The Holmes-Werkspoor Engine: Building, Testing and in Service*

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INTRODUCTION

The engine which forms the subject matter of this paper is of fairly conventional design of the four-stroke cycle trunk piston type with a cylinder bore of 390 mm. (15·3 in.) and a stroke of 680 mm. (26·8 in.). The author has not, therefore set out to present any new developments in marine Diesel engine practice but has dealt more with the practical aspects of the subject.

This engine has been built for a number of years on the manufacture in this country; the first engine was completed in June 1957 and went into service early in 1958.

In the naturally aspirated and pressure charged forms, the six, eight and ten-cylinder units cover a power range of 600 to 2,100 b.h.p. at speeds of 200 to 275 r.p.m. An eight-cylinder pressure charged engine developing 1,400 b.h.p. at 245 r.p.m. is shown in Fig. 1. Except for exceptional circumstances, the pressure charged engine offers distinct advantages and accordingly attention is devoted to this type only.

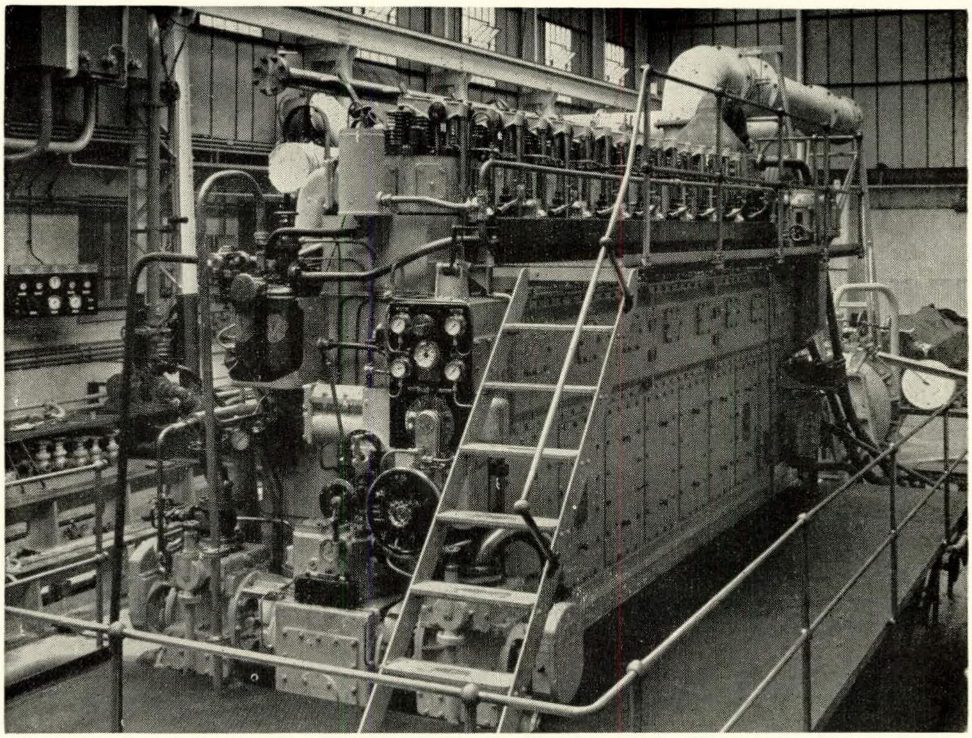

FIG. 1—*Eight-cylinder engine, 1,400 b.h.p. at 245 r.p.m.*

Continent and was designed as a marine propulsion unit of the direct reversing type suitable for the powering of coasters, trawlers and the like. In 1955 the company with which the author is associated entered into a licence agreement for its

At the present stage of development the exhaust gas driven turboblowers fitted to these engines have a compressor pressure ratio of approximately 1·35 which allows a power increase of 50 per cent of the naturally aspirated power. Under these conditions a saving in weight of about 16 per cent is obtained by pressure charging and the initial cost per horse power is reduced.

A limit is placed on the degree of pressure charging of an existing design by the allowable maximum cylinder pressure

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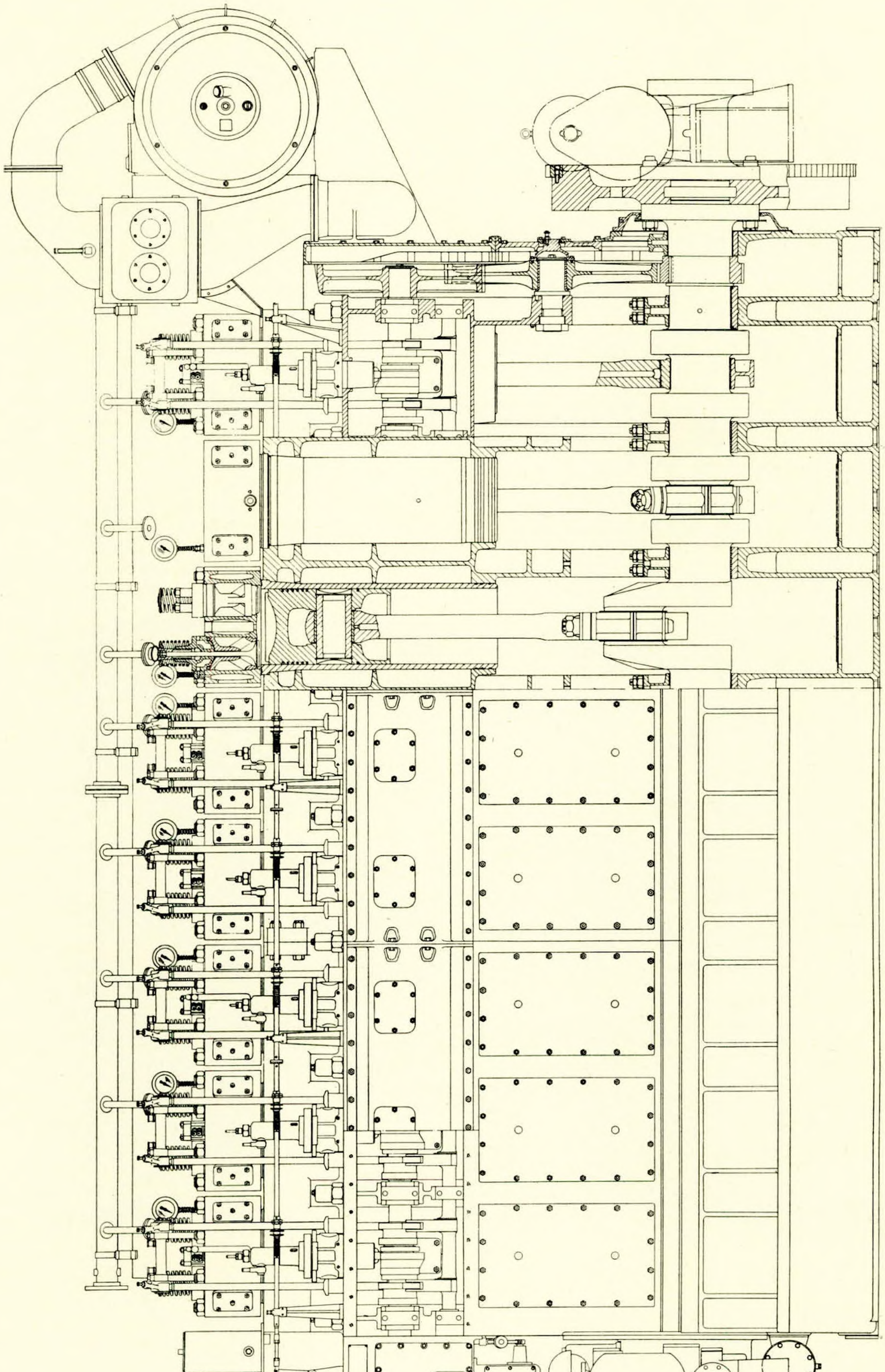

FIG. 2—*Eight-cylinder pressure charged engine, 390-mm. bore, 680-mm. stroke*

and rate of increase of pressure, as an increase in charge pressure leads to an increase in both these factors. This is necessary because the compression ratio of the engine has to be such as to allow for easy starting, when no advantage is obtained from the turboblower. The present compression ratio of the engine is more than adequate for starting and there is a margin for further increase in pressure charging.

As indicated by the title, the present paper is divided into three sections. The methods of construction and operation of the various parts of the engine are dealt with fairly comprehensively in the first part but it has not been possible to discuss in detail such parts as the governor, fuel pumps, which are of the normal "jerk" type, the turbocharger and other items. In addition to considering test bed trial results, some details of the equipment employed and methods of testing are included. Service performance data are of necessity rather limited but some comments on this subject are included in the concluding section.

BUILDING

Main Castings

The main structural components of the engine, namely, the cylinder block and crankcase and the bedplate are of cast iron. For large, slow running engines the use of steel fabrications for these parts is almost universal but for smaller engines cast iron is more economical; the engine at present under consideration is probably approaching the limit in size for cast iron construction in modern practice.

The main castings are made in dry sand moulds, the cores being of oil sand. Bedplates are normally cast in one piece, except those for ten-cylinder engines, which are jointed in the centre. An eight-cylinder engine bedplate has a cast weight of 9·5 tons and requires 12 tons of metal, which is poured from two ladles simultaneously in approximately 90 sec. The sump of the bedplate is formed in the top of the mould and the molten metal enters through a number of ingates at the bottom.

In the case of cylinder entablatures it has been found expedient, both from the point of view of casting and machining, to divide these parts into two or three sections, depending on the number of cylinders. As will be seen from Figs. 2 and 3 the main frame includes the cylinder jackets, air manifold and camshaft casings. These components are cast with the cylinder centre line horizontal and the air manifold to the top, the casting procedure being similar to that adopted for bedplates. After pouring, the casting is left in the mould for almost ten days to ensure slow cooling and prevent cracking.

The iron used for the castings just described has a tensile strength of approximately 12 tons per sq. in. and the analysis is closely controlled within the following limits: carbon 3·4—3·5 per cent; silicon 1·9—2·1 per cent; manganese 0·5—0·7 per cent; phosphorus 0·5—0·6 per cent. The relatively high silicon content is necessary to give soundness and fluidity to the iron in the thinner sections and the phosphorus must be maintained at as low a value as practicable to give strength and toughness. The phosphorus content of the iron has an appreciable effect on the cost owing to the scarcity of low phosphorus ores in this country and therefore it is kept to the medium value indicated.

Cylinder heads and other parts exposed to combustion conditions are made from a higher grade iron with a tensile strength of at least 17 tons per sq. in. In this case a typical analysis is: carbon 3·2 per cent; silicon 1·7 per cent; manganese 1·0—1·2 per cent; phosphorus, less than 0·3 per cent. This ratio of carbon to silicon gives a pearlitic structure which has high strength and is readily machinable. It will be noted that the manganese content is increased compared with the iron used for the large castings and the phosphorus is kept to a low value to increase both the strength and resistance to thermal shock.

Cylinder head castings are stress relieved after rough machining by slowly heating to a temperature of 450 deg. C (842 deg. F) and cooling in a closed furnace. Large castings are allowed to stand for as long as possible after rough machining to relieve internal stresses which would otherwise cause distortion of the finished component. In this connexion it might be mentioned that it is common practice to "weather" large castings in the as-cast condition, but it is the author's opinion that allowing the casting to rest for a few days after the initial rough machining is more effective than months of weathering.

Machining of large flat surfaces is usually carried out on a planer but where the cut is intermittent it has been found to be more economical to mill the surfaces owing to the fact that the depth of cut of a planing tool is limited under these conditions.

Wherever necessary to allow for interchangeability and to reduce individual fitting, parts are machined to close tolerances. In the boring of cylinder blocks and the main bearing pockets in bedplates, the principle of using a machine to provide only the drive and feed to the boring bar has been adopted. By supporting the bar in suitable fixtures the required dimensional accuracy is more readily obtained and for this type of work micro-boring tools are used to great advantage in final machining.

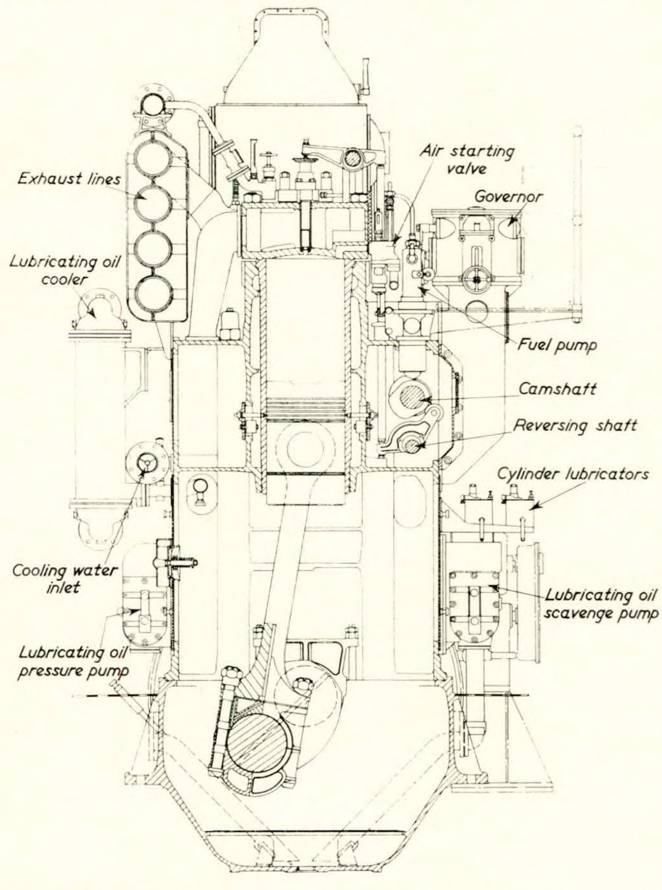

FIG. 3—*Section through pressure charged engine, 390-mm. bore, 680-mm. stroke (looking aft)*

The cylinder liners are centrifugally cast, which gives a close grain and fine structure. An unalloyed iron having an analysis similar to that of the cylinder head material is used; this has a tensile strength of approximately 17 tons per sq. in. and a Brinell Hardness number between 200 and 230. As will be seen from Figs. 2 and 3 the liner is held in place by the cylinder head, the top of the liner being an interference fit in the cylinder block. At the bottom the fit may vary from a slight interference to a clearance to allow expansion downwards. Three rubber rings seal the joint at this position and prevent leakage from the water jacket.

After final boring of the liners they are honed to eliminate

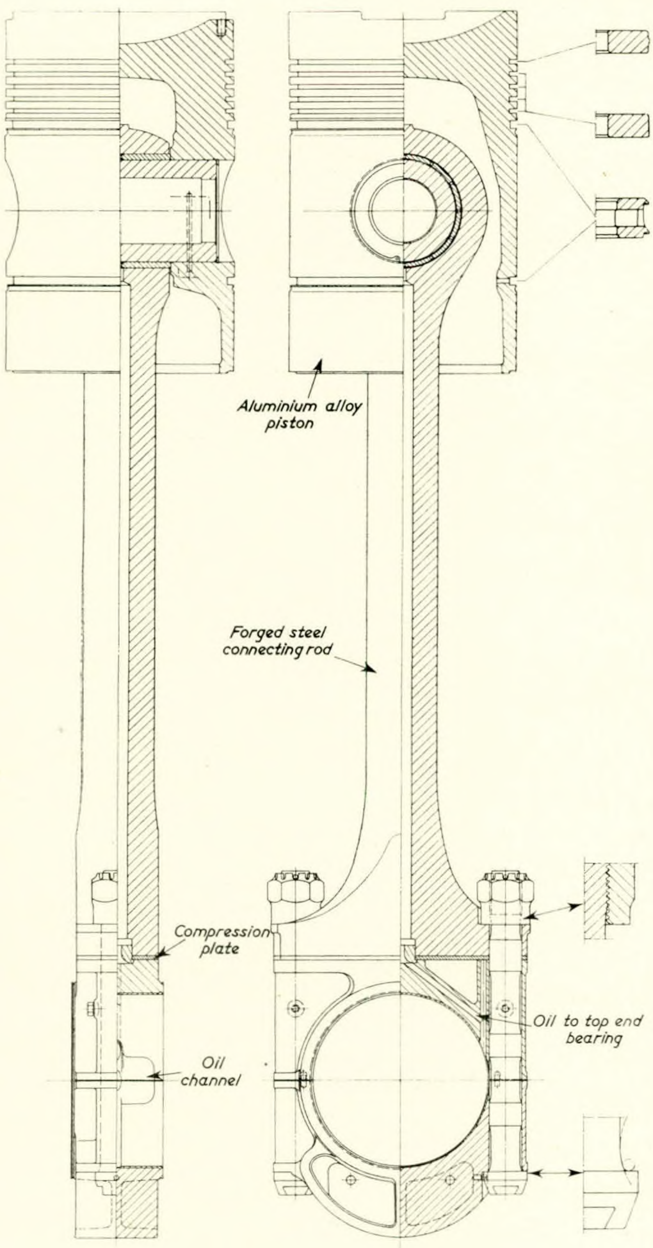

FIG. 4—*Running gear*

the "waviness" of the surface which results from the boring process. Differences of opinion exist on the ideal surface finish for the bore of liners, but it has been found that a finish with a C.L.A. value* of about 60 micro in. gives a satisfactory result and aids the retention of a surface oil film during the initial running-in of the engine. To facilitate further the running-in of the engine the liners are subjected to a phosphate treatment which produces a very thin crystalline coating of iron manganese phosphate. This acts as an extreme pressure lubricant to prevent scuffing and also has a high degree of oil absorption which prevents break-down of the oil film.

* The quality of surface finish is expressed in terms of the centre line average (C.L.A.) value. This index number is defined as the average value in micro inches, of the departures of the surface profile from its centre line, whether above or below it, throughout the prescribed sampling length. (1 micro in.= 0·000001 in.)

Two oil holes for independent cylinder lubrication are drilled through the liner on the athwartships centre line at a position corresponding to the level of the top piston ring when the piston is at the bottom of its stroke. Oil is supplied to these points by ratchet driven lubricators through quills which pass through the water space.

Running Gear

Details of the running gear are shown in Fig. 4. The connecting rod, in common with most steel components of any size, is made from forged steel with a tensile strength within the range of 33 to 37 tons per sq. in. A plain carbon steel with a carbon content of about 0·3 per cent is used; this is somewhat higher than that of the 28-32 tons tensile steel which is normally used in heavy marine engine practice.

The production of the connecting rod calls for little comment apart from the fact that the gudgeon pin bearing eye is chrome hardened to withstand the arduous conditions imposed by the oscillating motion and high bearing pressure. After pregrinding of the bore of the eye chromium is electrolytically deposited over the surface to a thickness of approximately 15/1000 in. The bore is then finally ground, giving a finished thickness of hard chrome of 7/1000 in. The gudgeon pin, which is of the fully floating type, is treated similarly on its outside diameter over the whole of its length. Particular care is taken to obtain a high quality surface finish of the gudgeon pin with a C.L.A. value of not more than 3 micro in.

The gudgeon pin bush, as will be seen from Fig. 4 is a plain bronze bush with only circumferential oil grooves in the centre to convey the oil to the gudgeon pin from the supply hole which is drilled up the centre of the connecting rod. In order to obtain the required physical properties, these bushes are centrifugally cast in bronze with a high tin content. For those who are unfamiliar with the process of centrifugal casting, it might be mentioned that the metal is poured into a cylindrical steel mould which in this case rotates about a horizontal axis at a speed of 1,000 r.p.m. In this way a very dense and fine grained casting is produced with a minimum of excess metal for machining and enhanced physical properties compared with a normal sand casting. All finished bushes are tested for hardness at a number of points on the surface and are rejected if the value falls below 110 B.H.N.; a normal casting in the metal used would have a hardness of 80 B.H.N.

The bottom end bearing is made from cast steel with a minimum tensile strength of 32 tons per sq. in. and is lined with tin base whitemetal having a finished thickness of 3 mm. (approximately 1/8 in.). In order to obtain a good bond between the whitemetal and steel, dovetailing is eliminated, but for satisfactory results the whitemetalling of the bearings has to be carried out under closely controlled conditions. It has been found that the best results are obtained when a high tinning temperature of 400 deg. C (752 deg. F) is adopted.[1] The shell is then reheated and retinned at 250 deg. C (482 deg. F.), this temperature being maintained whilst the whitemetal is cast. Rapid cooling from the shell side is applied to obtain the highest bond strength. After final machining the adhesion of the lining is checked with the aid of an ultrasonic crack detector.

As far as is known this technique has not previously been used in this application. It forms a rapid and very reliable indication of the continuity of bonding between the lining and shell, the principle of operation being that a high frequency sound wave is reflected back to a pick-up if there is a discontinuity in the bond between the two materials. This is readily observed on the screen of the cathode ray tube.

Bottom end bearing bolts are also components which receive special attention in manufacture. Although, in theory, these bolts are not subjected to fluctuating stresses of any great magnitude it is not unknown for fatigue fractures of running-gear bolts to occur in four-stroke engines. Accordingly, they are designed to avoid the introduction of stress raisers. In this connexion it will be noted that at the junction of the shank with the head of the bolt, which is one of the most vulnerable positions, the fillet is formed by two radii, the larger one having the effect of reducing the stress concentration factor at the most highly stressed position. At the other end the thread runs into a relieving groove in the shank of the bolt and also the threads run out before reaching the face of the nut, which has the effect of minimizing the stress concentration at the bottom of the threads. Nuts of low carbon mild steel are used to assist in promoting a more even load distribution along the threads in engagement.

The material used for the bottom end bolts is a special low carbon grain controlled steel containing small percentages of nickel and chromium, oil hardened and tempered to give maximum ductility, a tensile strength of at least 33 tons per sq. in. and a very high Izod impact value of over 100 ft. lb.

The pistons, which are of aluminium alloy, represent a departure from normal practice in this country for engines of the size under consideration, particularly in the field of marine propulsion. On the Continent, however, notably in Germany, it is now common practice to use light metal pistons for pressure charged engines of 400 mm. (15·75 in.) bore and larger. The reason for this choice is that inertia forces are reduced to a minimum, the material is less liable to cracking due to thermal stresses than ferrous materials, and owing to its high thermal conductivity oil cooling is not necessary.

An alloy containing approximately 13 per cent of silicon, together with smaller amounts of other alloying elements, is used to produce the sand castings from which the pistons are made. The use of silicon reduces the coefficient of thermal expansion of the alloy and provides greater resistance to ring groove wear than some other aluminium alloys used for pistons. As a matter of interest it might be mentioned that it is a similar alloy to that employed in Germany during the war

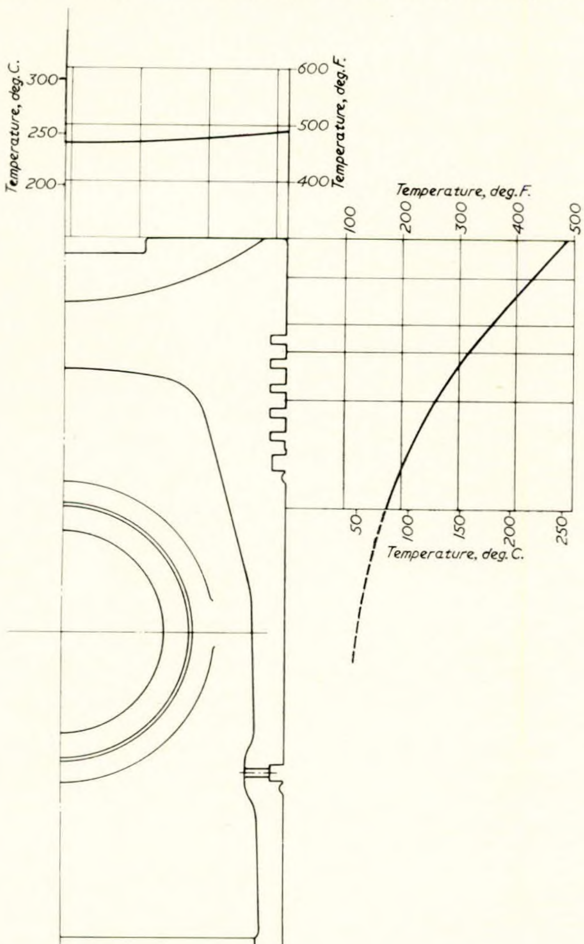

FIG. 5—*Piston temperatures under overload conditions*

in the manufacture of great numbers of pistons for submarine engines.

Heat treatment of the piston castings is an important part of the manufacturing process but is rather outside the scope of the present paper.

Although the design of the piston is very simple, as will be seen from Fig. 4, it represents the results of many years of development work by Continental engine builders. It will be noted that the thick section of the crown is gradually tapered-off to the skirt to provide an easy path for heat transmission and keep the temperature of the ring grooves as low as possible. The effect of this is illustrated in Fig. 5, which shows the temperature distribution under working conditions with the engine running on overload.*

The finished piston has a rather complex shape, being both tapered and with different degrees of ovality to allow for expansion under working conditions.

A chromium plated top ring, four plain compression rings and two slotted oil control scraper rings are fitted. It will be noted that the section of the rings differs from those commonly fitted in iron pistons, the radial width being greater in comparison with the depth to prevent blowby and reduce ring groove wear.

On assembly, the piston is heated in oil to a temperature of 80 deg.C (approximately 180 deg.F) which allows the gudgeon pin to be inserted by hand without difficulty; at the working temperature it is free to move, being restricted only in an axial direction by a circlip at each end.

Crankshaft

The crankshaft has been described as the backbone of the engine and the amount of work which goes into the making of this one item is probably greater than any other component; consequently, every stage of its manufacture is most rigidly controlled. It is solid forged from 0·3 per cent carbon steel with a tensile strength within the range of 33 to 37 tons per sq. in. Six-throw shafts are made in one piece, eight-throw may be in one or two pieces and the ten-throw in two sections.

The first stage of manufacture is to forge a straight rectangular slab with coupling ends from a 43-in. octagonal ingot which has a cast weight of 25 tons. At this stage the forging is examined for internal faults by means of an ultrasonic crack detector. The journals are then gashed out and rough turned before twisting the cranks into their correct angular positions. This is done by heating one crank at a time and while the adjacent one is held in the forging press the free crank is turned by means of an overhead crane into its correct position. Normalizing follows this process and subsequently physical tests are carried out on the material.

The pins and webs are machined from the solid rectangular slabs, the greater part of the material being removed by slotting and turning in a special purpose roughing lathe. Crankpins are finished in a crankpinning machine in which the tool rotates round the work.

To give some indication of the amount of machining involved, the rough forgings for an eight-throw shaft in two pieces weigh 18½ tons, whilst the finished shaft has a weight of only 7 tons.

Particular care is taken to ensure a high quality surface finish on the bearing surfaces and, also, in order to avoid introducing stress concentrations, at the run-out of fillets and oil holes. These holes are drilled through the shaft to convey oil from the main bearings to the bottom end bearings and subsequently to the gudgeon pin bearings.

Very close tolerances are laid down for dimensional accuracy and also for out of roundness, eccentricity and alignment of journals and pins. In checking a crankshaft it is mounted on a series of V blocks, as shown in Fig. 6, which are adjusted in height if necessary until the web deflexions are reduced to a minimum. Alignment is checked with the

* The data from which this diagram was constructed were obtained by fixing fusible plugs of known melting points in the piston at a number of positions over the surface of the crown and skirt.

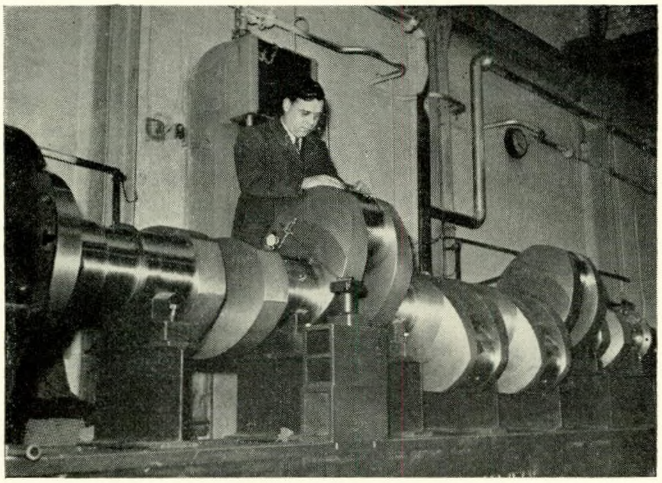

FIG. 6—*Inspection of six-throw crankshaft*

aid of a spirit level which is capable of registering an error of 0·06 mm. per metre (less than 0·001 in. per ft.). The readings so obtained are recorded and a similar procedure is adopted during the bedding-in of the crankshaft to the main bearings, to ensure correct alignment of the bearings.

Camshaft and Manœuvring Gear

The camshaft is produced with a series of steps in the diameter to allow for ease of assembly of the cams and is made in one or two pieces depending on the number of cylinders. The inlet and exhaust valve cams, each of which, of course, incorporate ahead and astern cams, are made of hardened steel. They are keyed directly to the shaft with a slight shrink fit and therefore require to be heated in oil before assembly. In the case of the fuel pump cams it is necessary to allow for adjustment, however, and they are carried in bushes which are keyed to the shaft. This construction is shown in Fig. 7.

Single helical gears transmit the drive from the crankshaft to the camshaft, which is carried in whitemetal-lined steel bearings fitted in pockets integral with the main frame.

Reversal is effected by longitudinal movement of the camshaft, the cam follower rollers being lifted clear of the inlet and exhaust valve cams during this operation. The fuel pump cam rollers are not lifted, but slide from one cam to the other during reversal of the engine. Space does not permit a detailed description of the construction and operation of the starting and reversing gear but the principle of operation is described briefly in an appendix to this paper. The construction of the manœuvring gear for a ten-cylinder engine is shown in Fig. 7; all manœuvres of the engine are carried out by movement of the one handwheel, shown in the stop position in Fig. 7.

The pilot valve operating cams are set on assembly so that movement of the handwheel through an angle of 16 degrees from the stop position operates the required reversing pilot valve. This admits compressed air at a pressure of approximately 250 lb. per sq. in. into the reversing cylinder, causing the piston to move and turn the reversing shaft through the rack and pinion drive. The stroke of this piston covers the full length of the cylinder and the reversing gear locking pawls are so adjusted that when it reaches the end of its stroke, corresponding to one complete revolution of the reversing shaft, one of these pawls is able to enter a slot in the rack on further movement of the manœuvring handwheel. This locks the reversing gear in position and prevents starting of the engine if the gear is not full over in the ahead or astern position.

Further movement of the manœuvring handwheel to a position 33 degrees from the central position operates the air starting valves, as described in the appendix.

Manual regulation of the output of the fuel pumps is effected by movement of the plunger *B* (at the top of Fig. 7) which is actuated through a system of levers by the eccentric *G* on the manœuvring handwheel spindle. This is so designed that during movement of the manœuvring wheel to the starting position the fuel pump racks do not go beyond the dead rack setting, but on further movement of the wheel the fuel pumps are brought into operation. Until the engine is turning on air in the correct direction, however, the locking device incorporating lever *A* in Fig. 7 prevents movement of the wheel to bring the fuel pumps into operation.

It will also be noted from Fig. 7 that a drive is taken from the end of the camshaft which operates both the tachometer and indicator driving mechanism. This last mentioned equipment reproduces, on a small scale, the movement of the crank and transmits this motion to a steel tape to which the indicator cord is attached.

Main Assembly

Engines are normally built on foundation girders in the position in which they are tested.

After levelling of the bedplate the semicircular main bearing shells are bedded to the main bearing pockets. These bearings are of steel with whitemetal linings 3 mm. thick, similar to those of the bottom end bearings; in order to maintain an unbroken bearing surface there are no oil grooves. The bearing surfaces are then bedded to a mandrel which has a diameter equal to that of the crankshaft journals plus oil clearance. Final adjustment of the bearings is carried out, if necessary, after the crankshaft has been tried in place, in order to ensure level readings on the journals and pins and crankweb deflexions close to those obtained during checking of the crankshaft after manufacture. The main bearing top halves are then fitted and adjusted to give the required oil clearance.

Testing of the water spaces of the cylinder block to a pressure of 30 lb. per sq. in. is carried out after pulling in the liners and fitting of cylinder head studs and lubricator quills. Before the main entablature is lowered on to the bedplate, the camshaft and reversing shaft bearings are fitted and bedded to their respective shafts.

The main through bolts, which pass from the top of the entablature to the bottom of the main bearing bearers in the bedplate, are also fitted into the bedplate before lowering the cylinder block into place. After lining up the block the tightening of these bolts is the next stage; no other bolts are used to connect together the two main castings and since they transmit the whole of the cylinder load from the cylinder heads to the main bearings they are subjected to high loading which varies from zero to a maximum at every compression stroke. The magnitude of the fluctuation in stress in the through bolts, and consequently the liability to fatigue failure, can be reduced to a fraction of that which would otherwise obtain if the bolts are adequately prestressed. (The fluctuating component of stress depends on the relative elasticities of the bolts and bolted parts and is dealt with more fully in a paper(2) read before this Institute a few years ago.)

To ensure these conditions the throughbolts are tensioned by means of hydraulic jacks, as shown in Fig. 8, the nuts being only lightly hardened up whilst the pressure is applied to the hydraulic cylinders.

After assembling the cylinder heads it is usual to fit the aftermost piston and connecting rod to enable an accurate top dead centre to be obtained for synchronizing the camshaft with the crankshaft. This is done by putting the piston at top dead centre and setting the camshaft, using a spirit level and an angle plate fixture attached to the camshaft gear wheel. The intermediate gear wheel between the crankshaft and the camshaft wheels is then fixed in position. The longitudinal setting of the camshaft is adjusted to suit the fuel pump driving gear and then the reversing shaft is located so that the cam follower rollers line-up with the corresponding cams. At this stage the worm on the reversing shaft, shown at *D* in Fig. 7 is not fixed; before this can be locked the reversing

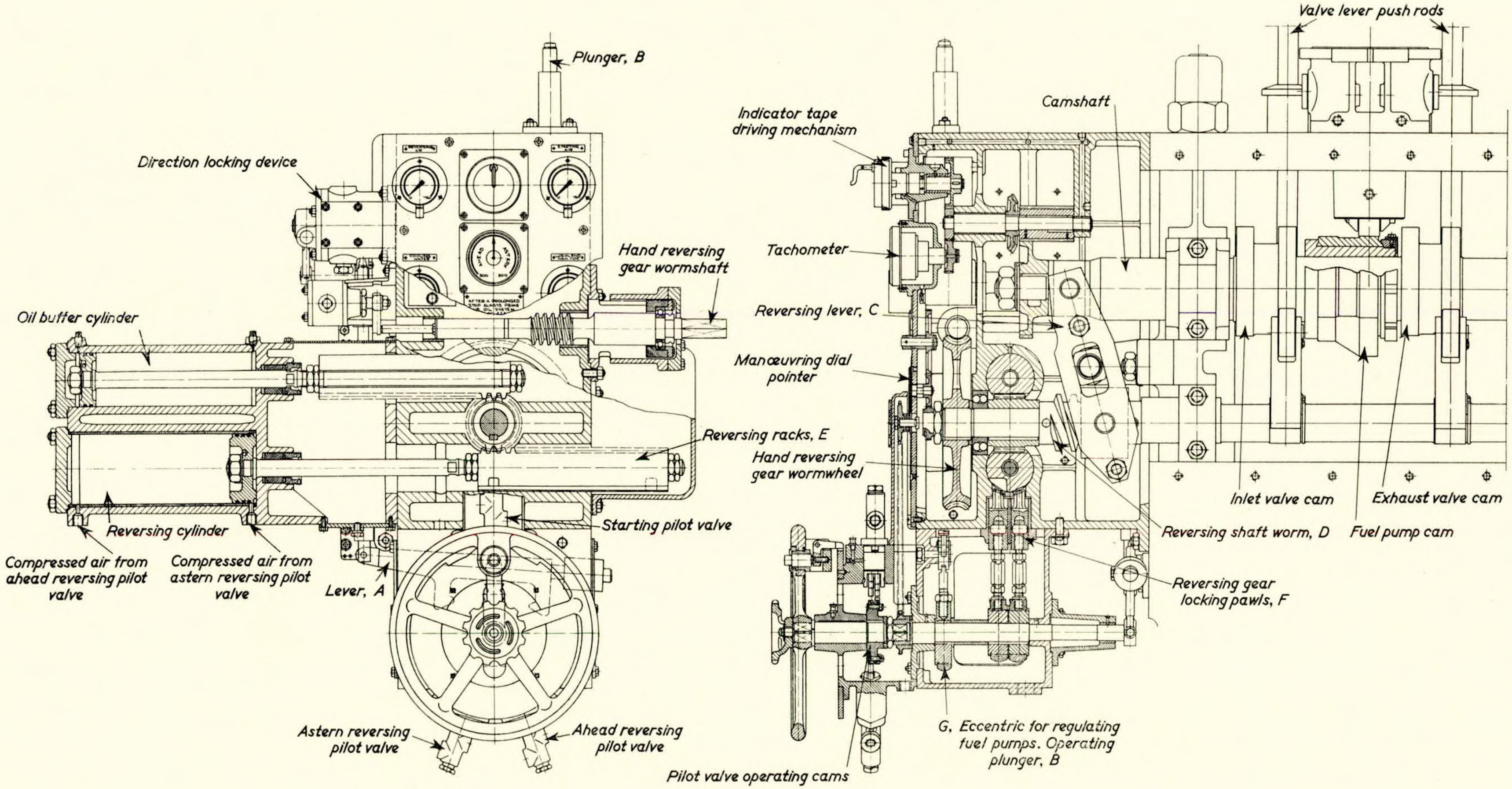

FIG. 7—*Manœuvring gear*

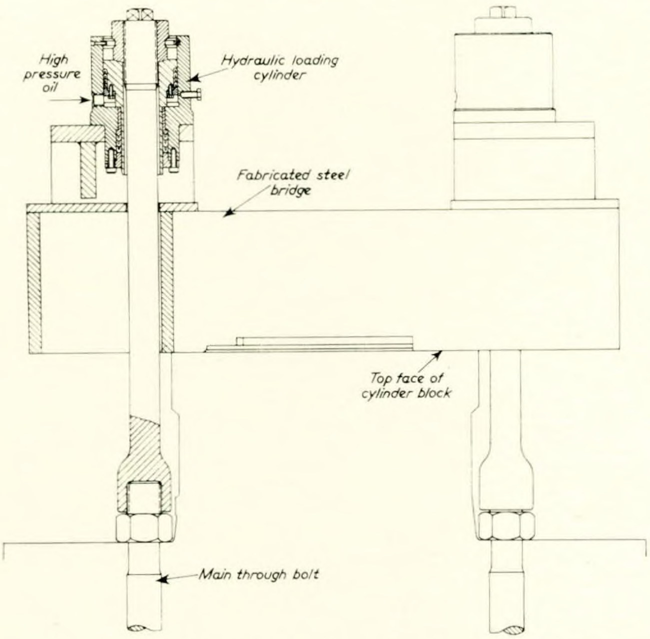

FIG. 8—*Hydraulic tensioning of main through bolts*

shaft has to be set to its correct angular position and then with the camshaft in its ahead-running position the scroll and reversing levers are set. (This mechanism is shown clearly in Fig. 7.)

The whole of the reversing and manoeuvring gear, which has been assembled previously, can now be mounted on the front of the engine, the reversing racks being set in their correct positions in relation to the pinion on the reversing shaft, as shown in Fig. 7.

The construction of the fuel pump operating gear and air starting valve mechanism is shown in Fig. 9. After assembly of this part of the engine, setting of the fuel pumps can proceed. This is carried out in two stages; in the first place the cams are set to their correct angular positions to give the same point of injection for ahead and astern running and then the lengths of the fuel pump operating plungers are finally adjusted to give the designed fuel pump timing. Both these operations are carried out with the fuel pumps in place, the point of port closure being obtained by observing the movement of a column of fuel oil in a capillary tube fitted in place of the high pressure pipe from the delivery valve of the pump. This is considered to be more accurate than spill timing or working from the timing marks on the pump; it also has the advantage that the fuel pump delivery valve has not to be removed as in spill timing.

After setting of the fuel pump cams they are locked in place by dowels fitted between the cam bush and the back half of the split cam. They are finally secured by tightening the cam bush nuts, which have conical faces.

The air starting valves are actuated by the fuel pump cams through the levers *X* and *Y* in Fig. 9. During running these are, of course, inoperative and are therefore retained by a spring so that the end of lever *X* is clear of the collar on the fuel pump operating plunger. When starting the engine, compressed air is admitted to the chamber *P* through the hole in the centre of the valve, which depresses the piston and operating spindle of the valve, bringing the end of lever *X* into contact with the collar on the fuel pump operating gear. This mechanism is set so that the starting valve opens when the piston is just over top dead centre, air being admitted for 130-deg. angular movement of the crank.

The assembly of the exhaust system can proceed as soon as the cylinder heads and turbocharger have been fitted. This last mentioned unit is completely self-contained apart from cooling water connexions, and consists of three casings, together with the air filter silencer, which are bolted together with the branches in the required relationship to each other. The turbine inlet casing has two, three or four entries which are separated right up to the turbine nozzle ring. By grouping the exhausts from different cylinders into these individual entries to the turbine, interference of the exhaust pressure wave from one cylinder with the scavenging of the cylinder that has fired previously is prevented. Thus, in the case of an eight-cylinder engine, the exhausts of four pairs of cylinders are separated by fitting four exhaust lines to the turbocharger. Expansion of each of these lines is accommodated by fitting stainless steel bellows pieces in the pipes; these are considered to be less troublesome than the sliding expansion pieces commonly used.

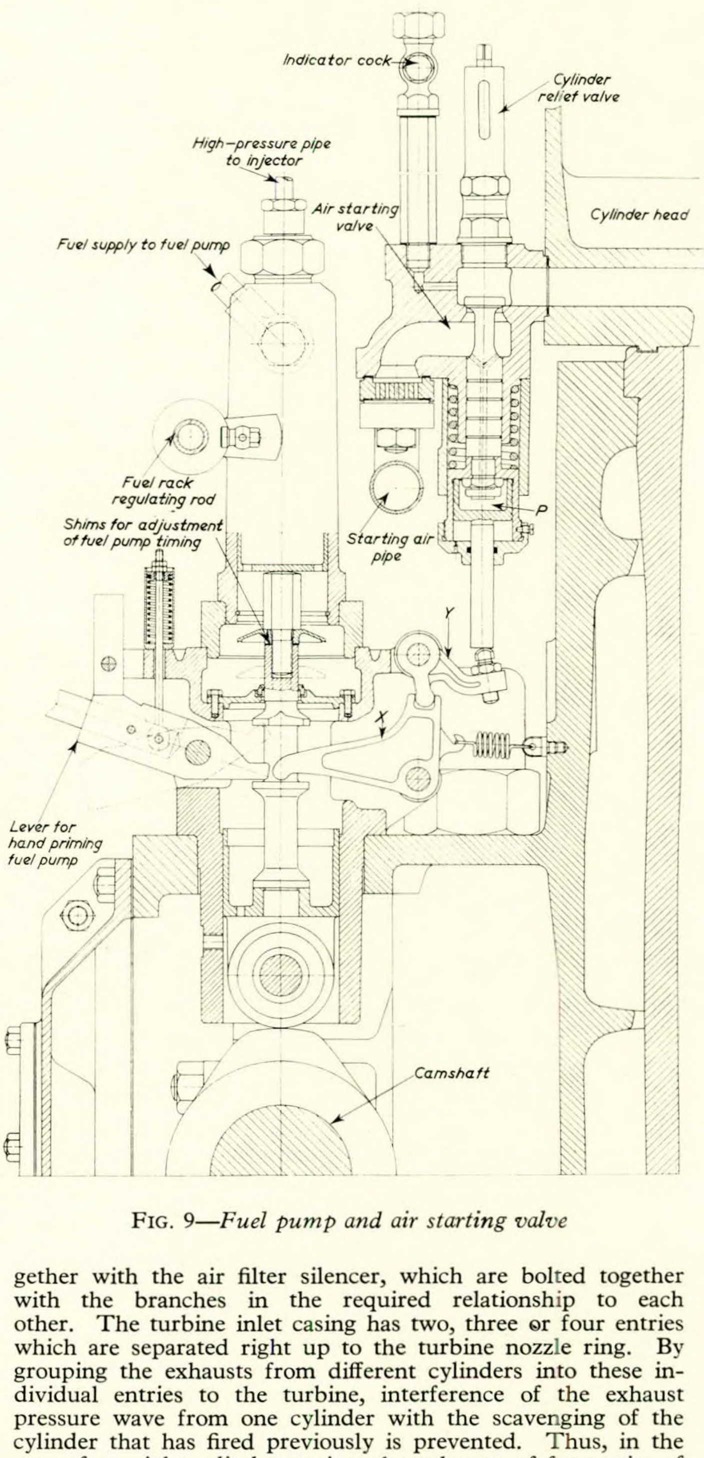

FIG. 9—*Fuel pump and air starting valve*

The axial flow single-stage exhaust gas turbine of the turbocharger is mounted on the same shaft as the radial flow air compressor. Various combinations of the individual components can be adopted to give correct matching, which is determined on tests of a prototype engine. The turbocharger is self-regulating so far as speed is concerned, and no controls are required, the speed of rotation depending on the energy imparted to the turbine by the exhaust gases and the power absorbed by the compressor.

TESTING

Test Equipment

Test bed trial data are normally presented without reference to the methods adopted and equipment used. In this case it was thought that some details of the test bed and associated equipment would be of some interest, and therefore, before dealing with the results obtained from engines running in the shop, the test equipment is described briefly.

As a new shop for the erection and testing of Diesel engines was to be built, various alternative layout schemes could be considered. The one finally adopted was to lay down two erection beds alongside each other so that it would not be necessary to move the engines for testing and allow for easy transport of the finished engines from the shop.

The first problem to be encountered was in the design of the foundation for the test beds. A test bore at the site proved that the ground was far from ideal for this purpose; indeed, to a depth of 23 ft. the subsoil consisted of very soft warp with a moisture content of over 50 per cent. Apart from the danger of settlement, this type of ground favours the transmission of vibration, which, in a built-up area, assumes great importance. With this problem in mind tests were carried out to determine the natural frequency of vibration of the ground and this proved to be within the range likely to be induced by engines running at speeds between 200 and 250 r.p.m.

To avoid resonance with the ground and minimize settlement, the design for the foundation evolved by specialists in this type of work consisted of a heavily reinforced concrete raft 4 ft. thick supported by thirty-two piles which were driven into a stratum of boulder clay underlying the silt. To ensure an homogeneous mass all the concrete for the test bed, except the top layer in which are embedded the bolting rails, was placed in one day, amounting to approximately 200 tons. This mass of concrete is isolated from the surrounding floor by slab cork and it has been found to be entirely satisfactory, vibrations even adjacent to an engine on test being imperceptible.

To provide cooling water for the cylinder jackets and lubricating oil and air coolers, together with water for dissipating the power produced by an engine of 2,000 b.h.p., would require a continuous supply of about 50 tons per hr. at a temperature of 50 deg. F. As a copious supply of water could not be relied upon it was therefore decided to adopt a recirculating system incorporating a forced draught water cooler.

A diagrammatic arrangement of the water piping is shown in Fig. 10, from which it will be seen that the system is divided into four sections which are interconnected. The cooler circuit consists of the main sump of 6,000 gal. capacity which is divided to within 6 in. of the surface of the water into the hot and cold sections. A pump draws from the hot sump and discharges to the water cooler, which then drains to the cold sump. This equipment is designed to cool 21,000 gal. of water per hr., from a temperature of 157 deg. F. to 97 deg. F. under summer conditions.

The hydraulic dynamometer which can be seen in Fig. 11 is supplied with water at constant pressure from the main header tank of 1,000 gal. capacity, mounted at a height of 35 ft. above floor level. This tank is fed by a pump which draws water from the cold sump, which, as mentioned above, can have a maximum temperature of 97 deg. F. After passing through the dynamometer the water drains to the hot sump.

Cooling water for the engine is drawn from the 200-gal. mixing tank by the circulating pump, which, after pumping the water through the engine system, discharges back to the same tank. The outlet temperature of the water from the engine is maintained at the required level by admitting water from the main header tank into the mixing tank, the excess overflowing to the hot sump.

Water for the lubricating oil cooler and the air cooler, which is normally fitted on pressure charged engines, flows by gravity from a third tank mounted at a height of 28 ft. above floor level. This "sea water" system is fed from the main header tank and, to bring the temperature down to the required level, mains water is also admitted to this tank. This supply is the only water drawn from the mains and the system is so designed that the quanti·y required does not exceed 5,000 gal. per hr. under summer conditions. After passing through the cooler circuit this water drains to the cold sump.

The lubricating oil system follows the pattern of a normal

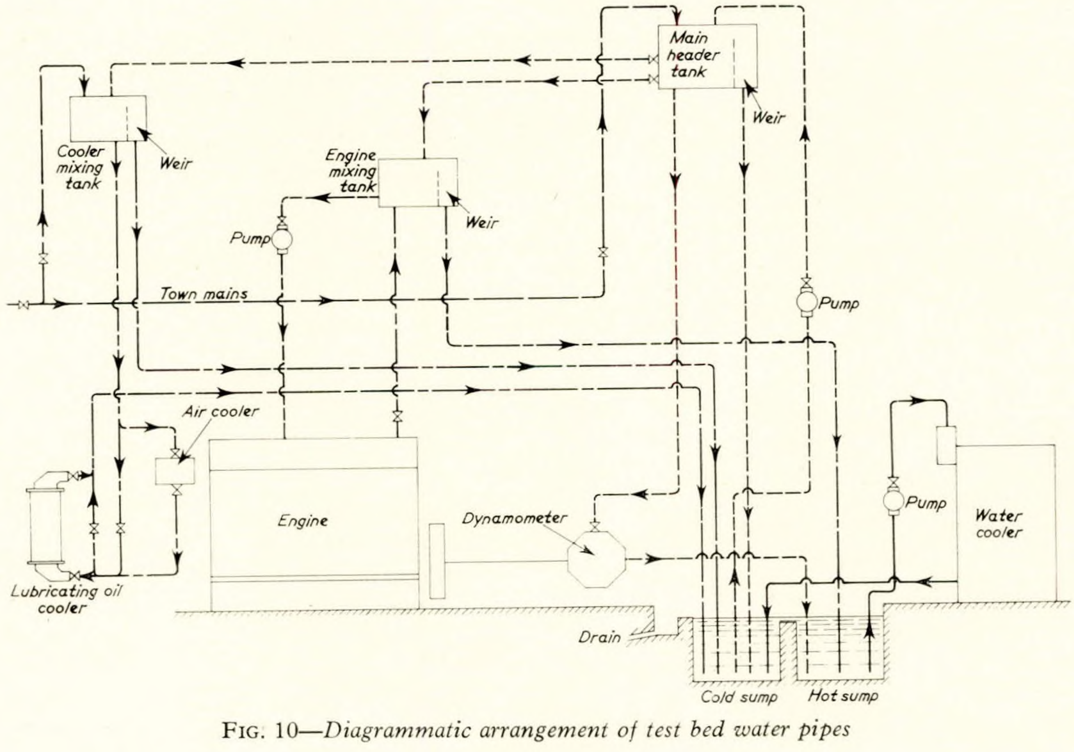

FIG. 10—*Diagrammatic arrangement of test bed water pipes*

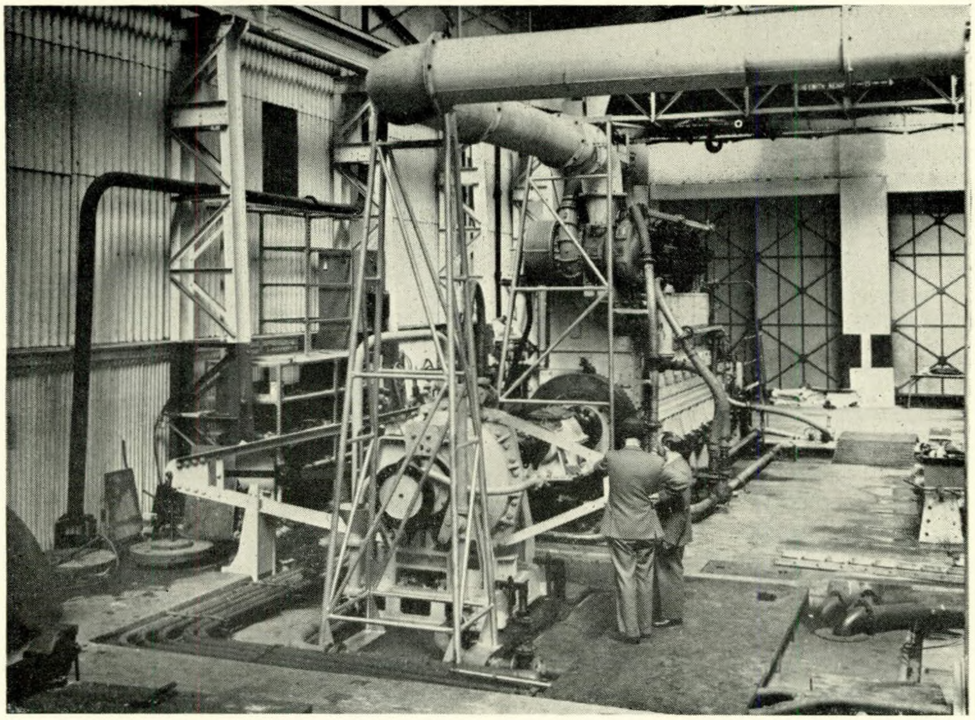

FIG. 11—*Engine connected up to hydraulic dynamometer for testing*

shipboard installation with a service tank holding 600 gal. of oil, a scavenge pump which draws oil from the engine sump and discharges to the service tank, and a pressure pump which delivers oil from the tank through the cooler, edge type filter and magnetic filter to the filter mounted on the lubricating oil inlet to the engine. A centrifugal purifier is installed for continuous treatment of the oil.

Fuel oil is drawn from the main storage tank by a pump which is automatically controlled by a float operated switch in the settling tank. From this tank the fuel oil passes through a centrifuge to the daily service tank which is located at a suitable height above the engine for the fuel to flow by gravity through the filter fitted on the engine to the fuel pumps.

The equipment for measuring the fuel consumption of an engine on test may be of particular interest as it is designed to record automatically the number of engine revolutions and time required for the consumption of a given weight of fuel. In this way the mean speed of the engine and the specific fuel consumption can be determined with great precision. Details of the system are shown diagrammatically in Fig. 12.

As mentioned above, the fuel normally flows by gravity to the engine through the change cock *P*, but when the fuel consumption is to be measured the cock is turned so that the

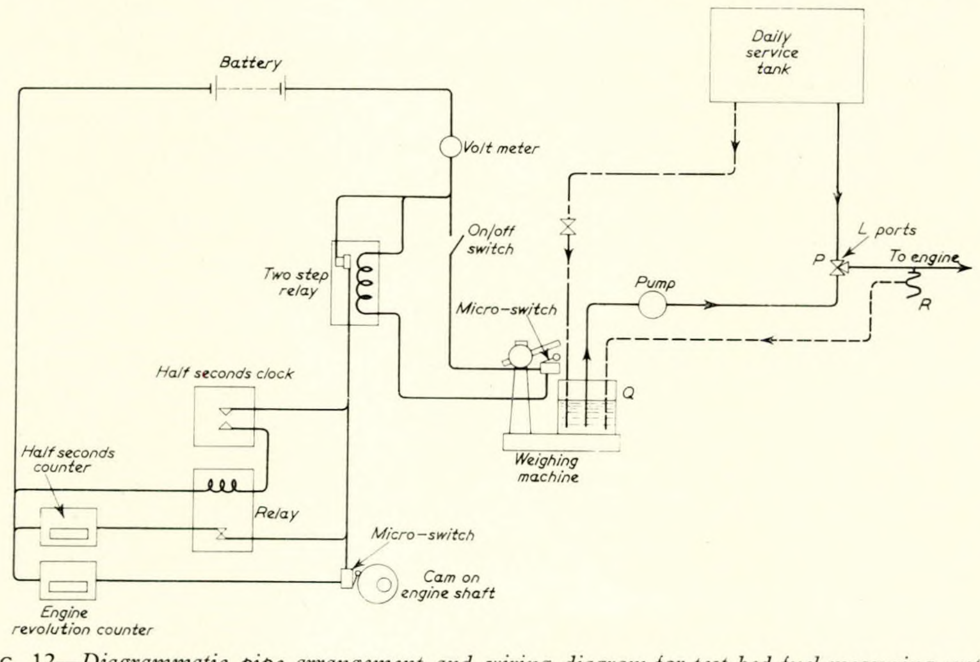

FIG. 12—*Diagrammatic pipe arrangement and wiring diagram for test bed fuel measuring system*

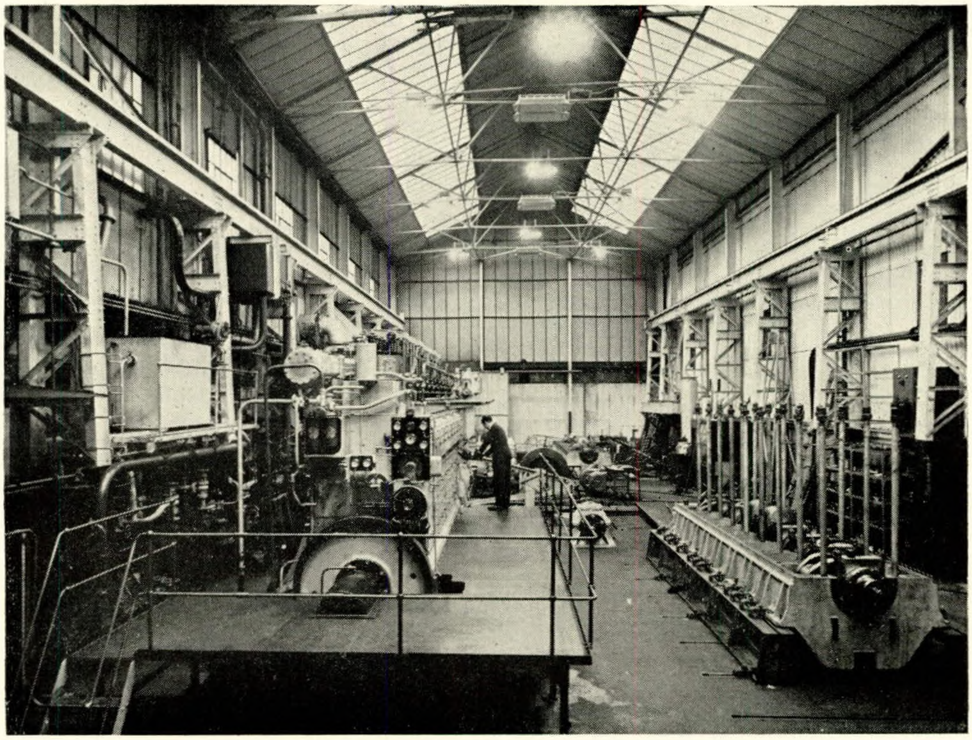

FIG. 13—*General view of the test bed*

fuel is pumped from the tank Q built on to the platform of a weighing machine. The pump supplies fuel to the engine in excess of its requirements, the surplus being returned to the measuring tank through the spring loaded valve R. A microswitch operated by the downward movement of the steelyard of the weighing machine makes and breaks an electrical circuit consisting of

(i) a half-seconds clock and electric counter, and
(ii) a microswitch operated by a cam on the engine shafting and a second electric counter to record the total number of engine revolutions during the period that the circuit is closed.

In operation the measuring tank is filled to a point at which the weight of fuel is sufficient to lift the steelyard of the weighing machine, the poise weight having been placed previously at a suitable setting. As the fuel is consumed, the arm of the weighing machine drops and starts the time and revolution counters. The steelyard is then lifted manually and the poise weight moved back by an amount equal to the weight of fuel to be measured. When the set amount of fuel has been used, the master microswitch again operates, this time stopping the counters simultaneously. The specific fuel consumption is calculated from a simple expression involving the number of revolutions made by the engine during the measuring period, the weight of fuel consumed and the brake load.

Other measuring equipment built into the test bed installation includes flow meters for indicating the quantity of water being circulated through the engine, the lubricating oil cooler and the pressure charge cooler. These instruments, together with various distant reading thermometers, are mounted on a central panel located at the control position of the various regulating valves. A comprehensive system of alarms is also installed, with signal lamps and klaxon horn, to give warning of failure of any of the equipment.

A general view of the test bed, on which can be seen some of the equipment referred to, is shown in Fig. 13.

Engine Testing

On first starting a new engine the normal procedure is to run without any load for about 20 min. to check the lubricating oil system, fuel injection equipment, starting valves, etc., the operation of which cannot be tried out before the engine is started. After a careful inspection of the running gear, the engine is started again on light load and gradually run up to full load over a period of about 9 hr. After running on full load for a further 4 hr., the fuel injectors are removed for inspection, as it has been found that there is sometimes an initial drop in the release pressure of the nozzles. During this period, compression and firing pressures are taken and any adjustment of the fuel pumps to balance the exhaust temperatures from the various cylinders are made.

Although the method adopted for timing the fuel pumps is considered to be as accurate as any, it is almost invariably found necessary to make some adjustment to the settings to bring the maximum pressures of the various cylinders within 15 to 20 lb. per sq. in. of each other. This is carried out quite simply, as mentioned earlier, by adjustment of the shims fitted under the fuel pump operating plungers.

After further running-in of the engine and the completion of routine tests, a series of tests are carried out to meet contract requirements at 50 per cent and 75 per cent full load, full load and 10 per cent overload. Typical results from such a test on an eight-cylinder engine developing a continuous power of 1,450 b.h.p. at 250 r.p.m., with a corresponding brake mean effective pressure of 116 lb. per sq. in., are summarized graphically in Fig. 16. It should be noted that the loading of the engine is in accordance with the propeller law in which it is assumed that the power output of the engine varies as the cube of the revolutions per minute. Although in Fig. 16 the curve representing brake horse power appears to be nearly a straight line owing to the scale adopted, it is, in fact, represented by the expression b.h.p. $= KN^3$ where K is a constant determined from the full power conditions and N the revolutions per minute. Thus, for the engine under consideration, half power is developed at a speed of 197 r.p.m. The brake load, which is applied by weights at the end of the balance arm of the dynamometer, is adjusted to give the corresponding torque.

The flow of water through the dynamometer is regulated automatically by the outlet valve to keep the torque balance weights floating and it is only necessary to adjust the water supply, which is at a constant head, to avoid an excessive rise in temperature.

Indicator cards are normally taken at each of the loads mentioned above but it is the author's opinion that even with the most reliable mechanical indicator such cards are of little use for determining the mean indicated cylinder pressure of an engine of the type under consideration. Accurate phasing of the movement of the indicator with the motion of the piston is extremely difficult and when coupled with the errors introduced in measuring the areas of the diagrams it is doubtful whether the indicated horse power value so obtained can be guaranteed within the limits of ±10 per cent. Nevertheless, out-of-phase cards are most valuable in order to check compression and firing pressures and operation of the fuel injection equipment. It is recommended that such diagrams should be taken at regular intervals in service.

Before considering the results plotted in Fig. 16 reference should be made to Fig. 14, which shows diagrammatically the variations in temperature of air and gas in its passage through the engine. After passing through the air filter silencer fitted to the intake to the turbocharger, the air enters the compressor which is driven by the exhaust gas turbine. On being compressed, the temperature of the air rises and if no air cooler is fitted it enters the cylinders at this temperature (shown in Fig. 14 by the chain dotted line). It is then compressed in the cylinder, reaching a temperature of the order of 1,100 deg. F. before combustion commences, thereafter attaining a temperature of over 2,000 deg. F. During expansion the temperature of the gas falls, and before the end of the exhaust stroke the air inlet valve opens, allowing a flow of cool air through the cylinder to the exhaust. This overlap period, during which time both the inlet and exhaust valves are open, occupies a time equivalent to a crank movement of approximately 150 degrees.

It will be seen, therefore, that the temperature indicated by the exhaust gas thermometer fitted at the outlet from the exhaust valve must be considerably lower than the real temperature of the exhaust gases, as it is subjected successively to impulses of hot gas and cool air, coupled with the fact that during the greater part of the time there is no gas flow at all past the thermometer. Thus, for a higher speed engine the exhaust thermometer reading approaches more closely to the true gas temperature. It should be borne in mind that this temperature cannot be used as a basis of comparison between different engines, depending as it does on speed, specific fuel consumption, excess air ratio, etc.

Temperatures of the exhaust gas recorded at the inlets to the turbine are considerably higher than those at the exhaust valves, this apparent rise in temperature being shown taking place in the exhaust pipe in Fig. 14. This rise in temperature can be accounted for partly by the conversion of kinetic energy in the exhaust gas impulses to pressure energy at the turbine but can also be attributed in large measure to the mixing of the gases in their passage to the turbine. In passing through the turbine some of the heat energy is converted into power to drive the compressor, with a resulting fall in temperature; the temperature at the exhaust from the turbine however, is still higher than that indicated by the cylinder exhaust thermometers.

When a sea water/air pressure charge cooler is fitted, the temperature of the air from the turbocharger can be reduced practically to the inlet temperature before it passes to the air manifold of the engine. This is illustrated by the dotted line in Fig. 14 from which it will be seen that this reduction in temperature is reflected in the temperatures throughout the remainder of the engine. By fitting a pressure charge cooler it will therefore be seen that for a given power the thermal loading on those parts of the engine subjected to combustion conditions is decreased, or conversely the power developed by the engine can be increased without introducing higher temperatures.

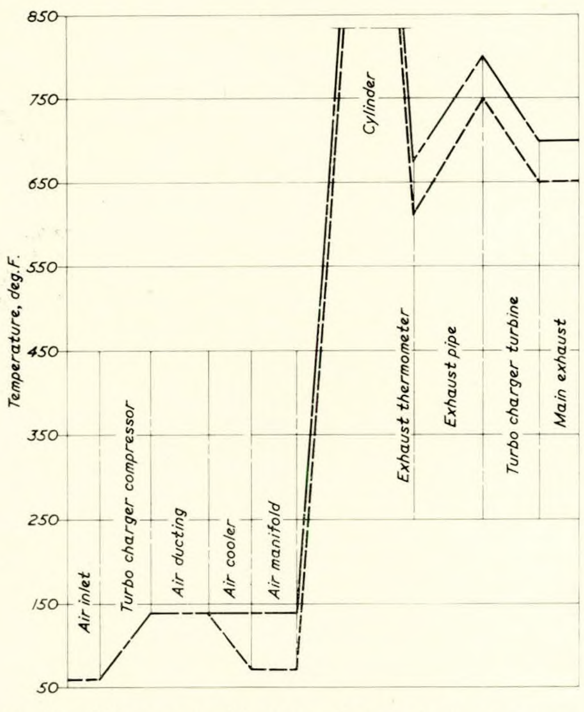

FIG. 14—*Diagram to illustrate changes in temperature of air and exhaust gas through engine*

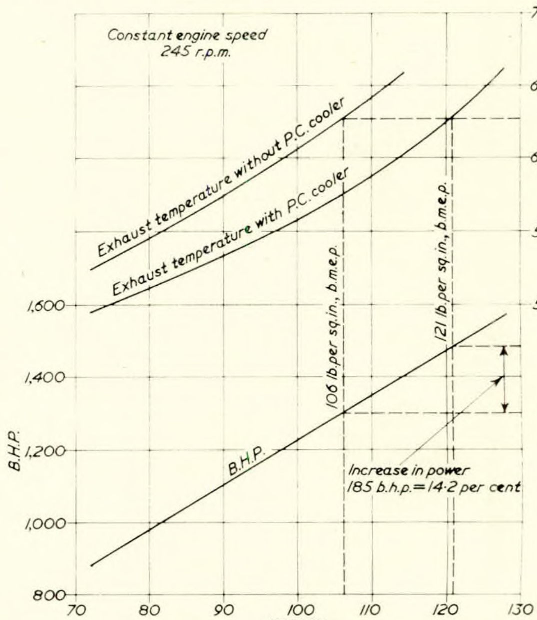

FIG. 15—*Effect of pressure charge cooler on output—eight-cylinder engine*

This point is illustrated in Fig. 15 in which curves are plotted from the results of tests on an eight-cylinder engine with a rating of 1,480 b.h.p. at 245 r.p.m. In this diagram exhaust temperatures are plotted against b.m.e.p. with and without the P.C. cooler in operation for a constant speed of 245 r.p.m. It will be seen that by cooling the air entering the cylinders and in this way obtaining the twofold advantage of increasing the air density and reducing the exhaust temperature, the b.m.e.p. can be increased from 106 to 121 lb. per sq. in., with the same exhaust temperature. This represents a power increase of over 14 per cent.

Turning now to the test results plotted in Fig. 16 (a), a detailed examination of curves *G* and *H* shows that the ratio

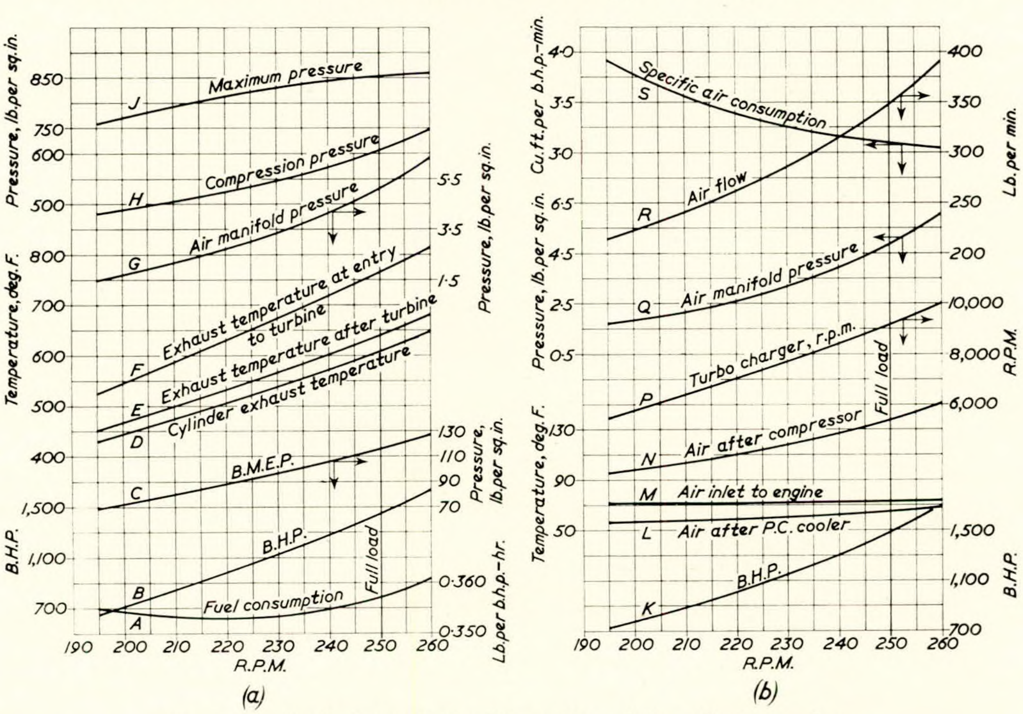

FIG. 16—*Typical test bed trial results—eight-cylinder engine*

of compression pressure to air manifold pressure is practically constant over the range shown, the exponent in the expression PV^n=a constant being equal to approximately 1·37. It will be noted that the lowest specific fuel consumption occurs at about 75 per cent full load, which in this case corresponds to a speed of 227 r.p.m. At loads above this value there is a slight rise in the specific fuel consumption, expressed in terms of lb. per b.h.p. hr., and there is an appreciable increase above full load. It is significant in this connexion that the difference in firing pressure and compression pressure is a maximum at about 75 per cent load with a value of 285 lb. per sq. in. (20 atmospheres). This confirms the generally accepted rule that for optimum efficiency, and hence minimum fuel consumption, the rise in cylinder pressure during combustion should be of the order of 20 atmospheres.

Referring to curves *D*, *E* and F in Fig. 16 (a), these again illustrate the phenomenon previously discussed of an apparent rise in exhaust temperature between the exhaust valves and turbine inlet branches.

Curves *L* and *M* in Fig. 16 (b) show that in these particular tests the air, in passing through the P.C. cooler, was cooled to a temperature lower than that of the air entering the turbocharger. At full load the effectiveness of the cooler, which is the ratio of air temperature drop divided by the inlet air temperature minus the inlet water temperature, had the high value of 85 per cent.

It is not usual to carry out air flow measurements as a routine procedure when engine testing but the results of such a test are included in Fig. 16 (b). Curve *R* gives the quantity of air passing through the engine in lb. per min. while curve *S* shows the variation in specific air consumption (the number of cu. ft. of air at N.T.P. per b.h.p. per minute) with load. At full load this has a value of 3.1 which is a relatively high value for this type of engine and has the effect of lowering the temperature of the parts subjected to combustion conditions and reducing the exhaust temperature.

IN SERVICE

Only a limited amount of service data has yet been obtained and it is therefore rather early to deal adequately with the question of service performance. It is not yet possible, for example, to give any reliable figures for the rate of liner wear and other facts relating to maintenance.

Although some companies have turned over almost completely to Diesel propulsion, in general the operators of smaller vessels in this country have had relatively little experience of the running of motor vessels. This will be appreciated when it is noted from the statistics issued by Lloyd's Register of Shipping that of British registered vessels of less than 1,000 gross tons only 34 per cent are fitted with Diesel propelling machinery compared with 80 per cent in Germany, 95 per cent in Holland and 80 per cent in Norway. Operators are therefore faced with difficulties in regard to manning and maintenance but the importance of routine maintenance of the machinery by adequately trained men cannot be overstressed. The amount of maintenance necessary is obviously related to the amount of wear and tear, which in turn depends on the

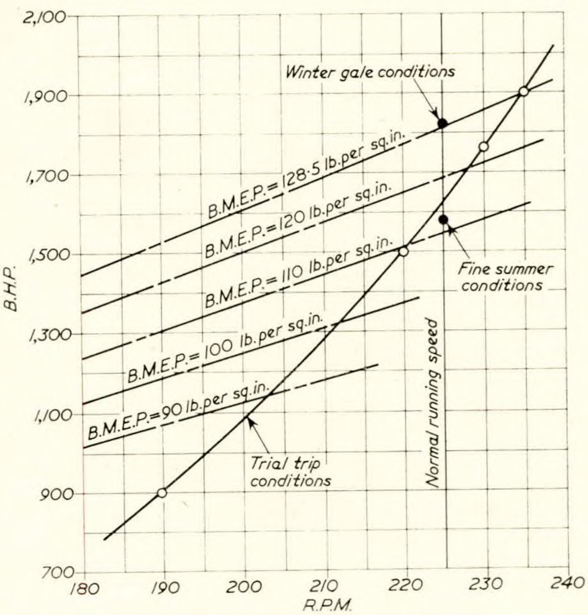

FIG. 17—*Trial trip and service powers—ten-cylinder engine*

rating at which the engine is operated. It would therefore appear prudent to specify two distinct ratings, namely (1) the power and revolutions under trial trip conditions, and (2) normal service conditions.

It is an accepted fact that Diesel propelling machinery is more sensitive to heavy weather conditions than either steam reciprocating or turbine machinery and when an engine is on governor control, vigilance is necessary to prevent overloading under such conditions. Some indication of the increase in power required to maintain a constant propeller speed under adverse weather conditions is given in Fig. 17. In this diagram the curve shown is plotted from trial trip data obtained from a distant water trawler fitted with an engine rated at 1,700 b.h.p. at 225 r.p.m. It will be noted that under trial trip conditions a maximum power of approximately 1,900 b.h.p. was developed at 235 r.p.m. with a b.m.e.p. of 129 lb. per sq. in.; this corresponds roughly to 10 per cent overload. Under normal service conditions in fair weather the estimated power developed is 1,580 b.h.p. at 225 r.p.m. but in a strong gale a power increase of the order of 15 per cent is necessary to maintain the same engine speed. A fairly reliable indication of the b.m.e.p. is afforded by the exhaust temperature and it is therefore recommended that a limit should be fixed on this value by reference to the test bed results.

A point of interest arising from the results plotted in Fig. 17 is that over the speed range from 190 to 230 r.p.m. the power absorbed by the propeller varies as the revolutions raised to the power of 3·5. In other words, the power absorbed by the propeller increases more rapidly with increase in engine speed than is normally assumed.

It has been found in service that manoeuvring of the engine can be carried out very rapidly. This is a characteristic of the design of the reversing mechanism which allows the camshaft to be moved rapidly into the reverse position in relation to the direction of rotation of the engine, and before the engine stops turning starting air can be admitted to the cylinders to stop the rotation of the propeller and start the engine in the opposite direction. In this way it is possible with full way on the ship to go from full ahead to turning over astern in as short a time as 18 secs. At reduced speed reversal can be carried out even more rapidly but this procedure is not recommended as a regular routine. It is evident that very rapid reversal imposes undue stresses on the moving parts and it is questionable whether, in fact, stopping of the ship is effected any quicker. The stopping of ships is a subject which has received considerable attention in recent years and tests[3] have shown that in order to bring a ship to rest in the shortest distance, ahead power should be shut off as quickly as possible and as soon as the propeller revolutions have dropped to " slow " the engine should be started dead slow astern. Full astern rotation whilst the ship is moving ahead merely causes cavitation of the propeller and the effective thrust is practically nil.

Control of the engine cooling water temperature presents no problem under steady running conditions but where the service of the vessel involves considerable manœuvring and running at reduced power it is difficult to prevent overcooling. This has a detrimental effect on the engine and undoubtedly contributes to excessive liner wear. A number of installations have been fitted with a thermostatically controlled bypass valve on the fresh water cooler and this has been found to function extremely well, maintaining the cooling water outlet temperature from the cylinder heads within a few degrees of the recommended figure of 140 deg. F.

Thermostatic control of the lubricating oil temperature is not considered to be necessary as the amount of heat carried away in the lubricating oil is very small. A normal figure for this heat dissipation for the type of engine under consideration is 130 B.t.u./b.h.p./hr. but it has been found from tests to be actually only about 70 B.t.u./b.h.p./hr.

It has been found to be advantageous to use a detergent oil for cylinder lubrication and only in exceptional cases have there been any signs of ring sticking after the normal running time of 2,000 hr. between piston examinations. Some engine builders recommend the use of a detergent oil for the crankcase but for an engine with uncooled pistons there does not appear to be any advantage in using other than a high quality straight mineral oil.

The lubricating oil consumption, excluding cylinder oil, has been found to average just under 6 gal. per 24 hr. over a period of 4,000 hr. for an engine developing 1,400 b.h.p. at full power. For cylinder lubrication the output of the mechanical lubricators is regulated to approximately 2 gal. per 1,000 b.h.p. per 24 hr. after the first 500 hr. running.

APPENDIX

Operation of the Starting and Manoeuvring Gear

Referring to the diagrammatic arrangement shown in Fig. 18, a movement of the manoeuvring wheel (12) to the left or right of the central stop position turns the cams (27 and 28) which actuate the reversing pilot valves (13 and 14). If the camshaft is not in the correct position for the direction of rotation desired, the corresponding pilot valve is opened, automatically admitting compressed air from the reducing valve (15) to the reversing cylinder (16). The piston in the reversing cylinder moves the rack (17) to the right or left, turning the pinion (18) which is keyed on to the reversing shaft (8), through one complete revolution. An oil buffer cylinder (19) ensures that this movement proceeds smoothly and at the correct speed.

The reversing shaft is provided with eccentric journals on which are mounted the cam follower levers (9) which transmit the motion of the cam rollers through the push rods (10) to the inlet and exhaust valves in the cylinder heads.

Rotation of the reversing shaft causes the camshaft (5) to move longitudinally by means of the double-start screw and nut and lever mechanism (20—23). At the same time the rollers are lifted clear of the inlet and exhaust cams (1 and 3) by the eccentric motion imparted to the cam follower levers by the rotation of the reversing shaft. After one complete revolution of the reversing shaft the movement of the camshaft

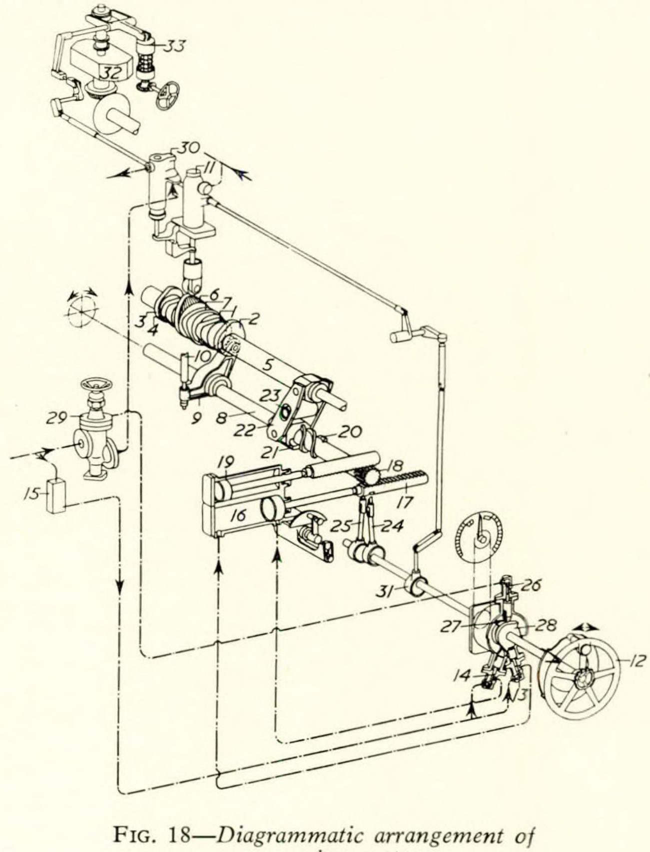

FIG. 18—*Diagrammatic arrangement of manœuvring system*

corresponds to the spacing of the ahead and astern cams; the cam follower levers return to their original positions and therefore the rollers come into contact with the cams corresponding to the opposite direction of rotation of the engine (2 and 4).

The fuel pump cam rollers are not retracted during reversal; when the camshaft moves they slide across the incline which connects the ahead and astern cam profiles (6 and 7).

The manœuvring handwheel can only be turned to the starting position if the reversing gear is in the correct position. This is ensured by the locking pins (24 and 25) which prevent movement of the handwheel unless the one corresponding to the required direction of rotation is in engagement with the rack; in this way the reversing mechanism is locked during running of the engine.

Further movement of the handwheel to the starting position moves the cams (27 and 28) to open the starting pilot valve (26). This, in turn, allows the automatic starting valve (29) to open, admitting compressed air to the starting valves (30) which then come into operation. The automatic starting valve opens only when the manœuvring handwheel is in the starting position; for any other setting the valve remains closed, the starting air line being vented by slots in the valve. A handwheel is also fitted to the automatic starting valve so that when the engine is shut down the valve can be locked in the closed position and there is no possibility of the engine being started accidentally.

After the engine has been started on air, further movement of the manœuvring wheel automatically brings the fuel pumps (11) into operation and stops the air supply to the starting valves. Regulation of the fuel supply can be effected in two ways:—

(a) by means of the manœuvring wheel, the spindle of this wheel being fitted with an eccentric (31) which operates the fuel pump racks through a system of levers; or

(b) by means of the centrifugal governor (32). Under normal conditions the quantity of fuel is automatically regulated by the governor to maintain a constant speed when a change in load occurs. Alternatively, the speed can be varied between specific limits while the engine is running by adjusting the speed regulating device (33) in the governor.

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