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TESTING AND DEVELOPMENT OF THE SEAHORSE ENGINE

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The Doxford Seahorse engine was described in a paper read before this Institute in November 1971[1]. Because the concept involved building an engine with the same power per cylinder as current medium/large slow speed engines, but with half the weight and more than twice the revolution speed, it was anticipated that new problems would arise which could only be solved by extended test bed running.

A 4-cylinder prototype of 7450 kW (10 000 hp) was built and has been running since late 1971. This paper describes the problems which actually arose, with their solutions, and information gained from highly instrumented test bed running which has led to improved performance and the prospect of building production engines lighter, simpler, and more accessible than the prototype.

INTRODUCTION

In the process of building the ever more powerful engines demanded by increasing ship size and speed, air and fuel flow must be increased, mechanical parts be strengthened, lubrication of components in relative motion be improved, and cooling of parts heated by combustion be intensified. When power increase is associated with higher crankshaft revolutions, balance and freedom from vibration can only be achieved by refinement in design and manufacture.

In all these areas, design stage work by calculation and on the drawing board is the basis of ensuring optimum performance and long life, but new ideas must still be proved on a running engine. Because designers are fallible, testing will disclose weaknesses which must be corrected by redesign.

The paper describes the improvements made on the Seahorse engine during the test bed running period. There are five main sections dealing with air and fuel flow, mechanical components, lubrication, cooling, and balance and vibration.

AIR AND FUEL FLOW

Air Flow

The engine was originally designed to use two exhaust gas driven turbochargers and an additional mechanically driven compressor delivering ten per cent of the total air. Critics forecast that the high speed mechanical compressor drive would be troublesome and indeed this proved to be the case. However, development both on the engine and on a separate test rig has ironed out the difficulties and an improved and simplified gearbox has been designed and will be ready for fitting if development of the engine to higher power calls for more air than can be delivered by turbochargers alone.

Some time after the prototype had been built two factors combined to make it possible to dispense with the mechanical compressor. The first was the availability of turbochargers with about ten per cent higher efficiency than those originally fitted. The second was the result of dynamic measurements of scavenge space, cylinder, and exhaust pressures which showed that the effective blow-down area of the exhaust ports could be reduced by 32 per cent without the cylinder pressure exceeding entablature pressure at the time of scavenge port opening. This change enabled the scavenge port height to be increased to give an effective scavenge area degree integral 23 per cent greater.

The engine was rebuilt in the summer of 1973 with the new turbochargers and modified ports. Results were immediately good with a specific air flow at full load of 10·2 kg/kW h (7·6 kg/bhp h) and fuel consumption 0·198 kg/kW h (146 g/bhp h).

Further pressure measurements demonstrated that keeping the exhaust ports open for a longer period after closing the scavenge ports had little effect on the amount of fresh air retained in the cylinder, so production engines will be built with zero exhaust crank lead. Apart from the practical advantages arising from this change, described under *Balancing*, theory indicates that the specific fuel consumption should be reduced by a further 1 to 1·5 per cent.

With constant pressure turbocharging, even high efficiency turbochargers cannot by themselves deliver enough air for slow speed manoeuvring; in the absence of the mechanically driven compressor the engine is therefore fitted with a low pressure electrical blower of only 34 kW (45 bhp) to supplement the air supply at dead slow manoeuvring speed.

The electrical blower is arranged to operate automatically below a predetermined control position and to deliver air to the entablature through a flap valve also automatically controlled. The system had been developed previously for direct drive engines but the flap valve had to be completely redesigned to withstand the much higher entablature pressure of the Seahorse engine. This valve assembly (Fig. 1) has two linked plate valves, one tending to close with entablature

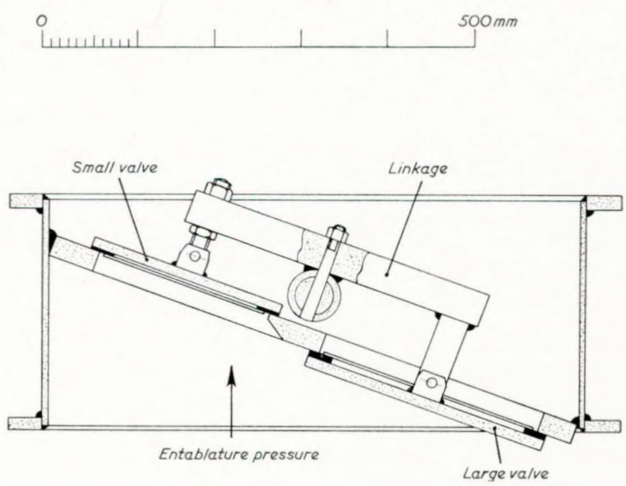

FIG. 1—*Auxiliary blower flap valve*

* Technical Director, Doxford Engines Ltd.

pressure and the other to open, the first being slightly larger to give an overall bias towards remaining closed under normal running conditions. To ensure air tightness of both valves over the pressure range required, compression of the seating of the large valve plus the bending of its lid must equal the deflexion of the linkage plus the bending of the smaller valve lid and the growth of its seating. In practice this requirement calls for a soft seating on the large valve with a very stiff linkage and a firm seating on the small valve. With too great a linkage stiffness the large valve would be held open by increasing pressure whilst over flexible linkage would fail to keep the small valve closed.

Fuel

At the design stage it was known that in order to obtain an outstandingly good fuel consumption this engine would need an instantaneous volume rate of fuel injection twenty per cent higher than on the current 76J engine. At the same time the flatter combustion space resulting from the shorter piston strokes would make it very difficult for fuel from the normal two fuel valve injectors to reach all the available air without excessive air swirl.

Accordingly, the engine was provided with four equally spaced fuel valves around the periphery of the combustion space and some thirty different sets of nozzles were obtained for evaluation. In the trials a specific fuel consumption at least as good as that of any known engine was obtained both with nozzles with 4 holes 0·8mm diameter and those with 5 holes of the same diameter, showing a most unusual tolerance for variation in nozzle flow area. This tolerance is confirmed by the fact that one set of injectors remained in the engine for over 1500 hours without the slightest sign of performance deterioration. For best results the 4 hole nozzles were so fitted that two sides of the square formed by the sprays were parallel to the piston crowns to minimize impingement. The 5 hole nozzles gave results equal to the best of those with 4 holes with the added advantage that total vertical spread of the sprays is almost constant, regardless of the fitting angle, so there is no need for reference marks or dowels.

The other side of the coin was shown by trials with single hole fuel nozzles. Such nozzles would be attractive from the point of view of simplicity and resistance to choking, but, unfortunately, they had a most adverse effect on engine performance. The boost pressure rose by 0·25 bar, the fuel consumption by 4 per cent, and the maximum cylinder pressure by 7 bar.

Further tests with 3 hole nozzles in a fan shaped spray gave results intermediate between those with the single hole nozzles and the standard 5 hole nozzles.

Varying degrees of air swirl were also tried by fitting different sets of swirl vanes around the scavenge ports. Again the combustion system showed great tolerance and similar results were obtained with the vanes originally fitted, and with others giving 43 per cent less swirl. Cylinder liners used later, with more efficient turbochargers, had larger scavenge ports designed to give an intermediate swirl rate 30 per cent less than the original. This ensured adequate margin against insufficient swirl, which might otherwise occur with fouled turbochargers and, on the other hand, kept the swirl rate as low as practicable to reduce heat flow to the pistons and liner.

The original common rail fuel system on this engine was designed to impose a very low pressure drop between timing valves and fuel injectors. Early measurements of valve spindle lift and fuel pressure at the injector showed that the rather large volume of the system delayed the necessary reduction of pressure after injection to an extent likely to promote dribble.

By reducing the bore of the pipe between the timing valve and four-way distribution block from 13mm to 10mm and of the pipes between the distribution block and the injectors from 8mm to 6·2mm, together with some re-routing, the volume after the timing valve was reduced from 0·507 litres to 0·276 litres. This gave a better pressure pattern and clean opening and closing of the fuel valves as shown in (Fig. 2).

Further consideration led to the design shown in Fig. 3b which has a corresponding volume of 0·24 litres and four other advantages i.e. neatness, easier accessibility, equal short pipes between distribution blocks and injectors to simplify

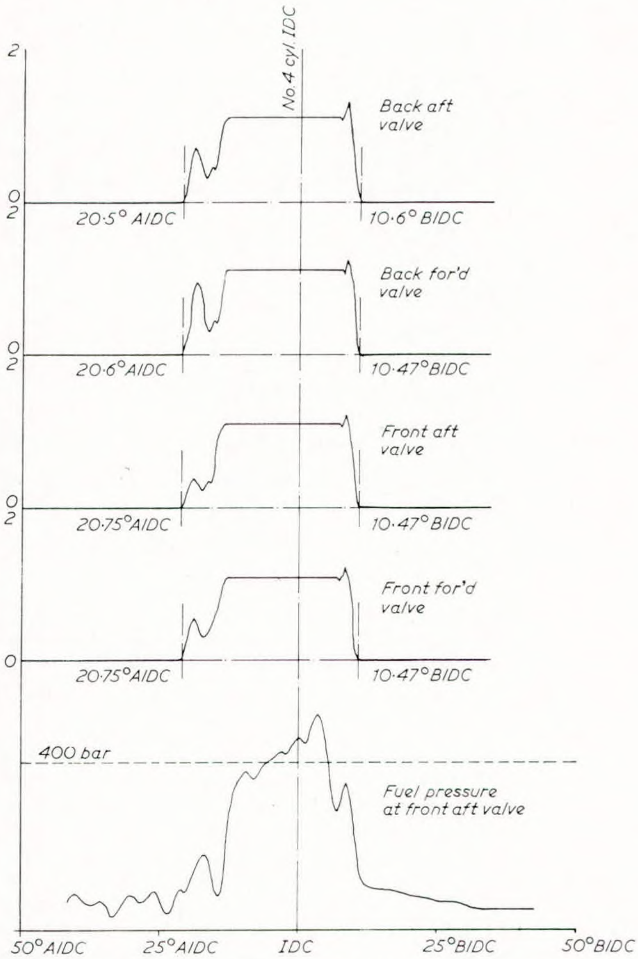

FIG. 2—*Fuel injector valve lift and pressure diagrams*

injector maintenance, and carrying the fuel from the back flow system for warming up nearer to the injectors.

The fuel pump used with the common rail system is based on the opposed cylinder design which has been in use on Doxford engines since 1955. On earlier engines, fuel delivery quantity was controlled by turning the pump plungers which were made with helical cut-off scrolls.

When bridge control of main engines became common in the 1960's pneumatically loaded spill valves were introduced to control the fuel pressure, allowing the quantity of fuel to be determined entirely by the timing valves. The control arrangements for the fuel pump plungers were retained for emergency use in slow speed engines, but by the time the Seahorse was designed sufficient service experience had been gained to justify dispensing with separate pump control.

The pump was, therefore, built with plain plungers which can tolerate larger clearances because of the greater sealing length between plunger and barrel, thus making the pump less affected by temperature difference arising when changing from diesel oil to heavy fuel.

For safety, to cover the possibility of control air failure, a third spill valve is fitted in addition to the working and standby pneumatically controlled valves. The third valve is kept in connexion with the system at all times but is set at a higher leak off pressure than the working valve. In this way it acts as a safety device to prevent excessive pressure developing. At first the extra valve was operated by lubricating oil pressure but as this had a sluggish action which made engine control difficult, a manually operated valve with screw and handwheel has now been developed and this is both simple and satisfactory.

Normal spill valves have had a tendency to squeal unpleasantly under some operating conditions, particularly when using heavy fuel. A modification incorporating a captive synthetic rubber disc between valve plunger and diaphragm has

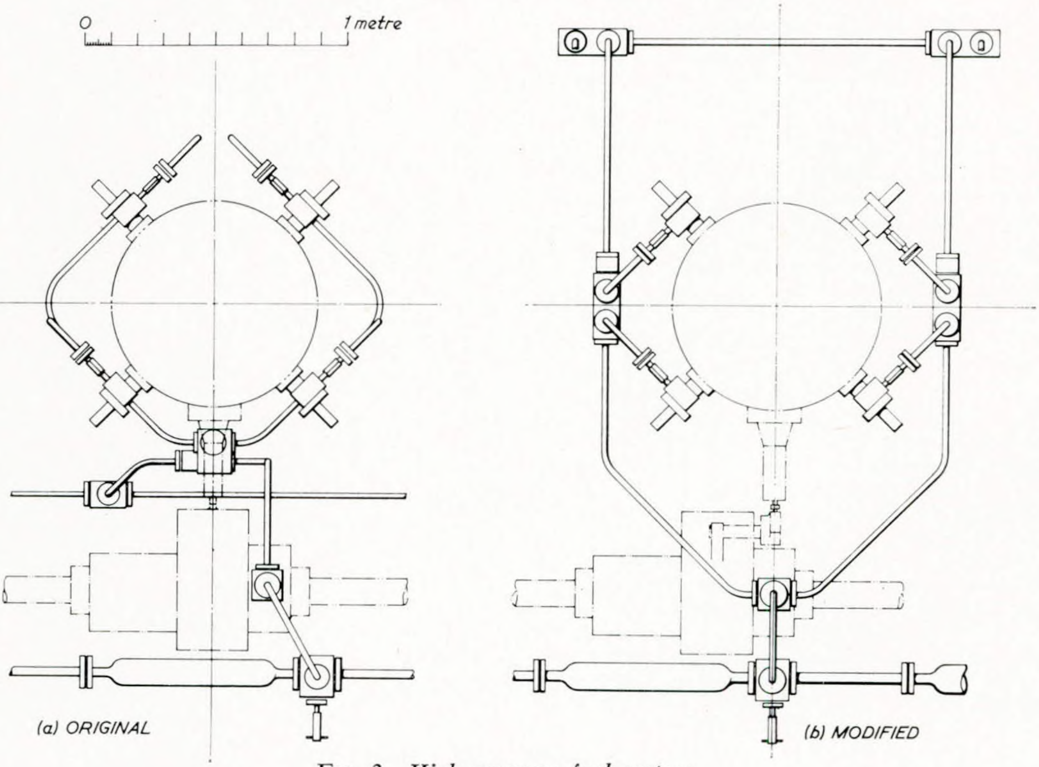

FIG. 3—*High pressure fuel system*

undergone extensive testing on the prototype engine and has cured this problem.

Provision was made in the protoype engine for fitting jerk pumps instead of the timing valve system. Because the flow rate required was higher than had been used in any current engine a separate test rig was set up to discover and correct any possible sources of weakness that might otherwise arise in service. A number of improvements were made, particularly in the cam and roller arrangement and over 1500 hours running was carried out, including a full output test of 1000 hours. Because of the successful operation of the timing valve system the jerk pumps have not yet been tested on the engine.

Air Starting

The air starting system used on this engine has proved to be very successful—accelerating the engine very quickly to over 100 rev/min, and giving a peak speed of 125 rev/min on air only. However, in its original form the rather large cover over the valve mechanism impeded access to the cylinder and was, therefore, redesigned as shown in Fig. 4b, giving a reduction of 22cm to the height of the main mechanism, and the final operating lever for the cylinder valve is now mounted on an outrigger bracket. The fuel timing valve may thus be located on the cylinder centre line to give a completely symmetrical piping system to the fuel injectors.

MECHANICAL COMPONENTS

The robust design of the engine is underlined by the fact that there has been only one failure due to mechanical stress despite gross overloads which inevitably occur during experimental testing; the exception was a side rod which cracked in the threaded connexion to its crosshead. This rod had not only suffered excess stresses due to the combination of upper piston movements described under *Balancing*, but also had to withstand the effect of an upper piston seizure resulting from inadvertently running at high power without piston cooling water. Following this incident the side rods were strengthened and a better water flow alarm system has been developed.

However, the robustness of the running gear initiated new alignment problems.

Alignment

Early marine engines were very tolerant of misalignment between cylinders, crossheads, and crankshaft because their long slender connecting and piston rods had a high degree of flexibility. This tolerance was fully needed because bed plates and columns were also rather slender, permitting considerable change in engine shape with ship movement. Increasing

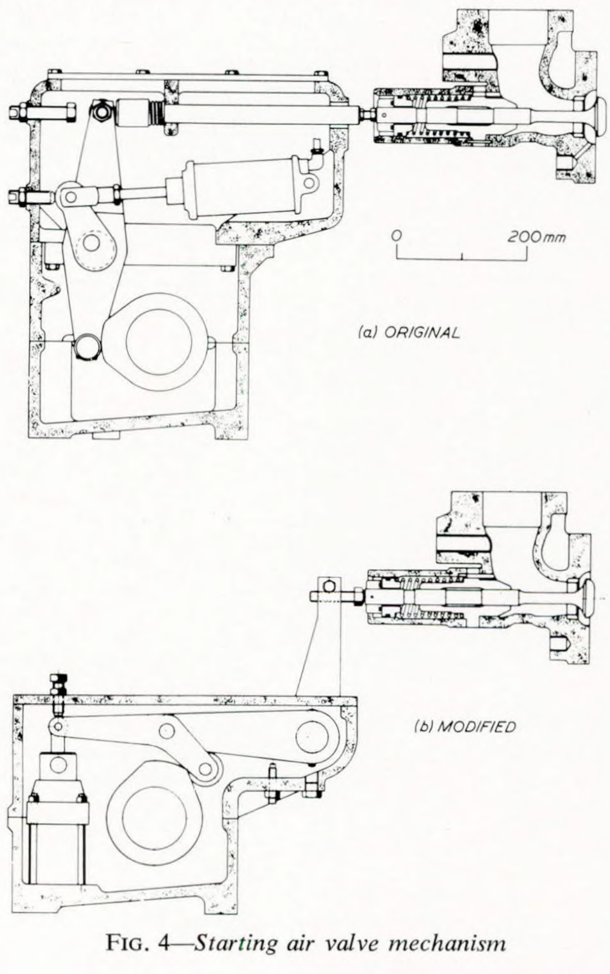

FIG. 4—*Starting air valve mechanism*

ratings following turbocharging necessitated more robust running gear and a higher standard of alignment had to be maintained. The short stroke and substantial connecting rods of the Seahorse carried this trend even further and some difficulties with piston alignment occurred in early running on the test bed. These were overcome by four improvements made at various stages:

1) The method used for setting pistons was refined: On first erection a telescope is set up on the upper face of the crosshead bracket with the crosshead urged into the normal running position by jacks. The cylinder assembly, which is attached to the entablature by a flange with large clearance studs, is then moved until correct alignment as shown by two targets fitted at top and bottom of the bore. The prototype engine liner lower jacket was dowelled to the entablature but, for simplicity in replacement, future engines will revert to the method used on J-engines in which a circular flange on the lower jacket is located between four shaped blocks welded to the entablature top after alignment has been made correct. Since the lower jacket is spigotted concentric with the liner, fitting a replacement simply requires dropping in the new liner and ensuring that clearances between jacket flange and blocks are as stamped on the blocks.
2) Unlike previous engines the cooling oil for the Seahorse lower pistons is carried from openings in the faces of the crosshead guide bars into and out of slots in the fore and aft faces of the crosshead shoe. The entry oil pressure of 5 bar tends to keep the crosshead shoe running firmly against the aft guide bar. On the prototype, however, it was found that the extreme stiffness of the short centre connecting rod in combination with natural slope variation of the centre crankpin could cant the crosshead under some running conditions and cause misalignment of the piston in the plane of the crankshaft axis. To overcome this the fore and aft side of the originally cylindrical connecting rod shank were milled away as shown in Fig. 5 Whilst leaving the rod with ample strength this modification reduced its bending stiffness by fifty per cent. Tests with proximity transducers fitted in the aft guide bars confirmed that with this change the crosshead and piston assembly did in fact move parallel to the crosshead guide.
3) With the original design of the crosshead in which the pin fitted into a semicircular saddle in the bracket, the combination of a crosshead pin at the lower limit of diameter tolerance with a bracket bored to the upper limit could result in the upper face of the bracket becoming convex to the extent of a few microns when the piston rod lower flange bolts were tightened. Because of the multiplying effect of the piston rod length, the order of tightening the flange bolts could therefore change the transverse alignment of the piston by an appreciable amount.
A temporary solution was found by grinding away the surface of the compression plate fitted between bracket and piston rod to leave only two fitting strips in way of the front and back pair of bolts. The final solution is shown in Fig. 6b in which the location between crosshead and bracket is by means of horizontal and vertical flat faces. This construction also permits a stronger cross-

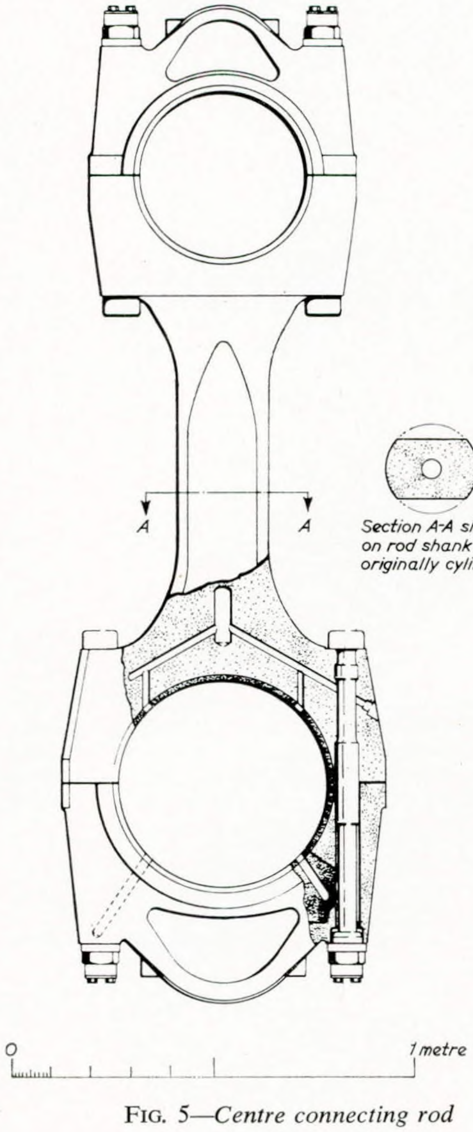

FIG. 5—*Centre connecting rod*

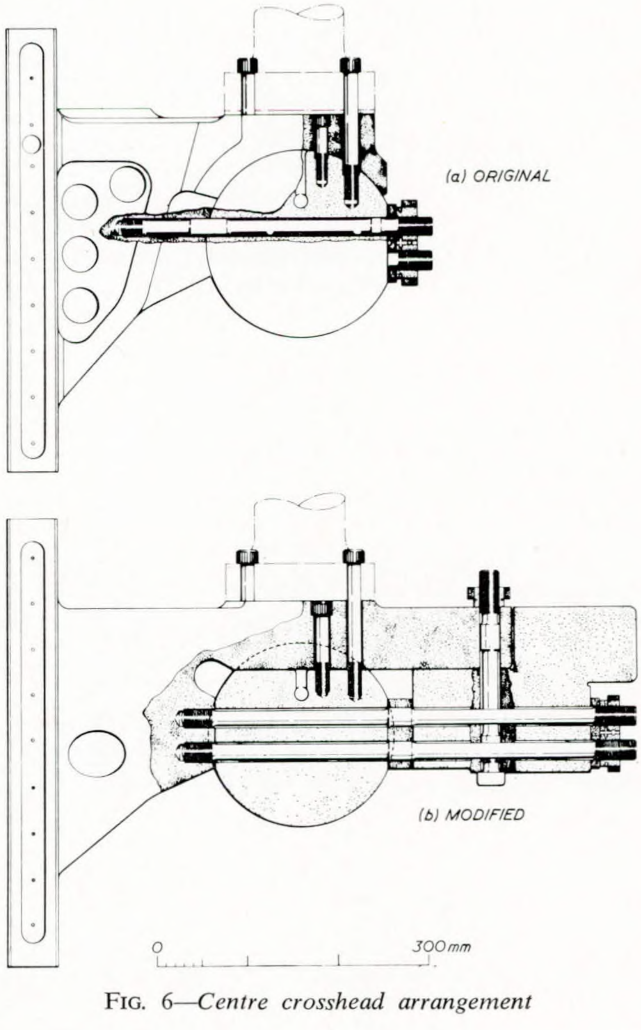

FIG. 6—*Centre crosshead arrangement*

head bracket for the same height and weight.

4) After many hours running, examination of the exhaust manifold expansion pieces between cylinders showed that they had hardened so severely as to prevent the cylinders maintaining their correct alignment during the heating and cooling cycle. They were replaced by the piston ring type expansion joints shown in Fig. 7 which are designed to retain their freedom of movement in adverse conditions and which are suitable for cleaning and overhauling after long service.

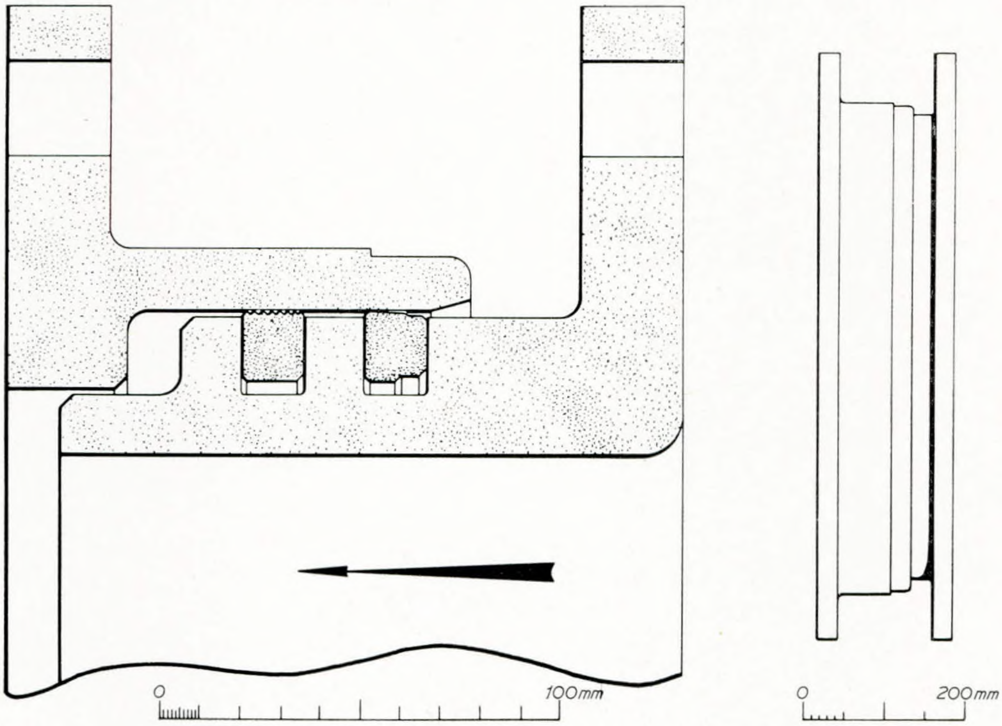

FIG. 7—*Exhaust manifold expansion joint*

Engine Mounting

It will be recalled from Reference 1 that instead of using the traditional multi-chock connection between bedplate and tank top this engine was designed to be supported on four sliding feet only. The objects of this change were to simplify installation and to save the engine both from hull deflexions and from the hogging effect resulting from a warm engine being rigidly connected to a cold double bottom.

The prototype has been running on the test bed for three years with the new mounting arrangement. Measurements have shown that change of shape between hot and cold conditions is negligible and that the controlled movement at the feet takes place as forecast. The synthetic rubber inserts in the sliding feet show no sign of deterioration.

In 1973 an alignment check showed that the crankshaft had become 1·3mm lower at the forward end than at the aft end. Naturally, the mounting feet were suspected but further investigation proved that no creep compression had occurred in these components and the decline was entirely due to subsidence in the concrete foundation.

This may have been due to heavy blasting in the adjoining shipyard reconstruction. The important point is that the alignment of the main bearings remained perfect although the whole engine had tilted through a small angle.

LUBRICATION

Bearings

For the same total power an opposed piston engine developing 1850 kW (2500 bhp) per cylinder has the same number of running gear bearings as a VEE 4-stroke engine of 670 kW (900 bhp)/cylinder. To maintain the advantage of simplified maintenance given by the smaller number of cylinders, it follows that the bearings must have a very long life and be easily accessible for checking.

In the Seahorse engine the clearance at all running gear bearings can be checked with feelers without dismantling and since they are all thin shell white metal bearings they are easily surveyed and replaced if necessary.

The "Achilles heel" of the slow speed two-stroke is the centre top end bearing but the higher revolution speed of this engine produces a load reversal and throughout the test running the centre top end bearings have remained as new.

Originally, half the side top ends had nitrided crosshead pins and half cast iron. The bearings running on cast iron showed some surface roughening which could perhaps be overcome by super-finishing the pin surfaces, but the white metal bearings running on nitrided pins have behaved perfectly so this combination has been standardized. One experimental side top end with a 40 per cent tin aluminium bearing running on a nitrided pin experienced a seizure which destroyed the bearing shell but not the pin. Investigation caused some doubt as to the quality of the original bond between the tin-aluminium and the steel backing. It is worth commenting that had the bearings been overlay plated with lead tin, as is current practice, the seizure might have been avoided, but the matter has not been pursued because the white metal bearings have been so successful.

The two extreme end side bottom end bearings and the two nearest the centre of the engine have experienced some damage but the other four have run satisfactorily since the beginning of testing. Since the damaged bearings all showed signs of edge loading an analysis was made of the crankpin slopes occurring along the shaft under normal running conditions. This indicated that the end bearings suffered greater slopes than the intermediate ones but did not explain the damage to the two bearings adjacent to the middle of the shaft. However, since there had been an accident during manufacture which had necessitated replacing the dog leg forging between cylinders 2 and 3, the side crankpins concerned were examined and they were found to be less accurate than normal.

Trueing these pins and fitting bearings lined with a stronger white metal containing chromium appears to have solved the problem for these two bearings, but in addition the side connecting rods have been redesigned as shown in Fig. 8b to give more than twice the bending flexibility and so reduce side loading.

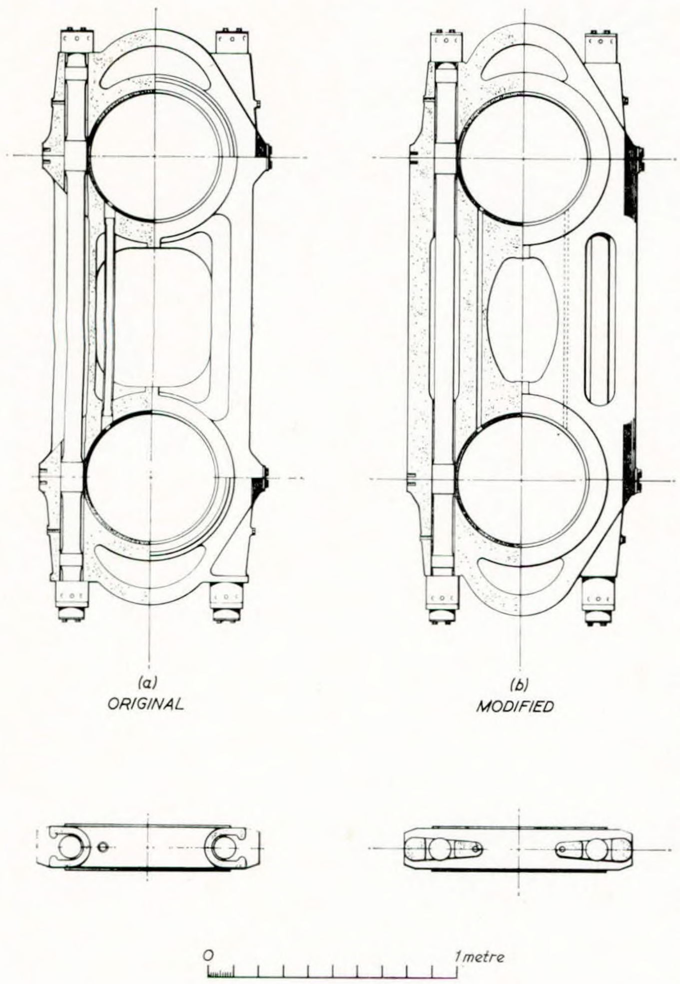

FIG. 8—*Side connecting rod*

Two of the centre connecting rod bearings have also shown signs of heavy loading which in this case suggested oil scarcity. For this bearing, oil is supplied from the top end through a bore in the connecting rod. Analysis of the inertia forces on the oil column showed that during that part of the cycle around inner dead centre the pressure at the top end, due to inertia, is greater than the supply pressure at this point, so oil will flow away from the bottom end. To provide a continuous supply of oil to the bearing the connecting rod keeps were drilled to form the oil reservoirs shown in Fig. 5; these are fed during the bottom half of the cycle and have sufficient capacity to maintain an oil film during the time when the oil supply is cut off by inertia.

The main journal bearings which are of large diameter and lightly loaded have shown no wear down at all and have remained undisturbed since the engine started.

Piston Rings and Cylinders

The heart of a reciprocating engine is the piston-ring-liner combination. Brutal shipowners want a piston ring to slide up and down the cylinder wall for a total distance equal to fifteen times round the equator, in a hostile environment with excessive contact pressure and minimal lubrication. Whilst travelling this journey it has to seal the hot high pressure combustion gas from the air below the piston and not wear itself more than 2mm or the liner wall more than a quarter of this amount. It is small wonder that piston ring scuffing is a problem in the development of all high output engines and Seahorse has been no exception in this respect.

In early running, piston misalignment with consequent high skirt wear and scuffing obscured the signs of piston ring misbehaviour. When corrected alignment permitted high loads to be carried the ring problem appeared. In some cases results were disastrous. For example, copper-coated high tensile cast iron rings gave excellent results until the copper coating wore off, but then caused liner wear as high as 1mm/100h. Similar rings in the upper pistons of J-engines with much lower rubbing speed give excellent results. Stepped rings with a barrelled rubbing surface also behaved well until the step wore away, but as soon as contact occurred with the relieved portion of the ring surface catastrophic wear developed.

At this stage discussions took place with engine builders and piston ring manufacturers from all over the world and it became clear that although a number of palliatives had been found, giving acceptable results in specific engines, there was no sovereign remedy which would give the required margin of safety in a large, high output, two-cycle engine.

By this time a large number of ring arrangements had been tried with varying degrees of success and the timed cylinder lubrication system had been refined to match the higher rotational speed and smaller oil quantity per revolution required in this engine. The distributor (Fig. 9a), used earlier, had a pawl carried in an anular ring around the ratchet wheel. Friction in this mechanism combined with the pressure drop in the fairly long pipe needed from actuating pump to distributor sometimes caused irregular operation at

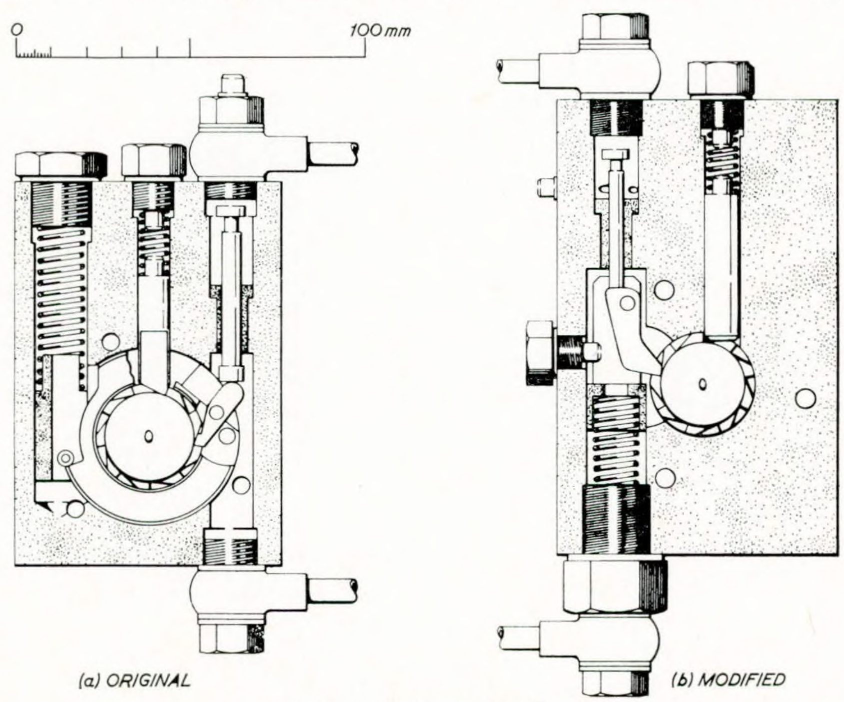

FIG. 9—*Lubricator distributor*

high speed, resulting in the oil being injected through only one quill instead of through each of eight in turn. This was cured by the redesign shown in Fig. 9b in which the pawl is carried on a separate plunger. Due to reduction of friction this design required 50 per cent less operating pressure. At the same time the operating mechanism was improved by providing the actuating plunger with substantial bleed ports opening at full stroke to permit the actuating pump to deliver excess quantity without over-travel of the ratchet, thus promoting a circulation in the actuating oil system.

With the original system, rig tests showed that output per cycle of the lubricator increased with speed so that with the setting correct for full speed there was a danger of insufficient oiling under manoeuvring conditions. A series of experiments, in which one factor at a time was varied, showed, eventually, that reducing the opening pressure of the lubricating oil injection from 27·5 bar to 20·5 bar and increasing the width of the pump spill port from 3 to 6 mm gave constant quantity per stroke from 50 to 350 rev/min. These changes, in combination with a reduction of the quill orifice size from 1·5mm diameter to 0·75mm also reduced the temperature sensitivity of the system to only a 20 per cent increase in output between 20°C and 80°C.

With the smaller plungers needed for the higher speed engine (9mm diameter instead of 12mm), some difficulty was experienced in priming the system. This was cured by fitting small non-return valves in each outlet from the distributor to reduce the effect of entrained air. The non-return valves also have the advantage of ensuring that should one quill valve become inoperative and admit gas to the piping, the effect is confined to the single injection point.

Piston examination showed that there was now ample oil available in the ring area and the problem was to ensure that sufficient remained between ring surface and liner wall to prevent metallic contact at the critical point in the cycle just after i.d.c. Baker, Dowson, and Strachen(2) gave the results of a computer calculation of oil film thickness between ring and liner and of oil transportation along the liner by the ring. A simplified approximate calculation was developed which could be extended to apply to a flat faced ring tilted in relation to the liner. This showed that a very small angular deviation, of one part in four thousand between the two, could reduce the minimum oil film thickness to one-tenth of its value, for a ring with its flat face parallel to the cylinder wall.

Consideration of cylinder liner and piston expansion due to temperature indicated that at full load there would be a taper near the outer end of ring travel on the liner of one part in a thousand and a downward slope of the first piston ring groove of four parts in a thousand.

Clearly these tapers would be disastrous for a flat faced ring and since it was known that rings tend to wear to a parabolic profile a study was made of the curvature needed to accommodate the thermal tapers. Reference 2) demonstrated that a large radius of curvature was needed to promote a sufficiently thick oil film and it became apparent that a satisfactory compromise was unlikely to be reached with normal flat based ring grooves.

A colleague of the author had previously calculated the couples acting in a piston ring due to gas pressure and a rig had been set up to measure the torsional stiffness of a ring. Correlation of these results suggested the possibility of a ring capable of aligning itself with the cylinder wall and led to the ring groove form shown in Fig. 10. At the time of writing, this ring arrangement has given very promising results but further testing is needed to confirm that a method of preventing scuffing in all conditions has been found.

The work of testing piston rings and assessing liner wear has been greatly facilitated by two experimental techniques described in(3),(4), and (5). The first, consisting of fitting thermocouples close to the gas side liner wall just below the position of the firing ring at i.d.c., provides a means of detecting scuffing or ring sticking before serious damage occurs. A section of such a record from three couples fitted in the upper section of No. 2 liner (Fig. 11) shows regular small peaks indicating the ring gap passing a particular thermocouple as the ring turns. Trouble with the ring would be indicated by a larger and higher temperature excursion.

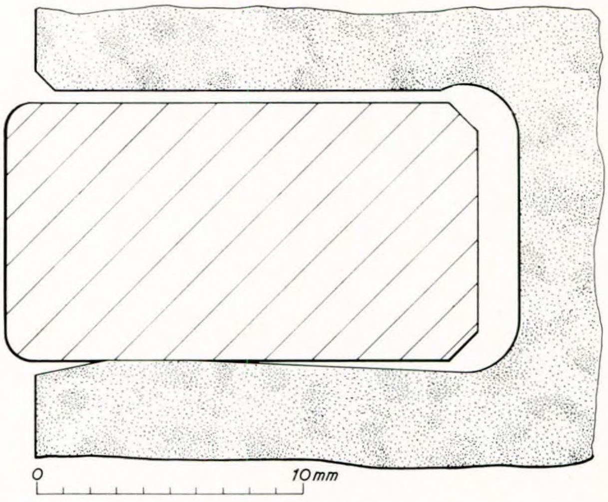

FIG. 10—*Piston ring groove*

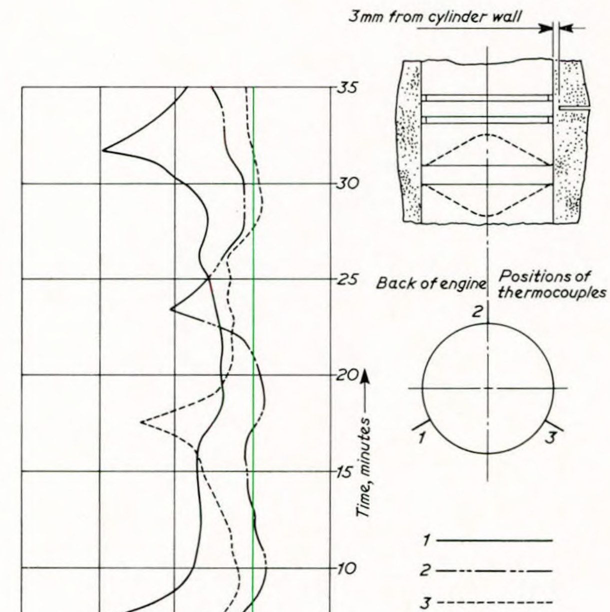

FIG. 11—*Liner thermocouple records*

The second technique (4) and (5) of cutting conical dimples in the liner wall and assessing wear by measuring their change of diameter has provided a means of measuring liner wear at least ten times more accurately than micrometer measurement and has made it possible to obtain useful wear figures after running periods as short as 100h.

Piston Rod Diaphragm Glands

Because of the relatively small crank chamber volume and high speed of the running gear in this engine, oil is thrown more violently from bearing edges than in a slow speed engine, and the density of oil mist is greater. This is an advantage from the point of view of safety from crankcase explosion because the mixture strength is further from the danger ratio but makes sealing the crank chamber more difficult.

The normal piston rod diaphragm gland shown in Fig. 12a has sealing rings to retain the scavenge air, separated by a cofferdam, from scraper rings returning oil to the crankcase. Cofferdam drainage to external sludge pots and the composition and quantity of the oil in the pots show any malfunctioning of either the sealing rings or scraper rings. With this type of gland on the Seahorse engine the leakage amounted to as much as 4 litres per cylinder/h. Meticulous fitting of the scraper rings and detail modifications to their drainage arrangements failed to reduce this figure to a satisfactory level.

Analysis of the drainings proved them to be pure crankcase oil showing that the scavenge air sealing rings were working satisfactorily but that the oil was not draining back to the crankcase from the scraper ring pack.

The modified arrangement shown in Fig. 12b with two cofferdams completely cured this problem. Here leakage into the lower cofferdam drains through pipes at the front of the engine down to the bottom of the bedplate where oil throw is less violent and the length of drain pipe prevents back flow. The upper cofferdam is drained to sludge pots in the normal way.

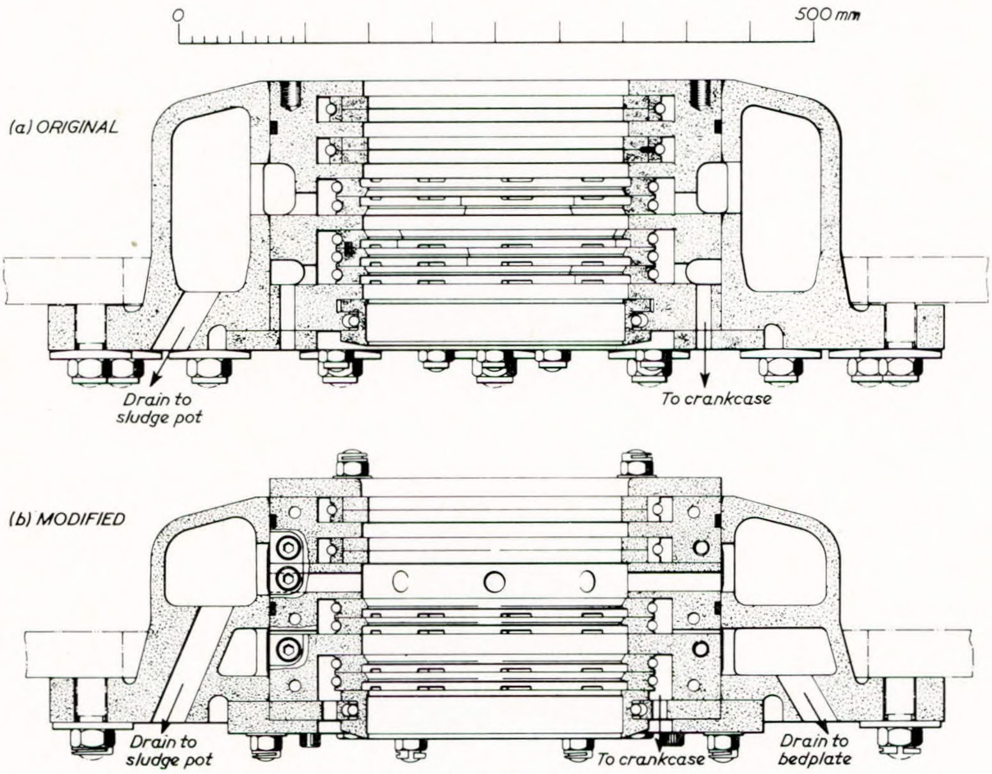

FIG. 12—*Piston rod diaphragm gland*

COOLING

Pistons

In an engine which has to run for long periods at high power it is essential that piston and cylinder liner temperatures should be maintained between upper limits set by lubricating oil carbonisation on the rubbing and oil cooled surfaces and thermal stress and fuel cracking on surfaces exposed to flame, and lower limits determined by the dew point of acid products arising from burning heavy fuel.

In the design stage the temperature and thermal stress pattern for the Seahorse was calculated from measurements on earlier engines by using a finite element computer programme. This calculation indicated that with temperatures shown in Fig. 11, Reference 1, thermal stresses would be low enough to give very long life for both pistons and liners.

On the prototype engine one cylinder and its two pistons were fitted with 26 thermocouples to measure temperature patterns, whilst the other cylinders were each fitted with six thermocouples measuring the temperature of the liner wall between the firing and second rings on upper and lower piston at i.d.c.

In addition, over 1200 piston temperature measurements have been made by means of "Templugs" to assess the effect of changes in cooling arrangement and varying airflow and load. These hardened steel screwed plugs 3mm diameter 4mm long produced by Shell Research Centre at Thornton can be fitted easily in positions where it is impracticable to fit thermocouples. After six or more hours running in the engine at a specified load the templug is removed and its hardness measured. The reduction of hardness from the original condition gives a measure of the temperature to which the plug has been subjected. Comparison with interpolated results of thermocouple readings on this engine suggests that the templug results agree with thermocouple readings to within $\pm$ 30°C in temperatures between 250 and 450°C on the piston crown surface, but tend to read higher by about 30°C on the lower temperatures between 100 and 200°C on the ring lands.

An example of the use of templugs is shown in Fig. 13 which gives a comparison of piston gas side temperatures between a normal upper piston and one fitted with triangular rods to increase water speed by 70 per cent in the cooling bores connecting inner and outer cooling spaces in the piston. The rods had the effect of reducing the hot surface temperatures of the piston periphery by about 30°C. Although measurements have not been made of the temperature of the hottest part of the oil cooled surface of the lower piston in the 'Cosy Corner' between crown and ring belt, it is to be expected that this modification will reduce the temperature at this point from the present measured 220°C to about 200°C. Since turbulence in this area is increased the margin against carbon formation will be considerably enhanced. After extended full load running there has been no carbon build-up in the oil cooling space but it is desirable to have as great a margin as possible to allow for possible adverse service conditions.

All the measured piston temperatures on the prototype engine are lower than the figures used in the calculation of thermal stresses, confirming that the piston should be free

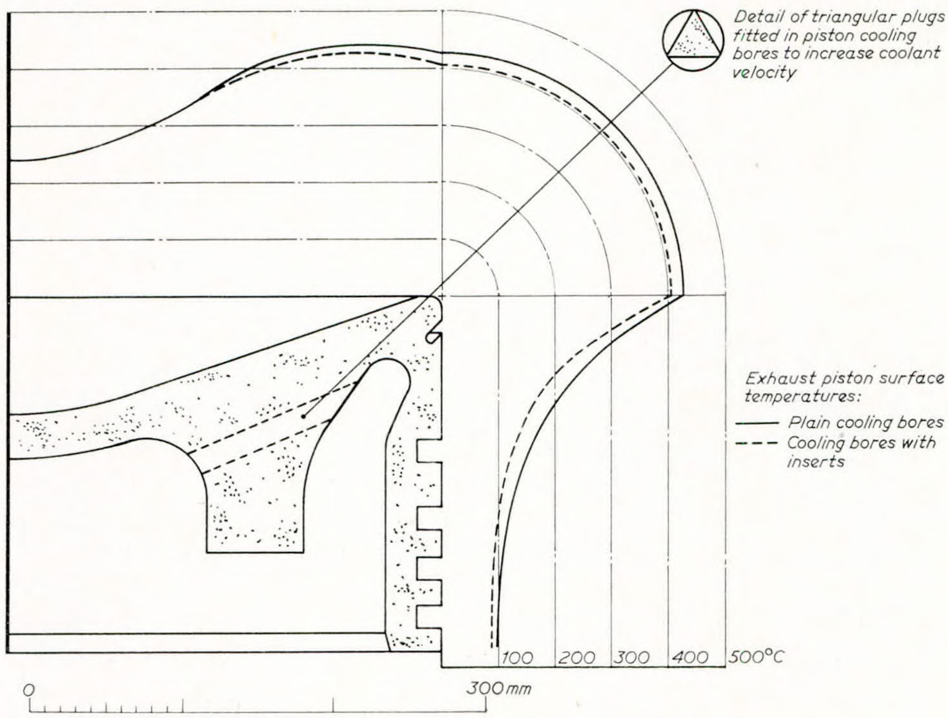

FIG. 13—*Exhaust piston surface temperatures*

from danger of cracking and that the measurements taken in this engine to improve cooling have been successful.

Cylinder Liners

In the liner, although the heat flux has been found lower than estimated, the highest figure being only 326 kW/m^2 compared with an estimated 462 kW/m^2, the position of the peak flux is just below the injector centre line instead of the forecast 5cm above. The result of the variation is to reduce the liner thermal stresses a little below those calculated but to increase the liner gas wall temperature by amounts varying up to 25°C at the hottest point in the combustion zone. The increase was confined to the area between 3cm above the injector centre line and the position of the lower piston firing ring at i.d.c. However, the highest temperature of the liner wall is still some 55°C below the level of 340°C at which the cast iron liner material begins to lose strength. The temperature in way of the firing ring is only 6°C above the estimated 180°C and well below the figure of 220°C at which cylinder oil is liable to form lacquer.

Future cylinder liners will have a modification to the cooling hole arrangement to bring them closer to the inner liner wall in the area concerned. This change will reduce the liner wall temperature in way of firing ring travel to about 155°C giving a very big margin against oil deterioration.

In the course of test running, four cylinder liners have cracked. The first two of these were from the first batch of castings made of this type and sectioning showed that the failure was due to major porosity in the heavy section around the combustion zone. On later castings porosity was avoided by modifying the chill arrangement in this area. The third and fourth failures resulted from overheating due to incorrect drilling of the cooling holes. Twelve of the slanting cooling bores passing from bottom to top of the combustion belt are interrupted by the fuel, air starting and relief valve pockets, and the connection between upper and lower bores is made by crossholes alongside the pockets. In these two liners the crossholes were insufficiently deep to make the connexion.

Traditionally, on marine engines, cooling water and oil flow have been shown by sight glasses because of their inherent simplicity and reliability. These have the disadvantage that the return flow line must be brought to a position where the sight glass can be easily seen, and the sight glasses must be vented to atmosphere to maintain a clear visible jet; the system is thus open, and extra pumping power is used in raising the coolant from supply tank to sight glass level.

After unsuccessful attempts to find a proprietary flow indicator that was entirely satisfactory in service the robust device shown in Fig. 14 was developed. With a pressure drop across the orifice of 0·6 bar the balanced diaphragms raise the flap indicator to show a green face instead of red and an eccentric on the spindle operates a micro-switch for additional remote indication.

Since the measuring orifice need not be adjacent to the indicator, and may indeed be an integral natural restriction in the cooling system, the indicator can be placed in the most convenient position for viewing. Simple shut-off cocks in the small pipes to the indicator permit servicing without stopping the engine.

BALANCE AND VIBRATION

Balancing

With the help of a contra-rotating balance shaft running along the back of the engine, primary and secondary balance is perfect. Nevertheless, problems which would not occur with a slow speed engine or with trunk piston engines arose due to the unbalanced inertia couples generated by the traditional offset crossheads. These couples produced quite heavy transverse loads of up to eight tonnes on each lower piston skirt, and bending stress in the side rods transmitting the load from the upper pistons. As a temporary measure the crossheads were lightened by drilling and milling away redundant material, but the final solution was to fit the lightened and balanced crossheads shown in Fig. 6.

In the interim period various types of piston skirt were tried in an attempt to find an arrangement capable of carrying heavier transverse loads than the normal rubbing band. Of these the most successful was a skirt composed of twelve cast iron segments arranged to pivot about their vertical mid point, rather like Michell type thrust bearing pads. This arrangement gave the bonus of providing a means of checking piston alignment in place, since with a correctly aligned piston all the pads could be moved by finger pressure through the scavenge ports.

The side crossheads were also lightened and fitted with balance weights and the opportunity was taken in the redesign to improve the threaded connection between side rod and crosshead. Careful analysis of the stress concentration at this connection showed that a 30 per cent increase in strength

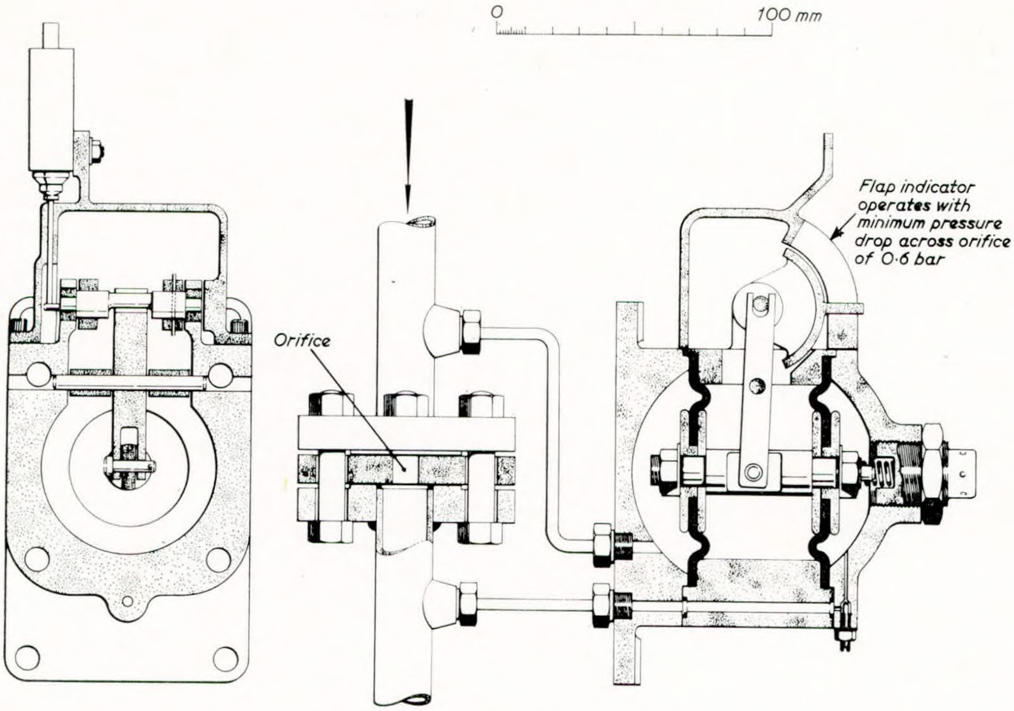

FIG. 14—*Coolant flow indicator*

would be achieved by reducing the cross section of the socket in the crosshead and using a coarser thread with a slightly larger diameter; the thread is heavily truncated to avoid the danger of seizure. These measures will provide a margin for future development of the engine to higher powers and maximum pressures.

Swinging links carrying the water to and from the upper piston, (Ref. 1, Fig. 14) produced a couple tending to twist the upper piston about the vertical axis twice per revolution. New forged links of about half the mass were made which alleviated the problem, but for continued full speed running a quick solution for the prototype was found by fitting slave balancer links at right angles to the transverse beams, producing a couple equal and opposite to that generated by the water carrying links. This solution is untidy and only suitable for the prototype. A design has since been evolved for production engines with diagonally opposed links producing no couple.

The prototype engine was built with exhaust cranks giving 8° exhaust piston lead. Tests described earlier under *Air Flow* showed that with constant pressure turbocharging this lead is unnecessary, so production engines will have main and exhaust cranks at 180°.

The lead on the prototype engine produces primary out of balance forces in each cylinder about 90° out of phase with the main crank, and it was the couple produced by these forces which was counterbalanced by offset crankwebs and by weights on the contra rotating balance shaft. Dispensing with the lead and reducing the exhaust piston stroke by 10mm on production engines enables the crankshaft weight to be reduced by nearly 4 tonnes without loss of strength and obviates the requirement for a balance shaft. This change also simplifies the manufacture of pairs of engines with oppositely rotating crankshafts.

Torsional Vibration

An unconventional design feature of this engine is the arrangement of the auxiliary drive at the forward end where torsional vibrations are normally more severe than at the aft main drive end. To avoid transmitting crankshaft movement to the auxiliary drive, the first gear wheel is mounted on a coupling supported in its own independent bearings and flexibly connected to the crankshaft. Inherent viscous friction in the coupling enables it to serve as a crankshaft damper for both torsional and axial vibrations.

Vibration conditions on the test bed are more severe than they would be in service because the dynamometer used for absorbing engine power has a heavy rotor on a relatively flexible shaft. This arrangement results in the fourth order natural frequency of the system occurring at only 360 instead of at over 800 rev/min as it would on a normal marine installation. There was consequently the possibility of fairly

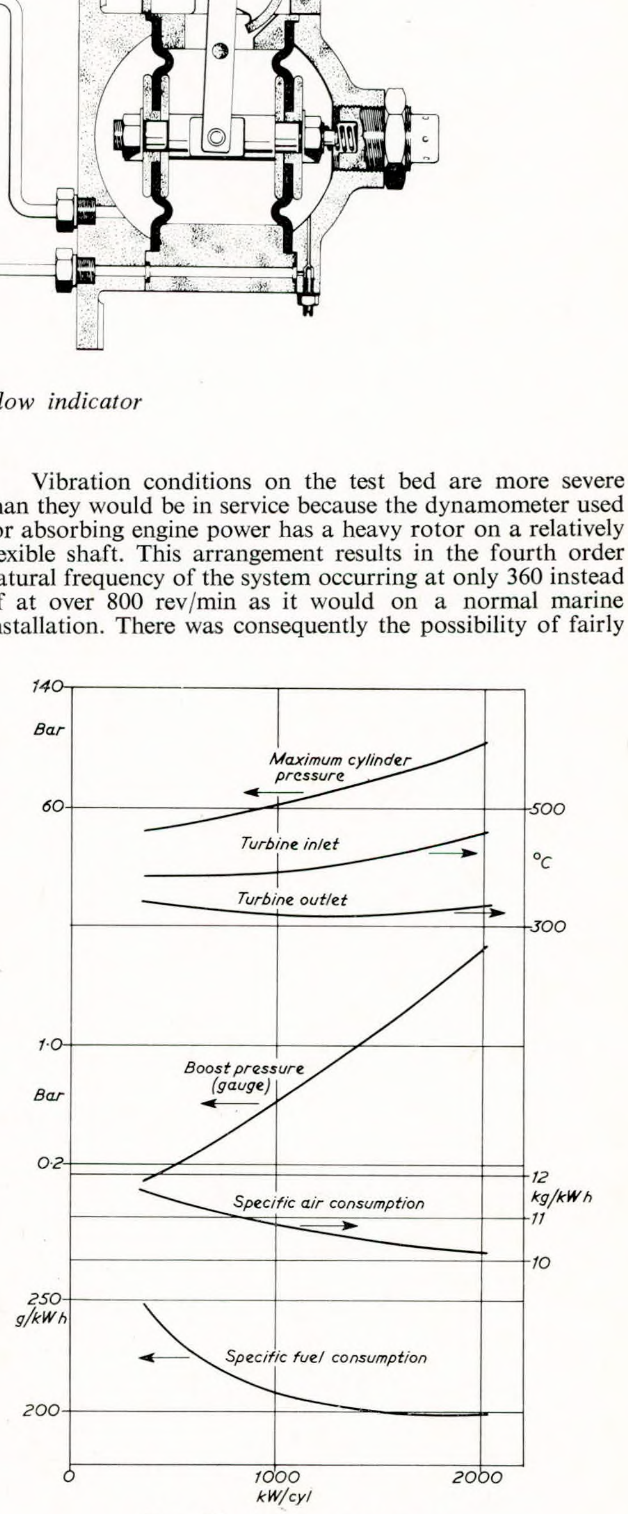

FIG. 15—*Performance curves*

heavy torsional movement at the running speed of 300 rev/min.

Vibration records were taken at the forward end of the crankshaft and on the balance shaft driven by the auxiliary gears. The measurements demonstrated the success of the arrangement in that the crankshaft twist was reduced substantially from its potential value and no measurable movement was transmitted to the auxiliary drive system from about 100 rev/min up to full speed. At the still lower engine speed of 70 rev/min there was a torsional movement of the balance shaft corresponding to the natural frequency of the coupling itself. This frequency has since been moved to well below the running speed range by fitting lighter springs in the detuner coupling.

The effectiveness of the coupling in isolating the auxiliary drive has also been evidenced by the silent running of the straight toothed auxiliary drive gears.

CONCLUSION

The objects of engine development testing are threefold: the first is to attain the best possible performance of a new design; the second to ensure reliability by subjecting the prototype to tests more severe than those likely to occur in service, and redesigning any components which show weakness until the weakness is eradicated; the third to ensure that every operation required for maintenance and overhaul has been carried out on the test engine without difficulty. The performance curves in Fig. 15 show that the first objective has been reached. The log book of inspections made during the running period has established the third objective and the engine is, at the present time, running a 500 hour test at full load on heavy fuel, to demonstrate reliability and to determine wear rates.

ACKNOWLEDGEMENTS

The author would like to thank all his colleagues who have made the paper possible by carrying out the design and development work described, and the many friends amongst consultants, oil companies, and engine and component manufacturers who have contributed to the solutions of problems which arose.

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Discussion

MR. A. HILL, F.I.Mar.E., began by thanking the author for his most interesting and lucid paper.

He said there was always a fascination in being able to see the evolution of a new machine through its various phases of development. So often the final solution looked so elegant and obvious that one was tempted to wonder why it was not used in the first place, but this seemed to be an inevitable part of any development work, and those who had been engaged in the design of plant knew the experience only too well. It was to the credit of the author that he had exposed so many of the development problems and shown the steps involved in arriving at a solution.

It was interesting to see that some of the redesigned components not only gave improved performance, but appeared to be simpler and, therefore, less costly to produce than the original version. This led once again to the general reflection—and this was not a criticism of Mr. Butler's work or his colleagues'—that perhaps not enough thought was put into details in the design stage. All too often pressure was exerted to proceed with production before the design alternatives had been investigated and evaluated. It was really necessary to prepare alternative designs and evaluate them from every aspect—functional efficiency, reliability, production cost, ease of maintenance etc. It was a common failing and often repeated, in spite of the knowledge of what happened.

The author had mentioned that the auxiliary blower flap valve was automatically controlled: was this simply by the differential pressure, or was there a mechanical control through the spindle?

The engine mounting was a bold change from orthodox practice, and it was very encouraging to see that it had fulfilled all expectations.

The author mentioned that the discovery of a drop in level of 1·3 mm at the forward end of the crankshaft was found to be due to foundation subsidence and that the alignment of the main bearings had remained perfect. It was no doubt a pleasant relief to see that the real cause was that the foundations had subsided. Would the author please describe the method used for checking the bearing alignment?

The use of conical dimples to measure cylinder lining wear was an elegant idea: what was the diameter of the dimples used, and what was the ratio of depth to diameter, or conical angle?

The author had mentioned an ingenious use of tilting pads to guide the lower piston skirt. Presumably this device would no longer be necessary where balanced crossheads were fitted, and therefore would not be fitted to the production engines. Could the author confirm this?

The author's remarks on bearings served to emphasize once again the extent to which the behaviour of highly loaded bearings was determined by external influences. As a bearing manufacturer, Mr. Hill said it might even be claimed that bearing behaviour was a monitor on the general design of the surrounding components.

He said it was interesting that reducing the fore and aft stiffness of both the centre and side connecting rods had been successful in keeping the bearing misalignment within limits tolerable to the bearing. In this context it might be of interest to point out that providing a bearing with a stronger, and, therefore, generally harder lining, would probably not necessarily solve the problem if misalignment were present.

It might be mentioned that the side connecting rod inner half bearings showed some signs of fatigue cracking at about 45° to the crown, due to the flexure of the rod eye. This was a not uncommon phenomenon in bottom end bearings. The modified design of rod had overcome this problem.

The addition of the oil reservoirs in the cap of the centre connecting rod bottom end was an ingenious way of supplying oil during the period when the normal supply was interrupted by inertia forces. It would be important to ensure that the holes were kept clean during servicing periods when stripped for maintenance. Being blind holes, people might think it did not matter if dirt got into them.

The author had mentioned roughening of the side top and bearings where they were running on cast iron pins.

This was not an uncommon occurrence. When nodular cast iron had been adopted for automotive crankshafts, there had appeared a serious wear problem. After considerable investigation, the cause was found to be due to the method of grinding the shaft journals. When this material was ground, and due to the discontinuaties in the surface caused by the graphite inclusions, the drag of the grinding wheel caused filaments, or whiskers, to be formed at the edge of the inclusions, so that the resulting surface was rather like the pile on a carpet. It was found that the grinding process was raising the pile in a direction opposing the motion, so that the pile would bite into the surface of the bearing. The solution was to arrange for the griding wheel to rotate in the same direction as the normal rotation of the shaft, so that the pile was trailing, and in this way the problem was overcome.

This solution would not of course be applicable to a crosshead bearing where the motion was reciprocating. It was probable that the pile could be removed by a super finishing or lapping process, providing it was done against the pile, otherwise the pile would just lie down.

Finally, Mr. Hill said, there was one aspect of this engine which had not been covered by the author but which nevertheless deserved mention. He believed it was a fair comment to say that this was a novel engine of advanced design which bade fair to combine the advantages of both low speed and medium speed diesel engines. Credit must be given to Messrs. Doxford Engines and Hawthorn Leslie, who had financed and carried out this project with only minimal assistance from Government sources.

Mr. E. P. Crowdy, V.R.D., M.A., F.I.Mar.E., said the Seahorse saga had started way back in 1968, as many knew. Towards the end of 1971 a paper[1] had been presented to the Institute of Marine Engineers describing the engine and its underlying philosophy, which had coincided with the commencement of full scale testing on the prototype engine.

Over three years had now elapsed since the prototype engine first ran, and in his paper tonight Mr. Butler had described how the technical problems which had arisen during testing had been overcome.

Mr. Crowdy said he would like to emphasize that the purely technical problems which Mr. Butler had described would in all probability have been overcome in a much shorter period of time if the project had enjoyed continuous administration by the original parent bodies. However, as they probably knew, one of the partners had undergone two changes of ownership during this period. The inevitable sense of insecurity during the transitional periods had made objective planning difficult and, at times, the identity of interest of some of those concerned had become obscured.

It was perhaps rewarding to review in retrospect the management of a major project of this nature. Mr. Crowdy said he used the word "major" advisedly, since the total expenditure on the project was comparable with the gross annual product of the sponsoring engineering concerns, and the single prototype alone represented approximately half a million pounds in today's currency and could be seen to represent a vital element in the organization's future prosperity.

The hazards of concentrating development into discrete step improvements, identified with single prestige prototypes could not be over-emphasized. If the magnitude of the project in relation to the resources of the promoters was such that only a single prototype could be contemplated, it was essential that wherever possible all innovative components should be subject to rig tests.

The original deliberate decision not to build a single-cylinder prototype had been taken on what, at the time, seemed to be sound reasoning, namely, that the extra design effort necessary would be diversionary and that the experimental results would be of limited value because the turbocharging and air flow arrangements would be so unrepresentative.

As far as the Seahorse project itself was concerned, it could never be proven as to which line of advance would have been most expeditious. However, there was now little doubt in his mind that the most prudent approach on a project of this nature involved the maximum possible devolution.

As regards technical development, he thought it fair to say that when the Seahorse engine project had been originally envisaged, the anticipated problem areas were concentrated round air flow and thermal stress. It would indeed be unfortunate if the remarkable success of the design in these respects should be in any way tarnished by odius comparisons with other engines and the time taken to overcome the piston ring problems.

Though the mean effective pressure of the Seahorse engine at 10·7 bar (10·9 kgf/cm^2) was conservative and represented but a modest increase on previous satisfactory practice, the increase in both piston speed and mean cylinder pressure might have instigated an earlier start on the necessary research programme, had the parent companies been aware of their proximity to a problem area. Subsequent research had revealed that many of their competitors had already become deeply involved in those problems, but had been remarkably successful in preventing their dissemination.

He said it was interesting to compare the exploitation of piston rings and journal bearings, both of which were used with complete success for many years before anyone had been able to advance a satisfactory explanation of their *modus operandi.* For years, journal bearings had been designed in accordance with empirical formulae in which the controlling parameters were totally misapplied, and it was not until Mitchell and Kingsbury had developed the convergent film thrust bearing that a true appreciation of journal bearing performance had been achieved. Likewise, Ramsbottom piston rings had now been successfully used for close on a century and an appreciation of their functioning was only just beginning to emerge.

For some time now the running-in period had been recognized as crucial, and the development of contour faced rings had gone some way to recognize the critical difference between worn and unworn rings. The face of the ring in contact with the cylinder liner was, however, not the only one subject to wear, and it was possible that the key to some of the piston ring problems currently experienced in high performance engines arose from the rigidity and wear resistance of the piston ring lands.

However, if the ring geometry was such that the ring was not torsionally constrained, it was possible that the operating régime could be so organized that an adequate oil film between the ring and liner was maintained throughout the cycle, despite the increasingly hostile environment of high performance engines.

He did not think the importance of Mr. Butler's contribution to the analysis of piston ring performance, as regards both the attainment of satisfactory ring profiles and the maintenance of adequate oil film thicknesses, could be overestimated or overemphasized. In fact, he thought that the author's contribution to the development of high performance engines in general would reach far beyond the satisfactory conclusion of the Seahorse programme.

Mr. J. H. Wesselo, F.I.Mar.E., said that his company had chosen the easier way of four-stroke engines, but for the same and even higher powered brackets in the market, they were still very much intrigued by the possibility of the two-stroke, opposed-piston principle, as he had stated previously in a contribution to the earlier paper by Butler and Crowdy[1]. Following the author's introduction today, however, his contribution was shorter than the earlier one; this maybe was due to the fact that their two-stroke experience was three years more in the past and he had made points much the same at the time.

As his company had been building uniflow scavenging two-stroke engines with exhaust valves, exhaust pistons were unknown, and these were considered to be the most risky component of an opposed-piston engine. In the Seahorse engine the exhaust piston had half the stroke of the bottom one and was water cooled, so direct observations were not conclusive as to the degree of difficulty. In his view, the pessimism about exhaust pistons was exaggerated, as heat load was more influenced by the high pressure and temperature phase than by the gas exchange phase of the cycle. Had the author taken heat flux measurements to compare upper and lower pistons?

He did not like to bother the author with a typical crosshead engine problem, but wanted to know the quantity of crankcase oil used in the new design. He knew a little more since he had heard Mr. Butler say it was a negligible quantity.

Why did the author's company dispense with the small fuel injection pump being used as a means for accurate and reliable cylinder lubrication?

In an old publication this had been mentioned; Mr. Wesselo's company had applied the principle to two-stroke trunk piston engines and it had been one of the means that had helped to solve ring scuffing problems.

Mr. Wesselo wished the author and his company much success with further development. But if difficulties arose, would he please keep in mind that the piston rings in the four-stroke engines produced by Mr. Wesselo's company were subject to a higher mean piston speed and, notwithstanding this high speed, were expected to travel not 15 but 20 times round the Equator.

MR. D. CARTER said that viewing the picture worldwide, starting in this country, they lived primarily in a four-stroke world. Certainly this was true of British development of diesel engines, with one important exception. In this worldwide picture, on the other hand, they had the dominating position of the big slow speed direct coupled marine engine, which was invariably two-stroke, built under licence in a large number of countries. Next in the worldwide picture, he would mention the fact that General Motors of America were the biggest makers of diesel engines in the world, maybe approached by Cummings, but they covered a wider field.

So, in spite of the domination of the four-stroke engine in this country as a whole, it was worth remembering that worldwide this was offset by the tremendous activity in the big, slow speed marine engine design, in countries other than Great Britain, to manufacture in many countries, plus General Motors, a smaller, but nevertheless sizeable, engine.

It was therefore in this picture, important for those in Britain to salute and recognize the contribution which Mr. Butler and his team in particular were making, and in which he was sure they all wished him and his team a great success, also they admired the pioneering effort, which too should be recognized.

Mr. Butler had dealt with various problems on the test bed: there were two kinds of problems on engines, the first group concerned problems on the test bed. When they had solved this, the next generation of problems started when they put the ship to sea. It was vitally important that both lots of problems should be mastered before large scale production commenced. They had all seen the damage done to the two-stroke engine cause, by engines being put on the market, two-stroke medium speed engines, test-bed tested, but bulk orders for service work being taken before even a single ship had been engined. A word or two from Mr. Butler on how his company would measure up to that would help the future success of the engine.

Mr. Butler had mentioned the problem of piston ring scuffing. This was at the heart of the troubles of any engine which was pushed up beyond the frontiers of hitherto known knowledge. One of the solutions which had been mentioned, but not yet proved, was that on a two-stroke engine burning heavy fuel, if there was a high rate of cylinder swirl or good combustion, this might throw heavy ash deposits on to the cylinder liner wall, setting up an abrasive action which would give an inherent problem. This was only a theory, and not proved, but there were engines in service with unsolved problems in this field.

Mr. Carter fully appreciated the work already done by Mr. Butler and his colleagues, and described, whereby piston rings could operate differently from the way they had done by the shape of the grooves and the way the ring could tilt, but if Mr. Butler could add a word of reassurance to engineers that the Seahorse engine would not have this kind of problem which had occurred on other engines in service, it would be appreciated.

MR. A. J. S. BAKER, F.I.Mar.E., in a contribution read by Mr. A. E. Franklin, F.I.Mar.E., said that the work described by Mr. Butler was of far greater importance than a recital of development problems inevitable to any new type of engine. It covered the basic development of a new breed of machines and in years to come might well be regarded as a watershed in the evolution of large prime movers. At the same time it could even now be seen as the outpost of British skill and endeavour in a colourless scene of current combustion engineering in this country. Mr. Butler and his collaborators such as Mr. Crowdy should, therefore, be recognized as courageous upholders of British tradition in marine engineering.

Despite some signficant disappointments in recent large two-cycle engines, Mr. Baker felt that the basic concept of the Seahorse engine offered every opportunity for it to reach an effective level of development in the near future, for it to find ready acceptance at sea and in many land based applications.

He made this prediction in the knowledge that there were three prime areas of difficulty associated with large two-cycle machines. These problem areas could be summarized as:

1) high thermal loadings compared with equivalent four-cycle engines;
2) lack of design criteria for two-cycle piston rings;
3) burning of exhaust valves with low cost fuel.

Concerning high thermal loading, experts usually concluded that heat flux measurements conducted in loaded engines operating within the smoke limit were higher in two-cycle than in four-cycle engines. However, what this really meant was that, provided the trapped charge density was sufficient for complete combustion, smoke would not be emitted even though total heat rejection to the chamber walls might be excessive. He was now quite convinced that in the two-cycle engine, at any rate, this was by no means the complete conclusion.

Usually, such heat flux experiments were performed in engines which, for valid reasons, provided little or no adjustment of the total weight of air passing through the engine. If, however, similar tests were performed in an engine possessing facilities for varying air throughout, it would be found that heat fluxes varied inversely with air throughout.

Mr. Baker had demonstrated this effect in a series of experiments on his company's laboratory two-cycle crosshead engine, Abingdon B-1, and appended the results to an IMAS 1969 paper at which he had shown a slide (see Fig 16). Subsequently, he had related his observations to some data presented by the late Hugo Scobel in another paper at the same conference.

Now, clearly, Mr. Butler would not be able to emulate the complete range of air/fuel ratio possible with the laboratory engine, in his large prototype. However, Mr. Baker believed that his observations could be of assistance in formulating advances if thermal loading became a problem. After all, increasing airflow from turbocharging should present less challenge than many of the developments already described.

Turning to piston rings, Mr. Butler had been quick to note the supreme importance of the radii of curvature developed on the face of a piston ring. Upon this depended the value of oil film pressures generated between the ring and liner and the wellbeing of the loaded surfaces.

As conceived, the normal Ramsbottom piston ring possessed immense powers of self-compensation as had been demonstrated in the work cited by the author. However, where these compensation factors failed, there

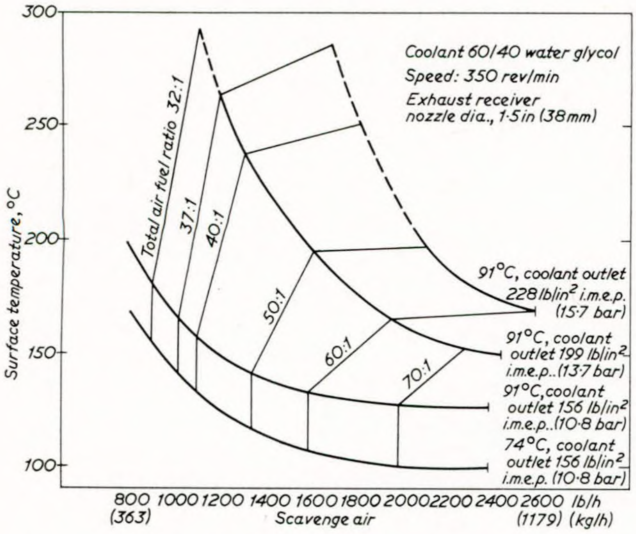

FIG. 16—*Abingdon B-1 liner surface temperatures—Opposite top ring at TDC*

appeared little that could be done in the shape of material modifications to avoid scuffing. It therefore became necessary to look again at the dynamic environment of the piston ring to try to find an easier situation. Mr. Butler's solution of the ramped groove to transfer ring guidance from the piston groove appeared eminently sensible and practical. Mr. Baker also gathered from a very recent conversation with him, that this modification appeared to produce rings with significantly increased radii of curvature on their sealing faces.

The ramped groove might well be criticized as being likely to wear rapidly, but he personally felt this to be most unlikely, especially if coated by modern chroming techniques, since the Hertzian load would be no higher than that which existed cyclically between the ring and the extremity of the groove in the conventional arrangements. In fact, the position of the zone of maximum load on the groove face was better supported with the ramp and might thus offer added fatigue resistance to the coating.

With regard to burning exhaust valves, Mr. Butler said that to eliminate the exhaust valves from an end-scavenged two-stroke, such as Seahorse, appeared to be an eminently desirable move. Doxfords had demonstrated quite clearly that the piston-controlled exhaust port was an entirely viable alternative. Moreover, while there now appeared avenues of logical development for piston rings, he failed to discern parallel activity in connexion with poppet valves. Simultaneously, most predictions of low cost fuel quality continued to be pessimistic in those directions relative to exhaust valve life.

In this short contribution, Mr. Baker had endeavoured to highlight the development freedom opened by Mr. Butler, in his Seahorse, to the difficulties confronting the two-stroke designer. He believed this engine to be a potential world beater in its class and the class of engine which it eventually represented would be much wider than was currently apparent. He wished Mr. Butler and Hawthorn-Doxfords every technical success, when their commercial success could only follow.

MR. A. J. WICKENS said that he would like to remind the author of a few other modifications which had been made to the Seahorse engine in the course of development. The first concerned the means for providing cooling water to the upper pistons. Members would remember the link arrangement shown in the original paper, in which two swinging links were arranged on opposite sides of the engine, one extending forward and the other extending aft. This had produced a problem because the inertia of the links had tended to impose a twisting moment on the upper pistons which could only be resisted by the side rods. One means of avoiding this was to swing one of the links round so that they both extended the same way. This converted the turning moment into a longitudinal force which was resisted by a reaction between the upper piston assembly and the liner wall. This was still undesirable so a modification was carried out on the swinging links to greatly reduce this inertia and to reinstate the orginal opposed arrangement. This had been successful, but plans were in hand to reorientate the swinging links to eliminate completely any horizontal unbalanced forces or moments.

By way of illustration of a detail Mr. Wickens said he would like to mention a minor problem with the guide bars. The philosophy of the construction of the engine was that there would be an accurately machined guide face which would form the ultimate register for all reciprocating components on the engine. On to this guide face were bolted the guide bars, but there had been some difficulty in securing them. It was particularly important to avoid any displacement of these members on the Seahorse engine, because the lubricating oil supply to the centre running gear was fed from, or to, the guide bars and any opening out increased leakage which could result in oil starvation. The matter was, however, finally dealt with by substituting fitted bolts for the previous arrangement of dowels.

As a further precaution a transducer was fitted into one of the bars to check that the shoe remained against its working face all the time.

One other important modification which had not been mentioned was applied to the cylinder liners. The drilled cooling arrangement around the combustion chamber belt was altered to provide 48–20 mm bore holes instead of the 56–16 mm bore holes originally used. This had given a better thickness of metal in the vicinity of the valve pockets and generally resulted in more satisfactory thermal conditions.

MR. G. VICTORY, Vice-President, I.Mar.E., said that Fig. 14 showed the coolant flow indicator; he assumed that it was in the main lubricating flow line. He would have thought a 0·6 bar drop across the orifice on the main lubricating oil cooling system was rather high, and would put up the power requirement of the lubricating oil pumps appreciatively. Was it necessary to have it as high?

Looking at the fuel pipe arrangement in Fig. 3, he wondered whether there had been any requirement that these pipes should be screened or fitted with double annular pipes to avoid spraying oil on to hot surfaces. This was usually a requirement for UMS installations.

No mention was made of the prevention of crankcase explosions. Had they provided crankcase monitoring arrangements, which were very necessary in UMS engine rooms.

It was very nice to see fashions change: it was a good thing when difficulties were experienced, to look at how things were done before. In this respect he was interested in the fuel system, which was rather similar to the old Doxford engine. In particular, the new air starting valve mechanism, shown at the top of Fig. 4 looked very familiar to old hands. He was very pleased to see Mr. Butler had retained an air-assisted arrangement for pulling the roller down on to the cam. In the first Doxford he had been on, this had had to be done manually by pulling a large hand wheel round against the resistance of the air starting valve. This was almost a "strongman act" and could well be why he had attained his present stature.

The CHAIRMAN (Dr. J. Cowley, F.I.Mar.E.,) said he had two questions. The first concerned the fuel system where

* A. J. S. Baker and G. L. Johanes. 1970. "Small Scale Test Engines designed to Predict Lubricant Requirements in Large Bore Marine Oil Engines". The Institute of Marine Engineers, *Proc. Imas* 69, Section 4(c)—Main Propulsion Machinery, p.4c/54.

the design included a short length of small bore high pressure pipe between the distribution block and the injection. Did not the introduction of the flanged joint (to facilitate removal of the injector) provide an additional potential source of leakage, i.e. a joint in addition to the joint at the distribution block and to the connexion to the fuel injector?

A number of serious fires had occurred due to leakage of high pressure joints and pipes, and he would ask the author whether any problems had been experienced with fuel piping failures.

Secondly, concerning noise levels, how did the Seahorse engine compare with other designs of medium speed engines?

Commander E. Tyrrell, R.N., F.I.Mar.E., said he could not agree with some of Mr. Crowdy's remarks concerning lack of information on experiences with engines developed in part at Government expense. The information was available and as far as he knew this had been made known to Mr. Crowdy.

Also Fairbanks Morse, who had been developing a similar engine which had been withdrawn, had supplied information on the difficulties they had experienced together with the solutions to these difficulties in certain cases.

The Government had often been criticized for not having provided sufficient finance to the marine engineering industry in this country to enable it to compete adequately in international markets. He wished to read a Press Notice issued by the Department of Industry at 10 a.m. that morning (10 December):

"Press Notice ref. 264 December 10, 1974
DEPARTMENT OF INDUSTRY
1, Victoria Street, London SW1H 0ET.
Tel. 01-215 3789

SEAHORSE MARINE DIESEL ENGINE

A contract has been placed by the Secretary of State for Industry as sponsor of the UK engine manufacturing industry with Doxford Hawthorn Research Services Limited for a 500 hour cumulative full power trial of the Seahorse marine diesel engine. The contract, which was placed through the Secretary of State for Defence, provides that the Department of Industry will bear half the cost of certain engine tests up to a maximum contribution of £110 000.

The Seahorse is a geared medium-speed, opposed-piston, crosshead diesel engine rated initially at 2500 horsepower per cylinder. A four-cylinder prototype is under test. It has been developed by Doxford Hawthorn Research Services Limited on behalf of Doxford Engines Limited and Hawthorn Leslie (Engineers) Limited. It is hoped that the engine will complete its 500 hour full power trial before the end of February 1975.

NOTE TO EDITOR

Doxford Engines Limited is one of the companies which were purchased by the Department of Industry from Court Line Limited earlier this year."

Commander Tyrrell continued that he was sure that all present would join with him in wishing the engine all success.

Correspondence

Mr. E. T. Kennaugh wrote that it seemed to him, judging from the author's response to the discussion which followed the reading of the paper, that he still had some doubts regarding the practicality of achieving a timed cylinder lubricating oil system by mechanical means using, one assumed, conventional lubricating oil quills, especially in view of the very small quantities of lubricating oil involved per injection. In fact, he did acknowledge the difficulties, and referred to some of the factors which affected an efficient realization of a really accurately timed cylinder lubricating oil system.

It was only after many experiments that Mr. Kennaugh's company finally achieved, under test conditions, a satisfactory timed lubricating oil system, and this was done by using a cylinder lubricator designed on the lines of fuel injection pump. This, in turn, required what amounted to small injection nozzles at each lubricating oil quill, combined with temperature control of the lubricating oil.

It was clear that the amount of accuracy required to produce the injection pump and the maintenance which would also be necessary to keep the apparatus in order, meant a system which was very costly and more complicated than practical. It was, therefore, abandoned, and research developed on the lines of the new accumulator system (see Fig. 17) now being incorporated in the latest design of engine, which achieved a timed injection of cylinder lubricating oil without any mechanical means, relying on the pressure in the cylinder to achieve this (see Fig. 18).

He submitted that the methods employed so far by many slow speed engine manufacturers who had claimed to have perfected timed lubrication, really referred to a timed mechanical drive, which did not necessarily mean that a really accurate timed injection had been achieved.

It could of course be argued that this might have been better than no timing at all, but it was felt that the claims put forward for so called accurately timed injection of lubricating oil in the past, had been rather dubious. Of

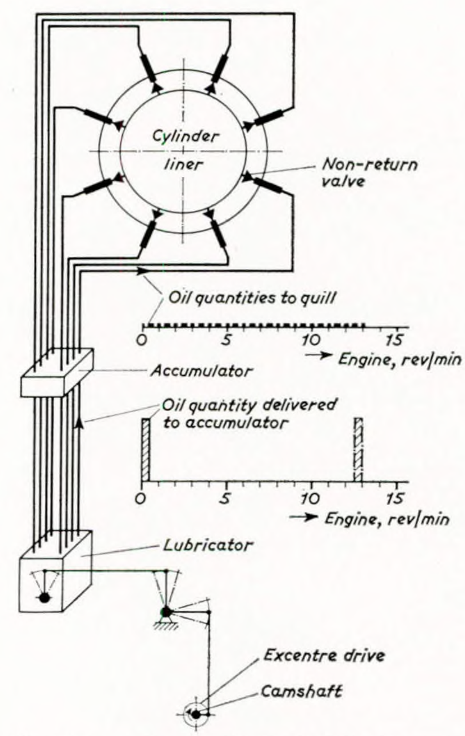

Fig. 17—*Accumulator cylinder lubrication system of RND-M engines*

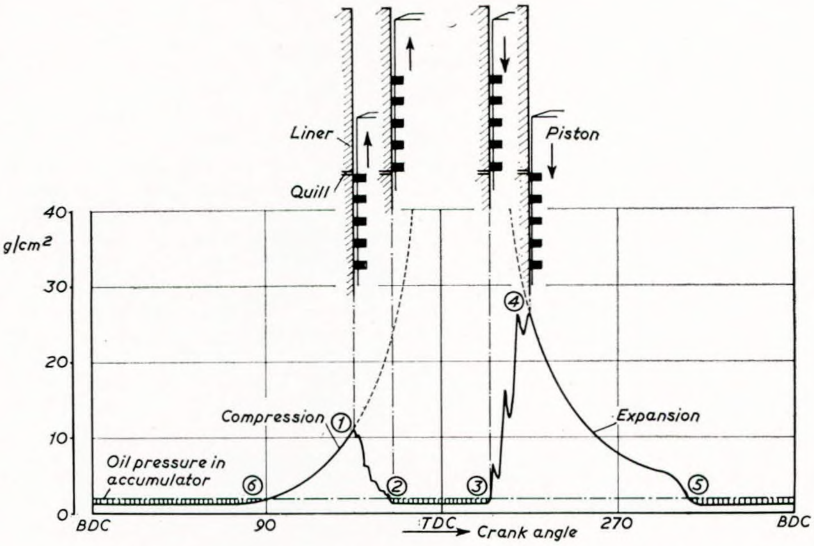

FIG. 18—*Pressure fluctuation at lubricating oil quills*

course the higher the engine speed, the more difficult it was to achieve accurate timing by mechanical means, as was acknowledged by the author in the paper.

It would be interesting in this context, if the author could give some details of the type of lubricating oil quill employed.

MR. P. MANSON, F.I.Mar.E., remarked in a written contribution, that in his contribution to the discussion on Mr. Bowers' paper* he had referred to the power rating (PME brake mep × CM mean piston speed m/s) and had related this to the specific air throughput, kg/bhp h, as a result of which some interesting facts came to light, where two, two-stroke medium speed engines having power ratings of 74·6 and 70·56 had air throughputs of 6·35 and 8·0 kg/bhp h respectively, indicating that the engine with the higher rating had a much lower air throughput than that

* Bowers, N. K. 1975. "Medium Speed Diesel Engines in Bulkcarriers." *Trans.I.Mar.E.*, Vol. 87.

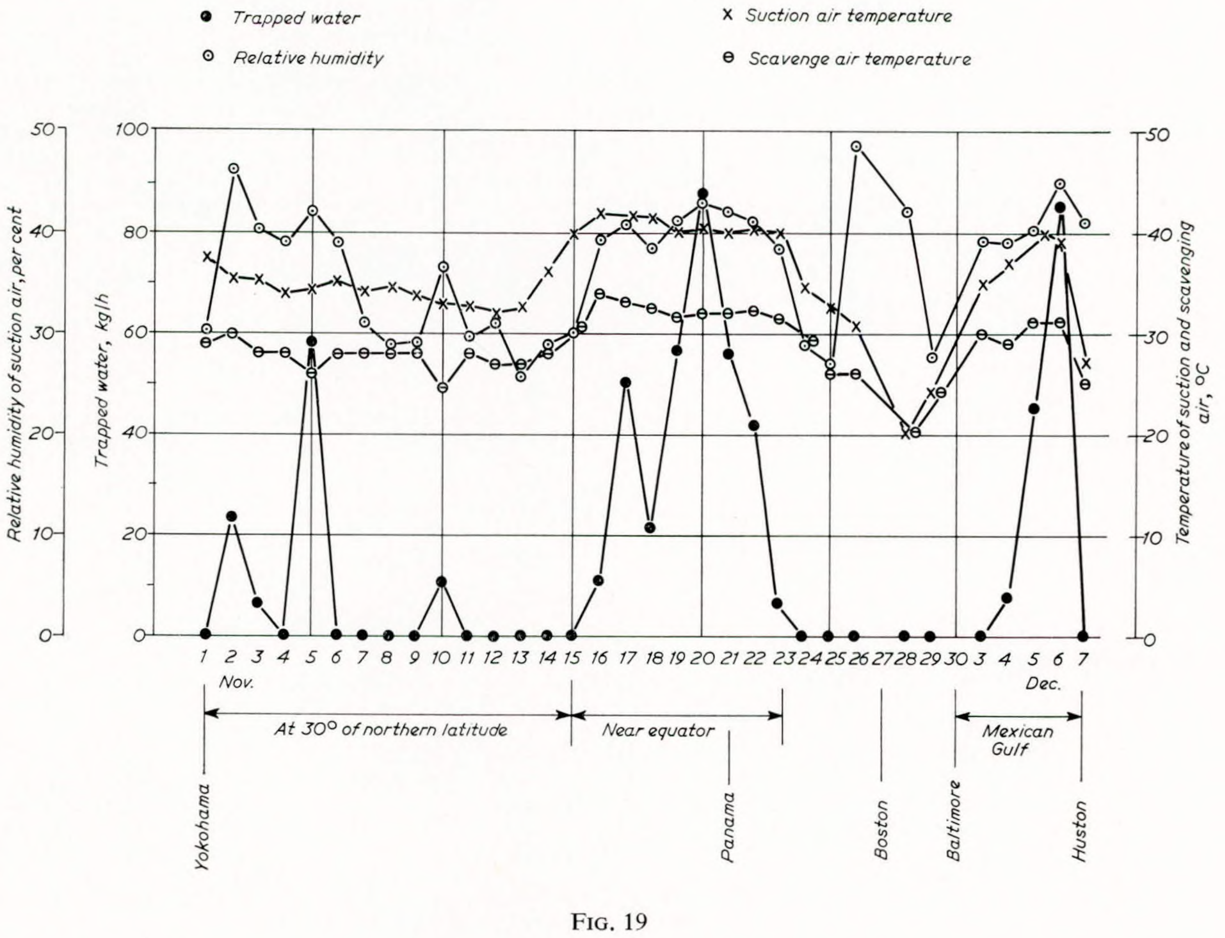

FIG. 19

of the lower rating, and in turn led one to believe that the higher rated engine would be subject to higher thermal loading, with its associated problems.

It appeared to him that a good many of the problems one got to know about from time to time, of excessive cylinder wear and piston ring fouling, were partly due to lack of scavenge air, resulting in excessive gas temperature, and carbonization of the cylinder lubricating oil. The power factor of the Seahorse engine by his calculations worked out at 95·9; this was based on a mean piston speed of 8·8 m/s. The scavenge air throughput was 7·6 kg bhp h, and he would appreciate Mr. Butler's views on the scavenge air supply in relation to the Seahorse engine.

Two leading marine engine makers had comparatively recently started to fit water mist separators to the scavenge air system. With the higher mep and ever increasing scavenge air pressure, the air delivered from the turbocharger had to be cooled through the air cooler before being delivered to the engine. Very often the air temperature was controlled in order to keep the charging air temperature above the dewpoint. However, there was a limit to this type of control, otherwise the exhaust gas temperature became rather high as a result of the increase in the air temperatures.

From figures published covering a six-cylinder B & W 6K74EF engine developing 85 MW (11 600 bhp) the effect of relative humidity in different areas of the world could be seen (see Fig. 19, published in the Bulletin of the Marine Engineering Society in Japan, Vol. 2, No. 1, 1974). A further report could also be seen in a paper by Amitani and Omotehara†. Further more, it had been proved that water droplets had an adverse effect on the engine, such as abnormal wear of piston rings and liner, and in fact increased cylinder lubrication was sometimes necessary. The author's views on this aspect would also be appreciated.

He had been privileged, from time to time, to see the Seahorse engine under working conditions and on his last visit it was behaving very well with practically no vibration. When standing on the top platform, if covers had been fitted over the upper piston running gear, the noise of the turbochargers, common to all diesel engines, would practically be the only means of telling whether the engine was running or not.

He congratulated Mr. Butler and his staff on the way they had faced up to the various problems inevitable in a completely new design of medium speed engine of opposed piston type. By a process of elimination, as a result of thorough testing and investigation, the engine was now achieving its designed output. One example of the development work was the elimination of the mechanically driven blower, as indicated by Mr. Butler, and one wondered if there was not still room for improvement in the turbocharger efficiency to achieve yet a higher air throughput to the engine.

Mr. Butler's problems had not always been of a technical nature, and Mr. Manson sincerely hoped that in the months ahead he would be able to achieve all his test programmes on schedule.

† Amitani, T. and Omotehara, I. 1973. "Some Experience with UE Type Diesel Engines." ISME Conference, Tokyo.

MR. K. A. LLOYD, B.Sc., and DR. R. W. WILSON, M.A., in a joint written contribution, remarked that their attention had been drawn to comments by the author concerning the accuracy of temperature determination by "Templugs". The latter could only give accurate results when used under particular, defined test conditions. In essence, the time that they were exposed to the steady state maximum temperature condition must be at least a quarter of the total running time, and preferably more. Many of those used in the Seahorse engine were not employed under these conditions. The author and his colleague, Dr. Orbeck, were advised of their concern several times in the course of the engine development work, and were invited to discuss the results. Unfortunately, owing to pressure of other work, they were unable to accept, and as a result the writers felt that their conclusions about the general accuracy of templugs, based on these particular measurements was misleading. When used under specified conditions they are capable of giving an accuracy of ± 5 deg C, i.e. much better than the ± 30 deg C cited by the author, as many users had proved for themselves by making controlled tests in laboratory furnaces.

In this case it was possible that the unsuitable test conditions were responsible for the differences of up to ± 30 deg C between temperatures determined by thermocouples and by templugs. However, when such a discrepancy occurred it was unwise to assume that thermocouples always provided a correct indication of temperature and that templugs must be wrong. As had been pointed out elsewhere, the structural modifications associated with the fitting of thermocouples often upset local heat flow far more than did the use of templugs, which could give more realistic values in these circumstances.

With regard to templugs reading high at the lower temperatures, they could only repeat what they had said previously elsewhere. A templug recorded the temperature to which it is exposed and the only way to soften the hardened steel from which it was made was by heating. If when fitted properly and used under specified conditions they recorded consistently high temperatures compared to thermocouples, then there must be a reason. The method of fitting the thermocouples, their adequacy of thermal contact in the region of the hot junction and the general instrumentation of the thermocouple system should be checked. The possibility of local heat "soak-back" to templugs on shut-down should be considered.

References

1) WILSON, R. W., Discussion on paper by C. C. J. French, *Proc. I. Mech. E.*, 1969–70, Vol. 184, p.540.
2) WILSON, R. W., Discussion on Session 3, I. Mech. E. Symposium on Thermal Loading of Diesel Engines, *Proc. I. Mech. E.*, 1964–65, Vol. 179, part 3C, p.157.

Author's Reply

Mr. Butler, in reply, said he would like to thank all the contributors for their kind remarks and for their contributions.

In answer to Mr. Hill's question, the flap valve had a mechanical control pneumatically operated. In the condition obtaining at 40–50 propulsion rev/min when the turbocharger pressure was nearly equal to the auxiliary fan pressure, a simple non-return valve would be liable to flutter and possibly initiate turbocharger surge.

For checking main bearing alignment, the engine was designed with openings in the main bearing girders in which sights could be placed which had plungers up to the crankshaft. These were used with a telescope.

Conical dimples were 4 mm diameter and they were right angled so the depth was half the diameter. References 4 and 5 described the method.

The tilting pad piston skirt was still in use on the prototype engine. Now that alignment problems had been overcome a simpler design would be tried.

Mr. Crowdy's extension to the paper was very helpful. Air flow and thermal stress had been expected to present problems since the power per square centimetre of cylinder bore was higher than any other comparable engine on the market. The development of the finally successful piston ring arrangement had been a team effort in which Mr. Crowdy himself had played a major part and to which many other colleagues had contributed.

To Mr. Wesselo, he said that it would be idle to pretend there were not problems with exhaust pistons passing over ports. Piston rings could break and the broken pieces could damage the turbochargers. Knowhow which came from experience was needed to design ports to avoid ring breakage, since it was always a matter of compromise between getting maximum possible port area and sufficient support for the rings. On the Seahorse, they had had no ring breakage problem, possibly because they had been over-careful.

The temperature fields in both upper and lower pistons had been measured. The upper piston had a higher heat flux than the lower piston which was partly cooled by scavenge air, but this was balanced by the better heat transfer characteristics of the water cooling in the upper piston compared with the oil in the lower piston. The tendency on this engine and previous engines had been remarkably similar; the general level of stress was just about the same in both pistons.

The small fuel injection type pump was still used in conjunction with the hydraulically operated distributor. With a normal system, with one pump per quill, the volume of oil delivered by each pump at each revolution was only between one-sixth and one-tenth of the dilation of the connecting pipe, so accurate timing was impossible. This problem was overcome by the distributor system in which nine times as much oil was delivered to each quill every nine revolutions. The hydraulically operated distributor simplified the system by reducing the amount of connecting piping needed.

Mr. Carter's contribution was most valuable in reminding listeners that although the four stroke engine tended to dominate the medium speed field in this country, the opposite was the case in America.

The author was fully appreciative of the problems that could arise at sea after satisfactory test bed performance had been proved. With conventional chocking arrangements, the crankshaft alignment of marine propulsion engines changed severely between hot and cold conditions and, in addition, the bedplate suffered distortions due to ship movement. Both these factors contributed to bearing damage in service, but were eliminated by the Seahorse four-point mounting which permitted the engine to expand independently of the cold double bottom and isolated the engine structure from hull bending.

There were other problems which could arise due to service conditions, but the builders were striving their utmost to eliminate these by prolonged testing under the most arduous conditions available and by deliberately maltreating components to ensure an adequate reliability margin. After the engine became proved on the test bed, it was the companies' policy to instal only a limited number of the new engines in ships until service experience had been obtained. In addition it was intended to continue running the prototype engine on the test bed at higher power than that authorized for service use.

Not only to avoid ash deposit on the cylinder walls, but also to reduce heat loss to pistons and cylinders, the Seahorse engine was designed to operate with minimum swirl. The provision of four peripheral fuel injectors helped by permitting low swirl and by avoiding the danger of a faulty injector spraying liquid fuel on to the cylinder wall.

The author regretted very much that Mr. Baker could not be at the meeting to present his valuable contribution and, equally, that he himself could not attend the Mechanical Engineers meeting at which Mr. Baker was at the same time describing a very advanced automotive engine.

Mr. Baker's graph (Fig. 16) showing the results of varying air flow on the Abingdon B1 engine demonstrated very clearly the importance of maintaining a low fuel to air ratio in high output engines. There was a limit to what could be achieved by intensive indirect cooling and the only other avenues open to the designer were to use materials capable of withstanding higher temperatures and temperature gradients, or to increase air flow to reduce cycle temperatures and provide direct cooling. Unfortunately, the range of structural materials available was limited because those which could withstand high surface temperatures tended to have low conductivity and thus suffered higher thermal stress for any given specific heat flux. In the case of the lubricant, which was essential to separate moving parts, there was as yet no practical substitute for hydrocarbon oil which imposed severe temperature limits because of carbonization.

Applying Seahorse figures of air to fuel ratio 50:1, water outlet temperature 76°C and indicated mean pressure 11·7 bar (169 lbf/in^2) to Mr. Baker's graph suggested a liner wall temperature of just under 140°C. The measured temperature opposite the top ring at i.d.c. was 186°C and it was anticipated that this would be reduced to 155°C by modifications to the liner cooling arrangements. The remaining difference could be due to the larger size and greater power of Seahorse compared with Abingdon B1. The graph was important in demonstrating that reduced air to fuel ratio would increase component surface temperature quite seriously.

The work done by the author and his colleagues on piston rings was given impetus by Mr. Baker's own work on the subject and it was gratifying to report that tests so far had shown no sign of abnormal wear of the groove face. What had happened was a high polish on both the hump and the mating portion of the ring, indicating an exceptionally good seal and excellent lubrication. The Hertzian load between ring and land was actually considerably lower than occurred with a normal groove when the ring bore at its inner edge.

Mr. Wicken's contribution added some very useful information to that contained in the paper. Perhaps the author had treated some of the development modifications a little briefly because, on looking back, it seemed the problems should have been foreseen and thus not have occurred in the ironmongery stage. Designers were human and in the pressure of getting a new design off he ground some factors did get overlooked, but fortunately hindsight was better than foresight, which was why development testing was so valuable.

An additional detail of some importance about the guide bar history was that, at the same time as strengthening this attachment, the connecting rods were made more flexible so the subsequent transducer measurements showed the effect of both increased strength and reduced cyclical loading.

In reply to Mr. Victory's question about flow indicators, Mr. Butler agreed that it would be preferable if they would work with a lower pressure drop, but this would necessitate more flexible diaphragms which would have a shorter life. It was felt that reliability was the first consideration and since the pressure drop was less than the loss associated with an open system using sightglasses it was considered acceptable.

In the initial engine design it had been considered that the complete enclosure of the space between scavenge entablature and exhaust belt would afford sufficient protection for the high pressure fuel pipes. Later consideration, however, in the light of more rigorous government and classification society requirements, provoked a change of policy and arrangements were being made for screening these pipes and providing fuel leakage tell-tales.

The prototype engine was provided with an oil mist detection system. Future engines would have either this or a bearing temperature monitoring system.

When the air starting system had been redesigned, the designers had borne in mind that they could not rely on always having the engineers as strong and powerful as Mr. Victory.

In reply to the Chairman, Dr. Cowley, the author agreed that the latest fuel system did include two more joints per cylinder than the original, but he felt that these were justified by the greater ease with which injectors could be changed and by the better accessibility of the joints, as well as by the fact that the pipes to the injectors were all similar, so that the number of spares needed was reduced.

High pressure fuel pipe breakage did occur on this as on other high powered engines. It was unfortunate that the total world demand for high pressure pipes of the size required was not sufficient to encourage pipe makers to make them in high quality steel. Engine builders such as the author's firm did everything possible by designing with an enormous safety margin, but there were still occasional failures due to faulty material.

At present, the prototype engine on the test bed emitted a fair amount of noise from the inlet pipes leading from the air measuring nozzles. With the normal air filter-silencers fitted, the noise level was about the same as the best direct drive engines and considerably below that of most medium speed engines. Provision was made on the design for fitting sound absorbing covers to reduce the noise to a level at which it would be comfortable to work on one engine of a pair, with the other running at full power.

The author agreed with Commander Tyrrell that much information was available on piston ring scuffing. Unfortunately perhaps, when the Seahorse was designed the builders had never had any trouble of this sort. Although later it was found that other engine builders had had serious problems, it was natural that they did not advertise the fact. Once the problem was identified and even before Government help was available, both Ruston and Fairbanks Morse had been completely open and helpful.

It was kind of Commander Tyrrell to read the press release and both firms concerned were extremely grateful for the Government's help. The contribution so far of £111 000 represented about seven per cent of the amount spent on the engine and both companies hoped that this represented only the thin end of the wedge and that very much more would be available for future development.

Mr. Manson's criterion of power rating which was in fact 150 times the indicated power per square centimetre piston area, was a useful guide to the level of thermal problems which had to be dealt with in an engine.

The higher this figure the more intensive must be the cooling arrangement around the combustion area and the more important did it become to have adequate specific air flow.

Heat flux was also affected in large engines very seriously by air to fuel ratio and it had been found desirable on slow speed engines to keep this figure about 45 : 1. On the Seahorse, with its higher speed, the air flow of 7·6 kg bhp h resulted in a fuel air ratio of over 50 : 1, which gave a very satisfactory margin.

The vapiur pressure of water in air increased in a ratio of nearly 3 : 1 for each 20°C temperature rise in the range of ambient temperatures up to 40°C. In theory, therefore, with a boost pressure of 2·5 atmospheres absolute, it would be necessary to keep the entablature scavenge air temperature about 15° above ambient to avoid the possibility of droplet formation in the scavenge entablature. In practice due to the possibility of supersaturation and, perhaps, to air movement keeping the water in the form of a fine mist, a smaller temperature differential sufficed, as was shown by Mr. Manson's interesting graph (Fig. 19).

Water separators inevitably produced some pressure drop which reduced air flow and this had to be balanced against the effect of higher scavenge temperature. In the case of Seahorse, the designers had elected to maintain the entablature temperature at about 45°C to avoid the use of separators. The opposite decision might have to be taken for engines with higher boost pressure or with lower air flow making exhaust temperature more critical.

Mr. Kennaugh's contribution outlining some of the work done by his company in their efforts to achieve timed cylinder lubrication was most interesting.

It was agreed that any system designed to inject very small quantities of oil regularly into a cylinder required extreme accuracy in manufacture. Because of the compression of the oil and the dilation of the pipes, the metering accuracy required could not be achieved if the pump used had to deliver only the amount of oil required by each quill each engine stroke. It was for this reason that the author's company developed a distributor system in which a single pump delivered nine times the quantity required per stroke at each quill to each of nine points in turn. It was noted from Fig. 17 that in the arrangement described by Mr. Kennaugh each pump stroke delivered enough oil for about 13 engine strokes, but the system had the complication of using eight pumps instead of one. The author agreed with Mr. Kennaugh's submission that many so-called timed lubrication systems amounted only to timed mechanical drives. Apart from the difficulty described above, which was overcome by the distributor system, there were many other problems which had only been solved, in the case of the author's company, by intensive research and development. The large number of engines at sea giving very good liner wear figures showed that a properly developed timed lubrication system was well worth while.

The distributor system had fewer working parts than conventional arrangements. This factor, in combination with a very simple quantity control and good filtration, actually reduced the maintenance requirement.

The normal type of needle valve injectors had the disadvantage in a lubrication system that they tended to open when the cylinder pressure was high and required an extra piping system to conserve the unavoidable leakage of oil past the injector needles. The author's company used very small, spring loaded, poppet valves opening towards the cylinder. In this way these disadvantages were overcome and, because the valve guides and springs were not exposed to hot cylinder gases, the valves operated very reliably.

In a running engine, the firing ring squeezed oil out towards the combustion space at the inner end of the stroke and the lowest ring squeezed oil out towards the bottom of the liner at the other end of the stroke. It was therefore important that oil should be supplied to the liner or piston between these two rings to make up the squeeze loss.

Mr. Kennaugh's Fig. 18 showed oil being delivered above the top ring and below the bottom ring. Presumably there was some mechanism by which it was transferred to the inter-ring area.

The author was particularly grateful to Mr. Lloyd and Mr. Wilson for their contribution and hastened to apologize for generalizing about the accuracy of "templug" measurements on the basis of tests carried out in far from ideal conditions.

The statement that the templugs registering temperatures between 100 and 200°C tended to read about 30°C high was based not only on comparison with thermocouple readings, but also on readings obtained low down on the piston skirt where it seemed inconceivable that the running temperature could be as high as that shown. It was quite possible that in this instance the templugs were affected by "heat soak" after the engine was stopped, as suggested by these contributors.