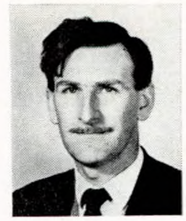
Paper read at the Institute on Tuesday, 22 January, 1974

Mr. Baker



Mr. Kimber

RESEARCH ENGINES FOR LOW AND MEDIUM SPEED APPLICATIONS

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The problems and conflicts associated with the scaling down of large engines, so as to preserve their operating characteristics, have been resolved to produce a pair of single cylinder laboratory engines of representative performance. The design and construction of these engines has been aided by the selective introduction of critical components from existing engines. The minimum modification necessary to replicate the essential feature of quiescent combustion has produced its own practical problems to avoid unacceptably high costs of design and manufacture.

The results obtained appear to be representative of the large engine types at which simulation was aimed, and an example of how such research simulation engines can be used to resolve problems is quoted. The prime purpose of the engines as units, about which marine lubricants can be developed, has been achieved; their usefulness as investigatory tools is only now becoming apparent.

INTRODUCTION

The entire pattern of motor ship propulsion technology has changed dramatically over the last two decades. Predominant in this change have been the increasing use of low cost residual fuel and the advances in operational output available for a given size of engine. Inevitably both factors have increased the critical constraints imposed upon all engine components and this has resulted in major advances in design analysis and experimental technique. For instance, few engine designers today would contemplate carrying out their work without access to thermal and mechanical stress data based upon input from realistic experimental studies.

Similarly, modern lubrication studies require direct measurements of operational data based upon realistic environmental conditions. These are, of course, available in actual engines in service and by choosing suitable vessels satisfactory experimental designs can be worked out[1]. However, before the tribologist can safely offer an experimental lubricant for ship testing, he must be very sure that any potential deficiencies or side effects are understood, otherwise he may endanger the operation well-being of the test ship.

In certain circumstances it is possible to arrange for lubrication experiments to be conducted in dynamometer tests ashore using full-sized marine engines. However, such work is very costly in both capital and operational terms. Moreover, the very existence of the full-size engine for this purpose may well imply some difficulty in imposing operating conditions in advance of the current field.

Recognizing the above situation, most marine fuel and lubricant research is carried out initially in much smaller laboratory engines. While such apparatus is by no means cheap to operate, it does enable systematic experimental programmes to be conducted on a continuous basis within the entire control of the research establishment. Moreover, it eliminates high cost risks to machinery when data are needed beyond the threshold of fairly established practice.

However, in general, smaller engine types regularly marketed have been designed for a totally different purpose and do not therefore reflect the operational characteristics of main propulsion units. Furthermore, they are normally multi-cylindered and as a rule, do not provide for very easy access to enable internal instrument installation.

To overcome these difficulties, it was decided to develop two engines representative in their operating characteristics of the most important main engines currently encountered. These are, of course, the low speed crosshead two-stroke on the one hand, and the medium speed four-stroke on the other. These experimental engines were planned as being as small as possible, while still preserving the most critical operating parameters of the larger machines they were intended to represent. Since heavy fuel is generally recognized as the most critical operating feature of current service, both engines were planned from the outset with a capacity of consuming the lowest cost fuel types available.

The present paper outlines the design philosophy, characteristics and operation of these two experimental engines, and demonstrates some comparisons of their operation against experiences gained with their full-scale counterparts.

COMBUSTION CHAMBER AND FUEL INJECTION REQUIREMENTS

A significant characteristic of large engines of 40 cm (16 in) bore upwards, is the relatively quiescent nature of the working fluid in their combustion chambers. This differs very sharply from high speed engine practice in which moderate to very high swirl rates are the rule in order to encourage rapid presentation of oxygen to fuel particles during the short period of time available for combustion. In medium and large bore practice, much larger parcels of air are involved, which do not respond rapidly to sudden accelerating forces. The period of time available for combustion is relatively increased, while the particle flight path is increased in length proportionately with bore diameter.

Because these differences are very much matters of fact, there had been little incentive to research them comparatively. However, some interesting comparisons were emerging at the

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beginning of this work in respect of combustion in both turbulent and quiescent chambers. Some of that research has since been published.(2, 3) Lustgarten and Dolenc(2) consider their quiescent chamber on the basis of a schematic heat release diagram as shown in Fig. 1. This appears to be very valuable in enabling a simple concept of the major variables to be grasped, and when referred to comparable events in the turbulent chamber.

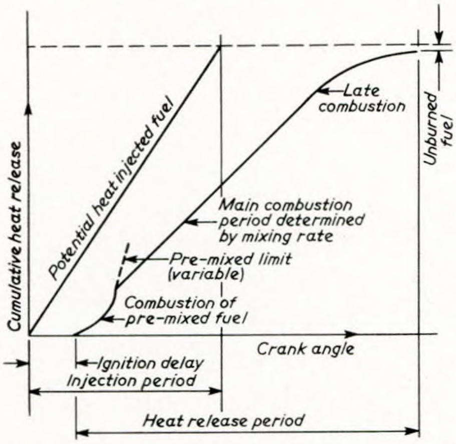

FIG. 1—*Schematic heat release diagram of the quiescent chamber (after Lustgarten and Dolenc)*

The first premixed combustion period in the turbulent chamber tends to be emphasized when compared to the quiescent, which involves a relatively lower initial rate of combustion close to the injector. Thus the bulk of the combustion takes place in the range of flight particles outside the combustion nucleus. This means that the far ranging particles may occupy increasing combustion periods related to the operating speed of the engine.

Conversely, of course, the parameter of piston speed enables sensible comparisons to be made between engines of different sizes, because it automatically compensates for the variation of combustion period as chamber sizes increase. However, the piston speed concept embraces a number of functions and assumes a degree of dynamic similarity which, although generally present in part, is not immutable. Generally, as engine sizes increase, so gradually do their swirl rates decrease, thus the change from turbulent to quiescent upon overall characteristics is scarcely discernible in values of general comparison. If, on the other hand, an engine type were to appear in the large engine range with high swirl, or in the small engine range with low swirl, it would almost certainly prove a commercial failure, but equally, it could be expected to reflect the characteristics of its combustion arrangement rather than its piston speed.

Accordingly, for experimental purposes, where the criteria of success are entirely different from those of commercial engines, some degree of cross-typing becomes permissible provided no cardinals of combustion are offended too greatly. Of these, the small engine is somewhat in danger of gross spray impingement, leading to the production of smoke particles in the absence of adequate swirl, unless special attention is paid to combustion chamber shape.

Fortunately, some work conducted by Scott(5) had explored the path of fuel sprays emanating from a single hole in an engine with a special wedge-shaped combustion chamber designed to simulate a segment of large flat combustion space. The work was based upon high speed photography of the combustion space through means of a transparent cover. This indicated that provided the piston could be so arranged as not to offer a significant target for the richer parts of the spray pattern, the vast majority of fuel particles could be consumed within a radius of approximately 8 cm (3·3 in) from the sort of injector envisaged for test engine purposes. The same work also enabled some approximate computation of the way in which the total injection could best be divided to ensure sufficient fuel for each cycle; this is needed to provide the combustion conditions needed to represent extreme ratings of full scale engines.

At this point it became apparent that the design criteria being sought bore only an incidental resemblance to good commercial engine practice. Unlike the latter, a research engine may be best considered as a test rig which happens to resemble an engine.

Having resolved the primary restraint of miniaturization to a bore diameter of not less than 16 cm (6 in) diameter, it became essential to next consider how the appropriate fuelling could be achieved in practical terms. A review of proprietary equipment quickly established that very little existed which would simultaneously meet either the physical size of injector applicable to small cylinder covers or the instantaneous fuelling rates, and be suitable for heavy fuel. However, preliminary work with heavy fuel in a Ruston 1 APC 20·2 cm (8 in) bore four-stroke had been made possible by a special partially cooled nozzle provided by the engine manufacturers. The particular nozzle had been designed to enable the APC engine to operate as a dual fuel system, but its possibilities when burning heavy fuel were quickly established through the good offices of Mr. Ernest Groschel of the Lincoln Company. Further discussions with Mr. Groschel were also fruitful in indicating a very practical development route for the injector which would ultimately enable it to be used to fuel both the two- and four-stroke experimental engines envisaged.

This, then, set the sizes and shapes of combustion chambers to be projected for the experimental engines and Fig. 2 shows their outlines. It will be seen that the maximum free flight path has been left for sprays emerging from the nozzles. The relative bore sizes are arranged to enable the same injector to be used in each case; this was felt to be necessary in the absence of any easily purchased production nozzles.

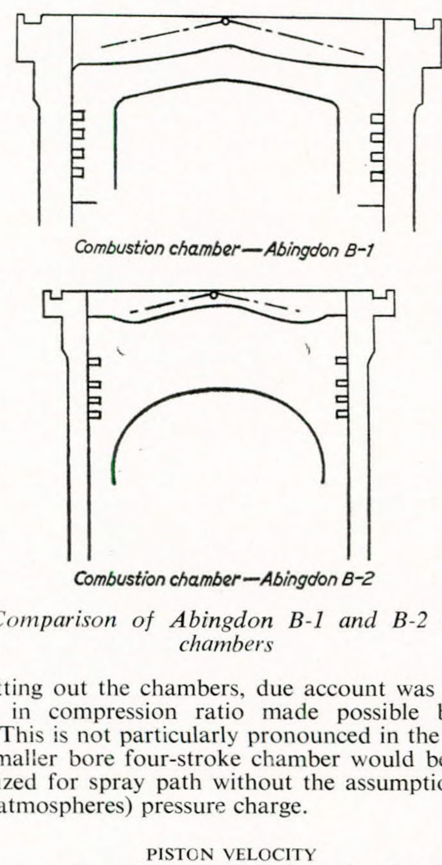

FIG. 2—*Comparison of Abingdon B-1 and B-2 combustion chambers*

In setting out the chambers, due account was paid to the reduction in compression ratio made possible by pressure charging. This is not particularly pronounced in the two-stroke, but the smaller bore four-stroke chamber would be hopelessly compromized for spray path without the assumption of about 2 bars (2 atmospheres) pressure charge.

PISTON VELOCITY

Having resolved the question of small chamber quiescent combustion geometry, consideration was given to dynamic differences. Enquiry amongst engine builders suggested that the frequency of firing cycles could not be scaled up directly in reverse proportion to bore size, although some increase in frequency was obviously needed for simulation. Information from various sources indicated that for the upper limit of fuelling cycles to be representative (around 400/min for the two-stroke and 600 for the four-stroke), the crankshaft speeds represented by these frequencies would be quite reasonable in relation to the other engine operating factors. In fact, excellent correlation with field engines has been obtained at 350 rev/min in the two-stroke and 1100–1200 rev/min appears about right for the four-stroke.

These considerations, of course, raised further questions as to the underlying relationships of cyclic frequency to the very low piston velocities which would be generated with reasonably proportional crank throw values. At that time our understanding of piston ring lubrication was even less than it is today, and the subsequent work in this area reported at ISME '73 by Baker *et al.*[4] would have helped in resolving the dilemma. However, even then, it seemed apparent to the authors that the primary mode of ring lubrication must be in some form hydrodynamic, so that velocity-induced ring and liner damage was more likely to be consequential to oil film failure than its cause. In these circumstances it seemed that a "lower-than-scale" set of piston ring velocities could only result in hydrodynamic conditions of ring sliding being somewhat more severe than in service; this being due to relatively thinner oil films and increased levels of asperity contact. This, it was felt, would lead to higher wear rates than those encountered in the field for given fuels, lubricants and operating conditions.

The operating speeds envisaged at the design stages were, therefore, set as 300–400 rev/min for the two-stroke and 800/1400 rev/min for the four-stroke. While the upper limit in the first case produces a nominal piston velocity well below the medial of full size practice, it was considered that the flexibility of the ranges afforded would be sufficient to enable a separate study to be conducted if required.

In the event, the two-stroke engine has only recently been released from more important work to allow consideration of the influence of piston velocity to be contemplated. That this has not been found essential, in the light of data produced by this engine over several years, may go some way towards justifying the arguments presented above.

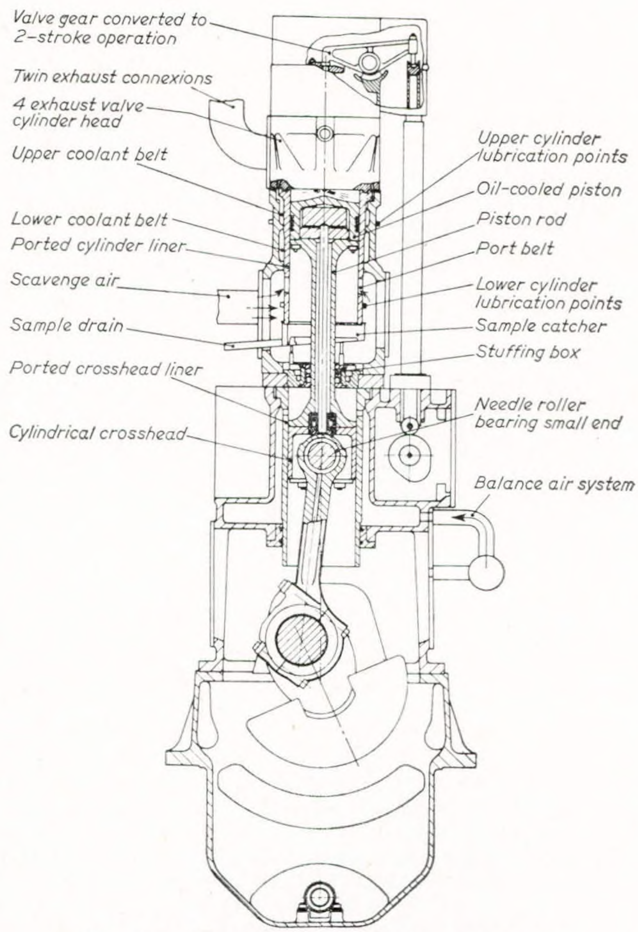

FIG. 3—*Abingdon B-1 general arrangement*

GENERAL CONSTRUCTION OF THE ABINGDON B-1

The two-stroke engine, designated Abingdon B-1 and shown in Fig. 3, is composed largely of Ruston 1 APC four-stroke components as these were found to provide very adequate margins of loading when applied to two-stroke operation. As has been described,[6] a new end scavenge two-stroke working cylinder was grafted above the original to enable the installation of crosshead and diaphragm. The new cylinder was provided with a constant loss, separate lubrication system flexible enough for alternative quill positions to be incorporated.

The open construction of the working cylinder housing shown in Fig. 4 is of special importance for experimental purposes. Care was taken in the design of this assembly to ensure that the four lantern apertures were as large as possible while still preserving an adequate measure of strength and rigidity. This was done to provide maximum access to the scavenge space around and below the port belt. Sampling devices and investigational instruments could then be inserted with minimum difficulty. Normally, the instruments can be built up on a spare cover plate and tested before assembly on the engine, which represents a considerable saving of instrument mechanic effort and engine down-time.

An example of how this access to the scavenge space can be utilized is given in Fig. 5. This shows a viewing port constructed to enable the ports and piston rings to be examined in the running engine. As the Perspex window is liable to obscuration by flying oil particles, the arrangement includes a steel wedge which can be interposed between the scavenge belt and the window while the latter is removed for cleaning. Further examples of the application of instruments to this area are given in Reference 7.

The same philosophy applies to the principle adopted for engine stripping, which in fuel and lubricant test work is generally confined to the working cylinder, piston and cylinder cover. In

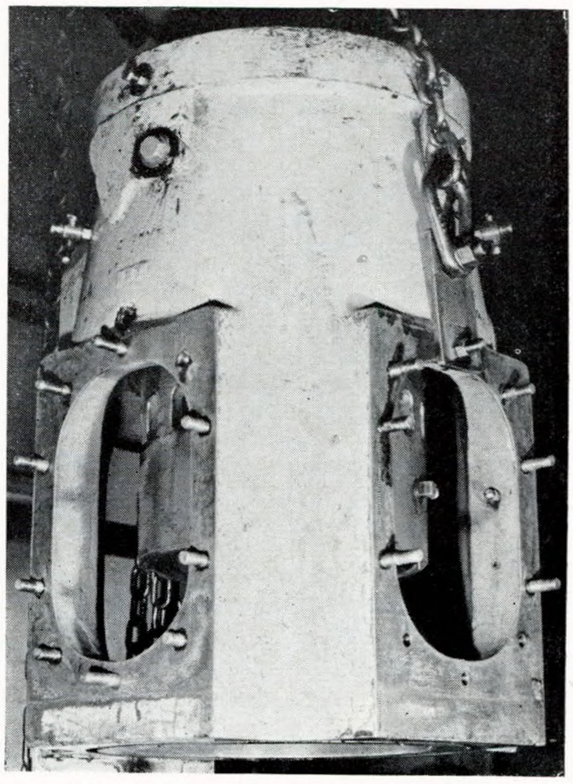

FIG. 4—*Abingdon B-1 cylinder housing*

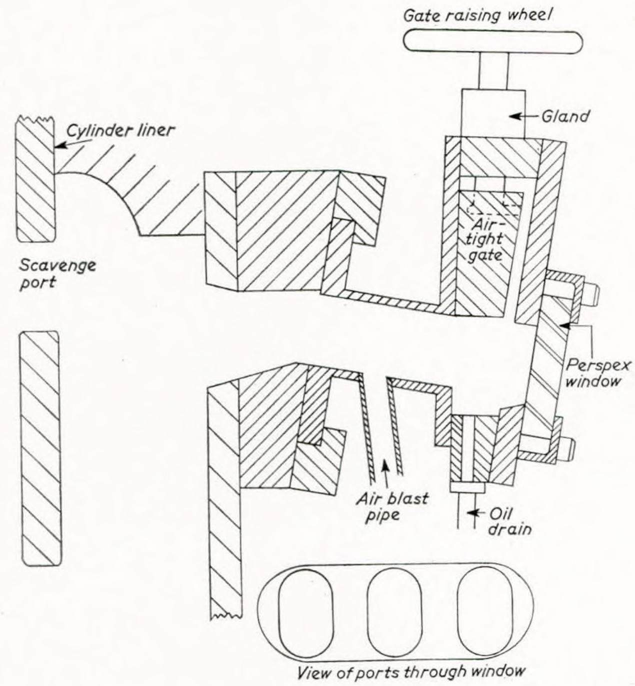

FIG. 5—*Scavenge port viewing arrangements*

this case, the complete cylinder housing is removed to expose the diaphragm for cleaning, while the used cylinder, still in its housing, together with the piston, can be taken to a suitable location for detailed examination. Meanwhile, a replicate cylinder housing, fitted with a new liner, is assembled on to the engine so that the next test can get underway. This facility enables a single engine to be kept running almost continuously, enabling test results to be accumulated without loss of data due to rapid turn-round.

The crankshaft and running gear, coming from robust four-stroke practice, have proved exceptionally reliable and inspections are limited to an annual shut-down period while laboratory services are overhauled and electrical power is liable to interruption.

The only departures from the standard four-stroke design in this area are the piston rod, diaphragm, crosshead and top end bearing. These too have proved perfectly capable of year-to-year operation. The piston rod has a chromed sliding surface and has shown no measurable wear in five years' operation, broken only by routine dismantlings. This may be due to the somewhat unusual construction of the stuffing box shown in Fig. 6. The use of a split rubber Walkersele beneath the metallic packings will be noted. The purpose of this element is to augment oil control from the crankcase and in practice it has proved extremely effective.

The use of a needle roller top end bearing occasioned some comment when first proposed and formed the basis of several pessimistic predictions. It was incorporated to avoid the difficulties encountered by plain bearings not subject to load reversal and has proved entirely trouble-free. Figure 7 shows the pin from this bearing dismantled after 6000 hours' operation. No measurable wear had occurred and the rolling surfaces had developed remarkably fine mirror-like textures. No special filtration of the lubricant reaching this bearing has been made and surprisingly, it appears more tolerant of adventitious debris than the plain bearings used elsewhere in the engine. This was demonstrated accidentally following a filter failure in the main oil supply. This was rectified as soon as it was detected and the engine continued until the next annual shut-down, when all the bearings were inspected. While the main and bottom end linings were sufficiently scored to dictate precautionary replacement, the top end roller bearing showed no signs of damage whatsoever.

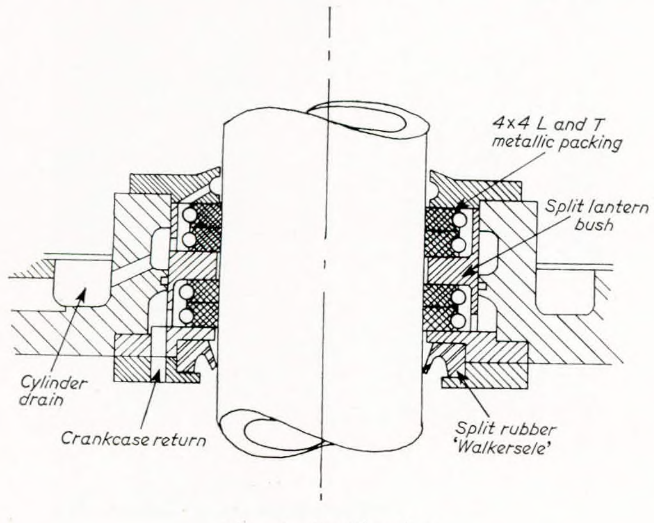

FIG. 6—*Stuffing box*

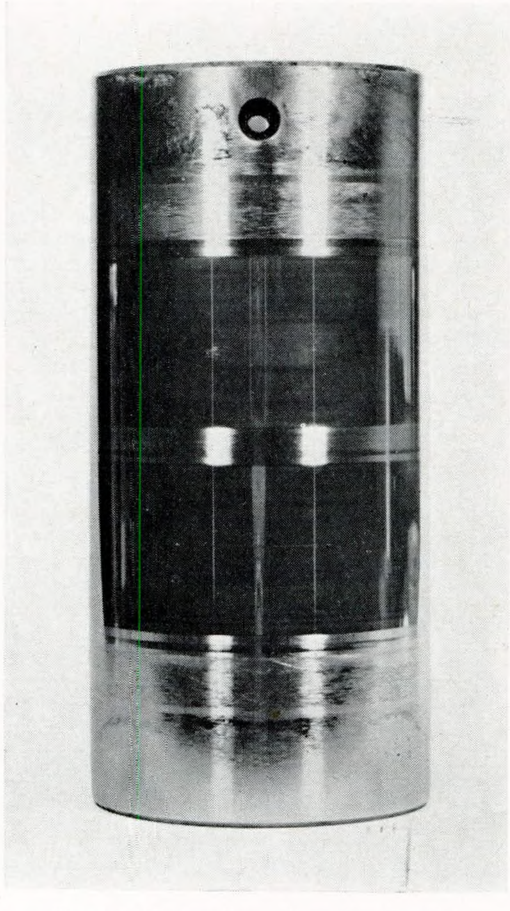

FIG. 7—*Top end needle roller bearing*

GENERAL CONSTRUCTION OF THE ABINGDON B-2

Unlike the two-stroke, there existed no range of small engines with maximum cylinder pressures well in excess of current medium speed practice. Accordingly, it was necessary to prepare a complete engine design having a load capability sufficient to cover the possible development of large four-stroke engines over the next decade. This, of course, does not mean that the authors are confident that maximum pressures will continue to rise at the rate of recent years, but a prime justification for the project was to ensure an installed capability to cover possible trends. A further consideration was to ensure that research programmes could continue uninterrupted by installation maintenance as far as possible. In this respect, the running gear of the research engine can be considered as being related to the installation, since such matters as bearing wear become merely delaying nuisances to potentially hinder data procurement.

For these reasons it was decided to limit the maximum gas force by reducing bore diameter to a minimum consistent with quiescent combustion limitations discussed earlier. Detail design, therefore, started with a review of existing four-valve cylinder covers of between 15 cm (6·22 in) and 18 cm (7 in) which might be modified to produce minimal port induced swirl. This, of course, embraced mainly high speed engines, the designers of which had been very careful to obtain a good degree of swirl by the basic lay-out of the ports. Moreover, since this class of engine invariably consumes distillate fuel, the head lay-out around the injector did not, in general, lend itself to the incorporation of the slightly "fatter" cooled injector.

The exception to these rules proved to be an early cylinder cover for an engine of 16·8 cm (6·625 in) bore. The main feature of interest was its lower swirl rate, brought about by almost straight tandem scavenge porting. The valve throat diameters were also somewhat small in relation to bore diameter; this allowed the installation of the cooled injector.

Accordingly, the running line was schemed for a bore and stroke of 16·8 cm (6·625 in) × 18·4 cm (7·25 in) and this resulted in the somewhat disproportionate looking assembly shown in Fig. 8. The extreme length of the connecting rod was dictated by the availability of a rod having adequate strength for the ultimate forces envisaged, which in turn led to a bottom end bearing diameter in the region of 15 cm (6 in). From the outset it was realized that a large ratio of rod length to crank throw, in the region of 7:1, would materially reduce the secondary cyclic disturbing force and that this could simplify questions of dynamic balance as discussed below. In the event, the rod proportions required fell so close to those of the Ruston APC used in the two-stroke engine that it was decided to use the same rod forging, modified only at the top end to incorporate the smaller top end bearing appropriate to the much smaller piston.

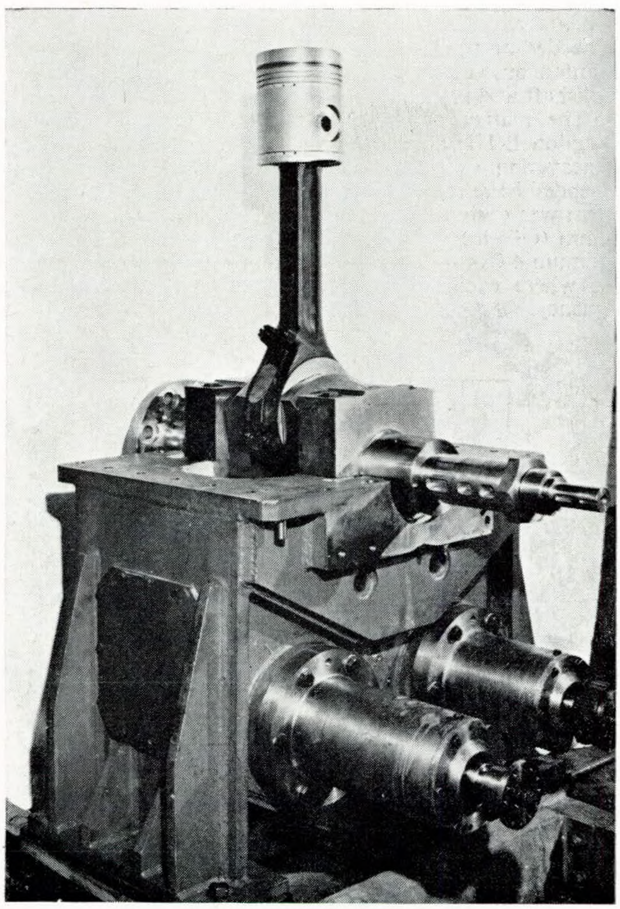

FIG. 8—*Abingdon B-2 running gear*

The next set of problems resulting from the incorporation of components from other engines was the unusual fixing arrangement of the cylinder cover. This had been taken from a multi-cylinder engine designed to a limited installation envelope for a lower order of maximum pressure than that now required. As will be seen in Fig. 9, the cover is held at six points, four locations of which were originally for shared load of the set screws locating adjacent covers. While it would have been feasible to use long studs tapped into the top plate of the crankcase, it was felt that such an arrangement would be a serious limitation if a change of cylinder cover were envisaged as a future development of the engine. For this reason it was decided to employ a heavy load band as part of the cylinder housing and retain this to the crankcase by four widely pitched studs of adequate length. This load band then provides a secure footing for six studs, to the cylinder cover bolt plan, which are extended in length to give a satisfactory load/extension characteristic necessary for good gasket control. The load band also contains machined water transfer passages between liner housing and cylinder cover.

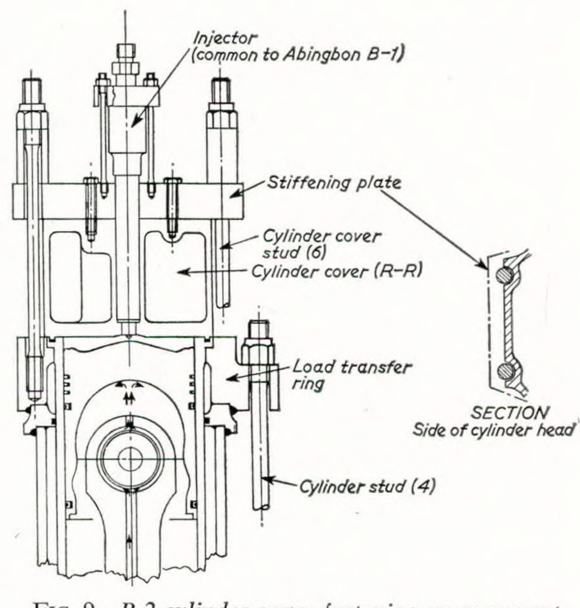

FIG. 9—*B-2 cylinder cover fastening arrangement*

Like the two-stroke, cylinder liner coolant is separated into two bands which can be piped up to separate cooling circuits to enable large temperature gradients found in some large engines to be simulated.

Figure 10 shows a cross-section of the completed design. The bedplate and entablature are completely fabricated from mild steel plate. This method of construction was chosen to avoid the high cost of pattern equipment for a single unit. The structures have been simplified wherever possible to enable them to be welded and machined simply and cheaply, special care being taken to minimize the number of co-ordinate dimensions. For this reason the cambox has been designed as a separate, bolt-on unit, mounted sufficiently remote to provide good access to the working cylinder housing.

A prime requirement for medium speed research is the study of the deterioration of alkaline crankcase oils, the potency of which decline to varying degrees in continuous service, the rate of deterioration being related to operational dosage. This can be expressed in terms of kW/litre (bhp/gal). Many engine test methods have been designed to accelerate this deterioration for comparative oil development purposes. However, these mainly involve unrealistic operating conditions such as elevated coolant temperatures which may lead to unrepresentative results.

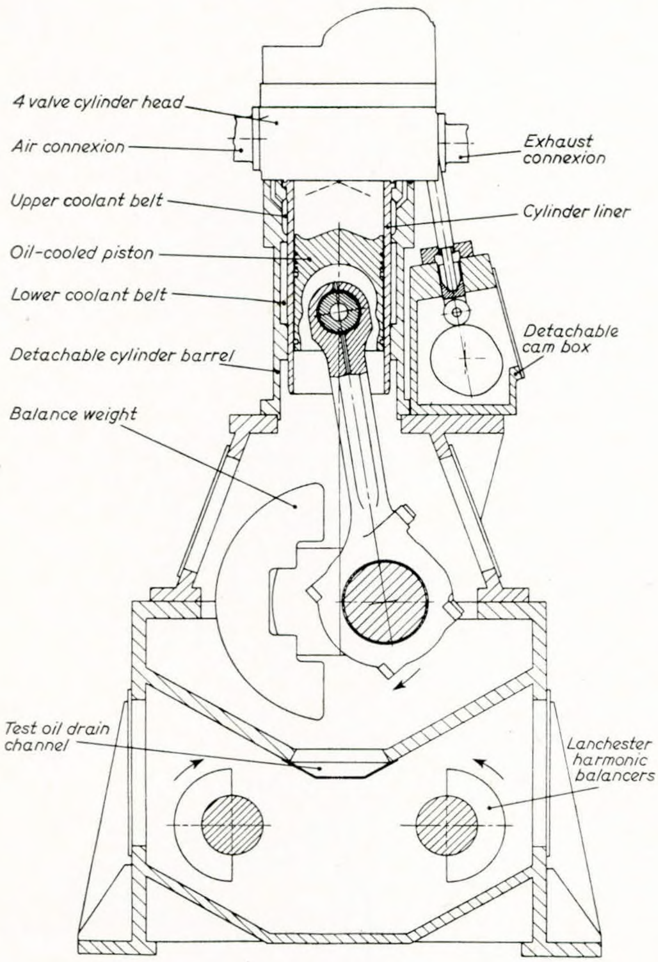

FIG. 10—*B-2 general arrangement*

To avoid this difficulty, but at the same time to accelerate the oil's ageing processes, the Abingdon B-2 is arranged to contain the minimum volume of test oil in circuit. This is achieved, firstly by a "dry" sump draining into the small capacity "top hat" catch tank shown in Fig. 11. Provision is made for automatically topping up this tank and recording make-up used during the test period. Secondly, all sections of the engine not essentially lubricated with test oil are linked to a separate slave recirculatory system carefully isolated from the crankcase proper. This ensures that the maximum proportion of test oil is used for piston lubrication at each pass through the engine. The operating capacity of the test oil system can thus be reduced to 4·5 litres (1·0 gal).

The piston and cylinder liner derive from the Rolls Royce D series engine, the latter being a standard component in every respect. The one-piece aluminium piston is also standard as far as the skirt and ring pack are concerned. However, fortuitously, the piston manufacturer was able to supply pistons fully machined in all respects except the crown, which could be "padded" up to permit experiments with different combustion chamber shapes. This enabled the chamber depicted in Fig. 2 to be machined, a route which has been followed for replacement pistons. In one respect the radial piston dimensions have been modified away from quick running practice. The crown land clearance has been opened up to give a volume ratio between its anulus and the combustion chamber volume in line with large engine practice. The effect of this modification has been to eliminate any running contact between the crown land and the liner, as will be noted later in Fig. 22.

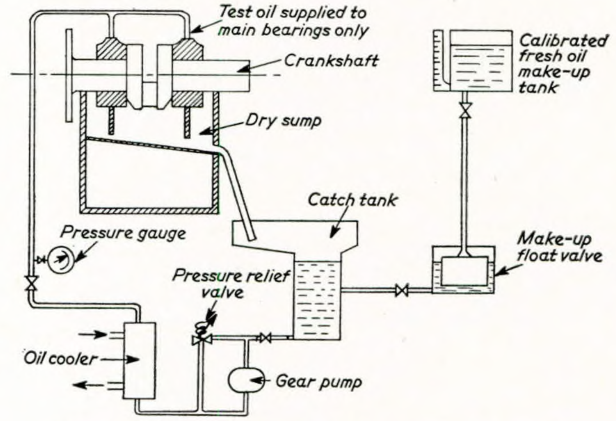

FIG. 11—*B-2 catch tank (lube oil) and oil circuit*

BALANCING

The primary vertical out-of-balance forces developed by both these single cylinder engines are quite high, as shown in Fig. 12, and as the engines are installed permanently it is necessary to isolate or cancel the forces to an acceptable level. Comparative schemes of dynamic balancing gear and suspended mass isolation were prepared for the two-stroke. However, as the engine had not run at that time and the shape of the cylinder pressure diagram could only be guessed at, it was difficult to arrive at a realistic level of cyclic crankshaft velocities. Good data are needed on this factor in order to complete the pattern of the instantaneous accelerative torques to be transmitted between the crankshaft and rotating balance weights.

The matter was resolved by the decision to house the Abingdon B-1 in a new building which presented no difficulty to excavating a pit below floor level in which to instal a spring suspended balance mass of adequate proportions. The suspended system was designed to permit a cyclic vertical displacement of 1·5 mm (0·06 in) on the cylinder centre line and an estimated maximum 4·0 mm (0·16 in) excursion during starting and stopping where exciting forces occur at lower than the critical frequency.

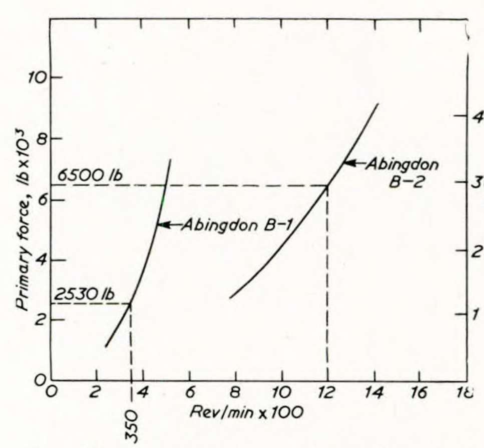

FIG. 12—*Comparative reciprocating forces*

The system proved very satisfactory once teething problems (in the shape of fractured scavenge and exhaust bellows connexions) had been sorted out. The design of bellows which was eventually found to be very reliable is known as the Palatine and consists of a series of convoluted plates in 0·3 mm (0·012 in) sheet of high nickel alloy, edge welded at the inner and outer peripheries to form very deep convolutions of exceptionally low axial stiffness.

The optimum number of five convolutions was found by experiment to give minimum self-excitation from gas pulses, yet develop sufficiently low elastic strain at the major excursions to avoid premature stress failure.

The Abingdon B-2, on the other hand, was allocated an existing running cubicle having a large cork mounted installation block which would have been costly and inconvenient to convert to spring suspension. Moreover, calculation indicated that the excitation forces to be generated would not be isolated from the laboratory structure by the cork mounting alone. In this case, it was decided to incorporate dynamic primary balancers from the outset, the cork mounting being considered adequate to isolate the rather small secondary disturbances occasioned by the long connecting rod.

The balance shafts are incorporated in an integral chamber below the dry sump and shaft extensions are carried forward to line up with a forward extension of the crankshaft. One shaft is driven direct by toothed belt from the crankshaft, the other via a pair of equal spur gears housed within the timing chest and then by toothed belt. Despite a tooth belt arrangement which was thought to be quite generous, some trouble was originally experienced due to belt jump at the maximum cyclic acceleration. Happily, however, this trouble was cured by increasing the nominal capacity of the toothed belt drives to a level approximating to that required for a power transmission duty of 70 kW (100 hp).

The final level of torque capacity appears quite surprising for a drive of nominally zero power transmission, but this experience seems to go some way in explaining gear and bearing problems which have sometimes been noted in the balancing gear of certain commercial engines. The moral of this episode would appear to indicate the need for development of a set of adequate loading factors for engine driven balance shafts.

The results of this development work have been well worthwhile in so far as the running of the four-stroke is exceptionally vibration-free with a noted absence of broken pipe fittings.

CONSTANT PRESSURE SCAVENGING

Supercharge scavenging of single cylinder engines inevitably involves some compromise compared with multi-cylinder practice. This is not too severe with the two-stroke which can be reasonably simulated with sufficiently large charge air and exhaust receivers. In the case of the Abingdon B-1 the charge air receiver represents 30 times nominal swept volume. The exhaust receiver is times 45 and the outlet is obstructed by a changeable flat plate orifice.[6] This system proved very successful in simulating large bore cylinder conditions and afforded great facility in changing conditions by varying the orifice diameter and charge pressure by adjusting the level of spilled air from the electrically driven compressor. A particular joy of the system is its complete freedom from surging and instability over the entire useful range of operation.

So satisfactory did the constant pressure system prove that it was used again on the four-stroke Abingdon B-2, virtually in the same form as for the two-stroke, as shown in Fig. 13. In this case, where scavenge pressures are generally in the region of 2 bars (2 atmospheres), the system has the additional advantage over restricted pipe pulse simulation in that it enables variations of air/fuel ratio to be accomplished very readily. Once again the system has proved entirely surge-free.

While no attempt will be made to justify constant pressure scavenging of the four-stroke on grounds of exact simulation, its stability, simplicity and ease of adjustment of charge are indisputable. Moreover, it would appear to remain open to demonstration that this departure from medium speed practice might result in grossly unrepresentative combustion for the same pressure maximum and total air/fuel ratios. Thus far, this has certainly not appeared to be the case.

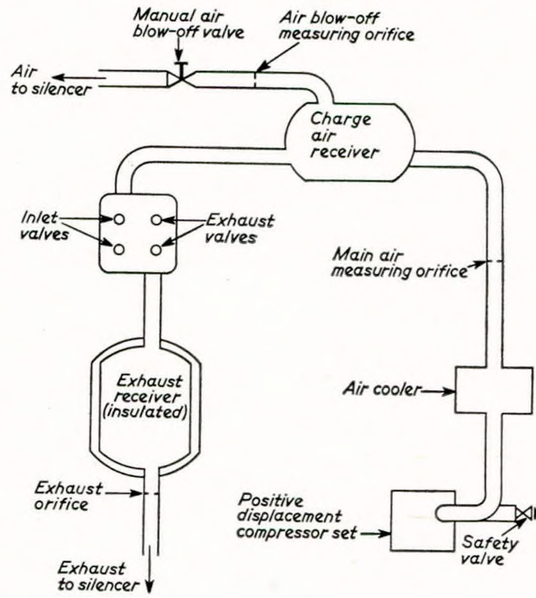

FIG. 13—*B-2 scavenge system*

RUNNING TEMPERATURES

Throughout both engines attention has been paid to the necessity for simulating surface temperatures at specific locations on the piston and cylinder liner. Consideration has also been given to ensure that most temperatures can be changed differentially to cope with significant pattern differences as these emerge in practice. The split coolant belts of both the Abingdon B-1 and B-2 cylinder housings have already been mentioned and the piston oil cooling of the B-1 is discussed in Reference 6. However, some mention of piston cooling in the Abingdon B-2 may be appropriate.

Piston cooling potential in this engine has been greatly simplified by the large connecting rod with large rifle drilling along its length. Moreover, the large top end boss of the unmachined forging provided ample scope for the development of a concentric parallel surface around the pin. As ratings mount, it is intended to incorporate a sliding slipper, bearing upon this surface in the same way as the B-1 transfers its piston oil. However, at the current ratings more than sufficient cooling effect can be obtained by allowing oil from the rifle drilling, obstructed by a metering orifice, to impinge upon the underside of the piston crown.

Figure 20 shows the temperature distribution in the Abingdon B-1 piston and liner at the normal test conditions used for much of the work. These data were obtained by finite difference methods using input from a number of thermocouples sited at strategic points.

Figure 21 giving the equivalent pattern for the Abingdon B-2 is somewhat more tentative as the number of discrete thermocouple values is small and the data have been augmented with determinations made by hardness recovery of inserted steel plugs.

CHARACTERISTIC PERFORMANCE

Representing as they do, no particular service equivalents, both engines have been fairly thoroughly characterized by normal observation of performance. These data have formed a useful basis of comparison against published data from the full scale engines at which simulation has been aimed.

Abingdon B-1

Some data concerning the two-stroke Abingdon B-1 have already been published,[6] however they are repeated in Figs. 14 and 15. These show the relationships established at a constant speed of 350 rev/min and increasing weights of fuel and air appropriate to clear combustion. Figure 15 showing the Willans line is included as a reminder of the importance of relating single cylinder engine performance to the indicated, rather than the brake mean effective pressure.

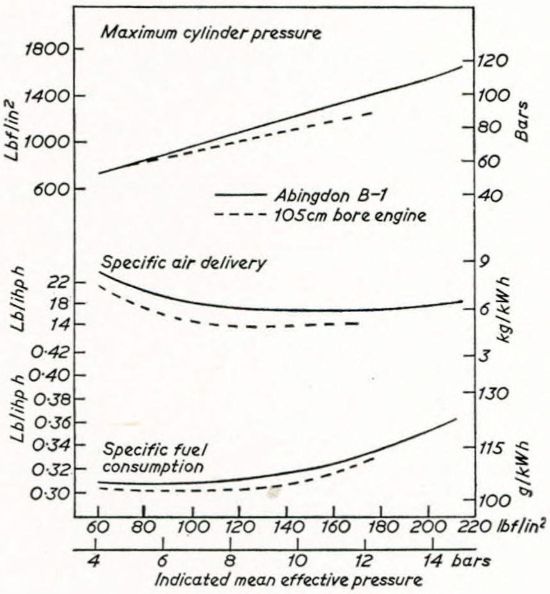

FIG. 14—*B-1 performance*

A certain reluctance to look at comparisons on the indicated basis has been noted, possibly because of difficulties in determining precise indicated data from cylinder pressure/displacement diagrams. However, a little thought shows that it is quite unreasonable to compare brake conditions directly between engine types which have differences in f.m.e.p. of 4:1 or more. Perhaps the rationale might prove more acceptable if, when interpreting the Willans line, reference were made to ablative mean effective pressure, rather than frictional mean effective pressure.

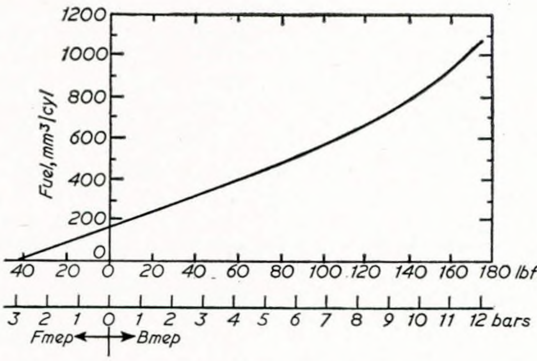

FIG. 15—*B-1 Willans line*

Figure 14 shows the range of performance developed at 350 rev/min. Although the greater part of the research and development testing performed so far has been conducted at about 11·8 bars i.m.e.p., this shows that ample scope in performance exists at the current level of combustion development for investigation in the regions beyond. Certain large bore engines are known to be operating experimentally at 13·7 bars and it is probable that simulations operating at somewhat beyond this pressure will be required shortly.

Looking further into the future, the pressure capabilities of the engine are ample for any further development and a maximum cylinder pressure of 207 bars (3000 lbf/in²) would

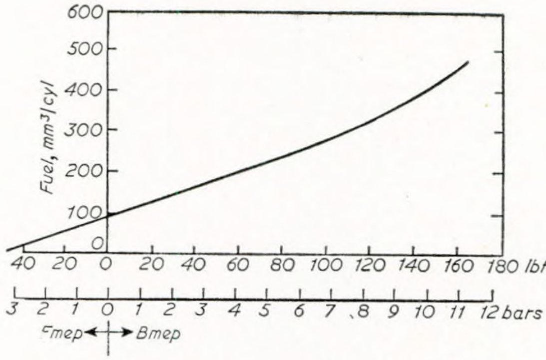

FIG. 16—*B-2 Willans line*

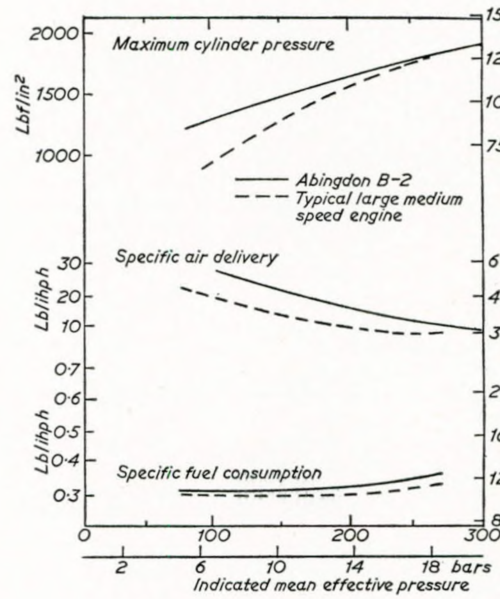

FIG. 17—*B-2 performance*

give rise to no great concern. Similarly, because of low cyclic speeds and relatively thin metal sections of the small cylinder, thermal loading is extremely light so that no advanced developments of components would be needed to avoid decreased reliability. However, it is anticipated that the existing fuelling rate, shown in Fig. 18, would need to be increased to keep the cycle in step with large scale practice. This too could be accomplished quite readily as the Ruston fuel pump is capable of receiving an element up to 40 per cent larger in cross-sectional area.

Abingdon B-2

Again, the ablative losses are very apparent as will be seen from Fig. 16 of the Willans line for the four-stroke. This is hardly surprising in view of the relatively large bearings and the additional losses incurred in driving the balancing masses and the unusually large cam gear. However, as mentioned previously, this is of no consequence in an engine whose entire useful work is discarded in warm water from a dynamometer. What is important is that a rational basis of comparison should be developed with the counterparts for which simulation is sought. While some effort and patience is needed to generate the data points for a good Willans line, this appears to be the best device yet conceived for the purpose and like many good marine practices, it requires no exotic instruments for its generation. Like the two-stroke, the characteristic performance has been

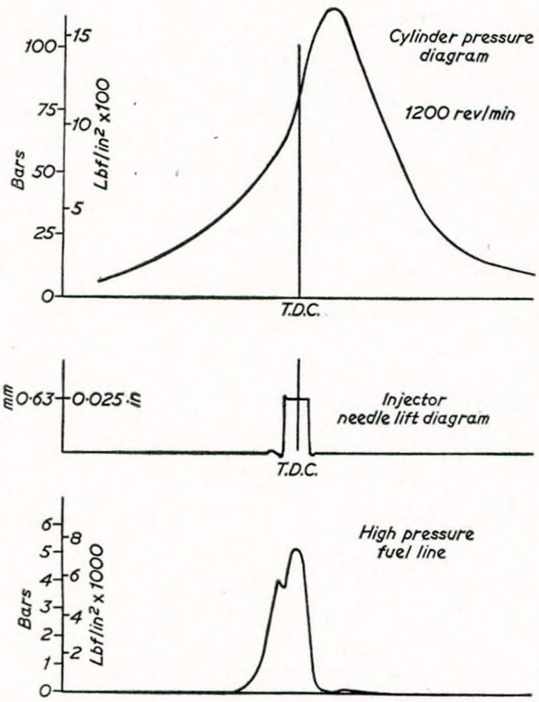

FIG. 19—*B-2 fuel injection and maximum pressures (Bars × 100)*

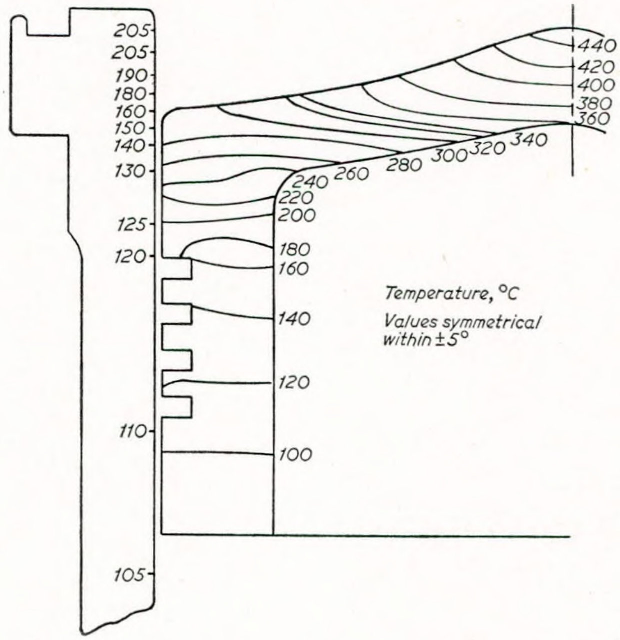

FIG. 20—*B-1 temperature distribution in the piston and liner*

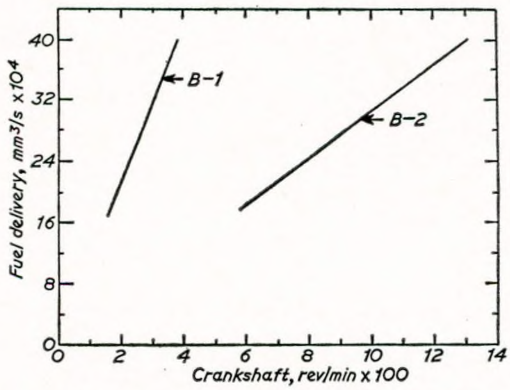

FIG. 18—*Comparative fuelling rates*

developed against performance published for engines of the type it is intended to simulate, in this case some typical high output medium speed engines in the medium-to-large part of the range have been found to give reasonably similar characteristics. Figure 17 shows how one such engine lines up against the B-2.

Development of this engine to date has been suspended at a maximum output of 20·7 bars (300 lb/in²) i.m.e.p. owing to limitations of quantity and pressure available from the charge air compressor. Unfortunately, the unit purchased for this purpose did not live up to its promised duties and this has limited engine output. It is being replaced by a more robust, if somewhat old-fashioned design which, when installed, will enable development to proceed. In the mean time, the maximum developed rating has been found adequate for comparative test purposes of oils destined for use in medium speed marine engines burning heavy fuel.

Figure 19 shows a selection of transduced observations of internal cyclic pressures taken at the cylinder and fuel line, together with the needle lift diagram from the fuel injector. Bearing in mind that this injector is identical to that used in the two-stroke, the uncompromised range of clean injection demonstrates the advantages of obtaining the benefit of sound fuel injection technology at the outset. It is probably true to state that the availability of the common fuel injection equipment has provided the greatest reduction in cost, time and effort, for both engines, in the entire project.

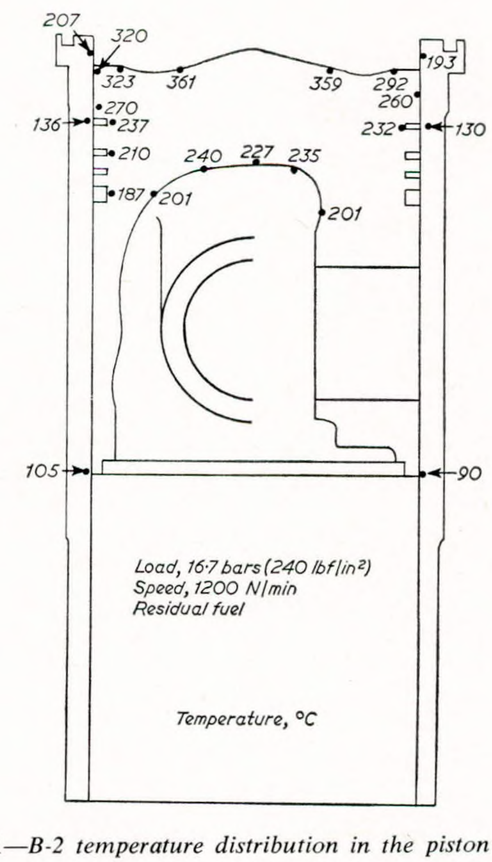

FIG. 21—*B-2 temperature distribution in the piston and liner*

RESEARCH AND DEVELOPMENT TEST RESULTS

The purpose of this paper is to describe some rather unusual combustion engine practice related to marine applications in terms appreciated by marine engineers. As such, the preponderance of test results, directed towards the development of new and better lubricants, probably has little interest, since for obvious reasons it would not be possible to relate such results other than by coding to hide their true identities.

In these circumstances it has been decided to limit the number of coded cylinder oils to be reported to a representative small number which have direct ship operation comparisons obtained under the optimized "key" ship test conditions.(1) Code letters used are common to both papers and comparative summaries of wear data are shown in Table II, from which it will be noted that the wear rate developed in the Abingdon B-1 is much higher than that obtained in the ships' engines. This is, of course, what would be expected in the light of considerations discussed in the previous section on piston velocity. However, what is important is that the Abingdon B-1 rates the performance of these oils in the same order that they appear in a very carefully organized series of ship trials. Moreover, all the oils reported represent formulae very close to the upper limit of performance amongst those currently available. In fact, were they to be run in a randomly distributed selection of ships, it seems doubtful whether their relative wear performance could be distinguished in much short of two years' continuous operation. Such is the measure of intensive development of highly alkaline cylinder oils for nearly two decades. Nevertheless, small though they appear, there is still room for these advances to provide significant operating benefits.

A similar brief summary of confidence is not yet available for the Abingdon B-2 engine as the cylinder wear rates of the best service examples of medium speed engines already approach the threshold of short term comparative measurement. The developments in crankcase oils for this field are mainly towards reduction in piston ring wear and internal cleanliness coupled to extensions in the useful oil life.

An idea of the progress made in medium speed lubricants can be judged from Fig. 22, which shows photographs of the Abingdon B-2 piston after 200 hours operation against a pattern of declining alkalinity shown in Fig. 23.

TABLE I—RELATED DATA FOR ABINGDON B-1 AND B-2 SINGLE CYLINDER RESEARCH ENGINES

	Abingdon B-1	Abingdon B-2
Cylinder bore	203 mm (8·0 in)	168 mm (6·625 in)
Cylinder stroke	273 mm (10·75 in)	184 mm (7·25 in)
Compression ratio (Geometric)	12·5:1	11·5:1
Allowable cylinder pressure:		
maximum	210 kg/cm² (3000 lb/in²)	210 kg/cm² (3000 lb/in²)
current rating	134 kg/cm² (1900 lb/in²)	141 kg/cm² (2000 lb/in²)
Operating cycle	Two-Stroke	Four-Stroke
Running speed range	300–400 rev/min	800–1400 rev/min
Nominal average piston speed	2·72–3·63 m/sec	4·9–8·6 m/sec
Scavenge system	Constant pressure	Constant pressure
Scavenge arrangement	End scavenge, ported liner, 4 exhaust valves in head	2 inlet and 2 exhaust valves in head
Combustion chamber	Open, without swirl aids	Open, without swirl aids
Fuel injection	Central, single fuel valve fully cooled to tip	Central, single fuel valve, fully cooled to tip

THE "LOW SULPHUR" FUEL PROBLEM

A more fruitful subject for investigation, for publication, concerns the area of problem solving in which a test engine facility of this type can be used. The disclosure of proprietary information is not in conflict with objective reporting.

Typical of this type of investigation is a problem which most marine engineers know of, though few have apparently ex-

TABLE II—CORRELATION OF ABINGDON B—1 WITH RESULTS FROM "KEY" SHIP OPERATION

Engine	Laboratory tests Abingdon B-1 203/273			Shipboard tests B & W 984-VT2BF-180		Man K9Z 70/120E		Sulzer 9RD90	
Cylinder oil code(i)	Top ring wt. loss gm/1000 h	Total 4 ring wt. loss gm/1000 h	Mean bore wear mm/1000 h	Top ring wt. loss rate %/1000 h	Mean bore wear mm/1000 h	Top ring wt. less rate %/1000 h	Mean bore wear mm/1000 h	Top ring wt. loss rate %/1000 h	Mean bore wear mm/1000 h
D	0.026	0.065	0.345	4.66	0.033	0.83	0.006	—	0.020
R	0.042	0.084	0.575	5.19	0.025(ii)	—	—	—	0.050
A	0.045	0.090	0.440	6.41	0.044	1.08	0.012	—	—
B	0.057	0.163	0.531	—	0.042	—	—	—	—

(i) Code as in paper by Baker, Casale and Breyer(1).
(ii) Results occluded due to lacquered liners.

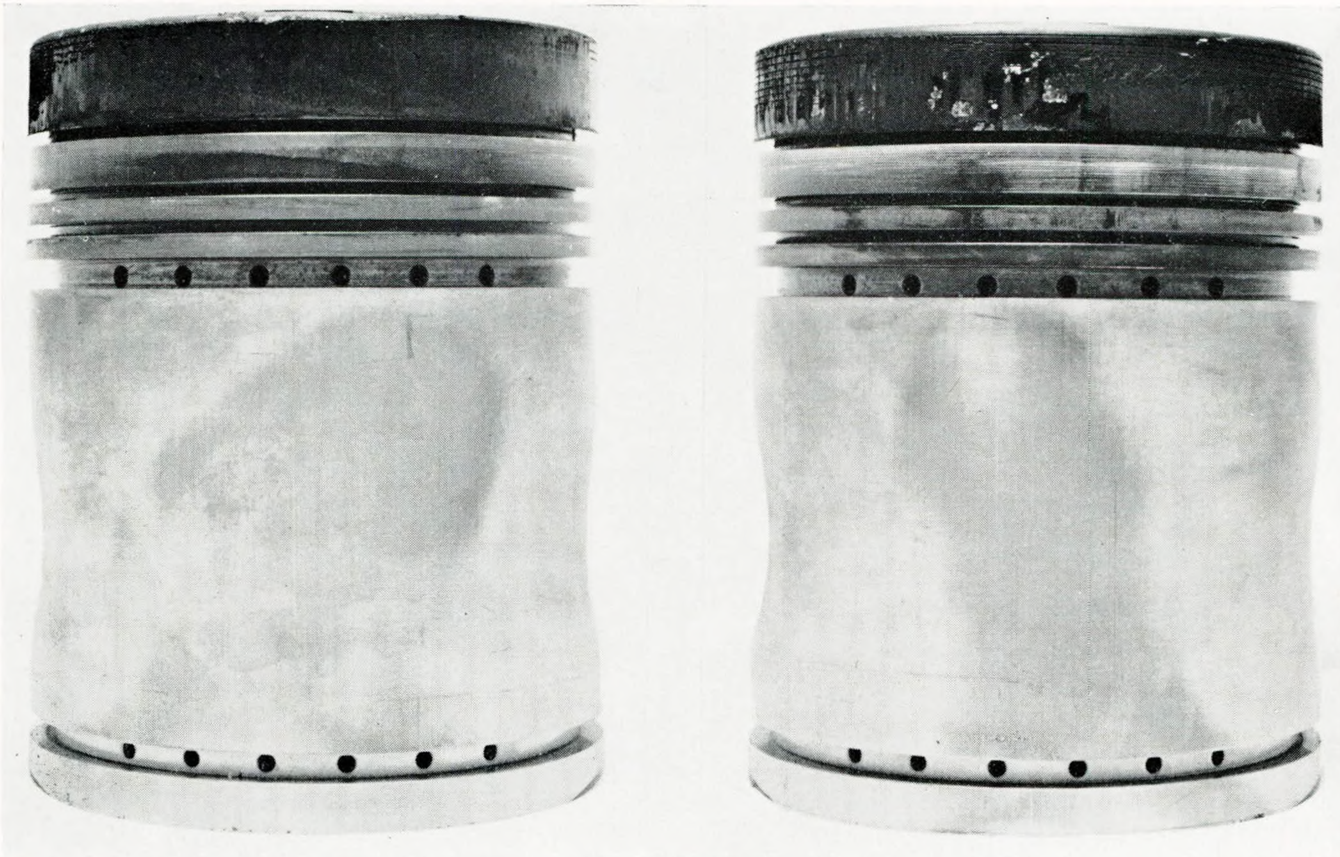

FIG. 22—*B-2 piston after 200 hours*

perienced. It concerns the difficulties experienced with engines using special supplies of fuel of a low sulphur content. So far as the authors have been able to ascertain, the precise origins of such fuels are outside the regular marine bunkering network of the Western world. However, their occasional appearance, associated with trouble, has been giving rise to concern by the Western suppliers of cylinder lubricants for at least eight years.

The trouble invariably manifests itself in reportedly disastrously high liner and piston ring wear rates, usually leading to unscheduled overhauls and acrimonious discussions with the operators of the unfortunate vessels. However, until very recently it had proved difficult to undertake any real investigation into the problem because samples of the difficult fuels were simply not available to Western petroleum laboratories. Even now, when (as will be described) one of these unusual fuels has been investigated, it still remains to be seen whether or not the findings are more than specific to that particular source.

Published reference has been made to an investigation in this area by a European engine builder,[8] in which it was concluded that the problem was due to unreacted alkaline material precipitated from highly alkaline lubricants in the presence of

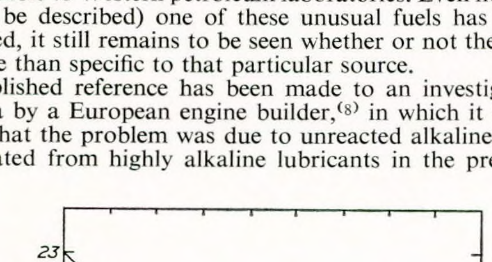

FIG. 23—*B-2 alkalinity*

fuels containing insufficient sulphur for reaction to occur. However, this observation did not agree with experience of many hours' operation in the Abingdon B-1 burning low sulphur distillate fuel and lubricated with premium alkaline cylinder oil in the context of different investigations. In the latter case, absolutely no trouble had been observed and liner wear rates had been measured at the expected low levels for this fuel and lubricant combination. By itself, of course, this evidence was insufficient to indicate an alternative mechanism to that put forward[8] as it was too easy to dismiss data obtained in a small test engine as being unrepresentative of full scale operation.

Eventually, a small supply of distillate fuel was obtained which was stated to have been involved in the "low sulphur problem". This fuel was inspected against two stock supplies of gas oil with the results shown in Table III. On this basis and the operating characteristics of the small turbulent combustion chamber cetane test engine, the fuel appeared to be of excellent quality and it is reasonable to suppose that in high speed automotive size engines with high turbulence it would perform

TABLE III—LOW SULPHUR FUEL—SUMMARY OF SIGNIFICANT DIFFERENCES

Fuel	Caterpillar reference	Regular gas oil	Low sulphur suspect (distillate)
Sulphur content, wt.%	0·40	1·00	0·06
Cetane number	41	53	53
Spectra by GLC	———	All very similar	———
Infra red spectra	———	All very similar	———
Distillation	Higher F.E.B.R.	— Very similar	—
Aromatics	66	43	43
Olefins	1·5	1·0	0
Paraffins	32	56	57
Total nitrogen	0·007/0·011	0·007/0·009	0·016/0·022

beautifully. However, as soon as it started to be burned in the Abingdon B-1, it was apparent that its combustion was inferior in this engine to the lowest quality heavy fuel experienced. At a rating of only 11·8 bars i.m.e.p., the engine smoked badly and required 10 per cent additional fuel to maintain the load. Varying jacket temperatures did nothing to help but eventually a fairly clear exhaust was obtained by shutting off coolant to the injector nozzle completely. This enabled 100 hours' operation to be completed with the low sulphur fuel and a premium quality alkaline cylinder oil. When stripped, the cylinder liner and piston rings were found to be severely damaged in an abrasive scoring mode, loosely termed "scuffing".

Routine observation of the scavenge space had been maintained during the run through the special window mounted on the working cylinder housing (Fig. 5), when it was noted that the occasional combustion flash had changed from its customary red-to-orange colour to purple and blue. To verify this observation the fuel was briefly switched to gas oil, when the normal flash colours returned within a few minutes.

Figure 24(a) shows the piston for this test alongside Fig. 24(b) of a piston which had completed 500 hours' operation on heavy fuel and the same cylinder oil at identical load and speeds. Prominent in Fig. 24(a) are the hard thick patches of carbonaceous deposit around the crown land. The edges of these patches of carbon were sharply fractured with occasional broken slivers just about to fall away. Clearly the damage to the rings and liners had originated by just such fragments becoming detached and trapped between the sliding surfaces. Sophisticated crystallographic techniques showed clearly that the carbonaceous structure contained exceptionally hard phases which were absent in carbon scraped from pistons with which conventional fuel had been used.

FIG. 24 (*a*)—*B-1 piston after 100 hours operation with low sulphur fuel*

FIG. 24 (*b*)—*B-1 piston after 500 hours operation with heavy fuel* (*same lubricant*)

Reference to Fig. 3 of the paper by Cotti and Simonetti[8] showed signs of a similar thick brittle deposit on the crown land of their large bore engine after 30 hours' operation, although their photograph had been truncated close to the upper ring position. Figure 25 is an enlargement of Cotti and Simonetti's Fig. 3A, which shows similar deposit features to those found on the crown land of the Abingdon B-1.

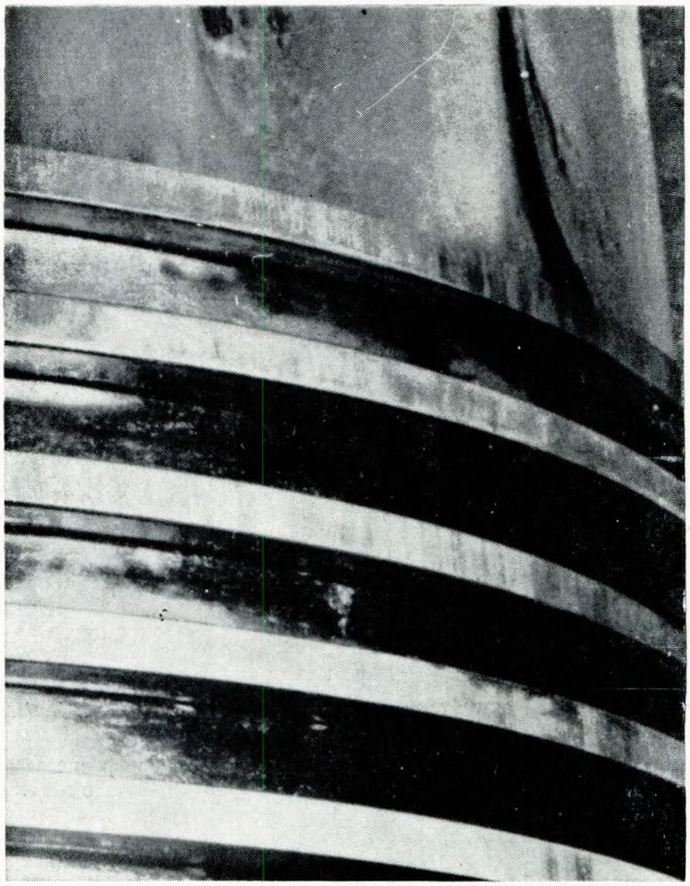

FIG. 25—*Enlargement of photograph of large bore Fiat piston* (*Cotti and Simonetti*)

Table IV summarizes the results obtained with three premium alkaline cylinder oils, A, B and R. The characteristic hard and fractured crown land deposit and varying degrees of

TABLE IV—LOW SULPHUR FUEL—ABINGDON B-1 TEST

Cylinder oil	Fuel	Ring/liner scuffing
Code[1]		
A (70 TBN)	3·0% S residual	O.K.
	1·0% S gas oil	O.K.
	0·4% S gas oil	O.K.
	0·06% S distillate	Scuff
B (70 TBN)	3·0% S residual	O.K. (Slight scratching)
	0·06% S distillate	Scuff
R (75 TBN)	3·0% S residual	O.K.
	0·06% S distillate	Scuff
Z (0·2 TBN)	0·4% S gas oil	O.K.
	0·06% S distillate	Scuff

(1) Oils "A", "B" and "R" are to the same formulae as those similarly coded in the paper by Baker, Casale and Breyer.(1)
Oil "Z" is a medium VI naphthenic base oil, without additives, in the SAE 50 viscosity range.

liner damage were apparent in each case. Also recorded in Table IV is the result with oil "Z", which differs from oils "A", "B" and "R" in that it contains no additives whatsoever, being a low V.I. naphthenic base straight mineral oil within the SAE 50 classification.

Results with the test fuel and cylinder oil "Z" were strikingly similar to those obtained with the alkaline oils "A", "B" and "R", both in terms of ring and liner damage and the occurrence of the hard, thick and fractured patches of carbon. Moreover, when analysed crystallographically, the same range of exceptionally hard phase carbon crystals was found as had been noted earlier.

DISCUSSION OF RESULTS

While the above study does not categorically contradict the findings of Cotti and Simonetti,[8] since the "low sulphur" fuel used for both pieces of work was not necessarily the same, it does throw considerable doubt upon them. Moreover, the investigation casts some considerable doubt on the combustion quality of the particular "low sulphur" fuel when used in quiescent combustion chambers. Further investigation into the influence of the differences noted in Table III has failed to reveal anything likely to make a significant difference to combustion. However, comparisons of low temperature oxidation between the three fuels of Table III indicate the low sulphur fuel to be very much more stable to initial oxidation than the regular gas oils at temperatures below 200°C and at pressures below 13 bars (13 atmospheres).

We would like to suggest the following possible mechanism to describe the difficulties encountered with this fuel. It is believed that small changes of sensible heat in the fuel at the injector may influence combustion in the Abingdon B-1, as discussed in a contribution by one of the present authors to the paper of Lustgarten and Dolenc.[2] Moreover, shutting off the nozzle cooling produced a marked reduction in smoking with the test fuel. It therefore follows that the rate of initial heating of fuel droplets may be close to a critical minimum needed for completion of combustion within the period of a normal cycle. In the case of the low sulphur fuel, this critical minimum sensible heat at injection may be somewhat higher than for normal fuels.

Whereas this lack of droplet heating would be swamped by impingement and higher exchange rates in turbulent combustion, the quiescent chamber appears to be less tolerant in this respect. Accordingly, in the case of the test fuel, partially burnt droplets may reach the chamber wall, which in the limiting case is formed by the upper end of the cylinder liner. In this event, combustion may continue in the annulus formed between the cylinder wall and piston crown land which is a very restricted space with regard to oxygen availability. This concept of combustion in the presence of limited oxygen around the piston crown annulus coincides with the observation through the port viewing window of blue-to-purple flashes described earlier.

Just how far the above experience covers other sources of difficulty remains to be seen, but the authors would be very grateful for any conclusive reports of this problem which can be tied to available samples of the associated fuel. In the meantime, the investigation appears to form a further piece of evidence which confirms the necessity for using a quiescent combustion chamber in any test engine used to simulate service units having this characteristic. It may also be worth a little thought by those concerned with improving low load combustion in engines which burn residual fuel. Could it not be that the efforts which have gone into reducing temperatures in the injector and its surroundings at high load have somewhat neglected initial fuel vaporization at low loads?

CONCLUSIONS

1) The scaling down of large engines to sizes suitable for laboratory usage can be achieved while preserving the essential features required for fuel and lubricant investigation.
2) A benefit of this is that a test engine can be constructed to be representative yet far more robust and accessible than would be possible by attempting to convert directly the combustion arrangements of a small engine.
3) By so constructing special test engines, ample margins can be allowed to permit future uprating and modification of the working cylinder which may well become a significant development in the field.
4) Scaling down produces low thermal loads, since operating speeds are generally much lower than those of commercially built engines of the test engine size. This can be of material benefit to the procurement of parts which are difficult and expensive to produce specially (cylinder covers, valves, etc.)
5) Lubricant performance in low and medium speed engines burning heavy fuel has now reached levels from which further advances require intensive development. Significant improvement appears possible in relatively small steps (see Table II), moreover responsible laboratories will continue to develop their own test methods, capable of reflecting such movements accurately as they occur.
6) Combustion initiation in quiescent chamber engines appears to be somewhat more critical than is generally realized.

ACKNOWLEDGEMENTS

The authors are grateful to the management of Esso Petroleum Co. Ltd. for permission to publish this paper. They also wish to acknowledge the contributions made by Mr. T. H. Aitchison and other members of the team at Esso Research Centre, without whose help the work would have been impossible.

Finally, but by no means least, the arguments and rationale behind the two engines described have come from a pot-pourri of useful exchanges with major engine builders around the world, too numerous to mention individually. Surprisingly not one of the busy people approached failed to respond enthusiastically to our search for information. In the authors' experience in other industries, this response is most unusual and it must surely reflect upon the character of engineers who are willing to earn a living by pitting their wits against the current difficulties of developing combustion engines.

Material Assistance

Mention must also be made of the organizations who have assisted in the design and construction of the engines.

We acknowledge the principal contributors in alphabetical order:

Associated Engineering Developments Ltd.
BICERI Ltd.
Bracknell Engineering Co. Ltd.
Clarks Crank & Forge Ltd.
Cross Manufacturing Co. Ltd.
Expert Heat Treatment Ltd.
Kayser Ellison & Co. Ltd.
Lancaster & Tongue Ltd.
Marsh Engineering Co. Ltd.
Norris Bros. Ltd.
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Ransome & Marles Bearing Co. Ltd.
Regulateur Europa Ltd.
Ricardo & Co. Ltd.
Rolls Royce Diesels Ltd.
Ruston Paxman Diesels Ltd.
Salford University (Numerical Machining Project).
James Walker & Co. Ltd.
Wantage Engineering Co. Ltd.
Wellworthy Ltd.
Woodward Governor Co. Ltd.

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Discussion

Dr J. W. A. SCHRAKAMP said that the development of I.C. engines was a difficult task. There were several "unfortunate careers" of newly designed engines to illustrate this point.

The amount of time and work which had to be put into the development of well-known and widely used single cylinder test engines was well known to all who were familiar with lubricant testing.

These points were mentioned specifically to underline that there should be the greatest appreciation for the achievement the authors had described in their paper. The inventiveness and perseverance shown by this interesting paper, as well as by earlier publications on the Abingdon test engines and related subjects, deserved profound admiration.

The idea of starting the development work by simulating the combustion characteristics of the low speed engine seemed very logical, at least as it was described by the authors. Nevertheless, it was a rather unconventional concept which demonstrated that considerable knowledge of, and feeling for, the cylinder lubrication process were already available before the construction of the test engines took place.

By comparing several performance characteristics the authors had shown that the test engines were representative models of their large counterparts in practice. But some questions remained, especially on the translation of laboratory results into reliable predictions of lubricant performance in practice.

The authors were not able to give more information on specific oils than they had in Table II. However, it would be of value to learn more about the influence of various T.B.N. levels on cylinder wear, and on the distribution of liner wear in a circumferential direction. It would also be of interest if the authors would give some information on the correlation between the B1 and the large-bore engines in regard to the two aspects mentioned above.

The information given on the "low sulphur" fuel problem was highly interesting, especially as combustion characteristics had been taken into account. This factor was too often omitted, and had greatly contributed to the confusion on this subject. There were obviously a number of factors which had to be considered, in addition to the fuel sulphur content and the lubricant alkalinity.

The speaker's experience of this problem seemed to be complementary to the authors' findings. In his company's Bolnes engine (normally aspirated, single cylinder, 190 mm bore), it had been found that under the full-load condition scuffing could be produced on a complaint fuel (residual with 0·3 per cent w sulphur) and a highly alkaline cylinder lubricant.

It had also been found, however, that the same phenomenon occurred when using a conventional and readily available gas oil with 0·1 per cent w sulphur. This enabled tests to be run with various additives at a high T.B.N. level, to check their response to low sulphur fuels, and to rank them accordingly.

However, a particular combination of fuel and lubricant which was satisfactory in the Bolnes produced scuffing marks on the cylinder when this oil was tested in a 680 mm bore Sulzer engine. Replacing the 0·1 per cent w sulphur gas oil by a gas oil containing 0·5 per cent w sulphur eliminated the problem completely in the Sulzer engine.

This example did not necessarily conflict with the authors' theory. It was agreed that combustion characteristics play an important role in the "low sulphur" fuel problem, but it would appear that other factors were involved as well. It would be of interest to have the authors' comments on the results described.

Much closer contact would be welcomed by the speaker between test laboratories on technical subjects as described in the paper. He thought that all parties concerned would benefit from the better and quicker solutions of some general problems in lubrication which were still with us.

MR. D. LYON, F.I.Mar.E., said he was particularly interested in the section of the paper dealing with low sulphur fuels. Although at the time of writing the paper the authors could not have been expected to predict the current crisis or the world shortage of fuel, they appeared, however, to have hit upon a topic of considerable interest, and one which was vital to the present state of famine in international marine bunkering. From the data concerning low sulphur fuel one deduced that it bore a marked resemblance to certain fuels emanating from China. They came from crudes having virtually no sulphur, so that it was not surprising to find the levels quoted by the authors. Perhaps they would confirm that the fuel described was of Chinese origin.

Bearing in mind that all aspects of fuel quality reported against Western standards showed that fuel to be ostensibly of excellent quality, it appeared extraordinary at first sight that it should give rise to trouble in large engines which one was used to considering as being capable of burning very poor quality fuels. However, as many people were aware, the marine chemical companies had provided fuel additives for many years to a very large number of ship operators who could vouch for their effectiveness as a means of offsetting low grade Western fuels. Thus, it was by no means surprising that fuel from a totally different source should require some specialized attention.

The authors stated that the only significant difference they could find for the low sulphur fuel they tested and Western gas oil was its remarkably high oxidation resistance at low temperatures. They went on to relate that difference to a relatively small increase in quantity of unburnt fuel reaching the crown land anulus. Mr. Lyon calculated that 2 to 4 per cent of the total fuel oxidizing slowly in that area would be quite sufficient to completely fill the crown land clearance volume in 25 hours' operation of a large engine. That estimate appeared quite in line with the authors' findings although, of course, the carbon should break up before total packing was reached.

The obvious solution to the problem was the introduction of a fuel additive to reduce the low temperature stability of the fuel in question. Some disastrous results had already occurred in some ships unwittingly bunkering fuel from Chinese ports and several millimetres wear in cylinder liners had been experienced during a single voyage. It would appear only prudent that vessels trading in those areas should have supplies of a suitable additive on board. Obviously the use of such an additive would be inapplicable to normal Western fuels, and careful arrangements would have to be made to disperse it only to the low sulphur supply as and when required.

In conclusion, the paper appeared to disclose the shape of an increasing difficulty to a number of Western ship owners who, through the good offices of the Institute, should be alerted and forewarned of the necessity for precaution in the manner discussed.

MR. G. MCCONNELL referred to the wear rates outlined in Table II, and said the authors mentioned a wear ratio of somewhere around 10:1 for the B1 versus the B & W, pointing out that it was somewhat higher for other engines. Mr. McConnell worked it out at between 30 to 80:1 for the B-1 versus the MAN engine which was very high indeed. The explanation for the high rates appeared to be that the piston velocity on the B-1 was relatively low, so that the oil film thickness was small, and the author would expect a fairly high wear rate due, presumably, to mechanical abrasion. What worried Mr. McConnell was that the engine was intended for screening oils which were usually used in service in engines which were affected predominantly by corrosive wear. That was the reason for using high alkaline oils. Therefore, if one were looking at those oils in an engine which was exhibiting high wear rates due to mechanical abrasion, was there any risk of screening the wrong property? Would one let slip through the net some oil which was very good in combating corrosive wear, but where anti-wear properties were masked in an engine which was responding to mechanically induced wear?

With regard to the low sulphur fuel problem, this was a particularly interesting piece of work, because one was now seeing evidence for the first time of a combustion effect. It seemed from studying the data in the paper that the authors were correct in assuming there was some sort of effective low cetane number in operation. Although the cetane number of the suspect fuel in the turbulent chamber appeared to be satisfactory, once the fuel was introduced into a quiescent chamber then combustion seemed to be slowed down. In that connexion, the blue-to-purple flashes which the authors observed were very important, because the blue flashes were probably indicative of a non-luminous flame. In other words, there was a long premixing time available for some of the fuel, which fitted in with the low cetane number concept such that one observed the blue flame occurring when the piston approached the ports. The red-to-orange flame normally observed occurred at the back end of combustion and was associated with a luminous type of flame. Scott of Ricardos in his papers had assigned temperatures to various types of flame, giving the luminous flame a temperature of the order of 1000°C and the blue flame one somewhere around 1800°C. One wondered whether there was a straightforward temperature effect in the B1 engine due to an effectively low cetane number of the fuel. This would give a high non-luminous flame temperature at the back end of combustion, giving rise to excessively hard combustion deposits due to hydrocarbons coking at 1800°C instead of at the normal 1000°C.

MR. G. H. CLARK, F.I.Mar.E., said that his initial comments would refer to lubricant rather than fuel and combustion investigations (although it was difficult to treat them in isolation as in practice they were so inter-related). It was becoming increasingly obvious that there was a widening gap between the lubricating oil requirements of large crosshead and trunk-piston marine diesel engines and those of conventional small bench test engines such as the Caterpillar, Petter, Poyaud, etc. This was particularly true when residual fuels were used. Furthermore, at present there were no really suitable official oil specifications which embraced the requirements of the large, highly turbo-charged production engines, both crosshead and trunk piston. Specifications such as the old Series 3, MIL-L-2104B, and the current MIL-L-2104C and DEF.2101D were intended only for evaluating lubricants for high-speed, automotive-type, 4-stroke diesel and petrol engines. All tests in the diesel specification test engines were run on premium quality distillate fuels rarely available in the marine field. Gasoline engine tests were of even less value when assessing performance requirements in large engines. Such specifications did not even mention alkalinity or TBN requirements—a vital characteristic in large engines burning residual fuels.

To a large degree, the Abingdon B1 was at present probably the most versatile test engine available. Results obtained from it appeared to correlate very closely to those obtained in production crosshead engines in terms of liner wear, piston ring wear and ring zone cleanliness. It was understood that although at present all published data related to the engine used with a uniflow scavenged cylinder, a loop-scavenged cylinder design had also been tested. Would the authors care to comment on any results obtained with the latter design as, to date, this engine type dominated the marine diesel crosshead engine field.

If these results were as encouraging as in the uniflow-scavenged design, the speaker would support the use of this engine as a standard laboratory test engine, around which suitable specifications could be developed to cover the lubricant requirements of even the latest highly turbo-charged engines in production, or on the drawing board. Such specifications could take into account fuel quality, engine loading and other important operating variables.

The comments on turbulent and quiescent combustion chambers were interesting, although it was perhaps rather surprising to read that conditions in a Sulzer RND 76 cylinder were such that turbulence was relatively low. In the smallest crosshead engines in production, namely the Bolnes DNL and the larger Smit-Bolnes 300 HDK, both of which were capable of operating satisfactorily on residual fuel, good combustion was obtained in a relatively small diameter cylinder, although special attention had been paid to the type and location of the fuel injector.

The recently developed Abingdon B2 engine still appeared to be relatively unproved, in so far as results from both the combustion and lubricating points of view could be correlated with the requirements of large bore trunk-piston engines such as the Pielstick PC3, the Werkspoor TM410 and the MAN 52/55, when burning residual fuel (see Table V).

TABLE V

	Bore mm	Stroke mm	rev/mm	bmep	piston speed m/sec
Bolnes DNL	190	350	500	9·6/	5·75
Smit-Bolnes 300 HDK	300	550	300	11·6/165	4·6–6·9
B1	203	273	300–400	—	2·7–3·6
B2	168	184	800–1400	—	4·9–8·6
Sulzer RND 76	760	1550	122	10·49/	6·3
Sulzer RND 105	1050	1800	107	11·7/166	6·48
B & W KGF98	980	2000	104	mip 11·1/158	6·8
Pielstick PC3	480	520	470	19·3/276	7·97
Werkspoor TM410	410	470	550	17·6/250	8·3
Mitsui V60M	600	640	370	20·2/287	7·8

The majority of such "2nd" and "3rd" generation engines were fitted with two-part pistons employing positive cocktail shaker cooling. The B2 piston with its thick crown and upper wall sections did not appear to give as efficient cooling as might be desired, especially in the ring zone. The top ring groove temperature of about 240°C would be considered excessive, judged by modern large engine standards. Undoubtedly this would impose much greater thermal conditions on the lubricant than would be normally encountered in practice and the speaker wondered whether the ring zone conditions were not too severe.

On the other hand, with much of the piston heat being transferred down past the ring zone to the skirt, the under-crown temperatures were lower than in the B1 test engine, but were, even so, quite high for modern trunk-piston engines.

Perhaps an inserted cooling coil, as in the PC2 engine, might reduce these temperatures to more realistic limits.

The investigations on the "bogey" of excessive wear, scuffing and seizure when burning residual fuels of very low sulphur content, in combination with high TBN cylinder oils using the B1 engine were most interesting. It was rather difficult to comment on results obtained with a simulated fuel of distillate type, when all known cases of this trouble had been on residual fuel, and these results could be misleading. It was agreed that it was difficult to accept that in actual cases of serious trouble reported from several different types of engines employing the low sulphur residual/high TBN oil combination, the initiating cause was unneutralized alkaline additive which was converted into much harder compounds such as calcium oxides, possibly inter-reacting with carbonaceous deposits to form even harder abrasive compounds. Even these harder deposits were believed to be appreciably softer than cast iron.

It was understood that this problem had only been encountered in slow-speed crosshead engines. It would be interesting to know whether or not this was only a coincidence.

MR. G. W. VAN DER HORST, said that the diesel engines currently used for marine propulsion were highly developed engines requiring sophisticated lubricants for optimum operation. The development of such lubricants was a major effort and was very time consuming. Before commercialization extensive evaluation of the lubricant in full scale equipment under service conditions was needed to demonstrate the desired performance. However, as the authors had indicated, prior to field testing extensive laboratory work was needed to sort out the most promising oils and obviously, although certain aspects could be studied in bench tests, this should be done in engine tests. The authors, not satisfied with existing engines, had decided to design and construct the engines required for their test work and he complimented the authors on their ingenuity and fine engineering.

When faced with similar problems as the authors the speaker's company had chosen, however, not to develop specific tests engines, since even with such engines no direct correlation with the field could be obtained and the best would be a good ranking of oils, as the authors had found. In their opinion this situation did not justify test engine development for laboratory work and two existing engines only slightly modified had been adopted.

For many years the Bolnes engine had been used for the development of marine cylinder oils. Initially as a two cylinder naturally aspirated engine(1), operating on 2·7 per cent sulphur residual fuel at a BMEP of 5·3 bar. To follow the increase in output of full scale engines, a supercharged three cylinder engine operating on approximately 2·8 per cent sulphur residual fuel with a BMEP of 9·8 bar was now used, and in the near future 11·8 bar was foreseen.

With a 72 hours procedure wear was evaluated, as were piston deposits and port blocking. The wear level was 0·31–0·65 mm/1000 hours' liner wear and 0·57–1·17 mm/1000 hours' top ring wear. Fig. 26 showed a typical wear pattern. (In connexion with the very low level of ring wear in the Abingdon B-1, could the authors give an indication of the repeatability of ring wear?) The wear was determined by liner measurement, ring weight loss and drop oil iron content. The latter had the advantage that stabilized wear could be evaluated while it was indicative for wear taking place as could be seen in Fig. 27. Although the general level of port blocking was not high in the Bolnes engine as operated, lubricant effects on port blocking could be evaluated. Since in some current models and makes of full scale engines port blocking was sometimes a problem, he would be interested to hear the authors' comments on the ability of the Abingdon B-1 to evaluate this aspect.

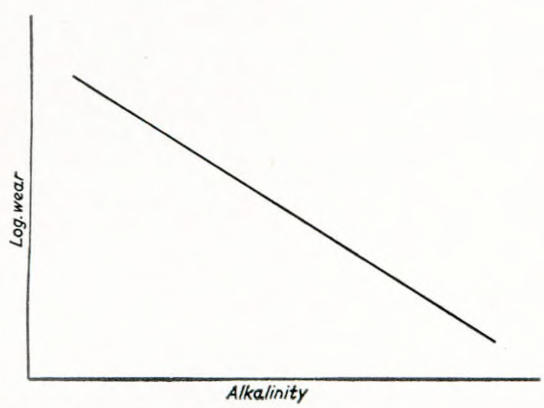

FIG. 26—*3 cylinder Bolnes engine wear*

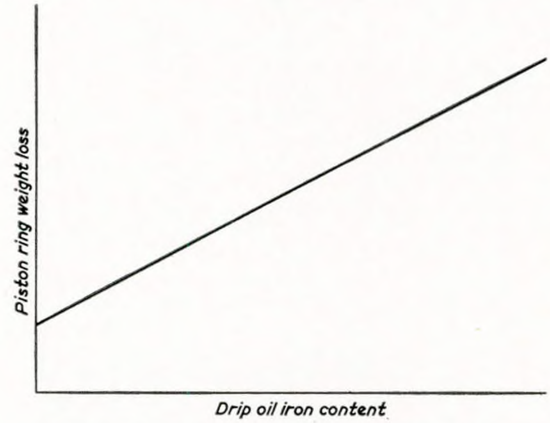

FIG 27—*3 cylinder Bolnes engine wear*

His company evaluated lubricants for medium speed diesel engines in a 1G Caterpillar engine uprated and modified to operate on residual fuel(2). With a 72 hours procedure significant piston deposits and crankcase oil deterioration were obtained. Piston deposits of oils A, B and C when tested in this Caterpillar test rank as expected on the basis of field performance, as indicated in Fig. 28.

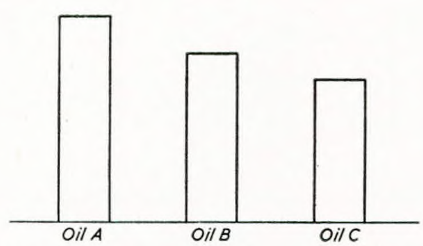

FIG 28—*RF-2 caterpillar piston deposits merit*

MR. J. F. BUTLER, F.I.Mar.E., in a contribution read by Mr. A. E. Franklin, said that the research engines described in the paper were most interesting examples of fitness for special purpose. He would like to ask the authors whether in relation to Fig. 5 they had used stroboscopic lighting to view the scavenge ports as he felt that this can produce useful information, particularly about the adequacy of the blow-down period.

What was the running life of the split rubber Walkersele in the piston rod stuffing box? For service engines one was inclined to fight shy of rubber for a duty like this but possibly designers were over conservative. In the test engine the life of the seal would be helped by the use of the chrome plated and ground piston rod, and by the large crank chamber volume which would reduce the intensity of oil throw.

It was many years since needle roller bearings had been used for top ends in an engine and in large engines difficulty would be experienced in obtaining the necessary housing accuracy in view of the need for a split housing for the purposes of assembly and dismantling. Nevertheless the authors were to be congratulated on obtaining such success with this type of bearing.

With regard to combustion of low sulphur fuel the relatively small combustion chamber of the B1 engine, together with the low piston crown temperatures and rather poor spatial matching of fuel spray and air, would tend to exaggerate the poor combustion qualities of the fuel. Engines with injection inwards from the periphery of the combustion zone towards the centre were more tolerant. This was not a criticism of the test engine as in this case it might be of advantage to have a combustion chamber particularly sensitive to fuel quality.

The authors had taken on a very difficult task in designing test machines to assess the merits of different oils in relation to the performance of the piston, ring, and liner combination. In high output engines piston rings had to perform an almost impossible task in sliding and sealing against the cylinder wall with high pressure loading, speed varying from zero to high with inadequate lubrication and in a hostile environment. To add to their discomfort they were restrained by the piston to lie at an undesirable angle to the liner surface which was itself made partly conical by temperature gradient. It was small wonder therefore that oil improvements were frequently masked by minute changes in ring geometry due to minor alterations in engine performance.

Fortunately a great deal of work was now in progress on assessing the conditions under which piston rings have to work, including the important contribution described by Mr. Baker himself and others [4], and there was some hope that in the fairly near future it would be possible to design rings and pistons in the confidence that they would not suffer random scuffing or excessive wear from other causes. Could the authors give their views on the possibility of modifying the ring and groove arrangements in the test engine as required to simulate the conditions obtaining in various good and bad engines in service?

Finally, Mr. Butler wished to congratulate the authors on the open-minded engineering which had enabled them to design and build test engines with such ample reliability margins on all components other than those to be tested. To the development engineer this had so far appeared as an unattainable Utopia but it could be that future far-sighted management might appreciate the advantage of building several such rigs to develop individual components, instead of concentrating all potential troubles into one vulnerable prototype.

PROFESSOR Y. WAKURI wrote that he was especially interested in the test engines built by the authors, because he had also been carrying out research work on the scuffing of cylinder liners and piston rings in large sized marine diesel engines, using a specially designed two-stroke cycle test engine.

This engine had a cross-head type construction, and was similar to the authors' Abingdon B-1 engine except for the piston velocity.

He wanted, therefore, to comment on one point concerning the piston velocity. It appeared that the piston velocity of Abingdon B-1 engine was unusually low compared with those of large sized marine diesel engines which were the subjects of the study.

With regard to this point, the authors had given as their opinion that the low piston velocity resulted in a somewhat more severe lubrication condition than high piston velocity, because the lower velocity resulted in a thinner oil film and consequently increased the level of asperity contact.

This explanation would be correct so far as the oil film thickness and the level of asperity contact. However, there was another opinion concerning the severity of lubrication condition.

According to an experiment[3] which had been carried out with a reciprocating piston ring tester, the result of the friction analysis was as follows.

Although almost all loading (over 90 per cent) was supported by hydrodynamic oil film pressure, the friction force acting between the cylinder surface and the piston ring surface was mainly attributable to the real contacting asperities.

In such a circumstance, heat generation per unit time at the severe contacting regions, which seemed to be the most important factor bearing on the occurrence of scuffing or mechanical wear, had to be almost dependent upon the piston velocity.

M. J. Neale had also shown in his survey of piston ring scuffing[4] that high piston speeds tended to make engines more prone to scuffing. As a result of these studies, it seemed that higher piston speeds would result in more severe damage than the lower piston speeds.

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Authors' Reply

The opening remarks of Ir Shrakamp were sincerely appreciated by the authors, coming as they did from so eminent an authority in marine lubrication as Mr. Shrakamp, the "father" of the extensive Shell Amsterdam marine engine test facility. Mr. Shrakamp had even longer experience of cylinder lubrication research than either of the authors, so that his encouraging comments and high level of agreement with their basic arguments on test engine requirements were a source of profound satisfaction.

Regarding his enquiry on the relationship of TBN and wear, this had received considerable attention by the authors ever since their earlier experience with a Bolnes engine had suggested a fairly strict proportionality between these factors. Such proportionality did not appear to be reflected very accurately in sea-going engines. Significant changes in TBN in otherwise similar oil formulae did, of course, produce corresponding differences in wear rates, but this pattern could not be closely replicated when the additive packages were dissimilar. Fig. 29 showed the relationships between TBN and wear in the Abingdon B-1 for 20 different oils. Some of these were generically similar and could be joined by firm lines, but the levels and slopes of the lines were not closely repeated between different types of formulae. This pattern related very closely to the shipboard data collected on those oils for which such data were available.

Figs. 30 and 31 showed the wear patterns in the B-1 liner for two oils of differing characteristics. These were diametral measurements taken in four directions. They gave an impression of the distribution of wear around the bore and it could be pointed out that there was no tendency for wear in one direction to predominate.

Although they had used cylinder draining evaluations extensively in former times, as reported in a paper by Baker and Davies[9] as long ago as 1964, more recently the authors had tended to use this method less for wear evaluation. Several factors had influenced this change. Firstly, the Abingdon B-1 had been specifically designed to permit rapid access to the working cylinder and piston rings and had shown itself capable of permitting meaningful wear measurements in only 100 hours' operation by the measuring methods which were now available. Secondly, wear determinations by drip drainings did not possess the required level of precision necessary to differentiate with certainty oils of current high performance which would thus need measured wear determinations in any event. Thirdly, from

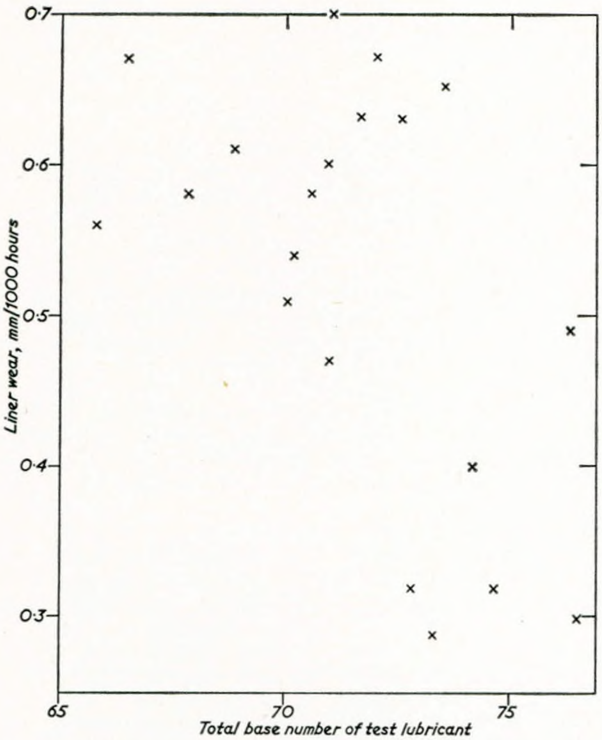

FIG. 29—*Abingdon B-1 tests showing relationships between T.B.N. and wear for twenty different lubricants*

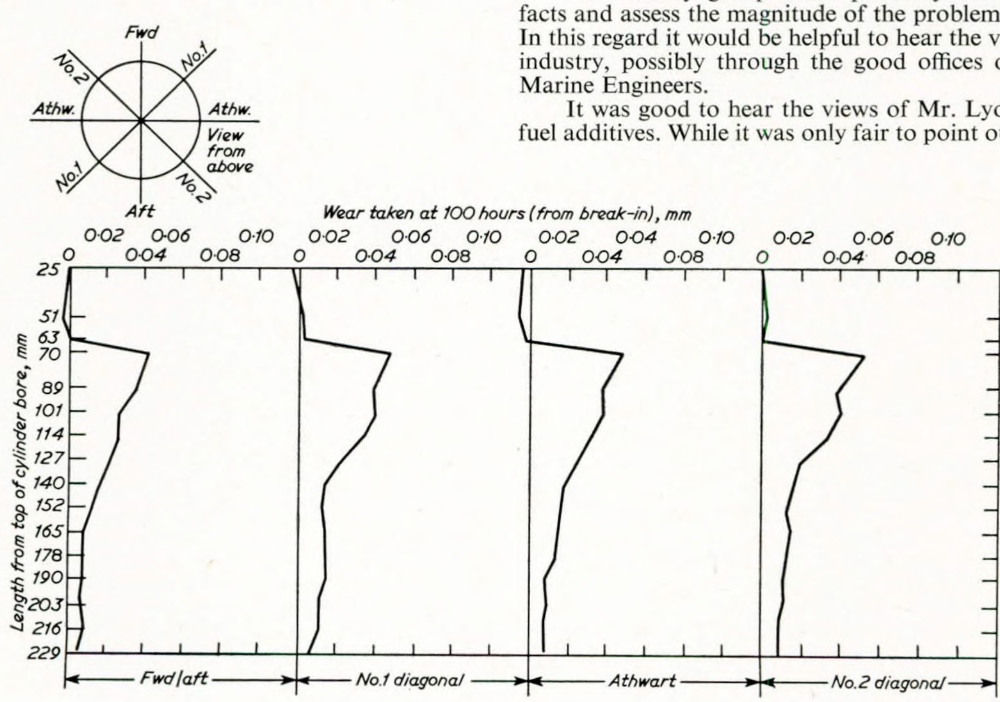

FIG. 30—*Abingdon B-1 liner wear AB 1/62*

previous discussions with Mr. Shrakamp, the authors knew he had expended considerable effort in sea trials to understand how best to sample drainings in full scale engines. This appeared to be a very wise course to take in Mr. Shrakamp's case as his prime test tool was the full scale Sulzer at Amsterdam, which would require more time and effort to dismantle and measure than the Abingdon B-1.

Nevertheless, the authors had not lost sight of the potential value of on-going evaluation of oil drainings and actually had some work on this aspect in their current programme which they hoped they might be able to publish eventually.

It was good to hear Mr. Shrakamp confirm the authors' observations on low sulphur fuel difficulties and their relationship to combustion. Obviously scale effect must be of importance to combustion, and it was largely with this fact in view that the initial work to relate the small Abingdon engines to full scale had been conducted. Moreover, the authors' experience of using the 0·4 per cent S gas oil quoted in the paper coincided with Mr. Shrakamp's experience with his 0·5 per cent S gas oil in his larger engine, in so far as no scuffing occurred.

On this evidence, it probably was true to suggest that fuels containing more than one-half per cent of sulphur by natural occurrence would not give rise to difficulties at sea. However, in the authors' view this was an entirely different matter from saying that the inclusion of 0·5 per cent sulphur in any fuel would remove the problem. From the evidence so far it appeared that the fact that the fuels in question happened to contain very little sulphur was merely a loose way of identifying them as potentially dangerous to engines with quiescent combustion.

The authors could endorse Mr. Shrakamp's comments on co-operation over matters of common concern to lubricant suppliers. The Norwegian Large Bore Group, which had emerged from the events he had recalled, had been a great success with a fall-out of usable data which had been of enormous value to the participants. However, the circumstances of the "low sulphur problem" were rather different. While the companies of Mr. Shrakamp and the authors were prepared to support a degree of work on this sort of problem and allow its results to be published freely for the benefit of marine trade and oil industry alike, it might be difficult to justify sufficient effort to research the problem fully within any one company.

An industry group could probably assemble the relevant facts and assess the magnitude of the problem most realistically. In this regard it would be helpful to hear the views of the marine industry, possibly through the good offices of the Institute of Marine Engineers.

It was good to hear the views of Mr. Lyon as a supplier of fuel additives. While it was only fair to point out that the authors had not been able to support most previous claims of combustion improvers, the evidence on low sulphur fuel to date was such as to suggest that Mr. Lyon could probably be of assistance to owners forced to use such fuel. Regretfully, the authors could not let Mr. Lyon know the origin of the low sulphur fuel as the European supplier had declined to make any statement on this despite repeated requests by the authors.

From the tone of Mr. Lyon's contribution it appeared likely

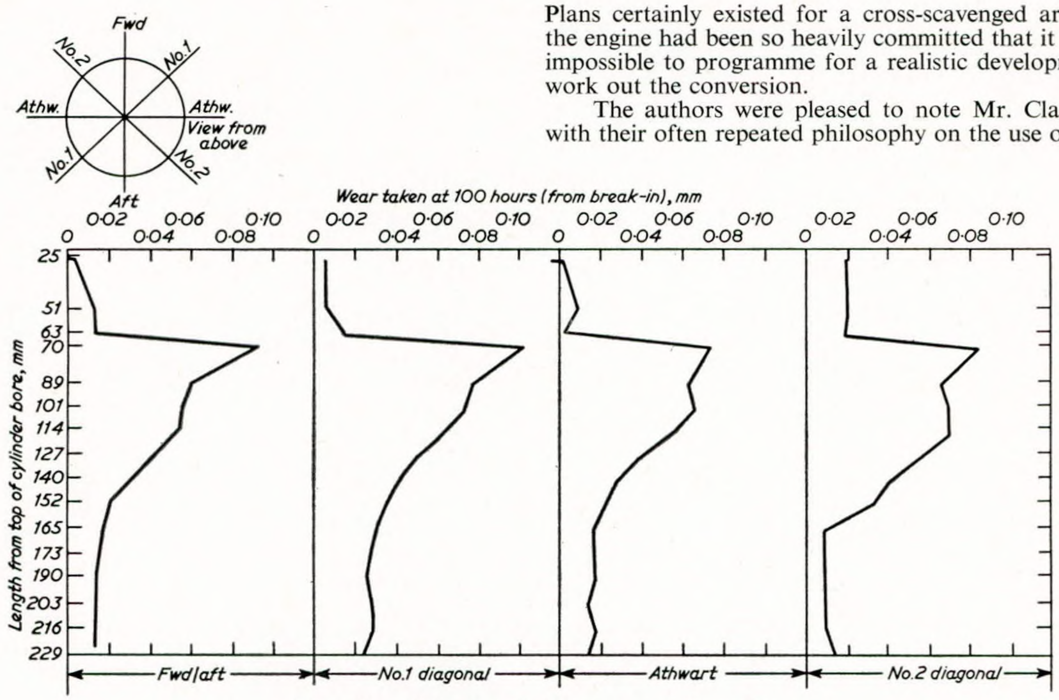

FIG. 31—*Abingdon B-1 liner wear AB 1/79*

that his company might undertake some combustion research to alleviate the problem with an additive designed to reduce low temperature stability of the problem fuels. Experiments along these lines appeared entirely logical to the authors and they wanted to express their wishes of success to his endeavours. At the same time they felt it necessary to point out that their own observations must be treated as evidence rather than proof that low temperature stability was the prime cause of the trouble. Thus it was to be hoped that Mr. Lyon would obtain independent corroboration of the cause and cure before offering a new fuel additive to the marine industry. Nevertheless, it was very encouraging to note the speed of response displayed by Mr. Lyon in extracting the message intended by the authors and translating it into a practical development route towards the discovery of a cure.

In reply to Mr. McConnell, the authors had certainly not intended to imply that the balance of corrosive to abrasive wear had been upset in the Abingdon B-1 engine. If this were the case they would have encountered ring and liner scuffing with oils which were satisfactory in service, but this situation had never been experienced. The reason that relatively smaller oil films gave higher wear appeared to be due to two factors. Firstly, the unit combustion dosage was increased, and secondly, the film was subject to higher instantaneous pressures. As discussed in reference[9], crystalline scale difference in surface topography and reactivity had to be expected. The net result of these differences had been of benefit to the present work in enabling wear rates to be exaggerated without significantly changing their relationships.

As regarded Mr. McConnell's remarks on flame luminosity, the authors had studied Scott's work with transparent combustion chambers and agreed the parameters of non-luminous combustion in such circumstances. However, they would point out that neither the oxygen availability nor the pressure at the point where the ports are uncovered related to Scott's observations. Moreover, the authors' hypothesis appeared to fit the known circumstances of precipitation of the hard carbon phases observed. However, they would stress that the mechanism they suggested was only a hypothesis at present and they would welcome any alternative mechanisms put forward for consideration.

The authors expressed sincere appreciation of the kind remarks of Mr. Clark, who, as a one-time colleague, had been a staunch supporter of the ideas behind the Abingdon B engines. Plans certainly existed for a cross-scavenged arrangement but the engine had been so heavily committed that it had been quite impossible to programme for a realistic development period to work out the conversion.

The authors were pleased to note Mr. Clark's agreement with their often repeated philosophy on the use of well designed test engines to predict future requirements. It had always been surprising to them that the advantages of down-scaling, such as the lowering of thermal loads at high cyclic pressure and the ability to use under-stressed components from other, larger engines, were not more frequently used.

Regarding the specific engine types mentioned by Mr. Clark, the Sulzer RND 76 did not fall outside the range of large bore engines in swirl number, and the authors did not understand why Mr. Clark thought this engine type to be exceptional. Both the Bolnes DNL and the Smit-Bolnes 300 HDK incorporated the unusual combustion system originated by Professor Kroon[10]. This system used very accentuated tangential injection which permitted a degree of impingement on the walls of the bowl. It obtained swirl from three directions, firstly from sharply tangential scavenge porting in sections of the liner having sufficient thickness to obtain reasonable guidance; secondly from combustion induced asymmetric injection; and thirdly as residual swirl from the blow-down.

The Kroon system was quite interesting as a prime mover, in that it permitted quite high smoke limits while using relatively low injection rates, and enjoyed a well deserved reputation in its own field. However, this form of combustion could in no sense be compared with large bore practice.

Mr. Clark's comments on the Abingdon B-2 piston temperatures were interesting. The authors were well aware of current methods used to cool medium speed engine pistons. However, the thermal loadings in the B-2 piston were still very low and it would be a long time before the need for such sophisticated techniques as Mr. Clark advocated would be felt.

Perhaps here it should be made clear that it was quite simple to change surface temperatures up or down in components with low thermal loading such as those used by the authors. As sizes and power outputs mount, heat fluxes increase to the point where the commercial builder had to incorporate special features. The actual temperatures depicted for the B-2 were chosen to represent a certain type of engine known to be critical to its lubricants. Should events dictate otherwise, considerable flexibility for temperature changes existed without any need to go to the lengths of cast-in oil tubes.

Unfortunately, the authors could not agree with Mr. Clark that all cases of damage with low sulphur fuel involved supplies of residual fuel. In fact several of the more recent cases of trouble had involved fuel which appeared remarkably like that tested.

They had, incidentally, heard of such a fuel causing difficulties in a Sulzer RND 76, if this could be confirmed it might convince Mr. Clark that this type really did employ quiescent combustion.

Mr. van der Horst's contribution presented some problems for the authors as he had not made it clear whether he agreed with their arguments on the need to simulate combustion and was merely using engines which happen to be easily obtainable, or whether he thought his Bolnes and Caterpillar had technically advantageous features over the Abingdon B-1 and B-2.

On the assumption of the former case, the authors would agree that the Bolnes engine could be quite useful for rough sorting. It was moreover a fine, reliable engine and spare parts were readily obtainable, particularly so in Mr. van der Horst's case. The Bolnes Company had been found to be most helpful when the authors undertook some development of their Bolnes engine some years ago, in a period which coincided with that company's development of the DNL. However, sound and useful though the Bolnes had proved to be, it contained a number of basic characteristics which prevented its conversion for the test purposes of the authors' company.

Firstly, the general scantlings were found to be insufficiently robust to permit operation above about 11·5 kg/sq cm on a continuous basis. Morever, this level of BMEP could only be obtained by adhering to existing combustion systems which generated somewhat lower maximum gas pressures than appeared to be representative of large bore practice. The principal reason for this was the rather low specific air throughput which the Kroon combustion system would tolerate without smoke. The relationship between specific air throughput and cycle pressure had already been demonstrated by one of the authors in reference.[6]

This meant that the most valuable method of surface temperature control, by the adjustment of air throughput, would be impractical without encountering dangerously high peak cycle pressures. Furthermore, the provision of an adjustable air supply presented a problem in the size of compressor and coolers which would be necessary for the independent supply to a 3-cylinder engine. Here again the difference could be seen between a good commercial engine, developed to minimize the influence of air supply changes due to fouled turbocharger nozzle ring and partly fouled ports, and requirements in a research engine.

As mentioned in reference[9], the cylinder air flow pattern in the Bolnes 2L was found to be somewhat asymmetric and this, coupled to the offset injection arrangement, resulted in non-regular temperature distribution. It appeared that this factor contributed to the tendency of some scavenge ports to plug rather readily. Mr. van der Horst wrote of the ability of the Bolnes to evaluate the port fouling tendencies of lubricants. However, unless this variable can be separated from others, such as wear, by the maintenance of constant air throughput it could result in the procurement of distorted data. Port fouling, under controlled conditions, could of course be induced in the Abingdon B-1 engine.

The next shortcoming of the Bolnes as a research test engine was the level of accessibility to the piston and liner. Delays due to stripping and inspection had been a prime factor in the development of the "sandwich" method of wear testing by iron recovery discussed in reference[9], but as mentioned earlier, this method of wear determination proved inadequate for later lubricant development. Delays were also experienced in installing internal instruments in the Bolnes, due to its monobloc construction and the need to make the entire installation on the engine itself. The procurement of direct data taken during operation had not appeared to be a large factor in earlier oil test work, but it seemed of importance to the authors who had since demonstrated its significance.

Finally, the Bolnes was only available as a multicylinder engine. Moreover, attempts to run more than one lubricant test simultaneously had not been successful owing to cross-over effects with the common scavenge chamber. While it might be argued that the availability of simultaneous data from more than one unit might substantiate results, experience showed more of a tendency to complicate and confuse. Moreover, in certain cases, the provision of sufficient test lubricant or test fuel to operate two or three cylinders could present difficulties in obtaining pilot or field test supplies.

Fig. 26 suggested entire linearity between lubricant alkalinity and the logarithm of wear. The authors agreed that their own Bolnes results indeed tended towards this pattern which appeared to be a general characteristic of turbulent combustion. However, as discussed in their reply to Mr. Shrakamp, such linearity was not true for full scale engines. If it was, as Mr. van der Horst suggested, the required data could be determined directly from an alkalinity determination, leaving little incentive to develop an engine test.

For medium speed test work, Mr. van der Horst advocated the 1G Caterpillar, an engine with about the highest turbulence found anywhere. With the greatest respect, the authors could only suggest that he considered the differences in combustion between this antechamber system and the quiescent open chambers used in medium speed engines. Certainly the former would relate acidic neutralization at the piston rings to alkalinity. However, in most other characteristics, the violent combustion reaction in the chamber and combustion nozzle were totally different from normal practice. A much better solution appeared to be the conversion of the 1G to direct injection as had been done in at least one laboratory. Such a conversion possessed limitations in cycle pressure and spray path length but must be far more representative of medium speed engine characteristics than Mr. van der Horst's.

In making this observation, it might be as well to point out that for data to be convincing to engine builders, the circumstances of their generation had to be made clear. Moreover, unless they referred to practice similar or relateable to the builder's, they would fail to provide more than a very rough indication to him. In these circumstances, the oil man should not be surprised if his data were insufficiently convincing to influence the decisions of the builder. Fig. 28 indicated this quite well. Mr. van der Horst claimed correlation on the basis of three oils and a similar pattern in service engines. However, for all anyone might know, the circumstances in which his ratings were gathered in the laboratory could totally omit an important characteristic of quiescent combustion.

Coming from the doyen of practising British engine designers, Mr. Butler's comments were particularly welcome to the authors. They had certainly used stroboscopic lighting to view and photograph the ports. However, they had not yet conducted any flow pattern observations by this method, but they agreed with Mr. Butler that such methods were entirely practical and potentially useful.

Two replacement Walkerseles had been fitted in five years, once because of crankcase oil reaching the scavenge chamber and once because the sealing surface appeared slightly worn. They had been surprised at the large quantity of oil in the space above the crosshead which needed to be controlled and returned to the sump, but agreed that it could well be rather less than that in large cross-head engines. They had not been trying to persuade engine builders towards needle roller bearings for top ends. Nevertheless they thought it useful to let the industry know of the good results they had experienced as such information had not been available previously.

Mr. Butler's comments on inward injection in the context of low sulphur fuel were very interesting. It might well be, as he suggested, that engines containing this feature could cope better with difficult fuels than those with central injectors. However, against this argument the authors had heard reports of at least one valve-in-head crosshead engine with radial injectors suffering from the problems of burning low sulphur fuel at sea.

The authors were very pleased to observe the keen interest which Mr. Butler had displayed in trying to understand the true functions of piston rings. In their view this represented a significant effort by an engine builder of note to develop design criteria for piston rings. The authors' company was resolved to analyse piston rings on the lines indicated in reference[4] and a good deal of effort had already been placed into differentiating between engines having rings which performed well and those which performed poorly. The Abingdon B-1 engine was well suited to demonstrate such differences provided it could be made available for this work in the circumstances of competing requirements.

Far from equipment to properly evaluate impinging variables being an unattainable Utopia in engine development, the authors felt that the levels of performance currently required from service engines made such equipment virtually essential. Nevertheless, since test equipment could only point the engineer in the right direction, it was understandable that management needed strong justifications before according to its procurement. However, in view of reported difficulties in several recent new engines, it was hoped that the climate of management opinion might become more receptive to proposals for such equipment. If the work described by the authors, as part-time engine builders of test engines, could be used to advance such concepts, they could be particularly satisfied. Nevertheless they realized only too well that every possible short-cut and loan of components from other sources must be sought to keep the overall cost of special test equipment to the minimum commensurate with its performing its tasks adequately.

The written contribution by Professor Wakuri of Kyushu University was welcome as it gave the authors an opportunity of expanding their viewpoint on piston velocity. Firstly, they agreed entirely that scuffing damage increased rapidly with engine speed. However, unlike Professor Wakuri's rig test, minimum oil film thickness in an engine piston ring was controlled not only by surface topography but also by the self-generated ring profile referred to in reference[4]. Until the profile was established under conditions of non-destructive wear, serious metallic interaction was likely and the severity of consequential damage was, to a large extent, velocity dependent. Thus, as Professor Wakuri pointed out, greater scuffing *damage* could be expected as velocity increased.

The authors recognized the necessity to understand the factors leading to a safe transition of surface profile and had studied this aspect separately in so far as it could be influenced by running-in procedures. However, the principal objective of the present work was to study lubrication after representative profiles had been established, thus it was essential to impose operational dosages upon the oil film representative of severe conditions in that mode.

It was now possible to analyse oil film thickness and oil transport rates by the methods described in reference[4]. Such analyses had confirmed that to operate at significantly higher piston velocities would have not only increased film thicknesses unrepresentatively but would also have produced increased net oil transport by the top piston ring, as shown in Fig. 32. In the latter circumstance the choice would have been to either operate at a representative oil feed rate and encounter oil starvation, leading to a predominantly scuffing situation, or to increase feed rate and accept a reduced oil dosage level.

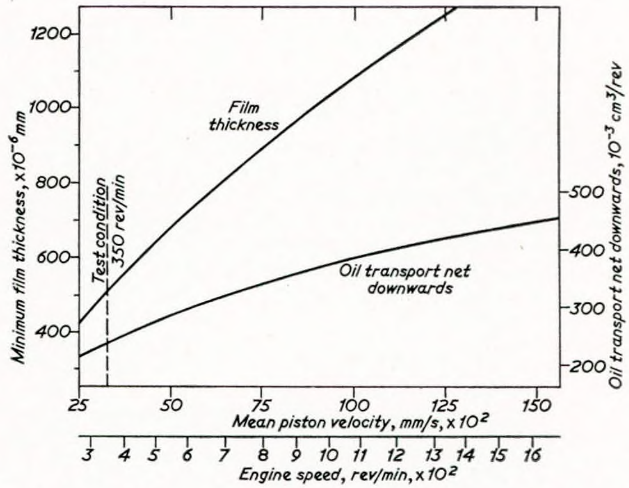

FIG. 32—*Abingdon B-1 engine influence of piston velocity upon top ring lubrication*

A further incentive to operate at low sliding velocity was the need to maintain reasonable similarity of combustion conditions, particularly in reference to swirl rates. Again, it was found that much increase in speed above 400 rev/min tended to produce increasing turbulence at an above linear relationship. In these circumstances the authors believed that they had made the correct choice of operational speed. Nevertheless they would look forward to reading of the considerations which led Professor Wakuri to obtain operational similarity with large engines at higher rotational speeds in his test engine.

ADDITIONAL REFERENCE

9) BAKER, A. J. S., and DAVIES, R. T. *J. Inst. Pet.*, vol. 50, No. 484.

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Related Abstracts

Filter Research Provides Effective Engine Protection

This article describes the equipment and methods used in the filtration laboratory of Vokes Ltd, at Guildford, Surrey, where for example microscopic slides showing the inlet and outlet conditions from experiments are compared in the Vokes Quantimet image-analysing computer, and efficiency readings obtained. In the marine field, the introduction of unmanned engine rooms has led to the use of mechanized strainers, which however suffer from progressive choking, as back-flushing arrangements rarely give 100 per cent efficiency. A further disadvantage is their inability to control the particle level of fine abrasive materials in medium-speed engine lubrication systems, but depth-type media such as felt control the maximum particle size and trap much of the fine contaminant; for some years the laboratory has been investigating the problems associated with the automatic cleansing of felt media, and a felt has been developed which can be back-flushed on a set time cycle, which has ten times the life of a conventional felt cartridge. Extensive research into the field of fuel-oil equipment has led to the development of a heavy fuel module for removing water and providing filtration to a 5-micron level, in fuels of up to 3500 sec Redwood 1. The results of a two-year test aboard a ferry using 200 and 400 Redwood 1 fuel in a Pielstick diesel engine are given. The synthetic depth material used, called Vaflon 203, is both resistant to sea and fresh water and very effective in removing solid and waxy contaminants. The filter/coalescer units are used with diesel engines because water contamination can cause serious corrosion of the precision-made fuel-injection equipment, and they also find use in decontaminating the oil in stern-tube systems.—*Mason, R. L. et al, Diesel and Gas Turbine Progress. June 1973, Vol. 5, No. 5, pp. 26, 27.*

Towards Wear Reduction in Engines Using Residual Fuel

Although this paper describes an investigation into the causes of wear in direct drive marine Diesel engines, it is equally applicable to medium speed four-cycle engines. For reasons of convenience and cost the work was carried out on two Crossley H.H.9 235 × 406 mm, (9·25 × 16 in) horizontal four-cycle trunk piston open crankcase mono-cylinder engines. The programme was sponsored by the British Ship Research Association, and took place between 1959 and 1968. The operating factors investigated were jacket temperature, the comparison of a lubricating oil of high alkalinity as compared with a straight mineral oil, fuel composition and the degree of centrifuging of the fuel. Trial of different carefully selected liner materials was made, as well as liners banded over the upper portion with hard surface materials. Ring materials included unhardened and hardened plain rings and inlaid rings sprayed with molybdenum and chromium. Results showed that a high jacket temperature was important, irrespective of whether or not an alkaline oil was employed. The sulphur content of the fuel had usually the most influence on wear but Conradson carbon value and ash content could also be important. Fifteen fuels of widely ranging composition were tested. Centrifuging only reduced wear in the case of one especially dirty fuel out of seven fuels regarded as typical of those bunkered in representative areas of the world.

High basicity additive lubricating oil reduced liner wear to about one tenth and top ring wear to about one sixth of that measured using a straight oil. In comparison with the standard build consisting of a vanadium titanium liner and unhardened rings, the following liner and ring combinations gave useful wear reductions:

- hardened top ring in standard liner;
- austenitic liner with unhardened rings;
- 2 per cent nickel/copper liner and unhardened rings;
- molybdenum banded liner, especially when used with hardened top ring.

An outline of rates of wear of liners, rings and ring grooves in direct drive and medium speed engines is given, together with suggestions for improvement arising out of the investigation.—*Burtenshaw, R. P. and Lilly, L. R. C. Trans.I.Mar.E. 1972. Vol. 84. pp. 389–424.*

Investigations of Cylinder Lubrication and Wear in Sea-Going Installations

Causes of the variations of wear rates observed in ship engines are reviewed and ways of reducing the variations are discussed. Basic reasons for the selection of two seagoing installations for research test work are given, and a design of experiment is described, together with an outline of an accurate and reliable method of liner wear measurement (referred to as Dimple Method and first reported at C.I.M.A.C. 1971), which was extensively employed in the investigations described in this paper. The statistical examination of the results from the experiment design is then discussed in detail with regard to significant effects of engine design and operation on liner wear pattern. Differential wear patterns between diametrally opposite liner areas are highlighted and the similarity of results from different types of engines, concerning the relationship of cylinder liner, piston ring and piston groove wear, is explored. A conclusion is finally drawn that further reduction of wear may be small if pursued through improved lubricant quality alone.—*Baker, A. J. S., Casale, P. G., and Breyer, H., Europort 73 Conference, pp. 28–33.*

Combating Wear in Large and Medium Diesel Engines Operating on Residual Fuels

The first part of the paper is a report on the experience gained by Fiat Grandi Motori in the operation of several two-stroke engines in respect of the important problem of wear, from the point of view of fuels, manufacture and lubrication. As ordinary anti-wear oils proved unsuitable when using residual fuels with low sulphur content (such as those of Russian, North African, and Argentine origin), the development and use of an anti-wear cylinder oil suitable for both high sulphur and low sulphur fuel oils are also described. The second part of the paper deals with wear in four-stroke medium speed diesel engines, referring particularly to service data of engines operating on residual fuels. Particular attention is paid to the behaviour of exhaust valves, to cylinder liners and to lubrication.—*Cotti, Dr. E. and Simonetti, Dr. G., Proc. IMAS 69, Section 46, pp. 15–33.*