The Harland and Wolff Pressure Charged Two-stroke Single-acting Engine

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It is over eighteen years since the author gave, before the Institute, an account of Diesel propelling and auxiliary engines. In the intervening years the changes and advances have been many and important.

The first single-acting opposed piston eccentric-type propelling engine was designed in Belfast and entered service in 1949. Since 1950, engines of the type aggregating over 700,000 horse power have been built. The first turbocharged two-stroke propelling engine was taken in hand in 1953, as the result of successful preliminary work on a four-cylinder eccentric-type opposed piston stationary unit of 370 mm. bore, 825 mm. stroke. Since then, the pressure charged propelling engines on order and in service have totalled 500,000 horse power. These results are indicative of the author's immediate background.

The present paper begins by describing, in comprehensive detail, the Harland and Wolff opposed piston, pressure charged engine. Then it proceeds to analyse and discuss some of the more important matters associated with the successful running of the engine type. It terminates with particulars of test bed trials, tables of engine powers, and some general notes.

INTRODUCTION

In this paper an endeavour is made to describe the Harland and Wolff single-acting two-stroke pressure charged marine Diesel engine in a manner likely to be acceptable to the various grades of engineer into whose hands it may come. For this reason many numerical particulars are included which do not usually fall within the compass of technical papers whose purpose it is to stress the advantages of, and the advances made in, the machine or machines described. The young engineer may see, in these data, opportunity for making exercises and for assimilating a sense of proportion; to the more experienced man they will indicate a level of practice; to the shrewd ship-owner and superintendent they can be a means, perchance, of equating value to price.

The metric system is used throughout the paper; but many leading and important dimensions are repeated in British units. In this connexion it should be noted that all horse powers quoted are British horse powers, i.e. 33,000ft. lb. per min. Quoted weights are British tons, except where specifically stated to be metric tons (1 metric ton = 1,000 kg. = 2,205lb.). Where bars, plates, etc., are quoted in mixed units, the implication is that the inch units are the rolled sizes and the metric units the dimensions after being machined. Where pressures are stated in atmospheres, 1 atmos. = 1kg. per sq. cm. (14·223lb. per sq. in.). Translations from one system of measurements to the other system are given accurately, despite the fact that this practice may lead to insignificant decimal fractions being attached to relatively large whole numbers.

ENGINE CHARACTERISTICS

In the creation of an engine and in the shaping of its evolution the designer must make up his mind at an early stage what it is that he is trying to do. An engine can be constructed to be of maximum lightness; or it can be made for occupying the least space, or to have—before and above all else—a low fuel consumption; it can be designed for a low first cost; or primary attention can be given to matters of overhaul and maintenance; and so on. But if one or other characteristic predominates, it can be expected that the remainder will tend to be recessive.

In the engine type with which this paper is occupied the best all-round compromise has been sought: that is, an engine of reasonable weight and space; of reasonable first cost and economy; an engine constructed of simple materials, and reasonably easy to overhaul and to maintain.

The advantages accruing from the exhaust turbocharging of a propelling engine include the following, viz.: (i) for engines of equal power, there is a reduction in length of 22 per cent, the height and width remaining unchanged; (ii) there is a reduction in weight of 27 per cent; (iii) the initial cost in pounds sterling per horse power is reduced; (iv) the specific fuel rate is less, to the extent of approximately 4 per cent. Expressed another way: for the same engine dimensions and weight, and for approximately the same initial price, 30 to 35 per cent increase in power is obtainable with no off-setting disadvantages.

The Harland and Wolff crosshead type of pressure charged two-stroke engine comprises, to date, two cylinder sizes, namely: 750 mm. (29·53in.) bore, 2,000 mm. (78·74in.) total stroke; and 620 mm. (24·41in.) bore, 1,870 mm. (73·62in.) total stroke. The numbers of cylinders per engine have hitherto ranged from four to eight; but a greater number of cylinders can be provided without any difficulty. There is also available a trunk design of the engine type, 530 mm. (20·87in.) bore, 1,180 mm. (46·46in.) total stroke, in four to ten cylinders per unit; but, as this engine is primarily intended for special craft, it is excluded from this paper.

Fig. 1 shows a transverse cross-section of the 750 mm.

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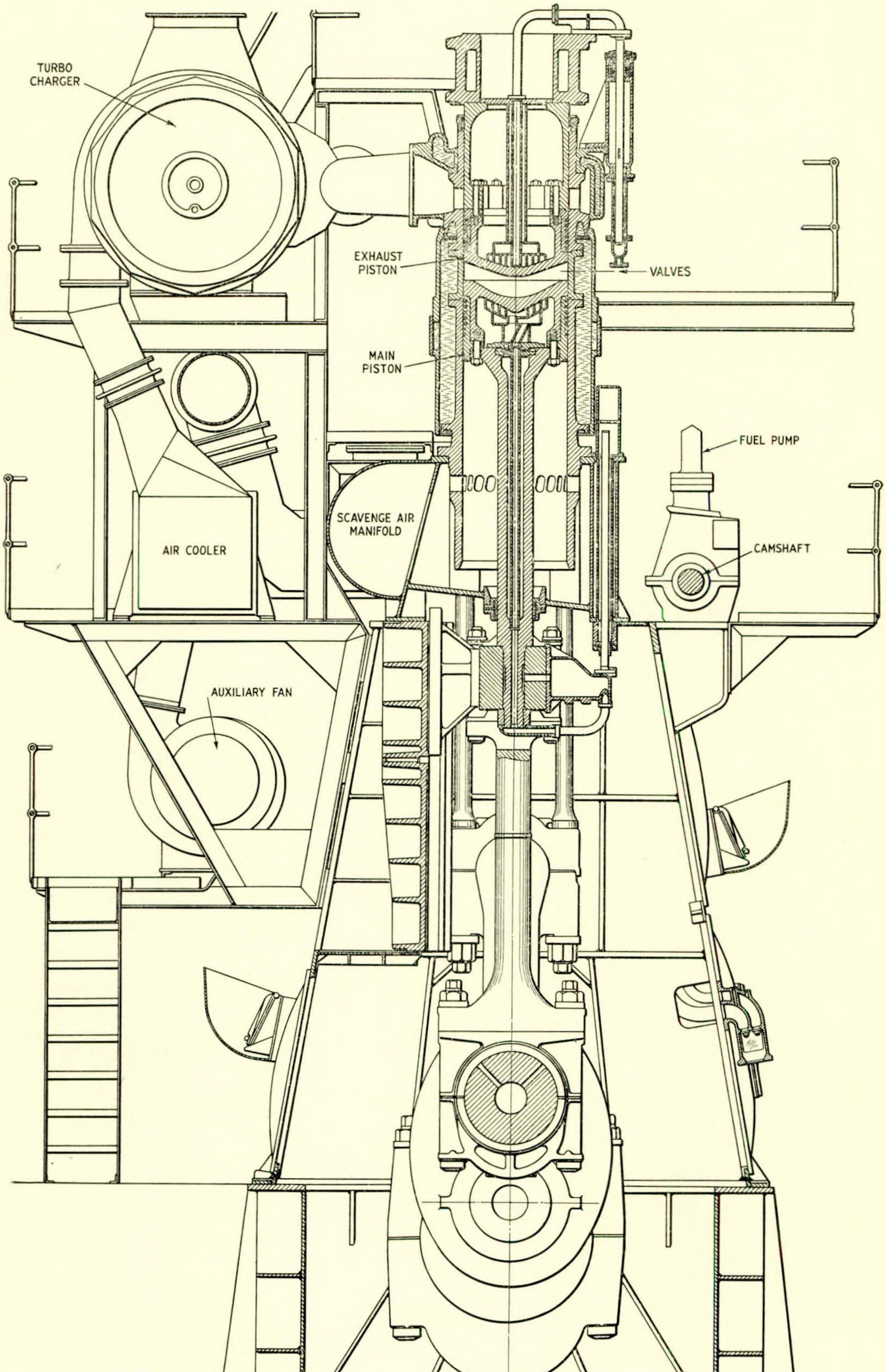


FIG. 1—*Turbocharged two-stroke engine of 750 mm. bore, 2,000 mm. stroke*

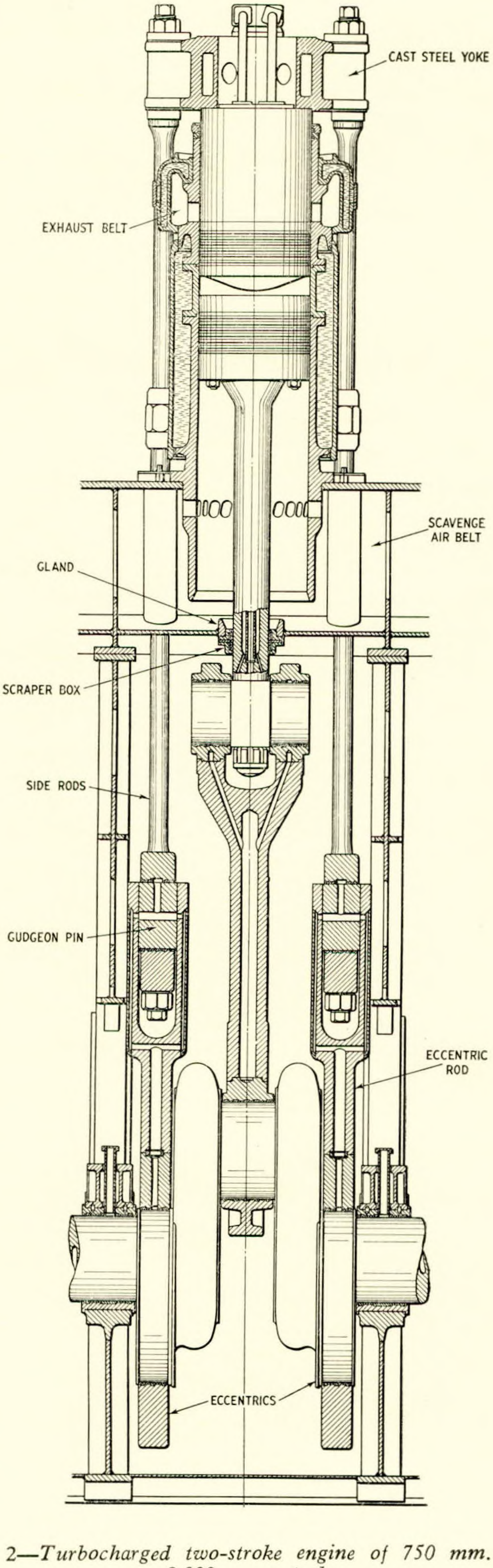


FIG. 2—*Turbocharged two-stroke engine of 750 mm. bore, 2,000 mm. stroke*

(29·53in.) bore, 2,000 mm. (78·74in.) stroke engine. Fig. 2 is a fore-and-aft view of one cylinder.

The engine is of opposed piston form; hence the characteristics of the design include the longitudinal scavenging of the cylinders and a stress system which is self-contained. The stroke of the upper, or exhaust, pistons is about one-third of that of the lower, or main, pistons; the exhaust pistons are controlled by an arrangement of eccentrics on the crankshaft. The crankshaft is strong and stiff; all important bearings are of simple design. The engine structure is stoutly built from fabricated steel plates; the crankcase is completely isolated from the cylinders; the ratio of connecting rod to crank is 4 : 1. The exhaust turbocharging blowers are carried by a suitable framework arranged fairly high on the back of the engine; there are no positively driven scavenge blowers.

It can doubtless be agreed that the backbone of an engine is its crankshaft. If the crankshaft is trouble free, if the bedplate in which it is carried is strong and robust, and if the double bottom underneath the engine is of requisite sturdiness, then anything else that may happen to the engine, however serious it may be, cannot be a root-and-branch affair.

The cylinder centres of an engine are determined by a build-up of bearing widths and crankweb thicknesses; the engine height by a summation of dimensions of which the crank throw is the basic unit. The transverse spread of the engine, however, is less fettered and therefore allows more scope to the designer. In Fig. 1, reckoning the height of the engine as extending from the bottom flange of the bedplate to the top of the cast steel yoke when an exhaust piston is at the upper end of its stroke, and taking the width of the engine as the distance over the bottom flanges of the bedplate, then the ratio of height to width is between the limits 2·6 : 1 and 2·5 : 1 —depending upon the engine cylinder size. That these proportions ensure a steady engine is confirmed by Fig. 62, which is a casually taken test bed photograph of a six-cylinder 620 mm. (24·41in.) bore, 1,870 mm. (73·62in.) total stroke engine when running at 118 r.p.m. and when developing its service power of 5,800 b.h.p. The exposure was 15 seconds. That is, during the exposure of the plate, the engine made about thirty complete revolutions. Apart from the upward blur at the exhaust piston path, the smudge at the turning wheel, and the position of the water brake pointer, there is nothing on a 10-in. by 12-in. print to suggest that the engine was not at a standstill.

DETAILS OF ENGINE COMPONENTS

For descriptive purposes the engine can be divided into: (a) its static components; (b) its running parts. Attention will be given to the two divisions in that order.

Cylinders

The main static structure of the engine consists of the bedplate, the "A" frames, the scavenge air reservoir, and the cylinders. The bedplate will receive attention when the engine running parts are examined. The "A" frames and the scavenge air manifold—which are mainly of interest to designers and builders—are fabricated from steel plates and bars. After completion, the frames and air reservoir are stress relieved and shot blasted.

The general form of the "A" frames is shown in Figs. 1 and 2. The general plate thickness is ¾in.; the top and bottom flanges are respectively 45 mm. and 50 mm., the outer sloping flange 1¾in., and the plate for the crosshead guides 2in. Each frame is secured to the bedplate by sixteen 1¾-in. bolts, total, of which four are fitted. At the top flange, where the air manifold rests upon the "A" frames, there are eight 1¾-in. bolts, total, of which four are fitted.

If the frame design is analysed it will be found to be a sound and economical compromise amongst the several conflicting technical and manufacturing factors, with low first cost as a predominant consideration.

The plates constituting the scavenge air manifold in Figs.

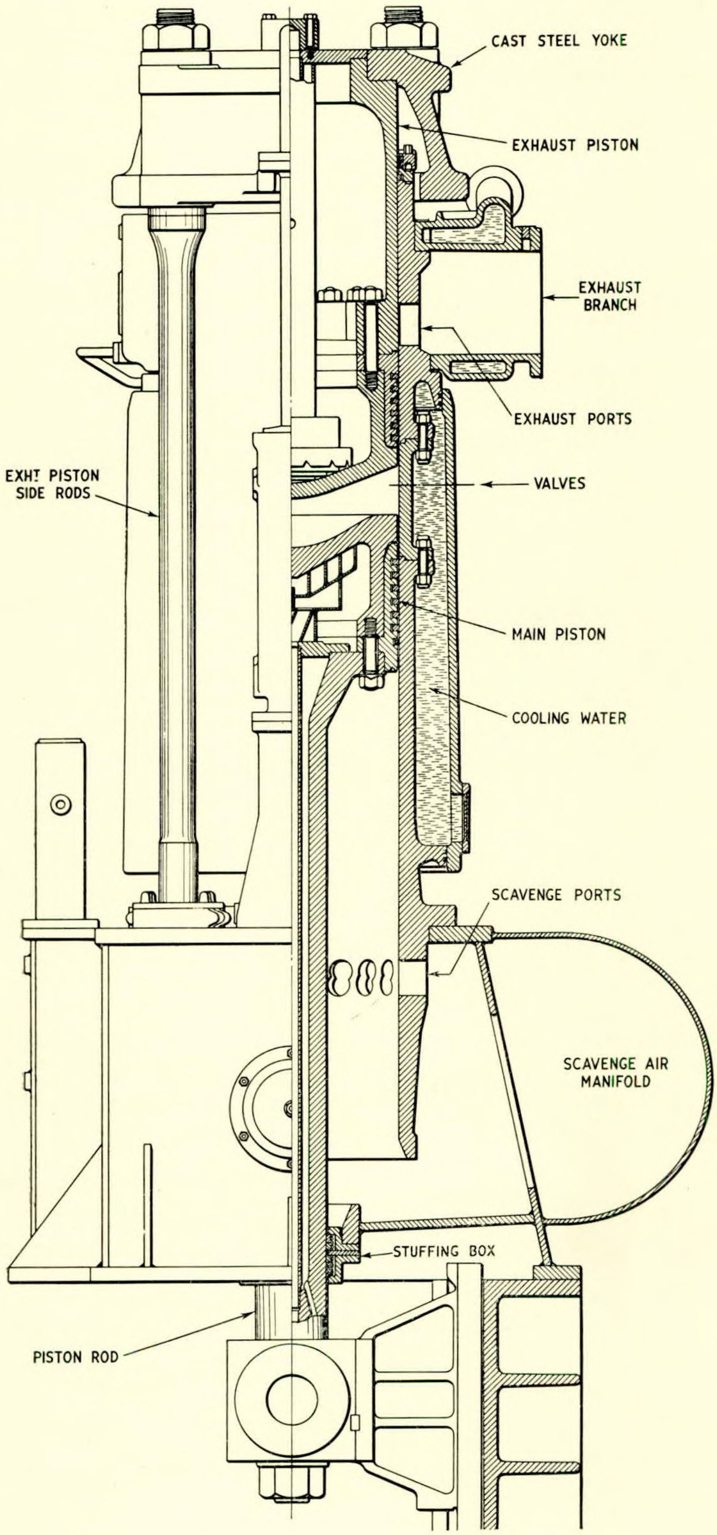


FIG. 3—*Section through cylinder*

1 and 2 are, in general, ⅜in. thick. The top plate is 48 mm., the bottom plate 1in., the bottom flanges 40 mm., and the semi-cylindrical belt ½in. thick. Normally there are two air manifolds per engine length.

Fig. 3 shows a vertical section through a 620-mm. (24·41in.) bore cylinder. The cylinder, which is free to expand vertically upwards, is secured to the top plate of the scavenge air reservoir by twenty 1¼-in. mild steel studs. The cylinder liner, in its present form, consists of three component lengths. These are, respectively: the main, or lower, liner, i.e. the part traversed by the main piston, which is of 0·2 per cent vanadium cast iron and has a length of 1,741 mm. (68·54in.); a centre piece 354 mm. (13·94in.) long, of cast steel, forming the boundary wall of the combustion chamber, in which are arranged the two fuel valves, the starting air valve and the relief valve; and the upper, or exhaust, liner 747 mm. (29·41in.) long, also of alloy cast iron, in which the exhaust piston reciprocates. The spigoted, bolted flanges of the cast steel centre section are constructed with the requisite combination of sturdiness and flexibility which will ensure pressure tightness under all conditions of service; and the twenty-four 1-in. double ended studs of 55/65 tons nickel-chrome-molybdenum steel, pitched at 100 mm. (3·94in.) are fitted with cap nuts of stainless steel. The very carefully ground joint surface is formed on annuli inside and outside of the pitch circle. The cast iron and cast steel flanges are respectively 44 mm. and 35 mm. thick. The normal thickness of the lower and upper cast iron liners is 46 mm., increased to 80 mm. and 62 mm. respectively at the belts of ports in lower and upper liners. The cast steel section is 36 mm. thick.

The liner cast iron has an ultimate tensile strength of 16 tons per sq. in., Brinell hardness 180-200. A hardness numeral as low as 160 or as high as 230 can be accepted, assuming that all other circumstances are propitious. The minimum acceptable tensile strength is 14 tons per sq. in.

Bolted to the open end of the upper liner there is a sealing box composed of two rows of scraper rings and one row of sealing rings, all segmentally divided and all held inwards by garter springs. Behind the rings there is a circumferential channel, with drain to atmosphere.

The lubricating oil admission points on each liner are eight in number. That is: a row of four points is arranged on the main liner—in the foundation flange—and another row of four points is drilled into the exhaust liner between the fourth and fifth piston rings, with the exhaust piston at the top of its stroke. The oil holes are 6-mm. bore, reduced to 3 mm. for the last inch.

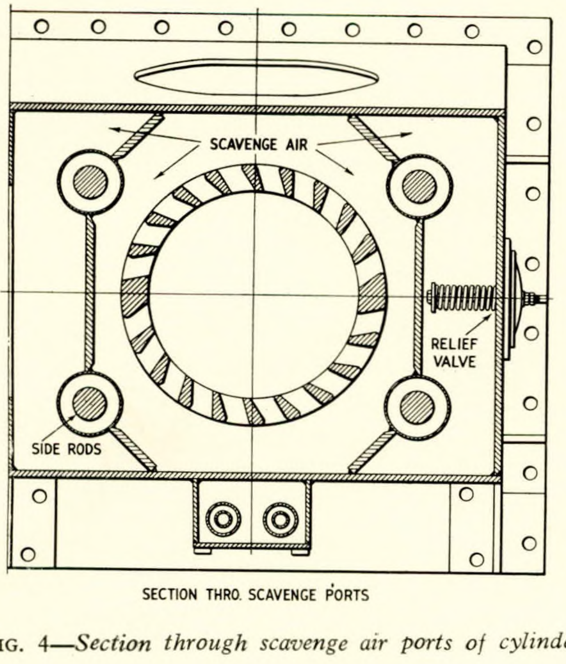


FIG. 4—*Section through scavenge air ports of cylinder*

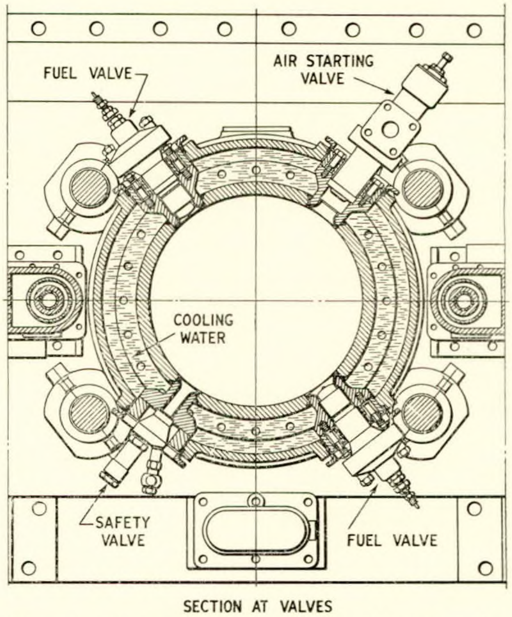


FIG. 5—*Section through combustion chamber*

The cast iron cooling water jacket is of simple construction. Two sets of rubber rings are arranged respectively at the upper and lower ends of the jacket. The cast iron, water jacketed, exhaust belt is held in place by steel tap bolts. The cooling water enters the cylinder jacket near the bottom, wells upwards to the top of the jacket, flows through an external pipe to cool the exhaust belt, thence to flow away through the outlet branch. The fresh water for cooling purposes is undistilled. Other details of Fig. 3 will be self-explanatory.

A horizontal section through the scavenge air ports is illustrated at Fig. 4 and in Fig. 6 the exhaust ports are shown. Both sets of ports are "swirled", as shown in the diagrams, and at both places there are undrilled bands over which the piston ring joints can pass.

The scavenge port area is composed of a single row of twenty ports, each port being formed by two 66-mm. (2·6in.) diameter holes, 37 mm. (1·46in.) centres; that is, the total height of each port is 103 mm. (4·06in.). At bottom deadpoint the rim of the main piston crown is in line with the bottom edge of the ports. The exhaust port area comprises one row of eighteen ports 60 mm. (2·36in.) diameter and one row of twenty ports 60 mm. (2·36in.) diameter, the perimeters of the two rows of ports being separated by 20 mm. (0·79in.). At top deadpoint the rim of the exhaust piston crown is 5 mm. above the top edge of the upper row of ports.

The stroke volume per cylinder is 565·5 litres (19·96 cu. ft.), the clearance volume 41·9 litres (1·48 cu. ft.), the total volume 607·4 litres (21·44 cu. ft.). The gas mean speed through the exhaust ports is 62·6 metres (205ft.) per sec.; the air mean speed through the scavenge ports is 78·65 metres (258ft.) per sec. If $K = A/RV$; where A = exhaust pre-opening area, sq. cm. deg.; R = r.p.m.; V = volume of cylinder after scavenge ports close (litres); then K (ahead) = 0·246 and K (astern) = 0·13.

In the ahead direction the exhaust ports open at 109·2 degrees from top dead centre and close at 239·8 degrees; the air scavenge ports open at 144 degrees and close at 216 degrees. In the astern direction the exhaust ports open at 120·2 degrees and close at 250·8 degrees; the scavenge ports open at 144 degrees and close at 216 degrees. The angle of advance of the eccentrics is 180 degrees + 5·5 degrees.

Fig. 7 shows the port opening diagram for the 750/2,000 engine. In this engine size the eccentric lead angle is 180 degrees + 7 degrees. The gas mean speed through the exhaust

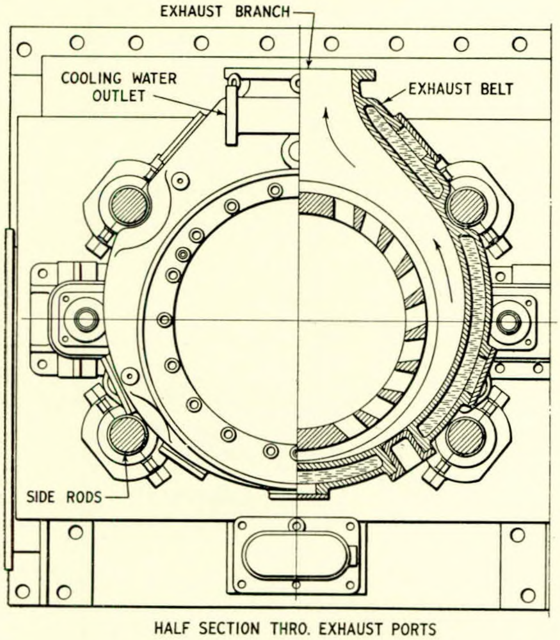


FIG. 6—*Section through cylinder exhaust ports*

ports is 75·5 metres (248ft.) per sec.; the air mean speed through the scavenge ports is 88 metres (289ft.) per sec. That part of the diagram which lies between exhaust opening P (or Q) and scavenge opening R (or S) is the part responsible for the impulses to the exhaust gases. It also determines whether or not the exhaust pressure will fall to sub-scavenge pressure before the scavenge ports are uncovered. The eccentric lead-angle determines the effective stroke of the engine. Angles and angular movement are more important than port areas.

Fig. 5 shows the disposition of the fuel and other valves. Steel branches are welded to the cast steel belt, with distance pieces, glands and stuffing boxes arranged on the cooling water jacket. Into the bores of the valve orifices, stainless steel "flared" sleeves are carefully fitted and then expanded.

The bottom of the scavenge air reservoir, or entablature, forms the top of the crankcase, as shown in Fig. 3. Where the piston rod intersects the reservoir bottom there is provided a gland which serves the double purpose of preventing egress of scavenge air and ingress of crankcase oil. The gland essentially consists of an upper element of two rings, each segmentally divided, and a lower element of three rings, also segmentally divided. Each complete ring is pressed against the piston rod by a garter spring. The complete gland box, which is divided vertically into halves, is bolted to the bottom of the scavenge air belt. Large access doors are arranged on the scavenge air manifold, one at each cylinder; a spring-loaded relief valve is also provided, as indicated in Fig. 4.

The cylinder liner construction illustrated in Fig. 3 may, as experience gathers, be modified and a reversion made to a simpler form.

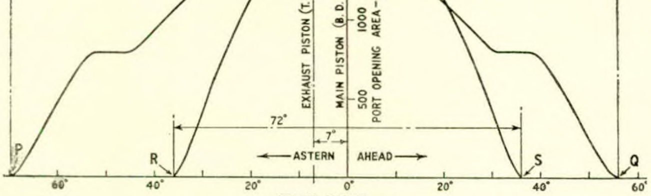


FIG. 7—*Scavenge and exhaust periods*

Vertical steel tubes are welded into the scavenge air manifold for accommodating the exhaust piston side rods, an oil scraper device being arranged on the top of the scavenge belt at the point of entrance of each rod.

Fuel Injection Apparatus

The standard fuel injection system embodies the long-established cam-operated "jerk" pump. The pump works equally well for Diesel oil and for heavy fuels of the whole range up to 3,500 secs. For operating on heavy fuel, the running clearance of the pump plungers is increased slightly; thus, for Diesel oil, the clearance may be 0·0002in. to 0·00025in., whereas for the heaviest fuel it will be 0·00035in. to 0·0004in.; otherwise there is no change. Normally, such fuel pumps run for five years or so without any need for appreciable maintenance. Accordingly there is no urge to make any alteration to the fuel injection system on the score of technical necessity. The substitution of gas compression fuel pumps for cam-operated pumps results, however, in some simplification. The fuel pump camshaft, with bearings and chain drive, is eliminated; but there is no saving in engine length by the discarding of the fuel pump chain drive.

Within the last two years, during shop trials, the Harland-B. and W. gas compression pump—which was originated some years previously—has been applied to several Belfast engines for observational purposes. Two ships are now at sea with such pumps running experimentally. It is only the author's habitual sense of caution which has prevented a more rapid application of the system. The ships mentioned are fitted with the gas compression pumps in addition to the normal jerk pump system. A switchover arrangement is provided.

In the author's opinion, gas compression fuel pumps should be timed. Otherwise, pressures can get out-of-hand, with detrimental effect upon cylinder liners; moreover, in ahead/astern movements, the timed pump is positive in action; also, there is no danger of a double charge of fuel, and it is not necessary to set relief valves any higher than for jerk pumps. The compression-injection pump shown in Fig. 8 is timed.

The gas cylinder, which is fresh water cooled, is made of a special grade of cast iron. It is attached to the cast iron body casing by mild steel studs, and the casing is, in turn, secured to the nitriding steel pump body. The gas piston is made of special quality cast iron and is fitted with high grade cast iron rings, the top ring being of double-seal type. Into the bottom of the gas piston is inserted the spring holder, of hardening steel, which also contains the collar end of the nitriding steel main plunger. A gunmetal fuel regulating sleeve, which encircles the main spindle and the upper end of the pump body, rotates on a collar which is recessed into the body casing. The lower, or release, plunger—also of nitriding steel—abuts against the bottom of the case-hardened steel cam roller guide, which is carried on the end of the pump body. The roller is held to the timing cam by an internal lower spring. The flame-hardened 0·55 carbon steel timing cam is keyed to the lubricator shaft. The flame hardened 0·45 carbon steel roller is carried on an eccentric pin, which permits of vertical adjustment by partial rotation.

Fig. 8 is to scale, but, for simplicity of explanation, there are certain omissions of detail. Thus, on an actual pump, the main and release plungers are carried in liners of special cast iron. The pump unit is made to the same dimensions for the 620/1,870 P.I. and the 750/2,000 non-P.I. cylinders. The gas cylinder is 135 mm. (5·32in.) diameter; the main plunger is 37 mm. (1·46in.) and the release plunger 24 mm. (0·94in.) diameter. The maximum effective stroke of the main plunger is 36 mm. (1·42in.); the total travel of the release plunger is 17 mm. (0·67in.).

The action of the fuel pump is, briefly, as follows; see Fig. 8(a). For priming, the main plunger *(1)* is held at the top of its stroke by the pressure of the spring upon the underside of the gas piston. Fuel enters through the lower suction

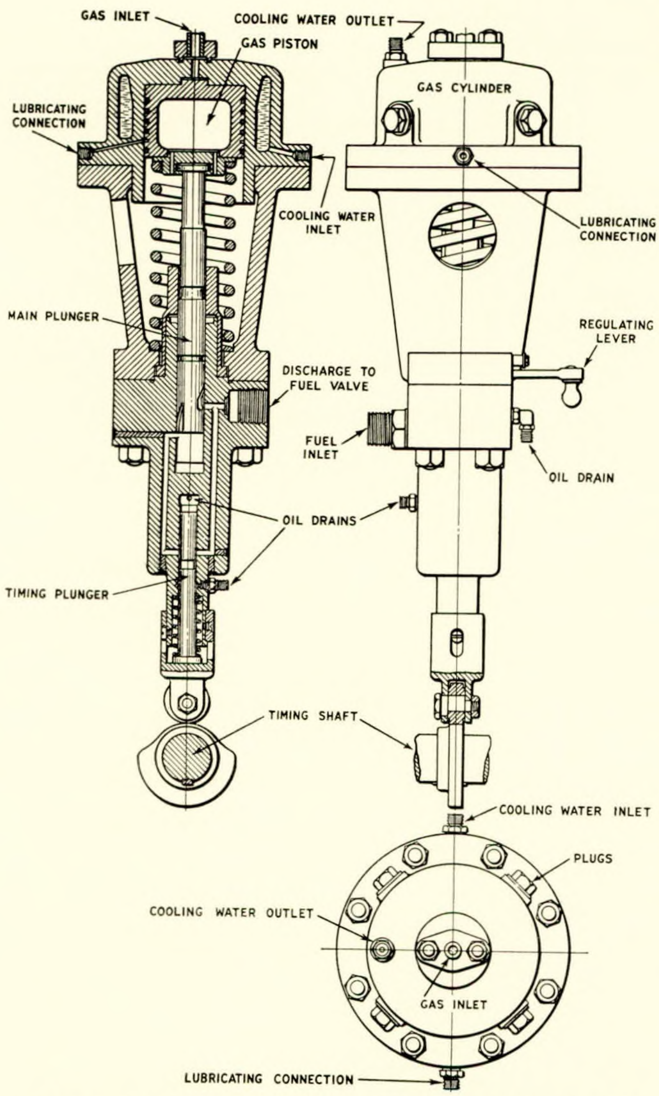


FIG. 8—*Harland and Wolff gas compression fuel pump*

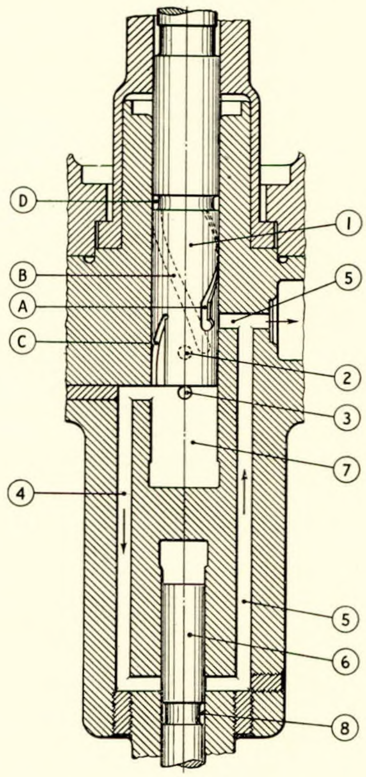


FIG. 8(a)—*Gas compression fuel pump details*

hole *(3)* and passes down the artery *(4)* to the release plunger *(6)*. The release plunger may be masking the access of fuel to the artery *(5)*—depending upon the position of the timing cam. If this be so, complete priming is accomplished by fuel entering the upper suction hole *(2)* by way of helix B, passing around annulus D in the pump plunger to helix A, and so continuing to artery *(5)* and, through the delivery pipe, to the fuel valves. In the "stop" position, helix B is open to the upper suction hole *(2)*, and helix A is open to artery *(5)*. When the engine is running, the two helices are arranged to unmask their respective ports simultaneously, at all positions of the main plunger.

Under running conditions, the rising compression pressure in the engine cylinder acts upon the gas piston. When the compression pressure has overcome the initial compression of the spring, the main plunger *(1)* begins to descend. Its movement continues until its bottom edge masks the bottom of the lower suction hole *(3)*. During this part of the travel the release plunger *(6)* masks communication between *(4)* and *(5)*, and fuel is therefore displaced back into the suction through the hole *(3)*. This movement is not part of the effective stroke of the pump plunger.

As the timing cam rotates, the release plunger *(6)* moves upward. Meanwhile the main plunger, having masked the lower suction hole *(3)*, is momentarily arrested. That is, the increasing gas cylinder pressure acts upon a trapped volume of fuel in chamber *(7)* and artery *(4)*. At a predetermined point—about 12 degrees before top dead centre—the upper edge of release plunger annulus *(8)* unmasks the lower edge of arteries *(4)* and *(5)*. The fuel, now under high pressure, travels through the delivery pipe to the fuel valves and injection at the combustion chamber begins.

The main plunger now continues its descent until helix A unmasks the discharge hole *(5)*, thus permitting the high pressure fuel to be suddenly spilled into helix A, then around annulus D to helix B and so on into suction hole *(2)*, thus terminating the period of fuel injection with a snappy closing of the fuel valve. The effective stroke of the main plunger varies in accordance with the power requirements of the engine, becoming greater with increase of load and *vice versa*. The main plunger spills for a very small portion of the stroke, then helix C masks the delivery hole *(4)*, thus trapping a quantity of fuel in chamber *(7)* and cushioning the downward travel of the main plunger. As the gas pressure in the engine cylinder falls, the main plunger is forced back to the upper limit of its stroke by the action of the spring against the underside of the gas piston. The pump is now ready for another cycle to begin.

In Fig. 9 a gas compression pump, with its connexions, is shown in place on the engine cylinder and, in Fig. 10, a fuel valve, in section, is illustrated.

The fuel valve body is a steel forging; the inner body which contains the spindle is nitralloy steel; the end nut is mild steel. The construction of the normal type of fuel inlet pipe connexion is shown, together with an alternative design. The fuel filter is of standard form. The pressure impulse of the fuel lifts the spindle against the loading of the spring. The nozzle is of non-shrink steel, ground to the nut; the flat top of the nozzle forms the seat for the end of the non-shrink steel spindle. The collar of the end nut is ground to the cylinder liner valve pocket. The valve is either water cooled or oil cooled, as may be required. Inlet and outlet connexions, shown in the plan view, lead to longitudinal holes in the valve body which constitute the circulating system. The fuel section of the valve is tested to 10,000lb. per sq. in.; the cooling space A is oil tested to 100lb. per sq. in. An air relief valve is provided for use with Diesel oil fuel only.

The silico-manganese steel spring is held between a tool steel cap and a case-hardened steel sleeve screwed into the valve body. The lift adjusting rod is contained in a steel tube, the latter limiting the lift of the spindle. The assembly of the cap for the lift adjusting rod is shown separately.

The same fuel valve is used for the 620/1,870 engine as for the 750/2,000 engine. The spindle, 14 mm. diameter, is the same. For the 620-mm. cylinder, there are four holes 0·035in. diameter, 2·5 mm. long, in each nozzle; the central hole is 2·8 mm. diameter, 20·5 mm. long; the total valve lift is 0·75 mm. For the 750-mm. cylinder, there are four holes, 0·043in. diameter, 3·1 mm. long, in each nozzle; the central hole is 3·0 mm. diameter, 23 mm. long; the total valve lift is 1·0 mm. Three of the spray holes are drilled in the same direction as the air swirl; the fourth is drilled in the opposite direction, thereby obtaining a more intimate admixture of fuel and air.

The length of the nozzle holes is about three times the diameter, experience having shown that this proportion provides good penetration. The bore of the central hole is designed for a fuel speed of about one-quarter of that through the spray holes. The central hole is as short as possible, to minimize dribbling; the actual length is determined by design considerations. The speed of the fuel through the spray holes normally lies between 200 and 220 metres per sec.; the fuel injection pressure, at full load, is usually about 6,000lb. per sq. in. The differential area of the fuel valve spindle is arranged in conjunction with the valve spring, so that the valve will open at a pressure of approximately one-half of the injection pressure. The spring ensures quick valve closure at the end of injection.

The axis of the fuel valve is set horizontally and the spray holes are arranged fanwise, with a downward dip, for directing the spray away from the relatively slow-moving exhaust piston and to prevent it from impinging upon piston surfaces.

If all spray holes are arranged to pass through the centre of the nozzle pip radius, they may overlap at their entrance to the central hole. To avoid this, some of the holes are offset in the drilling, although remaining parallel to the radial angle.

Bedplate

The bedplate is a fabricated steel plate structure and, as

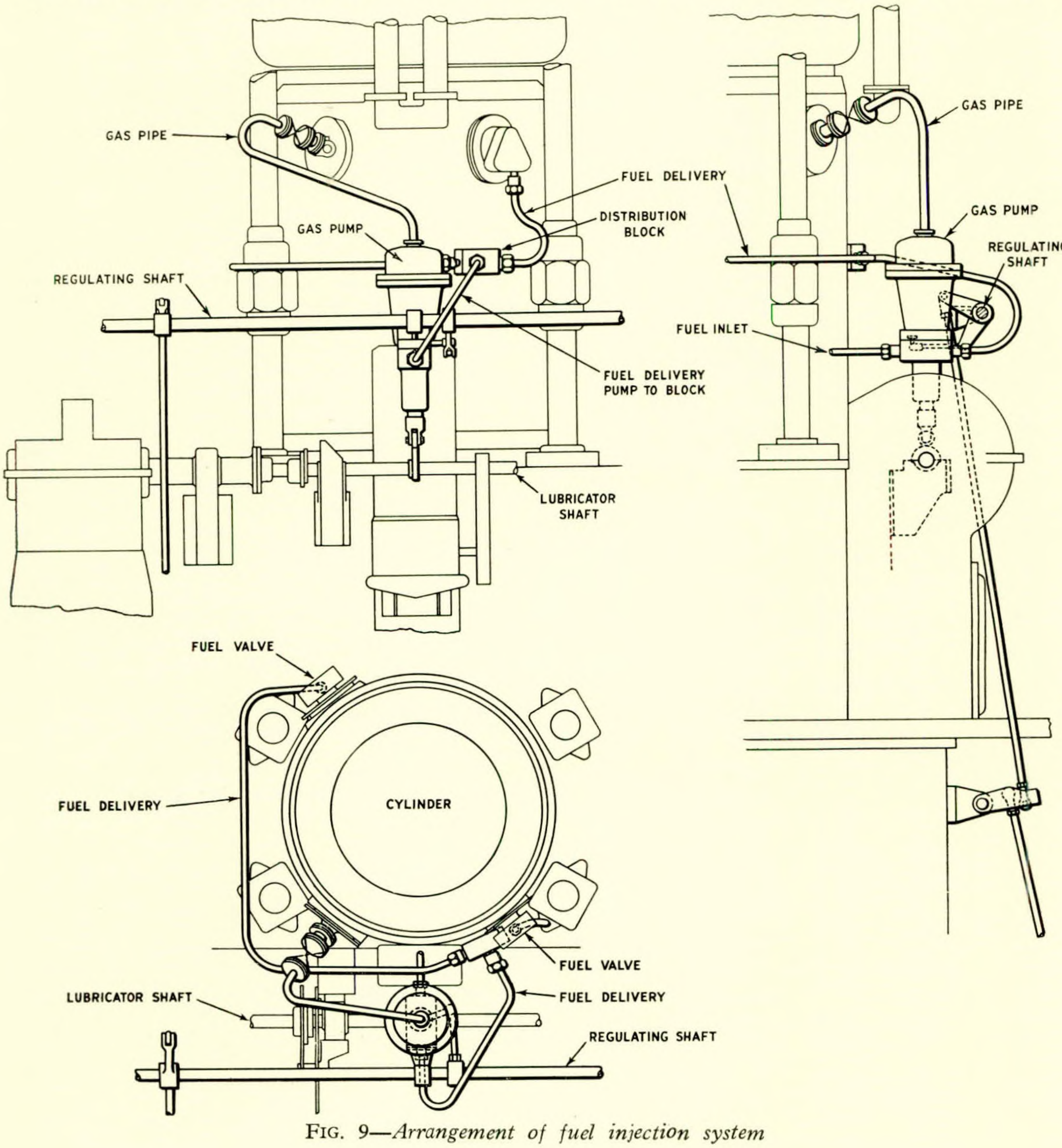


FIG. 9—*Arrangement of fuel injection system*

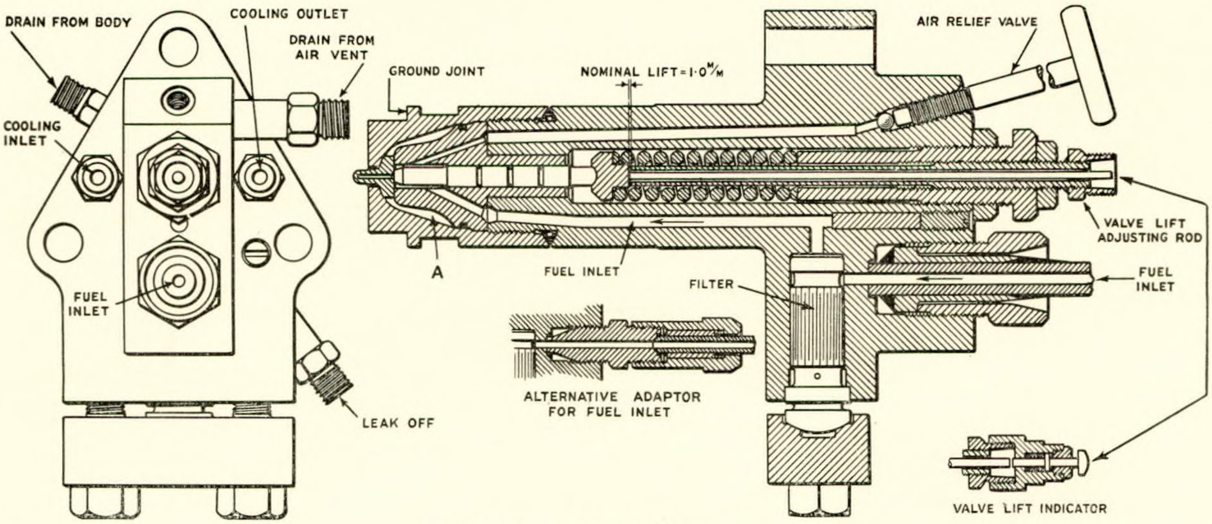


FIG. 10—*Fuel injection valve*

will be seen from Figs. 11 and 12, the transverse girders are continuous plates extending right across the bedplate, the fore-and-aft girders being short lengths welded to the transverse girders. Continuous lengths of plate constitute the top and bottom flanges of the longitudinal girders and these are stiffened vertically by two sloping ribs under each "A" frame. Two horizontal intermediate ribs stiffen the longitudinal girders. The top and bottom plates of the transverse girders are welded to the respective plates of the fore-and-aft girders.

The main bearing housings are steel castings. A heavy tapered circumferential rib underneath each housing ensures an effective weld to the transverse girder. Four sloping ribs on each face of the girder transmit a share of the load from the bearing housing to the bottom flange. Short vertical ribs stiffen the top flanges of the transverse girder.

It will be noticed that the ribs on the transverse girder are thicker than the plate. It is the author's opinion that if, for sustaining a heavy load, a plate requires ribbing, then the ribs should be of substantial dimensions.

In the bedplate of Figs. 11 and 12, the vertical plates of transverse and longitudinal girders are 1in. and $\frac{7}{8}$in. thick respectively. The top and bottom plates of the longitudinal girders are of respective thickness 42 mm. and 42·5 mm. (mean); the sloping and horizontal stiffeners are $\frac{7}{8}$in. thick. The diagonal and other ribs of the transverse girders are $1\frac{1}{8}$in. thick; the top and bottom plates are respectively 42 mm. and $1\frac{3}{4}$in. The bolted end flanges of each bedplate length are 40 mm. thick; the bolting consists of forty $1\frac{3}{4}$-in. mild steel bolts, of which twelve are fitted bolts. A $\frac{3}{8}$-in. steel plate welded to the transverse and longitudinal bottom flanges con-

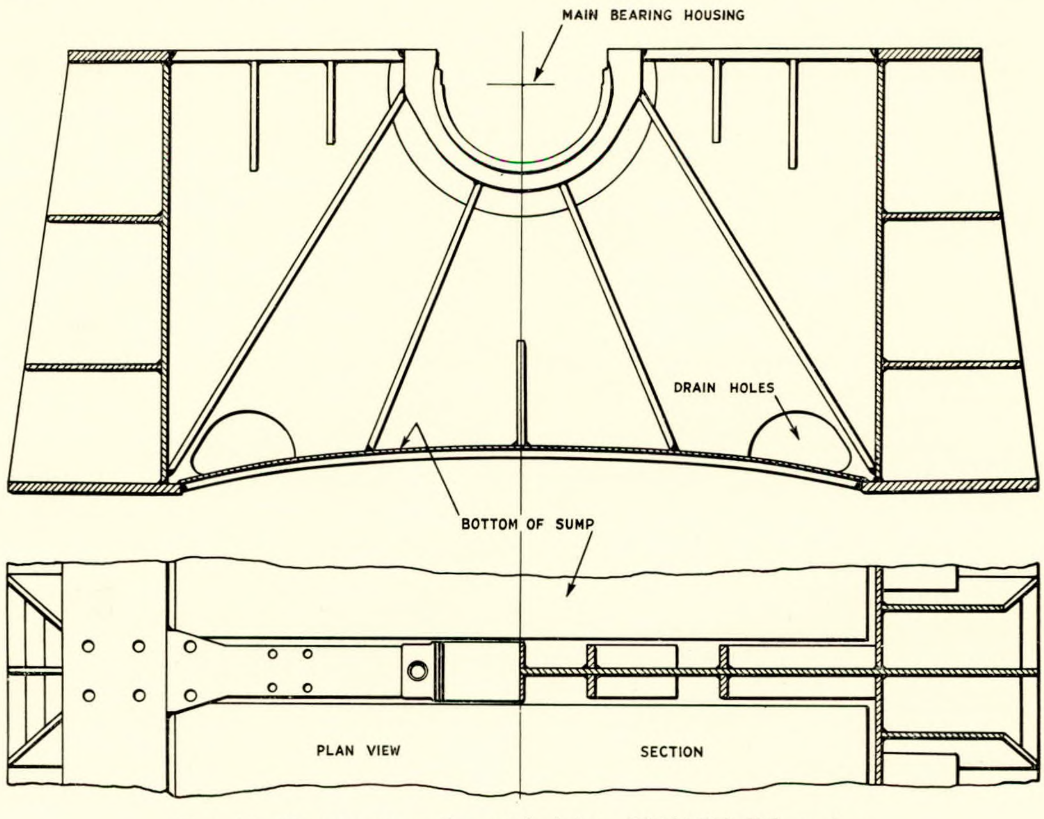


FIG. 11—*Transverse section of bedplate, 750/2,000 P.I. engine*

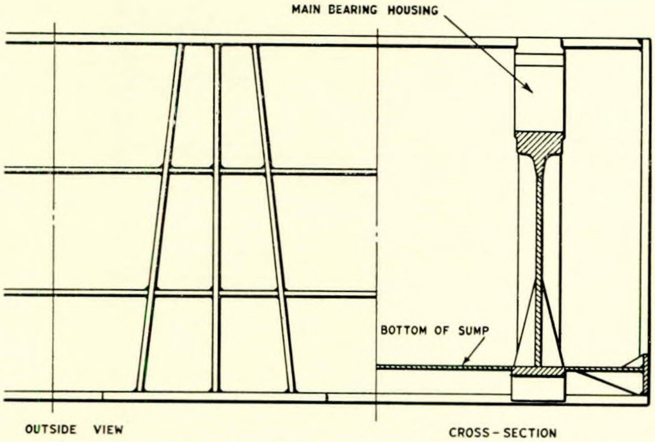


FIG. 12—*Longitudinal view of bedplate in 750/2,000 P.I. engine*

stitutes the oil sump bottom and large drain holes in the transverse girders allow the free flow of lubricating oil along the sump to the pump suctions.

The bottom of each bedplate is fully machined, with a 5-mm. slope downwards towards the centre, thereby facilitating the fitting of the cast iron chocks, which are similarly tapered. The number of holding-down bolts is adequate, but not over-generous. For example, for an eight-cylinder 750/2,000 engine there are 180 to 190 $1\frac{7}{8}$-in. diameter forged steel bolts. A steel side chock is arranged at each side of each transverse girder, welded to the tank top. There is also a chock at each end of each longitudinal girder. Steel wedges are driven between the chocks and the bedplate girder flanges.

After completion, each bedplate is stress relieved and thoroughly shot blasted.

Main Bearings

The construction of the main bearings is indicated in Fig. 13. The top and bottom half-bushes are of cast steel; the keep comprises two cast steel components, each held in place by two mild steel studs 3in. diameter. The ratio of bearing length to journal diameter varies from 0·5 : 1 to 0·6 : 1. In Fig. 13 the ratio is 0·58 : 1.

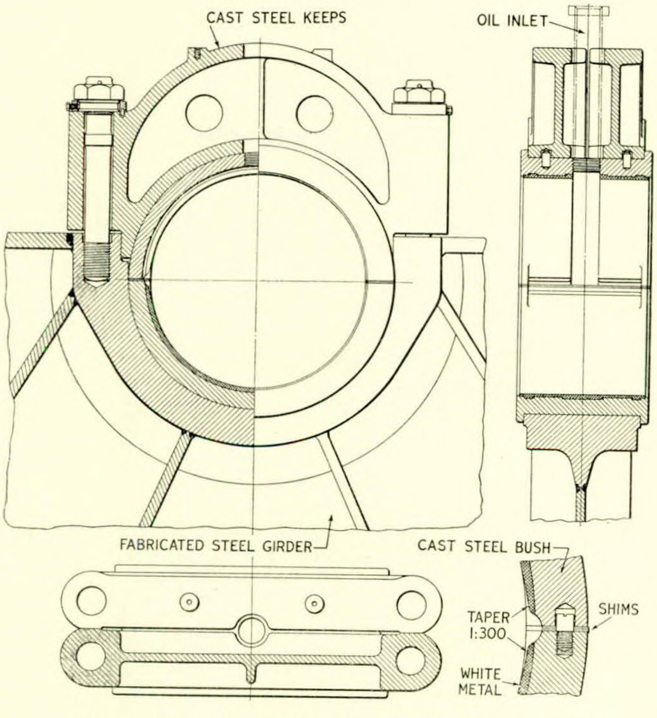


FIG. 13—*Main bearing in 750/2,000 P.I. engine*

The general features of the design are reproduced, approximately, in most other bearings of the engine. The white metal is 8 mm. (0·32in.) thick, reinforced by 4 mm. thick circumferential strips in the middle and near the edges of the bearing, minimizing the chances of boundary disintegration. The steel lip at each end of the bearing is 4 mm. below the surface of the white metal. There are six horizontal 4 mm. thick reinforcing strips—it would hardly be accurate to call them dovetails—on each half-bush. The sheet brass shims range from 0·2 mm. to 2 mm. thick; the initial aggregate thickness is 4 mm. (0·16in.).

Lubricating oil is admitted through a $2\frac{1}{2}$-in. bore pipe at the top of the bearing, whence it is led by a circumferential channel 76 mm. (3in.) wide, 21 mm. (0·83in.) deep, to a horizontal, semicircular-shaped oil reservoir, 15 mm. (0·59in.) radius, at each side of the bearing. The oil reservoir has tapered entrances to the lower and upper half-bushes, the taper starting at 0·1-mm. clearance and disappearing at 30 mm. The lower

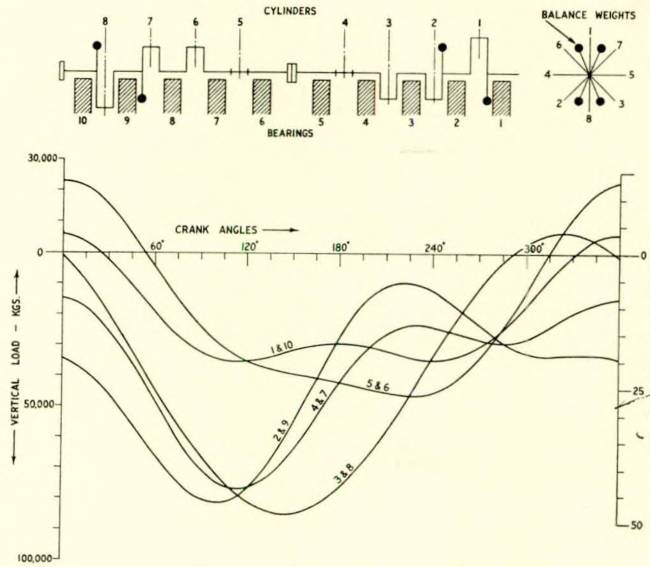


FIG. 14—*Main bearing loads for an eight-cylinder engine*

half-bush is plain and free from grooves and oil holes.

The centre of the outer circumference of the bushes is 1 mm. eccentric to the centre of the crankshaft. That is, when looking forward from the aft end of a twin-screw installation, the vertical centre line of each bedplate is 1 mm. outboard of the crankshaft centre line. For a single-screw ship the bedplate centre line is 1 mm. to starboard of the crankshaft centre line.

Because the cylinder gas pressure at the main and exhaust pistons is equal and opposite, the load on the main bearings is approximately that set up by the moving parts. Fig. 14 shows the main bearing loads for an eight-cylinder 750/2,000 engine. The diagram is self-explanatory.

Crankshaft

The crankshaft is of standard form. Fig. 15 shows a typical, fully built crank. The crankwebs are steel castings of 28-32 tons per sq. in. ultimate tensile strength, 25 per cent minimum elongation; the sum of the tensile and elongation figures must be 57 minimum; the yield point is required to be not less than one-half of the ultimate tensile strength. The journal pieces and crankpins are Siemens-Martin mild steel 28-32 tons per sq. in ultimate tensile strength. The shrinkage allowance is, as nearly as possible, one six-hundredth of the

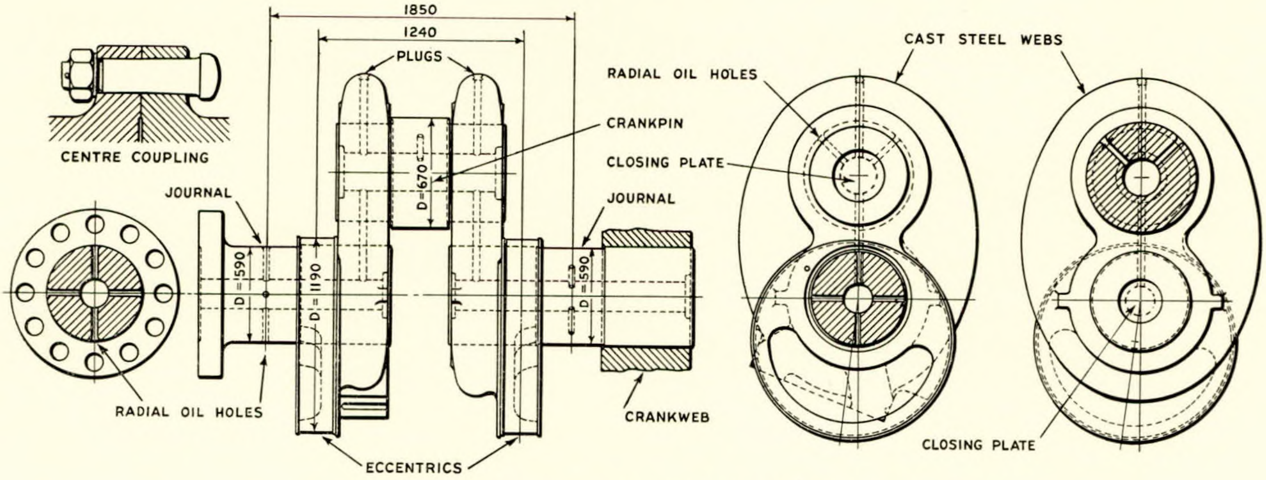


FIG. 15—*Fully-built crank in 750/2,000 P.I. engine*

diameter of the crankpin or journal piece, inside the crankweb. Thus, for a four-cylinder engine, the shaft diameter is 585 mm. and the hole 584·03 mm. Heating for shrinkage is done in an oven, not in a gas heater; each web is heated once only, the crankpin and journal piece being entered simultaneously.

The crankweb castings are heat treated twice. First, they are fully annealed after casting; second, they are stress relieved at 600 to 650 deg. C. after rough-machining—which includes

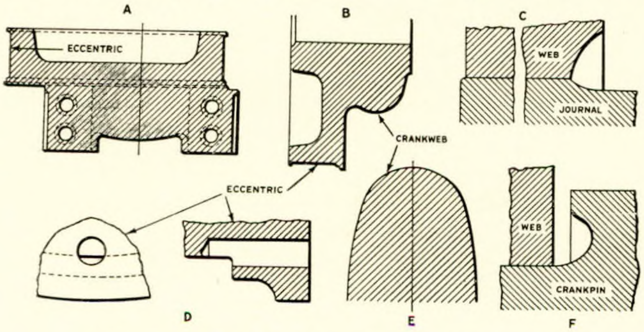


FIG. 16—*Details of crankshaft construction*

the boring of holes for journal pieces and crankpins. The crankweb contour, shown in Fig. 15 and at (E) in Fig. 16, has been standard for over twenty years; it ensures equalization of cooling stresses. Sections of the crankweb at the eccentric sheave are shown at (A) and (B) in Fig. 16.

A point to be watched in fully built crank designs is the thickness of the ligament between the holes. Thus, in Fig. 15, the centres are 750 mm. (29·53in.), the journal piece and crankpin diameters—in the web—are each 600 mm. (23·62in.), leaving a ligament of 150 mm. (5·91in.). To preserve this ligament and, at the same time, to obtain the bottom end bearing surface needed by the design—without spreading the cylinders—the crankpin diameter is stepped-up to 670 mm. (26·38in.), in the manner indicated at (F) in Fig. 16. The small circumferential space between the face of the web and the end of the crankpin bearing surface is intended to facilitate inspection. It was originally requested by one of the survey authorities.

To ensure a large and gradual fillet at the change of journal piece diameter where it enters the crankweb, the web is machined as indicated at (C) in Fig. 16.

Balance weights are normally made of cast iron, of 16-17 tons per sq. in. ultimate tensile, a material which, years ago, used to be called semi-steel, because of the addition of steel scrap to the cast iron mixture. The balance weights are secured to the crankwebs by the double process of heavy steel studs and stout countersunk dowel pins, each—by itself—being strong enough safely to withstand the maximum centrifugal and bending forces which could be engendered. This arrangement is economical in terms of first cost; but, sometimes, shipowners may specify integrally cast balance weights.

The eccentric sheaves are finished-machined after the crankshaft is finally assembled. Special care is taken to ensure that the two sheaves for each crank are machined true to each other and parallel to the crank journals. The maximum permitted tolerance is extremely small. The surfaces of journals and crankpins are highly finished in the machine and then polished. Eccentric sheaves are very carefully lapped and then given a high polish. The author does not favour the flame hardening of crankshaft surfaces.

Four 34-mm. oil holes are arranged radially at the mid-length of each main bearing, the holes being rounded by a 10 mm. radius; and two 34-mm. radial oil holes are also arranged in the upper half of each crankpin, at 45 degrees on each side of the vertical centre line. A 45-mm. oil hole is drilled vertic-

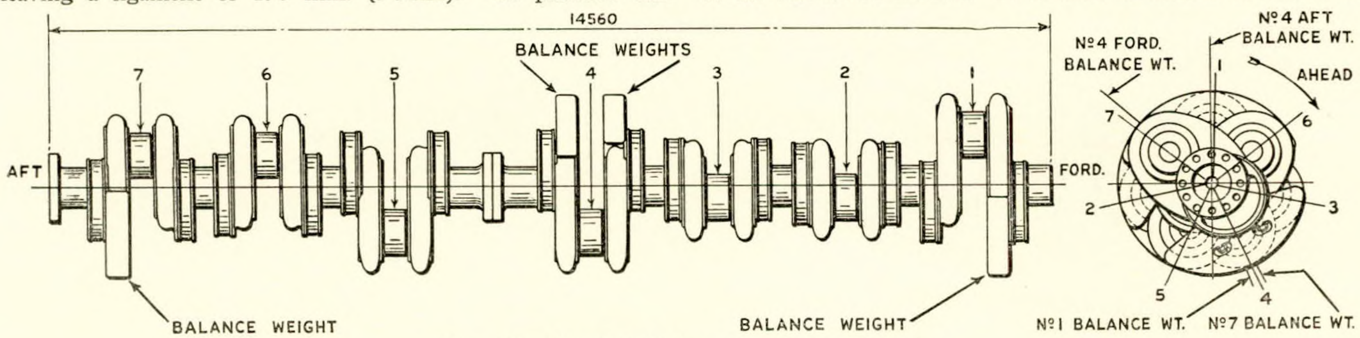


FIG. 17—*Crankshaft for seven-cylinder engine*

ally through each crankweb to connect the hollow journal piece with the hollow crankpin, the external end of the oil hole being plugged. For the removal of main bearing bottom half bushes, a 30-mm. hole, 40-mm. deep, is drilled in each web on the side adjacent to the main bearing. With the crank vertical, the holes are drilled at 30 degrees above the horizontal. Pins are inserted into these holes and the crankshaft is slowly rotated by the turning gear. As mentioned earlier, in connexion with main bearings, the centre of the bushes is 1 mm. eccentric to the centre of the crankshaft.

Crankshafts for four and five cylinders are built in one length: for six, seven and eight cylinders, in two lengths. Sometimes the five-cylinder shaft is made in two lengths. A ten-cylinder crankshaft would be made in two lengths. By way of example: Fig. 17 shows the crankshaft of a seven-cylinder engine, which is a commonly favoured size. It is 14,560 mm. (47ft. 9·23in.) overall and weighs 114 metric tons. The journal diameter is 590 mm. (23·23in.). An eight-cylinder crankshaft has an overall length of 16,410 mm. (53ft. 10in.) and a finished weight of 126 metric tons.

Table I classifies the usual arrangement of cranks.

TABLE I.—CRANK ARRANGEMENTS

Number of cylinders	Crank sequence (ahead)	Angle between cranks, degrees
4	1.4.3.2	90
5	1.5.2.3.4	72
5	1.2.4.5.3	72
6	1.5.3.6.2.4	60
7	1.7.2.5.4.3.6	51·43
8	1.6.4.2.8.3.5.7	45

Balance weights are arranged as deemed to be desirable. Thus, Figs. 18, 19 and 20 show the balance weight system for the seven-cylinder crankshaft of Fig. 17. Crank 1 has a single balance weight of 2·55 metric tons on the forward web; crank 4 has a balance weight of 3·04 metric tons on each of the webs; crank 7 has a balance weight of 2·55 metric tons on the after web. The reason for the rather complicated balance weight arrangement is simple. The balance weight needed for crank 1 is not heavy enough to justify subdivision, hence it is bolted to the forward web. For the same reason the balance weight for crank 7 is arranged on the after web. On crank 4 the required counterpoise, which is much heavier, is divided into two weights, these weights being mutually offset instead of being in line, thereby assisting the balance of each half-crankshaft. Each balance weight is secured by four 3-in. studs and two 40-mm. diameter steel dowels, with ends securely riveted over. Each pair of studs and nuts is secured against movement by a common, solid steel pin, 15-mm. diameter.

For certain combinations of cranks in engines located right aft, the incidence of torsional critical stresses can make necessary the substitution of semi-built cranks—with their less heavy revolving masses—for fully-built cranks. Each pair of crankwebs and the associated crankpin then constitute one steel casting. An equivalent result can, however, be achieved by introducing unequal crank angles and retaining fully-built shafts. The turning moment is then somewhat less regular than with the normal and preferred arrangement of cranks; but it is fully satisfactory.

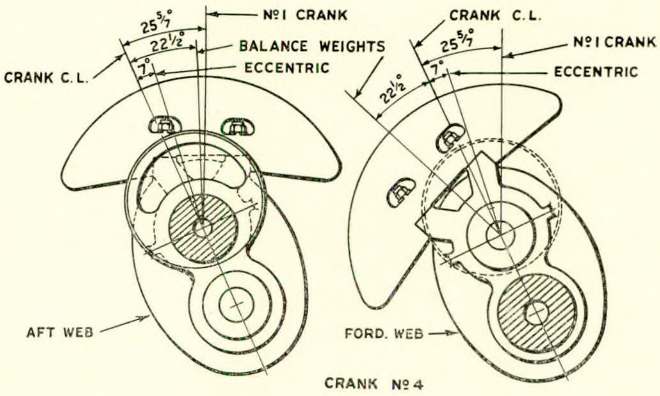


FIG. 19—*Balance weights on middle crank*

No crankshaft is designed for a strength less than 10 per cent above survey requirements. For convenience of manufacture the crankshaft components for six, seven and eight cylinders are made dimensionally the same. This equivalence extends to diameters and lengths of journals and crankpins, crankweb thicknesses, and so on. Crankshafts for four and five cylinders are devised similarly.

Each eccentric leads its crank by 180 degrees + 7 degrees when running ahead, and by 180 degrees − 7 degrees when running astern; this is for the 750/2,000 engine. For the 620/1,870 engine, the respective angles are 180 degrees + 5·5 degrees and 180 degrees − 5·5 degrees. As mentioned elsewhere, these angular leads are compromises to ensure maximum obtainable lag between exhaust port opening and scavenge port opening when running ahead, without jeopardizing the adequacy of the astern lag.

Figs 21 to 25 show the respective residual unbalanced couples in 620/1,870 engines having four, five, six, seven and eight cranks. The diagrams are drawn for 120 r.p.m. In all the engines the primary and secondary forces are balanced; and in the six- and eight-cylinder engines there are no secondary couples. For engines 35ft. to 45ft. in length and weighing, say, 325 to 425 tons, unbalanced couples of 50 tons ft.—as shown by the diagrams—are negligible. In other words, the running balance of the engine type is very satisfactory.

Turning moment diagrams for five and six crank engines of 620/1,870 size are shown in Fig. 26. In Fig. 27 similar diagrams are given for seven and eight crank engines. The diagrams are based on 120 r.p.m. and 7·65 kg. per sq. cm. (108·81lb. per sq. in.) m.i.p., this being a continuous sea service rating.

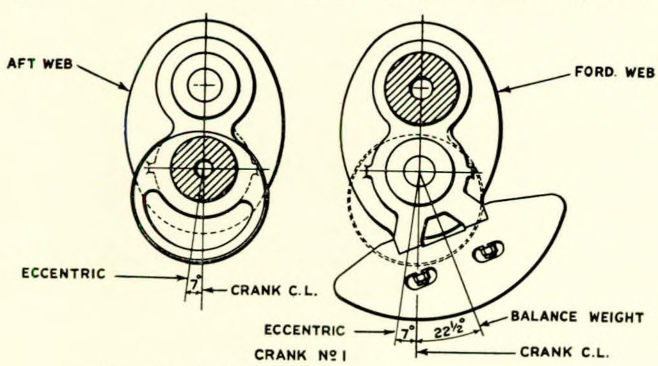


FIG. 18—*Balance weights on foremost crank*

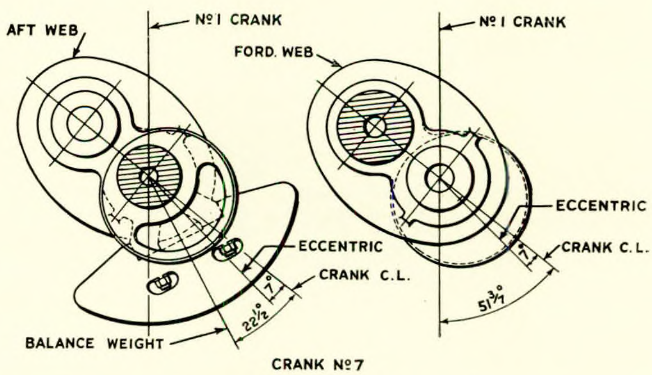


FIG. 20—*Balance weights on aftermost crank*

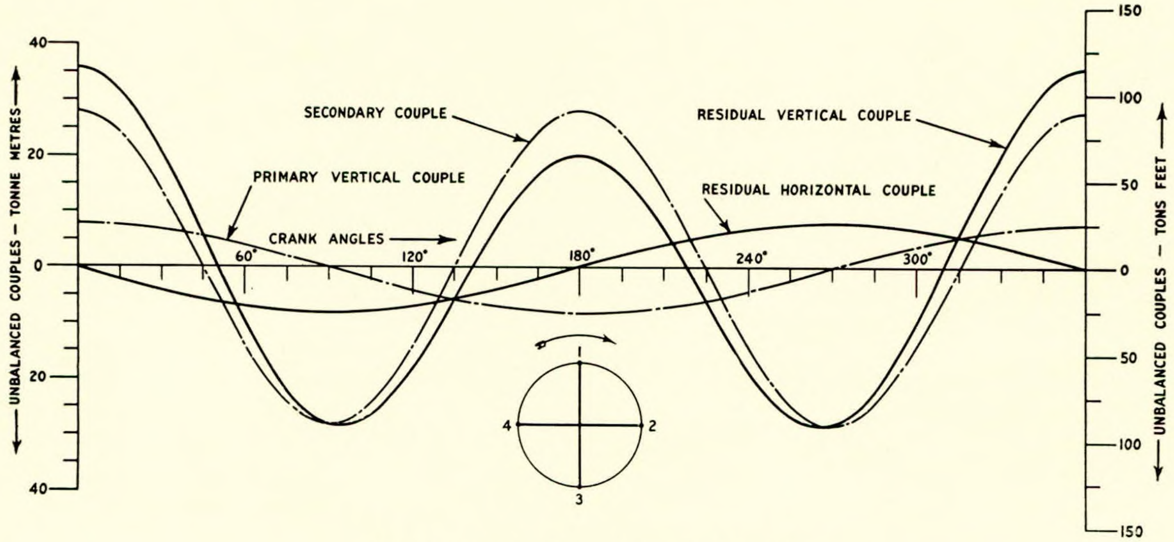


FIG. 21—*Balancing of four-crank engine*

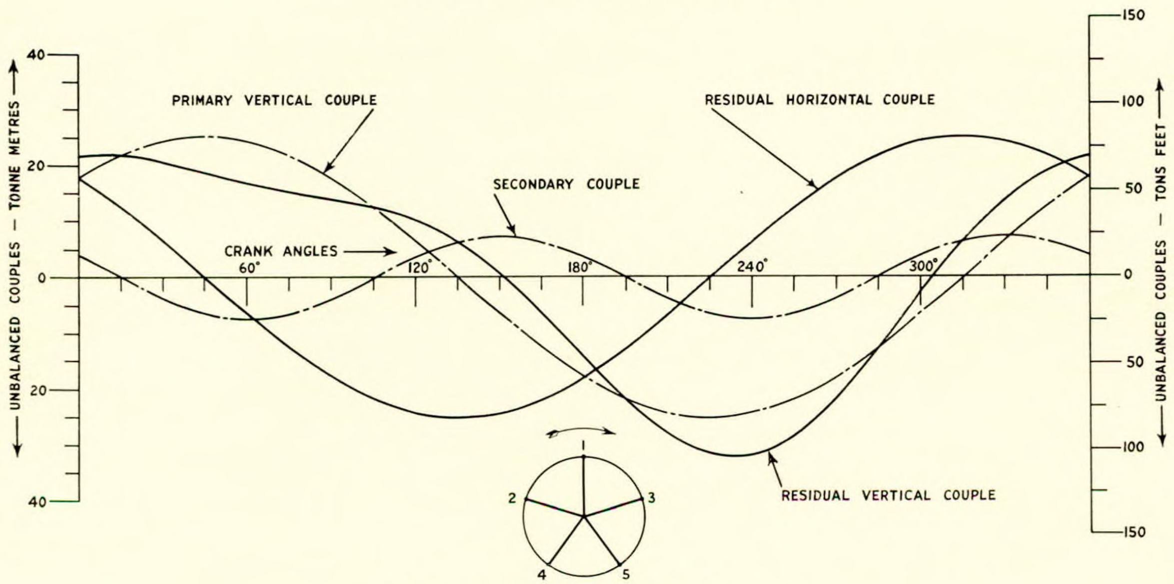


FIG. 22—*Balancing of five-crank engine*

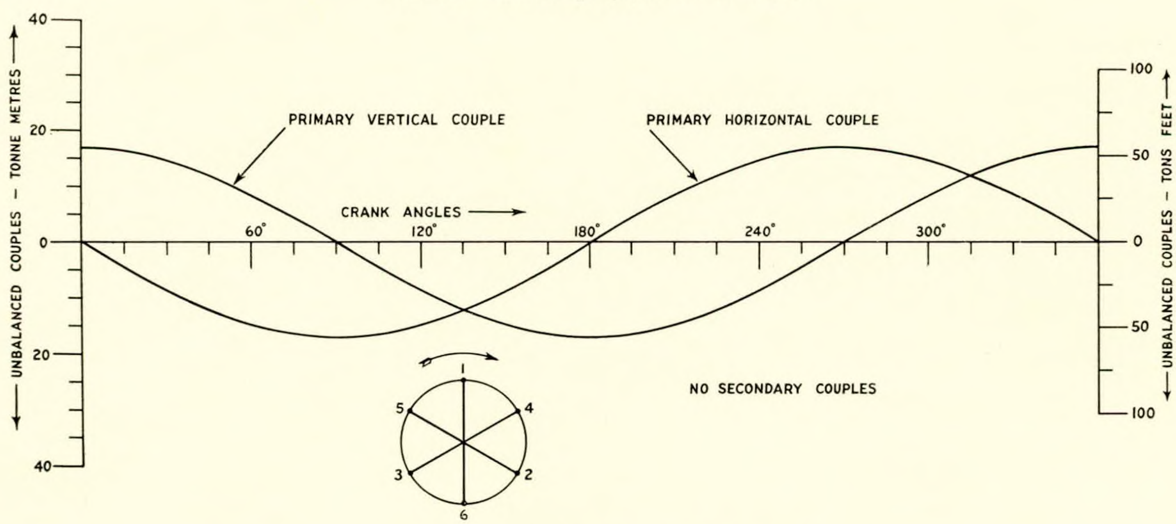


FIG. 23—*Balancing of six-crank engine*

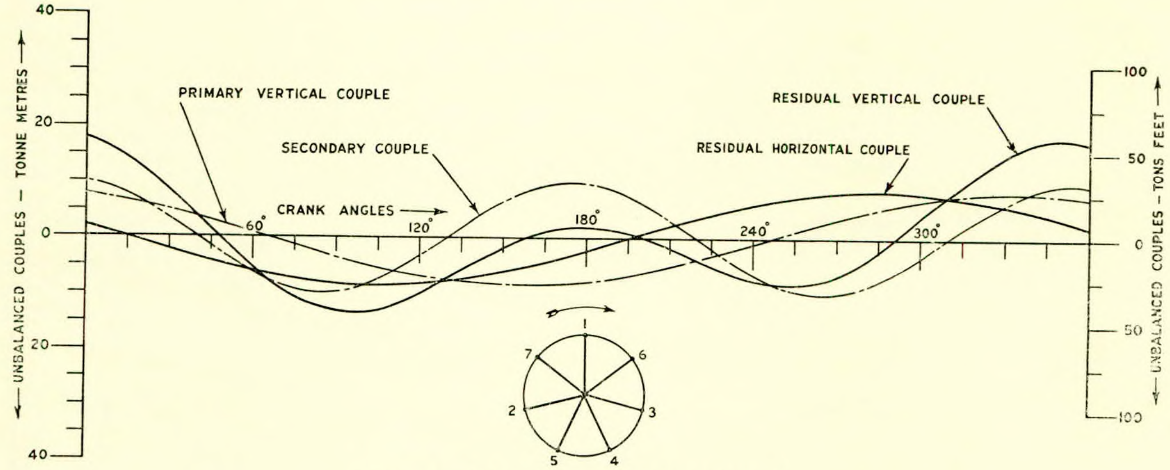


FIG. 24—*Balancing of seven-crank engine*

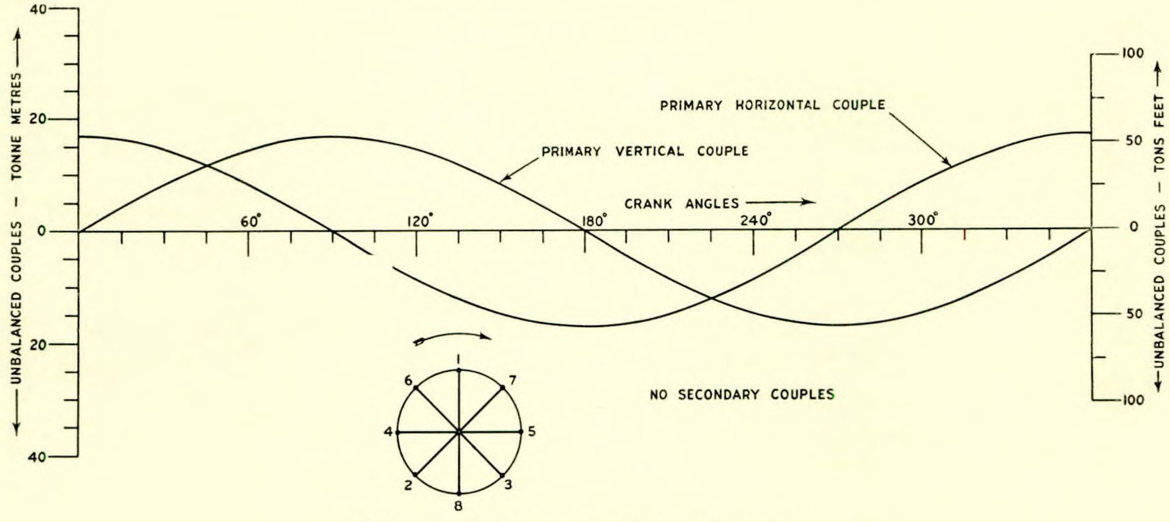


FIG. 25—*Balancing of eight-crank engine*

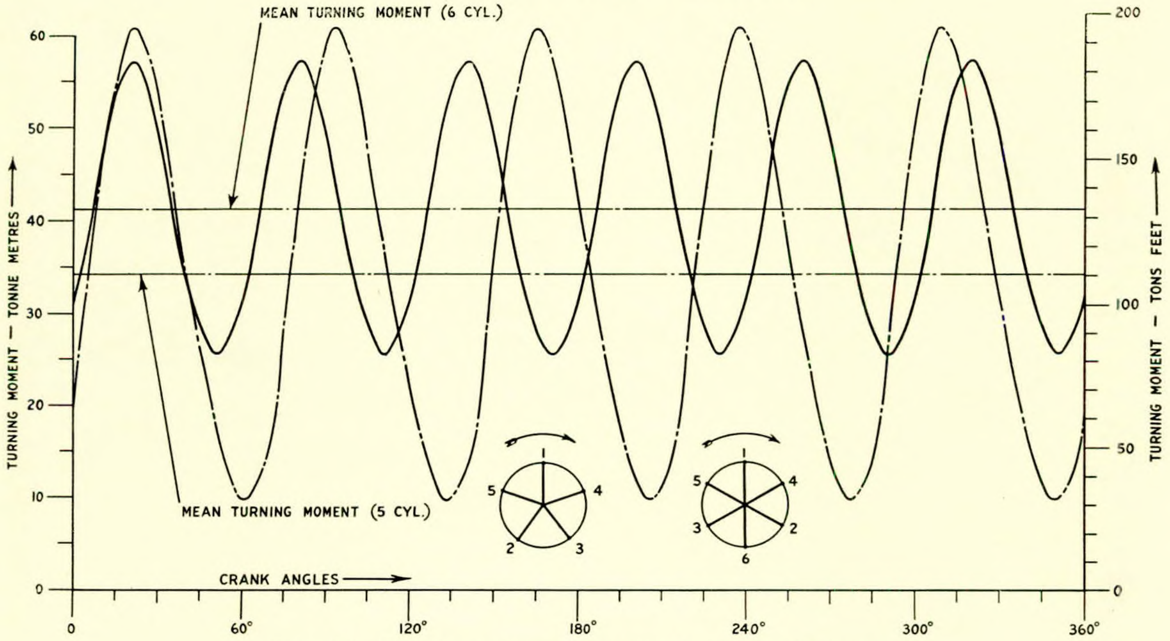


FIG. 26—*Turning moment diagrams for five and six cranks*

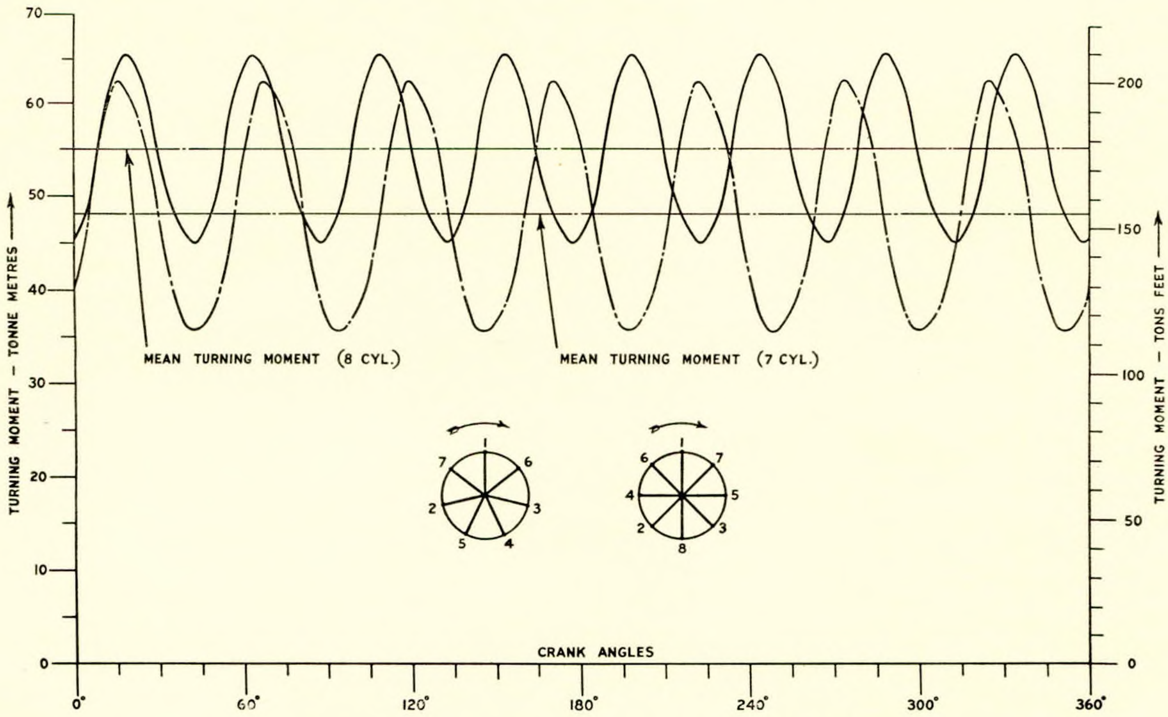


FIG. 27—*Turning moment diagrams for seven and eight cranks*

Connecting Rod

The connecting rod is of mild steel, of the builders' conventional design. The body of the rod is parallel, with a longitudinal hole almost one-half of the rod diameter in size. When calculated as a strut, based on the maximum piston load and on the ultimate strength of the material, with appropriate allowances for the form of the ends and the swing of the rod, the factor of safety is approximately 5·5 to 6·5. The ratio of length to diameter varies from 12:1 to 14:1. The ratio of connecting rod length to crank radius is 4:1. As the result of experiences, many years ago, with shorter rods, a ratio of 4:1 is the minimum that the author would offer to any shipping company. For the 750/2,000 engine the connecting rod is 250 mm. (9·84in.) diameter, a ratio of length to diameter of 12:1.

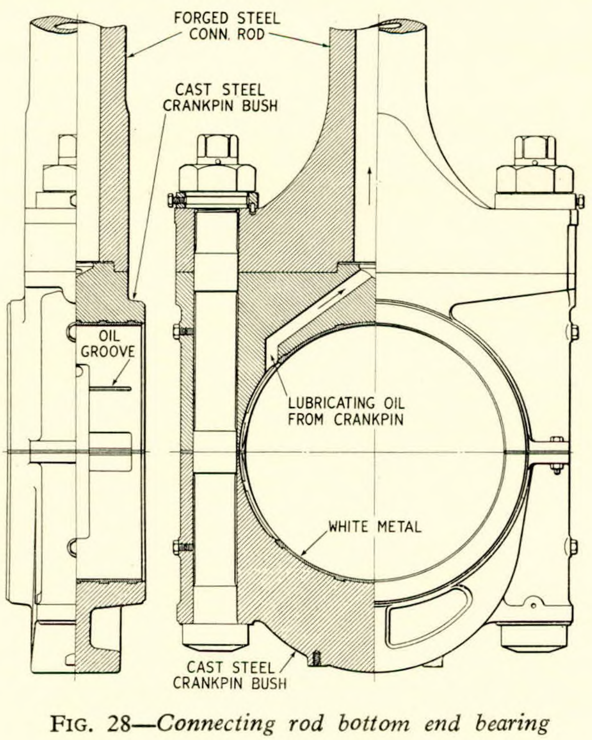


FIG. 28—*Connecting rod bottom end bearing*

Bottom End Bearing

Fig. 28 shows a connecting rod bottom end bearing of 670 mm. (26·38in.) diameter. The ratio of length to diameter of bearing is normally 0·375:1 to 0·5:1. The bushes are of cast steel; the $4\frac{1}{2}$-in. diameter bottom end bolts are forged mild steel, the nuts being low carbon steel, i.e. as near to wrought iron as is commercially possible. There is a longitudinal hole 46 mm. (1·81in.) diameter in each bolt; this is reduced to 16 mm. (0·63in.) in way of the thread. The pur-

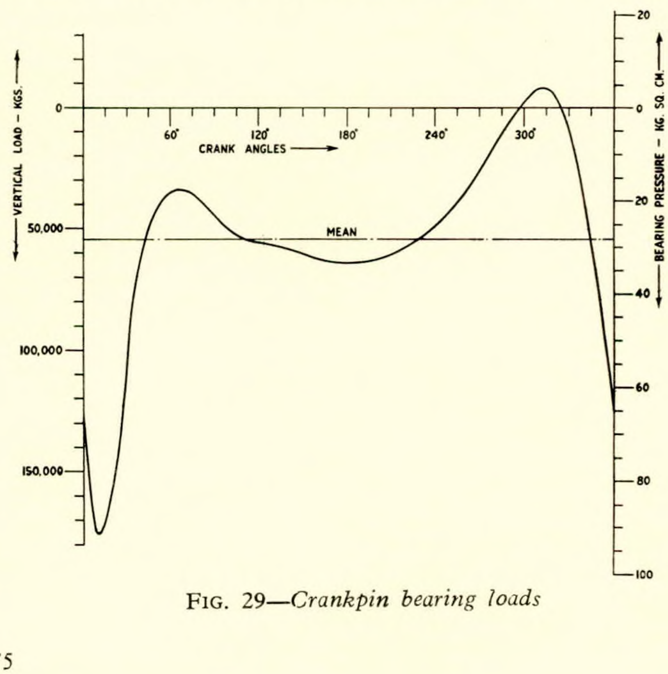


FIG. 29—*Crankpin bearing loads*

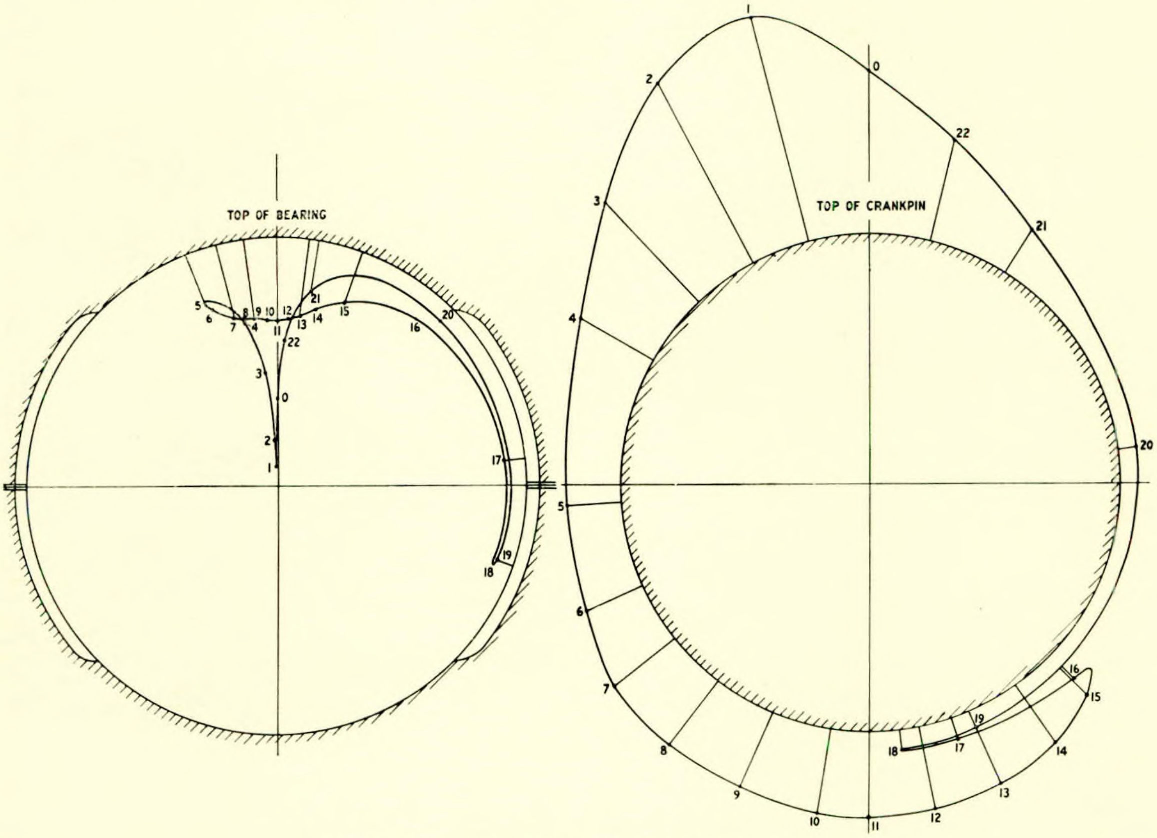


FIG. 30—*Crankpin and bottom end bearing loads*

pose of this differential hole is to increase the relative strength at the thread. To facilitate tightening-up, a hexagon is machined on the upper end of each bolt. As in other parts of the engine, a Penn type locking ring is arranged under each nut; there is also a split pin. Small tap bolts take the weight of the bottom end bolts in place and hold the halves of the bushes together.

The cast steel upper half-bush is of substantial thickness, being one-fifth of the crankpin diameter. The surfaces of connecting rod foot and upper bush, also the bolt and joint surfaces, are accurately machined and hand scraped. The white metal is 8 mm. (0·32in.) thick.

On each side of the bushes there is a circumferential oil channel, 70 mm. (2·76in.) by 17 mm. (0·67in.), subtending about 90 deg. of arc, i.e. 45 deg. above and 45 deg. below the horizontal centre line. Lubricating oil flows into this channel from the radial holes in the crankpin—at least one hole being always uncovered to the channel—thence into the

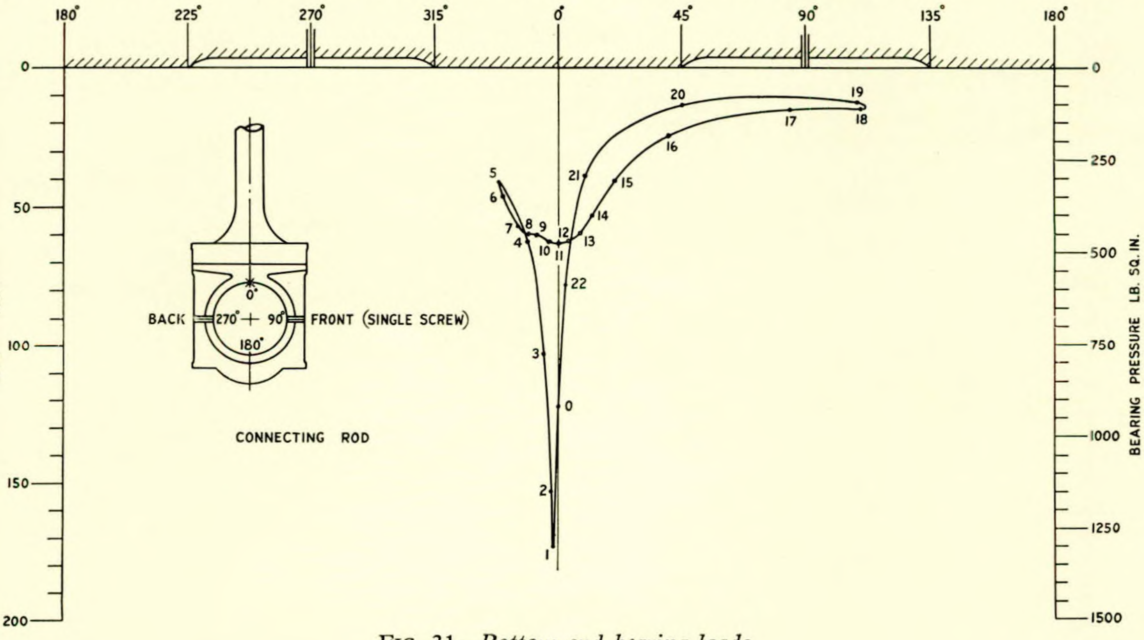


FIG. 31—*Bottom end bearing loads*

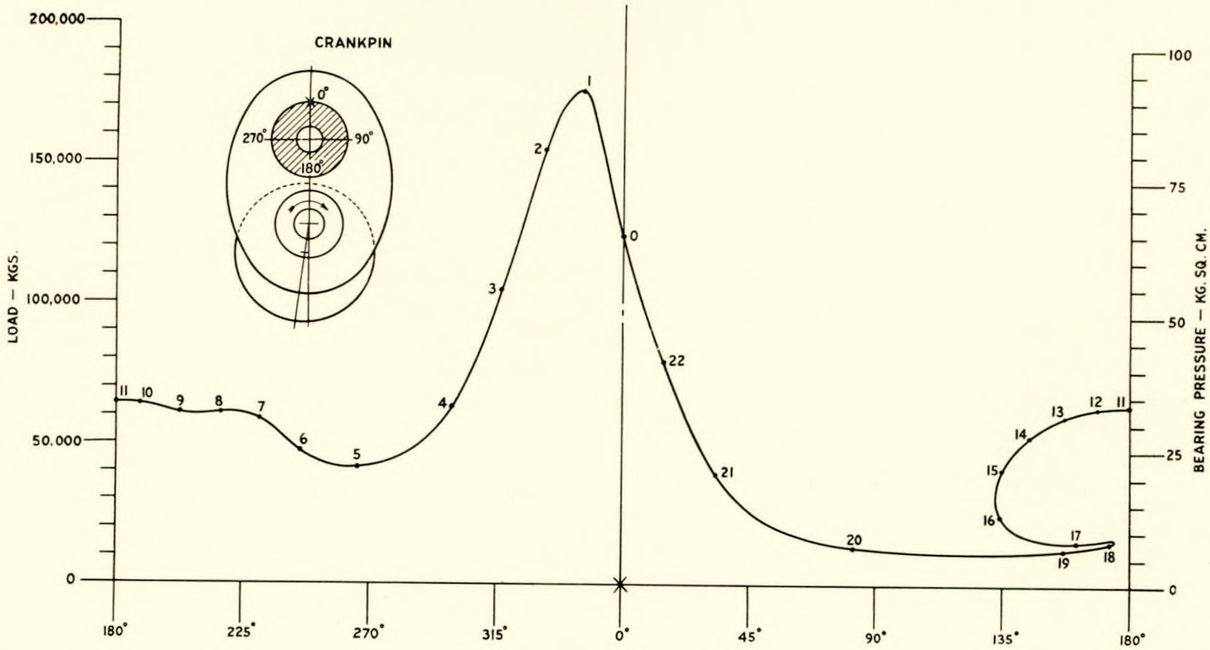


FIG. 32—*Crankpin loads*

horizontal oil reservoirs at the sides of the bushes, whence it is induced into the bearing. The oil escapes at the upper ends of the circumferential channel, through a sloping hole on each side of the vertical centre line, into the longitudinal hole in the connecting rod. There are two horizontal oil grooves in the upper half-bush, which is the surface sustaining the piston load.

Fig. 29 shows the usual form of diagram for the connecting rod bottom end bearing loads. The diagram is based on 112 r.p.m. and 7·5 kg. per sq. cm. (106·7lb. per sq. in.) m.i.p. Actually, however, the bottom end bearing oscillates and the crankpin revolves. Accordingly the diagrams of Fig. 30 show in a more realistic manner the magnitude, and distribution, of the loading on the crankpin bearing and on the crankpin. Diagrams prepared in this form can make clearer the reasons for local wear of white metal surfaces and the tendency to ovality in crankpins. In Fig. 30, 1 mm. of actual diagram = 5,500 kg.

In Figs. 31 and 32 the diagrams of Fig. 30 are redrawn, the circumference of the crankpin being expanded to a straight line base.

In calculating bearing pressures, the load is assumed to be taken by a projected area = bearing length × 0·87 diameter. From the projected area oil grooves and channels are deducted to obtain a net area. This rule is observed for all the bearings of the engine. For the main bearings the loading is a maximum when the revolutions are a maximum, because the forces are essentially inertia forces. The calculations are therefore based upon 120 r.p.m., i.e. the maximum for the engine. But for all the other bearings, e.g. connecting rod bearings, eccentric straps and gudgeon pin bushes, where the loads are the algebraic sum of piston pressures and inertia forces, the greatest loads are realized when the revolutions are below the maximum; hence 112 r.p.m. are assumed as a basis for the diagrams.

Top End Bearing

A top end bearing is illustrated in Fig. 33, the crosshead pins being 430 mm. (16·93in.) diameter. The bushes are of cast steel, the crowns being of ample thickness. The top end bolts 3in. diameter—which are omitted from Fig. 33—are of forged mild steel, the construction being not unlike that of the bottom end bolts of Fig. 28. A 33-mm. (1·30in.) hole is bored through each bolt, becoming 10 mm. (0·39in.) at the screwed end. The white metal lining is 8 mm. (0·32in.) thick.

Lubricating oil is fed to the lower half-bushes by the ascending holes from the connecting rod. The oil flows through a horizontal hole 24-mm. diameter and rises into the white metal surface, at bearing mid-length, through four vertical holes 12 mm. diameter and two vertical holes 20 mm. diameter. At the top of each hole there is a horizontal oil groove. At each end of each groove there is a 6-mm. hole drilled vertically through the lip of the lower bush, whence the oil freely escapes to ensure a continuous flow through the bushes in all circumstances.

Except for about 90 degrees of bearing surface at the bottom, there is a circumferential oil channel which extends from the uppermost oil groove in the lower bush at one side to the uppermost oil groove in the lower bush at the other side. There is an oil reservoir at each side of the horizontal centre line and there are four horizontal grooves in the upper half-bush.

The working load is taken by the lower bushes. This load varies in intensity throughout the cycle, but is always downward. Fig. 34 shows the magnitude and variation of the loads and pressures on the top end bearings, at 112 r.p.m. and 7·5 kg. per sq. cm. (106·7lb.. per sq. in.) m.i.p.

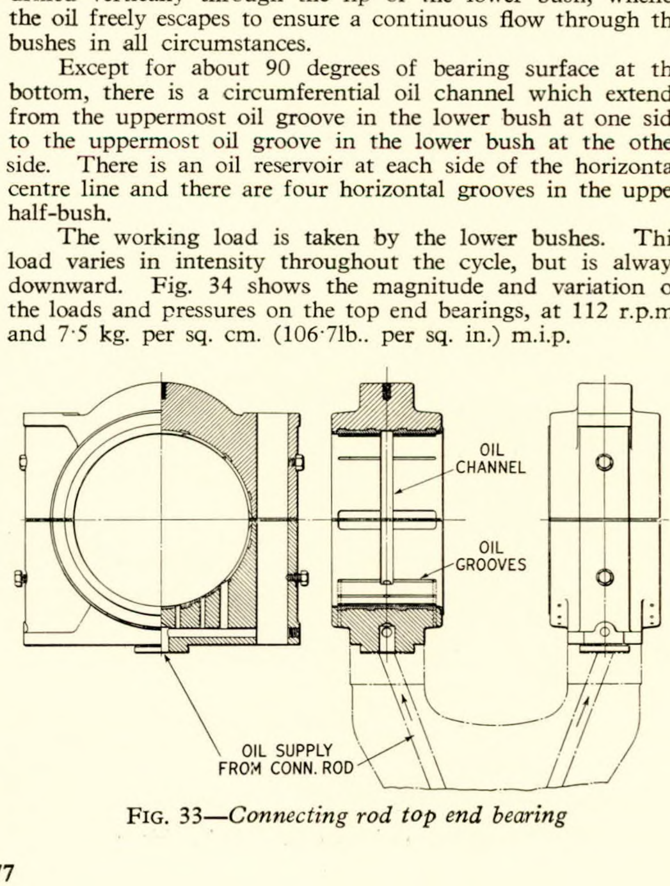


FIG. 33—*Connecting rod top end bearing*

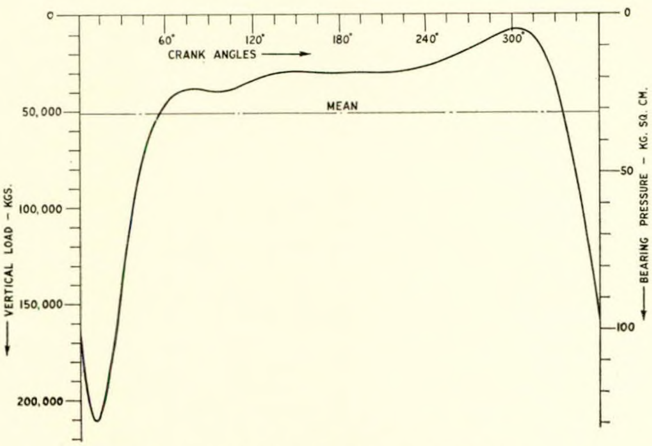


FIG. 34—*Loads on top end bearings*

Piston Rod and Crosshead

Fig. 35 shows a crosshead, piston rod end and guide shoe. The crosshead is of forged steel, 40-44 tons per sq. in ult. tens., 24-20 per cent elongation, with carbon content not less than 0·45 per cent.

The oscillatory movement of the connecting rod top end necessitates unusual care being given to the surface of the crosshead pins and to the lubrication of them. The crosshead pins must be truly circular and highly polished, these being complementary conditions. To obtain a surface hard enough to ensure a very high finish, the crosshead pins are flame hardened. Subsequently they are ground and superfinished to 3-6 micro-inches. For flame hardening, the carbon content of the steel must be at least 0·45 per cent.

The piston load is sustained by the collar which is formed at the lower end of the piston rod by reducing the diameter for the screwed end. As the upward inertia forces are insignificant in comparison with the continuous downward piston loads, there is no hammering at the collar surface. Had there been any suspicion of load reversal, the author would have discarded the collar and have substituted a fitted cone. The screwed end is 148 mm. (5·83in.) major and 139·87 mm. (5·51in.) minor diameter, 4 threads per in. The rod thread is truncated at 147·26 mm. (5·80in.) diameter and the nut at 140·61 mm. (5·54in.). The threads are ground and rolled. The piston rod nut is of low carbon mild steel.

Based upon the ultimate strength of the material and on the maximum piston load, the factor of safety of the mild steel piston rod, when treated as a long column with appropriate allowance for end attachments, is approximately 7·3. The factor of safety varies from 6·25 to 7·5. The ratio of length to diameter varies from 9 : 1 to 11 : 1. The piston rod and the connecting rod diameters are usually the same. Both rods are hollow, the longitudinal hole being almost one-half of the external diameter. In way of the screwed end the hole in the piston rod is reduced to one-third of the diameter at that place.

The guide shoe surfaces, ahead and astern, also the edges, are faced with white metal 6 mm. (0·24in.) thick. Lubricating oil enters the crosshead from the telescopic pipe connexion and, in addition to flowing upwards to the piston, flows horizontally through two holes to supply the eight guide shoe oil holes, which are located at the horizontal oil grooves. The white metal is taper-scraped from 0·12 mm. to 0 at a distance of 20 mm.

In Fig. 36 the guide shoe is shown in position relative to the eccentric rod crosshead shoes. The gross ahead surface of the guide shoe is 5,250 sq. cm. (813·75 sq. in.); the gross astern surface is also 5,250 sq. cm. (813·75 sq. in.). The variation

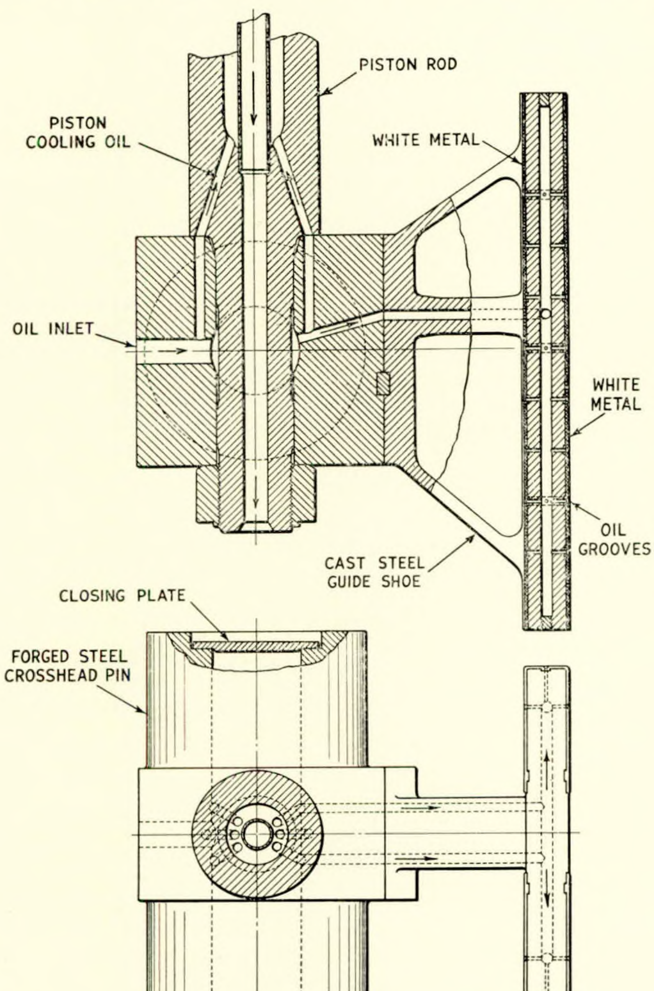


FIG. 35—*Piston rod, crosshead and guide shoe*

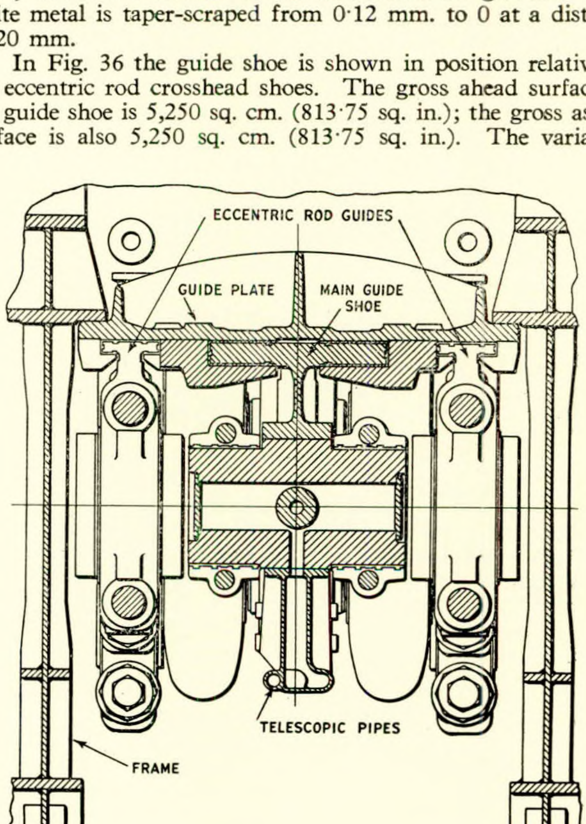


FIG. 36—*Arrangement of running gear*

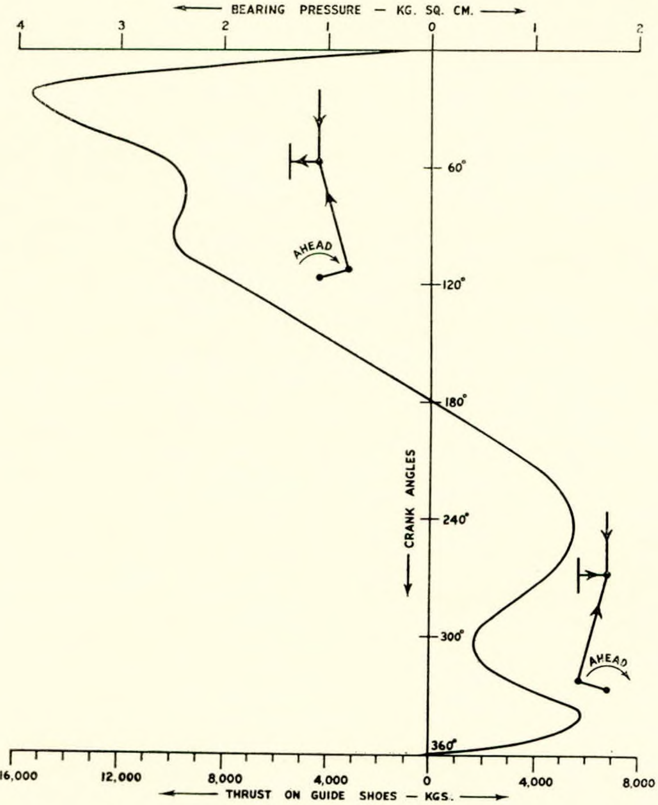


FIG. 37—*Thrust loads on guide shoes, ahead running*

in guide pressure is indicated in Fig. 37, this diagram being drawn for 112 r.p.m. and 7·5 kg. per sq. cm. (106·7lb. per sq. in.) m.i.p. It is instructive to note that, when running ahead, there is a reversal of loading. That is, for part of the cycle the pressure is taken by the astern guide.

The details shown in Figs. 28 to 49 are for the 750/2,000 engine.

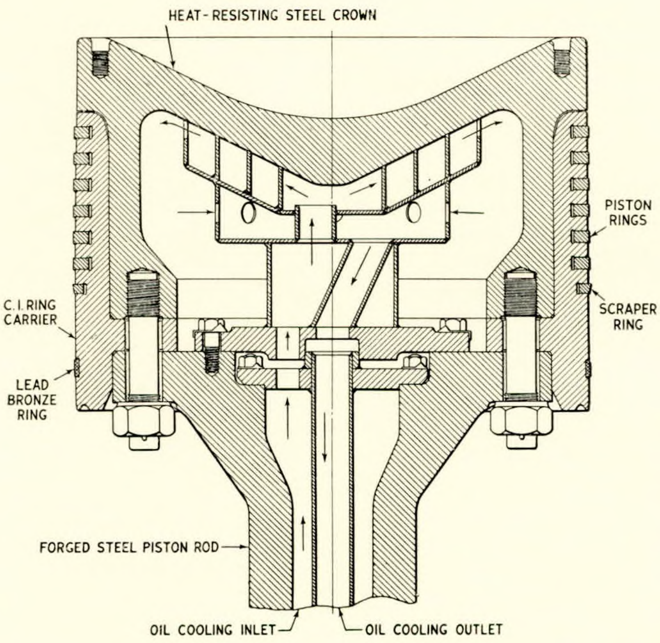


FIG. 38—*Section through main piston*

Pistons

Fig. 38 shows a 750-mm. (29·53in.) diameter main piston in section. The crown, which is 100 mm. (3·94in.) thick at the centre, tapering to 60 mm. (2·36in.) at the rim, is a high alloy chrome steel—i.e. a stainless steel—38-47 tons per sq. in. ult. tens., 18-30 per cent elongation, 31 tons per sq. in. yield point, 12 to 14 per cent chromium, 2 per cent nickel, 120-215 Brinell hardness.

The piston rings are carried in a nickel chrome iron casting sandwiched between the cast steel piston and the piston rod flange. There is a clearance space between the top of the annular carrier and the piston crown, to allow for differential expansion of the metals. The piston is secured to the piston rod flange by twelve, 2-in. mild steel studs. There are six cast iron piston rings of the anchored type, 23 mm. wide, 18 mm. deep, two of them—usually in the third and fourth grooves—with over-lapping ends of substantial design; also one cast iron scraper ring, 15·5 mm. wide, 15 mm. deep, with scraping edge directed upwards. A lead-bronze rubbing ring, 28 mm. by 7 mm., in three equal segments, is arranged near the lower end of the piston. There are four, 1-in. tapped holes on the rim of the piston for eyebolts. Although it is advisable to avoid interruptions of surface by the drilling of holes, it is nevertheless necessary to have a ready means of lifting a piston.

Fig. 38 shows the normal disposition of the piston rings. The end ring is placed as near to the rim of the crown as possible. This ring disposition is satisfactory for four-cylinder engines, where high compression is essential to ensure positive

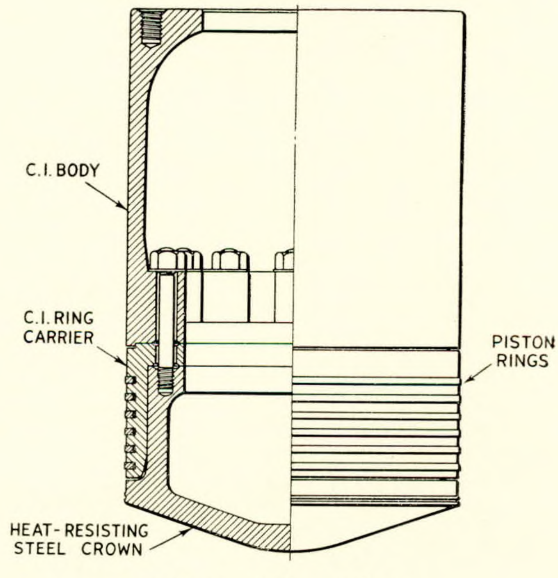
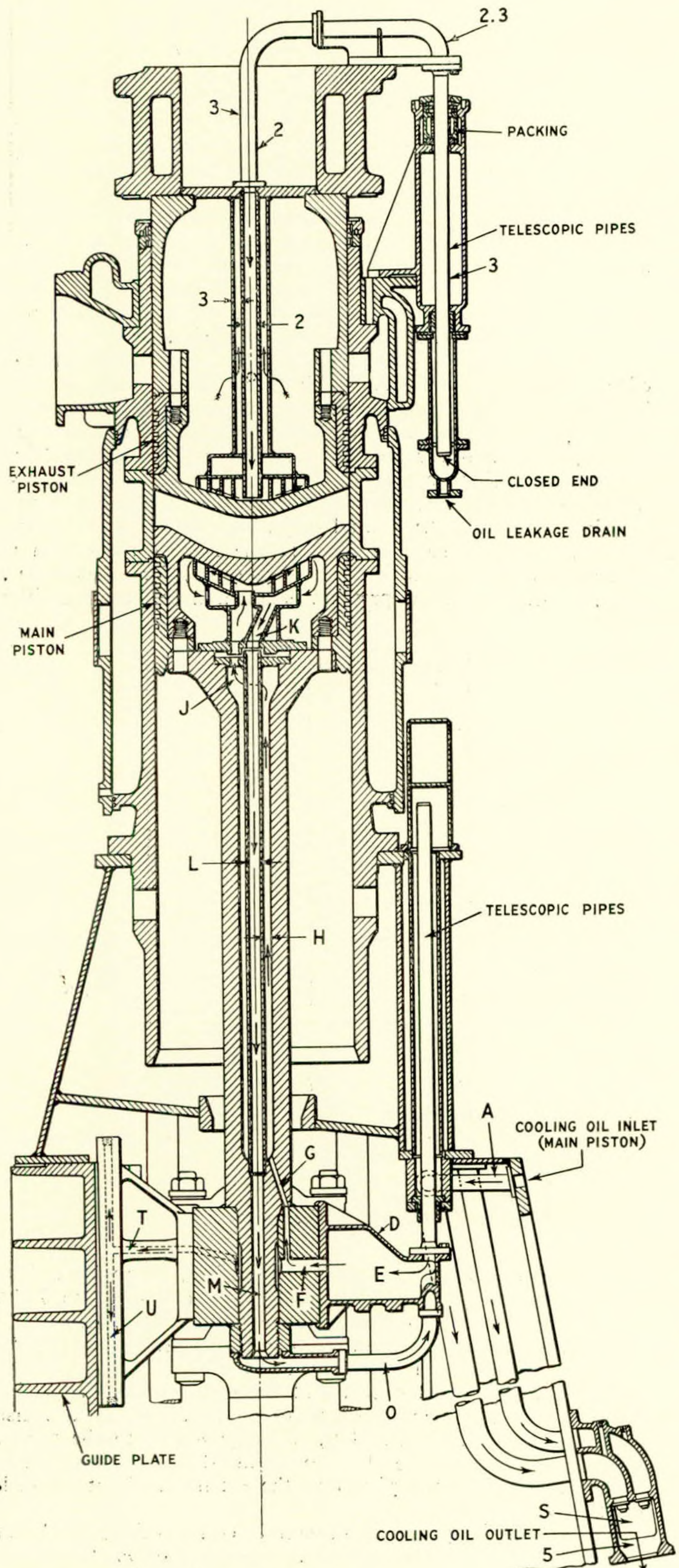


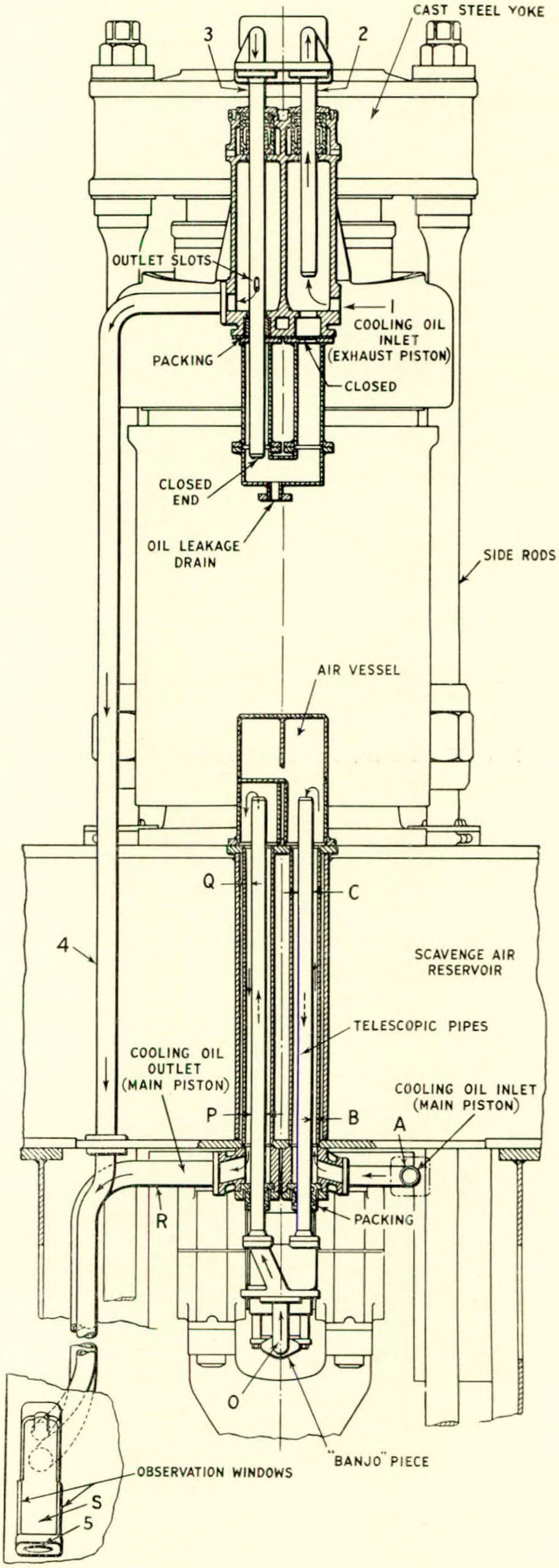
FIG. 39—*Exhaust piston*

starting, and it is therefore made standard for all engines. The diameter of the piston rim is 744·3 mm. (29·30in.) to 745·6 mm. (29·35in.), the cylinder being 750 mm. (29·53in.). The diameter continues to taper to 748·3 mm. (29·46in.) at the scraper ring, and then remains parallel.

The piston is oil cooled and the construction of the internal oil-swirling device will be clear from Fig. 38 and also from Fig. 40. It is carried by a flange bolted to the top of the piston rod flange and is so arranged that cooling oil is always retained in the piston to the extent of about one-half of the volume.

The exhaust piston is of simple construction, as shown in Fig. 39. In general features the design follows that of the main piston. The crown is a stainless steel casting; the cylindrical body—which is, in actuality, a guide—is of nickel chrome cast iron. The six piston rings, 23 mm. wide, 16 mm. deep, are arranged in a cast iron carrier ring, the flange of





FIGS. 40 and 41—*Piston oil cooling systems* which is gripped between the internal flanges of the body and the crown by twelve, 2-in. mild steel studs 622 mm. (24·49in.) long. The cast iron piston rings are of the anchored type, with strong overlapping ends. As with the main pistons, the end ring is arranged to be as close to the rim of the crown as possible.

The level of temperature in the exhaust piston is higher than that in the main piston. The cast iron body is cylindrical, its external diameter being 750·1 mm. (29·53in.), the cylinder liner being 752 mm. (29·61in.). At the crown the diameter is 746·3 mm. (29·38in.); at the bottom of the carrier casting the diameter is 747·7 mm. (29·44in.), tapering to 750·1 mm. (29·53in.) at the top.

For oil cooling, the high-alloy chrome steel crown is 53 mm. (2·09in.) thick, and the circumferential wall 45 mm. (1·77in.); the cast iron body is 39 mm. (1·54in.) thick. When fresh water cooling is specified, the piston is of the same general design, but the crown is a carbon steel casting 63 mm. (2·48in.) thick, with circumferential walls 45 mm. (1·77in.) thick; the cast iron body is 39 mm. (1·54in.) thick.

The cleaning of carbon from the interior surface of the oil cooled crown is a simple process, the yoke closing plate being taken off and the oil circulator withdrawn. A carbon layer can easily be mistaken for metal. Cleaning should be undertaken twice a year, if convenient. Should heavy accumulations of carbon be permitted to grow inside the crown, local external burning can be expected, despite care taken with the fuel spray.

As shown in Figs. 3 and 40, each exhaust piston is securely bolted to a sturdy cast steel yoke.

Piston Cooling System

The oil cooling systems for main and exhaust pistons are shown in Figs. 40 and 41.

Taking the main piston system first: from the cooling oil inlet connexion (A), near the top of the frames, the oil passes into the twin telescopic pipe casing (B), ascending the annular space between the casing and the telescopic pipe, and descending through the telescopic tube (C) to the fitting (D) attached to the piston rod crosshead. Flowing out through passage (E), the oil enters the crosshead through horizontal hole (F) and ascending holes (G). Thence it flows up the hollow piston rod, through annular space (H), reaching hole (J) and then circulating through the swirling device inside the piston, to descend through connexion (K) and tube (L) to hole (M). Passing through outlet (O), the oil ascends telescopic pipe (P) and descends through annular space (Q), to flow away through outlet pipe (R) leading to observation vessel (S) and so to the oil sump.

For the exhaust piston cooling system: the oil enters the inlet connexion *(1)*, ascends the telescopic tube *(2)*, descends the pipe also labelled *(2)* to the inside of the exhaust piston, then circulates through the oil swirler arranged inside the piston crown. Leaving the exhaust piston by the annular space marked *(3)*, and the pipe and telescopic tube also labelled *(3)*, the return oil flows through two vertical slots in the telescopic tube *(3)*, and so passes away by pipe *(4)* leading to observation outlet glass at *(5)* and eventually to the sump.

The main guide shoe is lubricated by oil which flows from hole (F) in the crosshead, through two horizontal holes (T) to two vertical holes (U), leading to horizontal grooves cut in the white metal of the guide shoe. (See also Fig. 35.)

To give an indication of sizes: for the cylinders shown in Figs. 40 and 41, the main piston cooling oil inlet and outlet pipes are respectively 2½-in. and 3½-in. bore; the telescopic tubes are 48-mm. (1·89in.) bore, 5 mm. (0·197in.) thick, hot-rolled weldless steel; the exhaust piston cooling oil inlet and outlet connexions are 2-in. and 2½-in. bore respectively; the telescopic tubes are 48-mm. (1·89in.) bore, 5-mm. (0·197in.) thick, hot-rolled weldless steel. With water cooling, the telescopic pipes are 32-mm. (1·26in.) bore, 8-mm. (0·315in.) thick, monel metal.

A suitable resilient, floating packing is provided for all telescopic tubes. If fresh water cooling is applied to the exhaust pistons the system is the same as for oil; but the packing must be of suitable design and quality. The fresh water supply is taken from the cylinder jacket outlet, i.e. at the point of highest temperature. It returns to the engine fresh water circulating line, by way of sight glasses arranged on the front of the engine. A deæration pipe is fitted to the line at each exhaust piston.

The lubricating oil system external to the engine is simple. In essentials it consists of a common oil main which subdivides to supply the main and exhaust pistons with cooling oil and the running parts with lubricating oil. Control valves are located at four points in the system, for regulating the respective supplies. The author's preference is for the piston cooling and the bearing lubrication systems to be separated. This means a duplication of oil pumps, coolers and filters, and not every client is prepared to pay the higher price.

Exhaust Piston Operating Gear

Each exhaust piston yoke is secured to four long side rods, i.e. two per side. These side rods terminate in a pair of eccentric rod gudgeon pin bearings. The eccentric rods are connected at their upper ends to the gudgeon pin bearings and, at their lower ends, are bolted to the straps which encircle the eccentric sheaves on the crankshaft. The various components in the eccentric drive will be seen in Figs. 1, 2, 3, 36, 42, 43 and 45.

An eccentric rod top end or gudgeon pin bearing, with guide shoe, is illustrated in Fig. 43. A steel casting forms the upper half-bush and the guide shoe. The lower half-bush is usually a forging but it may be a steel casting. The half-bushes are gripped by the mild steel side rods which transmit the exhaust piston load. The path of the lubricating oil will be clear from the drawing. The general pattern of the oil channels and grooves follows that of the main crosshead. The horizontal grooves in the lower half-bush terminate in small channels which continue to the edges of the bush, to permit of a continuous flow of oil through the bearing. Their purpose is thus similar to that of the small holes in Fig. 33. In Fig. 36 the eccentric rod gudgeon pin bearings and guide

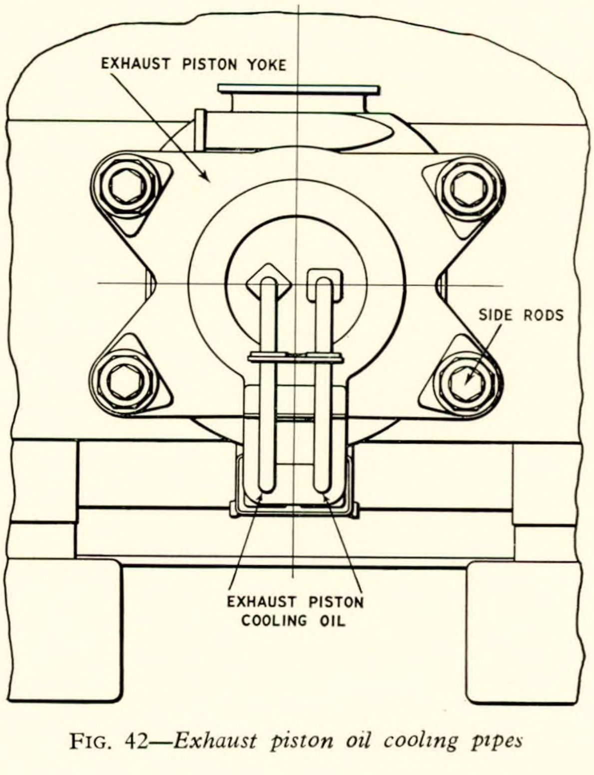


FIG. 42—*Exhaust piston oil cooling pipes*

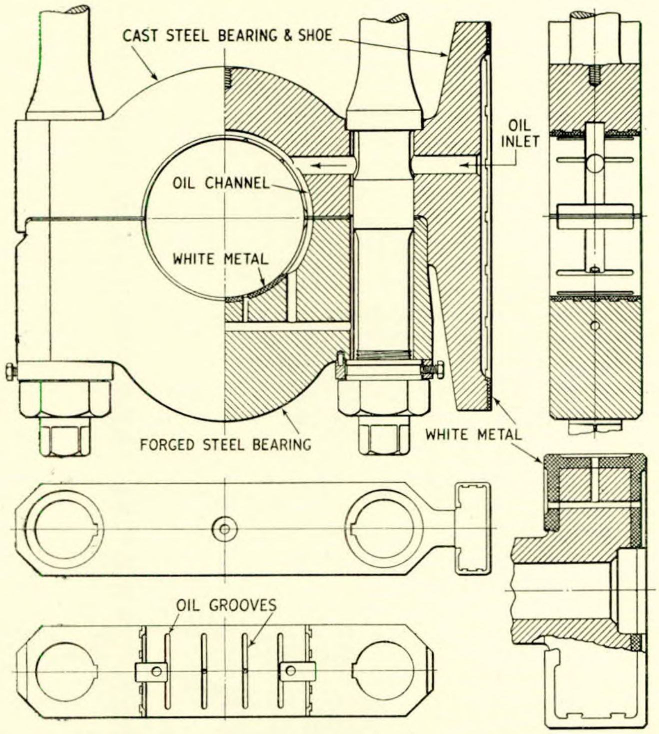


FIG. 43—*Eccentric rod top end bearing*

shoes are shown in position relative to the main piston rod crosshead and guide shoe. Fig. 44 shows the bearing loads on the eccentric rod gudgeon pin bushes.

Eccentric Rod and Strap

The construction of the eccentric strap and rod will be clear from Fig. 45. The ratio of length of rod to eccentricity of sheave is about 10·5 : 1; the ratio of crank radius to sheave eccentricity is 3 : 1.

Each eccentric strap is in two pieces which are bolted firmly together. The strap is of very heavy cross-section, to ensure exact preservation of geometrical form when running under load. By obviating any tendency to flexure the frictional loss is reduced to insignificance.

A hole is drilled in all eccentric sheaves, on the face remote from the main bearing, for the insertion of a feeler between sheave and strap. The hole may be 13 mm. diameter, 46 mm. deep. See item (D) in Fig. 16. A hole is also drilled in the eccentric rod, at the radius of the gudgeon pin, for insertion of a feeler between bush and pin.

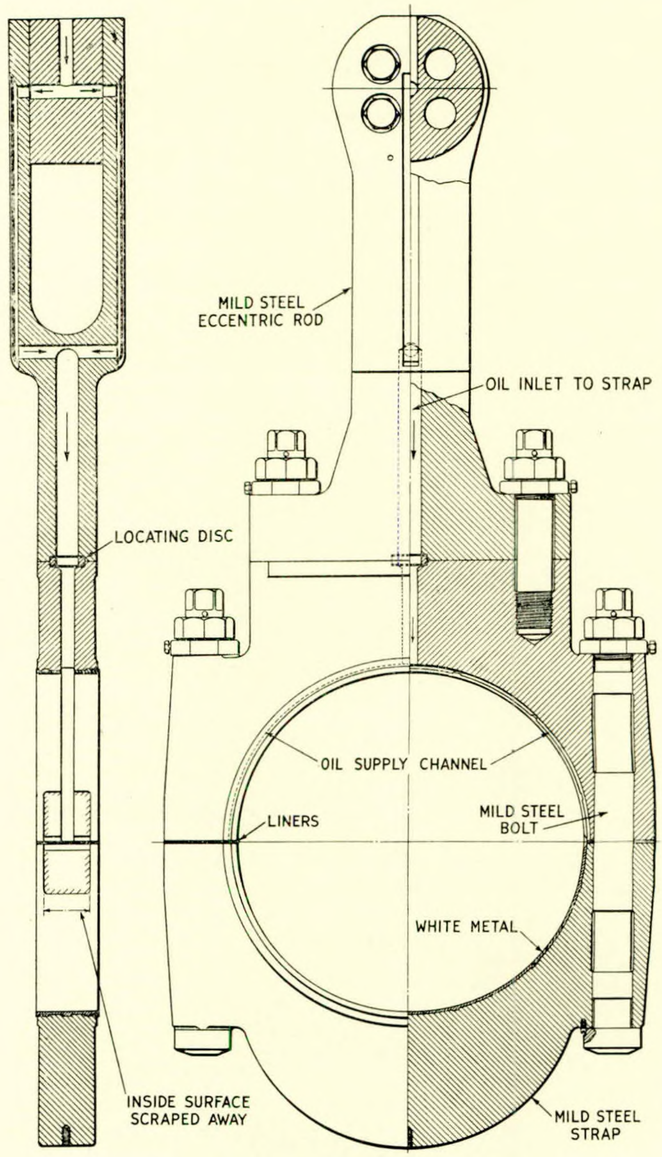


FIG. 45—*Eccentric rod and strap*

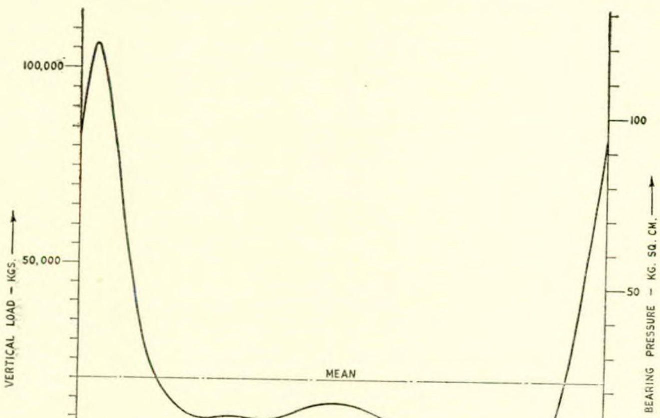


FIG. 44—*Eccentric rod gudgeon pin loads*

Lubricating oil flows down a longitudinal hole in the eccentric rod and continues through a vertical hole in the upper half of the strap. This hole terminates in a channel cut circumferentially in the white metal. The inside surface of the white metal is cut away to form an oil reservoir. The whole of the bottom half of the strap is free from channels and grooves, but there is a tapered entrance for the oil at each side of this surface. The white metal is 8 mm. (0·32in.) thick and extends 2·5 mm. (0·098in.) beyond the face of the strap at each side. There are a generous number of transverse dovetails, in addition to three circumferential dovetails. Hitherto the reasons in favour of dovetails have outweighed those adverse to them. Instead of a single vertical hole in the upper strap, there is an alternative arrangement consisting of two sloping holes and very short circumferential grooves. For the higher speed engines this alternative arrangement obviates the formation of ridges on the sheaves.

The total effective surface of two eccentric straps is 2·2 times that of one connecting rod bottom end bearing. The ratio of eccentric strap bearing width to diameter is 0·175 : 1.

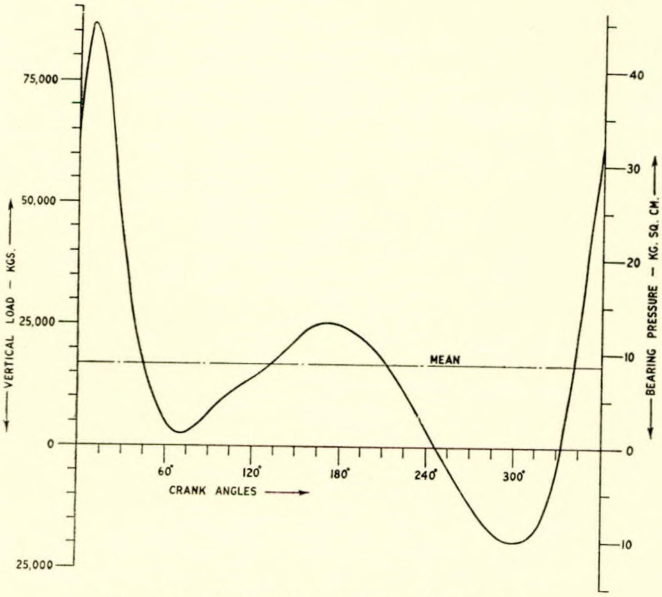


FIG. 46—*Eccentric strap bearing pressures*

Fig. 46 shows the cyclic variation of loading for an eccentric strap and sheave. The pressure is taken by the lower strap. In Fig. 47 the magnitude and distribution of the loading on an eccentric strap and sheave are shown. As implied earlier, the author prefers this form of diagram to the more conventional graph in Fig. 46. In Fig. 48 the diagrams of Fig. 47 are expanded on to a straight line base and are probably easier to follow. The diagrams are drawn for 112 r.p.m. and 9·2 kg. per sq. cm. (131lb. per sq. in.).

The eccentric straps are cut from mild steel slabs; alternatively they may be steel castings. If the latter, they are annealed at 925 deg. C. and then stress-relieved at 650 deg. C. The shims and liner at the horizontal joint have a total thickness of 8 mm. (0·32in.). The eccentric strap bolts and the eccentric rod studs are screwed 5¼in. diameter. Each strap bolt has a longitudinal hole 50 mm. (1·97in.) diameter, becoming 16 mm. (0·63in.) diameter in way of the thread.

The eccentric rod is a mild steel forging, with a deep jaw milled from solid for containing the gudgeon pin. The crosshead pin, 500 mm. (19·69in.) diameter, 250 mm. (9·84in.) long, is a 0·45 per cent carbon steel, 40-45 tons per sq. in. ult. tens., flame-hardened, the surface being super-finished to 3-6 micro-inches. The gudgeon pin is riveted to the eccentric rod jaws by four, 120 mm. (4·72in.) diameter carefully fitted pins, 28-33 tons per sq. in steel, with ends riveted over. The vertical oil channel on each outer face of the jaw is formed by a thin steel strip over a milled groove. The eccentric rod palms, strap faces and strap joints are smooth machined and hand scraped.

To assemble an eccentric strap and rod on the crankshaft, the bottom half-strap, with bolts in place, is lowered on to the top of the eccentric sheave, the bolts pointing downwards. The half-strap is then rotated, by suitable gear, until it occupies its correct place under the eccentric sheave. The upper half-strap is then lowered on to the bolts. The eccentric rod may be shipped with the upper half-strap, or it may be lowered

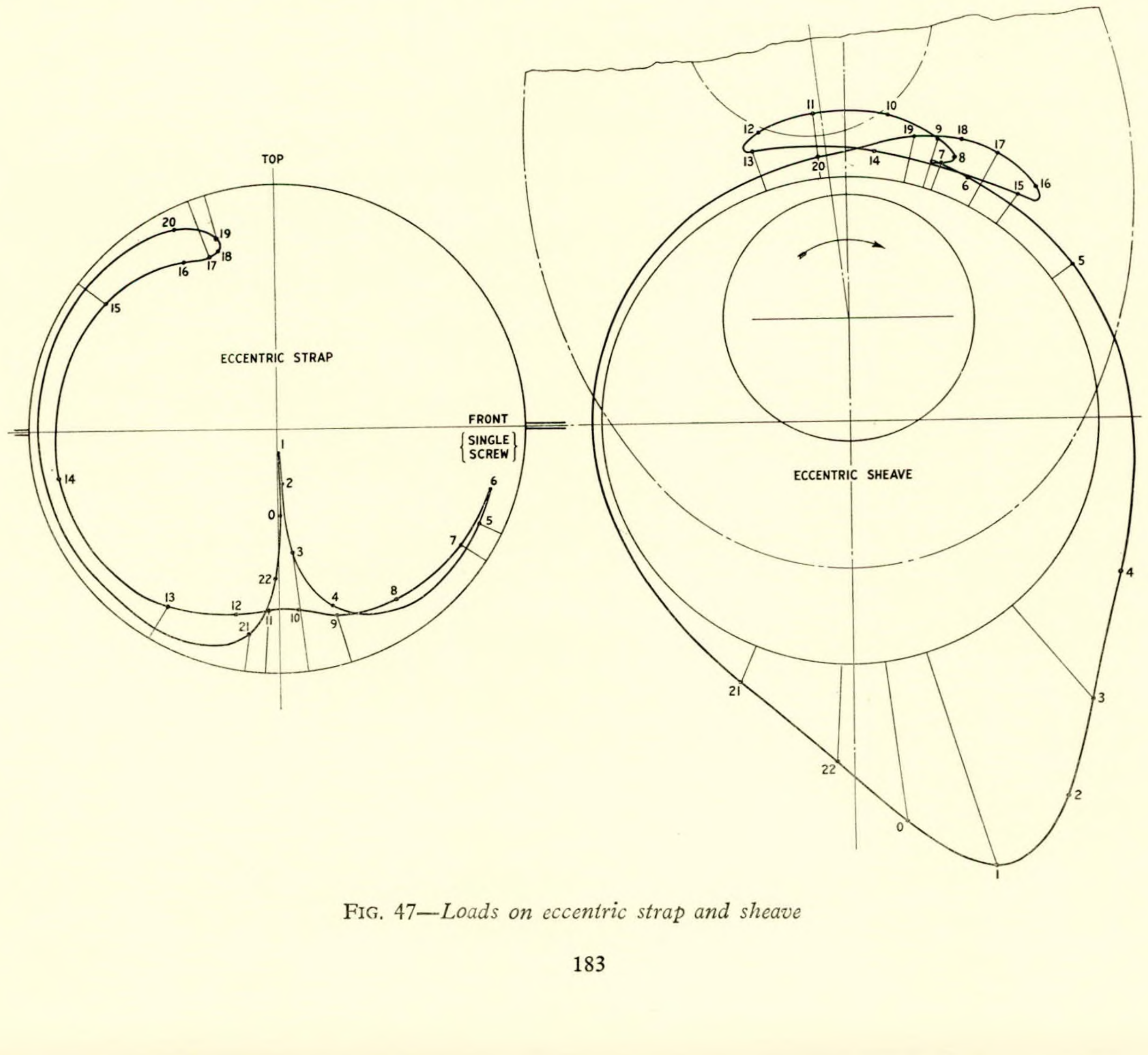


FIG. 47—*Loads on eccentric strap and sheave*

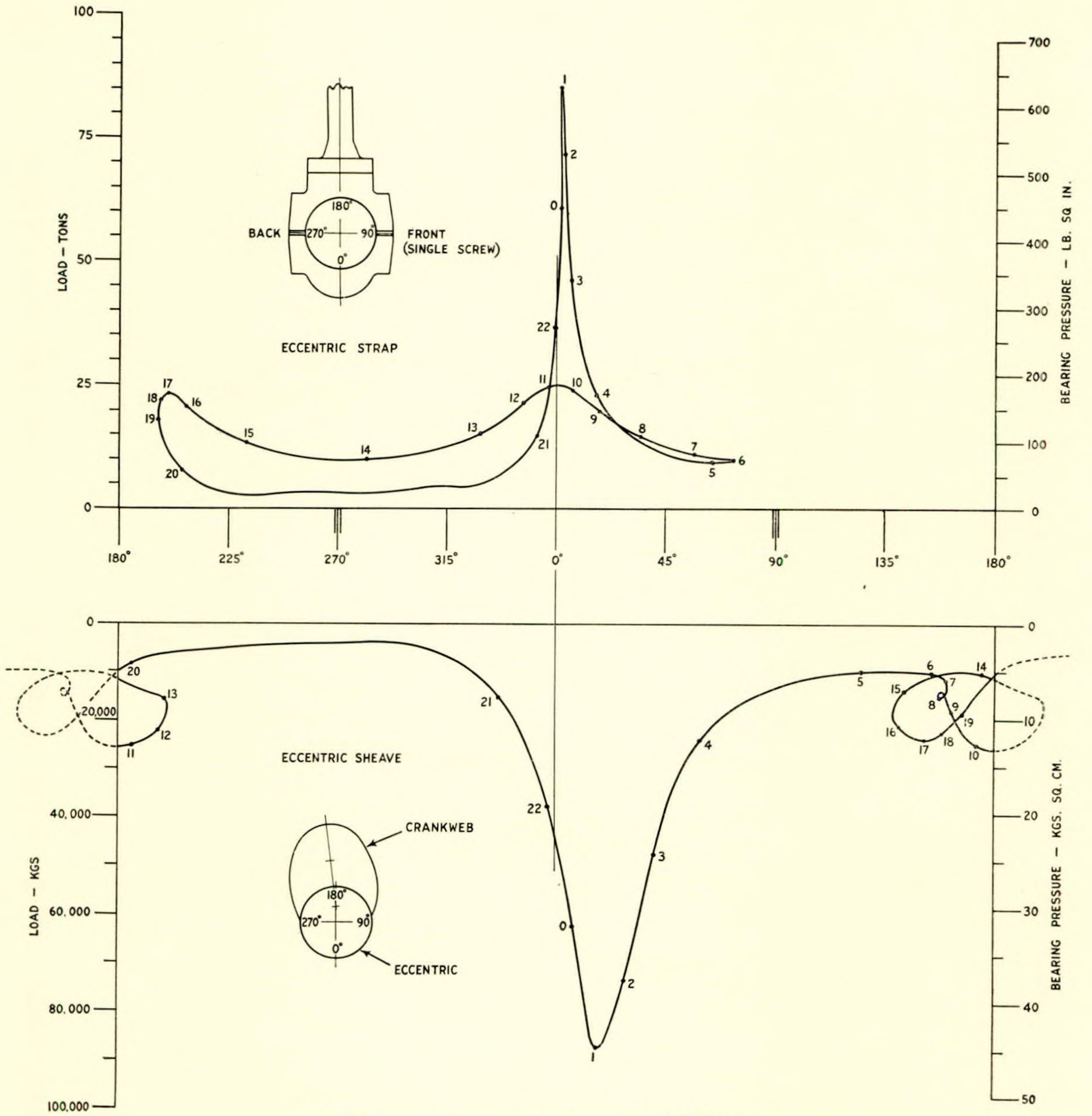


FIG. 48—*Loads on eccentric strap and sheave*

separately. Alternatively, the half-straps may be lifted sideways into position on the sheave, i.e. with the joint vertical. The bolts are then inserted horizontally into the straps.

Eccentric strap running clearances vary from 0·012 to 0·015in. for the larger engine size and from 0·010 to 0·014in. for the smaller engine size.

To determine the actual stresses in eccentric straps under full-power running conditions, strain gauges with all necessary connecting wires were recently arranged on the aftermost eccentric strap of a seven-cylinder engine running on the test bed. The contract full-power rating of the particular non-pressure induction engine available was 7,700 b.h.p. at 112 r.p.m. The firing pressure was 51 kg. per sq. cm. Fig. 49 shows the results of the strain gauge measurements. The strain gauges were arranged on the upper half-strap, as indicated at points *(1)* to *(5)*. On the lower half-strap there was a single strain gauge, at point *(6)*. The rotation of the eccentric sheave was clockwise.

In diagrams 1 to 3 the stresses are compressive, the stress being greatest at point *(2)*, where it is 1·5 tons per sq. in. At point *(4)* the stress changes to 1·4 tons per sq. in. tension. At points *(5)* and *(6)* the stresses are respectively 0·9 and 0·7 tons per sq. in. tension. In diagram 4 the compression is due to bolt tightening not being included in the measurement. Zero stress is calculated to occur at 50 degrees before exhaust piston bottom dead centre.

The bottom right-hand sketch shows the stress variation in an exhaust piston side rod. This diagram is used as a stress scale check on the other diagrams because the stress in the side rod can be checked by calculation and by reference to the flexure of a long thin steel rod. The exhaust piston pressure is always in the upward direction, hence the load is taken by the lower half-strap. The purpose of the experiment was to determine what happens at the upper half-strap.

Strain gauge readings at the same six points, to determine the tightening-up stresses when the corners of the nuts were rotated 125 mm., showed: at points *(4)*, *(5)* and *(6)*, compressive stresses of 1·47, 0·36 and 0·53 tons per sq. in.; at points *(2)* and *(3)*, tensile stresses of 0·12 and 0·44 tons per sq. in. At point *(1)* there was zero stress. The compressive stress at the front vertical face of the strap, outside of the bolts, was 1·32 tons per sq. in. The horizontal diameter of

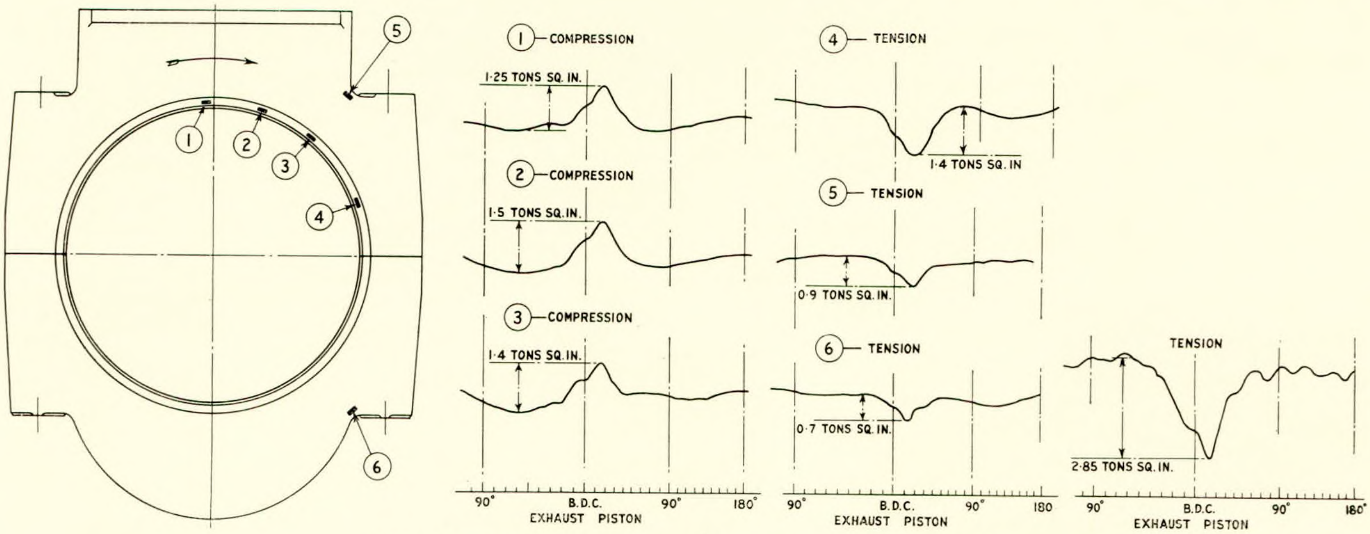


FIG. 49—*Strain gauge measurements on eccentric strap*

the sheave was increased by 0·01in.; the vertical diameter was reduced by 0·0173in.; there were points of zero change at 30 degrees to the horizontal. Diametral changes are corrected by re-turning.

Fig. 59 shows the result of a qualitative photo-elastic investigation into the loading of an eccentric strap model. The numerical figures are indicative of relative stresses. Thus, at point (*6*) the stress is three times greater than at point (*2*) and 1·5 times greater than at point (*4*); and so on. Zero stress occurs at points (*0*). This is almost the extent of the deductions which can usefully be made.

General

It is not necessary to refer to the manœuvring and reversing gear; the established arrangement remains standard and descriptions will be found in authoritative literature dealing with Harland and Wolff machinery. The thrust block, also, is unaltered. Where the Michell block is requested by clients, it is fitted; otherwise the Harland and Wolff block is supplied.

Engine room cranes and overhauling gear are matters of real importance. Expeditious overhauling requires, as a concomitant, a high degree of labour saving. It is not the size of an engine which makes overhauling difficult; it is lack of appropriate facilities. A large and powerful engine, well planned for easy overhaul and well provided with lifting and transporting gear, can be much easier to handle than a small engine having no equivalent gear.

Wherever possible, main pistons and piston rods should have a straight vertical lift. Ship considerations can, and do, override this desideratum, but, nevertheless, every effort should be made, in planning the engine room, to attain this result.

As this paper is confined to a description of propelling engines, no reference need be made to ancillary equipment. There is, however, one exception. If, in the burning of heavy fuels, maintenance costs are to be low, then the fuel oil purifiers and clarifiers must be of ample size. In the author's opinion, many purifiers are much too small. It seems to him to be prudent to choose purifiers having listed outputs of twice

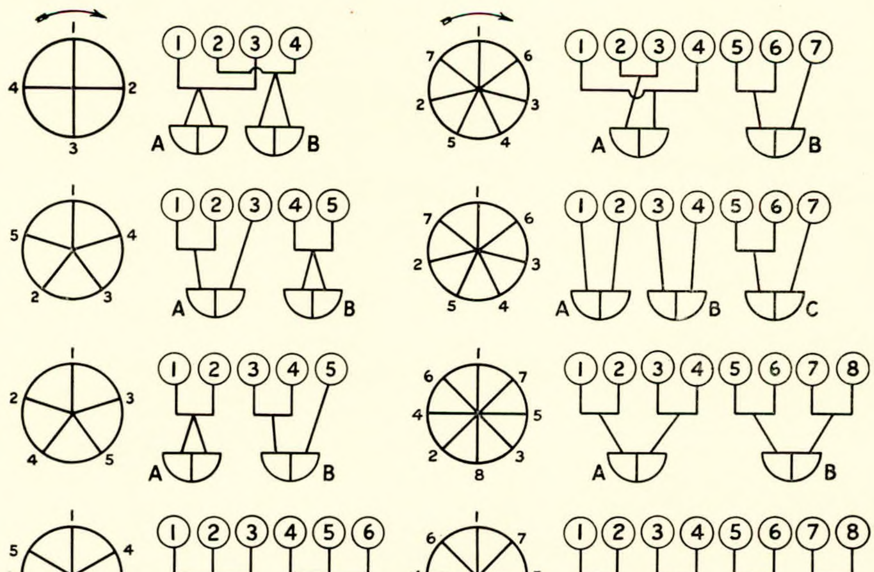


FIG. 50—*Diagrammatic arrangement of exhaust pipes*

what is required. The additional cost is well worth while.

Nothing need be said regarding crankcase safety doors and flame traps.

The consumption of starting air for the described engine type has been so low that, in some ships, the owners have taken out one of the cylindrical starting air reservoirs. The initial air pressure in the reservoirs remains at 25 atmos. (356lb. per sq. in.).

EXHAUST TURBOCHARGER ARRANGEMENTS

The Harland and Wolff two-stroke engine is pressure-charged on the principle of utilizing the exhaust gas impulses, as in the earlier Belfast-built four-stroke Büchi turbocharged engines—which, in their day, comprised many notable installations, including twin-screw sets of eight- and ten-cylinder units, 740-mm. (29·13in.) bore, 1,500-mm. (59·06in.) stroke. With two-stroke engines of few cylinders the problem of pressure charging is more complex and more subtle than it was with four-stroke engines of many cylinders.

It is essential that the exhaust turbines shall be located as close to the cylinder exhaust branches as possible; otherwise the energy in the exhaust impulses can be uselessly dissipated. In other words, the turbochargers must be mounted on the engine. The substantial increase in power, which is obtainable with a relatively low boost pressure, is mainly the result of utilizing the high energy impulses at exhaust release. The piping is so arranged that the exhaust impulses from one cylinder do not interfere with the impulses from other cylinders. Between exhaust release and scavenge air opening, the exhaust pressure must fall below the air charging pressure, ensuring an effective scavenge and a minimum remainder of gases in the cylinder. The air flow through the cylinder has a cooling influence, which should be reflected in the volumetric efficiency and low thermal loading.

In Fig. 50 is shown, diagrammatically, the disposition of blowers and the arrangement of exhaust pipes for engines of four to eight cylinders. The encircled figures represent the cylinders and the lettered semi-circles are the blowers. The turbines have each a double-exhaust gas entry at the inlet casing, with internal gas ducts arranged to keep the gas streams separated right up to the nozzle ring, i.e. until the instant when the gases impinge upon the turbine blades. For the four-cylinder engine there are two blowers (A) and (B); the pipes from cylinders 1 and 3 are taken to blower (A), the pipes from 2 and 4 to blower (B). The interval between the successive impulses from 1 and 3, or 2 and 4, is 180 degrees of crank angle. In the eight-cylinder engine with two blowers, the exhaust pipes from cylinders 1 and 2 are led to one inlet of blower (A), and those from cylinders 3 and 4 to the other inlet of (A). Similarly with blower (B) and the connected cylinders 5, 6, 7 and 8. The angular interval between successive impulses from cylinders 1 and 2, and from 4 and 3, is 135 degrees. And so on.

Figs. 60 and 61 show a two-blower arrangement on a six-cylinder engine.

Exhaust gas turboblowers are self-adapting to variations in engine load. Accordingly, cut-out devices, whether for overspeed or for overload, are not required. The only cut-out device necessary is the emergency overspeed governor which is mounted on the engine above the manœuvring platform. The speed of the turboblower depends upon the available flow of exhaust gas, upon its state of pressure and temperature, upon the power absorbed by the compressor, upon the component efficiencies of the turboblower, and upon other relevant but minor factors.

In the firm's non-pressure induction (non-P.I.) engines, scavenging air is provided by chain-driven Roots' displacement blowers mounted on the back of the engine. In the pressure-induction (P.I.) engines the chain-driven blowers are discarded; that is, for normal running the exhaust turbochargers must fulfil all scavenge and combustion air requirements. At low engine revolutions, however, there is insufficient energy in the exhaust gases to drive the turbochargers at the speed necessary to provide the required air manifold pressure and air mass flow. In this fact lies one of the problems inherent in two-stroke pressure charging.

The free air capacity for the blower can be computed from the aggregate cylinder swept volume and the appropriate excess air factor for the stipulated engine rating. Using this figure and applying the necessary pressure ratio, the basic frame size can be deduced for the particular exhaust manifold arrangement, together with the appropriate turbine nozzle area and compressor diffuser. For a pressure charged rating of 35 per cent above the normal non-P.I. rating, it is usual to provide a free air quantity of 1·5 to 1·8 times the aggregate cylinder swept volume.

For all the engines described in this paper, Napier turbochargers are used. The complete range of engine powers has hitherto been satisfied by two sizes of turbocharger, respectively designated MS.500 and MS.600 by the makers.

The Napier turboblower consists of four casings which are bolted together to house a single-stage axial flow exhaust gas turbine mounted on a common shaft with a centrifugal air compressor. The engine exhaust gases are led into the turbine inlet casing, where they are directed through a ring of nozzle

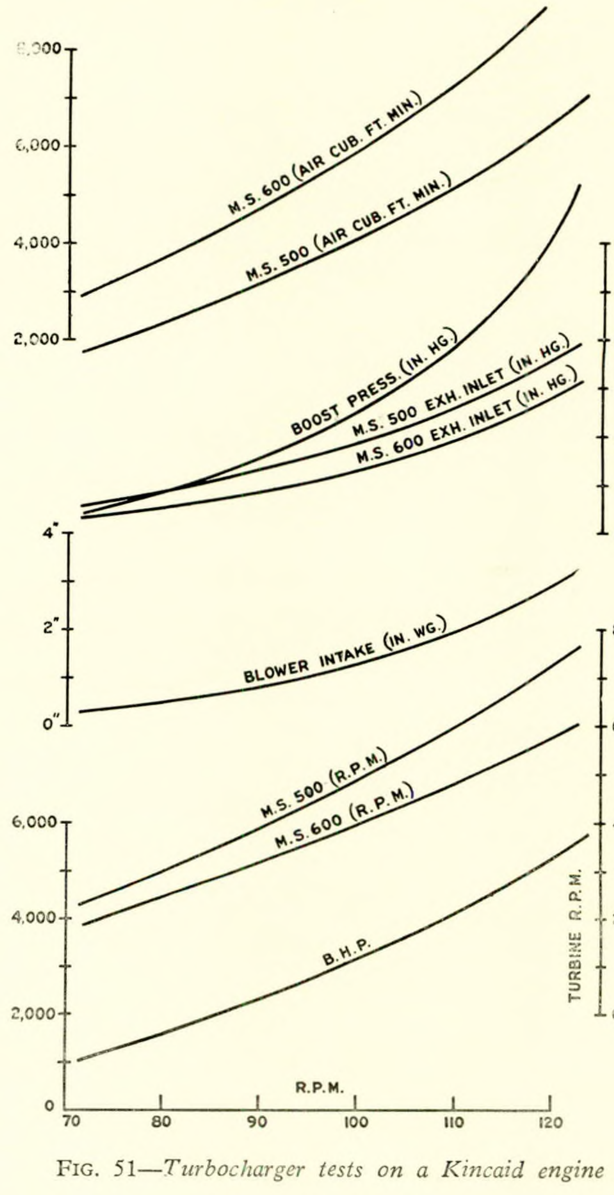


FIG. 51—*Turbocharger tests on a Kincaid engine*

blades on to the blades of a turbine wheel. After passing the turbine wheel, the gases leave the turbine outlet casing through a large rectangular aperture. Rotation of the turbine wheel by the action of the exhaust gases causes rotation of the air impeller. Air is thereby drawn from the atmosphere, through the blower inlet, compressed, and passed through the blower outlet casing, by way of a diffuser and a delivery volute, to the air inlet manifold of the engine.

From the data recorded at prototype engine trials, it can be determined whether or not the engine and turboblower are correctly matched. Should it prove necessary, adjustment can be made by a change of blower diffuser, turbine wheel, or nozzle assembly. The correct combination of compressor and turbine, once established, remains correct for all subsequent engines of the same type and rating.

The turboblower rotor is supported in plain sleeve bearings, which are mounted in spherical housings. The author will not accept ball bearings in these large blowers.

Lubricating oil of a quality suited to the turboblower is delivered by an independent pump through filters and a cooler to connexions on the blower. After passing through the bearings the oil drains into a suitable tank. A standby pump is provided.

The turbocharger can be dismantled, for inspection or overhaul, in either the horizontal or the vertical position.

The elimination of chain driven scavenge blowers enables the 750/2,000 engine to be shortened by 550 mm. (21·7in.) and the 620/1,870 engine by 350 mm. (13·8in.). The saving is effected between the two middle frames of the engine in the region where the fuel surcharging pump and the manœuvring gear controls are arranged. There is no saving of engine length by the substitution of gas compression fuel pumps for chain driven "jerk" pumps.

If a weight comparison be made between six-cylinder engines, respectively of P.I. and non-P.I. type, there is a saving of 20 tons in favour of the turbocharged engine. This saving applies both to the 620-mm. and the 750-mm. cylinder sizes. The pressure charged engine delivers 1·3 to 1·35 times more power on this slightly reduced weight.

The total saving in weight by the substitution of gas compression pumps for the standard chain driven fuel pumps is 5 tons for a six-cylinder engine of 750/2,000 size and 3·65 tons for a six-cylinder 620/1,870 engine; useful but not significant!

At top rating the turbocharged engine delivers, on an average, 25 b.h.p. per ton weight.

Whereas bearing pressures and working stresses in all important parts of current pressure charged engines are not any greater than for non-pressure charged engines, in the earlier years it was not so. The engine was then simply the non-P.I. engine, to which 30 per cent increased power was applied by pressure charging. Only the crankshaft was strengthened. The generous proportions of the design permitted the change. Based on a P.I. firing load of 56 atmos., compared with 50 atmos. non-P.I., and on 8 atmos. m.i.p. compared with 6·5 atmos., the increase in pressures and stresses was nowhere more than 8 per cent; for most components it was less. Non-P.I. engines running at the turbocharged output always gave the impression of ample reserve power. With the latest practice, therefore, the engines have an enhanced margin. On the basis of these facts the way may perhaps be opened for pressure charging up to 50 or 60 per cent increased power in due course. As a matter of interest it may be mentioned that recently, on the test bed, a standard P.I. engine developed 50 per cent increased power, by exhaust turbocharging, without such being the intention.

When the author has suggested to important clients, men whose opinions he greatly respects, that the engine design could advantageously be lightened, at least in some of its parts—e.g. the bedplate is rather massive—the reply has always been the same, always against any such suggestion and often delivered with emphasis.

A ready yardstick of comparison, which is regarded as useful by some superintendents, is the relationship between horse power and swept volume. An average figure, for the pressure charged engines under description, is 14·5 to 15 b.h.p. per cu. m. per min., a rating in which inherent reserve power is implicit.

Fig. 51 summarizes, in graphical form, the shop test results of a five-cylinder 620/1,870 P.I. engine. There are two turboblowers. The designed maximum continuous power of the pressure charged engine was 4,900 b.h.p. at 118 r.p.m. The blowers were respectively of the MS.600 and MS.500 sizes, the larger blower, i.e. MS.600, being coupled to three

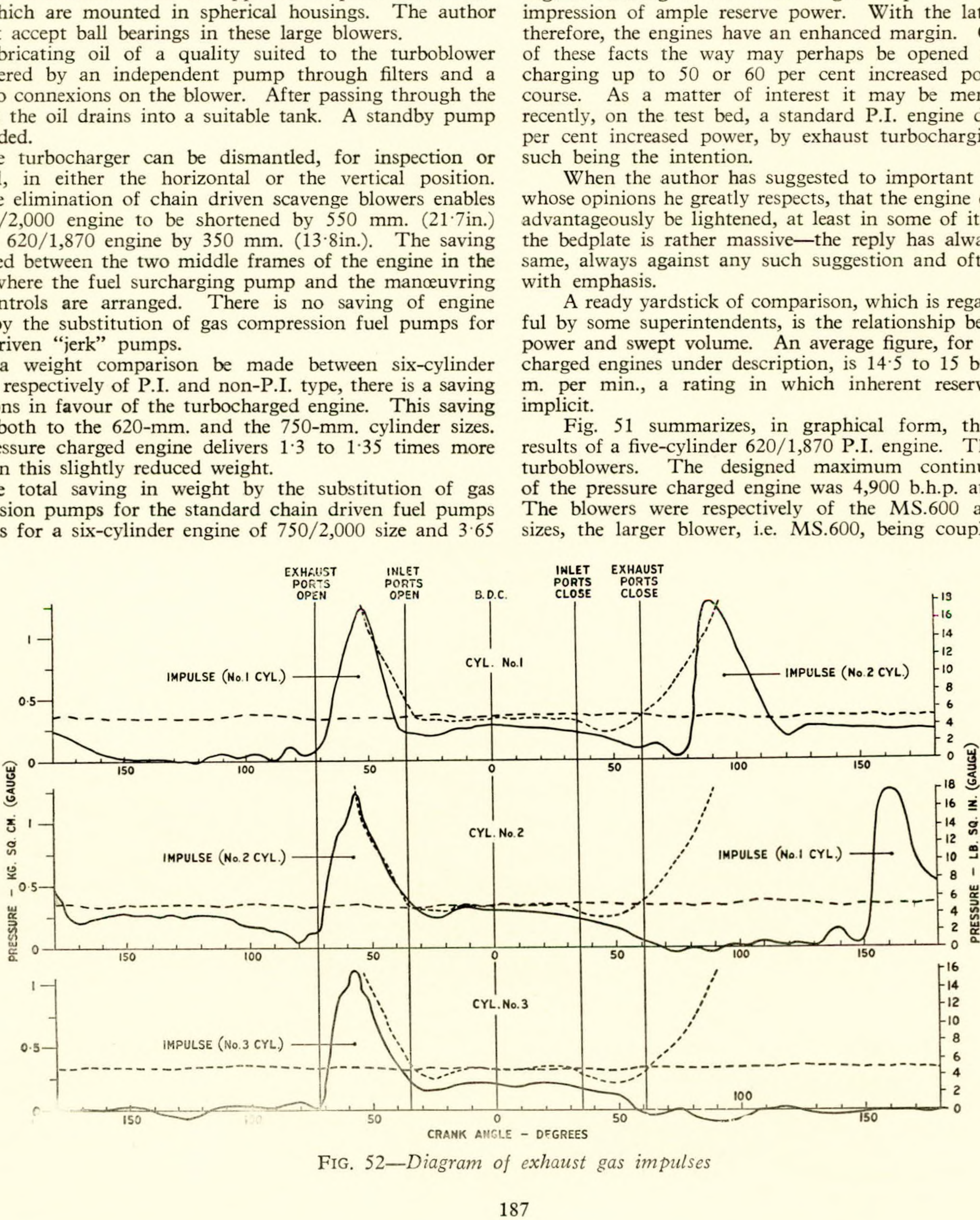


FIG. 52—*Diagram of exhaust gas impulses*

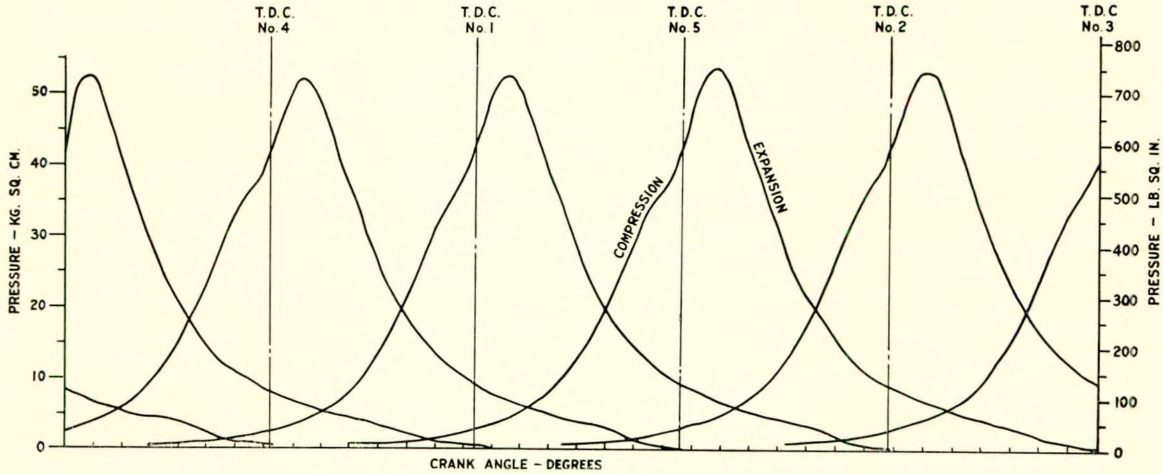


FIG. 53—*Indicator diagrams*

cylinders and the smaller blower, i.e. MS.500, to two cylinders.

The typical diagrams of Fig. 52 indicate the magnitude and duration of the exhaust gas impulses, at full load, for cylinders 1, 2 and 3; and in Fig. 53 the indicator diagrams are shown. There is not space to reproduce the other diagrams of the series, taken at full and fractional loads. The engine tests are summarized, later, at Table VIII and in Fig. 57.

The Farnborough diagrams of Figs. 52 and 53 were taken by Napier. It is difficult to obtain accurate top dead-centre lines on Farnborough diagrams. No. 2 cylinder of Fig. 52 is an example of this difficulty.

There are several ways of approaching the problem of the manœuvring and the continuous slow running of a pressure charged engine. One obvious way is to retain a positively driven blower. An alternative is the conversion of the lower end of the cylinder into an under-piston compressor, to be coupled to the turbocharging system. This the author has considered and, in fact, patents have been on his file for a decade or more. But such an arrangement, to him, is unattractive, apart from complications and cost. Other alternatives, more acceptable, have included the spinning of the turbocharger by a motor coupled to its spindle, a mechanical automatic cutting-in and cutting-out device being used.

Rightly or wrongly, it is instinctive to the author always to prefer a solution which is straightforward and simple to one which—although ingenious—is complex. Accordingly the use of a motor-driven auxiliary fan, as in the Napier system, is —for him—the present preferred arrangement.

Fig. 60 is a photograph of a six-cylinder engine, with turbochargers, auxiliary fan, charge air coolers and separators in position on the back of the engine.

SOME OPERATIONAL MATTERS

For the successful operation of turbocharged two-stroke propelling engines there are, *inter alia,* two basic conditions to be satisfied, viz.: (a) the effective ensurance of normal continuous running at sea; (b) positiveness in starting, in manœuvring, and in prolonged low power running. The first of these problems is a matter of design and construction. The second is one with which the marine engineer is closely concerned.

Exhaust turbocharged engines which lack the assistance of positively driven scavenge blowers must fulfil the following requirements, viz.: (i) when running at low powers—ahead or astern—for long periods, or when manœuvring, the engine must be supplied with a fully sufficient weight of air; (ii) if one or more turbochargers are out of use, the engine must be able to run satisfactorily at a predetermined percentage of full power; (iii) should it happen that the engine or/and turbocharger have been allowed to deteriorate, or, alternatively, should there be at least one cylinder out of action, the engine must still be able to run and be manœuvred satisfactorily.

The Napier system, which was evolved in collaboration with Harland and Wolff, consists, in essentials, of a motor driven auxiliary fan which is arranged to supply air in series with the turbochargers. The fan draws air from the engine room atmosphere and discharges into the air inlet ducts of the turbochargers. The motor may be a constant speed, or a two-speed shunt, unit. It is switched on at low speed, during manœuvring, and it continues to run until steady conditions are realized at the engine.

Non-return flap valves are arranged in the turbocharger air intakes. These valves automatically lift as soon as the turbocharger suction requirements outstrip the weight of air being delivered by the auxiliary fan. If one or more turbochargers should be out of use, the fan is run at top speed and the air discharged by the fan either passes through the turbocharger—the rotor of which is locked by means of plates —or it bypasses the turbocharger and flows direct to the air cooler or the air manifold.

Fig. 54 shows a six-cylinder engine arrangement. There are two exhaust driven turbochargers and one auxiliary, or emergency, motor driven fan.

When the engine is running normally the auxiliary fan is stopped. Each turbocharger then draws atmospheric air through its suction silencer and delivers through an air cooler and a water separator to the scavenge air manifold, thence through the scavenge ports to the cylinders. The exhaust gases are led from the engine cylinders to the inlet branches of the exhaust turbines, eventually to escape upwards to the funnel and so to atmosphere. In these circumstances the non-return flap valves (X) and (Y) are held open by the ingoing air. As the auxiliary fan discharge ducts are connected to the turbocharger suction boxes, a fraction of the air received by the turbochargers is drawn through the auxiliary fan from the atmosphere, thus causing slow rotation of the fan. Blank flanges are fitted at (B) and (C).

When the engine is running at low power for long periods and also during manœuvring periods, the auxiliary fan delivers air to the turbochargers, closing the non-return flap valves (X) and (Y). As, and when, the engine speed increases and the turbocharger speed "picks-up", the atmospheric air suction opens valves (X) and (Y). Blank flanges are fitted at (B) and (C). The auxiliary fan can be switched off, as may be desired, when the engine reaches a speed corresponding to about 25 per cent of full power.

With one turbocharger out of use—say the left-hand blower of Fig. 54—the auxiliary fan is run at top speed, delivering air to the scavenge air belt through branch (C), the blank flange having been removed. Additionally, air is delivered through the left-hand turbocharger—the rotor of which is locked—to the scavenge air manifold. Non-return flap valves (Y) are thereby closed. The right-hand turbocharger, running normally, draws air from the atmosphere through valves (X) and discharges to the scavenge air manifold. That is, the active turbocharger and the auxiliary fan function in parallel. For this operation, a blank flange is inserted at (A).

If desired, blank flanges could be inserted at (B), (D) and (F). By this means, the active turbocharger and the auxiliary fan would respectively supply air to separate sections of the scavenge air manifold. The application of this alternative must depend upon the number of cylinders, the power of the motor driven fan, and so on.

With both turbochargers out of use, the motor-driven fan is run at top speed, delivering air through branches (B) and (C) to the scavenge air manifold. Under these conditions there are no blank flanges at (B) and (C); nor at (A) and (D); nor at (E) and (F). Air will, incidentally, pass through the turbochargers—the rotors of which are locked—by way of branches (A) and (D). If the blank flanges were to remain in place at (B) and (C), then all the air would pass through the rotor-locked turbochargers, with accompanying loss of pressure.

Tests were made recently in Belfast on a six-cylinder 620/1,870 engine, which was pressure charged by two Napier turboblowers and fitted with a 30-b.h.p. motor driven auxiliary fan. At the engine service power of 5,800 b.h.p. and 118 r.p.m., the total pressure drop across the turbocharger inlet air filters and non-return flap valves was approximately 100 mm. (say 4in.) of water. With one turbocharger locked, and the motor driven fan running at top speed, an engine speed of 99 r.p.m. was obtained on the test bed, with a clear exhaust. With both turbochargers locked, an engine speed of 75 r.p.m. was obtained. Under normal running, with both turbochargers in use, the non-return flap valves were wide open at 75 r.p.m., with the motor driven fan running at low speed.

The arrangement shown in Fig. 54 affords full assurance that the engine will be adequately supplied with air in all circumstances. The changeover from normal running to engine manœuvring only requires the starting or the stopping of the motor driven auxiliary fan. It would be superfluous to provide a bypass arrangement for the turbochargers; nor is it necessary to fit blank flanges in the turbine exhaust pipes. The blank flanges shown in Fig. 54 can be handled easily and quickly; all are cold. There is no objection to exhaust

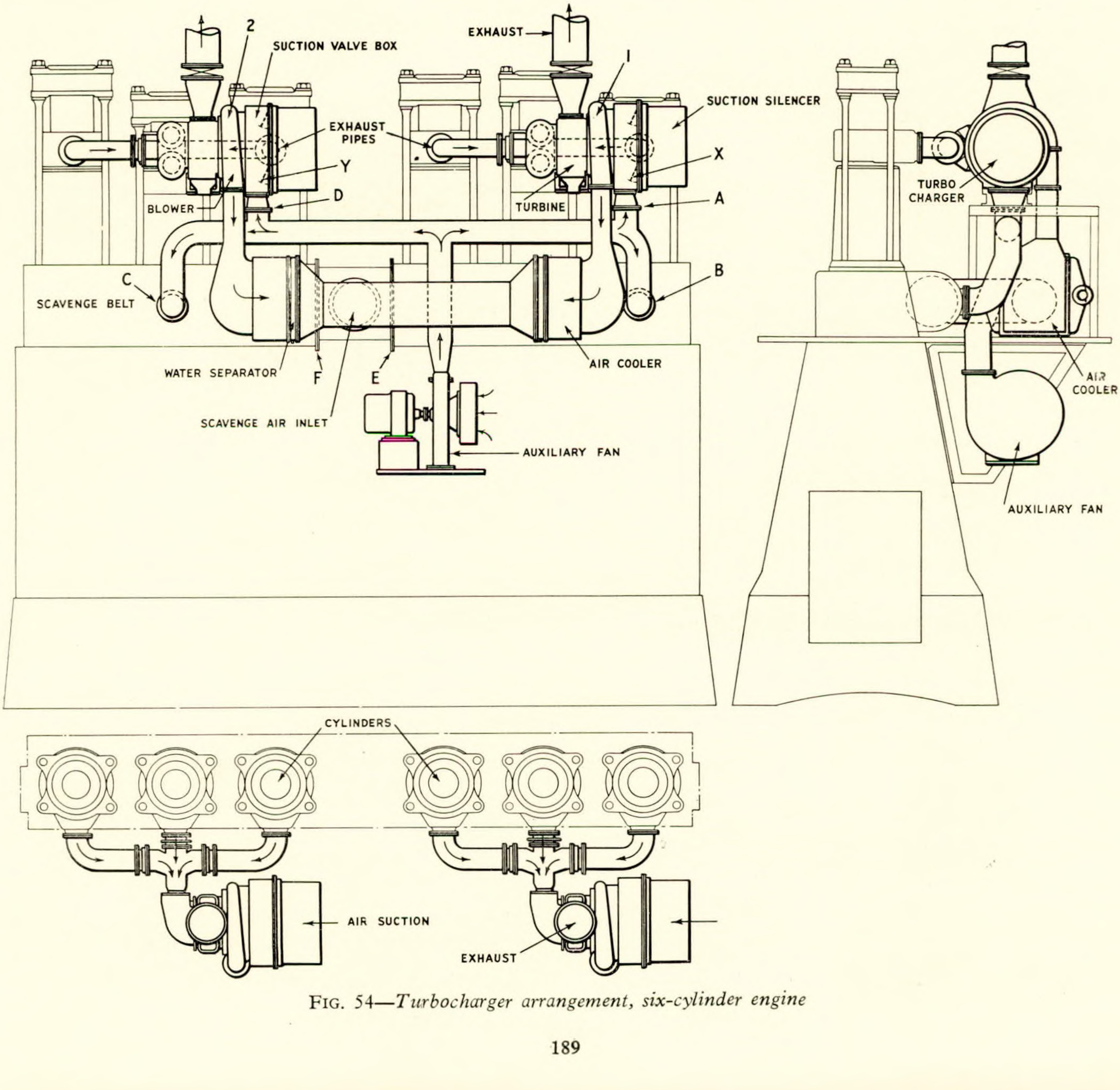


FIG. 54—*Turbocharger arrangement, six-cylinder engine*

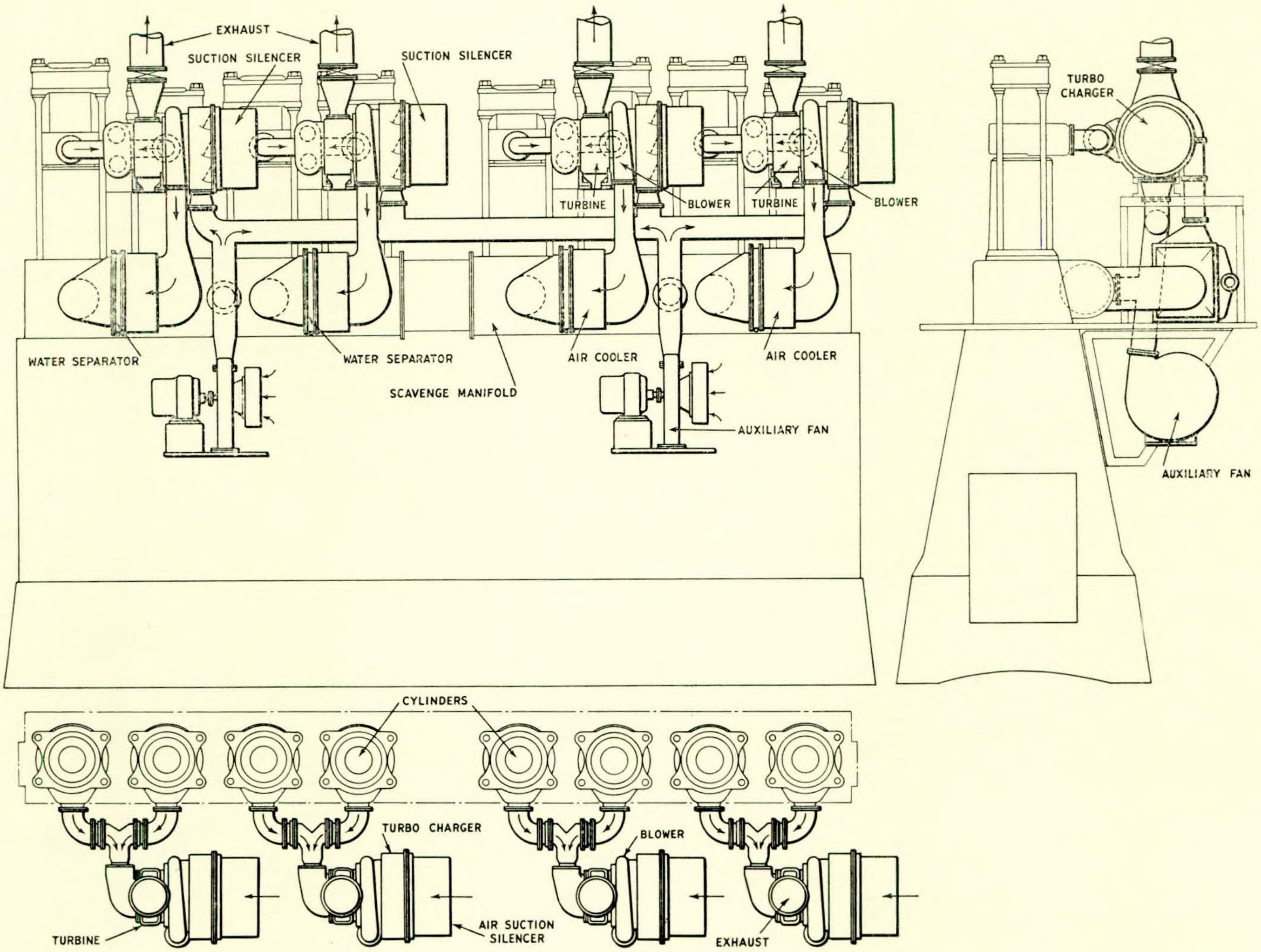


FIG. 55—*Turbocharger arrangement, eight-cylinder engine*

gases continuing to flow through a turbocharger which is out of use. The blower end seal remains air packed and the blower casing air pressure is great enough to prevent the exhaust gases from blowing past the labyrinth.

Fig. 55 illustrates an eight-cylinder engine with four exhaust turbocharging blowers and two auxiliary fans. The principles governing the arrangement are the same as those applied in Fig. 54.

In Figs. 54 and 55, between the turbocharger and the scavenge air manifold, an air cooler and a water separator are fitted. Charge air cooling has a doubly beneficial effect on engine performance. It increases the cylinder air density, thus increasing output; and also, by lowering the scavenge air temperature, it reduces the exhaust gas temperature for a given output. The pressure drop across the air cooler is small; the cooler is water circulated. The water separators shown in the diagrams are simple and compact.

Reference was made, earlier, to the shop tests of a 620-mm. bore, 1,870-mm. stroke engine, on which were arranged two turboblowers, respectively of the MS.600 and the MS.500 sizes, together with a 30-b.h.p. auxiliary fan, Fig. 51 summarizing the results. Different combinations of motor driven fan and blower on the shop tests gave the results summarized below.

(a) *MS.500 turboblower and auxiliary fan working*
Engine b.h.p. = 58 per cent of full power = 2,780;
R.P.M. = 97·5;
Turboblower r.p.m. = 3,290;
Air flow, cu. ft. per min. = 4,000;
Auxiliary fan, b.h.p. = 17;
R.P.M. = 2,150;
Air manifold pressure = 0·59lb. per sq. in. (gauge).
MS.500 exhaust turbine inlet temperature = 950 deg. F. bottom, 740 deg. F. top.
MS.600 exhaust turbine inlet temperature = 850 deg. F. bottom, 970 deg. F. top.
Appearance of exhaust gases = fairly heavily shaded.

(b) *MS.600 turboblower and auxiliary fan working*
Engine b.h.p. = 68 per cent of full power = 3,280;
R.P.M. = 102·5;
Turboblower r.p.m. = 2,950;
Air flow, cu. ft. per min. = 7,500;
Auxiliary fan, b.h.p. = 10;
R.P.M. = 2,300;
Air manifold pressure = 0·89lb. per sq. in. (gauge).
MS.500 exhaust turbine inlet temperature = 975 deg. F. bottom, 970 deg. F. top.

MS.600 exhaust turbine inlet temperature = 960 deg. F. bottom, 980 deg. F. top.

Appearance of exhaust gases = fairly heavily shaded.

(c) *Auxiliary fan, only, working (both turboblowers locked)*
Engine b.h.p. = 39 per cent of full power = 1,870;
R.P.M. = 85;
Auxiliary fan, b.h.p. = 22;
R.P.M. = 2,075;
Fan air flow, cu. ft. per min. = 5,600;
Delivery pressure = 8·3in. w.g.
MS.500 exhaust turbine inlet temperature = 667 deg. F. bottom, 670 deg. F. top.
MS.600 exhaust turbine inlet temperature = 660 deg. F. bottom, 700 deg. F. top.
Appearance of exhaust gases = fairly heavily shaded.

(d) *Both turboblowers working, one cylinder cut-out*
The MS.600 turboblower surged at highest powers; that is, at sea, the engineer would run both blowers at reduced output, or, alternatively, shut down the MS.600 turboblower and run the auxiliary fan with the MS.500 turboblower.

(e) *Both turboblowers working, minimum engine revolutions*
Progressive deloading showed that the engine would run at 20 r.p.m. with the fan running at slow speed.

(f) *Engine revolutions at which turboblowers become self-sustaining*
Progressive loading, in accordance with propeller law, showed that at 75 r.p.m. the MS.600 blower non-return valves began to flap; at 77·5 r.p.m. the MS.500 blower non-return valves began to flap, the MS.600 valves then being slightly open. The minimum sustainable engine speed, without the use of the fan and with clean exhaust, was about 60 r.p.m.

The auxiliary fans used on all engines to date are of Sirocco type, made by Davidson and Co., Ltd., of Belfast. Typical performance characteristics of such a fan are shown in Fig. 56. In the trials at (c) 39 per cent of full power was obtained with both blowers locked and the auxiliary fan only in use. For medium speed and high speed ships the margin thus shown should be satisfactory. But for low speed ships operating in bad weather a higher percentage power might be advisable. This would mean a larger fan and motor.

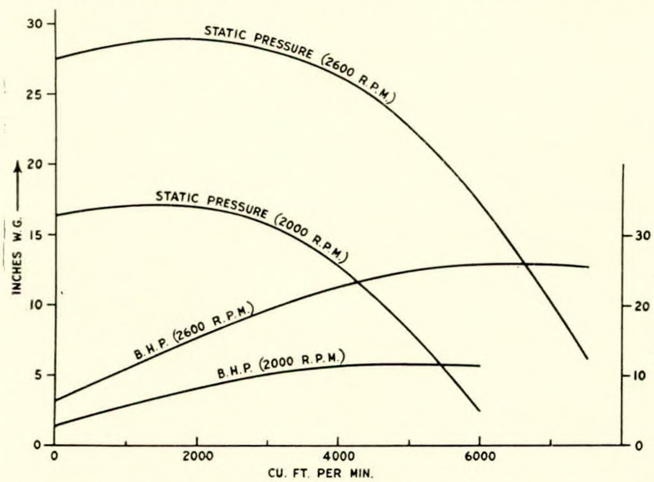


FIG. 56—*Characteristics of 30-b.h.p. auxiliary fan*

ENGINE RATING

As the maintenance costs are a function of the amount of wear and tear experienced by a marine Diesel engine, and as wear and tear is a function of the rating, it is advisable—in the author's opinion—for three levels of rating to be assigned to an engine, viz.: (a) normal continuous sea service rating; (b) maximum continuous service rating; (c) trial trip rating. On the basis of this experiential reasoning, Tables II, III and IV have been compiled.

TABLE II.—COMPARATIVE RATINGS: P.I. AND NON-P.I. ENGINES

	R.P.M.	B.H.P. per cylinder (non-P.I.)	B.H.P. per cylinder (P.I.)
(a) *620/1,400 + 470 engine*			
(i) Trial trip	125	850	1,150
(ii) Maximum continuous service	118	750	1,000
(iii) Continuous sea service	110	700	950
(*b*) *750/1,500 + 500 engine*			
(i) Trial trip	120	1,300	1,750
(ii) Maximum continuous service	110	1,100	1,500
(iii) Continuous sea service	100	1,000	1,350

TABLE III.—B.H.P. RATINGS FOR ENGINES OF 620 MM. BORE, 1,870 MM. STROKE (CYLINDER CONSTANT=1·237)

	Trial trip	Maximum continuous	Continuous sea service
M.I.P., kg. per sq. cm.	8·5	7·75	8·0
M.I.P., lb. per sq. in.	120·9	110·2	113·8
Piston speed, metres per sec.	5·8	5·5	5·1
Piston speed, ft. per min.	1,148	1,083	1,010
R.P.M.	125	118	110
4 cylinders	4,600	3,900	3,800
5 cylinders	5,700	4,900	4,800
6 cylinders	6,800	5,900	5,700
7 cylinders	8,000	6,900	6,700
8 cylinders	9,100	7,800	7,600

Total engine stroke=1,870 mm; i.e. main piston stroke=1,400 mm., and exhaust piston stroke=470 mm.

TABLE IV.—B.H.P. RATINGS FOR ENGINES OF 750 MM. BORE, 2,000 MM. STROKE (CYLINDER CONSTANT=1·936)

	Trial trip	Maximum continuous	Continuous sea service
M.I.P., kg. per sq. cm.	8·5	7·75	8·0
M.I.P., lb. per sq. in.	120·9	110·2	113·8
Piston speed, metres per sec.	6·0	5·5	5·0
Piston speed, ft. per min.	1,181	1,083	984
R.P.M.	120	110	100
4 cylinders	7,000	5,800	5,400
5 cylinders	8,700	7,200	6,750
6 cylinders	10,400	8,600	8,100
7 cylinders	12,000	10,000	9,500
8 cylinders	14,000	11,500	10,800

Total engine stroke=2,000 mm.; i.e. main piston stroke=1,500 mm., and exhaust piston stroke=500 mm.

The values for b.h.p., m.i.p. and r.p.m. stated in Tables III and IV are, of course, capable of mutual variation within reasonable limits, the horse power developed per cylinder being the product of the mean pressure, the revolutions per minute and the cylinder constant. The power and speed ratings are conservative. All the cylinder combinations named are either in service or in process of manufacture. The piston speeds quoted are those of the main piston only.

The specific fuel consumption of the pressure charged engine can be expected to be lower than that of the equivalent non-pressure charged engine. This follows mainly from the power saved at the chain driven blower, and from the relative improvement in mechanical efficiency, consequent upon a higher power output being obtained from the same engine dimensions as before, but with an almost unaltered amount of frictional horse power. There may also be a slight gain in thermal efficiency.

Particulars of routine test bed trials are given in Tables V and VI. In these trials, no attempt was made to go beyond contractual requirements. Superintendent engineers, and others, should insist upon being fully satisfied, in the negotiating stages of a contract, that there is ample power in the propelling machinery for all requirements. At the test bed trials and on the sea trials, when an engine is new and not run-in, commercial prudence suggests that no more power should be taken from the machinery than is necessary for immediate purposes.

Table VII summarizes the auxiliary fan tests for the engine of Tables V and VI. In Table VIII the shop test results are given for an engine built by J. G. Kincaid and Co., Ltd., who are sub-licensees of Harland and Wolff, Ltd. The engine is a five-cylinder unit, designed to develop a maximum continuous service power of 4,900 b.h.p. at 118 r.p.m. The tests for the engine are summarized in the graphs of Figs. 57, the results for the turbochargers having been already given in Fig. 51.

Mechanical efficiencies of over 90 per cent are recorded on shop tests. For example, one recent routine test of a six-cylinder 750/2,000 engine, the contract rating of which was 8,600 b.h.p. at 112 r.p.m., showed: maximum firing pressure 55-59 atmos.; fuel consumption 0·34lb. (154 gr.) per b.h.p. hour, the fuel being marine Diesel oil; mechanical efficiency 92 to 93 per cent. But the author dislikes citing figures of this order as if they accorded with everyday practice, because too often they are apt to be quoted by managerial and non-technical men to the discomfort of operating engineers. The log abstract, recorded on the high seas, must necessarily have a different criterion than that of the test log dexterously compiled on an engine works test bed.

For the 750/2,000 engine, at 116 r.p.m., with 6·6 kg. per sq. cm. (93·9lb. per sq. in.) b.m.e.p., a fuel consumption of 0·35lb. (159 gr.) per b.h.p. hour, at 52 kg. per sq. cm. (740lb. per sq. in.) firing pressure, is reasonable. Typical indicator diagrams for P.I. and non-P.I. engines are shown in Fig. 58. A fuel consumption of 0·34lb. (154 gr.) is obtainable, with a firing pressure of 60 atmos. (853lb. per sq. in.). At this pressure, a compression of 600lb. per sq. in. (42 atmos.) is reasonable. Most superintendent engineers prefer not to go

TABLE V.—6–620/1870 SINGLE-ACTING TWO-CYCLE PRESSURE INDUCTION
ROUTINE ENGINE TEST, USING DIESEL FUEL

Load, per cent.	25	50	75	86·5
B.H.P.	1,675	3,350	5,025	5,800
R.P.M.	78	98·3	112·5	118
B.M.E.P., kg. per sq. cm.	2·88	4·57	5·99	6·58
Mechanical efficiency	0·71	0·81	0·86	0·87
Fuel consumption, lb. per b.h.p. hr.	0·41	0·37	0·36	0·35
Exhaust at cylinder, deg. F.	398	466	549	587
Exhaust before turbine, deg. F.	555	650	740	785
Exhaust after turbine, deg. F.	440	520	610	640
Compression pressure, kg. per sq. cm.	28·6	32·2	35·6	37·4
Firing pressure, kg. per sq. cm.	42·4	49·1	52·4	54
Scavenge pressure, lb. per sq. in.	0·89	2·12	3·83	5·07
Total air supplied, cu. ft. per min.	6,780	11,570	16,220	18,460
Air ratio	0·86	1·16	1·43	1·54

Fuel oil=Diesel quality; sp. gr. 0·874 (60 deg. F.); viscosity 44 sec. (100 deg. F.); gross cal. val. 19,370 B.t.u. per lb.

TABLE VI.—6–620/1870 SINGLE-ACTING TWO-CYCLE PRESSURE INDUCTION
ROUTINE ENGINE TEST, USING HEAVY FUEL

Load, per cent	25	50	75	86·5
B.H.P.	1,675	3,350	5,025	5,800
R.P.M.	78	98·3	112·5	118
B.M.E.P., kg. per sq. cm.	2·88	4·57	5·98	6·58
Mechanical efficiency	0·71	0·81	0·86	0·87
Fuel consumption, lb. per b.h.p. hr.	0·42	0·38	0·37	0·36
Exhaust at cylinders, deg. F.	441	491	550	604
Exhaust before turbine, deg. F.	595	677	753	804
Exhaust after turbine, deg. F.	465	547	647	635
Compression pressure, kg. per sq. cm.	28·9	32·5	35·8	36·8
Firing pressure, kg. per sq. cm.	42·2	48·8	52·5	54·2
Scavenge pressure, lb. per sq. in.	0·81	2·22	3·87	5·10
Total air supplied, cu. ft. per min.	6,800	11,790	15,880	18,720
Air ratio	0·87	1·18	1·40	1·56

Fuel oil=boiler quality; sp. gr. 0·965 (60 deg. F.); viscosity 1,163 sec. (100 deg. F.); gross cal. val. 18.505 B.t.u. per lb.

TABLE VII.—AUXILIARY FAN TEST

(i) *Engine particulars*
Size=6·620/1870; service b.h.p.=5,800; r.p.m.=118.

(ii) *Turbocharger particulars (design)*
Number=2; size=MS.500; r.p.m.=6,900; b.h.p.=223 (each).
Free air per blower=9,800 cu.ft. per min.; air pressure=5·9 lb. per sq. in.

(iii) *Fan particulars (design)*
Size=30 b.h.p.; r.p.m.=2,600; variable speed=2,000 to 3,000 r.p.m.
Air quantity=3,000 cu.ft. per min.; pressure=1·5 lb. per sq. in. (gauge)

(A) *Manœuvring conditions*
Number of fans=one; air quantity=3,300 cu.ft. per min.; air pressure=10 in. w.g.; fan r.p.m.=2,060; b.h.p. absorbed=11·6.

(B) *Emergency conditions*
Number of fans=one; air quantity=6,320 cu.ft. per min.; air pressure=17·6 in. w.g.; fan r.p.m.=2,600; b.h.p. absorbed=26·5.

TABLE VIII.—5–620/1870 SINGLE-ACTING TWO-CYCLE PRESSURE INDUCTION
ROUTINE ENGINE TEST

Load, per cent	25	50	75	100	110
M.I.P. (average), kg. per sq. cm.	3·4	5·0	6·4	7·6	8·0
R.P.M.	73·3	91·3	104·7	116·8	120·3
I.H.P.	1,529	2,826	4,138	5,476	5,964
B.H.P.	1,199	2,370	3,562	4,810	5,280
Mechanical efficiency	0·78	0·84	0·86	0·88	0·89
Maximum firing pressure (average), kg. per sq. cm.	40·4	46·3	50·6	52·3	53·4
Compression pressure (average), kg. per sq. cm.	27·2	29·7	32·1	35·0	36·4
Exhaust temperature (average), deg. F.	457	510	563	648	680
Induction air, lb. per sq. in.	0·4	1·6	2·1	4·9	6·2
Lubricating oil before filter, lb. per sq. in.	40	41	40	42	44
Lubricating oil after filter, lb. per sq. in.	27	31	26	30	30
Lubricating oil at engine, lb. per sq. in.	19	19	17	19	19
Piston cooling oil (rail), lb. per sq. in.	20	20	18	20	20
Lubricating oil before cooler, deg. F.	96	104	96	100	104
Lubricating oil after cooler, deg. F.	87	98	87	92	97
Main piston cooling oil return, deg. F.	100	108	100	103	105
Exhaust piston cooling oil return, deg. F.	100	105	94	99	99
Fuel injection pressure, lb. per sq. in.	4,000	4,210	4,050	4,590	5,340
Fuel, lb. per i.h.p. hour	0·32	0·31	0·29	0·31	0·30
Fuel, lb. per b.h.p. hour	0·41	0·38	0·34	0·35	0·34

Quality of fuel=Diesel; cal. val. 19,300 B.t.u.; specific gravity 0·85 at 60 deg. F.

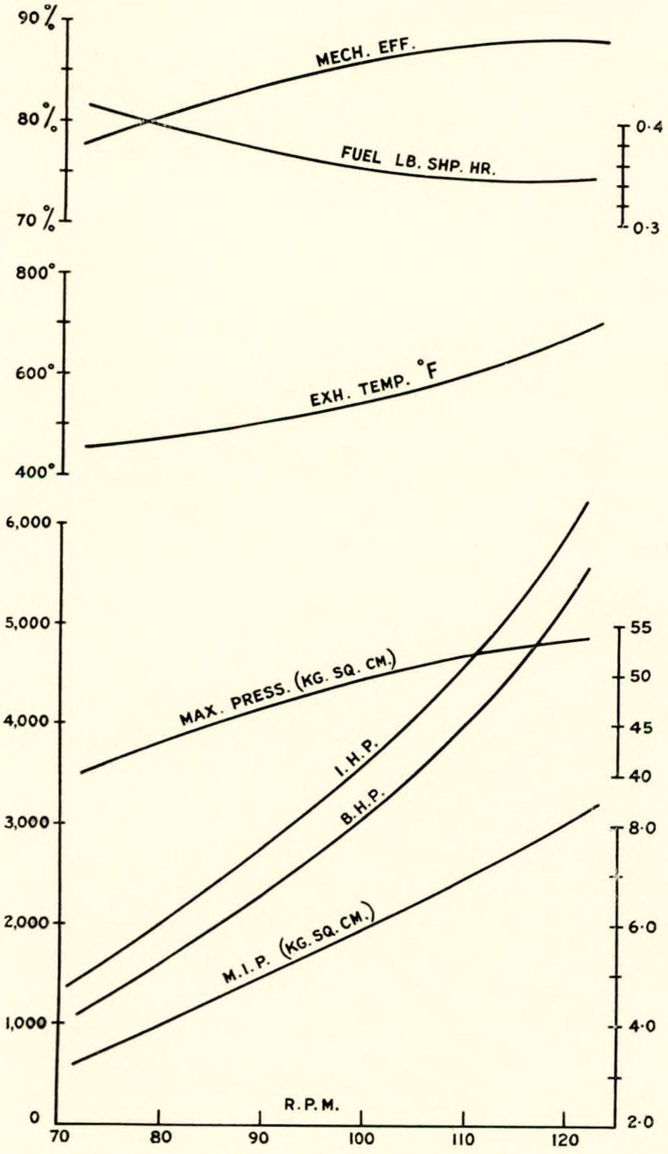


FIG. 57—*Test results of a five-cylinder 620/1,870 engine*

higher than 55 kg. per sq. cm. (782lb. per sq. in.). By advancing the fuel cams slightly, the higher pressure is easily obtainable. The author, for the time being at least, prefers the firing pressure not to exceed 800lb. per sq. in. (56·5 atmos.). The crankshaft is strong enough for 850lb. per sq. in. (60 atmos.). With a compression of 37 to 38 atmos. (say 525 to 540lb. per sq. in.) the engine will start in either direction without the use of the auxiliary fan; but it is advisable quickly to increase the engine revolutions on the fuel handle. A compression of 39 to 40 atmos. (say 550 to 570lb. per sq. in.) adequately allows for worn piston rings.

When the auxiliary fan is running under standby conditions, the air from the fan can pass into the turbocharger through one of the cylinders whose exhaust ports are open. The turbocharger can thus reach a speed of about 1,000 r.p.m., making easy the starting of the engine. With an eight-cylinder engine having two or more blowers the air from the auxiliary fan can spin more than one blower. On the shop trials of six-cylinder units, the engine speed has been reduced to 17 r.p.m. with the auxiliary fan in use. One superintendent recently told the author that, at sea, with the auxiliary fan in use, his six-cylinder engines could run at as low a speed as 12 r.p.m.; without the fan, the lowest speed was 50 to 60 r.p.m.

In computing the electrical load for an installation care must be taken to include the auxiliary fan. For all except the smaller cargo vessels this point will not be of significance.

Some engineers find it puzzling that the mean indicated pressures for the main and exhaust pistons should be different. There are two points to consider in this connexion. First,

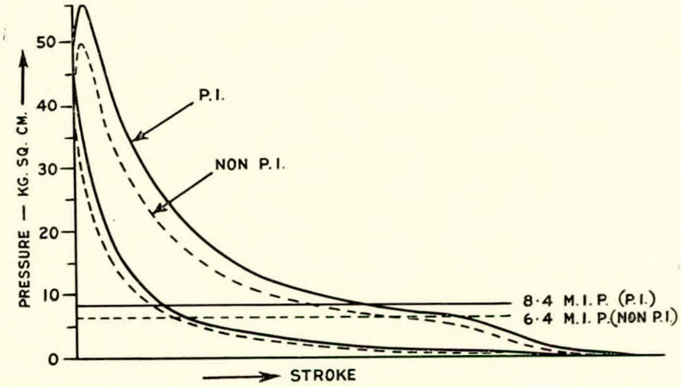


FIG. 58—*P.I. and non-P.I. indicator diagrams*

there is the difference in the connecting rod/crank ratio for the main and exhaust pistons, the former being 4 : 1 and the latter 10·48 : 1; second, there is the effect of the exhaust piston lead. The result is seen when the path of the pistons is followed. Thus, for a downward crank angle of 90 degrees the main piston has travelled 56·25 per cent of its stroke. The eccentric, in the same time, has moved 97 degrees from bottom dead centre and has travelled 53·75 per cent of its stroke. That is, for a point of equal pressure inside the cylinder, the main and exhaust pistons are not at the same relative point in their stroke. The difference, at the particular point named, is 2·5 per cent. Assuming an indicator card is 77 mm. long, then 2·5 per cent = 1·9 mm. An increase in the width of the indicator card of this amount raises the m.i.p. from 7·5 kg. per sq. cm. to 9·2 kg. per sq. cm. And so, for a cylinder m.i.p. of 7·9 kg. per sq. cm. (112lb. per sq. in.), the main piston m.i.p. is 7·5 kg. per sq. cm. (107lb. per sq. in.) and the corresponding exhaust piston m.i.p. is 9·2 kg. per sq. cm. (131lb. per sq. in.).

The indicator cam, to be correct, must impart to the indicator the same relative travel as the piston. On an actual engine the indicator cam ratio is designed to show a mean o the indicated pressures on the main and exhaust pistons.

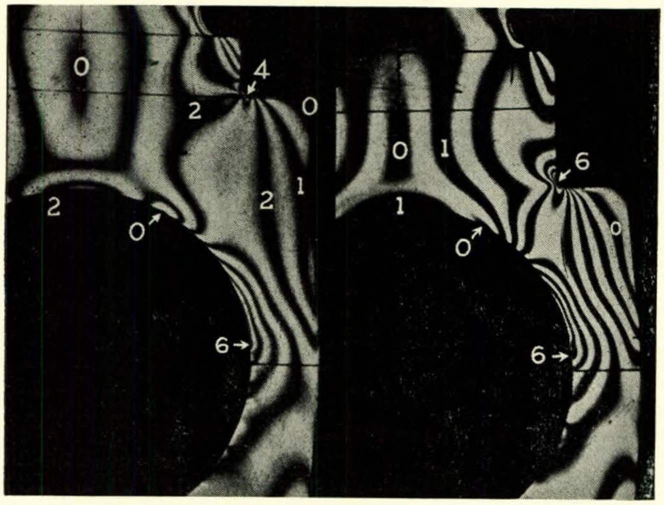


FIG. 59—*Photo-elastic study of eccentric strap*

Engine tests have shown the actual m.i.p. to be lower than that normally recorded. The implication is that the exhaust piston does more work than the ratio of strokes would lead one to expect.

A question which is frequently put to the author refers, naturally enough, to the amount of maintenance, in the overall sense of the term, likely to be needed for exhaust turbocharged engines in comparison with non-pressure charged engines.

In the present state of knowledge the author can only answer by *a posteriori* inference from experiences with four-stroke turbocharged machinery.

Following the turbocharging in Belfast, in 1926, of a two-year old cargo vessel called the *Lochmonar,* many single- and twin-screw ships, up to 12,000 s.h.p. and over, were propelled by pressure charged engines. The turbochargers for these vessels were built in Belfast and included the largest blowers of their kind ever made for marine engines.

Experience with this wide range of vessels showed that the engine upkeep was less than for engines having atmospheric induction. Seldom had the turbocharged engine liners an effective life below 1·5 times the life of similar liners in non-pressure charged engines. Often the life was doubled. Two or three cargo liners of the period are still in service—having survived the war—and their owners have repeatedly

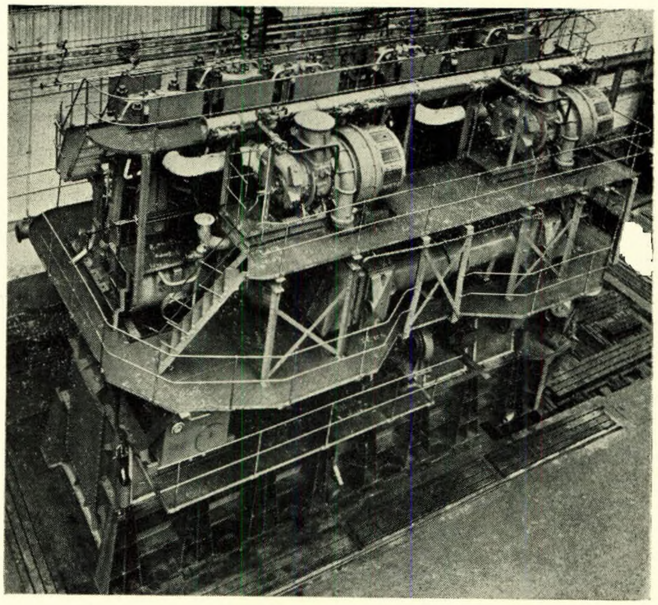


FIG. 60—*Arrangement of turbochargers on engine*

told the author that the engines were the finest they had ever had, especially as regards trouble-free running and low cost of maintenance. The turbocharged four-stroke engine was a *kindly* engine.

With two-stroke engines the results can hardly be expected to be so attractive, but the same trend should be apparent. In the Harland and Wolff engines the auxiliary fan may be said to be a minor complication. But against that must be set the expensive, and sometimes difficult, chain driven scavenge blower. The two-stroke turbocharged engine will almost invariably run on heavy fuel and some turbine deterioration can therefore perhaps be expected. Even so, it seems to the author that, on the lowest plane of reckoning, maintenance should be

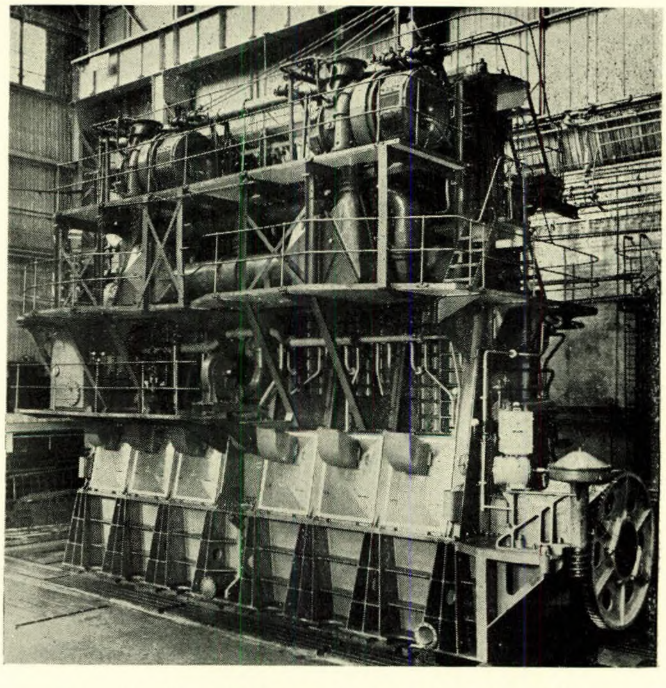


FIG. 61—*Six-cylinder turbocharged engine 620 mm. bore, 1,870 mm. stroke*

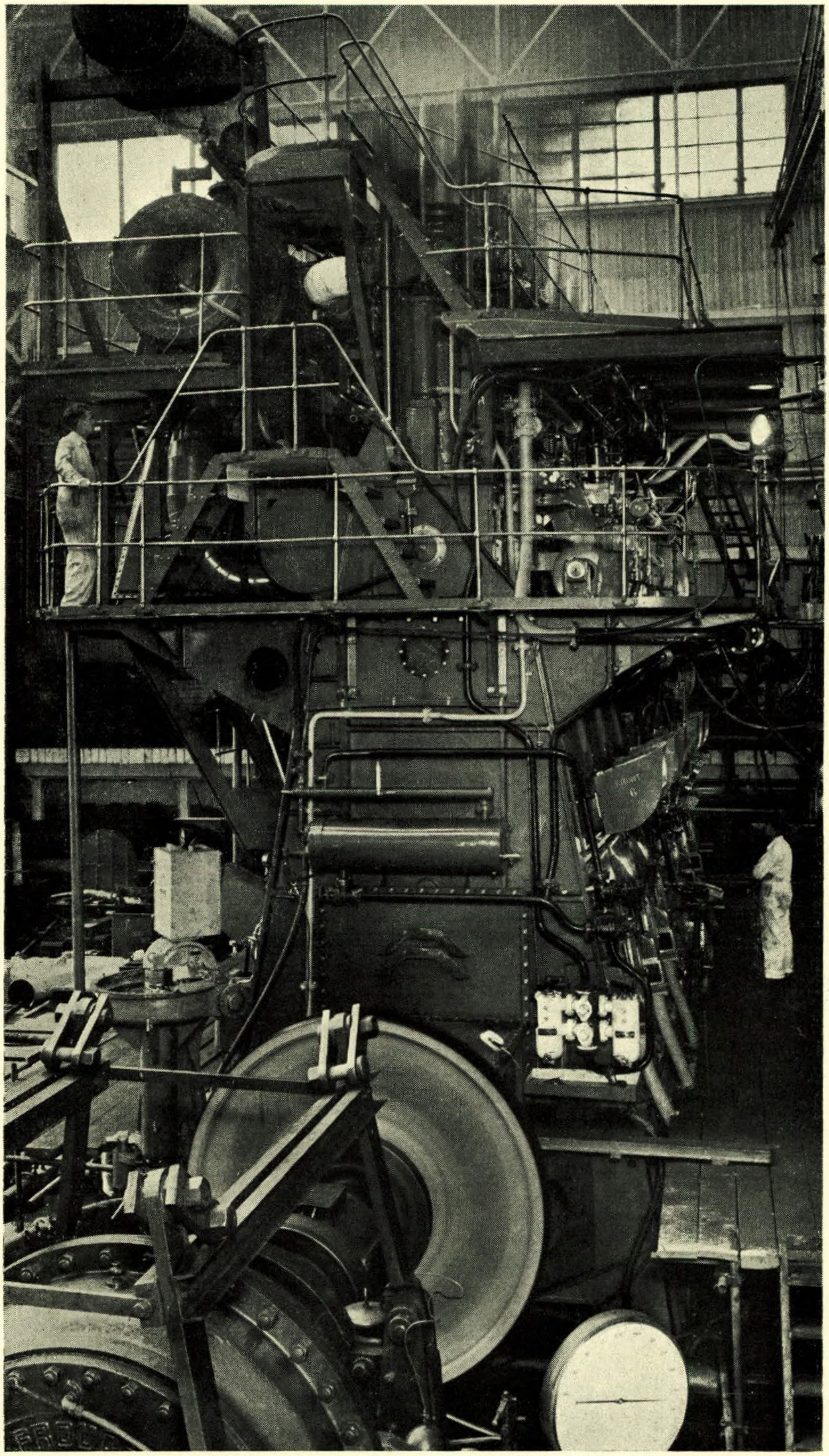


FIG. 62—*5,800 b.h.p. engine running at 118 r.p.m.; 15-sec. exposure of photographic plate*

no more costly than for the same size non-P.I. engine running on the same kind of fuel, despite the 30 to 35 per cent extra power gained.

CONCLUDING OBSERVATIONS

There is no difficulty in building a ten-cylinder engine, 750-mm. bore, 2,000-mm. stroke, developing a maximum continuous power of 15,000 b.h.p. A twelve-cylinder engine—while offering no technical obstacle, either in construction or running—the author would be inclined to pass over in favour of fewer cylinders of a larger size.

The recent substantial stepping-up in the dimensions of oil tankers has naturally been accompanied by an increase in the power of the single-screw engines used for propulsion. But it has not been a proportionate increase, as the very large tankers are not speedy. Accordingly, the increase in breadth, depth, and length of ship should permit of the installation of Diesel engines larger than those described in this paper.

So far as the author is concerned, he would have no hesitation in building engines up to 880 mm. (34·65 mm.) bore. There has been nothing in his experience to suggest that 750 mm. (29·53in.) is, in any sense of the term, a limiting size; rather the contrary.

At the present time there seems to be a spreading and pervading notion that 750 mm. is the rational limit of bore for a cylinder; that, beyond such a size, disproportionate difficulties can be expected. The experiences of different men, with different engines, naturally will not all be the same. The author, therefore, can only speak from the background of the varied engine types with which his work has lain.

In developing the non-P.I. engines which immediately preceded, chronologically, the P.I. engines described in this paper, and in taking the further step of turbocharging the engine type, nothing has surprised the author more than the negligible number of problems which have arisen in service. At the outset it was only to be expected that he should have spells of anxiety. For example, the application of eccentrics for operating the full-sized exhaust pistons of 8,000 s.h.p. engines—without any prototype—required an amount of doggedness to establish that was not negligible. Upholders of the arrangement, initially, were few and the critics many. But the tangible results have justified the latent faith.

There have been a number of instances of broken white metal in the eccentric straps when opened-up for survey. The straps ran fully satisfactorily, but as soon as they were lifted the crazy pieces fell out. The irony of this situation has been that almost all such straps were white-metalled by outside firms of specialists of the highest repute, to whom the work was given to ensure that the results would be beyond any peradventure. In all, however, the straps involved—of which the author has knowledge—constituted but a negligible number compared with the straps in service. That phase, in all probability, has now passed. It is an example of the operation of that natural law which seems to dominate new ventures, namely, the Law of Cussedness. Eccentric straps white metalled in Belfast and Greenock have always been fully satisfactory.

One exhaust piston cast steel yoke, in service, was found to have a serious casting fault. This incident seems to have gained publicity. Judging from his day-to-day experience with different types of prime mover, the author has yet to learn that bad steel castings are a monopoly of the Diesel engine.

The foregoing two matters are, however, but venial faults.

There have been a number of cracked cylinder liners in the non-P.I. engines. Some were undoubtedly due to loss of jacket cooling water. Some of the remainder arose from imperfect technique in the application of cast iron chills in the foundry; others were probably caused by unequal internal stresses from over-heavy bosses at the fuel and other valves, augmented by too heavy a use of the expanding tool on the stainless steel sleeves. The bosses were removed and the use of the expander severely controlled. Some of the cracks were serious; most were incipient and superficial, the liners lasting their full term. But the important point, for present purposes, is that none of the things mentioned was bound up with the cylinder size. Equivalent occurrences have been experienced, in times past, in cylinders considerably smaller than 750 mm. bore.

An eight-cylinder engine of 880 mm. bore and suitable stroke would conservatively develop 20,000 b.h.p. The finished crankshaft would weigh about 200 tons. The length of engine over the crank ends would be approximately 70ft., the breadth of bedplate 16ft., the height from centre of crankshaft to top of exhaust piston yoke 35ft.; the height from centre of crankshaft to crane-hook 43ft.; total weight of engine 950 tons. The size of the component parts of a powerful engine—as already mentioned in another connexion—does not necessarily constitute a problem; it is a matter of providing appropriate, easily managed, power driven, overhauling gear.

The rating mentioned, namely 20,000 b.h.p. on eight cylinders, is based upon current levels of accepted practice. With a higher level of turbocharging and with an adjustment of cylinder dimensions, an output approaching 24,000 b.h.p. could be attained without hazard.

The author does not state that a considerable increase in cylinder size and in developed power of direct coupled engines *will* be the next step. He simply draws attention to the complete practicability of successfully constructing engines more powerful than those at present in service. Direct coupled engines, when all is said, are simpler in construction, lower in initial cost, more economical in fuel, and more easily capable of burning the heaviest fuels, than are any of the Diesel engine alternatives.

Discussion

MR. A. G. ARNOLD (Member) said he was very pleased indeed to be asked to open the discussion on Mr. Pounder's most interesting paper; it was an honour he very much appreciated. It had been his pleasure to have worked with Mr. Pounder for very many years and he was able to recognize the great work he had done for engineering in general and the internal combustion engine in particular.

Mr. Pounder reminded them that it was quite a long time since he wrote a paper for this Institute but he was sure they would all agree that the paper was well worth waiting for, and in the main he agreed wholeheartedly with what Mr. Pounder had to say about the engine and its performance.

Mr. Pounder's remarks concerning the crankshaft being the backbone of the engine were profoundly true, and he hoped that he and other designers would not yield to the craving for reduced weight of the bedplate, which was, of course, as essential for successful operation as the crankshaft. The bedplate must be stiff enough to ensure perfect alignment of the crankshaft under all conditions of service. He himself believed that engine builders should never have discarded the cast iron bedplate; however, he thought it would not be long before a return to this type was made. In Fig. 1 he was pleased to note that Mr. Pounder had now adopted the shallow type scavenge air manifold, and felt many members would be interested to know the reason for this change; could it be due to the excellent results achieved in Messrs. Alfred Holt's fleet, who had only these shallow manifolds? Furthermore, could Mr. Pounder tell them the reduction in weight of the engines due to this major alteration, and perhaps cost? The reduction in depth from 1,900 mm.-1,180 mm. was considerable; could Mr. Pounder say whether this alteration had any appreciable effect on the centre of gravity of the engine? This aspect in cargo liners, with the advent of supercharged engines, was becoming increasingly important.

On page 163 Mr. Pounder mentioned that scavenge air manifolds were mainly of interest to designers and builders but he maintained that the engineer in charge of the plant should have an intimate knowledge of these parts of the engine in order to reduce the possibility of fires, and to deal efficiently with them should they occur.

On page 165 Mr. Pounder gave most informative details concerning the area of the ports, etc., also the point of opening and closing. This interesting paragraph prompted him to ask Mr. Pounder if he had ever considered constructing this engine to work in conjunction with a variable pitch propeller.

On page 166 reference was made to the fuel injection system. Mr. Pounder would know how interested he had been in the fuel injection system in this type of engine and others in his company's fleet. In conjunction with this, he would like to say how pleased he was to read that Mr. Pounder had now improved the engine by fitting gas operated pumps. He thought, judging by the illustration on page 167, that the system was more complicated than it need be. He was sorry, however, that he had seen fit to retain the camshaft and timing. He was also very sorry to note that he had retained the flat seats for the fuel oil valve. It was noticed that there had been two unnamed vessels at sea fitted with this system for two voyages, but Mr. Pounder would also know that Messrs. Alfred Holt and Company had nine vessels with non-supercharged engines and three vessels with supercharged engines which had made a total of 122 voyages, 121 of these being entirely successful, and that these engines had no camshaft. Records showed that there was no excessive firing pressure, and, furthermore, there had been no complaints from responsible people about the manœuvring of the machinery. In fact, marine superintendents, masters and pilots had often commented favourably upon this aspect of these vessels. The engine, being able to run at twelve revolutions for an indefinite period and to start at the remarkably slow speed of 35-40 revolutions, was often commented upon. A further important point to be made was that the first three vessels with this type of machinery were fitted with the builders' fuel injection equipment; fires occurred in the scavenge air manifold, making it necessary to replace the builders' equipment with gas operated pumps and pilot injection fuel oil valves, and he was happy to state that since this was done not one fire had occurred in any one manifold of any one vessel.

Table IX gave the records for Voyage 3 of m.s. *Dolius*.

Fig. 63 showed the difference in the dimensions of the crankshaft for the two classes of engine to which he had referred.

The m.s. *Dolius* and the engines referred to were built by Harland and Wolff, Ltd., Belfast. The results, he thought, were an interesting study for the marine engineer and naval

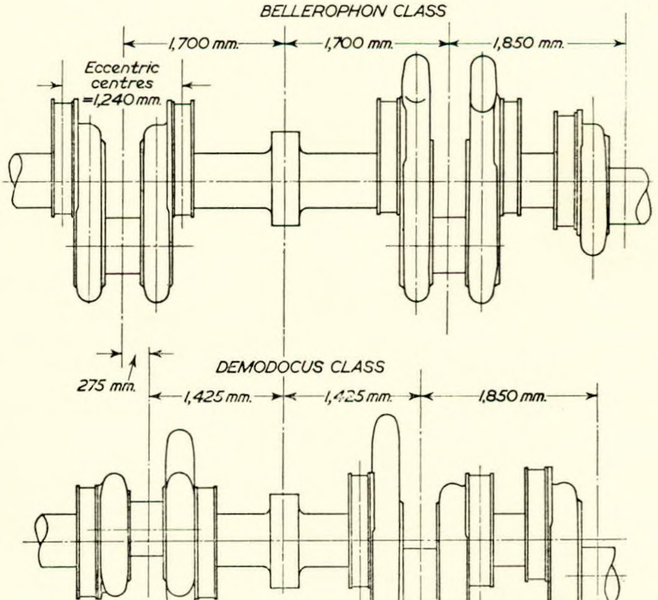


FIG. 63—*Reduction in length between* Bellerophon *and* Demodocus *class crankshaft centre sections*

architect. Incidentally, they were the best voyage records ever to have been received in the office during the long history of his company.

Mr. Pounder mentioned that the elimination of the camshaft reduced the weight by a very little, but would Mr. Pounder tell them how it affected the cost of the engine?

On page 166 the author said that no change in the length of the engine could be expected, due to elimination of the camshaft; he thought this statement was correct but would Mr. Pounder say whether, by removing the gears for the operation of the Roots blower alone, this would enable the engine to be shortened were the camshaft retained?

PROFESSOR A. F. BURSTALL, D.Sc., Ph.D., expressed appreciation of the opportunity given him as a visitor to speak on what was, he thought, an historic occasion. It was certainly an historic occasion to him to be allowed to drink, as it were, of the Pounder philosophy—if he might call it that—on the design of marine engines.

There were not many pressure charged marine engines about and that Mr. Pounder had given in a single evening such a wealth of philosophy, facts and diagrams was something of which the Institute would be very proud.

It was difficult for anyone like himself who had not had experience of operating these engines to comment or criticize what a designer had done. Most of his contribution consisted in asking for more information. They had had a good meal, but like Oliver Twist they wanted more.

What had interested him most about the paper and the author's comments at the outset were the reasons which led to the choice of the 30 per cent supercharge. He thought those were the units in which one usually spoke of this supercharging. He understood it was about 30 per cent or 6lb. per sq. in. The choice was, he thought, 14·5 : 1 for the compression ratio. This figure came from page 165 where the stroke volume and clearance volume were given.

Did this represent the limit of what could be done with the present materials in the sizes for which these engines had been built? As he understood it, in the remarks at the end of the paper Mr. Pounder spoke of very large engines indeed—24,000 b.h.p.—and presumably this also related to 30 per cent supercharge. He did not understand why. Perhaps this was due to his own ignorance, but he thought others also might like to hear why this was regarded as the present limit of the amount of supercharge.

He asked whether Mr. Pounder would care to offer any reasons as to why the wear of the pressure charged engine was less than with atmospheric induction. Other people had made the same statement, and it would be interesting to know why.

Finally, would it not be an advantage to have some idea of the scavenge efficiency at that particular scavenge ratio for the conditions of the tests, if Mr. Pounder had that information?

MR. P. PLUYS said it was with much profit that he had read the paper, not only because it gave a detailed description of a new engine but also because there was a real abundance of numerical figures which many people would find interesting. He would like to ask the author a few questions.

The cylinder liner described in the paper appeared to be of a new design, being in three parts, with the central part round the combustion chamber in cast steel. This should normally improve the safety margin. He well remembered three or four years ago that he had come over from Antwerp to this country to obtain Mr. Pounder's advice on the very delicate problem of liner failures in engines of similar types, but double-acting. Subsequently, the design of liners was improved by reducing to a minimum the bossings for the valves. A stress relief was also applied with good results as there was now a definite reduction in failures, even in turbocharged engines with all cast iron liners. Had the present construction, which was more complicated, been designed specially for turbocharged engines or was it also needed for normal engines?

The second point of interest concerned the exhaust piston. They had had a lot of trouble with burned heads, and for the latest engines that had been ordered they had gone over from oil to water cooling. This proved to be a radical solution. In 8,000 hours of service in a supercharged engine the exhaust pistons—which did not require any opening for cleaning—were absolutely as new. Unfortunately, there was a drawback in generalizing that system. Several times on the test bed an increase had been measured in fuel consumption from 3 to 4 per cent with water cooled pistons, the cooling being really too efficient. For oil cooled pistons they were a little afraid to use stainless steel as basic material, and they had rewelded the burned-out heads with three or four layers of special electrodes, using heat resisting steel in such a way that after heat treatment and stress relieving the piston heads were much more resistant to burning than new parts in chromium-molybdenum steel. What was Mr. Pounder's own preference, and what did the customers of this new engine in England ask for—oil or water cooling, especially for the large-bore engines?

On the turboblowers for pressure charging, the author seemed to be averse to the use of ball bearings. His company was making its own experience on this subject as they had blowers with ball bearings and others with sleeve bearings. The former had to be renewed at regular intervals. The latter required an independent oil circuit for bearing lubrication and cooling. Did the author not think that the friction losses were reduced when ball bearings were used and that consequently starting and slow running were improved? There was a case, by accident, where a new engine could not be started until the length of the sleeve bearings had been reduced to half, which proved that the losses with these bearings were not at all negligible. He understood that in a short time high quality ball bearings would be available with a service life of over 10,000 hours instead of the present 5,000 hours. Nevertheless, he wished to add that up to the present there had been no trouble at all in his firm with the big size Napier blowers in service. Would Mr. Pounder give his reasons for preferring sleeve bearings?

MR. P. JACKSON, M.Sc. (Local Vice-President), said he had read the paper and listened to the author with great interest. He had never previously seen a paper containing so much detail of the construction of an engine; even dimensions and thicknesses of plate had been given. If a good designer ever required to build a Harland and Wolff engine it would not be necessary to take out a license!

He had been very interested in Mr. Pounder's remarks on page 161, outlining the aims of a designer. He supposed that most designers had these aims and ideas at the back of their minds when designing an engine but none of them had put his aims into writing so concisely. All marine engines must have a low fuel consumption and must be designed for a low first cost, but primarily attention must be given to the minimum of maintenance and ease of overhaul. During his long association with William Doxford and Sons, Ltd., Mr. Purdie had always maintained that it was well worth spending £1,000 to reduce the fuel consumption of an engine by one per cent.

Mr. Jackson had gone through the paper very carefully and had marked a number of points on which he would like further explanations. Mr. Pounder stated on the first page that the reduction in weight of a supercharged engine relative to a normal engine of the same power was 27 per cent, and he himself would agree with this figure, but later on, on page 187, Mr. Pounder stated that the weight of the supercharged engine was 20 tons lighter than the weight of the corresponding normal induction engine, due presumably to a reduction in weight of the blowers and that the supercharged engine gave 30 to 35 per cent more power. This did not quite tie up with the 27 per cent saving in weight and he would like Mr. Pounder to give a further explanation.

He would agree that the specific fuel consumption on a supercharged engine was 4 per cent less than that of a normally aspirated engine. This could be proved on the test bed, but experience at sea seemed to show that the fuel saving was more like 10 per cent. He wondered if Mr. Pounder had had the same experience and if he had any explanation?

With regard to three-piece linear construction, it would appear that all those concerned with the construction of opposed piston engines had been thinking along similar lines. Mr. Arnold had shown a slide of a three-piece liner some two years ago; Mr. Andresen showed one for a B. and W. engine last year, and he himself had shown one for a Doxford engine. Now Mr. Pounder was also showing one. He supposed that they all had had the same difficulties and were trying to remedy them in the same way. There was one point about Mr. Pounder's liner which he himself did not like. The joints were internal in the water space, and if a joint became defective water would get into the cylinder.

Mr. Jackson said he was interested in the dimensions given for the gas operated pump on page 166, where it was stated that these dimensions were common to both sizes of engine. This gas pump seemed to be much smaller than the jerk pump which Mr. Pounder had described in his paper* to the Institution of Mechanical Engineers, some years ago. This pump for the large engine was given as 42 mm. diameter × 65 mm. stroke, whereas the gas pump was of only 37 mm. diameter × 36 mm. stroke, which would appear to be too small for the large engine and particularly when supercharged.

Again, he was intrigued by the balancing of the seven-cylinder engine. His company had never built a seven-cylinder engine so he had not had to go into this question in great detail, but he noted that the balance weights on the centre of the crank were heavier than those on the end, certainly only half a ton heavier, but it seemed to him that with equal phase angles between the cranks the crankshaft was inherently balanced as regards rotary forces. If heavier balance weights were put in the centre it would create a slight revolving force which was not beneficial, and what purpose could it serve? In other respects he assumed that the horizontal residual couple shown and the vertical residual couple were those given by the rotary couple due to the eccentrics being some seven degrees in advance of the centre crank. This rotary couple could have been balanced by some offset weight on the balance weights attached to No. 1 and No. 7 cranks, and he wondered why this had not been done.

The firing order which had been chosen for the four-cylinder supercharged engine was also interesting. He believed that the firing order of the normally aspirated H. and W. engine was similar to Doxfords, being 1, 2, 4, 3, or 1, 3, 4, 2. On the supercharged engine Mr. Pounder had chosen a firing order of 1, 4, 3, 2, which enabled the turboblowers to be coupled up very nicely, whereas the normal firing order introduced a long pipe connecting up cylinders 1 and 4. On the other hand, the best order of firing to suit the turbochargers would be 1, 3, 2, 4, which however gave a fairly big out-of-balance secondary couple. He presumed that it was this couple which had influenced Mr. Pounder's choice. The orders of firing for other sizes of engine were much the same as had been considered and investigated by many designers over the past three years, since the turbocharging of two-cycle engines had become so fashionable.

Mr. Pounder had outlined concisely and explicitly the methods of obtaining slow running on a turbocharged engine. He himself was not quite sure that the arrangement given by the Napier system was the simplest. The one fitted by Mr. Arnold to the *Demodocus,* of a motor on the end of the turbo blower shaft, was a very simple arrangement. He understood, however, that there had been difficulties with the Airflex clutch used with this arrangement, but doubtless if Mr. Arnold were to adopt this system again he would do it with a mechanically operated clutch from the speed control lever, which would then give a very simple solution. The arrangement of under piston pressure charging also gave a very simple solution and he wondered what the patents were which Mr. Pounder had "had on his file for a decade or more". The Napier system involved a considerable amount of large diameter piping on the back of the engine with non-return valves and blank flanges which had to be taken out for running under various conditions. Both the motor on the end of the turboblower shaft and under piston pressure charging assistance to a turboblower gave these various conditions automatically.

With regard to maximum pressures, he was in agreement with Mr. Pounder that these were bound to increase on the pressure charged engine. The compression pressure had been given as 30 atmospheres, but from the diagram, Fig. 58, it appeared to be more like 40 atmospheres, and it was well known by those who had experimented with combustion that to obtain a good fuel consumption and good exhaust, the maximum pressure must be some 20 atmospheres greater than the compression pressure. Fuel consumption would certainly be detrimentally affected if attempts were made to reduce the maximum pressures by retarding the fuel injection, and the compression ratio had to be high enough to secure good starting and therefore the compression pressure would automatically increase on a supercharged engine with increase of air charging pressure.

MR K. MOLLER said it had been very interesting indeed to hear Mr. Pounder's detailed description of the turbocharged two-stroke engines being built by Harland and Wolff, Ltd., and by John G. Kincaid and Co., Ltd., Greenock.

On page 163 the author mentioned shot-blasting of welded structures. This was undoubtedly good practice, as when efficiently done it introduced compression stresses in the surface of the welding and closed minute surface cracks that might occur in the welds. Further, it reduced notch sensitivity in the transition zone between welds and plates. In Copenhagen the welds were magneto-fluxed to ascertain faultless surfaces.

On page 170 Mr. Pounder stated that steel wedges were driven between the bedplate and the side chocks. This also was considered good practice, as it created compression stresses in the bedplate when the vessel was fitting out. Usually it had a small draught when it was fitting out. Therefore, tension stresses caused by bending upwards of the ship's bottom due to increase of hydrostatic pressure at full draught was avoided or at least reduced by driving in wedges.

In Copenhagen there had been cases where such tension stresses due to too lightly fitting side chocks seemed to have been at least a contributory factor in cracks in the joint between longitudinal and cross girders in the bedplate.

In Fig. 45 on page 182 the eccentric rods were shown. On the Continent an alternative design was now being introduced. This design had two separate steel straps bolted to the lower part in the same way as the crosshead pin was bolted to the gudgeon pin at the upper end of the rod. This was a cheaper solution with the available production facilities. The design had earlier been used successfully in Burmeister and Wain double-acting opposed piston engines.

On page 186, under exhaust turbocharger arrangements, the author remarked that the substantial increase in power which was obtainable with a relatively low boost pressure was mainly the result of utilizing the high energy impulses at exhaust release. That was not quite clear to the speaker. In the opinion of his company the magnitude of the boost pressure was determined out of regard to obtaining a sufficient amount of air trapped in the cylinder for the load in question. Here air cooling played a very important part, as it increased the density of the air charge. Thus, at a relatively low charging pressure it was possible to reduce or at least maintain the heat flow through the cylinder walls as compared with non-supercharged engines.

The energy in the exhaust release period was a considerable part of the energy necessary for driving the turbine. By correct design of the engine timing, arrangement of exhaust pipes, turbine areas and turbine heat drop characteristics, it was possible for the opposed piston eccentric type engine to

* Pounder, C. C. 1949. "Some Current Types of Marine Diesel Engine." Proc.I.Mech.E., Vol. 160, p. 312.

obtain sufficient air under retention of a good fuel economy and without an additional engine driven blower. Therefore, an auxiliary blower was needed only for starting and dead slow running.

The next point related to the description given of the auxiliary and emergency blower arrangement and the various possibilities of continuing the operation of the engine in emergencies where one or perhaps two of the superchargers of an engine had been rendered inactive. His company had many turbocharged engines in service with turbochargers of different makes, and according to their experience these turbochargers were very reliable. The first turbocharged engine went into service in October 1952 and since then well over 150 had been put into service. Until some weeks ago there had been no emergencies where it had been necessary to use the emergency blower. About four weeks ago, however, it was found necessary on board a ship engaged in the coal transporting business. This ship had, after a trouble-free voyage, stopped at some distance outside the harbour of destination, waiting for the pilot. When the engine was to be started again it was quite impossible to make the turbochargers move. Therefore, the necessary steps were taken to operate the engines with the emergency blower only.

When the ship arrived in harbour and the rotors of the turbochargers had been dismantled, it was found that there were heavy deposits on the blower side as well as the turbine side. The deposit on the blower side was a black oily substance. The deposit on the turbine side was a dark grey brittle deposit. Which deposit actually caused the seizure of the rotors was not discovered. But as regards the deposit formed on the turbine side, his company were under the impression that the formation of deposits in the turbines was more likely to occur where some of the new neutralizing cylinder lubricants were used for the engine cylinders. The ship in question used neutralizing oil and it was running on heavy oil. The same high rate of deposit had not been found on other engines running on heavy oil but which did not use neutralizing cylinder oils.

Mr. Pounder had dealt with two engine sizes—750 and 620 mm. bore engines. Smaller engines of the same design, turbocharged, had been built in Oslo and Copenhagen. The bore was 500 mm. and the length of the combined stroke was 1,500 mm. There were already six engines of this type in service and a further twelve or thirteen were under construction or on order.

Regarding the steadily increasing demands for higher engine outputs, for instance, for big tankers, besides the engines with four to eight cylinders of the larger motor type referred to in the paper, there were three nine-cylinder engines on order in Oslo, each with a continuous service output of about 15,000 b.h.p. at 120 r.p.m.

Owing to the wishes expressed regarding still higher outputs, it had been decided in Copenhagen to build a poppet valve turbocharged engine with a cylinder bore of 840 mm. and a length of stroke of 1.800 mm. This engine would, on twelve cylinders, have an output of about 20,000 b.h.p. At the moment, negotiations were going on with Norwegian, French, and Japanese owners about the delivery of 10- and 12-cylinder engines of this type for tankers of 45,000 and 65,000 tons deadweight.

No decisions had been reached as yet about the building of turbocharged opposed piston engines of increased dimensions, as mentioned by Mr. Pounder.

Mr. W. D. Ewart (Member) said that masterpieces had one thing in common; they might be approached from all angles and still satisfy. The same might be said of Mr. Pounder's excellent paper of which all thirty-six pages would be read with equal profit by probationer student and member alike. This was not always the case as some papers were unavoidably specialized and would lose their interest if expanded to suit all tastes. One of the many outstanding features of the paper was that it was about a British-built product, the most powerful main propulsion Diesel engine built in this country. This fact should be borne in mind. It placed this country on equal terms with European and Far Eastern engine builders.

This was not a projected engine. It was in service and had proved under normal working conditions to be both reliable and economical. It represented the result of the steady development of the two-stroke single-acting unit with the added advantage of exhaust gas driven turboblowers. The upper limit of the engine's power range was not yet fixed, but at 12,500 b.h.p. it would appear to be worthy of serious consideration for twin screw installations where 25,000 b.h.p. was a useful proposition.

Such an arrangement would carry with it numerous advantages when applied to 40,000 to 50,000 ton tankers where it had been proposed to use single-screw turbine installations. When the immense value of a tanker's cargo was considered in conjunction with the vessel's high daily rate of earning, it seemed an unnecessary risk to place it all at the mercy of a single prime mover which could easily be immobilized.

It might be argued that the loss of efficiency involved in fitting twin screws, together with the higher capital cost, outweighed the advantages, but was this so? The major charge in a tanker's running account was the fuel bill, and this would always remain the prime factor.

When problems of limited deadweight capacity imposed by restrictions of draught in certain localities were considered, there was the added attraction of a gain in carrying capacity due to the reduced bunker requirements with Diesel propulsion.

Recent analyses of steam- and motor-tanker tonnage indicated the continued preference of Scandinavian tanker owners for Diesel-engined tonnage. Admittedly, they had proceeded cautiously in placing orders for twin-screw Diesels for the larger ships, but the orders were growing. British-built ships were a valuable export, and it was encouraging to know that this country was in a position to meet its customers' demands with regard to both hulls and engines.

It would not be out of place to enquire if a speed of 15 to 17 knots was an economical one for a twin-screw 100,000-ton tanker. Should it not be lower and consequently within the power range of the turbocharged two-stroke?

Rear-Admiral W. G. Cowland, C.B. (Member) said previous speakers had said justifiably nice things about Mr. Pounder and some not quite so nice, and some extremely pertinent questions had been put. These criticisms and questions would be given due consideration. His paper suggested a new line in papers as compared with anything he had known before. An eminent builder gave full and frank information about his product, and Mr. Pounder was to be congratulated on his frank approach. Some of the information in the way of curves and out-of-balance forces and so on would not—he was afraid—be of great use to the "hewers of wood and drawers of water", of whom he was one and was proud to be. But it contained a lot of meat which would give food for questions by professors and examiners. Any students who were present were strongly advised to read the paper very carefully: he felt sure some examination questions would be based on it.

The paper might be described as one by an experienced father who was fond of his offspring but had no delusions that they were prodigies. Although he called his engines P.I., that did not stand for prodigal infants. He explained how they had developed. He mentioned the troubles he had had and modestly implied that he did not want to send his ever-increasing offspring into the world without giving them the benefit of his ever-increasing knowledge, so that they would be ever the better able to earn their living.

He gave a hint that he was still young and vigorous and might become the parent of an exciting new addition to his family. This latest addition was conceived, and if it were born it would undoubtedly take a prominent place in the expanding and increasing powers required for ship propulsion. He wished Mr. Pounder luck and every success with it.

It was fuel economy that made the Diesel engine so attractive and he believed that up to powers of about 20,000 h.p. it would compete as regards initial cost and maintenance with the super high pressure, high temperature steam plant, which in its effort to compete with the Diesel in economy had to accept increased complications, higher stresses and lower factors of safety; all of which tended to increase maintenance bills.

He personally wondered why, especially for tankers of high power, single as opposed to twin screws were almost always specified. It would seem that for a 20,000-h.p. tanker, twin screws driven by two 10 000-h.p. Diesels was a most attractive proposition, taking all factors into consideration. Perhaps the author would comment.

On page 192 he rejected any desire to claim remarkable fuel consumptions. Like Gallio he "cared for none of these things".

Mr. Pluys had referred to ball bearings as compared with plain bearings on slow running. The ball bearing ran, of course, with a somewhat lower coefficient of friction than that of the plain bearings. The order of saving was about 0·2 or perhaps 0·3 h.p. on a big blower at low speed. This was not a very serious point; but if the ball bearing blower had also to drive its own little pumps there was really nothing much in it. There was always liable to be a rogue in any family, and the ball bearing was not peculiar in this respect.

Mr. Jackson asked about the turboblower and the help required. He suggested that an electric motor to assist the blower would be satisfactory. That was proved by Messrs. Alfred Holt and Company in their first turbocharged ship, the *Demodocus*, and it was entirely satisfactory. There was, however, one objection: namely, that with two blowers and two motors, if both blowers became inoperative there was no air for the engine and the ship could not go. He felt sure that was why Messrs. Alfred Holt and Company had abandoned this system.

On page 188 Mr. Pounder gave the basic conditions for the successful operation of turbocharged two-stroke engines. In his (Admiral Cowland's) opinion these were important. The "Napier" system had been described in some detail in the paper. Figs. 64-68 presented the method diagrammatically and

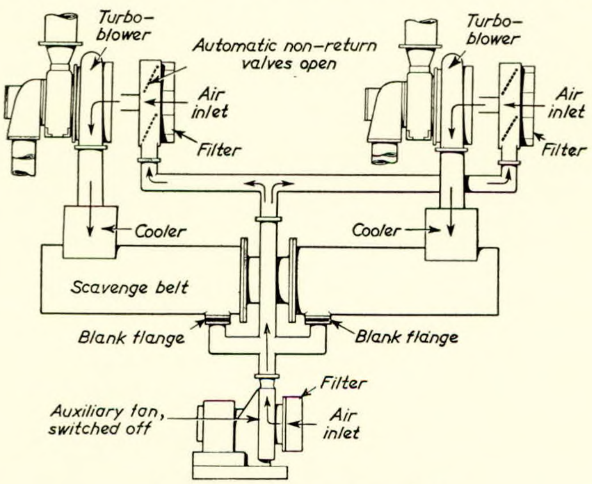


FIG. 64—*Normal running*

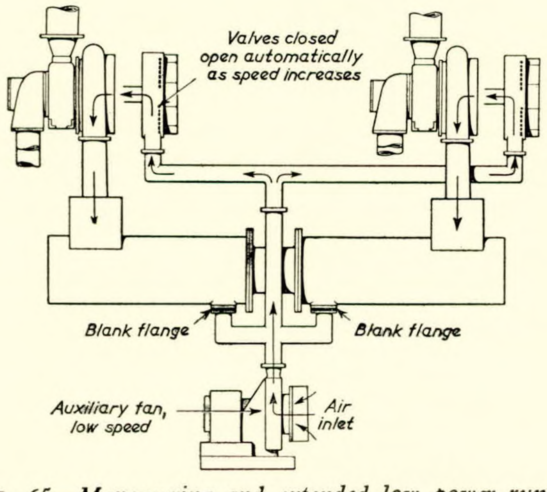


FIG. 65—*Manœuvring and extended low power running*

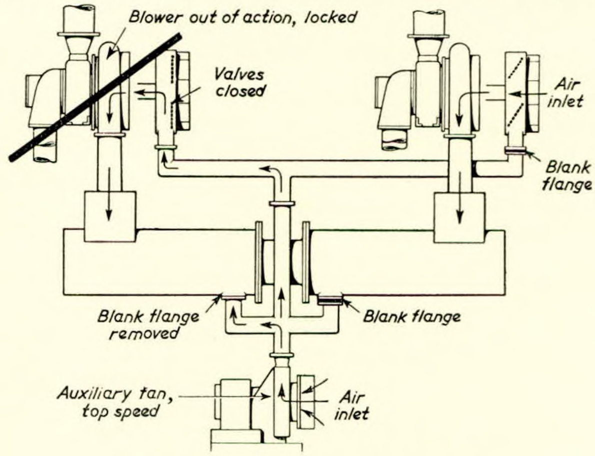


FIG. 66—*One blower out of action*

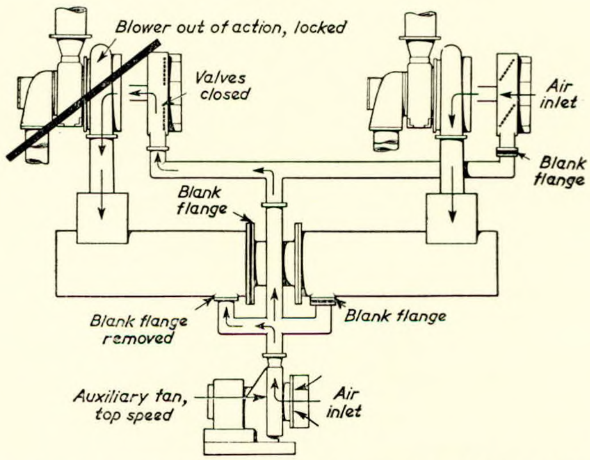


FIG. 67—*One blower out of action (alternative to Fig. 66)*

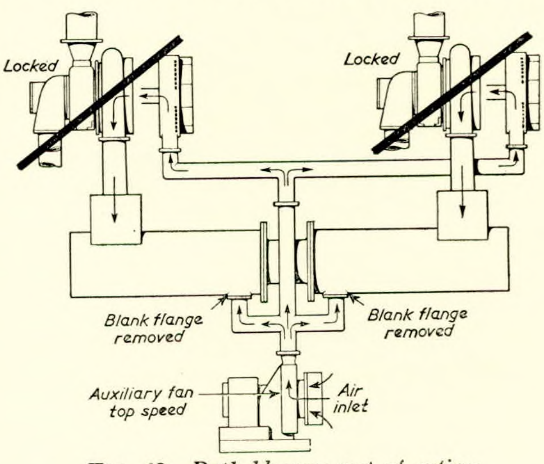


FIG. 68—*Both blowers out of action*

might amplify and perhaps simplify the diagrams in the paper.

He thought that the important points in favour of this system were: —

(a) In the event of trouble a quick changeover was possible.

(b) It was not necessary to bypass the exhaust.

(c) The vessel could continue to run indefinitely at reduced power.

The actual power obtainable was of course dependent on the kind of trouble in the engine or turboblower and the size of the auxiliary fan. As an example, however, it was worth noting that a fan taking only 22 h.p. was just sufficient to give 75 per cent of full r.p.m. with both blowers locked out of action on the five-cylinder SA6 620/1870 Harland and Wolff engine described in the paper.

Admiral Cowland admitted to surprise at the low power necessary to give sufficient air for this speed.

MR. N. GRAM thanked the Institute for allowing him, as a Danish visitor, to say a few words.

Although he was only slightly connected with Diesel engine building, construction and operation, he had read the paper and listened to the author with great pleasure. It might be of further interest to make one or two comments on bearing design for Diesel engines. He did not propose to make any criticism of this engine design, however, since he understood the engine was performing very well. He would only point to certain differences in design practice between the system outlined in the paper and that used by other builders.

The sleeve bearings, or plain bearings, were not described in much detail. How could they be, when the whole engine had to be covered in the paper? But Mr. Pounder had referred several times in his paper to the thickness of the white metal layer. This had generally been 8 mm. with reinforcing strips 4 mm. thick. Scandinavian and other Continental engine builders were tending to use thinner white metal linings. For instance, on crosshead bearing there was only half that thickness, that was, approximately 4-5 mm. or even less, of white metal in a bearing 400 mm. in diameter. Since his company were suppliers of white metal, they would prefer engine builders to use much thicker linings; but it must be admitted that bearings performed better with comparatively thin linings of white metal. This held for bearings that were exposed to blows, as were Diesel engine bearings. The fatigue strength of the lining was improved with bearing surfaces more closely connected with the steel back or steel shell. In itself, the white metal had no remarkable fatigue strength, but this did not apply so much when the layer was thin and when it was metallically bonded to the steel shell with a first-class tinning. Dovetails linked the white metal in position but that did not make the steel and the white metal a bimetallic composition. Proper tinning did the job.

There was one thing that could be done—and in his opinion should always be done—to improve the fatigue strength of the white metal itself. It should be allowed to solidify quickly in order to obtain a fine-grained structure. Experience, which had been confirmed by certain research experiments, had shown that white metal with a fine grain had a better fatigue strength and better anti-wear properties in respect of boundary lubrication conditions that would prevail during starting and during the boosting of the engine.

Mr. Pounder did not refer to the composition of the white metal and he himself would not go further than to say that in many cases the application of a thin white metal layer, properly bonded to the steel shell, and cast with a fine-grained structure, had allowed for the use of much cheaper white metal alloys than would otherwise have been used.

There was one point about oil grooving. In Fig. 33 Mr. Pounder showed the lubricating system used in connecting rod top end bearings. It was interesting that these horizontal oil grooves connected with both oil inlets and oil outlets. There was no doubt that this would give an adequate oil supply for these oscillating bearings, even if the lubrication of such bearings might very often be critical. He would like to take this opportunity to point out that Mr. Barwell of the University of Glasgow had recently developed a new type of oil grooving for such oscillating bearings. He applied horizontal grooves, as usual, at distances corresponding to the amplitude of the movement of the shaft (or bearing). But instead of connecting these grooves with one or more circumferential grooves and instead of using a system like that described by Mr. Pounder he connected these horizontal grooves with one helical groove

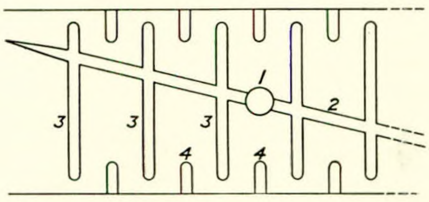


FIG. 69.—*Suggested oil grooving for oscillating bearings (Barwell)*

(1) Oil inlet; (2) Helical groove; (3) Horizontal grooves; (4) Blind grooves

which had a supply channel for oil (Fig. 69). Then he provided a new feature in that he added certain blind oil grooves extending from the edge and a little way between the horizontal grooves. It seemed—at least he said so—that these grooves, oil outlets they might be called, secured a better oil flow over the edge of the bearing from the feed grooves, the horizontal grooves, to the outlet grooves.

He himself did not know whether this lubricating system had been tried out on a larger scale, but he thought it worth mentioning on this occasion.

MR. N. MACLEOD (Member) said that not only the members present that evening but all engineers whose duties included the running and maintenance of these engines had good reason to thank the author for presenting this paper and including so much detailed description of the design. The points which he wished to make were only very minor ones based on experience of this and similar engines.

In Fig. 4, the section through scavenge ports, the relief valve for scavenge air reservoir was shown with an internal spring. On one ship, not turbocharged, which had a series of scavenge fires, it was found that these springs collapsed under heat, allowing valves to open and add to the general unpleasantness. A simple modification converted these valves to external spring and, though only one scavenge fire had occurred since, it was felt that this modification had eased the situation. In a turbocharged engine a scavenge fire, should it occur, could be expected to be more violent and the external spring relief valve would be preferred.

On page 183 the author mentioned that "the eccentric straps were cut from mild steel slabs. Alternatively, they might be steel castings". After one case of a cast steel eccentric strap breaking because of a casting defect—fortunately without causing any damage—straps made from mild steel slab would definitely be preferred. The lubricating oil system to turbochargers with working and standby pumps plus emergency gravity tank maintained full by a float valve and heated by an immersion heater seemed rather elaborate. Perhaps the author would say why this system was chosen rather than a simple gravity system in which a tank maintained overflowing by pumps feeding the bearings directly.

His next question had been partly answered already. He would like to ask if, after some time in service on heavy fuel oil, any build-up on turbine blades or nozzles had been encountered. One Continental builder had provided a nozzle for introducing saturated steam to the turbine to wash off heavy deposits which were experienced.

Towards the end of the paper the author mentioned that

"The turbocharged four-stroke engine was a *kindly* engine". During three years' service on the *Lochmonar* he had never heard the engine described as kindly; but if their, at present, limited experience of the turbocharged two-cycle engine bore out first impressions, he was sure that that description would apply.

Correspondence

MR. W. F. JACOBS (Member) stated that, some years ago, there was reason to believe that cylinders of about 24-in. bore ran without any trouble, that those of about 28-in. bore had some trouble, and those of, say, 32-in. bore suffered from a great deal of trouble, the mean effective pressure being the same in all cases. Apparently the cracking was, and continued to be, due to thermal stresses, because if the cylinders were subjected to similar varying pressures at normal room temperatures, by hydraulic pressure for instance, there would be no cracking.

Assuming that the heat released in a cylinder varied somewhat as the square of the bore, then the cooling area varied at a much lower rate, thus causing the inner wall of a large cylinder to be hotter than that in a small cylinder.

Also, in a large cylinder, the thickness of the wall must be greater than the wall thickness in a small cylinder, when using the same design and materials; that is, the heat flow must take longer and the difference in expansion between the inner surface of the cylinder wall and the outer surface must b larger the greater the diameter of the cylinder. This would cause cracking in a large cylinder when it would not cause cracking in a small cylinder.

It was possible that, with improvements in materials and design, cylinders could now be made larger in diameter than hitherto.

There was a point in connexion with the exhaust temperatures in Tables V and VI that he could not understand. In the final column of Table VI, dealing with the results when running the engine with an 86·5 per cent load, the following exhaust temperatures were given: —

Exhaust at cylinders, deg. F.	...	604
Exhaust before turbine, deg. F.	...	804
Exhaust after turbine, deg. F.	...	635

As far as he could see, the temperatures would be hottest at the cylinder (presumably just beyond the exhaust ports), then a little lower at the turbine entry and lowest at the final discharge from the turbine. From the figures given it would seem that the gases picked up heat after they left the cylinder and before they entered the turbine. Transposing the figures did not seem to give a satisfactory result and in any case the figure of 805 seemed high, to say the least. With modern fuel injection one assumed, of course, that all the fuel was burned in the cylinder.

MR. E. V. JONES (Member) remarked that there was one point in the paper which he failed to understand. He referred to Tables V and VI on page 192. Both these tables gave three sets of exhaust temperatures for four different engine loadings. In each set of figures the temperature of "Exhaust after turbine" was higher than the temperature of "Exhaust at cylinder" and there was a very considerable rise in the temperature between the "Exhaust at cylinder" and the "Exhaust before turbine", this latter difference being 200 deg. F. at 86·5 per cent of the full load conditions.

In the text of the paper the author stressed that the turbine should be placed as near as possible to the cylinder and from this he anticipated that there would be little or no difference in the temperatures of the cylinder exhaust and turbine inlet.

No doubt there was some simple explanation for this temperature rise, but as he could not see the reason he would be grateful if the author would satisfy his curiosity.

MR. W. E. MCCONNELL (Member) noticed that this very complete exposition of a famous and successful design was explained by its author as having been given in an endeavour to provide opportunities for young engineers for exercise in the elements of design, for experienced engineers as an indication of a level of practice, and for the shipowner or superintendent to make comparisons. Discussion in the usual form was not to be expected, for the paper described a design which had been in successful operation for several years.

The value of the paper lay chiefly in the very careful and detailed explanation given by the author and it was perhaps reasonable to suggest that the young engineer, whose interests had always claimed a large share of the author's attention, would reap most benefit from it.

Experienced engineers would recall the improvements in details which were developed after the design was first put into service and they would note that even the seemingly trivial items in their present form represented the approved result of trial and experiment in the patient pursuit of a reliable and efficient engine. Only those who had followed these experiments could realize what a difference in performance could be produced by a change in the tangential angle of the scavenge ports. One component alone, the complicated steel casting for the combined crank web and eccentric sheave, required much research before sound castings free from shrinkage defects due to the heavy changes of section could be ensured, and the solution of this problem was a commendable achievement.

It was noteworthy that the feature described by the author as before and above all others—low fuel consumption—was the most elusive when compared with the improvements made in steam engines, which could show a reduction of some 30 per cent in fuel consumption since the oil engine was first sent to sea. In discussions at the Institute about that time a figure of 0·4lb. of oil per b.h.p. per hour was quoted, even for the earlier Burmeister and Wain designs; the figure given for the newest design was 0·35lb., a gain of 12½ per cent. The history of the Scott Still engine showed how difficult it was to achieve any large improvement of this feature in an engine which was, to begin with, the most economical prime mover available.

With vivid recollections of the trouble from broken piston rods, which was finally eliminated by the device of the articulated guide shoe, it was noted that Fig. 35 showed a fixed guide shoe. It would be interesting to know the reason for reverting to the earlier design.

MR. K. MCKENZIE (Member) commented that, as always, Mr. Pounder had presented his latest paper with meticulous detail and in his usual masterly fashion. They were indeed conscious of his continual striving for improvements, not only in the Harland and Wolff engine, but for the benefit of the industry in general.

He would like to ask the author for his opinion upon the cooling of the upper pistons with water in lieu of the accepted method of oil. The Harland and Wolff opposed piston engine had always, at least in so far as marine units were concerned, been cooled with lubricating oil. It had been found in practice that the upper piston crowns had a tendency to burn, and he was of the opinion that this peculiarity very probably resulted from inefficient cooling due to the formation, internally, of carbon from the oil deposited around the crown. It had been found very necessary, after an operating period of nine months, to open up the exhaust pistons and remove carefully the hard carbon which had been deposited, and this in no small measure, he believed, had prevented burning of the piston crowns. Whilst it appeared to be a retrograde step, water cooling of the upper pistons only might prove advantageous.

On page 166 Mr. Pounder made reference to the cylinder liner construction illustrated in Fig. 3 and stated that as experience gathered the liner might be modified to a simpler form and an indication of his thoughts in this direction would indeed be interesting. It would be recalled that liner design in some of the earlier engines enabled a section of the lower cylinder to be lowered into the scavenge belt, so permitting the lower piston to be exposed and rings renewed without the necessity for withdrawing the piston. In tankers, the time available for piston overhauling was extremely scanty and any method whereby overhauling could be undertaken with the minimum of time and effort was of value. It would be of great interest to learn of the reason for the cessation of this practice. It must be appreciated that the designer was striving continuously to improve the performance and reliability of his particular engine, but ease of maintenance should also be given a great deal of thought.

The section of the paper dealing with fuel injection apparatus and the simplification which could be anticipated from the gas operated fuel pump would, no doubt, create a good deal of anticipation, and Harland and Wolff were to be commended upon their investigations, design and production of a pump of this nature which had the distinct advantage of being timed. As Mr. Pounder remarked, "jerk" pumps had performed in a highly satisfactory manner for some time but it might well be that the time was not so far distant when gas operated pumps would become the rule rather than the exception. Any reduction in weight, however small it might be, was a definite asset; furthermore, there must inevitably be some reduction in the cost of the engine, due to the elimination of chain drives, camshaft, cams and the jerk pump itself, which must be more expensive to produce than the relatively simpler gas pump. Cooling the gas pump with fresh water might present difficulties due to deposition of scale and it might well be that an independent system, utilizing a light oil as a cooling medium, would prove more advantageous.

Turbocharging the two-stroke engine had indeed opened up a new field, particularly in connexion with the large tankers now being built and eliminating the necessity to adopt turbine propulsion. Although, at the present time, 30 per cent to 35 per cent appeared to be the limit, it was refreshing to learn that Mr. Pounder was already thinking ahead in terms of 50 per cent to 60 per cent pressure charging and a step in this direction would most certainly extend the field of future installations.

It was to be hoped that the present method of approach to the manœuvring problem by the adoption of an electrically driven blower was retained, the alternative of converting the lower end of the cylinder into an under-piston compressor being undesirable from the viewpoint of cleanliness and purity of the air supplied in this manner. The former method was simple, maintenance was negligible and the electrical power absorbed of minor consequence during manœuvring.

The disclosure of broken white metal in eccentric straps which had been lifted for survey could be most disconcerting and extremely expensive, not only in the actual cost of remetalling but in loss of hire. The writer had experienced this particular defect but in all fairness would point out that the original metalling had been undertaken by an outside contractor. They had since been remetalled "Belfast fashion" and after having seen the method adopted, it was not anticipated that further trouble of this nature would develop.

Mr. Pounder stated very definitely that he would have no hesitation in building an engine having cylinder bores of 880 mm. and indeed why not if higher powers were to be attained on a single screw. Any attempt to reduce the number of cylinders was worth while, even if only with a view to maintenance and cost of renewals. Perhaps Mr. Pounder might care to enlarge on this particular point, as to whether serious design and preparations had as yet taken place.

Mr. F. J. van Asperen, Dipl.Ing.E.T.H. (Member), if asked to give his opinion on the paper, would certainly state that he very much liked the detailed way in which the author gave away (if he might use the expression) his secrets. He thought that none of the author's engine building friends had ever given so many interesting figures for constructional dimensions and load figures or had so thoroughly explained design details. For students and for various grades of engineers it certainly contained many valuable data for checking calculations and for comparison with other marine engines.

On reading the paper, however, the following points occurred to him. If the author's prescriptions for liner hardness called for 180-200 Brinell, he wondered if owners, consultants or classification inspectors would accept a figure of 160, even if all other circumstances were within acceptable limits. Personal costly experience made him feel rather doubtful in respect of certain clients.

Referring to Figs. 21 to 25, the amplitude of the free horizontal moment had an exactly equal value to that for the free vertical primary moment. Did this mean that the reciprocating masses did not develop a primary vertical moment, assuming that the vertical moment was the total of the moments initiated by the rotary and primary reciprocating masses? This would be in accordance with the given phase angle of 90 degrees as in the graphs for the four, five, six and eight-cylinder engines. For the seven-cylinder engine the phase angle, however, remained the crank angle of 51 deg. 43 min., which was not quite clear in view of the above assumption.

Could the author, in addition to the data for the free moments, also indicate values for the internal moments? It was assumed that the explanation of this part of the problem could be found in the separate balancing of the crankshaft halves, as described on page 172 (Fig. 19).

As to Mr. Pounder's preference for separate piston cooling and bearing lubricating systems (p. 181), it was difficult to see the exact reason for this. His experience was that difficulties from contamination of lubricating oil occurred only in cases where there were insufficient quantities of circulating cooling oil or when only splash piston crown cooling was applied. In Mr. Pounder's case, with the application of well-guided circular oil flows in both pistons these difficulties should not arise, and extra costs for a duplicate oil system could certainly be avoided.

The author's remarks on page 188 on operational matters had his full support, especially as concerned the prolonged low power running. The solution for supplying air by the auxiliary fan to the blower suctions seemed to him a very simple and straightforward method for security at sea, if needed, and he thought Harland and Wolff and D. Napier and Son were to be congratulated as to first cost and overall efficiency on the adoption of this system. Had this solution ever been compared with that using the bottom parts of the lower piston with non-return valves between the common air receiver and the separated under-piston spaces, like Sulzer Brothers did?

A very important feature was that the gases could be allowed to flow through the blocked turbine during longer periods in the case of the gas pulse system. For the equal pressure system this could also be stated to be true, but of course flow conditions in the latter case could be assumed to be less severe on the turbine rotor blades.

Mr. F. J. Welch (Member) remarked that although Mr. Pounder's excellent paper on the latest Belfast engine did not invite comment on the unpleasant subject of crankcase explosions, and indeed his brief reference at the top of page 186 might even suggest that nothing more need be said about it, he would like to comment on an aspect of the matter which had been prompted by an examination of the section of the engine in Fig. 1.

The figure showed that ample relief area had been provided for the crankcase and the engine was well protected against explosion damage. The figure also showed that deflexion hoods were fitted to ensure that any flame issuing from the relief valve would not envelop any of the watchkeeping engineers who might be on the bottom platform. Suppose, however, that some slight but unusual noise, smell or wisp of smoke was sensed by the watchkeeper and his investigation led him to the middle gratings, he might be exposed to the flame of an explosion of which the clue he was following was in fact the warning.

A study of the papers* read early in 1956 by Freeston, Roberts, Thomas and Mansfield suggested that the insertion of a gauze flame trap under the flange of the relief valve where it was bolted on to the crankcase doors would be a very simple and effective precaution. The various tests which had been made on gauze flame traps had, he believed, all been made under conditions which were not those of a normally running engine in which a hot spot had developed and possibly quite rightly the conditions had thereby been more onerous than any which could occur in practice. Moreover, there was a danger that too many thicknesses coupled with too small a mesh might prevent the relief valve fulfilling its function with a large and therefore relatively weak door. His own choice would be for one 10-mesh gauze with three gauzes between 20 and 30 mesh for an engine of the size described by Mr. Pounder.

Further, one of the papers referred to made it clear that an oil wetted gauze was a much better flame retardant than a dry one and since any gauze on the inside of the relief valve would be expected to remain thoroughly oil wetted in an engine of the type shown in Fig. 1, the maximum benefit from the gauzes suggested would be expected.

MR. G. YELLOWLEY (Member) was interested in the great detail Mr. Pounder gave of his three-piece cylinder liner as the firm with which he was connected had developed a similar type of liner for another type of opposed piston engine. In this case the centre portion was a casting in spheroidal graphite cast iron and the upper and lower cylinders were of alloy cast iron. The parts were joined with spigoted, bolted flanges 55 mm. thick, using twenty-four 1-in. stainless steel studs and nuts pitched at about 105 mm., the ground joint surface being on the inside of the pitch circle only; this method of attaching liner to cylinder head had withstood the test of service over many years on four-stroke engines, and would appear to have some advantages in simplicity over the construction described in the paper.

Mr. Pounder referred to the position and number of the oil admission points on each liner, i.e. four top and four bottom, and as a recent paper before the Institute stressed the value of good distribution of cylinder lubrication, especially when burning residual fuel, the writer would ask if any oil grooves were cut in the liner when only four points were provided in these large bore cylinders.

The writer had always been grateful for Mr. Pounder's publications on the balancing of engines and the large amount of information now given was a valuable addition to previous work. There was, however, just one point the writer would ask and that was the magnitude of the vertical forces in the line of each cylinder, as although the primary and secondary forces were balanced for the engine as a whole, the out-of-balance forces per line must be transmitted through the engine frame to achieve this complete balance.

Mr. Pounder expressed himself very forcibly regarding the ratio of connecting rod length to crank radius as a result of experiences many years ago with shorter rods. A brief account of those experiences would be helpful to enable one to judge fairly the merit of the 4 : 1 rod as the mechanics of the problem did not lead one to anticipate any dire results from the adoption of a slightly shorter rod and in fact there were many modern engines in service with a ratio less than 4 : 1 running very successfully and it was the writer's opinion that there were other factors such as alignment, rigidity, etc., which were much more important than a slight divergence from the 4 : 1 ratio.

The section dealing with the subject of engine ratings was a very valuable contribution to this sometimes vexed problem, but it should be realized that the three levels of rating given in the tables—(a) normal continuous sea service; (b) maximum continuous service; (c) trial trip—were not related according to propeller law and would therefore require three different propellers to achieve the (a) (b) and (c) ratings in any one vessel. Perhaps the author would state which rating was generally chosen for the propeller design.

* Freeston, H. G., Roberts, J. D., Thomas, A., and Mansfield, W. P. April 1956. "Crankcase Explosions: Protective Devices". Proc.I.Mech.E., Vol. 170, No. 24.

Author's Reply

Mr. Arnold referred to the proportions of the scavenge air manifold.

The original design of manifold, as incorporated in the earliest non-P.I. engines, was of the so-called shallow type. For the 750/2,000 engine this manifold had a vertical depth of 1,180 mm. On one side of the manifold there was a horizontal semi-circular duct, introduced to augment the manifold volume. The cylinder liner was a simple, continuous casting. The design was what the author regarded as an appropriate one for the engine, both as regards dimensions and form.

But, as Mr. Arnold would know, an engine builder had perforce to be all things to all men, so far as it was possible. Accordingly, when certain powerful interests requested the design to be modified in such a way that piston rings could be inspected and renewed in place, and main pistons disconnected and withdrawn sideways inside the scavenge space, the cylinder liners were arranged to have bolted-on bottom skirts. These skirts were lowered *in situ,* the piston rings were exposed and, if necessary, the pistons were withdrawn sideways on skids introduced into the scavenge spaces. To make this general construction workable, the scavenge manifold was increased in depth from 1,180 mm. to 1,900 mm. The increase in manifold volume rendered superfluous the semi-circular duct and the deep scavenge manifold therefore became a rectangular structure.

The purpose behind this change was to provide certain clients with the type of construction that had been standard for the firm's single-acting four-stroke under-piston pressure charged engines.

For a period the two designs, namely the shallow or original design and the deep or modified design, were contemporaneously in use. Later, when it had become generally understood how relatively simple an operation it was to lift the main piston upward through the cylinder liner, the popularity of the loose skirt decreased. With the introduction of pressure charging, advantage was taken of the incidental changes to discard the deep manifold completely and revert to the original conception of a shallow manifold.

It was not, therefore, so much a matter of seeking a reduction of height and weight as it was a question of avoiding accretions and complications. The difference in weight was approximately one ton in favour of the deep manifold, because of its simpler construction. There was no difference in cost. The slightly higher centre of gravity of the deep manifold was counterbalanced by its slightly lower weight. Accordingly, the effect on the whole engine of a change in manifolds was completely negligible.

In the author's opinion, Mr. Arnold showed wisdom in insisting upon the shallow type of manifold for his engines. Nevertheless, as he would no doubt notice, one of the contributors to the present discussion, a superintendent engineer of sound experience and judgement, and withal a strong personality, was still in favour of the deep manifold.

Mr. Arnold asked if proposals had ever been made to couple the engine to a variable pitch propeller. The reply was that, for a variable pitch propeller, the only question which could arise would be one of arranging suitable interlocking controls. The author assumed, however, that a reversible propeller was implied by the question. If this were so, the answer was that no serious suggestion of this kind had ever been made. With a reversible propeller the engine would become uni-directional, and the angle of advance of the eccentrics could be increased to, say, 15 degrees. This by itself would be an advantage. But it would be outweighed by the general difficulties of the proposition.

Regarding fuel injection systems: the author was of opinion that Mr. Arnold might possibly have misunderstood the timing shaft detail shown in Fig. 8. There was no camshaft. The text explicitly stated that the fuel pump camshaft, with bearings and chain drive, was eliminated. Timing was obtained, as shown in Fig. 9, by the simple expedient of fitting a small cam to the lubricator drive shaft. In other words, timing and all its advantages were obtained without any kind of design complications. The lubricator driving shafts were of themselves a necessity, irrespective of types of fuel injection gear. All that was done, therefore, was to mount small cams upon the shafts.

Mr. Arnold expressed regret at the retention of flat seats for the fuel valves. The *pros* and *cons* of flat-seated valves *versus* mitre-seated valves would involve too long a story for citation here. So far as the author was concerned, there was no bias either way. His business was to provide the several types favoured by the clients of his company, so far as this was reasonably practicable.

Mr. Arnold referred to three ships fitted with the engine builders' standard fuel injection equipment and stated that fires occurred in the scavenge manifolds; the equipment was later replaced by gas-operated pumps and pilot injection fuel valves; and subsequent to, and consequent upon, the change there were no scavenge manifold fires. The author was of opinion that there might have been alternative explanations.

The author was naturally gratified to see the particulars which were tabulated by Mr. Arnold regarding the machinery performance in m.v. *Dolius* and to note his very favourable comments.

In his last paragraph Mr. Arnold returned again to the subject of camshafts. In the paper, at the end of the first paragraph dealing with fuel injection apparatus, it was stated: "The substitution of gas compression fuel pumps for cam-operated pumps results in some simplification. The fuel pump camshaft, with bearings and chain drive, is eliminated; but there is no saving in engine length by the discarding of the fuel pump chain drive". If this were read in conjunction with what was later mentioned in the section dealing with exhaust turbocharger arrangements, a complete picture would be obtained. There it was stated: "The elimination of chain driven scavenge blowers enables the 750/2,000 engine to be shortened by 550 mm. and the 620/1,870 engine by 350 mm. The saving is effected between the two middle frames of the engine, where the fuel surcharging pump and the manœuvring gear controls are arranged. There is no saving of engine length by the substitution of gas compression fuel pumps for chain driven jerk pumps".

The centre distance at the middle of the engine was determined by the shaft coupling flanges and the coupling bolts. With a camshaft drive, the chain wheel was located on the coupling; hence, with or without a camshaft drive, the centres were the same.

The particulars regarding crankshafts given by Mr. Arnold in Fig. 63 were the same as those given in the course of the paper, because ultimately they were derived from the same working drawings. Fig. 63 showed clearly the amounts and the location of the dimensional savings.

To the author, it was a matter for real regret that Mr. Arnold had now retired. Men of his enthusiasm, vitality and courage were all too few in number.

Professor Burstall was invariably so much *en rapport* with marine engineers that the author was always glad to have an opportunity to co-operate with him. The last time he was on the platform with the Professor was at the delivering of the second Stephenson Lecture before undergraduates and others at King's College, Newcastle upon Tyne, the subject being certain aspects of marine engineering.

Professor Burstall asked for the reasons which led to the choice of about 30 per cent of supercharge. The answer was simple.

As happened so often in progressive marine engineering, a designer became torn between what he would like to do and what prudence dictated that he ought to do. In seeking acceptance of the principle of pressure charging a two-stroke engine, it was more prudent to offer shipowners something that was well worth while but likely to be trouble free, than something which might be even better worth while but which might carry with it an element of uncertainty. If an engine builder could offer a shipowner something lighter or something more compact, something more economical to run and easier to maintain than what he had had before, at the same price as hitherto, he was pleased. But if there were growing pains which were other than insignificant, his pleasure was apt to deteriorate into querulousness. If shipowners could be assembled in a line and graded, at one end there would be those to whom even the slightest change would imply experimentation. At the other end there would be a few who would dislike being offered the same installation twice in succession; that is, they would expect a significant improvement each time they were in the market for new tonnage. Between these two extremes was poised the engine builder, endeavouring to be anathema to as few potential and actual clients as possible. And so, in endeavouring to establish a pressure charged, opposed piston, two-stroke engine, it seemed to the author that about 30 per cent increase in power on the same engine weight was a sufficiently attractive advance in one step to be likely to find acceptance. While he would have preferred 50 to 60 per cent supercharge this would have involved a substantial amount of redesigning and the present *tempo* of life was such that emphasis must necessarily be brought to bear on early establishment at reasonable level.

Regarding the observations at the end of the paper, to which Professor Burstall referred: two levels of practice were implied. Thus, an output of 20,000 b.h.p. could be obtained on eight cylinders of the dimensions indicated. This was on the basis of the present level of supercharging, i.e., about 30 per cent. But it was also stated that, with a higher level of pressure charging, the output could be raised to the order of 24,000 b.h.p. without hazard. Upon review it would seem to be more practicable and practical to design for, say, 50 to 60 per cent supercharge. If it were known with certainty that a unit of 20,000 b.h.p. would fulfil all requirements, the cylinder dimensions could be reduced for this higher supercharge rate.

To the author the *experimentum crucis* was the maximum power per screw which was likely to be required during, say, the next ten years, because the design and construction of a very large engine was not lightly to be undertaken.

Nobody in the course of the discussion had referred to a single-screw geared drive, and the reasons were fairly clear. An aggregation of four engines running on heavy oil, transmitting a total power of, say, 24,000 b.h.p. through electromagnetic or hydraulic couplings and gearing, would, in the author's opinion, present a greater number of problems than would a simple straightforward direct-coupled engine. Neither technically nor commercially was such an arrangement likely to be attractive. The alternative of a Diesel-electric drive could be technically more satisfactory than a geared drive, but hardly any more attractive commercially.

Professor Burstall asked why the cylinder liner wear of pressure charged engines had often been less than that of normally aspirated engines.

For four-stroke single-acting propelling engines there could be no doubt about this phenomenon, because, after pressure charging had been established for some years, it became a matter of general experience that liners were lasting longer. Observations seemed to confirm that the chief factor was the higher temperature of the intake air, which was thus removed further from the region of the dewpoint. At this early period Harland and Wolff engines were running on blast injection and the blast air compressors sometimes showed an unreasonable amount of cylinder wear. Notably was this so on certain large Diesel engine driven blast air compressors in two important passenger liners. After investigation it was decided that the water jacketing of the three-stage compressor cylinders was too effective; instead, therefore, of the cylinders being water cooled they were water heated, by rearranging the water jacket piping. The improvement in compressor cylinder life was remarkable. As the result of these experiences the author assumed that a similar principle operated with pressure charged Diesel engine liners.

Referring to the comments of Mr. P. Pluys, it had been a pleasure to the author to think that the chief technical men in his company, the Compagnie Maritime Belge, should turn periodically to Belfast for an opinion on important matters relating to high powered marine Diesel engines. Especially in this connexion did the author recall many excellent meetings with his chief, Mr. G. Dufour, technical director.

The cylinder liner design described in the course of the paper had been devised for turbocharged engines. It could, of course, equally well be fitted to non-P.I. engines if so desired; but there was nothing in the author's experience to indicate that this more expensive construction was necessary for such engines.

Since the inception of the pressure charged engine, and indeed with non-P.I. engines also, it had not accorded with the author's outlook to insist upon oil cooling for exhaust pistons. A considerable number of 530-mm. bore trunk-type opposed piston engines for land installations in various parts of the world had been water cooled at the request of the respective clients. But by far the greater number of exhaust pistons hitherto made had been oil cooled. Where fuel sprays, in themselves correct, had been correctly set in the combustion chamber, and where carbon deposits inside the exhaust piston crowns had been periodically removed, oil cooling had been very satisfactory.

With approved types of packing, and with the cooling system arranged as described in the paper, water cooling had also been fully satisfactory. But, as Mr. Pluys pointed out, with water cooling an appreciable increase in fuel consumption could be expected, his quoted figures of 3 to 4 per cent being average values, unless special precautions were taken.

Lately there had been a tendency, even on marine engines, for clients to ask for water cooling of the exhaust pistons. The reasons for the request could not usually have been experiential, because the superintendents chiefly concerned had not hitherto had Harland and Wolff single-acting opposed piston engines. The request, of course, could be a carryover from experiences elsewhere.

The piston materials and forms of construction detailed in the paper had been standard for a long time.

Mr. Pluys was correct in stating that, on the basis of his experience, the author was averse to the use of ball bearings. This aversion was not confined to turboblowers; it extended to all parts of the engine. An essential objection to their use was the "brinelling" of the ball races. If, in service, the ball bearings were renewed at proper intervals, the chief objection would be removed. In engine rooms fully free from vibration there could be no objection to ball bearings. But there were no such engine rooms, at least not on board ship.

Mr. Pluys reviewed very clearly the respective matters which constituted his remaining remarks. Comment upon them would therefore be superfluous.

Mr. Jackson quoted certain numerical figures regarding savings in weight, length, etc., which appeared to him not to be in alignment.

The author was of opinion that Mr. Jackson might possibly have misunderstood the numerical figures in their respective contexts, because actually they were completely consistent; in fact they were all taken from working drawings and from accurate returned weights. The author was not very clear what he could offer as further explanation. The 20 tons quoted was the difference in weight between two six-cylinder engines; one was a P.I. engine having turboblowers and the other was a non-P.I. engine provided with engine driven blowers, the engines otherwise being the same. The 27 per cent referred to two different engines of the same cylinder size and type, one P.I. and the other non-P.I., both developing the same power. Stated another way: there were two equi-cylinder engines of the same overall dimensions. One engine, pressure charged by turboblowers but stripped of engine driven blowers, weighed 503 tons; the other engine, non-P.I., with Roots' type scavenge blowers mounted on the back of the engine, weighed 523 tons. That is: the weight difference in favour of the P.I. engine was 20 tons, and the power 30 to 35 per cent in favour. Again, if two engines were chosen, of the same cylinder dimensions, the number of cylinders being respectively arranged so that the power output was the same in both engines, then the P.I. engine showed a saving in weight of 27 per cent.

The author agreed that the specific fuel consumption of a pressure charged engine at sea tended to show an improvement of up to 8 per cent and even 10 per cent. But, as the paper was almost exclusively directed to what took place in the builders' works, he had confined attention to the stated figure of 4 per cent saving on the test bed. Various partial explanations to account for the differences had occurred to the author, as no doubt they had to Mr. Jackson also, but at present he was not satisfied. Accordingly, a definitive explanation must be deferred for the present.

Regarding the construction of the cylinder liner shown in Fig. 3, the change was dictated rather by caution than by experience. It was correct, of course, that the liner joints were located inside the water space and, if a joint were to become defective, water *could* enter the cylinder. But this had been true of numerous designs of very successful liners, with a history going back over many years. Thus, the original type of liner incorporated in the single-acting four-stroke crosshead engine had its joint inside the water space. Although the author might dislike, and indeed did dislike, such joints, because the construction fell short of the ideal, this relatively simple design—which, to date, had not given difficulties of the kind which Mr. Jackson might have in mind—was preferable to resorting to heavier, more complicated constructions simply to avoid the internal joints. The cooling water was undistilled fresh water, a completely innocuous medium.

The fuel pump dimensions for the engine in the paper to which Mr. Jackson referred were too generous for non-P.I. The same dimensions were used for P.I. cylinders of 1,750 b.h.p. It was Belfast practice to reduce the number of pump sizes to the minimum. The gas pump dimensions quoted were suitable for 620-mm. P.I., 750-mm. non-P.I., 750 mm. with 20 per cent P.I.; for 750-mm. P.I. the bore and stroke were 45 mm.

The firing order for the four-cylinder engine was a compromise between the conflicting demands of good balancing and a suitable scheme for the turboblowers. Regarding the balancing of a seven-cylinder engine, it was the practice to balance each half of an engine, if this was at all practicable. The balance weights were placed to suit the unbalanced couple in each half of the crankshaft and, if the balance weights were not diametrically opposed, their sizes were adjusted to balance the forces. The unbalanced couples which were indicated in Fig. 24 were the couples which would be present in any seven-cylinder engine which had a firing order of 1.7.2.5.4.3.6.

Mr. Jackson stated that he was not quite sure that the Napier system was the simplest of the alternative arrangements which were possible. The author agreed that the attachment of a self-acting electric motor to the turboblower appeared attractive. The objections that had been directed against the arrangement, however, emanated from the electrical engineering experts of the shipping company which had initially favoured the scheme. An automatic mechanically operated clutch was a workable alternative to the air operated clutch, but the objections to the rather involved electrical gear contained in the motor drive still remained. There was, additionally, the objection that the availability of the turboblower was dependent upon the availability of the electric motor and its gear, the automatic clutch, and so on. It seemed to the author that if one apparatus were intended to serve as standby to another apparatus it was only prudent to make the first named apparatus independent of that for which it was a standby; all of which implied an independent blower of some kind.

Mr. Jackson stated the relationship between compression pressure and maximum pressure correctly. Actually in the two diagrams incorporated in Fig. 58 it had been desirable, for reasons of clarity, very slightly to shift the full-line compression curve, which had led to the effect mentioned by Mr. Jackson.

Mr. Knud Moller's long and valuable statement, descriptive of his company's practice, contained numerous useful references to the very successful turbocharged poppet valve engines which were the mainstay of his company's marine business. As the statement was self-contained and non-controversial, there was little or nothing for the author to comment upon. He had been interested in the remarks of Mr. Knud Moller regarding the construction of a turbocharged poppet valve engine to develop about 20,000 b.h.p. on twelve cylinders. This was indicative of the current, but independent, trends of thought that were beginning to take hold of engine designers, influencing them towards larger propulsion units. The tendency required the restraint of prudence and wisdom, for, in the world of shipping, fashions could grow and seize hold of men with the tenacity of a contagion. Mr. Moller would be familiar with the story of the lemmings!

Mr. Ewart's remarks constituted a well-reasoned statement against relying upon single screws for the propulsion of large and costly tankers.

In the author's opinion it had been wise for shipowners, in the post-war years, to have fostered the retention of single screws for cargo vessels and tankers requiring up to, say, 15,000 s.h.p. in service. But the retention of single screws for powers of say 25,000 s.h.p. was quite another matter. No passenger liner requiring, say, 20,000 to 25,000 s.h.p. for its propulsion would be offered to a shipowner as a single-screw vessel. It was noteworthy also that the large refrigerated cargo liners of well-established companies renowned for their acumen, which required less than 20,000 s.h.p. for their service schedules, were twin-screw ships. There might be a debatable range between, say, 15,000 s.h.p. and approaching 20,000 s.h.p. when sometimes the overall requirements might point to single screws and sometimes to twin screws. But, for 20,000 s.h.p. and above, the overall balance of commercial advantages was with twin screws, or so it seemed to the author.

With the range of turbocharged single-acting engines at present available, based on existing cylinder dimensions, an aggregate output of 30,000 s.h.p. on twin screws could be obtained without difficulty.

The author, as an engineer engaged in the design of high powered Diesel engines, would welcome with acclamation the chance of building an eight-cylinder unit of 20,000 to 25,000 s.h.p. But if he were a shipowner, faced with the running of vessels requiring 25,000 s.h.p. for their propulsion, he would certainly decide upon twin screws, irrespective of the type and nature of prime mover installed.

With reference to the remarks of Rear-Admiral Cowland, the success of any exhaust turbocharged engine, in the ultimate, depended upon the knowledge, skill, experience and co-operation brought to bear upon the scheme by the two parties to the venture, namely, the engine builders and the turbocharger builders. Success for one, without equal success for the other, meant ultimate success for neither. The excellent results achieved, to date, by the engines with which the paper was concerned were due in no small measure to the enthusiastic efforts of the leading technical men in the firm of D. Napier and Son, Ltd. Well in the forefront of the Napier band stood Admiral Cowland.

The Admiral, from his wide and long experience, was clearly exercised in his mind as to the doubtful wisdom of propelling by single-screw machinery the larger and more powerful tankers now projected and likely to be built. It was therefore interesting to observe his suggestion that, where a power of 20,000 s.h.p. was needed for the propulsion of a tanker, twin-screw machinery sets of 10,000 s.h.p. each were to be preferred to a single-screw set of 20,000 s.h.p.

Admiral Cowland's comments upon ball bearings versus sleeve bearings for turboblowers were valuable. The larger Napier blowers had sleeve bearings—no doubt out of deference to the expressed requirements of shipowners and their representatives—but for smaller blowers, where these inhibiting influences were not a factor, ball bearings seemed to be standard.

The other remarks which constituted Admiral Cowland's contribution to the discussion were self-explanatory; comment from the author would be gratuitous.

Mr. Nils Gram, of Messrs. Paul Bergsöe and Son, the well-known suppliers of white metal and other alloys, commented upon white metal thicknesses. Mr. Gram's contribution was notable for the many useful remarks which he made on the technique of white metal application.

During the war, and in the immediate post-war years, white metal thicknesses were restricted by Government edict to 3 mm. for the largest bearings used in marine Diesel engines. In consequence a number of problems arose, which space did not permit of detailing. The bonding of the white metal was sometimes particularly difficult. In pre-war years it had always been possible to obtain commercial steels of the requisite degree of purity to ensure satisfactory bonding, other things being equal. But present day British Standards allowed percentages of impurities in excess of the acceptable range, a fact which did not tend to diminish bonding problems.

The author used to find, e.g., in large guide shoe surfaces, that to obtain well bonded white metal 3 mm. thick it was often necessary to cast-on the metal at twice this thickness and then machine away the surplus; otherwise the quality of the adhesion could be unpredictable. The author's repeated references to thicknesses of white metal, as noticed by Mr. Gram, were to some extent provocative.

It was perhaps regrettable that the activities of the firm of Bergsöe were confined to supplying white metal and offering incidental advice and service to purchasers. If their activities had included the lining of bearings with white metal, the results would have been interesting and valuable. The author was familiar with the work of Dr. Barwell and with the experiments to which Mr. Gram referred.

Mr. Macleod referred to the design of relief valves for scavenge air reservoirs, shown in Fig. 4 of the paper. He was correct in preferring an external spring to an internal spring and there was no reason why his proposal should not be embodied in standard practice.

Mr. Macleod mentioned the failure of an eccentric strap steel casting, because of an inherent foundry defect. This occurrence had not been known when the author itemized his experiences in the last section of the paper. It was only to be expected that a shipowner, having had a failure of this kind, should desire forged or rolled steel slabs for future straps. But it would be imprudent to interpret this request as a rigid requirement for all time. If slabs and castings could be obtained from steel works and foundries with equal readiness, then the author would agree that steel slabs should be always used. But if, as so often happened, suitable delivery times for slabs were hard to extract from the steel makers, then it would be a pity if the use of steel castings were precluded.

What was required was that greater skill should be shown in the making of steel castings; increased care should be exercised in heat treatment and X-ray examination; and greater accuracy applied in the machining and inspection of the castings, thereby ensuring that no stress raisers remained. In the occurrence to which Mr. Macleod referred it was the upper, i.e., the non-loaded, half-strap which failed. Subsequent investigation revealed the presence of a heavy stress-raising defect near the nut facing, it being here that a crack originated, to creep diagonally across the strap.

The author agreed that the turbocharger lubricating oil system was perhaps too elaborate. Mr. Macleod's suggested arrangement should be adequate for the purpose. Regarding the building-up of deposits on turbine blades and nozzles, the insignificant amount of such deposits, at least to date, had been one of the remarkable things about the turbocharging of engines running on heavy fuel oil.

Mr. Macleod's comment upon his three years' service in *Lochmonar* was interesting. As mentioned in the text, the propelling unit had been the first Belfast four-stroke engine to be pressure charged. The engine had not been built for pressure charging. As the author well remembered, the supercharging of this engine was in the nature of an experimental effort. Writing from memory, the author was of opinion that *Lochmonar* had been the only pressure charged four-stroke engine in the fleet of Mr. Macleod's company. The appellation "a kindly engine" had certainly been true after the system had become established, according to reports from shipowners over many years.

What Mr. Jacobs wrote regarding cylinder liners was qualitatively sound; cylinder cracking, when it occurred, was caused by thermal stresses. It would be seen, from a scrutiny of the dimensions given in the text for the constituent parts of cylinder liners, that the distance pieces at the combustion spaces, being of cast steel, could be made relatively thin, with benefit to the heat transmission rate. By an application of the principle of similitude, the results mentioned by Mr. Jacobs could be expected. But, in the practical design of engines, principles of similitude had often to be set aside and each design treated on its merits. This was simply another way of saying that, even in these days, processes of trial and error had a not inconspicuous place.

The author's essential life work had necessarily lain with engine types designed and built in Belfast. Accordingly he was not, by any means, always familiar with the problems associated with other makes of engine. This would explain his inability to comment satisfactorily upon the opening sentence of Mr. Jacobs's contribution, as this statement did not align with his own experience of the many varied engine types and sizes which had been built by his firm.

Mr. Jacobs traced the passage of the exhaust gas from the cylinder to, and through, the exhaust turbine and the author noted his comments. Perhaps the simplest analogy was a heavily flowing sea tide. The waves surged forward until they struck the sea wall. Suddenly halted, their kinetic energy became transformed into pressure energy, shooting upward a tall column of spray. The spray descended; the potential energy, now somewhat diminished by mechanical and aerial friction, was retransformed into a lesser amount of kinetic energy and away flowed the outward return wave. So it was, very roughly, in the exhaust pipe. The kinetic energy of the hot gas leaving the cylinder was converted, in part, into additional heat energy as it adiabatically compressed the column of gas ahead of it until, at the turbine inlet, the temperature exceeded that at the cylinder branch. At the turbine, some of the heat energy was transformed into horse power, lowering the gas temperature somewhat, and the gas flowed into a boiler. The turbine exhaust temperature could accordingly be higher than that at the cylinder branch.

Mr. Jones also referred to this increase in temperature of the exhaust gas between the exhaust branch of the engine cylinder and the inlet branch of the exhaust turbine, and asked for an explanation of the apparent anomaly.

The explanation, as already indicated, was that the kinetic energy of the exhaust gas from the cylinder caused adiabatic compression of the column of gas *en route* for the turbine, thus raising its temperature. This explanation might not, perhaps, always be the whole truth. Some experts maintained that the temperature recorded by the thermometer at the cylinder was a time average figure; also there was a cooling exhaust pipe water jacket. Moreover, during the exhaust/scavenging stroke, the last thing to emerge from the exhaust ports was a slug of cold air which could hang around the thermometer stem for about two-thirds of a revolution, thus influencing the reading towards a lower temperature. Whatever influence these other factors might exert, adiabatic compression could be accepted as the essential cause of temperature difference.

Mr. McConnell, who until his retirement was a Vice-President of the Institute, had daily opportunity in his work for the Ministry of Transport to see with the eyes of an interested outsider the evolution of Belfast engine types over a quarter of a century. His brief survey indicated a few of the important points as he saw them.

Mr. McConnell referred to failures of screwed ends of piston rods in double acting two-stroke engines. At one phase the superintendent engineer who was chiefly involved in this problem suggested that there should be a hinge or articulated joint between the guide shoe and the crosshead, to permit of self-alignment. For a long time there was reason to think that this device would prove the solution to the problem of failures. But it did not.

The piston rods of the engines to which Mr. McConnell referred, being double-acters, were exposed to alternating stresses. In the single-acting engines with which the present paper was concerned the piston load was directed continuously downwards and therefore, strictly speaking, there were no working stresses on the screwed end in ordinary circumstances.

To the author, the piston rod problem mentioned by Mr. McConnell stood out as an example of something that was perfectly sound on paper but utterly unsound in practice.

Mr. McKenzie's comments confirmed that, if the exhaust pistons were periodically cleared of the hard carbon which became deposited on the inner surfaces, burning of the piston crowns was prevented. He vouchsafed the opinion that, while it might appear to be a retrograde step, water cooling of the upper pistons might prove to be advantageous.

The author agreed with Mr. McKenzie. In his opinion, water cooling of the exhaust pistons was not to be regarded as a retrograde step. In fact, as mentioned in the text, designs were equally available for oil cooling and water cooling, but, other things being equal, oil cooling was to be preferred. If, however, for one reason or another, the exigencies associated with the running and maintenance of machinery made it necessary to use every available artifice to reduce time and labour, then the increased fuel consumption which was a concomitant of the water cooling of exhaust pistons should be accepted. With suitably designed exhaust pistons and the established arrangement of telescopic tubes, there were no physical disadvantages associated with water cooling.

Mr. McKenzie referred to the earlier practice of providing bolted-on cylinder liner skirts which could be lowered into the scavenge spaces, so permitting the main pistons to be exposed and the piston rings to be renewed without the necessity for withdrawing the pistons. In this connexion, and to prevent repetition, the author drew attention to his reply to the contribution of Mr. Arnold. Anything that achieved speed and simplification of engine maintenance necessarily had the author's unqualified support. In many cargo vessels the length of stay in port, as determined by cargo handling, exceeded the time necessary for engine maintenance purposes. But in a tanker fleet the stringent requirements of the service placed a stronger emphasis upon the need for rapid overhaul.

The author was most glad to have had the spontaneous co-operation of Mr. McKenzie in the experimental application of the gas-operated fuel pump to one of his propelling engines. His experience had shown the necessity for reconsideration of the present arrangements for the fresh water cooling of the gas cylinder.

Not the least refreshing of Mr. McKenzie's remarks were his comments upon the manœuvring of turbocharged engines. It was clear that he was an advocate of the separate electrically-driven blower and was not enamoured of any proposal to convert the lower end of the cylinder into an under-piston compressor.

Mr. van Asperen wrote about cylinder liners. The figure for liner hardness which the author quoted, namely 160, he expected one or more contributors to comment upon. The figure was given as indicating a level which, other things being equal, could be just within the range of acceptability, according to the author's metallurgical advisers. Whether the author himself would be prepared to accept such a figure was more than doubtful, irrespective of the circumstances.

It was normally correct to assume that each half of the crankshaft was separately balanced, as indicated by the relative magnitudes and positions of the balance weights in Figs. 18, 19 and 20. The engines were balanced for total revolving plus one-half of the reciprocating moments, thus introducing the same order of unbalanced couples in both the horizontal and the vertical planes.

It was not by any means every client that agreed with the principle of separating the piston cooling and engine lubricating systems. If a ship's engineers were prepared to make a study of the common cooling and lubricating oil system and painstakingly to regulate the respective valves so that the pistons received their full proportion of cooling oil, then there was no need for separate systems. The double system was necessarily more costly and did not always achieve the end for which it was introduced. But, against this, some of the most experienced shipowners preferred it and were prepared to pay for it. Where the exhaust pistons were water cooled the need for separate systems disappeared.

Without naming competitive engines, the author's present opinion, after repeated and independent surveys of alternative possibilities, remained in favour of the principles which were embodied in the Napier system.

Mr. Welch's statement on crankcase safety precautions was something for which the author was grateful.

It must be twenty years or more since Mr. Welch and the author first collaborated on problems of crankcase safety. The incident that brought them together originally was an explosion in a double-acting engine in which a camshaft thrust bearing at the top of the engine had seized, causing pellets of molten steel to fall the full height of the engine enclosure, thus initiating an explosion low down in the crankcase.

The present position, as the author saw it, could very simply be stated. He had no fear for the engines themselves. Crankcase doors were stoutly constructed and heavily secured; relief doors were large in size and generous in number; all crankcases were sealed and provided with an exhaust fan system which ensured a slight sub-atmospheric pressure; and so on. But while he was confident about the integrity of the engine, he could not speak with the same assurance regarding the safety of engineers if an explosion should occur with men standing in the path of released flame and scorching gases.

There could not be many contrivances available which the author had not investigated. But, hitherto, for one reason or another, none of these had provided the sought-for certain safeguard. And so one turned again to the simplest and most direct of the likely solutions, namely, a gauze flame trap attached to the self-acting relief door. This, it should be noted, was also the conclusion reached, in principle, by Mr. Welch. In fact, the only difference between the proposal described by Mr. Welch and that offered by the author was the geometrical shape of the gauze sheets. Mr. Welch described a flat circular composite sheet of gauzes stretched across the inner face of the relief door orifice. The author's proposal was a shallow conical structure, also composed of sheets of gauze, attached at its rim to the circumference of the relief valve orifice, the apex of the cone pointing into the crankcase. The cone ensured stiffness and permanence, and provided an enhanced free area. The author agreed with Mr. Welch's suggestion of one sheet of 10-mesh gauze, with three sheets of gauzes of mesh 20 to 30 per inch.

A proposal frequently made, but one which seemed impracticable to the author, was that gauzes should be stretched across the large openings in the "A" frames. An engine might leave the builders' works thus, but it was doubtful if the gauzes would survive, in an effective form, after the first overhaul involving attention to main bearings, etc.

Engineers interested in Diesel propelling engines, whether as builders or as operators, seemed divisible into three categories, viz.: (i) those who did not believe that their engines could ever have an explosion; (ii) those who believed that an explosion *could* occur, but did nothing about the matter, being prepared to accept whatever slight risk there might be; (iii) those who were explosion conscious and who would go to almost any lengths to ensure the safety of the operating staff, but were uncertain how to achieve the desired result. In this third class the author was to be found. In any adverse eventuality the hind sight prophets could always prove the engine builder to have been remiss—whatever he did!

The risk of fatality by explosion was very small; but statistical probabilities, while definable in comforting general arithmetical terms, could not have any relation to the individual and to the particular.

In the author's opinion it was high time those authoritative public bodies who were responsible for the issue of safety and seaworthiness certificates should make a pronouncement on flame traps and kindred contrivances for the guidance of engine builders and shipowners. It was not sufficient to promulgate rules for the size of relief doors. Relief doors simply facilitated the escape of flames and hot gases. In the meantime Mr. Welch, as an eminent, highly experienced engineer, had given his personal opinion with commendable frankness.

Mr. Yellowley, in commenting upon cylinder liner construction, mentioned that, in a three-piece liner type made by his company, the centre portion was a casting of spheroidal graphitic cast iron.

The author had given thought to the use of this material for the combustion chamber section of the cylinder liner as his firm was a licensee for the nodular iron process. But, as he had not been fully satisfied of the adequacy of this material for the purpose, and as there were many liners to be made, it seemed prudent to use cast steel.

Regarding lubricating oil points: normally there were no oil grooves in cylinder liners but, occasionally, where an owner specifically desired them, grooves were provided.

The remarks in Mr. Yellowley's ante-penultimate paragraph were not clear to the author. The out-of-balance forces per cylinder line were, of course, transmitted through the engine structure as in an engine having cylinder covers, except that such forces in an opposed-piston engine were relatively much less important.

Regarding the ratio of connecting rod to crank: the words which Mr. Yellowley used were not those of the author, either in letter or in spirit. The author was not conscious of having expressed himself forcibly on the subject; nor was he conscious of having prophesied any dire results from the adoption of a slightly shorter rod. At two places in the text there was mention of the ratio of connecting rod to crank. The first was when referring to Fig. 2; and the relevant sentence stated: "The engine structure is stoutly built from fabricated steel plates; the crankcase is completely isolated from the cylinders; the ratio of connecting rod to crank is 4 : 1". At another place, in referring to connecting rod details, the relevant clauses were: "The ratio of length to diameter varies from 12 : 1 to 14 : 1. The ratio of connecting rod length to crank radius is 4 : 1. As the result of experiences, many years ago, with shorter rods, a ratio of 4 : 1 is the minimum that the author would offer to any shipping company".

The first time the author proposed a reduction in connecting rod ratios was for some small, high speed steam propelling engines of forced lubricated enclosed design for certain Admiralty craft. The proposal came to nothing. The next time was in connexion with certain Belfast-built single-acting four-stroke crosshead type Diesel engines for single-screw vessels. The intention was to standardize the engine type on a reduced ratio of connecting rod/crank if experience proved favourable. Experience did not prove favourable. The engines oscillated more freely than was acceptable, and this in turn tended to deterioration of a kind that was more general than intense.

The standard engine type of the period had a height/base ratio of 2·4, and a connecting rod/crank ratio of 4·5; the engines were characterized by remarkable steadiness in operation. The engines in question were given a longer stroke than normal, to ensure a high propeller efficiency; and, to recover some of the height, the connecting rod/crank ratio was reduced to 3·75. The firm's double-acting engines from the beginning had had a ratio of 4 : 1, and this ratio prevailed for all sizes and proportions of engine. That was over thirty years ago.

Mr. Yellowley, referring to Tables III and IV, asked which rating was generally chosen for the propeller design. No simple answer could be given to this question; inevitably there were variations from installation to installation, in accordance with the terms and requirements of each specific contract. Sometimes one rating was dominant in the mind of the purchaser, sometimes another; but it was always possible to offer an acceptable proposal within the boundaries of the relevant guidance tables. Tables III and IV, and the paragraphs relating thereto, were hardly capable of closer definition, having regard to their purpose.

The author was deeply appreciative of the kindly personal references made by contributors to the discussion. There were many points of affinity between the engine designer, who brought new engines into existence and fostered their infancy, and the marine engineer who nurtured their adolescence and saw them through maturity to old age. Both were strong individualists; both pursued their vocations in relative isolation; both were engaged in occupations which the outside world but dimly apprehended, that world which was apt to class them as sons of Jeshurun; and, consciously or not, both were ultimately sustained by the contributive impulsion enshrined in the old-time words: "Although they shall not dwell where they will . . . nor be sought for in public counsel . . . nor sit on the judges' seat . . . they will maintain the state of the world . . . for all their desire is in the work of their craft". *Pernoctant nobiscum peregrinantur!*

INSTITUTE ACTIVITIES

Minutes of Proceedings of the Ordinary Meeting Held at the Institute on Tuesday, 12th March 1957

An Ordinary Meeting was held at the Institute on Tuesday, 12th March 1957, at 5.30 p.m., when a paper entitled "The Harland and Wolff Pressure Charged Two-stroke Single-acting Engine", by Mr. C. C. Pounder (Vice-President) was presented and discussed. Mr. T. W. Longmuir (Chairman of Council) was in the Chair and 174 members and visitors were present. Nine speakers took part in the discussion.

A vote of thanks to the author, proposed by the Chairman, was accorded by acclamation. The meeting ended at 8.0 p.m.

Calcutta

A meeting of the Calcutta Section was held on Wednesday, 27th March 1957, in the Examination Room at Marine House. Mr. J. Connal (Local Vice-President) was in the Chair and thirty-five members and visitors attended, also forty-six cadets from the Marine Engineering College.

A paper was presented by Mr. D'Abreo, which he illustrated with lantern slides. A discussion followed regarding the relative efficiency of the constant pressure or common rail system and the jerk pump system, in which Mr. J. Stevenson and Mr. H. K. Gupta took part.

The Chairman proposed a vote of thanks to the author.

British Columbia

The Annual Spring Dinner Meeting of the British Columbia Section was held at H.M.C.S. *Naden* in Esquimalt, Vancouver Island, on 15th February 1957. The entire arrangements were handled by the Victoria Committee in their usual capable manner. The attendance figure was very gratifying as a total of forty-five members and guests attended, with a large number of Vancouver members making the overnight journey involved.

Attending as a guest was Mr. E. Drake, British Columbia President of The Society of Naval Architects and Marine Engineers, who spoke briefly and extended an invitation to the members to attend the meetings and functions of the Society, which were scheduled to commence in Victoria on the following day.

Capt.(E) C. I. Hinchcliffe, R.C.N., was Chairman of the dinner meeting and the speaker of the evening, Dr. William H. English, B.A., Ph.D., the Head of the Marine Physics Section of the Pacific Naval Laboratory, H.M.C.S. Dockyard, was introduced by Mr. James McPherson of Victoria.

After the dinner Dr. English lectured on the subject of "Marine Hydrodynamics". He covered the subject in a most interesting manner, from the widest application of the principles involved to the various fields of practical interest to naval architects and marine engineers. The lecture was made doubly interesting by the excellent series of projections showing some of the experimental apparatus in use in the Naval Laboratory under his direction.

The lecture was followed by a discussion and question period in which members participated with great interest.

A vote of thanks to the author was moved by Mr. W. Dey and the meeting adjourned at 11.30 p.m.

South Wales

A meeting of the South Wales Section was held jointly with the Institute of Welding on Thursday, 14th March 1957, at the South Wales Institute of Engineers, Cardiff. Mr. J. Wormald, B.Sc. (Chairman of the Section) was in the Chair and there was a fair attendance.

Mr. J. A. Dorrat presented his paper on "Welding in Marine Engineering", which was enthusiastically received and many questions were asked in the discussion that followed.

A vote of thanks proposed by Mr. D. Skae (Vice-President), seconded by Mr. Lewis on behalf of the Institute of Welding, was carried by acclamation.

West Midlands

The Fourth Annual Dinner of the West Midlands Section was held at the Imperial Hotel, Birmingham, on the 14th March 1957.

The Chairman of the Section, Mr. H. E. Upton, O.B.E., was in the Chair, and members had much pleasure in welcoming a number of guests, among whom were Mr. T. W. Longmuir (Chairman of Council), Dr. A. L. Clarke, Mr. J. A. Kendall, Capt. H. K. Hodgkin, R.N., and the Secretary of the Institute, Mr. J. Stuart Robinson.

Members and guests attended an informal reception before dinner and after the Chairman had proposed the toast to the Queen, the gathering of ninety-six members and their guests sat down to an excellent meal.

The after-dinner speeches were given by the Chairman, by Dr. Clarke, who proposed "The Institute of Marine Engineers", and by Mr. Longmuir, who responded; finally, Mr. Robinson added an amusing and interesting account of his visit to India.

The cabaret provided a very agreeable interlude and at the end of the evening members were unanimous in voting the evening a success and in particular wished to extend their thanks to the guests, many of whom had travelled considerable distances.

Election of Members

Elected 15th April 1957

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John Barnes Dick
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Albert Fowler
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Lacy Townsend Gaskill
Frederick Gills
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Bih-Shung Hah
James G. Hayward
James Howard

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Alexander Caldwell Hutchinson, B.Sc.(Eng.) London
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William Orkney Williamson
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R. Boorsma
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Elected 14th May 1957

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Stanley Maxwell Pilling
Walter Glyn Rhys
Rajendra Lall Sachchar
Michael Francis House Singer
Roy Maurice Spencer
John Martin Stark
Derrick Frederick Streeton
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Harold Frederick Cook

TRANSFER FROM ASSOCIATE TO ASSOCIATE MEMBER
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Clifford Soranson

TRANSFER FROM GRADUATE TO ASSOCIATE MEMBER
Thomas Dalgleish

TRANSFER FROM STUDENT TO GRADUATE
Gordon William Pinhey, B.Sc.(Durham)

TRANSFER FROM PROBATIONER STUDENT TO STUDENT
Charles Allan Caldwell

OBITUARY

FREDERICK ALEXANDER CHAMBERLAIN (Member 8343) was born in 1894. He was an indentured apprentice with Alfred Dodman and Co., Ltd., King's Lynn, from 1908/13, and continued in this company's service as an assistant engineering draughtsman for another year. From 1914/28 he was chief engineering draughtsman for Crabtree and Co., Ltd., Great Yarmouth, from 1917 onwards serving them also as chief ship draughtsman. From 1918/28 he was occasional surveyor for Lloyd's agents in Great Yarmouth.

From 1928 until his death on 5th April 1957 Mr. Chamberlain was managing director of Claxton and Co., Ltd., Ramsgate; he also worked independently as a marine surveyor and naval architect. During the Second World War he carried out many repairs for the Admiralty on small craft operating in the English Channel.

Mr. Chamberlain was elected a Member of the Institute in 1937.

ALLEN OWEN ELLIS (Member 12735) was killed at the age of thirty-eight on 14th March 1957 in the Ringway air disaster. For the last two years he had been employed as a senior service engineer by the National Gas and Oil Engine Co., Ltd., and was returning home from Rotterdam where he had been visiting a vessel in which the firm's engines were fitted when the disaster occurred.

Mr. Ellis served an apprenticeship with Grayson, Rollo and Clover Docks, Liverpool, and then spent ten years at sea, from 1939/49, as junior to second engineer with the Elder Dempster Lines. He obtained a First Class Ministry of War Transport Motor Certificate in 1944. From 1949/54 he was marine engineer in charge for the Cameroon Development Corporation, British Cameroons, West Africa.

Mr. Ellis was elected to Membership of the Institute in 1950.

GEORGE FRANC FISHER (Member 7580) was born in 1890 in Earlston, Berwickshire, and was educated at Earlston Public School. He was apprenticed from 1906/11 with Peter Readman and Sons, engineers of that town, and went to sea in 1914 in the s.s. *Hampstead,* which was owned by Watts, Watts and Co., Ltd. The following year he was employed by the Currie Line as third engineer of the s.s. *Weiman.* From February 1916 until he was demobilized in 1919 he served in minesweepers, at first as engineer sub-lieutenant, being promoted engineer lieutenant in 1918.

In 1920 Mr. Fisher joined the Colonial Service and was appointed second engineer with the Lake Albert Marine Service, Uganda. In 1923 he was transferred to the service of the Kenya and Uganda Railways and Harbours and served on Lake Victoria, Kenya, and Lake Kioga, Uganda, as second and chief engineer, until in 1936 he was appointed engineer-in-charge, Lake Kioga, where he remained until he retired in 1944 on medical grounds.

In July 1945 Mr. Fisher was apointed a surveyor with the Ministry of War Transport but was again obliged to resign in October 1946 due to ill health. He joined the staff of Flannery, Baggallay and Johnson, Ltd., nine months later but was only able to remain at work until October 1949. He died on 27th March 1957. He had been a Member of the Institute since 1934.

JAMES GORDON (Member 14098) was born in 1890. He served a six-years' apprenticeship with Barry and Hendry and Co., Ltd., and then joined the Shaw Savill and Albion Co., Ltd. (in 1913 the Aberdeen and Commonwealth Line), remaining with them until his retirement forty-three years later, in 1956. He was chief engineer of the t.s.s. *Moreton Bay* at the time of his retirement. Mr. Gordon was superintending alterations on 13th February 1957 at the home in Sussex to which he was about to move when he had a heart attack and died immediately. He was elected a Member of the Institute in 1953.

ROBERT SYDNEY POTTS (Member 6000) was European representative for the Maritime Overseas Corporation, New York, at the time of his sudden death at the age of fifty-six on 5th March 1957. He served an engineering apprenticeship with Swan, Hunter and Wigham Richardson, Ltd., and then spent eighteen months in their drawing office. For the next seven years, 1922/29, he was a seagoing engineer with the British Tanker Co., Ltd., and obtained a First Class Board of Trade Steam Certificate with Motor Endorsement; in 1933 he was awarded an Extra First Class Certificate. For five years after leaving the sea he was assistant consulting engineer with Swan Macfarlane and Company, Newcastle upon Tyne, and from 1935/38 he was assistant superintendent engineer to the United Africa Co., Ltd., now the Palm Line. Mr. Potts then set up in business on his own account as a consulting engineer and naval architect, later taking a partner, and undertaking work for the British Corporation. This was interrupted by the late war and in 1940 he was appointed a member of the technical advisory staff to the Norwegian Shipping and Trade Mission in London and only returned to his own business in 1949.

Mr. Potts was elected to Membership of the Institute in 1928; he was a Fellow of the Society of Consulting Marine Engineers and Ship Surveyors, an Associate Member of the Institution of Naval Architects, and a Member of the North East Coast Institution of Engineers and Shipbuilders, and the Society of Naval Architects and Marine Engineers in New York.

JAMES ROBERTSON (Member 14144) was born in 1894. He served an apprenticeship from 1910/15 at the Central Marine Engine Works, West Hartlepool, and then served as a junior engineer for five years with Furness, Withy and Co., Ltd. From 1926 he served in ships owned by F. C. Strick and Company, as chief engineer for many years, until discharged from the Merchant Navy in 1944 on account of war injuries. He obtained a First Class Board of Trade Certificate in 1928. In 1944 he joined the Beldam Packing and Rubber Company at Middlesbrough, as assistant manager, and three years later was appointed manager at the Hull branch of this company. He died on 27th March 1957.

Mr. Robertson was elected a Member of the Institute in 1953 and in 1955 was elected to Membership of the Kingston upon Hull and East Midlands Section Committee.

LEO ROSS (Member 2669) was apprenticed to Mordey, Carney and Co., Ltd., Southampton, from 1897/1902. For the next year he sailed with the Red Star Line and for a further five years after that with the Royal Mail Steam Packet Company, Southampton; he obtained a First Class Board of Trade Steam Certificate. He then worked for Messrs. Topham, Jones and Railton as second and chief engineer on steam hoppers and dredgers until the end of 1911, when he was appointed engineer superintendent to the company, and spent seven years in Rosyth superintending the dredging of the Naval base. On returning to Southampton in 1920 he joined the James Dredging Towage and Transport Co., Ltd., as superintendent engineer and works manager and retired from this appointment thirty-two years later on 31st March 1952. Mr. Ross died on 14th April 1957, aged seventy-four years. He had been a Member of the Institute since 1912.

FRED COX SMITH (Member 11360) was born in 1898. He served an apprenticeship with Alexander Shanks and Sons, Arbroath, and then spent twenty years at sea, with Messrs. Alfred Holt and Company for twelve years and with the Currie Line as second and chief engineer for the remaining eight. He obtained a First Class Board of Trade Steam Certificate in 1930. In 1942 he was appointed an engineer surveyor with the British Corporation Register of Shipping and Aircraft and remained with them (and later with Lloyd's Register of Shipping when the two associations merged) until his death on 9th March 1957. Mr. Smith had been a Member of the Institute since 1947 and was also a Member of the North East Coast Institution of Engineers and Shipbuilders.

GEORGE RODHAM UNTHANK (Member 6433) served an engineering apprenticeship with a rubber company in Silvertown, the India Rubber Gutta Percha and Telegraph Works Company, from 1899/1906, and continued in their employment as draughtsman and eventually as assistant maintenance engineer for another eleven years. On leaving this company he worked for a time for a firm engaged in aviation work, in the manufacture of propellers, but in 1919 he joined R. and H. Green and Silley Weir, Ltd., as consulting engineer, an association which lasted until his retirement in 1952. For several years he specialized in the replanning and rebuilding of many of the company's machine shops; later, he was responsible for many new buildings, including the Victor Engineering Works at Rainham, the original general offices of the company at the Royal Albert Dock and the Jubilee Flats in Manchester Road, Millwall, which were opened by H.R.H. The Duchess of Gloucester in 1936.

It was in the early 1920s that Mr. Unthank started to work on the problem of oily water separators and in 1924 the first installation was completed. One of his major projects in this field was the 1,000-ton tank cleaning installation at Falmouth, which has been the model for similar installations all over the world. He continued to take an active interest in the separator even after his retirement. Another development which concerned him was the use of pulverized coal in ships and he developed much of the equipment which went into the *Hororata* prior to her round-the-world voyage, using this type of fuel. His considerable inventive powers were also brought to bear in the design of hook couplings and other marine equipment and he was largely responsible for designing the vacuum plant of the *Interknit* and patented two suction nozzles, one of which has been installed on a tanker cleaning barge recently commissioned in a Northern French port. Mr. Unthank died on 10th April 1957, aged seventy years. He had been a Member of the Institute since 1930.

CECIL ST. CLERE WILLIAMS (Member 15494) was born in 1899 at Plymouth and was the son of a Royal Navy gunnery officer. He entered the Royal Navy as a boy artificer in 1914, graduating as an engine room artificer three years later. During the next ten years his service was mainly in coal burning destroyers. In 1927 he transferred to the Royal Australian Navy as a chief engine room artificer to take up instructional duties at the Naval College, Jervis Bay, New South Wales. After his promotion to warrant engineer in 1934 he served in H.M.A.S. *Brisbane* on her last voyage to England. In 1937 Mr. Williams retired to the Emergency List and became proprietor of an automotive service station at Merrylands, New South Wales. He was recalled to service a few days before the outbreak of the second World War and appointed to H.M.A.S. *Australia,* 10,000-ton County Class cruiser in which he had served previously and which set sail without delay for Home Waters. Later, he served under great stress in the Pacific as damage control officer in the same ship during actions in which accompanying American and Australian capital ships were sunk. He was mentioned in dispatches in 1943 for distinguished services in the Pacific. In 1944 he was appointed assistant to the squadron engineer officer and in 1948 assistant to the general manager at Williamstown Dockyard, Victoria. His final appointment in the Australian Navy was as assistant to the general manager, Garden Island, New South Wales. In 1950 he was promoted Lieutenant(E).

On reaching retiring age in 1954, Lieutenant Williams was employed as works engineer by a Sydney plastics manufacturer, a position he held until ill health forced his retirement in 1956. Whilst in England for treatment he died at Callington, Cornwall, on 8th December 1956.

Lieutenant Williams had been elected to Membership of the Institute in 1955.