that would eliminate this risk?

Author's Reply

Mr. Kilchenmann wished to thank those present for the intense interest shown in the paper and for the questions which had been asked.

Dealing first with Mr. Ross' question as to the provision of heavy lifting gear, the author confirmed that suitable gear and specially designed tools were provided to facilitate dismantling and handling of heavy components. Anchorage points were also located within the crankcase to ensure the gear was disposed to the best advantage.

Referring to the problem of cracking in welded components of engines, this had been experienced many years ago in early designs of small high-speed locomotive engines. At that time, quite a lot of cracking was encountered, but solutions had been found and he could now say that these engines were operating satisfactorily and were free from this problem. On the large marine Diesel the change to welded design took place only about ten years ago. Prior to this, cast iron was used and since the change to welded design, they had not experienced cracking. It was agreed that it was not only the engine with a cylinder cover which suffered from cracks in the welded parts. The same trouble had been experienced in opposed piston engines. In one case this had led to the adoption of a double wall construction, in way of the bearing saddle, where previously a single wall subsequently reinforced with ribs had been used.

It was absolutely correct, as Mr. Ross had stated, that the tie-rods were so tensioned that even during the period of maximum firing pressure, the whole welded structure was under compression. The author also agreed that cracking was mainly found in the bedplate rather than the column of the A-frame. He could think of one firm that had built engines with tie-rods and then modified the design to exclude the rods. Subsequent cracking caused so much trouble that the rods were re-introduced. Fractures, in the columns of another company's design without rods, had also been quite common. For these reasons, the author's company had decided, when the change to welded construction was made, to introduce tie-rods. Tie-rods had not been in use when cast iron was used, but they were absolutely convinced that the absence of cracks in his company's engines was due to the use of these rods.

Referring to the modulus of elasticity of steel and cast iron, considering two similar pieces, one of steel and one of cast iron, and applying forces of equal magnitude upon them, the cast iron would deflect more than the steel. If cast iron and steel pieces of the same area were exposed to the same tension, then the E for steel being greater, the elongation will be smaller for the steel piece.

It would be possible to make cast iron parts with less deformation than steel parts, but much more material would be needed. With the double wall arrangement, which from a design point was very rigid, they obtained a much stiffer part with less material than if cast iron was used. This not only proved theoretically true, but also practically. The RD90 engine with a 35½in. bore and welded construction, stood like a rock while running on the test bed. It did not move at all.

Regarding the necessity of stiffer foundations in ships for welded bedplates, Mr. Kilchenmann did not think this was required, inasmuch as the theory used in construction was to make the bedplate itself very stiff. The very high and rigid longitudinal side beams took the deformation and did not rely upon the foundation. With this design no special reinforcement built up on the double bottom was asked for.

His company did not favour spherical bearing shells and he noted that another manufacturer had now omitted them in his new engines.

Mr. D'Ottavio had raised the question of annealing or stress relieving the bedplate cross-members. The author confessed that the difference was not understood. After further explanation from Mr. D'Ottavio, the author answered by saying it was stress relieving. Columns and longitudinal beams were not stress relieved, the cross-members only being so treated. They were the parts which really had to bear the forces from the bearings. If tie-rods were not fitted and the columns took the load, then it would be necessary to stress relieve the whole structure.

Regarding welds in the engine structure, a special full penetration weld was employed at highly stressed parts.

Referring to operation at 70 per cent of ship speed with scavenge blowers inoperative, where a figure of 50 per cent excess air to cylinder volume was mentioned, the author would say that this figure no longer applied. Since the advent of supercharging it was preferable to have the air blown in the cylinder instead of having it blown through. This was shown in the high speed film where air entered and pushed out the exhaust gases without too much mixing. What was needed if mix-up could be avoided completely was about the volume of the cylinder. It was for this reason that so much work had been carried out with 40 port configurations in the scavenge model. The best ones had been adapted to the engine. At 70 per cent ship speed corresponding to about 35 per cent full load, the amount of fuel injected was small and combustion was sufficiently good to avoid the danger of smoke and dirt in the cylinder.

Commenting on the question of explosion in the space below the piston, the author could say that the separation in the scavenge receiver had been pointed out and that when the piston came down, the scavenge pressure was increased to nearly double that supplied by the blower. At the moment when the piston opened the scavenge ports, the pressure on the scavenge side was as high as that in the cylinder, so that there was no flow back to the scavenge space. This had two advantages, firstly, fire had never occurred, because there was no flow of exhaust gas to this space; secondly the scavenge ports remained absolutely clean. He could recall the old double row scavenge system, with non-return valves and the difficult job of cleaning them.

Considering the question of relying upon turbo-charge only for the supply of scavenge air under all operating conditions, the author stated that his company had, in the test stages, three experimental engines. One of these had a uniflow scavenge and another a cross scavenge system. The uniflow engine was able to run at 10 per cent full speed without assistance, but it had been necessary to shift the camshaft to open the exhaust valves earlier under certain speeds. There was another arrangement upon which they were working but it was a little early yet to predict the results in this case.

With regard to the hydraulic pump developed for prestressing the piston rod crosshead nut, the author went on to give a diagramatic sketch of the arrangement.

With regard to piston design, the author said that the solution of having internal ribs was not possible with oil cooled pistons because of the need for smooth machined surfaces; the ribs, of course could not be machined. The arrangement of the water cooling system was similar to a cocktail shaker, the inlet was lower than the outlet and, due to the action of the piston, water was freely splashed around the internal surfaces.

The author went on to describe by means of the blackboard, some of the design principles of the piston with particular reference to heat stresses and avoidance of subsequent cracks.

Mr. Baird had mentioned that tie-rods had returned, after having disappeared. The author had commented previously on his company's reasons for re-introducing them and he could only add that experience had fully justified this decision.

The crankshafts used in the RD90 engine had journals of about 25in. diameter and no special manufacturing methods were required. Similar construction applied, as in smaller semi-built up shafts, two webs and a crankpin making a solid piece. Journals were shrunk in and the shaft was produced under Lloyd's Rules. Problems of a physical nature were the same as in smaller shafts and could be calculated. Bearing pressures were the same with this engine, as with the smaller engine and the shrink fits were quite capable of transmitting the required torque.

As to the question of rotary exhaust valve bearings and overheating, these were kept relatively cool, partly due to the main body of the valve being hollow. The bearings were situated at the outer ends of the shaft and were well lubricated, so that excessive heat was avoided.

Turning to the carrier rings on the piston, the lower landing was usually chrome-plated and in some models a special shaped carrier ring was used. The author was not fully convinced as to which was the most satisfactory solution and experiments were still being carried out. Material used in construction of pistons was an alloyed cast steel.

Regarding the maximum power available from the RD90 engine, 25,000 h.p. could be attained and even exceeded. This should suffice, for some time, for the present day freighter and tanker. However, the advantage of high power lay not only in extending the power range. It would be possible to reduce the number of cylinders for a certain power; a 6 cylinder engine could be used, instead of an 8 cylinder engine.

Concerning propeller speeds 115 r.p.m. was quite a high speed at which to absorb 30,000 h.p. and a small loss of efficiency must be accepted.

Mr. Wilkinson was not alone in his criticism of the crosshead section in Fig. 19 and the photograph in Fig. 20; other gentlemen had commented on this. In the photograph there were vertical ribs and in the section there were none. The section was intended to show the deformation and principle, not the design. The author then explained with the aid of slides, the design of a crosshead bearing working at a firing pressure of 76 atm. This was a much higher pressure than had been used up to some five years ago. The bearing shell was relatively elastic so that ribs were required for support. They had changed the design many times, until the best conditions for deformation and deflexion had been found. Deformation was still inwards with the bearing loaded and from practical experience the design had proved satisfactory. The crosshead pins were relatively thick and deflexion was very small.

The cross-section of the engine, Fig. 16, showed two oil inlet lines, one leading oil to the crosshead and the other to the main crankshaft bearings. Supply for each inlet was from the same pump, the required pressure at the crosshead being $3\frac{1}{2}$ atm. From the crosshead bearings oil passed down to the bottom end, through a hole in the connecting rod.

Bearing halves of the crankshaft were identical and interchangeable, hence the oil and dowel holes in each half. Distance pieces did not touch the crankshaft, so that it was not necessary to have a separate oil piece for the top half-bearing. The bottom ends and main bearings had oil pockets at a high level and were scraped, with side clearance, so that by rotation of the shaft, oil was taken in.

It was mentioned that the crosshead bolts were extremely long. As the question was understood by the author, it was for this reason that fear had been expressed that they might loosen because of their length. Commenting on this, the author would say: "The longer the better, if they were long they had more elongation and when tightened kept their tension much longer". However, if Mr. Wilkinson meant by his query that these bolts might become misaligned or bent in time, he should have considered that after the bearing cap had been dismantled, only a very short length of the bolt was protruding above the main bearing body, in which they were guided over three-quarters of their length.

Mr. Graham had brought up the question of chrome plated liners. Requests for such liners were received by the author's company and they suggested, if others were prepared to pay, they were prepared to supply. The cost of chrome plating was about the same as the liner. With the newer lubricating oils, wear in the ordinary liner was about ·002 to ·004in. per 1,000 running hours. Up to ·004in. was considered good and up to ·008in. was acceptable. Wear had been so reduced, with the adoption of the newer lubricating oils, that the author's company would not consider chrome plating to be really necessary. Their larger engines had rarely used chrome plating, but it was used on some of the smaller engines.

The author would suggest that where chrome plated rings were not used, then the ring material should be a little softer than the liner. Rings were easier and cheaper to replace than liners.

Soft metal inserts had been favoured for a long time, but they were no longer used because the author's company had found they had a tendency to break more easily than ordinary rings. Inserts were wonderful for running-in purposes. To help running in, in some cases, a thread was cut on a ring, copper-plated and then built up with copper.

Mr. Boyce had asked for comments regarding flexing, in relation to the tie-rods. Tie-rods were fitted between two blocks, the blocks were bolted together longitudinally and there was no relative movement.

The procedure for pre-stressing the rods was to tighten two rods at the same time, starting at one end and going through the engine, setting it to half tension. This method was then repeated, increasing it up to the full tension. This ensured equal tensioning of all tie-rods.

The rods were manufactured from Siemens-Martin steel, having approximately similar qualities to that used in the engine crankshaft.

Could an exhaust gas boiler be used? To this the author would say yes. However, back pressure at the turbine was limited to 300 mm., about 12in. of water. This was the back pressure specified on the test bed.

Referring to the rotary exhaust valve and blow through of broken piston ring pieces. If a valve blade were bent and no longer tight, then when the next cylinder exhausted, the escaping gases would flow into the scavenge space under the piston, but only for an extremely short period. At all other times the pressure in the scavenge side was higher and scavenge air flowing out would mean that a wave or cloud of exhaust gas would come in and pass out again immediately.

Discussion in Vancouver

FRIDAY, 17TH MARCH 1961

MR. L. L. LAWRIE (Member) opening the discussion, asked several questions:

1) In the construction of the bedplate, was it stress relieved after welding in one piece, or was each cross-member annealed separately, as stated on page 105 of the paper?
2) One would take from the paper that the tie-rods were completely encased in the A-frame and bedplate. Was there any possibility of corrosion taking place to the tie-rods? Was there also a period after which it was necessary to pull the tie-rods for examination?
3) What was the maximum wear allowed before cylinder liners required renewing and what would be the average life of a liner running approximately 28 to 30 thousand hours per year at normal full power?
4) What was considered an average mileage or period of time running at normal full power between cylinder liner inspections?

Mr. Kilchenmann replied as follows:

1) Only the cross-members were stress relieved. The heavy bearing loads were taken up by the through tie-rods. The structure outside of them was considered to be loaded only by small remaining forces.
2) The tie-rods were completely enclosed except for drain holes. There was no experience to date of tie-rod failures and there was no experience of corrosion.
3) The life of a cylinder liner should approximate 50,000 running hours. Wear measurements per 1,000 hours should not exceed ·008in. Maximum wear in a 30in. diameter liner should not be more than 5 mm.
4) In a recent case pistons had been pulled after about 9,000 running hours and found in perfect order, all rings being used again. He suggested 6,000-10,000 running hours between inspections.

MR. J. A. STEWART (Member) asked if any trouble had been experienced with turbo-blowers.

The author said that considerable trouble had been experienced with some blowers in the past, due to relatively great blade length, resulting in blade vibration. He explained that this had now been cured by the use of dampening wires.

MR. D. B. ROSS (Member) wanted to know if there had been very much change in metals, due to the higher pressures and temperatures being experienced in modern Diesel engines.

Mr. Kilchenmann replied that, in the case of his firm, over the years, piston crowns and heads had been changed over to cast steel. He had found it better to use water cooling of pistons to avoid the introduction of fancy and expensive alloys.

MR. F. DALGARNO asked if heavy fuel could be used with the Sulzer engine, in ships employed on coastal runs, where manœuvring of the main engines was very frequent.

Mr. Kilchenmann's reply was that heavy fuel was used for such purposes, but care should be taken to use the equipment supplied for keeping the fuel at correct temperature.

MR. R. W. BROWN (Member) then put the following questions:

1) Were there any Sulzer trunk piston engines using heavy fuel?
2) Was it usual practice to dowel piston rings or let them run free?

On these two points the author said that in the smaller range, trunk piston engine tests were being conducted using heavy oil. He pointed out that in the case of four-cycle engines, the exhaust valves were not too suitable for use with heavy oil. The piston rings were left free.

MR. T. TAYLOR (Member) said that as engines became larger in power and bore, with increased loading of bearings, it would appear to follow that the deflexions in these units would increase. In this respect he would like to ask the author if any thought had been given to replacing the conventional crosshead, as they knew it, with a central top end bearing, which would transmit the piston load directly down the piston rod to the connecting rod and thence to the crankshaft. To him this would appear to be a feasible possibility with the present day single acting Diesel engine and he would welcome the author's remarks about this.

Mr. Kilchenmann said that considerable research had been conducted around this interesting idea. Small models had been operating on the test beds, utilizing a ball connexion at the top of the connecting rod, together with a rotating piston. He explained that this had, to date, no practical use with respect to the larger engines.

MR. J. D. G. KINVIG (Member) asked if there was any difference in the proportion of bearing surface between top and bottom ends, resulting in difference in bearing loads.

Mr. Kilchenmann replied that the bearing load per square cm. was similar in all cases, resulting in equal bearing loads.

Discussion in Toronto

Thursday, 22nd March 1961

Mr. J. H. Edlund (Associate Member) said that he was a great protagonist of the smaller Diesel engines and it occurred to him that development costs and any errors, which might occur due to development of these very large engines, must be extremely expensive. He wondered whether the increased powers and decreased weights required would not be better obtained by multiple Diesel engines running at higher speeds, operating through reverse reduction gears, a method of propulsion extremely familiar in Canada.

Mr. Kilchenmann replied that his company had produced many installations of this kind, the largest one being for the Dutch passenger ship, *Willem Ruys*. There were eight engines, totalling about 45,000 h.p., driving two propellers through electro magnetic couplings and gears. A central control stand was provided, to permit each engine to be started independently, or all at once; also each engine could be separately stopped and the load varied for each engine, separately or all at once. The vessel was a classical example of a geared installation. In spite of the fact that the owner was very satisfied with this installation, the author believed that, if such a vessel had to be built again, direct drive engines would be preferred; since instead of having 64 cylinders, there would only be two engines totalling 20 or 24 cylinders altogether. It seemed to the author that having 64 cylinders was a bit of a handicap, as there were too many moving parts.

Mr. Kilchenmann then quoted from a sheet showing the difference in this respect for a vessel of 12,000 h.p. Taking a supercharged engine of 750 r.p.m., there would be between 100 and 120 cylinders, compared with 6 cylinders each developing 2,000 h.p. with a direct drive engine. The multiple engine consisted of many more pistons, cylinders, connecting rods, etc.: this being the main reason why, in Europe, the ship owners preferred the direct drive. Sulzer Bros. had engined about ten large ships with two engines, connected to one gear case, driving one propeller. The owners of these ships were also satisfied, but at the present time, one of them was building new ships and had ordered slow speed direct drive Sulzer engines, because the installation was simpler, not only from the point of view of multiple cylinders, but the whole engine layout was much simpler and cheaper.

Mr. G. W. Oman (Associate Member) asked whether any manufacturer had produced a successful engine in excess of 35½in. bore.

The author did not think there was one engine in service with a bigger bore than 900 mm., but to answer the question he had to say yes. His company had produced an experimental engine having 1 meter bore, however, they were absolutely convinced that 35½in. bore was as safe as a 30in. bore, because with their new piston, with its internal ribs, as described in the paper, they were convinced that the piston crown stresses were not higher than in the smaller engines. The same applied to stresses in the cylinder liner with steel rings surrounding and supporting the liners, stresses had been measured and they were entirely satisfied that there was no greater danger in an engine of this size than in a smaller engine.

Mr. Oman then said that a rotary valve was used in the older engine, but more recently, an automatic valve was employed. Why had Sulzer gone back to the rotary valve?

The author said that the rotary valve should be distinguished from a sleeve valve. They returned to the rotary valve about ten years ago because it enabled them to make the engine lower and facilitated a short piston. The valve was made up of steel segments with a very narrow edge, making it impossible for carbon deposits to cause the valve to stick. Steel segments were used in the event that if a piece of broken piston ring went through the rotary valve, only one of these segments would be bent. A piece of broken piston ring would only be exhausted during the pre-exhaust period, at which time the valve was open and they had never experienced damage to the rotary valve due to broken rings. Breakages of piston rings had been practically eliminated by a new method of machining the piston rings. Rings were no longer machined concentric, but in such a manner that they became perfectly round at running temperature. For example, an engine was recently dismantled for the first time after 9,700 running hours and all the piston rings were used again, not a single one being broken.

Mr. F. A. E. Redman asked if the author would clearly illustrate how pistons were lubricated, since they were now separate from the crankcase.

Mr. Kilchenmann showed a slide illustrating the location of cylinder lubricators. Eight of these were distributed around the cylinder, being fed by a lubricator driven from the camshaft. The slide showed that each lubricator discharged into an inverted vee shape groove in the cylinder wall. Oil scraper rings were no longer necessary to keep the cylinder lubricating oil from engine crankcase.

Mr. R. Neilson requested the author to give some detail of the cooling arrangements for cylinder jackets and pistons, and to say what media was used.

Mr. Kilchenmann replied that sea water was no longer used for cooling any part of the engine. Fresh water was used for both cylinder jackets and pistons; a closed circuit fresh water cooling system was used, involving heat exchangers cooled with sea water. Under no circumstances would Sulzer Bros. permit any licensee to manufacture their engine, utilizing direct sea water cooling.

Mr. Neilson further wished to know whether Sulzer Bros. manufactured any trunk piston engines and also whether they manufactured four-cycle engines; he felt there was a future for high speed engines, for propulsion of small Lakers in the order of 3,000 to 5,000 h.p.

The author said that his company manufactured a large number of two cycle trunk piston engines, for river boats on the Rhine; also this type of engine was produced for auxiliary generators. Regarding small Lakers of the size mentioned, he felt the best solution would be a direct-coupled engine.

Mr. D. F. MacDonald (Associate Member) asked if any special attention had been given to the profile of threads on the large engine bolts. Some manufacturers ground the thread form to minimize "notch" effect on such threads, due to the presence of water.

Mr. Kilchenmann replied that trouble of this nature was confined to double acting engines, some years ago. The present piston rod and large end nuts had been in existence for the last ten years and they had not experienced one case of trouble with the thread. He illustrated the manner in which the large nuts were easily loosened, enabling one man to carry out the work with a minimum of effort. A hydraulic pump and ram set was provided to produce a predetermined force at the nut. Tightening of the nut was carried out by pumping up the pressure and screwing the nut down by hand, the pressure then being released, and the nut becoming automatically tightened to the correct tension. The same method of operation applied also to the tie-rods. The reverse was being followed for dismantling the engine.

Mr. G. R. Gibson asked whether "noise fatigue" had been experienced, in the research and development of the engines. It was an advantage to keep the noise level down; was there a maximum allowable figure?

The author replied that he was unable to give a decibel figure, but confirmed that engine noise created no problem; the high noise level being confined to the superchargers. Fortunately this problem could be easily coped with by adequate insulation on the suction and discharge side of supercharger. In engine rooms of this type, it might be said that most of the noise came from the auxiliary machinery and his company had never experienced problems associated with noise level.

Mr. E. F. Kay (Member), referring to the hydraulic system for tightening the large nuts, asked if there was an alternative arrangement. He also wished to know whether the author had experienced any problem with the rotary exhaust valve in regard to clearances and burning-up?

Mr. Kilchenmann said that, in addition, an ordinary standard spanner was supplied for the nuts and could be used in the conventional manner. Clearances for the rotary valve would be about half a millimetre on each side and no problem had been experienced with this clearance, or with burning of valves. The actual diameter of valve was determined by running experiments, in which, an oversize valve was used which wore down during operation, giving the correct size required.

Mr. A. C. Waldie (Member) requested Mr. Kilchenmann, to tell him whether problems had been experienced with the use of heavy fuel, and whether he had comparable costs for heavy fuel and Diesel fuel for his company's engines?

The author declined from quoting relative costs on the two types of fuel oil because these varied considerably from port to port. With regard to burning heavy fuel, this question presented no problem at all today. Three points were very essential when burning heavy fuel—firstly, heating the fuel to reduce the viscosity; secondly, the dirt must be removed; thirdly, the high sulphur content must be counteracted by use of special lubricants in engine cylinders. The engine fuel valve was designed to provide additional cooling at the valve tip. Operating results showed that cylinder wear of ·002 to ·004in. was rarely exceeded during 1,000 running hours.

Mr. Edlund admired the author's dismissal of the problems connected with burning heavy fuel oil, very greatly and asked if he had any difficulty with vanadium and sodium contained in heavy fuel; also, had he experienced any difficulty with vapour locks developing in fuel pumps due to high temperature? Finally, ·002 to ·004in. cylinder wear during 1,000 hours running time, on a slow speed engine, seemed rather high. Would not a figure of ·001in. be more appropriate?

Mr. Kilchenmann said that the degree of cylinder wear mentioned could only be regarded as extremely low when bearing in mind the large cylinder bore in question. Also, it should be remembered that these marine engines were running constantly on full load. Vanadium had created problems in connexion with gas turbines. Vanadium might possibly have had some influence in burning on the pressure side of the piston crowns when oil cooling was utilized and the pistons being carbonized on the inside. The present day water cooling systems had completely eliminated those problems. No vapour locks were possible within the oil system, which was constantly heated and maintained under pressure at all times.

Mr. A. Newland (Local Vice-President) said that many of them were familiar with the trials and tribulations of sea water cooling for pistons. Appreciating that this was something of the past, would the author tell them something about water treatment for present day systems? Further, what maximum exhaust temperatures were experienced, also the type of scavenging used on the engine illustrated was surely not applicable to a double acting engine? Finally, for what reason were there two interruptions in diameter on the cross-section of the tie-rods?

Mr. Kilchenmann replied that he was not entirely familiar with the various methods of water treatment; a matter which was usually left to the owners. Interruptions in the cross section of the tie-rods only occurred where the thread had been undercut to facilitate threading. What appeared to be an interruption of cross-section in the middle of the tie-rod was a ring circling the rod to prevent vibration. For RD90 engines rated at 2,000 h.p. per cylinder, an approximate temperature at the supercharger exhaust would be 570 deg. F.; before the supercharger approximately 750 deg. F.

Mr. J. Boyles (Member) asked the author if he would kindly expand on the lubricating system of the engine referred to in the paper.

Mr. Kilchenmann illustrated, by means of slides, the main lubricating system and emphasized that the system did not employ holes in the crankshaft, oil being supplied at two different pressures from a common pump. While illustrating the crosshead lubricating system, the self aligning crosshead bearing was shown and he drew their attention to the fact that the slippers were not fixed but were self adjusting, being free to float in their guides.

Mr. E. L. James (Member) said that for Great Lakes operation, ships in the order of 8,000 h.p. and propelled by steam turbine were most familiar, but their naval architects were continually endeavouring to reduce the weight of propelling machinery. Did Mr. Kilchenmann have any figures showing the weight for an equivalent Sulzer engine?

The author gave an approximate figure of 270 metric tons for 8,000 h.p. and then went on to give comparisons in regard to original cost of Diesel and steam turbine installations, indicating that the Diesel enabled considerable cost savings to be made.

Mr. Kay remarked that, going back a number of years, Diesel engines of this type were susceptible to trouble when operating on overload, and asked if overloading was a problem with the engine produced today, or was there a margin of safety available to the operator?

Mr.Kilchenmann answered that these engines were supercharged, which meant, that increased load resulted in more air being supplied to the cylinder. Since more fuel could be burned, the engine did not become overloaded, hence the load limit was not so much a question of combustion, rather it was a question of maximum permissible bearing loads.

Mr. Boyles said that the problems of crankcase explosions were very much before them today. Was there anything incorporated in the Sulzer engine design to counteract this problem and would the author care to enlarge on it?

Mr. Kilchenmann replied that in the present design, the piston did not enter the crankcase, thereby eliminating explosions caused by hot pistons. In the event of a hot bearing, suitable crankcase explosion doors were provided, with deflectors, so that personnel would not be burned. No explosions had been experienced with this type of engine. It should be noted that air was constantly passing the underside of the piston, preventing the build-up of oil mist at this point.

Discussion in Halifax

SATURDAY, 24TH MARCH 1961

MR. G. JOHNSON (Associate Member) asked the author if he would agree that the normal system of two-stroke turbo supercharging, with an electric motor-driven turbo-charger taking over the duties of the exhaust turbo-charger, at low speeds, had many advantages.

Mr. Kilchenmann replied that he was aware that some firms supplied electric motor-driven superchargers and he knew of one company that had built an oil driven hydraulic turbine for main engine blowers at low speeds. This was a kind of solution, but he favoured the system that did not require any auxiliary drive with motors or clutch gear, such as the Sulzer under piston system, which acted as a scavenge pump, and it was possible that a vessel could operate at 70 per cent full speed, even if all turbo-chargers had failed.

For uniflow engines with poppet valves in the cylinder head, good acceleration could be obtained, by opening the exhaust valves earlier and stealing a little energy from the main engine to drive the turbo-blower.

The author added that his firm had, on the test bed, a small 32 cm. bore two-cycle engine on which they had reached a mean effective pressure of 15 Kg./sq. cm. (213lb./sq. in.) and this machine could operate at 10 per cent of full speed, without any aid for the turbo-charger, by moving the cam shaft a small amount, to advance the opening of the exhaust valves. The cam shaft was then returned to the regional setting, by hydraulic servo-motor, for normal operation.

MR. J. D. CLARKE (Member) asked what was the heat effect on the exhaust valves when the cam shaft was advanced?

The author replied that the advancing of the cam shaft had an adverse heat effect and the exhaust valves would tend to overheat at full loads, but this practice was only to be recommended at light loads.

Mr. Johnson asked the author to clarify the design of the fuel nozzle in respect to the cooling medium being brought very close to the injector tip.

The author demonstrated that a stainless steel sleeve was shrunk onto the inner body of the nozzle to separate the water passage from the outer body (Fig. 5). He added that Mr. Johnson no doubt was apprehensive about water leakage to the cylinder, and gave an example of the testing of this type of valve in a machine shop, where the operators mistakenly connected the water passage to the high pressure fuel connexion. The stainless steel jacket was blown up to form a ball shape and was permanently distorted, but not a drop of liquid leaked out of the assembly.

MR. K. JONES (Member) asked the author to give more details on the construction of the two piece cylinder head and of the rotating exhaust valves of the RD90 engine.

Mr. Kilchenmann answered the first part of the question by projecting the slide of the combustion chamber, Fig. 26, and continued with a description of the construction of the rotary exhaust valve, i.e. it was made from a length of steel pipe with a welded block section, on which were bolted the valve reeds, constructed of 5 mm. steel section.

The steel reeds were sub-divided in their length to prevent carbon deposits and pieces of broken rings jamming the valve. Mr. Kilchenmann added that it could be assumed that they were prepared to expect many broken piston rings, but on the contrary, the piston ring problem was absolutely different to what it was a few years ago.

Recently there was an engine opened for the first time after running about 9,000 hours on heavy fuel and not one single piston ring was replaced; the secret was in the machining of the piston rings.

The older type of rings were machined concentric. When they became warm the butts protruded through the ports and were chopped off. With the modern design the rings were machined slightly eccentric when cold, and when expanded under operating conditions they formed a true circle, thus eliminating the possibility of the ends being broken off.

MR. DEREK HUGHES (Member) wanted to know whether the author knew if any of the companies operating these modern Diesels had any difficulties in obtaining the services of qualified engineers and would he give some comparative figures on relative costs of modern Diesels and steam turbines.

Mr. Kilchenmann replied that his company were not shipowners, but engine builders, and as far as he knew there appeared to be no shortage of ships' engineers for these engines. However, the question of experienced engineers seemed to be a universal problem and he was not in a position to comment on the solution. In reply to the second part of the question, he quoted a Scottish shipbuilder whose name must remain anonymous:

First example: In the range of 7,000 to 8,000 b.h.p. the cost of the steam turbine complete installation, exceeded the complete installation of the Diesel by 16 per cent.

Second example: In the power range of 15,000 b.h.p. the cost of a steam turbine plant exceeded the Diesel installation by 11 per cent.

He went on to give an example of a Dutch shipowner, who operated different makes of propulsion machinery of steam and motor. The shipowner stated that the maintenance costs over the years were about the same, but the fuel consumption of the Diesels was only two-thirds of the steam plants.

MR. MURRAY OSBORNE (Member) said that there were five Sulzer Bros. engines of 2,800 h.p. in his company's fleet and they had found that maintenance costs were lower than similar steam plant installations.

MR. L. M. MATHERS (Member) asked the author to explain briefly the materials used in crankshaft and top and bottom end bearings and how the bearings were lubricated.

Mr. Kilchenmann replied that a standard babbitt was being used for bearings on the RD90 engine. For lubrication, the main bearing was fed separately from the sump and also the top and bottom ends were separately lubricated, thus there were no drilled holes in the crankshaft. An interesting item about the RD90 engine was the loose slipper at the cross-head guide, which did not require close tolerance fitting for alignment, but still offered a complete bearing guide in operation.

Mr. Osborne then said that his company had experienced lubricating problems on the trunk piston Sulzer engine, when using light mineral oils, resulting in piston seizure and laquering, but since using a detergent oil this had been eliminated. Would the author like to explain this solution?

Mr. Kilchenmann replied that he would like to explain this by an example of using heavy fuel oils in engines.

There were three basic problems concerning the fuel:

a) the viscosity must be reduced by fuel heating;
b) impurities must be removed by centrifuging;
c) the sulphur content must be counteracted by using a highly alkaline lubricant.

The use of a good quality, high alkaline detergent oil appeared to be the solution to the last of the above stipulated points, as well as to the problems brought up by Mr. Osborne. Mr. Kilchenmann concluded by thanking his audience.