Design Methods and Development of Medium and High-speed Oil Engines

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The Diesel engine activity of the Ruston Paxman Group has been concentrated in two sources of design, development and manufacture, located at Lincoln and Colchester.

Ruston and Hornsby, Ltd., in Lincoln, produce Diesel engines ranging in power from four to 8000 hp and Davey, Paxman and Co. Ltd., in Colchester, produce Diesel engines ranging in power from 100 to 3000 hp.

This paper covers two ranges of engine from the group's activities, but, in introducing these, it is necessary to describe a little of the background and philosophy which led up to the introduction of these two engine ranges.

Ruston and Hornsby, which was formed in 1918 as a result of the acquisition by Ruston Proctor and Co., of Lincoln, of Richard Hornsby and Sons, of Grantham, have been manufacturers of heavy, slow-speed, solid injection oil engines since Richard Hornsby and Sons acquired the world manufacturing rights of the Herbert Akroyd Stuart oil-engine patents in 1891. Through the years the traditional market has developed from small lighting sets through to large municipal power stations and, with the growth of the maritime fleet, all forms of commercial marine application where medium to slow-speed operation has been required.

Davey, Paxman and Co. Ltd., who have been manufacturing Diesel engines since 1925, became a member of the Ruston Group in 1939. Engine development has been directed towards the production of high-speed, small-bulk engines having an ever increasing specific output aimed at the world's naval requirements and at the rapidly developing commercial market for this type of engine in the industrial rail traction and marine fields.

With customer acceptance of higher specific output engines, Ruston and Hornsby embarked upon the design and development of the AT range of engines and Davey Paxman upon the Ventura range. Initially both ranges covered roughly the same horsepower bracket, but were totally different in speed, weight, bulk, etc. The design of the AT range was started in 1956 and that of the Ventura range in 1957. The paper deals with specific design features of the two engine ranges.

PART I—THE HIGH-SPEED ENGINE

by A. G. Howe, M.B.E.

The Paxman Ventura engine is the latest member of the family of vee-engines first developed in 1935 and follows the 7-in bore range known as the RPH and YH engines which were developed from the TP engine so widely used in tank landing craft during the Second World War.

The development of a new engine taxes the skill of the professional engineer. This was particularly so in the case of the Ventura, where certain parameters were established. Briefly, these were that the overall dimensions and weight of its predecessors were not to be exceeded and that the output should be increased by approximately 50 per cent. This seemed like trying to get 1½ pints out of a pint pot and it was this challenge which made the whole development so exciting.

Being a member of a family, certain hereditary features were bound to prevail. Full advantage was taken of past experience and availability of production facilities so that, broadly speaking, the Ventura looks much like its predecessors in that it has a normal-type crankshaft, fork and blade rods and conventional piston.

Generally speaking the design modifications which have made this major change possible fall under four headings:

1) a welded steel engine housing (crankcase and cylinder blocks combined);
2) the enlargement of the bore and stroke to give a 23 per cent increase in cylinder capacity within the same overall length;
3) the careful studied use of high grade materials with the appropriate attention to surface finish and hardness;
4) the use of high-pressure turbocharging and intercooling, which could be fully exploited by designing for higher cylinder pressures.

The Ventura range has a bore of 7¾ in and stroke of 8½ in and comprises six, eight, twelve and sixteen-cylinder engines

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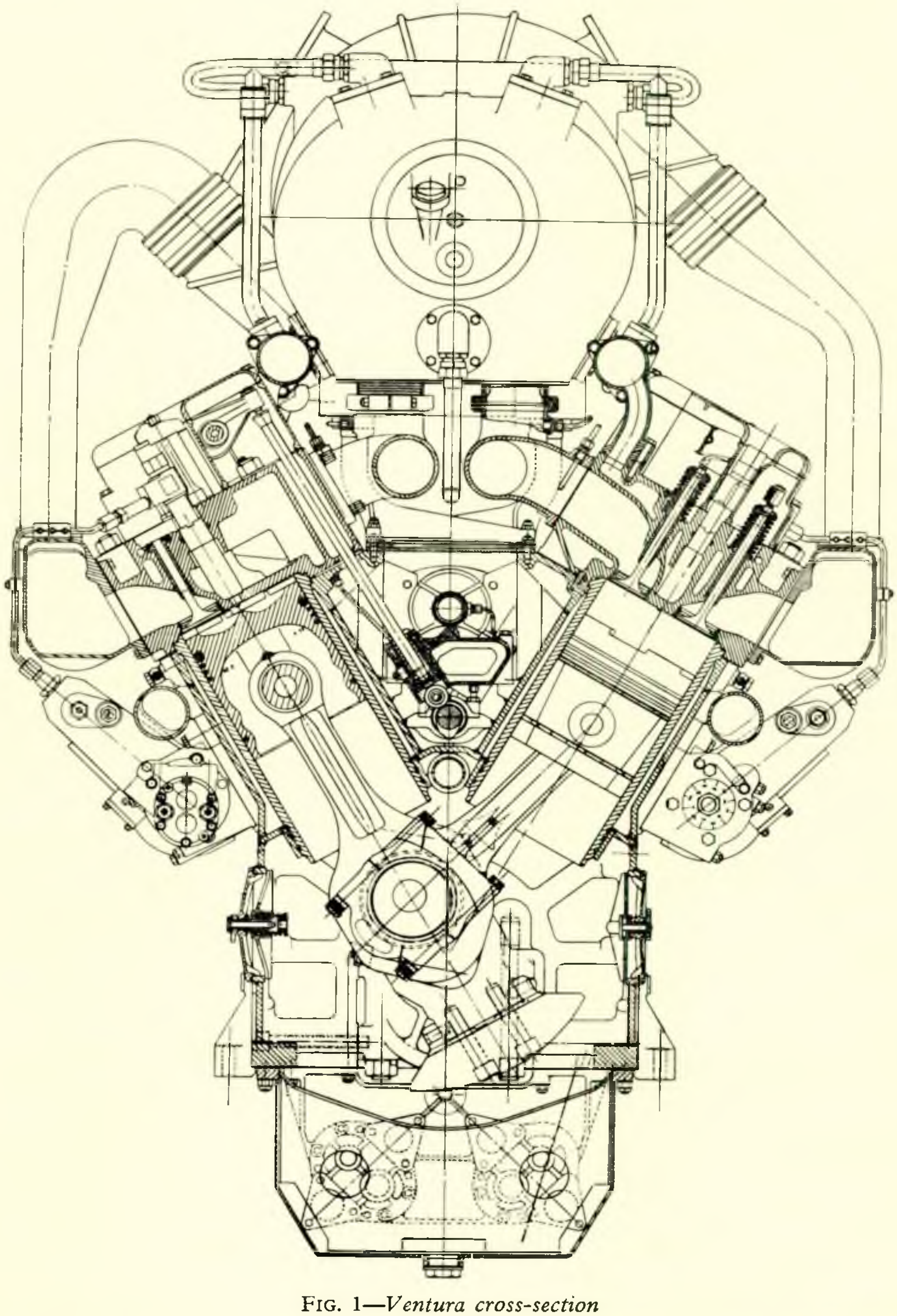


FIG. 1—*Ventura cross-section*

with ratings of 500/2400 bhp at 1500 rev/min. They have been designed for continuous operation at 1500 rev/min and some are already in use for base-load power generation at this speed. A 20 per cent overload can be taken for short periods, such as in high-speed marine craft applications.

The b.m.e.p. increases from 160 lb/in^2 at the continuous rating to 200 lb/in^2 b.m.e.p. at the marine pursuit rating. The engine weight based on the full load rating is 6·3 lb/bhp. Fig. 1 shows the cross-section arrangement of the 16-cylinder turbocharged and intercooled engine with a maximum rating of 2400 bhp at 1500 rev/min.

DESIGN FEATURES

Before discussing in detail some of the design aspects of this engine, it is worth mentioning the need for accessibility to assist maintenance, as this can control other features in the design. When designing a compact high-output engine which will be used in a wide variety of applications, accessibility for maintenance must be considered at all stages of the design. To achieve this the author's company tried to keep the design simple although compact. The cylinder heads can be removed individually without removing the air or exhaust manifolds, and quite conventional camshaft followers and rocker gear are used to operate the four valves. Those parts requiring attention, such as injectors, fuel pumps, filters and control gear are kept on the outside of the engine, whereas the exhaust system is within the vee, which gives the shortest possible exhaust system to the turbocharger and keeps the hot exhaust well out of the way.

The big-end bearings are of the thin-wall, lead-bronze type and these can be dismantled for examination through the crankcase doors. The main bearings, also lead bronze, and the crankshaft, although underslung, can be inspected by using special tools and jacks designed specifically for this purpose. The complete gear train is situated at the drive end of the engine and, although this may not be the best location from the point of view of maintenance when using a close-coupled generator, the drive is smoothest at this point and the gears should require no maintenance between major overhaul periods.

The question of oil leaks is also connected with maintenance. This can be a problem on a high-speed engine and during the design of the Ventura close attention was paid to this aspect. Most of the flange connexions, such as on the water rails and oil supply pipes, are made with 0-ring seals, the flange faces being metal-to-metal, and these have proved quite successful. Wherever possible, right-angle joints, i.e. having to make a joint on two faces at the same time, have been avoided. Some of the more difficult joints to keep tight over a long period have been the various ferrule connexions between cylinder head and cylinder block, particularly where these transfer oil under pressure, such as the oil supply to the rocker gear. One reason for this difficulty is that owing to the accumulation of drawing tolerances the gap between the cylinder head and block can vary as much as 0·030 in. A successful design was finally achieved by using synthetic rubber bushes approximately ½ in long, recessed into both head and crankcase. These have the advantage of taking up considerable variations in length and seal on the side faces as well as the end faces.

CRANKCASE

Welded crankcases have been a feature of certain Paxman engines since 1930, so that quite a lot of knowledge and experience contributed towards the successful behaviour of the Ventura crankcase (see Fig. 2). The Ventura crankcase is quite

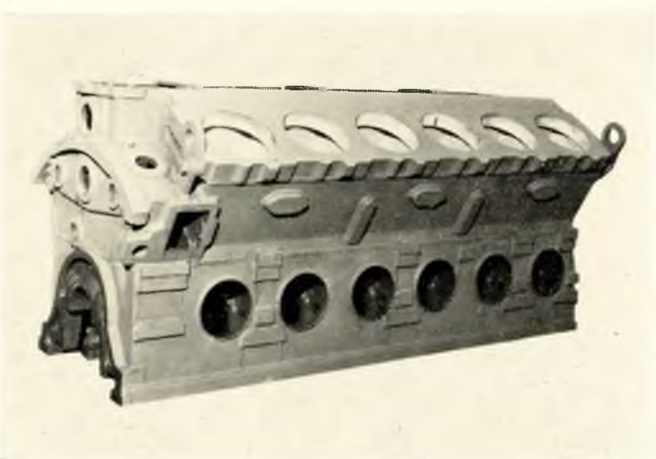


FIG. 2—*Crankcase fabrication for 12-cylinder engine*

a small fabrication, made from a combination of relatively thin plate and steel castings, and the size and section of this crankcase has influenced the design. In a large crankcase, such as described for medium-speed engines, several welders can operate at the same time and relatively little distortion will occur. However, with a small crankcase this is not possible as firstly, it is not feasible to have several welders operating together, secondly, severe distortion would result and, thirdly, the welds on a small fabrication, if made as one assembly, would not be so accessible. Hence the Ventura crankcase is designed to be made in three sections consisting of two individual cylinder blocks and the lower half crankcase. These can be made as quite independent assemblies and annealed before being brought together and finally welded to form the complete crankcase.

The whole assembly is stress relieved again for a period of two hours at 600°-650°C (1112°-1202°F) brought up to temperature by automatic control.

The assembly fixture should allow the crankcase to be moved in any position to permit down hand welding and it is the practice of the author's company to test all the plates before assembly ultrasonically, and crack detect the completed crankcase using one of the dye penetrant methods. Any weld subject to fatigue should be a fully prepared butt weld and the stress should not be above two ton/in². Strain gauging techniques, both on a perspex model and on the actual crankcase were adopted to prove the crankcase under both static and dynamically loaded conditions, and a careful study of the disposition of stresses and material in the manufacture of these components has probably contributed more to the weight saving in, and overall dimensions of the engine, than anything else.

CRANKSHAFT

Any engine designed for continuous marine duty must meet the rules of the classification societies and those rules relating to the crankshaft can well control the overall engine size. It is, therefore, important in designing a compact engine to make maximum use of good-grade material and surface hardening techniques to give minimum wear rates under continuous maximum-speed conditions. The surface finish of the crankshaft is important and the pins and journals are held to eight microinches (see Fig. 3). The detail design of certain features of the

FIG. 3—*Crankshaft*

shaft requires careful attention, for instance, in the prototype shaft the oil holes in the crank pin were drilled at an angle to the pin surface to link up the journal and pin with a single drilling, this had been used before in soft shafts quite successfully. However, with an induction-hardened shaft, it was found that in some cases the hardening had entered the oil hole and caused a hardening crack under the edge of the hole. This propagated and eventually broke the shaft. Fig 4 shows how this crack propagated.

Production shafts now have the oil hole at right angles to the pin, even though this has complicated the drilling. This failure was investgated very thoroughly and the National Engineering Laboratory carried out some fatigue tests for the author's company on small-scale samples of unhardened material with different angled oil holes. These tests showed that changing the oil hole angle from 30° to the pin surface to 90° to the pin surface gave an increase in torsional fatigue strength of about 60 per cent.

To take care of torsional vibration a viscous damper is

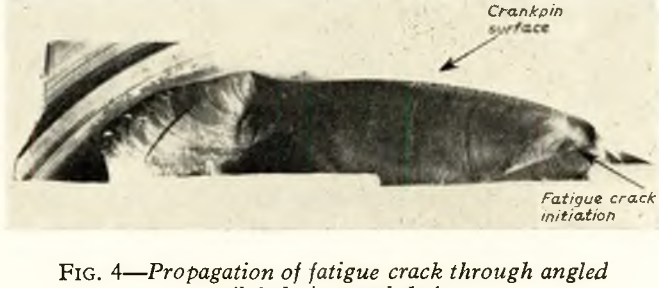


FIG. 4—*Propagation of fatigue crack through angled oil hole in crankshaft*

fitted to the free end of the crankshaft and enclosed within the free-end cover of the engine, and to ensure correct overall balance the completed shaft is dynamically balanced to within four oz in.

CYLINDER HEAD AND PISTON

The satisfactory operation of an engine of this type is closely allied to the thermal conditions in the cylinder head and piston. During the development stage these are studied closely to ensure they are within acceptable limits. Tests have shown that for loads up to 200 lb/in² b.m.e.p., a single-piece

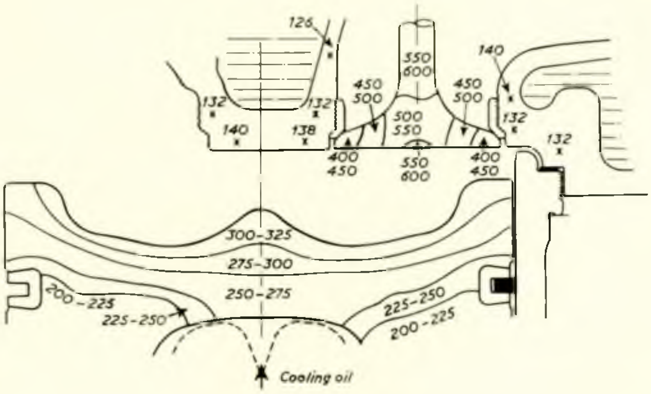


FIG. 5—*Piston, valve and cylinder-head temperature distribution*

cast aluminium piston is satisfactory. Oil cooling is used to keep the piston and ring temperatures within a satisfactory range. This cooling takes the form of an oil jet directed from the top of the rod on to the underside of the piston crown. The oil is supplied to the rod from drillings and grooves in the large end assembly and is arranged to feed oil to the piston from 70° before top centre to 70° after top centre. Due to deceleration, as the piston approaches the top of the stroke a pressure of 160 lb/in² is generated in the column of oil in the rod.

The effect of this oil jet on the heat flow in the piston is shown by the temperature contours in Fig. 5.

For instance, the horizontal nature of the 275°—300°C (527°–572°F) temperature contour shows the effectiveness of the oil jet in removing heat from the piston crown. The same diagram shows the metal temperature of the aluminium cylinder head.

Minimizing ring-groove wear is an important factor in good piston-ring performance, and all the company's high-speed engines have an Alfin-bonded austenitic iron insert surrounding the top ring groove.

As loads increase with the adoption of higher firing pressures with more severe thermal conditions in the cylinder, the conventional piston will no longer be able to cope with these conditions and the adoption of "stepped" connecting-rod small-ends, to allow more area for the piston bosses and cast in cooling passages whether by tube or soluble cores, will be used more widely.

CONNECTING-RODS

The construction of the blade and fork rod will be seen from Fig. 6. This design is basically a continuation of Paxman vee-engine practice and several important features contribute to the success of this arrangement. Firstly, the design is such that the bolts only carry tension loads. The rods and block half are located by serrations and dowels are used to locate the block to the palm face of the rod. In this way fully-waisted bolts can be used, overcoming the possibility of fatigue failures from fretting which may occur on a highly loaded connecting-rod bolt when used to locate the rod halves. The abutment step at the fork-rod joint-face on which the blade-rod bearing oscillates must be held within close limits. This is checked during assembly by means of a clock gauge between a pair of rollers which are run across the joint line and the reading must be within 0·0002 in. Finally, the rods are machined all over and polished by tumbling in a rotating drum filled with abrasive. The blade rod is split at an angle of 40° to enable it to be passed up through the bore.

During the initial design and early prototype stages the proposed rod design was investigated by both rubber and photo-elastic models. Fig. 7 shows two alternative types tested. As soon as the actual prototype rod became available this also was subjected to tests to determine the stress level by means of strain gauges and the bearing bores are checked for distortion (see Fig 8).

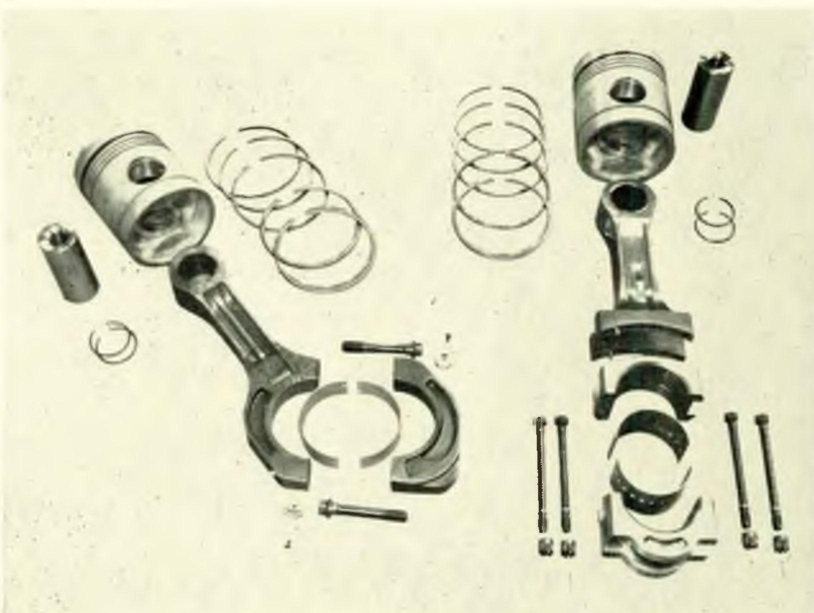


FIG. 6—*Blade and fork-rod construction*

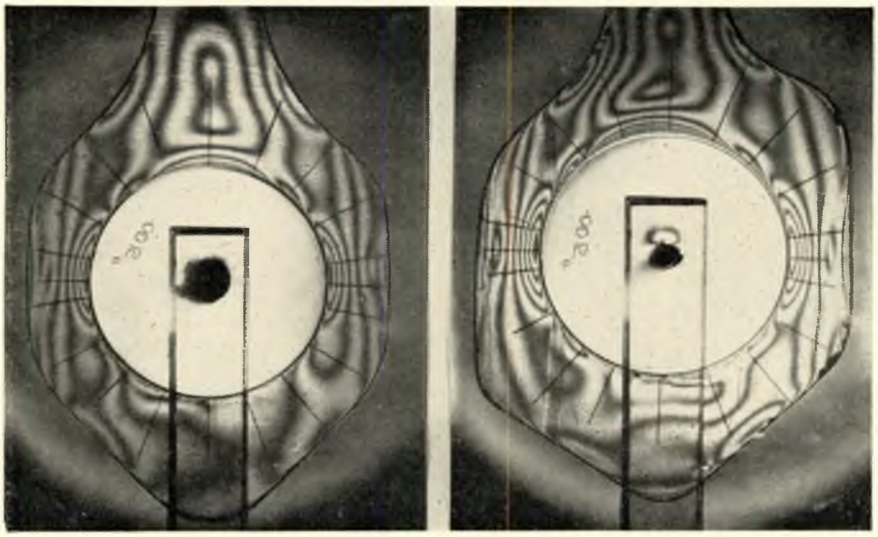


FIG. 7—*Photo-elastic tests on alternative connecting-rod profiles*

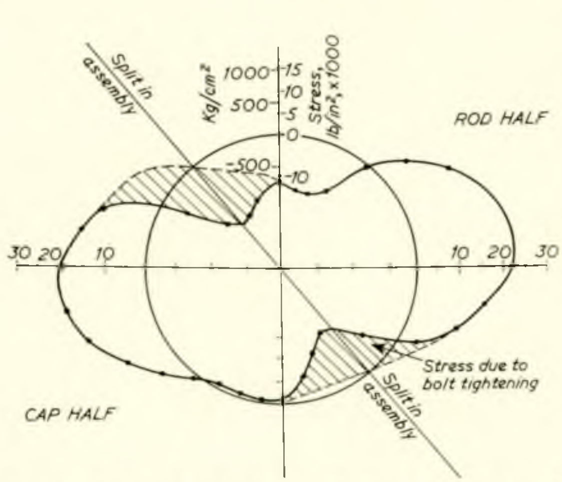


FIG. 8—*Stress distribution in blade rod at overspeed*

COOLING SYSTEM

The careful design put into an engine to reduce liner wear and increase overhaul periods will be wasted if the cooling system does not operate correctly. Experience has shown that excessive liner wear can be traced to low operating temperatures. The water flow through the Ventura is 15 gal/bhp-h, giving a temperature rise across the engine of about 10 deg F at full load. The oil coolers mounted on the engine are cooled by the engine water, ensuring a rapid warm-up and enabling the whole cooling system to be controlled by one thermostat. This thermostat is considered to be part of the base engine and is always supplied, even if other means such as variable-speed radiator fans are fitted for temperature control. This ensures that the engine operates under the design conditions irrespective of the remainder of the installation which is not always under the control of the engine builder. In a compact engine using high water flow rates it is important that coolant passages are not too small. Although better cooling will be obtained with high velocities, the small coolant passages can in practice lead to trouble. For instance an engine prone to cavitation erosion on the cylinder liners will often show an attack on the crankcase if the cooling passage is too small. Water treatment is also important, and in the author's opinion the selected treatment should be one which is not too sensitive to the quantity used. Some treatments such as chromate can increase the rate of attack if the solution is too weak. It has been found in practice that soluble oil is quite an effective treatment. It is not too sensitive to quantity used and gives protection to both steel and aluminium. One precaution, however, is that all rubber in the system must be oil resistant, otherwise it will be attacked by the soluble oil.

ENGINE PERFORMANCE

The performance of the Ventura engine has been developed to suit a wide variety of duties, these include both generating and marine propulsion applications. In some respects these requirements differ. For a generating engine good constant

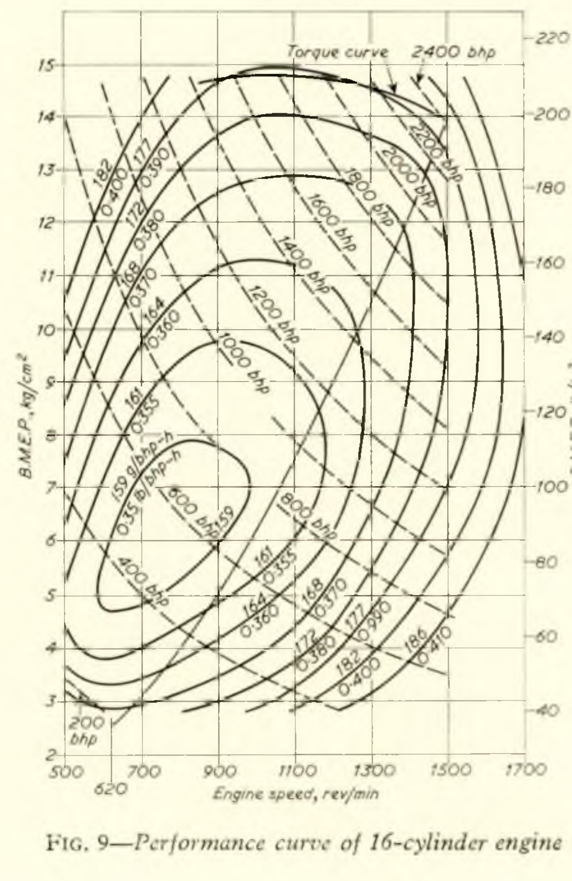


FIG. 9—*Performance curve of 16-cylinder engine*

speed performance is required with a flat specific fuel consumption curve covering as wide a part of the load range as possible. Speeds other than full speed, such as idling, are not of particular importance in this duty. However, with a propulsion unit a wide speed range is required and to ensure that the top speed is not excessive the idling speed must be as low as possible, otherwise the minimum speed of the ship will be affected and create difficulty in handling. To this end, quite a lot of development has been directed towards ensuring a consistent low idling speed and part-load operation down to 450 rev/min and a good all round performance up the propeller law curve to the maximum sprint rating of 200 lb/in^2 b.m.e.p. at 1500 rev/min. In some cases a compromise is required as conditions for 200 lb/in^2 b.m.e.p. at full speed can conflict with those required for smooth idling at low speeds, for instance to overcome secondary injection at high fuel input a large unloading volume in the fuel pump delivery valve may be required, but this is not conducive to good idling.

The performance curve of a 16-cylinder Ventura engine is shown in Fig. 9.

ENGINE APPLICATIONS

Ventura engines have been in production on an increasing scale since 1960 and to date about 300 engines have been supplied, or are on order. One base load application involving engines running at 1500 rev/min has accumulated a sum total of 140 000 hours running.

The first application was in a passenger express locomotive operating on the Western Region, some six years ago. Two engines were fitted to one locomotive and these are still in operation. As a result of this further engines of a similar type were supplied to the Scottish Region of British Railways.

The Ministry of Defence has adopted the Ventura as its Standard Range II for the Royal Navy as a successor to the Paxman YH 7 in bore. The generator is flange mounted to the engine and the whole unit carried on flexible mountings positioned under the centre of gravity of the engine and centre of gravity of the generator.

Various types of generator set are in service with Ventura engines and provide a particularly compact high-output set for standby applications. A 12-cylinder engine generator set will produce 1000 kW at 1500 rev/min within a length of 9 ft 6 in and width of 4 ft 6 in.

In the marine field, the Ventura increases in popularity for propulsion duties as well as for marine auxiliary generator sets, particularly in relation to patrol craft, typical of these being those supplied to Vospers for vessels for Trinidad, Kenya and Libya. Each craft has two 12-cylinder engines rated at 1500 hp at 1500 rev/min driving through Z.F. gear-boxes.

The company has also supplied 12-cylinder Ventura engines as main propulsion units for survey vessels using Diesel-electric drive. Among the interesting applications has been that for the Canadian hydrofoil, at present under development.

The general purpose of the foregoing list of applications is to show that the engine is versatile and this is quite intentional as each application benefits from the other, and one is able to build a progressive engine from experience gained in various fields, to the advantage of all.

PART II—MEDIUM-SPEED ENGINE

by H. Watson, B.Sc., Ph.D.

INTRODUCTION

While engine development at Davey Paxman and Company has been directed towards the production of high-speed, small-bulk engines with ever-increasing specific output, the traditional market of Ruston and Hornsby Ltd. has been in the medium-speed engine field and the product has had a long established and treasured reputation for reliability and durability. Any uprating or new design has therefore been considered within this context and from the outset it has been a prerequisite that, even with high specific output, this feature of reliability and durability must be maintained.

It is of interest to consider the rate of development of the medium-speed oil engine over the past 45 years.

Fig. 10 shows values of piston speed and brake mean effective pressure, and the product of these two values taken from company catalogues covering medium-speed engines over the period under review. The values shown are general to the Ruston medium-speed engine range and are in general typical of the industry as a whole for this type of engine.

Fig. 10 shows the unit cost of horsepower available from a 12 in bore engine in absolute values over the period.

The rising cost curves immediately show the main incentive for uprating. In terms of absolute purchasing value, the cost of power has decreased by a factor of 20 per cent between 1932 and 1966, which is a measure of the effectiveness of development.

When the rate of increase of specific output is considered, it is apparent that, during the period from 1920-1940, rating increased comparatively leisurely, with succeeding designs only slightly modified from their predecessors. Under these conditions, new designs could, in fact, be sold straight from the drawing board with a secure knowledge that reliable operation would be achieved without any requirement for development.

Moderate pressure charging was introduced in the late 1930s, but for the next fifteen years specific output remained static.

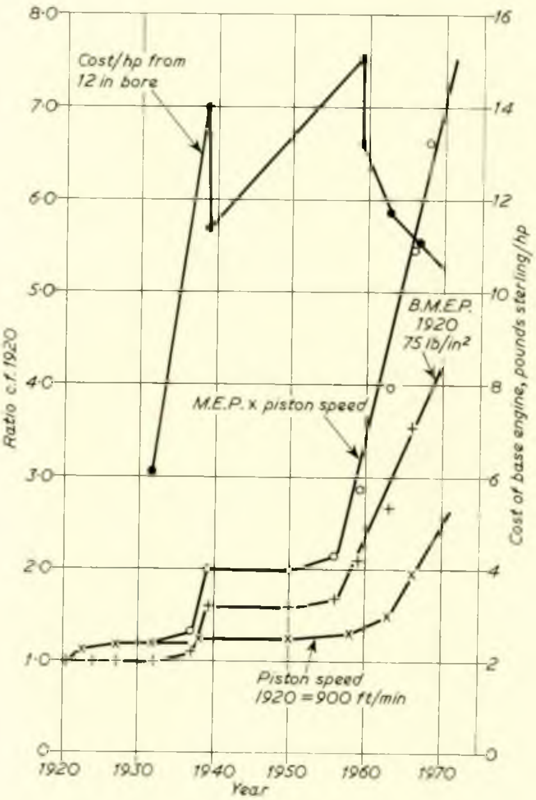


FIG. 10—*Power increase in medium-speed engines and comparative cost/unit power*

Since the mid 1950s power output increases have been achieved at an increasing rate and, Fig. 10 shows, have more than trebled over the last ten years.

This rate of increase, which has been achieved without any sacrifice in reliability and durability, has only been made possible by the introduction of improved design techniques which have been dependent for their success on the establishment of advanced specialist design teams and by introducing development testing as a logical step in the design process. Facilities for development testing to be undertaken on the necessary scale have been established progressively by Ruston and Hornsby and, in 1957, on a site separated from the normal production facilities, a special development centre was completed and commissioned.

THE RUSTON AT ENGINE

The following outlines some of the techniques which have been developed and uses a specific engine to illustrate what has been achieved by the use of these techniques.

The engine, designated AT, has a bore and stroke of 12½ in and 14½ in respectively and is manufactured in in-line form with six, eight and nine cylinders, and in vee-form with 12, 16 and 18-cylinder configurations.

A longitudinal section of the in-line engine is shown in Fig. 11 and a transverse section of the vee-engine in Fig. 12. A complete engine is shown in Fig. 13. Cylinder-centre distance is common to both vee and in-line engines.

The main structure of the in-line engine is machined from an iron casting, designed in the form of a deep U-section bedplate which supports the crankshaft. A cast-iron cylinder block bolts directly to the bedplate. The vee-engine structure is fabricated, the crankshaft being supported in the underslung position, and the cylinder block for each bank is carried on the top plate of the frame.

Both in-line and vee-form designs incorporate a single-piece, continuous grain flow, forged crankshaft.

Stamped steel connecting-rods are employed, the in-line version utilizing an automotive-type rod with a diagonal split to facilitate removal through the cylinder bore. The vee-engine incorporates an articulated rod arrangement, with a vertical

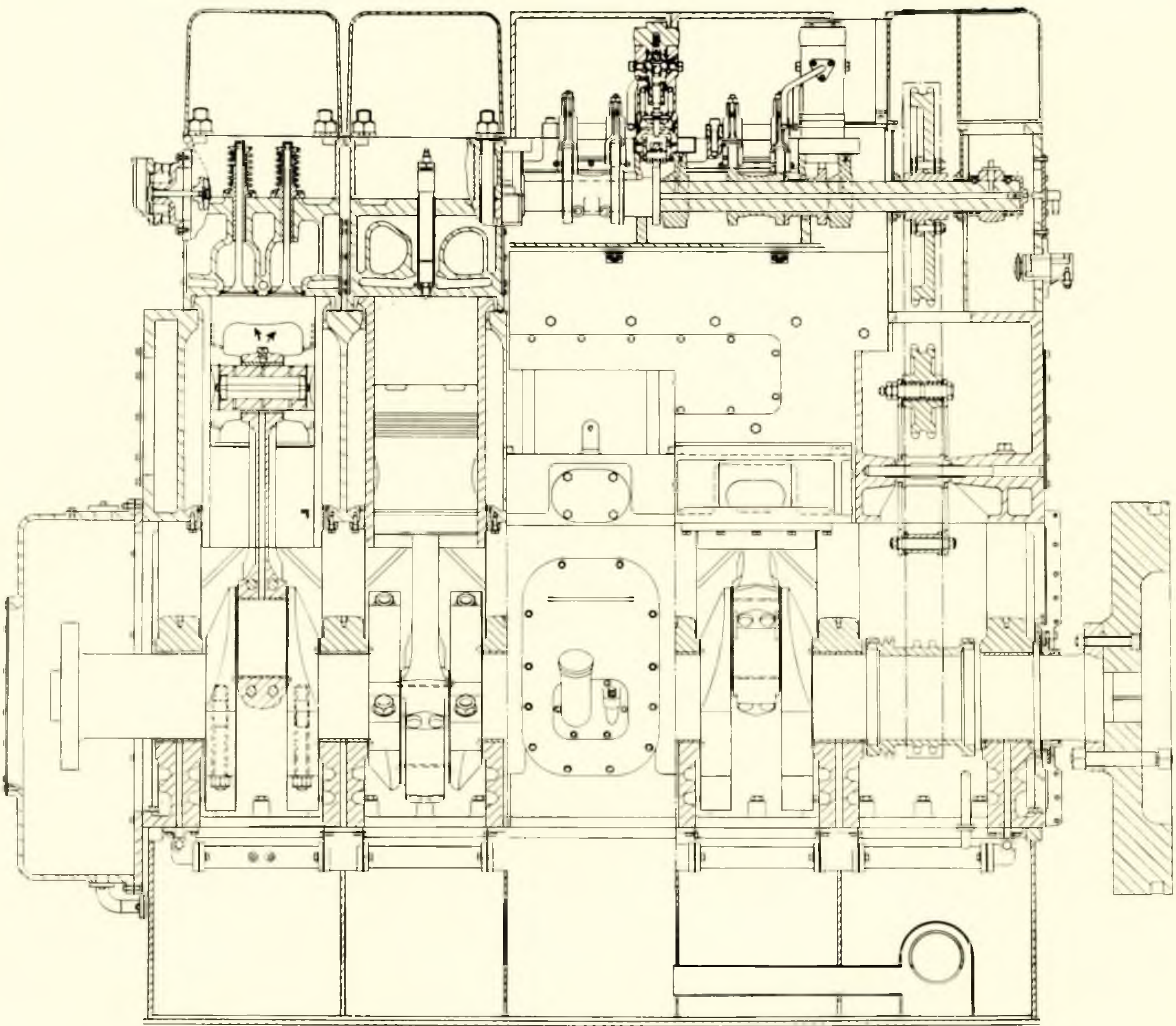


FIG. 11—*Longitudinal section of AT engine*

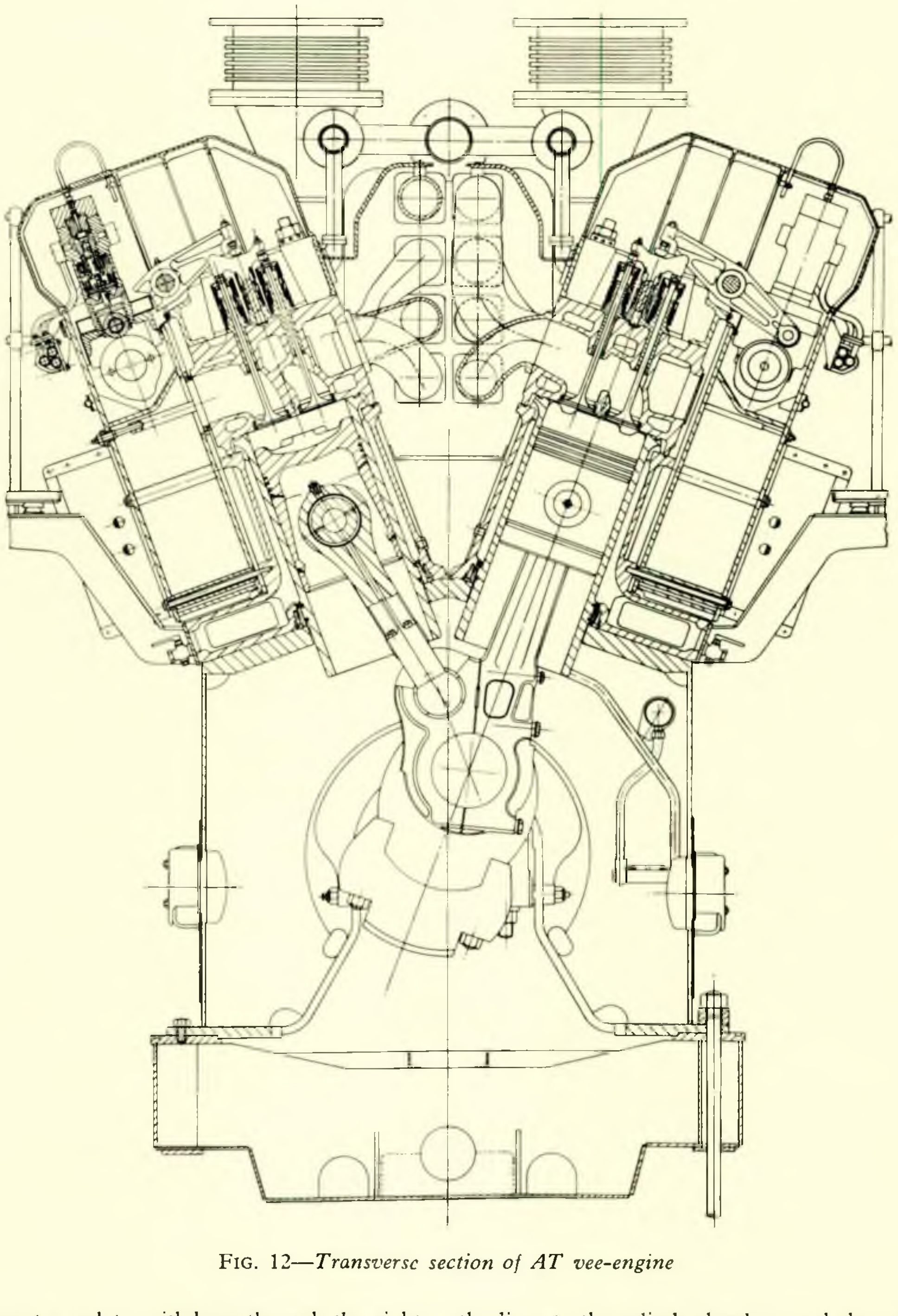


FIG. 12—*Transverse section of AT vee-engine*

split enabling the master rod to withdraw through the right-hand cylinder and the slave rod, together with the master-rod cap, to be withdrawn through the left-hand cylinder.

Aluminium tin thin-wall, steel-backed bearing shells are utilized for main and crank-pin bearings, and lead-bronze bearings in the small-end and articulated bearing on the vee-engine slave rod.

Cast-iron pistons are articulated to the connecting-rods, via fully floating gudgeon pins, and under-crown cooling is provided by oil jets fed under pressure via drillings in the respective connecting-rods.

Cast-iron liners, chromium plated on the running surface, are housed in the main cylinder block. The design of liner, cylinder head and cylinder block allows water transfer from the liner to the cylinder head around the whole circumference and allows the liners to be cooled effectively throughout their length and around the upper flange.

Cast-iron cylinder heads incorporating a double-deck construction achieve effective flame-plate cooling and are attached to the engine by four studs. The strain energy stored in the studs and the depth of cylinder head assembly ensures that adequate sealing force between cylinder head and liner is achieved and maintained in operation.

Each cylinder head accommodates two inlet and two exhaust valves actuated through rocking levers and bridges directly from a high-level camshaft housed in a fabricated structure which also forms the engine air manifold.

Individual fuel pumps are located in the cam-box structure

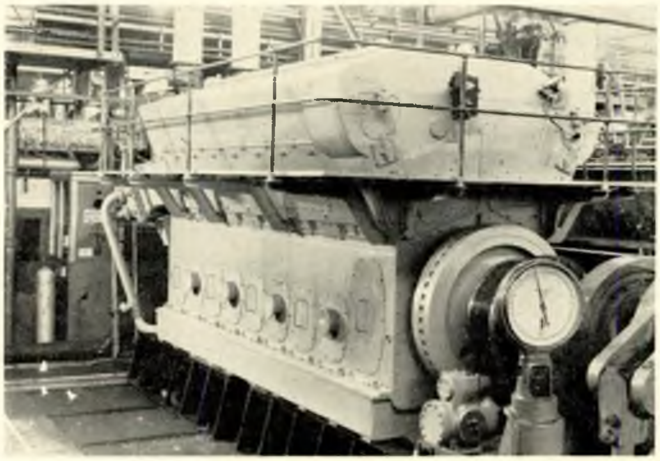


FIG. 13—*Complete 16-cylinder AT engine*

and are actuated through integral tappets from the camshaft. Fuel is delivered directly to the combustion chambers from a "jerk" type pump through low inertia nozzles.

On all engines, exhaust-driven turbochargers and charge-air coolers are mounted at the free end.

Design work commenced in 1956 and, after extensive development work undertaken on prototype units, the engine was first released for service in 1959, operating at a continuous output of 160 lb/in^2 b.m.e.p, at 500 rev/min. In 1962 the continuous b.m.e.p. was increased to 200 lb/in^2 and in 1965 the speed was increased to 600 rev/min, together with an increase in b.m.e.p. to 205 lb/in^2 on six and nine cylinder combinations.

Indications are that the basic form of engine will eventually achieve a continuous b.m.e.p. of 300 lb/in^2 at speeds up to 750 rev/min. The four successive ratings correspond to 180, 225, 278 and 500 hp per cylinder respectively. At the present rating, the largest unit in the range, namely the 18-cylinder engine, develops 5000 hp at 600 rev/min.

PRELIMINARY CONSIDERATIONS

The majority of components in a reciprocating internal combustion engine have to perform a complex duty and design of the main component groups is, of necessity, a compromise between several conflicting requirements. Basic data available to the engineer have increased rapidly during recent years and the availability of modern computers allows design to become more exact.

The process of design however remains still one of successive approximation with development operation an essential step and, in the following arguments, it is impossible to differentiate between design and development. The whole process does become extremely involved and, due to the time scale required for the production of any new medium-speed engine, modern planning techniques have to be applied to achieve a compact overall programme and to maintain punctuality.

Fig. 14 shows the initial steps which are necessary in the design process and gives some indication of the complexity introduced by present rates of development. The time cycle has a critical path time, depending on the magnitude of the rating step, of from four to seven years from the decision to proceed and the achievement of an engine capable of satisfactory service operation.

SPECIFIC DESIGN METHODS

As a very broad generalization, three main types of design problem exist, namely:

1) performance prediction;
2) problems associated with purely mechanical loading, usually dynamic in character;
3) problems where a component is subject to a combination of mechanical and thermal loading.

In each range of problems, the accuracy of prediction increases progressively as experience is built up and new approaches are evolved. Each problem is itself dependent on the existence of a preliminary design which can be analysed with the techniques available, followed by a more accurate design which it is then necessary to test by development operation to give a measure of the accuracy of the design, thereby feeding back empirical data to allow successive design assessments to become more accurate.

The first category of design problems is dependent on empirical data from engine operation.

In the main the second set of problems can be solved by calculation and the use of static rigs, although where loads are applied dynamically the engine is the only true rig.

Problems in the third category are the most difficult and at present rely on complex computations and require rig and engine testing in the overall design process.

Full justification for the overall design can only be obtained after considerable periods of operation under actual service conditions and, when it is considered that engines of the AT type are expected to achieve up to 175 000 hours operation during normal working life, the requirements of any design programme assume considerable magnitude.

PERFORMANCE PREDICTION

History records that Dr. Diesel, in the last decade of the 19th Century, arrived at his engine process purely by an analytical approach to cycle calculations. From this time until the 1950s the analytical approach to performance prediction has tended to be overlooked in preference to the experimental approach.

The availability of digital computers in the company coincided with the commencement of design work on the AT range of engines. In the initial design stages it was considered worth while to study a number of independent variables and to attempt to estimate the individual and collective effects of these upon fuel consumption and thermal loadings in order to establish design parameters.

The variables considered were:

a) turbocharger pressure ratio;
b) intercooler outlet temperature;
c) effective engine compression ratio;
d) maximum cylinder pressure;
e) trapped air/fuel ratio;
f) engine expansion and compression ratio;
g) scavenging air losses;
h) fuel injection rates.

A computer programme was written based on straightforward thermodynamic relationships and including simplifying assumptions, such as heat release rates and heat loss in the cylinder. This programme was run with different combinations of individual parameters.

The results allowed performance trends to be predicted and showed that for an engine rating of 200 lb/in^2 b.m.e.p. and 500 rev/min a realistic design with satisfactory performance could be obtained by setting the following parameters:

i) maximum cylinder pressure 1500 lb/in^2;
ii) engine nominal compression ratio 13 : 1;
iii) intercooler outlet temperature 40°C (104°F);
iv) boost pressure ratio 2 : 1.

Further calculations at ratings of 300 lb/in^2 b.m.e.p. indicated that maximum cylinder pressures would rise to the order of 2000 lb/in^2. Fig 15 gives an example of results obtained for the 300 lb/in^2 b.m.e.p. condition.

The results of such a broad assessment do not predict performance accurately, but in themselves they do predict the qualitative effect of any design variable.

In the case of the AT engine, development running of an initial build was then undertaken to allow a datum performance level to be measured and with the predetermined trends predicted by calculation, individual parameters were varied to achieve the required performance.

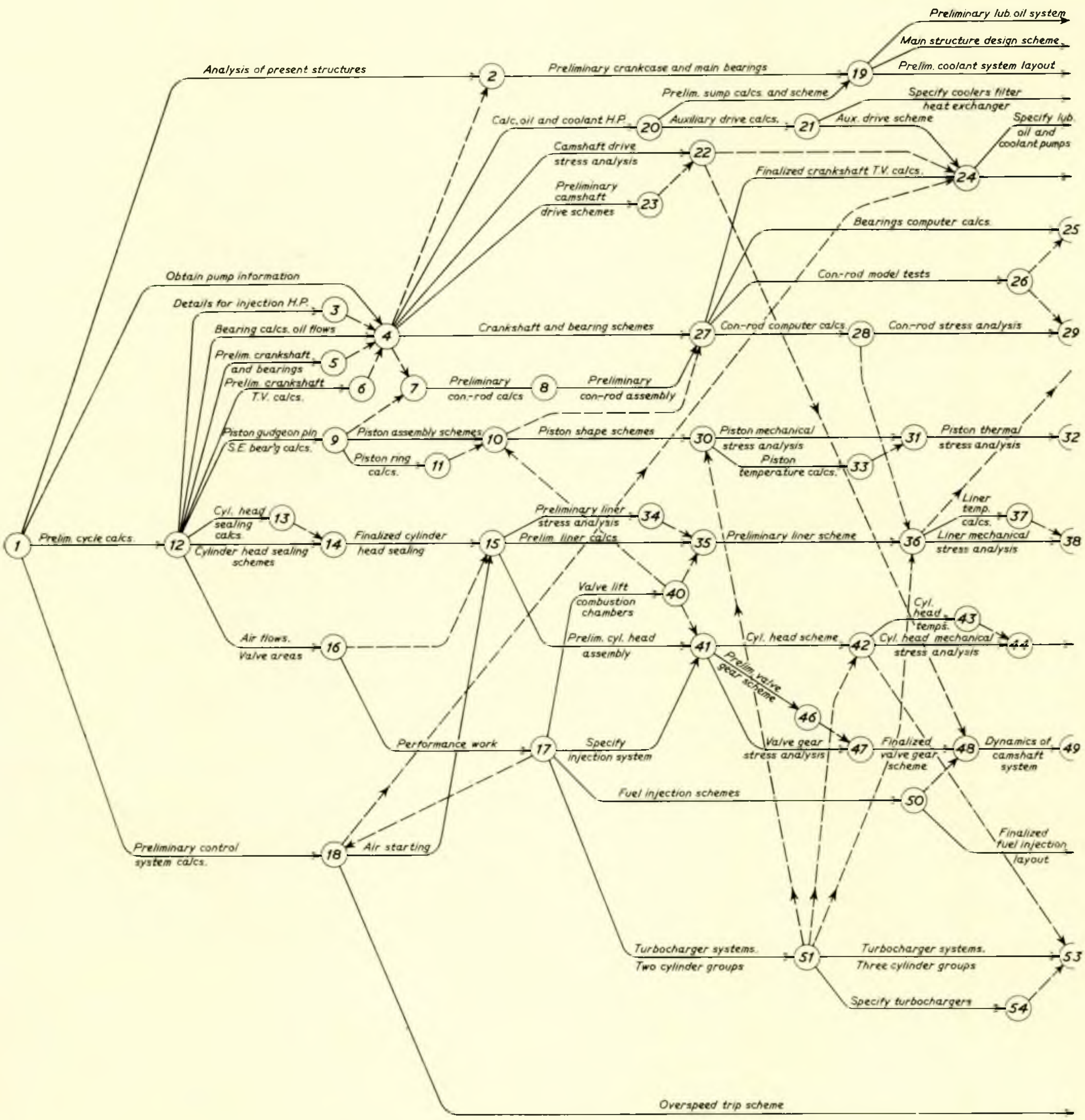


FIG. 14—*Initial steps in engine design programme*

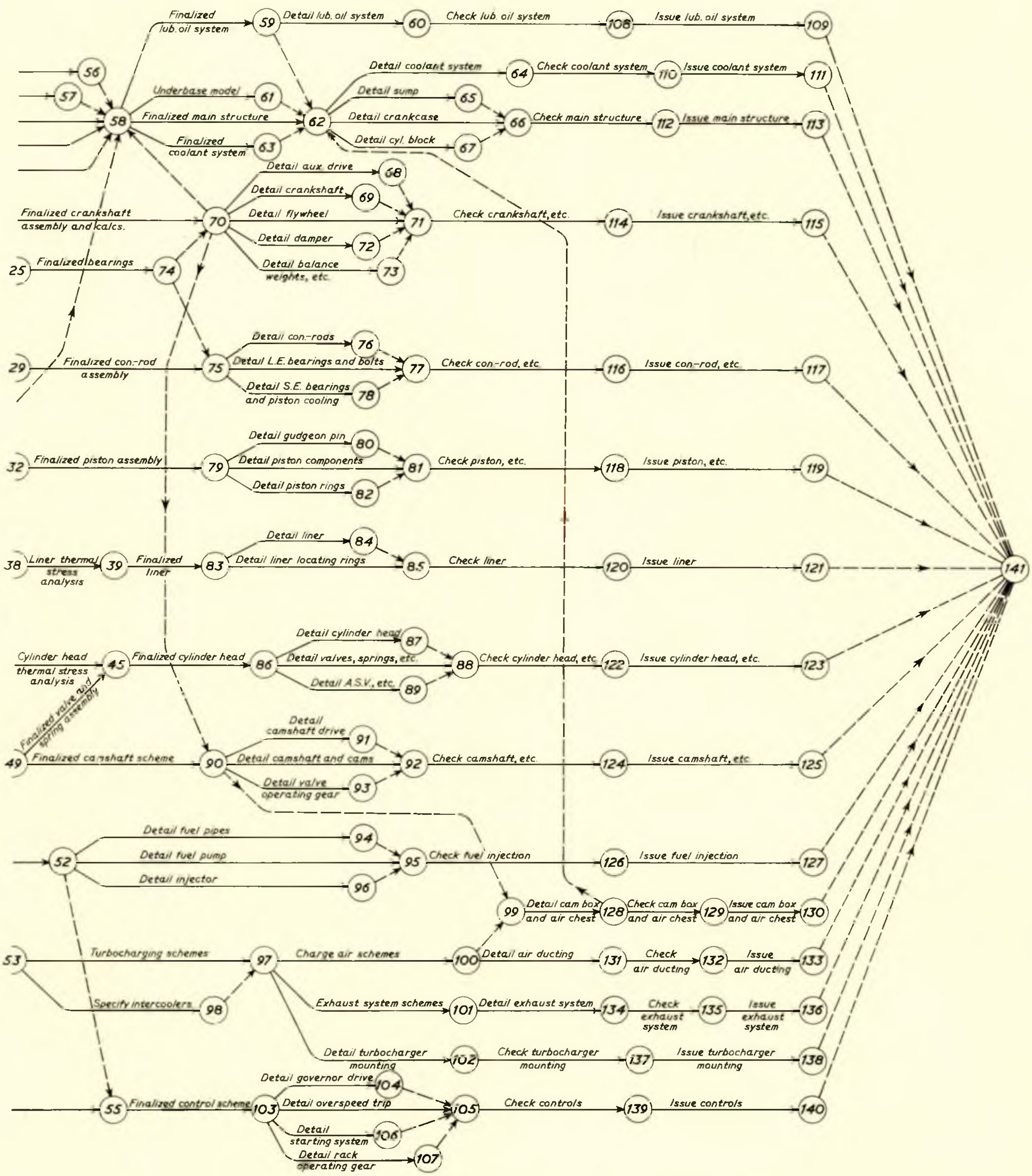


FIG. 14—*Initial steps in engine design programme*

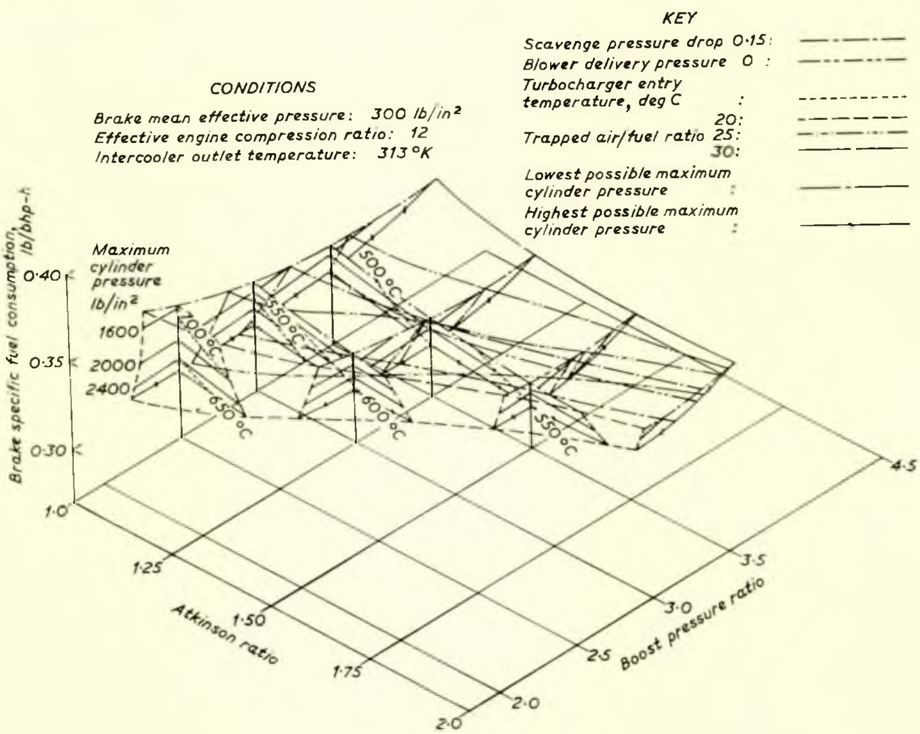


FIG. 15—*Typical performance predictions for AT engines at 300 lb/in² b.m.e.p.*

Actual engine tests were also fed back to the theoretical programme, which, in turn, allowed more accurate performance predictions to be made.

Further developments in the computer programme have since taken place in which events in the cylinder are more accurately predicted and the full cycle is calculated, including energy exchange processes in exhaust pipes, turbine compressor and inlet manifold. It is now possible to predict actual values rather than trends. Table I shows a comparison of measured and computed values for a particular build of engine.

TABLE I.—COMPARISON OF PREDICTED AND MEASURED RESULTS FROM THE AT ENGINE AT 200 LB/IN² B.M.E.P.

	Actual engine results	Theoretical match
Rev/min	500	500
Indicated mean effective pressure, lb/in²	—	222·19
Brake mean effective pressure, lb/in²	200	199·79
Indicated specific fuel consumption, lb/bhp-h	—	0·3179
Brake specific fuel consumption, lb/bhp-h	0·351	0·353
Indicated specific air consumption, lb/bhp-h	—	13·73
Brake specific air consumption, lb/bhp-h	14·4	15·2
Mechanical efficiency per cent	—	90
Input temperature, °C	25·0	24·0
Exhaust temperature, °C	390·1	396·1
Boost pressure ratio	1·86	1·86
Maximum cylinder pressure, lb/in²	1523	1506
Heat lost in cylinder per cent	15·0	15·0
Heat lost in exhaust per cent	31·5	24·6

Much remains to be done, however, with theoretical calculations, particularly where the turbocharger is concerned.

Initial computer calculations do allow basic design parameters to be given to the designer and enable an overall engine layout to be produced in scheme form. It is then necessary to consider each component group in isolation to analyse the main limiting factors, subsequently considering each group within the complete design to enable overall limiting requirements to be estimated. The designer must be capable of assessing here which factor can be ignored as capable of satisfactory solution at the detail design stage, within the limitation of the fixed requirements introduced during scheming. Rate of convergence of the design process is closely dependent on the sequence in which the main component groups are considered in detail.

EXAMPLES OF DESIGN TECHNIQUES

At present few engine components can be designed without experimental verification. The various items are interrelated and must be mutually compatible.

For the crankcase, whether cast or fabricated, theoretical estimates of stress and deflexion are at best very approximate and it is necessary to resort to some form of experimental analysis. Model techniques, in varying degrees of sophistication, have been developed and are used to finalize the design(1).

These techniques have been developed to the point where they permit engine components to be finalized with fair certainty of success, thus greatly reducing the development effort with obvious economic advantages. Rubber models can be produced quickly and cheaply and modified with minimum effort. These readily high-light stress concentrations or excessive deflexions and give an immediate estimate of material utilization. Fig. 16 shows a series of rubber models of an engine bedplate diaphragm.

Coating of the models by a suitable lacquer which cracks

(a) original (b) first modification (c) final design

FIG. 16—*Models of bedplate diaphragm*

under the strains produced by the loading system, allows the presence of any stress concentration to be observed and, under controlled conditions, the magnitude of stresses in the actual prototype engine structure can be predicted within 20 per cent.

Fig. 16 (a) shows an original design of diaphragm, not in this case the AT engine, which produced main bearing failures under particular conditions of operation.

Production of the rubber model immediately showed the reason for this. Under the action of the firing load, the diaphragm deflects, allowing the bearing horns to contract and nip the crankshaft. This model also shows which parts of the diaphragm in the affected region are fully utilized.

The design shown in Fig. 16 (b) eliminates the initial defect, but again this model shows that only a portion of the material is contributing to the overall stiffness of the component. The design shown in Fig. 16 (c) eliminates this further undesirable feature and, by comparison with the original, results in a structure having three times the stiffness without any local high deflexion and without any unnecessary stress concentration within the same overall weight. This type of technique has been used in the design of components for both in-line and vee versions of the AT engine.

A further development in model techniques is the use of scale perspex and epoxy resin models in which a scale reproduction of a complex component, or of a complete engine, is built up, the epoxy resin composition being adjusted to have the same elastic properties as the perspex. Fig. 17 shows a scale model of the vee version of the AT engine in a loading frame capable of applying scale firing load to the engine,and which enables the stresses and deflexions due to torque reaction to be measured.

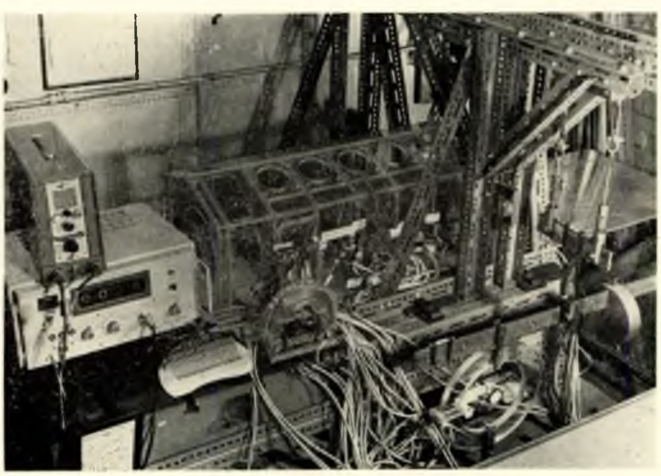


FIG. 17—*Scale model of AT vee engine in loading frame*

Model deflexions are measured directly. The process of stress analysis is to coat the model with a water-soluble brittle lacquer under controlled conditions. Load is then applied, the resultant strains causing the lacquer to crack. The direction of the cracks indicate the orientation of the principal stresses in the member under consideration and the density of the cracks give a measure of the magnitude of the stress under the particular load. The overall stress picture can be predicted accurately by means of strain gauges attached to the model at the areas of stress concentration as revealed by the brittle coating. The stresses in the prototype are derived by simple non-dimensional analysis. Accuracies of the order of plus or minus 10 per cent are usual. Modifications can be readily incorporated in the model and re-testing undertaken in a very short period, the process being repeated until stress levels are within acceptable limits and all points of major stress concentration eliminated.

Fig. 13 shows a full-size 16-cylinder engine, designed using the principles outlined above. The cost of the crankcase-model work in this particular instance amounted to £2900 and occupied some 80 man-weeks. It enabled the structure to be specified with a high degree of certainty, avoiding the production of a costly and possibly useless prototype.

The main connecting-rod dimensions of an engine can be designed with a degree of certainty. The total stress in the connecting-rod shank is made up from the direct stress due to the component of applied forces parallel to the axis, bending stress induced by friction moments and bending stress due to the neutral axis of the connecting-rod being offset from the line of action of the forces parallel to the rod axis. The major loads allow stresses to be calculated through the engine cycle and the magnitude of maximum values determined. Superposition of the minor effects gives a complete stress analysis at the critical point.

Eccentricity of loading can occur in both tranverse and longitudinal planes due to initial imperfections in the connecting-rod itself, to float at the small end, to bearing deflexion and to bending of the rod by applied loading. The maximum stress produced by these effects can be estimated by a reference to dimensional tolerances and limiting behaviour imposed by adjacent components.

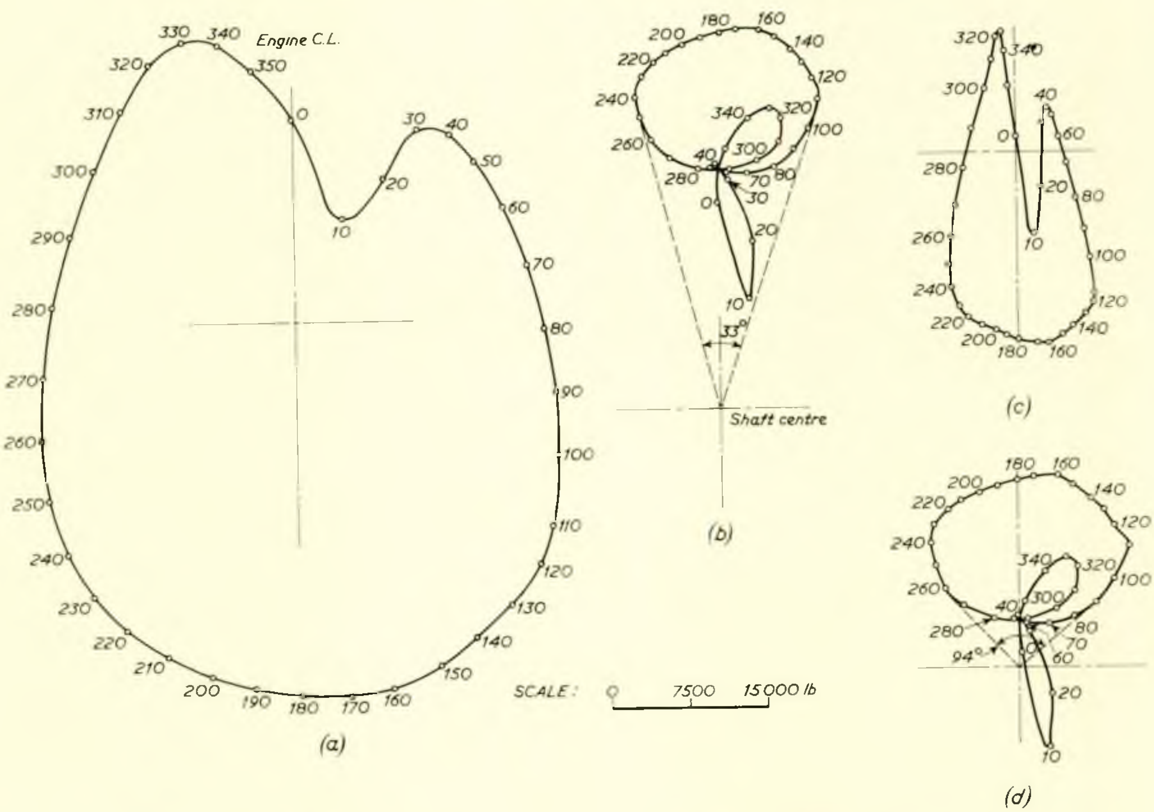
The maximum and minimum values of stress obtained from the calculation at the crucial section can be converted to a fluctuating stress imposed on a mean stress value.

Data exist for a range of suitable materials to allow the effect of material composition and surface finish, mean stress and type of loading to be assessed under cyclic loading conditions. It is, however, necessary to make additional allowance for contingencies, namely material defects, manufacturing errors and scatter of experimental fatigue data. By logical argument it is possible to account for these features either as a safety factor increasing the computed stress or the safety factor reducing the allowable stress.

The design method outlined in the foregoing appears simple. The calculation is, however, a prime example of design by trial and error since, in order to calculate loadings, it is necessary to assume connecting-rod dimensions which in themselves are determined by the magnitude of loading and the magnitude of bearing deflexions which occur.

While the method does allow the main stresses in the connecting-rod shank to be estimated accurately, determination of bearing stiffness presents a more complex problem and in this respect it is necessary to resort to rig testing using either scale models or a loading rig capable of accommodating full-scale engine loads. Unfortunately, the method of applying loading in the rig cannot strictly simulate engine conditions. In the rig, the loads have to be applied to the bearing through a mandrel, whereas in the engine they are transmitted through an oil film. Over the years, techniques have been developed which allow results from the mandrel-loaded rig to be correlated with a deflexion which is known to give satisfactory operation in service.

Accurate predictions of deflexion, in general within the

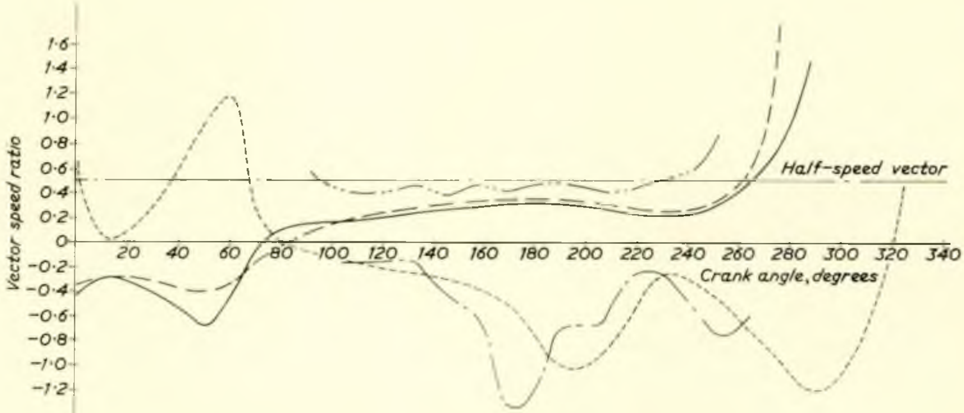


(a) original relative to bearing
(b) original relative to crank journal
(c) modified relative to bearing
(d) modified relative to crank journal

FIG. 18—*Bearing load diagrams*

bearing bore and in particular at the joint, have allowed a satisfactory design to be evolved which gives accurate location between the cap and the main body of the connecting-rod without any discontinuity or fretting. These techniques are similar to those described for the engine frame and progress through the various stages of rubber models, simple metal models and full-scale components. Similar measuring techniques are also used during the static tests, the design being finally proved by strain-gauge testing in an engine, under operating conditions. Predicted and measured stress correlate within plus or minus 10 per cent.

Prediction of bearing loading can, to a first approximation,



Curve	Speed, rev/min	Firing pressure, lb/in²	B.M.E.P., lb/in²	Balance wt. wr. lb-in
— — —	500	1750	200	1150
———	600	1750	200	1150
—··—··	750	1750	200	1150
—·—·—	750	1750	200	2240
- - - - -	750	2240	250	2240

FIG. 19—*Vector speed ratio for main bearing*

be considered separately once the design of the rod has been fixed and in very recent years large advances in prediction of bearing performance have been made.

In the past, where white-metal bearings were the general rule, loading applied to this type of bearing was limited by fatigue considerations in the main and in any design it was normally sufficient to limit the maximum applied specific loading to give satisfactory operation.

This state of affairs no longer exists with modern hard bearing materials such as aluminium tin or copper lead and an entirely different approach to bearing design is now necessary.

With the new approach, simple considerations of maximum specific loading, even utilizing accurate loading diagrams, is impossible. The assessment of an accurate load diagram is, of course, dependent on knowledge of component masses associated with piston, connecting-rod and crankshaft and on having an accurate indicator diagram.

In any modern approach, it is necessary to consider the load pattern, which is of far greater importance than maximum loading.

Fig. 18 shows load diagrams from a low-rated engine which has in the past produced bearing failures. It can be seen from Fig. 18(a) that the maximum loading is only 37 500 lb, giving a specific load of 1080 lb/in^2. When the load diagram is drawn relative to the crank pin, see Fig. 18(b), the reason for bearing failure becomes apparent. The load rotates at very nearly pin speed for the majority of the cycle and loading is applied over a very small arc of the pin. This results in overheating of the pin in this region, causing oil film breakdown and subsequent failure. Figs. 18(c) and 18(d) show the same two diagrams with modified balance weights. The specific load has been reduced to 570 lb/in^2, but, more important, over the major portion of the cycle the extent of the loaded arc of the pin has been increased by a factor of three.

When the load vector rotates at half the speed of rotation of the shaft, the oil film thickness and hence the load-carrying capacity are theoretically zero. In the case of large-end bearings, the superimposed oscillation due to the motion of the connecting-rod must be taken into account. The "vector speed ratio", i.e. the ratio of angle moved by load vector to angle moved by journal, is shown in Fig. 19 for the centre main bearing of the AT engine at various operating conditions. This shows that as speed is increased, the "half-speed vector" condition is approached. By a change in balance weight size it is possible to alter the load pattern, and hence vector speed ratio, to achieve a satisfactory system.

With an oscillating load on the bearing, this simple approach is not always sufficient and account must be taken of the previous locus of the shaft. The shaft locus must be calculated by an iterative process(2). A computer programme has been devised and developed to carry out this process. Adoption of this detailed approach can give a good qualitative assessment of bearing performance and can be used to show up the effect of changes in reciprocating or rotating weights, alterations in bearing design and effect of engine speed. When allied to practical experience of satisfactory and unsatisfactory bearings, the approach becomes a powerful design tool.

Fig. 20 shows the resultant eccentricity ratio diagrams for a particular bearing, illustrating how the theoretical approach can show the effects of various changes. Eccentricity ratio (n) is related to the minimum film thickness (h min) by the following equation.

$$h \text{ min} = c\,(1 - n), \text{ where } c = \text{radial clearance.}$$

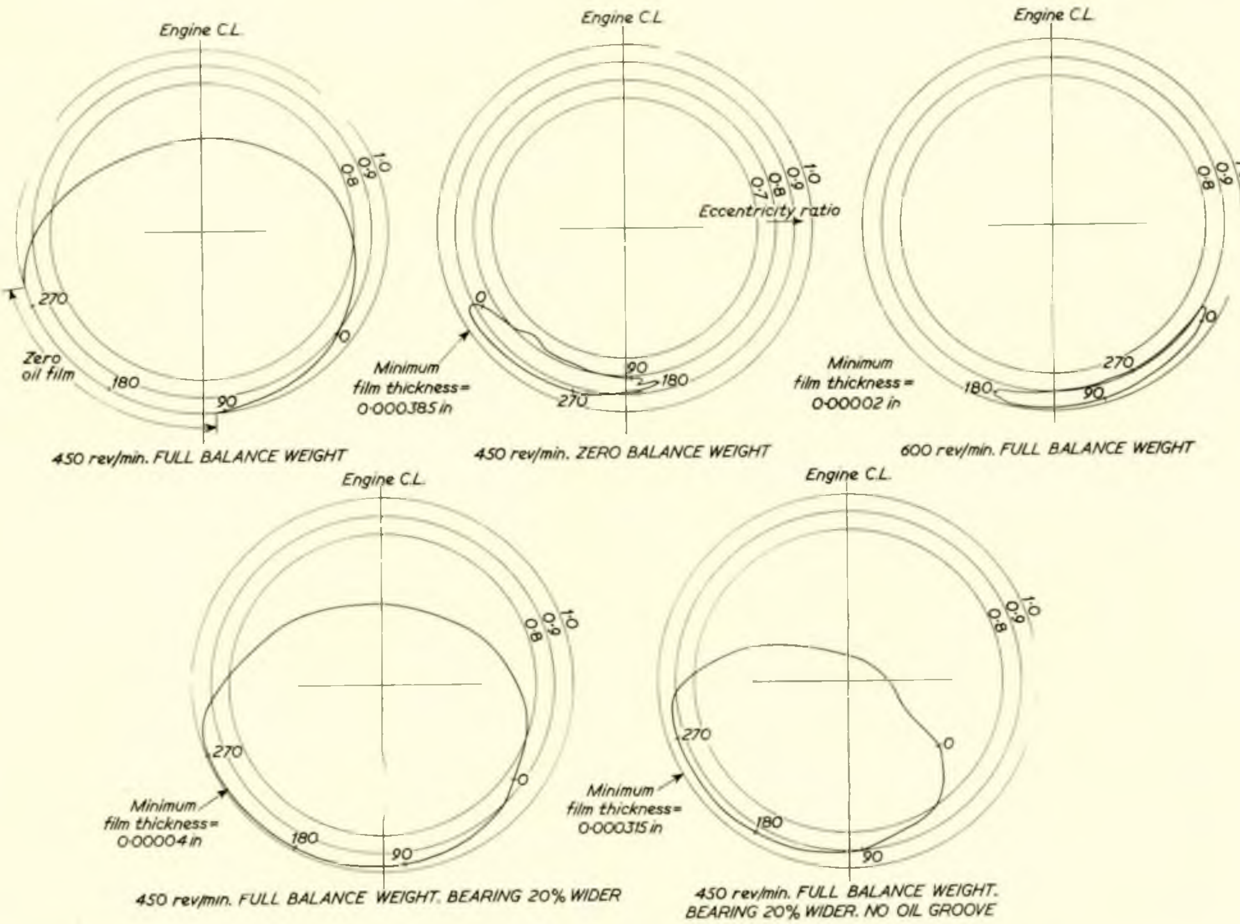


FIG. 20—*Eccentricity ratio for bearing under various operating conditions*

Hence the closer the eccentricity ratio is to 1, the closer the film thickness becomes to zero.

DESIGN WITH COMBINED MECHANICAL AND THERMAL LOADINGS

As an example of a design problem involving a combination of thermal load and mechanical load, the cylinder head is worthy of consideration in some detail.

Until the introduction of the AT range, the company had, for simplicity, retained the two-valve cylinder head arrangement. It was realized that, for the AT range to achieve the high specific output aimed at, a four-valve head arrangement was necessary and, as the engine was intended to digest inferior fuels of high viscosity and containing undesirable impurities, caged valves which had been present in previous designs were rejected.

The adoption of four valves and the elimination of valves in cages provides two advantages. A larger valve area per square inch of piston area is obtained, than with a two-valve arrangement, thus allowing a higher density of charge, and the valve seats are cooled more effectively, thereby enabling inferior fuels to be burned more readily with acceptable valve-seat life.

It is worth while considering the steps in the design process, the first of which is to produce a scheme design incorporating all known desirable features, taking these to the maximum possible extent. At this stage there is considerable compromise between conflicting requirements.

The initial scheme design must then be subjected to theoretical analysis to calculate thermal gradients and mechanical loadings in the system. These calculations are complex and rely on assumptions based on previous practice, especially in the estimation of point-heat fluxes used. This type of exercise can show up any obvious undesirable features and from this it is then possible to produce a second-stage design in some detail, taking into account any manufacturing difficulties which are likely to be involved, after which prototype units can be built.

Having produced prototype units, it is then necessary to measure actual temperatures in the critical regions of the cylinder head under operating conditions and to measure stresses of a purely mechanical nature. The latter can be obtained from static tests either on the engine or on a specially constructed rig. Thermal and mechanical stresses can then be computed and superimposed to give an estimate of total stress.

FIG. 21—*Instrumented AT cylinder head*

Several methods of increasing complexity can be applied to measure temperatures under operating conditions. Fig. 21 shows a sophisticated method where a cylinder head has been fitted with a high degree of instrumentation utilizing fixed and traversing thermocouples. The head shown has 32 fixed thermocouples inserted in the cylinder head material in defined positions, eight traversing thermocouples mounted in a horizontal direction and operated by an electric motor with push-button control and seven traversing thermocouples mounted in the vertical direction. Data-logging equipment is necessary to record temperatures during operation.

The use of such a cylinder head does allow the effect of changes in load and speed on metal temperatures to be accurately recorded over a very short period and precludes any deterioration of engine condition which could occur over a lengthy period of running.

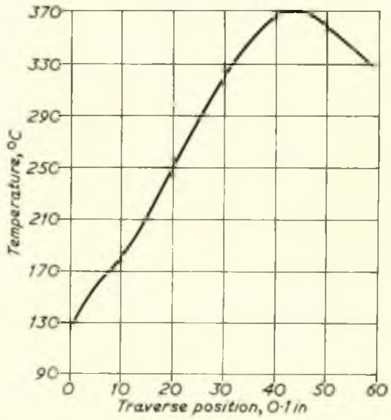


FIG. 22—*Results of traverse taken across an AT cylinder head*

Fig. 22 shows an example of results of an actual traverse taken across the bottom deck of the cylinder head through the region between exhaust valves; position 0 represents the outside of plate at liner radius.

During the particular investigation under consideration, approximately 7500 individual readings were taken during tests and, while complex analysis of the results has been necessary[(3)], this has been undertaken and it has allowed the AT cylinder head to be designed to eliminate failures in service due to operation at the high ratings of which the engine is capable.

It is of course necessary, before placing any component into service, to thoroughly test under adverse conditions on a full-scale engine. With thermally-loaded components around the combustion chamber, failures occur due to high thermal stress at sustained high power and due to fatigue loading brought about by rapid changes in load. It is the policy of the company to operate engines, in general at ratings some 15 per cent higher than operated in service, for periods up to 5000 hours.

In the particular case of the AT engine, a system of thermal cycling was adopted to try to induce fatigue failure. This consisted of running the engine for a short period at maximum overload conditions, followed by a longer period of shut-down with extreme external cooling applied to the engine.

This procedure was adopted and followed for hundreds of cycles and did contribute to an understanding of thermal problems associated with cylinder heads and combustion chamber components in general.

Although the foregoing has been quite a lenthy discourse, it has only covered a small proportion of design and development involved in a medium-speed engine.

Overall, the new approaches to design and development adopted by the company have allowed engines to be put into service at the higher rating without any reduction in reliability or durability by comparison with their predecessors.

At the various ratings there are now some 250 AT engines either in service or in course of manufacture, approximately 40 per cent being applied to marine duties and the remainder for land-based power generation. At the present time the early units put in service have individually operated for periods of up to 40 000 hours.

ACKNOWLEDGMENTS

The body of this paper describes results of co-ordinated effort by design, development and research teams at Ruston and Hornsby Ltd. and Davey, Paxman and Co. Ltd. over the last ten years, and all credit for the features mentioned lies with the individuals concerned.

The authors wish to acknowledge permission granted by their respective companies to present this paper and to draw on the group's resources for material contained therein.

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Discussion

MR. J. CALDERWOOD, M.Sc. (Honorary Vice-President) said that when he was asked to open the discussion he was a little worried, as the paper was concerned mainly with statements of fact and had very few statements of opinion. Therefore it left little room for discussion.

One rather interesting point was that he remembered having discussions with the late Mr. Oswald Wans of Rustons about the tendency for a great increase in engine speeds and piston speeds, not that Mr. Wans was against progress at all, but he felt that, in general, engine makers at that time—the middle 1930s—were trying to move much too fast. Mr. Wans fully realized that higher speeds were coming, but that progress should be step by step. Today both revolution speed and piston speed were about twice the speed common at the date of his talk with Mr. Wans.

The Ventura engine was made in six, eight, twelve and sixteen cylinders in vee formation. He wondered how the balancing of the six-cylinder engine, and even more the eight-cylinder engine, was achieved, because vee engines with such numbers of cylinders were extremely awkward to balance thoroughly. Had the eight cylinders a 90° shaft or a 180° shaft? He wondered what additional balancing facilities, such as countershafts, balancing weights, and so on, there were.

In the paper, he would have liked to see more information about the welding of the Ventura. What had struck him was that very heavy sections—for instance, the foot of the frame and the top of the cylinders appeared to be about $1\frac{1}{2}$-$1\frac{5}{8}$ in—were welded to side plates of approximately $\frac{3}{8}$ in, with no gradual sloping off, such as he had noticed in the Ruston engine, where the thin plates and thick plates joined. It must have been quite a problem in the welding not to get excessive stresses at these welds.

He would be grateful for a little more information about the welded structure.

Mr. Howe had said that stress relieving was carried on for two hours. His experience was that the length of time at the soaking temperature was relatively unimportant in stress relieving. What really mattered was the speed at which heating and cooling proceeded and that was dependent on the relative thickness of the various sections. He would have thought that Mr. Howe could not have heated up or cooled down a structure such as that safely at the rate of more than 40 deg C per hour, which would mean a 24 or 28 hour complete cycle. He would be interested to have more information on this.

Mr. Howe had said that all the plates were tested ultrasonically. He wondered whether one could get useful results from ultrasonic testing of very thin plate.

Turning to the crankshaft, the question of oil holes was very interesting. He could see the argument, but even with the additional sketch that Mr. Howe had put on the blackboard, he could not see how the oil holes ran. He wondered whether they ran into the centre of the pin, the bore. If not, he could not quite see how one could get straight right angle holes through to the position where Mr. Howe wanted to get them.

As a slight criticism of the paper, under the heading "Cylinder Head and Piston", not a word was mentioned in the text about the cylinder head. He understood that the cylinder heads were of aluminium, although that was not mentioned in the paper. He wondered whether there had been any trouble arising from the relative expansions of the aluminium head and the cast-iron liner and the steel block. Also, he wondered what material Mr. Howe used for the valve inserts in the cylinder head in order to prevent them becoming slack due to differential expansion.

With regard to the pistons, the diagram in the paper was not quite the same as the one that Mr. Howe had shown on his slide. According to the diagram in the paper, there was a top ring groove temperature of 250°C, which would almost certainly cause considerable trouble from carbon formation. It might be that there was a mistake in the diagram. However, Mr. Howe had certainly shown that that system of piston cooling give nice horizontal lines for the isotherms, which was just what one wanted.

Mr. Howe had mentioned that he and his colleagues were thinking of the built-in cooling coils primary behind the ring grooves. It was a system of cooling that he had always disliked. While it cooled the grooves, it left the centre hot and gave rise to temperature stress. Many people thought that oil temperature as such was not quite so important as many used to consider. Mr. Howe seemed to allow an oil temperature of about 200°F.

He had one important point of disagreement with Dr. Watson over his part of the paper. Dr. Watson had said that in the old days one could take an engine straight off the drawing board with safety. He did not think that such a condition had ever existed. In those days the engine was certainly not as complicated as it was today, but if one had not given it a good testing first in the works and then in service, one could not be sure about it, however good the designer might be. He knew that some people had taken engines straight off the drawing board, but the poor customer had certainly paid for it by troubles in service.

With regard to the costs diagram, Dr. Watson had figures for the 1930s. It was a pity that he had mentioned them at all, because they were completely misleading. People were selling engines at any price at the time in order to get a little work into the factory, often the price was far less than the engine cost. However, the rest of the curve was extremely interesting. It was interesting that the cost per bhp of Diesel engines had remained, within fairly narrow limits, practically constant since before the first war. This was a remarkable confirmation of technical progress when one considered that the cost of labour and materials had risen by about 10 per cent over the period.

As Dr. Watson was using aluminium tin bearings, he wondered whether the crankshaft had hardened. Dr. Watson had not mentioned that.

A more important point was that it appeared to him from the drawing that Dr. Watson had shown that the fuel pumps and valves were all inside the casing, and there was forced-fed lubrication on the valve gear. He wondered whether there was not a risk of contamination.

Dr. Watson referred to a further increase in the rating of the engine. It seemed doubtful to him whether the piston cooling system that Dr. Watson had on an engine of that kind would be satisfactory when the ratings were above 200 lb/in^2.

In conclusion he thanked both authors for their interesting and valuable paper.

MR. C. C. J. FRENCH, M.Sc. (Member) said that under the general heading of thermal loading, as the authors had said, thermal loading increased with increased load and he would agree with the greater use of stepped or of tapered small ends to improve the small-end condition. In so far as piston temperatures were concerned, however, and cast-in cooling passages, the improved cooling would presumably have to be applied both behind the top piston ring and under the whole area of the crown. That would mean a spiral cooling tube under the whole crown and he wondered how much of a casting problem that would be, even with soluble cores. The alternative was a two-part piston. He invited any comments which the authors might have on those alternatives.

For cylinder heads of highly rated engines, his company recommended cooling passages drilled in the bottom deck, to give the most effective cooling. He noticed that the Ruston AT engine had a double-deck construction to give effective flame plate cooling, whereas on the AO engine it had gone to a separate flame plate. He asked for Dr. Watson's comments on the relative thermal effectiveness of the two methods.

With regard to additives in engine coolants, problems could arise due to the thermal barrier imposed by quite thin layers of material deposited on the metal surface from the additives and, in bad cases, very considerable increases in metal temperatures could result from the use of inappropriate additives. The deposits laid down were not the result of corrosion, but were apparently laid down as part of the corrosion prevention mechanism.

Turning to the engine structure, Mr. Howe had added some data in his presentation, and Mr. Calderwood had already commented on it. The authors quoted a limiting fatigue stress of 2 tons/in^2 for a welded joint. Compared with the strength of the parent metal, that was not impressive, but he was sure that the low figure was realistic, as had perhaps been emphasized by British Railways published experience on the crankcases of Diesel engines from a variety of manufacturers.

With that low operating stress level, cast iron still looked attractive and he wondered whether the authors knew of any work that might be in progress anywhere to raise the stress levels for welded structures. One might even consider alternative methods of gluing the structure together.

So far as rating was concerned, Fig. 10 was fascinating. Dr. Watson had commented further in his presentation, but Mr. French asked him to be a little more definite. He had done a short extrapolation of Fig. 10, and it had given about 315 lb/in^2 b.m.e.p. at 2250 ft/min piston speed by 1970, with a continuing reduction in cost per brake horsepower. It was interesting to speculate whether there was some limiting rating above which the increasing sophistication resulting from the increased rating might result in increasing specific cost. Had Dr. Watson had any additional information on that point?

Finally, perhaps he might be forgiven for asking Mr. Howe a rather tongue in cheek question. Paxmans had stuck to blade and fork construction for the big ends through, he suspected, thick and thin, but it had obviously given them good service. It was, however, a complicated construction, and if the firm were starting from scratch on a highly rated engine, did not Mr. Howe feel that side by side rods, or even an articulated construction, might be a better choice?

CAPTAIN W. A. STEWART, C.B.E., R.N. (Member) said that he was with the British Railways Board.

In the paper, Mr. Howe had mentioned the four main features of the design, which seemed to have been more of a development of existing design experience, rather than adopting anything new. Captain Stewart first referred to the welded engine frame, with which, as Mr. French had said, British Railways had had such a disastrous time—in fact, so much so that he was afraid that he might be considered slightly biased about it.

The opening speaker had mentioned the joining of two quite different sizes of plate directly by welding. Captain Stewart entirely agreed with what had been said; he thought that it was bad practice and should be avoided, if one had to use welds at all.

He understood that the design of the welded frame for the Ventura was done considerably in advance of the work that had been done recently, by Rustons, Dr. Watson and other workers, on predicting stresses and generally in bringing in their new philosophy on frames, such as the space frame on the AO engine. He would have liked to ask Mr. Howe for his personal views on the decision to have welded frames for railway engines. However, he thought that the latter had really answered the question in his talk by saying that the engine had to be designed for a multiplicity of uses and could be supplied with a cast iron frame. On the railways, they felt very strongly that the weight saving that accrued from welding steel frames had now become an unreliable type of saving and should not be adopted by anybody who was not frightfully sensitive about the total weight. The percentage increase in weight for cast iron had not been stated anywhere, but it was a matter of a few hundred pounds and, in the case of British Railways, as a locomotive weighed about 100 tons, it did not seem to be very formidable. In any case, British Railways needed the weight for traction adhesion.

Mr. French had pointed out in a previous paper the characteristics of the world's leading high output locomotive Diesels. The figure that interested British Railways was the specific volume/c.v. rather than the specific weight. If one could gain reliability, long life and repeatable good quality over a large production range, it seemed appropriate to have something very much better than welded frames for highly rated engines. The art of stress relieving had not absolved the welder from doing certain things during manufacture which might cause trouble over a long life. The sort of life that British Railways was talking about was 10^{10} loaded cycles.

British Railways now considered that 1·5 to 2·0 tons/in^2 was the maximum stress that could safely be used because of low scatter and imperfections in the welds themselves, especially in fillet welds which should always be avoided. Owing to the multiplicity of welds, it was almost impossible to check them all. So it depended on the skill of the operator; Captain Stewart had often found that to be disastrous. British Railways considered that the Ventura engine was so good with regard to specific volume that it could well afford to carry the extra few hundred pounds required for a cast crankcase in railway service.

He understood that recent developments in bearing materials had led Paxmans to change to 20 per cent tin/aluminium for these. He would very much like to know whether Mr. Howe considered that that material was less sensitive to dirt and, particularly, whether the oil films were very thin.

Also with regard to bearing materials, he would like to know whether Mr. Howe considered that the new material justified the expense and complication of fork and blade rods, rather than the more commonly used side by side rods in other similar machines. Spare gear was always a problem, and those present had seen what a complicated thing the highly stressed rod became.

With regard to high cylinder pressures and high supercharge, apart from increasing bearing loads, the high cylinder pressures resulting from higher supercharger pressures caused trouble to the cylinder block and widened the stress cycle in load carrying welds. Generally they led to trouble after about four or five years. He wondered whether Mr. Howe's firm had given due consideration to varying the compression pressure with load by one or other means.

Mr. Howe had not mentioned the cylinder liners of steel, nor the cylinder head of aluminium. Captain Stewart believed that both of those were novel features in the design of such engines, although valid from previous experience. The cylinder head of cast iron had been found to be excellent in all respects. It had shown no trouble whatever. On the other hand, the aluminium head had given rise to some trouble and British Railways were gradually getting rid of it. That was just as a result of the experience that British Railways had had. It might be due to casting troubles or flexing of the head, or it might be something to do with the difficulties with the valve seats. All in all, British Railways had found the cast-iron head in the engine very successful indeed, and that it went well with the whole of the cast-iron block.

He was interested in the curves that had been given, and would like to know whether Mr. Howe could explain why the "valley" of lowest consumption was so far down the b.m.e.p. and rev/min scale, as the declared continuous rating for the 16 YJC was 1850 hp at 1500 rev/min and 10·8 kg/cm^2. It would seem that the engine would be better run at 1300 rev/min and 12 kg/cm^2 as at present designed or was its volumetric efficiency all that it should be?

With regard to the section of the paper written by Dr. Watson, the cost per horsepower seemed steadily to have come down and Captain Stewart wished to know where the Ventura engine appeared on the curve.

Mr. W. Lowe, B.Sc. requested from Mr. Howe, the oil and water operating temperatures for the Ventura, as, he said, the cooling system described must give very tight restrictions. It seemed to him to be fundamentally unsound to use a system in which minimum oil temperature must be substantially above maximum water temperature; either excessively low water temperature or high oil temperature must result, neither of which would have a beneficial effect on the engine. Since thermostatic control was used on the jacket-water system, there was no reason why the same method of temperature control should not be used on a separate air cooling system.

He was surprised to notice on the AT engine a fundamental structural difference between the in-line and vee-engines. With an in-line engine, the underslung crank construction made a very attractive design, in that the load path could be led directly from the cylinder head down to the reaction at the main bearings. The vee-engine was far less attractive in that a direct path was not possible. The AT engine, however, had the underslung principle on the vee-engine, whereas the in-line was a bedplate construction. He would be interested to know the reason for that philosophy.

He had a very great admiration for the model work and computer cycle calculation work carried out and, in many cases, pioneered, by Ruston, and the examples given in the paper were of great interest. He dared to suggest, with almost no basis but engineering intuition, that the 13 : 1 compression ratio of the AT was rather high for a rating of 200 lb/in^2 b.m.e.p. and more so at the proposed 300 lb/in^2 b.m.e.p. The engine thermal efficiency might well be better at the same maximum cylinder pressure with a rather lower compression ratio and a higher boost.

Mr. J. H. Attwood (Member) said that his comments would be general in the sense that they applied more to the application of high and medium-speed Diesels for main propulsion and were intended to cover some of the problems that the shipowner might be faced with in selecting the type of engine for a particular application.

Generally speaking, the medium-speed Diesel covered the range over which his organization was presently interested, in as much as the power output of the high-speed Diesel was rather too low for present and future requirements.

The widest use of the Diesel in the medium-speed range would probably be in the field of coastal carriers. Such organizations as his had certain requirements and these included compactness of the engine-room machinery and low weight, as clearly the penalty for cargo shut-out with this class of ship was very great indeed.

The maintenance must be low and the reliability must be extremely high; these two requirements were not always compatible.

Another requirement was to get as high an efficiency as possible consistent with the other requirements. In this respect there were several problems and he would divide them into two parts. First there was the problem that on the coastal trade with a high vessel utilization, there was a great deal of manœuvring, requiring operation over a wide power range. In addition, when discharging cargo the shipowner wished to use the main propulsion Diesels to drive an alternator of sufficient capacity to run the cargo pumps. This meant that a main Diesel would be running comparatively low down its load range when in harbour.

Another problem relating to efficiency arose out of the propulsive power requirements of a particular vessel. Initially these power requirements were obtained from tank tests of models, taking into account the required ship's speed and the amount of cargo to be carried. However, the power obtained was related to the clean hull condition and this condition would not be maintained once the ship went into service. If the ship's speed was to be maintained the shipowner must have additional power available to feed into the system to compensate for the ship slowing down due to hull fouling, and so, in many ways, a compromise had to be reached. Even with having this additional power available it might be that as the ship became dirtier the engine would be quickly up to its maximum b.m.e.p. and, if this condition was continued for an extended time, there might be conflict with one of the essential requirements, which was low maintenance coupled with high reliability.

It might be said that one could get that power margin by reducing the revolutions in the initial stages and increasing the revolutions later, thereby keeping the b.m.e.p. constant and merely increasing the revolutions to compensate as the hull fouled. This would mean that during the time before the hull became extensively fouled the propeller would not be operating at its designed condition, thereby incurring a financial penalty for the shipowner over a considerable part of the ship's operating life.

In the light of what he had said, he wished to ask Dr. Watson whether he had any feel for the maximum b.m.e.p. that one could design for in such conditions and with the stated operating requirements. It was accepted that the paper talked about 300 b.m.e.p. and other figures had been mentioned at the meeting, but there were these other criteria, which the shipowner still had to meet, and he wondered whether the shipowner could afford to run continuously at the maximum designed b.m.e.p. of the engine.

Dr. W. P. Mansfield said that Dr. Watson's Fig. 10 showed, in impressive manner, the acceleration that had occurred in development towards higher specific power. There was every indication, he believed, that the rapid increase in b.m.e.p., which had made the major contribution, would continue and that b.m.e.p. would reach at least double the present rated values in the not very distant future.

Both authors had pointed to the fact that, even at the present stage of development, the air supply equipment on the engine—the turbochargers and the after-coolers—were beginning to occupy rather a lot of space. Clearly, if the size of the engine were to be halved, that situation would appear to be still worse—or it would do if those concerned did not do something about it. It was an area in which there was considerable scope for improvement. At present the general method was to accept the turbochargers and heat exchangers more or less in the shapes in which they were supplied as stock items by the component makers, mount them near the engine and couple them together with large pipes. He thought that in the future a different line would have to be taken. It would be necessary to design the items to fit in with the contour of the engine or even to adapt the contour of the engine to suit them and, in effect, design the air-supply system with the same attention to compactness as was applied to the design of the rest of the engine.

Of course, if there was that continuing rapid increase in b.m.e.p., maintenance of present levels of reliability would depend largely upon the control of mechanical and thermal loading. He would not enlarge upon the BICERI method of dealing with that problem as that was well known, but he would point to the fact that the simple aluminium pistons used in the Ventura engine could not be expected to give satisfactory service at the greatly increased ratings

The experience of several companies developing engines well above the 200 lb/in^2, b.m.e.p., level had shown the need for a crown and ring belt portion of iron or steel. The requirement could be met by bolting an iron or steel cap on to an aluminium piston; that solution had been adopted in several cases. That construction resulted in a piston of weight similar to that of the lighter designs of variable-ratio piston, in which a similar cap was arranged to move relative to the rest of the piston. Hence, it appeared that, whether fixed or variable ratio pistons were used, the connecting-rod assemblies of the 400 lb/in^2, b.m.e.p., engines would need to withstand higher inertia forces than were produced by simple aluminium alloy pistons. Those forces would be increased still further by any further increase of engine speed.

That brought him to the type of connecting-rod assembly adopted for the Ventura engine. According to Mr. Howe's opening remarks in his paper, the fork and blade rods were inherited from the YH engine. Dr. Mansfield's question was whether that was regarded as the right choice for the future. It was interesting to note that the entirely different articulated type of construction was used in the AT engine. Three of the most advanced vee-engines in the world, the Daimler Benz MB 873, the Caterpillar VHO, and the Continental AVCR 1100, all had side by side rods. The Caterpillar engine, which used a disc-web type of crankshaft and fixed-ratio pistons with a pre-combustion chamber, was rated at 258 lb/in^2, b.m.e.p., which resulted in a peak cylinder pressure of 3000 lb/in^2, and the Continental engine, which had variable-ratio pistons giving a minimum ratio of 10: 1, was rated at 373 lb/in^2, b.m.e.p.—in spite of an air manifold temperature as high as 121°C (250°F)—with a peak cylinder pressure of 2200 lb/in^2. As the variable-ratio pistons in the Continental engine had a pearlitic malleable iron outer portion, of the earlier heavy design, their use added about 46 per cent to the total reciprocating weight, yet they ran in the five inch stroke engine at a rated speed of 2800 rev/min. It thus appeared that the side by side rod arrangement, which presumably was less expensive, could accommodate both high gas loads and high inertia loads. The authors must have given much detailed consideration over the years to the choice of rod arrangement and it would be most interesting to hear their present views.

Mr. C. A. Beard said that he was interested to note that nuts used for the big-end bolts of the Ventura engine were integral with a thick washer having an outside diameter equal to the "across corners" dimension of the nut. He would be interested to know the evolution of that practice, which, in his opinion, could be used more widely since it increased the area of the nut seating face.

These days there was an increasing tendency to use high tensile bolts and studs, and to tighten them to correspondingly high stress levels. Rightly from the fatigue angle, soft, low strength nuts were usually used on those high strength bolts. However, an important point often overlooked in his experience was whether the nut face and its abutment on the clamped member could stand the applied load.

Usually the threads had sufficient strength in shear for the applied loads, but cases might be met, with certain standard nut sizes, where the bursting strength of a low strength nut was inadequate. Similarly, he had met cases where the normal loading of a nut face, allowing for the chamfer at the end of the hole in the clamped member, had equalled the ultimate tensile strength of the nut material.

Similarly the stress levels and deflexions in the clamped member should be considered. Cases could be found where yielding of cylinder head bosses had occurred under the nut face loads. That was not a happy state of affairs. Again, he had recently met a case in which the cylinder head bosses shortened by a calculated 0·011 in, which must impose additional stresses and deflexions in the top and bottom decks of the cylinder head.

Clearly, more was involved than tthe mere tightening of the nut to provide a load corresponding to a certain stress level in the bolt or stud. He thought that this was worthy of consideration by nut and bolt standardization committees as well as by designers.

Turning to Dr. Watson's section of the paper, he wished to query Figs. 18 and 19. The text referred to the load diagram —18(b)—drawn relative to the crank pin and then to Figs. 18 (c) and (d), showing the effect of modified balance weights. As crankshaft balance weights would not affect big-end loads, he suspected that those diagrams really applied to a main bearing—probably the centre main bearing of a six or eight throw crankshaft.

Also, the title of Fig. 19 appeared to be wrong and should read "Vector speed ratio for centre-main bearing". He wondered whether Dr. Watson could confirm that.

Mr. N. H. New said that he also wished to query the point raised by the last speaker. He believed from experience that it was a centre bearing.

Dr. Watson had mentioned that the improvement in the bearing was due to the fact that it was no longer heating at one point on the pin and also that the oil film was not allowed to collapse. That might add some contributory factor, but he rather felt that a more important effect was the fact that the load oscillated in magnitude and there was an effective squeeze action, which resulted from the change in radial load.

What was shown in Fig. 18(a) was very close to a pure rotating load of the order of 1000 lb/in^2 and a pure rotating load of that order was extremely large, giving rise to thin oil films.

The eccentricity ratio diagrams in Fig. 20 were of great use to those engaged in design. He and his colleagues were obtaining experience using such diagrams, and design and simplified methods of obtaining oil film thicknesses were at present being worked on. Messrs. Love, Campbell, Rafique and Martin would be presenting a paper on that subject later in the year at the Institution of Mechanical Engineers.

He was very interested in the basis on which the designer decided whether he should have a welded steel or cast iron crankcase. The reason why he raised the point was that distortions occurring within the bearing environment could considerably modify the load-carrying capacity of the bearing and, with the very thin oil films upon which the bearings now operated, the crankshaft and housing stiffness would appreciably alter the load-carrying capacity of the bearings. He wondered whether the authors had made any tests to compare the characteristic stiffness of welded steel and cast iron engines of similar design.

Correspondence

Mr. E. Rimmer (Member) wrote that his company now had over 500 000 hours experience of ATCM engines.

Due to the very short turn round time of trawlers, they expected to run 14 000-16 000 hours between engine overhauls, which covered approximately two years in service.

Between these overhaul periods, they did not expect, in fact time did not allow for, any maintenance other than routine crankcase inspection, checking and adjusting timing chains, cleaning superchargers every 8000 hours and changing fuel valves.

To date, it had been possible to run these periods without any problems. He asked Dr. Watson whether, when the b.m.e.p. was raised to 300 lb/in^2, he considered that they could still obtain this sort of period without having to remove heads for valve grinding or drawing pistons.

Mr. E. C. Younge (Member) commented, in a written contribution, that comparing the two types of engine, he was struck by the residual design features perpetuated in each engine and the apparent lack of interchange of experience between the two branches of the company. There were many examples, but he had selected two on which the authors might care to comment.

In the vee-design of the AT engine the connecting-rod configuration was entirly different from that of the Ventura and one wondered just which design was the better, or whether there were any basic underlying reasons why one design was superior for a 600 rev/min engine and another for an engine running at 1500 rev/min.

The liners on the AT engine were cooled completely at the upper flange, whereas on the Ventura engine the same flange was not cooled. Unless there was some exceptional virtue in cooling the top flange of the liners he considered the adverse feature of gas leakage past the cover-to-liner joint, passing directly into the cooling water and the cooling water passing into the cylinder, particularly when the engine was stopped, might be hard to justify. His company ran some Ruston VOX engines where this particular feature had proved to be most troublesome and which was only finally overcome by separating the gas joint and the cooling water. The long cover studs of the AT engine must certainly help to keep the gas joint tight, being able to deal with the copper joint creep in a more satisfactory manner than in previous engines.

He noted that the Ventura engine retained copper lead bearings whereas the AT engine crank pin and main bearings were of aluminium tin. Would the authors give the reasons behind the selection of these two types of bearing? His company's experience with the bearings of earlier Ruston engines showed that the greatest improvement was achieved by fitting full-flow filters to the bearing lubricating oil, rather than by changing the bearing metals. The original bearings of white metal gave untold trouble due to crazing, wiping and final failure of the large-end bearings. Copper lead bearings were substituted and these reduced the cracking, but the subsequent wear of the crank pin caused bearing failures due to pin ovality. After fitting the full-flow micro-filters his company had reverted to white-metal bearings with good results. Their experience led them to believe that the lubricating oil filtration on both the Ventura and AT engines was a vital factor in the successful running of these engines.

Mr. Howe's remarks on crankshafts brought out the importance of taking all factors into account when applying new techniques, such as were experienced when induction hardening of the crankshaft was undertaken. He thought that many had experienced crankshaft failures due to drilled oil holes, even in soft shafts, and realized the importance of carefully rounding off the edges of the hole to prevent initiation of stress cracks. Dr. Watson did not mention whether the crankshafts were hardened on the AT engines, but it would be interesting to learn whether it had been required to modify the angle of the oil holes.

He thought that all users of engines would be heartened by the amount of research and development which went into the design of modern engines and which had so ably been brought out in the paper. His own company's experience, with the AT engine in particular, had been most encouraging, giving them confidence in the modern approach to the ever increasing b.m.e.p. and speed.

Mr. H. E. Tune (Member) wrote that the paper set out in comprehensive form the complex and detailed investigations necessary in development of engine designs capable of increased performance, both in new initial approach, and extension of established designs with much experience in service. With the intermarriage of the Paxman experience in the high-speed, high specific output field with Ruston expertise in the medium-speed range, the authors' organization was well placed to meet individual requirements over the whole range of user requirement. This was an undoubted advantage in terms of available specialist advice.

It was noted that the Ventura engine employed gear train drive from the drive end of the crankshaft and thus avoided the problems associated with crankshaft free-end axials. With regard to the AT engine, he was under the impression that these drives and also attached pump drives were arranged at the free end of the crankshaft and that certain troubles were initially experienced with the gear take-offs at that position. Advice as to the measures taken to overcome this problem and as to present experience would be welcomed.

In regard to water-cooling systems, it was noted that Mr. Howe considered inhibition important and it would be helpful to have development of this point. Presumably water-system inhibition could be regarded as a penalty of the search for higher ratings, in terms of increasing heat flux values and the maintainance of adequate heat transfer margins. The writer's company had carried out considerable experimentation with cooling system inhibition over some years and could only agree that where inhibition was required, the simplest and least critical system was to be recommended. The point in rating upgrading, where treatment became necessary, was of interest. Had Mr. Howe any views on this?

On the question of piston cooling by lubricating oil jet, whilst it was recognized that, under conditions approaching 200 lb/in^2 b.m.e.p., some form of piston cooling was a requirement, it had been found that the system described, whilst simple and effective in terms of heat transfer, did introduce some complications on a class of engine in his company's service, at one stage, in that the amount of extra free lubricating oil in the running gear system contributed materially to increased lubricating oil carry-over into the combustion space, with all its attendant difficulties.

In this respect, as the machinery class was conservatively rated, with requirements specified to be met within a maximum of 140 lb/in^2, b.m.e.p., in conjunction with the engine builders, they had been able to dispense with piston-crown cooling, up to these limits, and the situation had been simply resolved. It would be helpful if Mr. Howe could advise on his experience in this direction and indicate the specific lubrication oil rates of the Ventura up to full load.

Finally, with regard to the Ventura class of engine, Mr. Howe had stated in his opening remarks that the conception of this high-speed vee-twin type of machinery was directly aimed at high-output capacity, tailored to meet difficult service limitations, presumably mainly on space.

This subject of course introduced the major controversial field of high-speed versus medium-speed machinery, in which the writer had no intention whatsoever of becoming involved here.

However, in view of the excellent development and service experience now available, this class of machinery was an obvious candidate for the unit replacement philosophy and no doubt Mr. Howe intended its claims in this direction.

It was interesting, therefore, to note his remarks concerning generator sets, towards the end of the section. It might be conveyed from these that the engine was most favourably regarded for standby applications only. No doubt however these remarks were taken out of context in this interpretation.

In connexion with Doctor Watson's comment on bedplate design features, Fig. 16 (a) depicted an "old friend".

The writer's company had a number of VGBM engines in service with this type of girder construction, some of them for many years. Failures had been experienced in individual girder units in certain cases, involving fracture of the lower girder ligaments between the cast opening in the girder and the web lower edge. Only in two per cent of the cases had fracture more seriously extended upwards into the main bearing housing proper.

In all instances repair had been effected, quite satisfactorily to date, by metal stitch methods, buttressed in the more serious cases by shaped steel strong-back clamp plates, secured to restore sectional strength by fitted bolts.

In no case had bearing troubles, as such, been experienced prior to discovery of the fractures, which had been disclosed purely by routine inspection methods. There was, in fact, some evidence, in some instances, that the fracturing of the ligaments had been present for some time prior to notice.

The type of main bearing arrangement in this design provided for clamping of the liner type main bearing upper and lower shells by a main bearing keep, which itself abutted solidly to the girder top plate on hardening down. Correct tensioning of the bearing studs was intended by the provision of an initial clearance between the main bearing keep and the girder top plate, usually 0·006 in to 0·008 in prior to hardening down.

This arrangement, very adequate in theory, was very difficult to achieve satisfactorily in service and virtually impossible over a long term basis, where datum surfaces suffered wear and tear. At the best, a good deal of effort and care were required to be expended, under difficult conditions, to achieve the required standards and, in many cases in practice, achievement could only be regarded as less than satisfactory.

On opening out main bearings, for routine overhaul prior to discovery of the fractures, it had been noted that some relative movement of bearing keep and shells had been taking place.

The investigation at the time, carried out with Dr. Watson's company, pinpointed the weakness in girder design described by Dr. Watson. There was no doubt that the flexing of the girder under load had been initially responsible for the loosening of the main bearing arrangement.

In the writer's opinion, however, the sensitivity of this main bearing arrangement to disturbing influences, together with the difficulties of restoring and maintaining basic adjustment conditions, following some-time running under slack conditions, contributed materially to the problem overall and played no little part in hastening ultimate girder fracture.

On this basis, where applicable, in the instances quoted, and as discussed with the engine builders, repair of the fractured girder was supplemented by modification of the main bearing assembly. The keep was machined to provide solid clamping of the main bearing shells, with a positive clearance between the keep and girder top plate, the bolts being loaded by stretch gauge. It was considered that this arrangement provided a more positive method of achieving satisfactory loading margins sufficient to withstand the flexing influences and to avoid bearing fringe parting load conditions.

Fortunately, the engines were moderately rated and, as noted, so far these repairs had been satisfactory. Subject to experience it was intended to continue this modification on a progressive basis throughout the machines concerned.

Mr. J. H. Gilbertson (Associate Member) wrote that both authors had expressed the need to maintain complete reliability, together with accessibility, to assist maintenance and one could see that the development of the medium and high-speed oil engines could be compared with the car industry, where the engines were becoming smaller and smaller and the accessories larger and larger.

One would also agree with the comments that design should be kept simple, but, as the engine output had increased by approximately 50 per cent within the same overall dimensions and weight of its predecessors, it had become a fully developed engine and much greater reliability was placed upon the accessories.

Referring in particular to the charge air coolers which played an important part in the development of the engine, one would like to see consideration given to ensuring that adequate space was available for installation and access for cleaning and maintenance. In addition, one must also accept the fact that such equipment must be capable of maintaining its performance over a long period, hence the access for cleaning *in situ* was most essential.

He agreed with the authors that a selected water treatment should be one which was not too sensitive to the quantity used. Very often the control carried out on board ship was not adequate and over-addition of reagent could promote attack on copper-based alloy heat exchangers, just as weak solutions could promote attack of the system in general. Also, different alloys theoretically required different concentrations of inhibitor for complete protection and, because cooling systems were of mixed alloys, a balance had to be reached. Variations, which occurred due to use and degradation of the inhibitor, could, therefore, be more significant in sensitive water treatments.

In the past, heat exchangers had proved to be very reliable and had not generally suffered from corrosion from the fresh water side. Because of their reliability, the copper-based alloys used in heat exchangers had very often not been considered to the extent to which they deserved. As a result, there had been an increasing number of failures by corrosive attack in conditions where no attack would have occurred if the water treatment had not been used. It was thought, therefore, that the copper-based alloys and heat exchangers, used in the cooling systems, should deserve more attention when considering the application of water treatments. The authors had mentioned steel, aluminium and rubber, but not copper-based alloys.

Mr. J. J. Stennett wrote that each of the engines described illustrated interesting features with regard to cooling water circulation.

The Paxman engine had its water inlet manifold at the upper end of the jacket in a position which his company had found beneficial in obtaining good circulation and hence uniform temperature distribution around the top of the liner. The AT engine had its water outlet in the more usual position at the lower end of the liner and achieved the same result by the use of multi-port water transfer from the jacket around the liner flange.

For a fairly modest expenditure his company had produced life-size transparent models of cylinder heads and jackets to enable the flow of cooling water to be studied and these techniques had revealed a surprising number of air pockets and stagnant regions, particularly in cylinder heads when inclined at the vee-angle.

By minor modifications to the model it had been possible to eliminate these undesirable features and ensure good circulation around the important injector and valve seat regions before committing the final design to the pattern shops.

Whilst the position of the inlet manifold on the Paxman obviously fitted in conveniently with the outline of the engine, he would be interested to know if this was the sole reason for its position, or whether the question of water circulation was involved.

In Fig. 5, the outside diameter of the liner flange appeared to be screwed. Was this for the application of a liner withdrawing tool?

Presumably the figure 0·990, in Fig. 9, should be a specific fuel consumption of 0·390.

Fig. 10 was most interesting and, from an engine manufacturer's point of view, would be even more so if the curve of cost per horsepower were split up into engine builders' content and suppliers' content (i.e. bought-out turbochargers, bearings, cooling equipment etc.).

Were the captions in Fig. 20 correct? Comparison of these diagrams and similar diagrams for engines of his company's manufacture suggested that the fitting of balance weights had had an opposite effect on the eccentricity ratio diagram. He would have expected the broader diagrams 1, 4 and 5 to have been without balance weights and diagrams 2 and 3 to have been with balance weights.

Authors' Replies

PART I—Reply by A. G. HOWE, M.B.E.

Mr. Howe said that Mr. Calderwood had mentioned the question of the balancing of the six and eight-cylinder engines. There were problems. The method used at the moment was a countershaft arrangement. The problems were much more intense with the six-cylinder engine than with the eight-cylinder engine and, in consequence of that, it was something that had been looked at extremely carefully, because it had a serious effect on the immediate surroundings as well as the engine.

Fig. 1 showed a cross-section of the engine. Mr. Calderwood had mentioned the joining of the relatively thin side plates to a very thick top deck. The top deck was illustrated in the figure and its edges had been hollowed out so that they were relatively the same thickness as the side walls of the metal block. The weld could be seen on the right. So the two thicknesses were just about the same. Mr. Calderwood had also mentioned the relative thickness at the base and he had a point there, but it did not alter the fact that no problems were experienced.

Mr. Calderwood was obviously right in what he said about the period of stress relieving. The period of two hours was the time after the appropriate temperature had been reached. This temperature was then retained for that period. It was then allowed to cool gradually, but under control.

However, it was important to remember that each section of the crankcase had been separately stress relieved. If one read the paper carefully, one would find that the period mentioned was really to deal with the joining welds after the three pieces had been joined together.

Each section was stress relieved for a longer period, possibly 8-10 hours, so each piece would experience a long period of stress relieving before being joined together. Therefore, the finishing figure of two hours covered the whole structure.

The plates were ultrasonically tested for laminations. Plates sometimes came from the manufacturers with laminations; as one could not see them, it was horrible after having cut a plate and starting to weld, to see how the plate opened up along its axis. So all plates were tested for laminations.

With regard to the question of the oil hole in the crankshaft, the hole went into the bore of the hollow pin, which bore had to be plugged at each end.

With regard to the question of the cylinder head and pistons, Mr. Howe was sorry that he had not made any specific mention of the cylinder head in the paper, although he had done so in his opening remarks. Fig. 5 showed an aluminium head. Generally speaking, the problems connected with expansion of the block were not very serious, but as the ratings went up certain problems occurred in connexion with the strength of the head at the temperatures at which it was working, and, for that reason, cast-iron heads were coming in at the higher ratings.

Mr. Calderwood had asked about the material of the seats. They were in aluminium bronze. It was very interesting to remember that the first effort by the author's company to manufacture aluminium cylinder heads was in 1938, when valve seats in that material were introduced, screwed in exactly as at present. Little reason had been found for modifying them. They had given very good service over that long period.

Regarding the piston temperature, up to now, at the ratings that the author's company had adopted, the piston cooling had been by means of a jet on to the underside of the head. The amount of experience and knowledge at the present time (the author was six months out of date with some of these detailed points now that he was retired) in regard to other types of cooling of internal passages and so on was relatively limited. However, Mr. French mentioned the question of a two-piece piston. Mr. Howe had to confess that he liked the method of using a two-piece piston so as to incorporate a kind of controlled cooling. If he were designing he thought he might well choose this method possibly in conjunction with a variable compression piston, because it also answered one or two other points that had been raised in the discussion, such as the desirability of having a piston which would be stronger, more heat resistant, and so on. These things seemed to add up.

Mr. Lowe had mentioned the question of lubricating oil temperature. The water temperatures usually adopted on the Ventura engine were certainly in the range of about 175°F (80°C) in, and about 185°F (85°C) or a little more out, so that with a great deal of water being circulated there was only about 10 deg F (5 deg C) temperature rise. The lubricating oil would be about 15 deg F (8 deg C) higher than that—around the 200°F (93°C) mark. It had been found from experience that this was very beneficial, which was rather contrary to what Mr. Lowe had implied. It had been found that keeping the oil at a high temperature reduced the wear in the cylinder. Much of the wear could take place at low temperatures and certainly on starting. Consequently, if the oil was made hot quickly the wear was prevented.

He had dealt with the two-part pistons to which Mr. French had referred. Mr. French was wise in drawing attention to the question of additives in the cooling water. This was something that must be treated very carefully and with expert knowledge. One tended to think that it was only water, but water seemed to vary in many cases. It needed as much careful attention as possibly lubricating oil because fun and games went on inside those spaces one could not get at, possibly with the deposition of scale and also corrosion and erosion or cavitation on the outside of the liner. Some of the additives were helpful in that respect, but must be carefully used.

Captain Stewart had referred to the welded structure and the stress levels and also the figure of two tons. A great deal had been written about welded engine structures. Mr. Howe had to confess that he had tended not to say too much about these because so much had been written recently. One tended either to corroborate or contradict or simply extend what had already been said. Two tons was a well accepted figure. However, taking the matter broadly, the problem seemed to arise in the fillet welds of welded structures. If one could find a way of ensuring that fillet welds were either dispensed with entirely or made much more reliable, then this would be a very valuable contribution. As had been mentioned in the introduction, efforts were being made in that direction by means of a greater use of steel castings whereby as many fillet welds as possible could be eliminated by their being incorporated in the steel castings. Butt welds were easily made, and being fairly readily examined, were, in consequence, much more reliable.

Captain Stewart would be interested to know that the Perspex model that was made subsequently to the earliest welded frames confirmed, very closely, the stresses that had been found, or calculated to exist, in the actual engine frame.

There had been reference to tin/aluminium bearings. He was not very close to the results obtained with tin/aluminium bearings in the engine, and would prefer not to comment. He had already mentioned the cylinder liner of steel and that aluminium cylinder heads might well be replaced by cast iron at the higher ratings.

The question about performance comprised about six questions in one, and he proposed to answer in writing.

Mr. Attwood had made an interesting contribution on the subject of the need to determine the right power for the job—so as to have reserve as and when one was needed. He would not elaborate on this question; it was the owner's or the ship designer's responsibility to determine at what rating he would run his engine and what he would allow for fouling and so on. The high-speed engine described in the paper was intended rather for special applications and not for general applications for propulsion. It was obviously for those cases where low weight and great compactness were essential.

Dr. Mansfield had given food for thought. He had put forward a challenge in respect of some of the engines that existed overseas. He thought that they could thank Dr. Mansfield for having brought these matters to their attention. Dr. Mansfield could be satisfied that Mr. Howe and his colleagues had not missed this point.

A number of speakers had asked about the general construction of the engine shown in Fig. 2. Mr. Howe had explained in the paper that this general form had been developed by his firm through the years from about 1936. Therefore, they thought they knew something about it. It was surprising how much development, time and money could be spent in making changes, just for the sake of making changes.

He had been asked what would happen in the future if there were big jumps in b.m.e.p. It might well be that side-by-side rods would have to come into the picture. He had described how difficult it was to accommodate the right stiffness for the bearings, the big-end bearing particularly, within the confines of the crankcase.

He had not been sure that he understood Mr. Beard's remarks about the bolt and the big nut, but if Mr. Beard was referring to what appeared to be the washer under the bolt head, it was not a washer but part of the bolt head construction, and it had a special spanner shape above the plain section so as to ensure that a spanner could be got on to it from the crankcase door.

Mr. New had asked a very useful question about when, or how, or why a choice was made of a welded steel engine, or a cast-iron engine. His organization had adopted it largely on the basis of weight and reliability and disposition of material. As to the question of distortion, some interesting work had been done on that subject, upon which Dr. Watson would comment in his reply.

In reply to the written contributions Mr. Howe first referred to Mr. Younge's comments relative to the design of large end bearings. Mr. Howe had already dealt with this point in the paper on the basis of continuing to use a method which was understood and which he knew could be made to work without development, at the same time appreciating its limitations. The juxtaposition of the crankcase, liners and crankshafts in the Ventura also prevented any major change in shape. Contrary to Mr. Younge's thoughts there was a very full interchange of experience between the two companies.

Referring to bearing material, here again with the Ventura, past experience determined the choice of material, thereby eliminating needless time and trouble on further development.

In answer to Mr. Tune's questions regarding water-cooling systems, Mr. Howe said that the treatment of circulating water was intended to cover a number of features, probably the most important being that of ensuring an inert fluid from the corrosion standpoint, with a *p*H value of 7·5; the reason for this was obvious. Another reason could be to reduce surface tension in the water with a view to reducing or eliminating the likelihood of vibration erosion on the outside of cylinder liners. For this latter purpose soluble oil was used as an additive, but care was taken to ensure that joints and hoses were not effected. These problems could exist on most engines but were likely to show up more on higher speed engines, but not necessarily as a function of rating.

It was fair to suppose that with piston cooling there would be an increase in consumption of lubricating oil, but it seemed that a figure of one per cent of the fuel burnt was maintained bearing in mind that piston cooling was needed at the higher ratings with a consequent greater output and quantity of fuel burnt.

Mr. Tune had taken the author's remarks on the application of the Ventura engine somewhat out of context in that there were already many applications operating on a full-time duty basis and the engine was designed and intended for such applications. He agreed that it was a suitable engine for standby duties, but it would be an uneconomic proposition if it were applied only for this purpose.

Replying to Mr. Stennett, Mr. Howe confirmed that in Fig. 5 the outside of the liner flange was screwed to take a removal tool and that in Fig. 9 the fuel consumption figure of 0·990 should read 0·390.

Captain Stewart's question regarding general performance and consumptions could be answered in a number of ways, but generally some measure of compromise had to be used when applying an engine. It was a well known fact that the torque curve of an engine, although fairly flat, peaked at a given point. It could be said that this was the point at which full load or top rating and speed should be fixed. This could be a waste in the utilization of speed, however, as while the torque curve might be dropping at the top speed allocated by the designers, it frequently happened that generators, gearboxes and other driven equipment were the more economically applied at the higher speeds.

The reasons for the shape of the torque curve were well known, including not only mechanical loadings but pumping losses as well.

PART II—Reply by H. WATSON, B.Sc., Ph.D.

Dr. Watson accepted Mr. Calderwood's point concerning prices of engines in the early 30s. Fig. 10 in the paper had been compiled from available records within his company and it had to be admitted that the first point on the price curve was suspect. In reply to Mr. Stennett, from the company's records it was only possible to determine an approximate division between engine builders' content and bought-out equipment in the total price make up. The prices shown in the graph were for the base engine only, i.e. without any ancillary equipment. Indications were that at the present time Ruston and Hornsby were responsible for 80 per cent of the total content in the AT engine. It was likely that in 1930 this proportion was approximately 90 per cent.

The aluminium tin bearings on the AT engine operated on a soft crankshaft. The material had a Brinell hardness of 250 which represented the limiting hardness for the method of manufacture adopted. Individual engines had now operated for periods in excess of 45 000 hours without any problems associated with shaft wear.

Mr. Younge had suggested that lubricating oil filtration was a vital factor in the successful running of the AT engine and Dr. Watson was in full agreement with this. Full flow filters capable of achieving 15 microns filtration were necessary with aluminium tin and copper lead bearings. While the AT engine had been fitted with aluminium tin bearings through all stages in development, other Ruston engines had been fitted with copper lead bearings. Choice between the two alternatives was generally dependent on individual designers preferences. Aluminium tin bearings were marginally cheaper. Under ideal conditions both materials gave satisfactory performance on medium-speed engines. Copper lead bearings with overlay plate would appear to cope with minor misalignment and dirt more readily than aluminium tin.

Mr. Younge's comments on copper lead bearings versus white-metal bearings were noted. Dr. Watson wondered if Mr. Younge had referred to an older type of engine and in this case perhaps the comparison was a little unfair, although Mr. Younge's comments in this case were valid. The fitting of copper lead bearings on the older engine without attention to filtration would produce high wear. Reversion to white-metal bearings and the ability to operate with these bearings however, resulted from a change of bearing and housing design.

In the AT engine design a certain amount of leakage could occur from the fuel injection system. The engine incorporated a separate lubrication system for the valve operating gear and the cylinder head. A pump, relief valve and supply tank were built into the cam box, and by this system crankcase dilution was eliminated.

With regard to Mr. Calderwood's comment on piston cooling it should be remembered that the cross-section shown in Fig. 12 was for an engine operating at 200 b.m.e.p. The same type of piston cooling need not necessarily be adopted as uprating proceeded.

In answer to Mr. Calderwood, Mr. French, Dr. Mansfield and Mr. Tune, the type of piston cooling as shown in Fig. 12 had proved to be very effective. Alternative designs of multi-piece pistons having controlled oil flow were incorporated in other Ruston engines. Temperature profile achieved with the more complex system had been measured and shown to be only marginally better than a simple design.

For four-stroke engines operating at 300 b.m.e.p. a separate flame plate head construction was likely to be adopted and this would incorporate drilled passages as adopted for the Ruston AO engine. Ability of this design to accept high thermal loading had been proved. Experimental results indicated that a separate flame plate with cooled passages immediately behind the valve seat was capable of operating with a thermal flux some 70 per cent higher than the double deck cast iron head.

The low stress level in welded structures, providing the structure was designed correctly, was not an embarrassment. The main function of the engine frame was to maintain bearing alignment under the action of firing loads and support reactions. In a correctly designed structure the required stiffness did imply a low stress which was compatible with the weld requirements. The problem with many welded structures arose from the introduction of stress raisers in the design and where these coincided with a weld an embarrassing situation occurred. Where a uniform stress system was achieved the stress level required to maintain a satisfactory deflexion would in all cases be below the fatigue limit.

Mr. New had questioned the reasoning behind welded construction, and Mr. Stennett had indicated some doubt as to suitability. In general, in adopting a welded construction the designer was attempting to achieve a compact engine and to take full advantage of the higher Young's modulus afforded by the steel. In isolated cases the designer could be misguided in considering welding to be a cheap form of construction, but only where adequate stiffness could be achieved by cast iron. Experience indicated that bearing distortion did considerably lower the load capacity of main bearings. The highly rated engine did in fact benefit from the generally shorter bearing in this respect.

Tests had been undertaken by Ruston and Hornsby to consider the characteristics of welded steel and cast iron engines but these had not been of similar design. Investigation of successful designs exhibited the same order of bearing deflexion under the firing load for which the engine was designed and this would appear to be in agreement with Mr. New's contribution.

Mr. French's extrapolation of Fig. 10 to 315 b.m.e.p. and 2250 ft/min piston speed by 1970, implied that Fig. 10 was possibly over optimistic. At the lower end of the medium speed engine range 300 lb/in^2 m.e.p. and 2000 ft/min piston speed by 1970 could certainly be visualized and within this range a continued cost reduction should be possible.

It would appear logical to assume that there must be some limiting rating at which increased sophistication produced an increase in specific cost. This was likely to be at the rating where a two stage turbocharger was required. It could be argued that when this point was reached a two-stroke cycle then became a more economic proposition than the four-stroke cycle, even for engines operating on distillate fuel.

Mr. Attwood and Mr. Rimmer had discussed maximum ratings in their contribution. The optimum rating for any duty always tended to be a thorny problem between customer and engine builder. The engine builder sold a maximum continuous output which the customer could accept for his required duty. Engine testing in Ruston and Hornsby was in general undertaken at 10-15 per cent above the maximum continuous rating in order to allow for deterioration of engine components between infrequent overhauls and to ensure that contract rating could be maintained indefinitely.

It should be borne in mind that the present high ratings were achieved in many cases with a higher degree of reliability than older engines achieved under naturally aspirated conditions. Mr. Rimmer's comments on the periods between overhauls certainly provided evidence for this and in answering his questions specifically, design and development would be such that when the m.e.p. was raised to 300 the same degree of reliability would be maintained.

Mr. Lowe had commented on the high compression ratio on the AT engine and suggested that thermal efficiency of the engine might well be improved by a lower compression ratio for the same limit on maximum cylinder pressure. At the increased loads contemplated, 13 : 1 compression ratio would certainly be too high and more recent calculations had shown that with a limiting firing pressure of 1500 lb/in^2, adjustments of engine parameters, including a reduction in the compression ratio, could result in an improved engine efficiency.

Considering Mr. Lowe's second point, the underslung construction adopted for the AT vee-engine was definitely more economical than the bedplate structure. Based on results from another Ruston engine, the vee-engine in underslung form did give a saving in material of some 25 per cent by comparison with the bedplate version. The saving was not so great in the case of the in-line engine. Figs. 11 and 12 did not unfortunately highlight this feature. In effect the in-line version of the AT did have a bedplate, although a cross-section of this tended more towards the underslung design than the conventional type of bedplate structure.

Mr. Tune's written contribution pointed out some of the difficulties associated with the cast bedplate. One peculiar feature of the failures he mentioned was that the bearings functioned satisfactorily after fracture of the girder. The authors company had expended a considerable degree of effort in attempting to ascertain the dynamic stress system in a bedplate structure at the joint mentioned by Mr. Tune without finding any stress which could explain failure. Mr. Tune's comments on the main bearing arrangement on older type engines were of interest and any criticism made by him was fully justified. The modifications which had been adopted for the main bearing assembly by Mr. Tune's company referred to thick walled bearings and did involve the principle adopted for modern designs incorporating thin steel shell bearings. In the later design the nip on the thin walled shells could be controlled and was achieved consistently. The main bearing bolts were stretched to a predetermined value, and compression of the bearing shell required only a small proportion of the bolting load.

Dr. Mansfield and Mr. Younge had both questioned the reasoning behind the different connecting rod arrangements adopted for the AT and the Ventura engines. The final choice depended mainly on individual preference. Both systems had features which required careful design. There would appear to be nothing inherent in either system which made it more suitable for one engine than the other, except that the fork and blade arrangement did allow a shorter connecting rod length to be achieved. This feature would affect engine height and was of greater significance in the Ventura than the AT.

Mr. Beard and Mr. New had raised doubts as to the validity of the captions and Figs. 18 and 19. Fig. 18 did of course refer to main bearings and, as Mr. New had pointed out, specifically to the centre main on the six-cylinder four-stroke engine. Fig. 19 referred to main bearings and not to a large end.

Mr. New had suggested an alternative reason for the bearing failures and the cure indicated in Fig. 18. Obviously, the change in load pattern achieved by adding balance weights must have some effect on the durability of the bearing. Experience on other engines however, did support the argument given in the paper, i.e. that local heating of the shaft was the main contributing factor to the type of bearing failure discussed.

Mr. Stennett's comments on liner cooling were of interest and Mr. Younge's point about gas leakage was valid. On the AT engine the gas joint and the water joint were separate and did not rely on close tolerance in manufacture. Intuitively the fully cooled liner should produce a system capable of higher thermal loading than an uncooled liner but unfortunately the stress system involved was extremely complex. As rating increased liner cooling over the complete length became a virtual necessity.

The camshaft drive on the AT engine was taken from the drive end of the crankshaft. Early AT engines were placed in service with an auxiliary drive for pumps only at the forward end. This system employed a rubber coupling to provide torsional isolation between crankshaft and the pumps. The rubber was also intended to maintain concentricity of the crankshaft and pump system and this represented a weakness. Radial gear movements occurred with the original system, resulting in pinion failures on the driven auxiliaries. The problem was overcome on early AT engines by redesigning the system to provide positive location of the axis of the driving gear relative to the driven pinions and with this system flexibility between crankshaft and the auxiliary drive was achieved by a large diameter coil spring. Further uprating which had been applied to the AT engine has increased power requirements of the various auxiliaries and had in some measure made the multi-drive impracticable. The latest 200 m.e.p., 600 rev/min engines had lubricating oil and fresh water pumps mounted at the drive end of the engine. In recent installations separate electric drive was adopted for additional pumps.

Dr. Mansfield had raised the question of size of auxiliaries in relation to the engine and it was gratifying to see from Mr. Gilbertson's contribution that the ancillary equipment manufacturers were backing up the engine builders in providing more compact equipment.

Minutes of Proceedings of the Ordinary Meeting Held at the Memorial Building on Tuesday, 11th April 1967

An Ordinary Meeting was held by the Institute on Tuesday, 11th April 1967, at 5.30 p.m., when a paper entitled "Design Methods and Development of Medium and High-speed Oil Engines" by A. G. Howe, M.B.E., C.Eng. (Member) and H. Watson, B.Sc., Ph.D., was presented by the authors and discussed.

Mr. R. R. Strachan, C.B.E. (Chairman of Council) was in the Chair and one hundred and four members and guests were present.

Eight speakers took part in the discussion which followed.

The Chairman proposed a vote of thanks to the authors which was greeted by acclamation.

The meeting ended at 7.30 p.m.