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Some Recent Diesel Installations and their Characteristics.

READ

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CHAIRMAN: MR. R. RAINIE, M.C. (Chairman of Council).

Synopsis.

THE choice, scope and treatment of the subject have been influenced very largely by questions raised by shipowners, superintendents and sea-going engineers.

Typical recent installations for cargo and passenger ships are referred to in very brief outline, mention being made of such features as pillarless engine rooms and resilient foundations for main engines. Then follow descriptions of propelling engines of double-acting two-cycle, single-acting two-cycle and four-cycle, supercharged types; four-cycle and two-cycle auxiliary engines are also briefly described.

Points of general interest then receive attention, such as engine rating, manœuvrability, crankshaft alignment, balancing, piston rings, exhaust and silencing systems, etc. The paper closes with examples showing the progress made in recent years in reducing the weight and space of Diesel machinery. The Diesel engines described are of Harland-B. & W. type, as developed by Messrs. Harland & Wolff, Ltd., to the requirements of British shipowners.

The paper is divided into three parts. The first part outlines very briefly a number of representative installations built during the last two years; the second part describes the propelling and auxiliary engines; the third part refers to a number of points of general interest.

I.—OUTLINE OF TYPICAL INSTALLATIONS.

(a) Double-acting Two-cycle Installations.

Single-screw Cargo Ship.—The first example is the machinery of a vessel of 14,000 tons displacement, deadweight carrying capacity 10,000 tons, speed 14 knots.

The main propelling engine has 6 cylinders 530 mm. (20·87in.) bore, 1,250 mm. (49·2in.) stroke, 4,500 s.h.p., 110 r.p.m. The engine is very compact, the cylinder centres being only 2·36 times the cylinder bore. All pumps are direct-driven from the engine, for use at sea, and steam requirements are met by the utilisation of the exhaust gases in a composite oil-fired and waste-heat boiler. At sea, therefore, the engine is completely self-contained. In harbour, an oil-fired Scotch boiler supplies winches, ship's heating, etc. The engine pumps are duplicated as independent units.

After comparing capital cost, economy, etc. of steam-driven auxiliaries with Diesel generators and electrical auxiliary plant, some owners have adopted the former, thus appreciably reducing the initial outlay.

A number of features in the design of such ships have attracted attention, but reference need only be made herein to the funnel and deckhouses, which comprise one structure (see Fig. 1). The principle underlying this arrangement is likely to be developed further. The deckhouses are rounded.

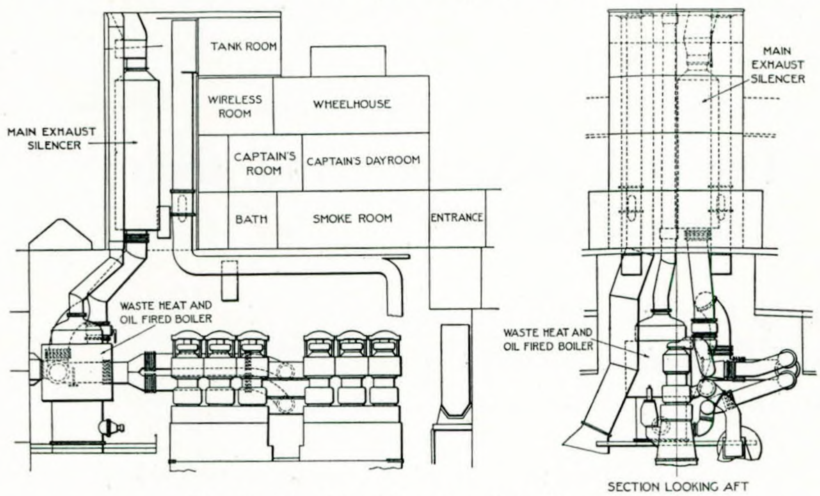


FIG. 1.—Combined funnel and deckhouses.

The bunker capacity—assuming for the purpose a generous overall daily consumption at full speed—enables ships of the class described to voyage 16,000 miles without re-bunkering, with 15 per cent. allowance in excess.

It would, perhaps, be more accurate to describe ships of this type as cargo liners.

Large Twin-screw Refrigerated Cargo Vessel.—The second example is a twin-screw refrigerated cargo ship of 20,000 tons displacement, gross tonnage 11,000, speed 17 knots.

Each propelling engine has 6 cylinders, 620 mm. (24·41in.) bore, 1,400 mm. (55·12in.) stroke, with an aggregate s.h.p. of 14,000 at 110 r.p.m. All the pumps are independent, motor-driven. Where the refrigerating machinery is electrically-driven, four Diesel generators are fitted; where operated by horizontal, self-contained, Diesel engines, three are fully sufficient. The generator engines are usually 6-cylinder 4-cycle sets, 300-330 kW., 270-300 r.p.m., arranged in the engine room wings.

Comparison of such ships with similar vessels previously built is interesting. The latter have twin 10-cylinder, 740 mm. (29·13in.) bore, 1,500 mm. (59·00in.) stroke, 4 C. S.A. engines, Büchi supercharged, s.h.p. 12,000 at 119 r.p.m.

The saving in engine room length in the latest vessels is 18·5ft., the reduction in main engine weight 160 tons.

As two of the auxiliary engines named above are together as long as a main engine, their adoption affects the engine room length. Machines of the 2-cycle type illustrated later would produce a further saving in length of engine room.

By adopting the pillarless engine room construction referred to presently the lay-out is considerably improved.

High-powered Passenger Liner.—For this example is chosen a twin-screw liner, 36,000 tons displacement at load draft, gross tonnage 27,000, speed 20 knots.

Each propelling engine has 10 cylinders, 660 mm. (25·99in.) bore, 1,500 mm. (59·00in.) stroke; at 102 r.p.m. the output is 12,000 s.h.p. The thrust blocks are integral with the engines, as in the previous examples. The propelling and generator engines are located in contiguous rooms. The engine room pumps, etc., call for no special comment. There are four independently-driven main scavenge blowers, each of capacity 880 metres^3 (31,000 cub. ft.) per minute at 0·24 kg. cm.^{2} (3·4lb. sq. in.) air pressure, the direct-coupled motors being rated at 590 b.h.p. each, speed range 2,200 to 2,800 r.p.m. The blowers discharge into a ring main, two blowers being capable of supplying the full quantity of air for two engines. For the largest engines it is sound policy to avoid driving pumps, blowers, etc., from the engine. This used to be also the policy with steam engines; it prevents propelling engine stoppages from causes associated with them. Each engine exhausts to a 1,800 sq. ft. thimble-tube type boiler, raising 2,500lb. steam per hour at 100lb. sq. in. pressure. There are also two

850 sq. ft. oil-fired boilers each capable of supplying 5,000lb. steam per hour. There are five 1,000 b.h.p. 2-cycle 6-cylinder auxiliary engines, 350 mm. (13·78in.) bore, 620 mm. (24·41in.) stroke, 260 r.p.m., coupled to 700-kW. dynamos. The adoption of independent blowers naturally increases the auxiliary load. The engines are of the design shown later; the pistons are oil-cooled.

In the building and finishing of such a vessel unusual care is given to ensure that in the passenger spaces there is freedom from vibration and noise. Special designs are adopted for many ship details; main and auxiliary exhaust pipe arrangements, machinery space insulation, and so on, all receive special attention.

A noteworthy feature lately has been the adoption of the Harland & Wolff patented construction of pillarless engine room.

One of the factors in preventing engine room vibrations from being transmitted to accommodation spaces is the freeing of the tank top—with its small elastic deformations—from all connection with accommodation decks, except at the shipside and bulkheads, where there is no elastic deformation. This is achieved as outlined in Fig. 2.

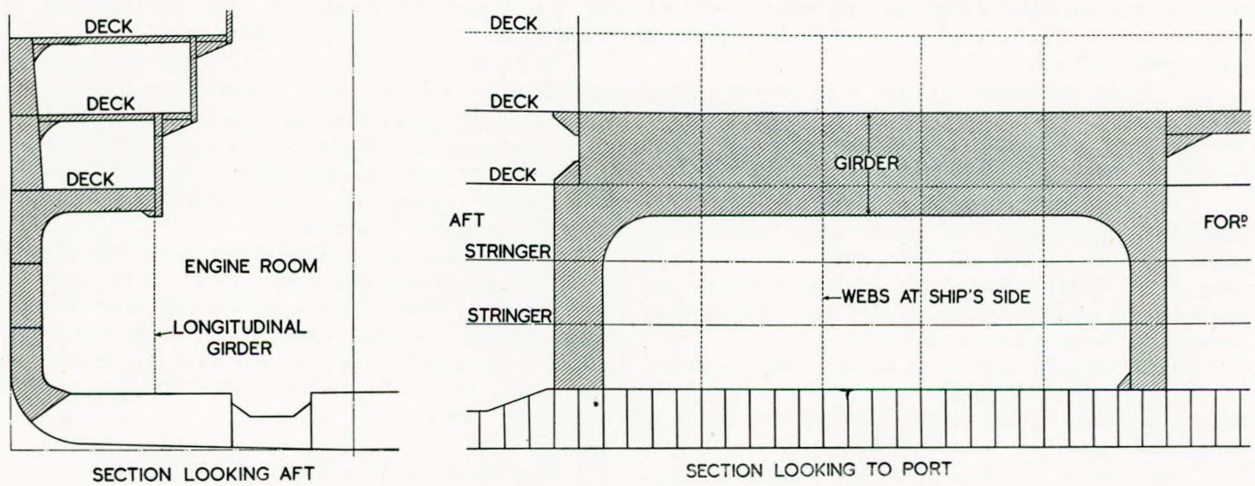


FIG. 2.—Pillarless construction of engine room.

The arrangement consists, in essentials, of the elimination of all pillars which ordinarily carry the decks from the tank top, by providing transverse and/or longitudinal girders adapted to carry the decks and casings from the transverse bulkheads and from the deep frames. Thus, in Fig. 2 the lowermost deck, in way of the engine room, is supported by a transverse girder system extending inwards from the shipside to the inboard edge of the decks. Fore and aft girders are also provided. The fore and aft casings are strengthened, to form part of the system. In a moderately-sized ship either transverse girders or longitudinal girders may be sufficient.

The depth of the double bottom in way of the machinery spaces is increased to raise the frequency of vibration well above the prudent limit; the increased tank capacity so obtained is an advantage.

The system has been very successful to date and the cost involved is little, if anything, above that for the usual pillared construction.

(b) Single-acting Two-cycle Installations.

Machinery of the type described herein has been fitted in many ships ranging from small barges with four cylinders, to powerful twin-screw vessels.

It is an especially useful engine where low head room is desired. For instance 12,000 s.h.p. can be obtained on two shafts with a vertical height of 26ft. from tank top to deck beams, allowing ample head room for overhauling; on four screws 32,000 s.h.p. can conservatively be obtained with 31ft.

The advantage to shipowners of this characteristic is fully appreciated by those who have considered the matter; it provides them with one, often two, additional decks in way of the engine spaces.

Cargo Vessels and Small Craft.—One example is a 170ft. single-screw coaster, speed 10 knots. The engine has 6 cylinders, 280 mm. (11·02in.) bore, 500 mm. (19·69in.) stroke, 240 r.p.m. Bilge, sanitary, fuel surcharging and lubricating oil pumps are driven from the engine; other auxiliaries are electrically-driven. There are two 50-kW. generating sets.

Another vessel, 220ft. long, for passenger service in the East, has twin-screw engines with 6 cylinders 350 mm. (13·78in.) bore, 620 mm. (24·41in.) stroke, 240 r.p.m. normal, but capable of 20 per cent. overload at increased revolutions. The pistons are oil-cooled. The electric sets are two of 60-kW. output with a 5-kW. emergency set.

A last example is a 275-ft. single-screw cattle ship having a 6-cylinder engine 500 mm. (19·69in.) bore, 900 mm. (35·43in.) stroke, rated at 2,100 s.h.p. at 160 r.p.m. The engine room pumps are all motor-driven; the generating plant consists of three 75-kW. sets.

Salvage Tender.—This is a Diesel-electric vessel.

The engines develop their maximum power at constant revolutions while the propeller revolutions vary, depending upon whether the ship is towing or running free; the system gives great flexibility of power.

The main engines comprise two 8-cylinder, 220 mm. (8·66in.) bore, 370 mm. (14·57in.) stroke sets, 550 r.p.m., having self-contained pumps; each is direct-coupled to a generator. The fire-fighting equipment includes a high-lift three-stage centrifugal pump. Current for engine room and ship services is supplied by two 45-kW. sets, 700 r.p.m., of the same general type as the main engines. The two main generators, each 410 kW., 350 volts, 550 r.p.m., supply direct current to the two main propulsion motors, each of 500 s.h.p. There are two exciter sets, one for each main generator and propulsion motor.

The speed and direction of the vessel are completely controlled from the bridge.

Normally, each generator drives its own propulsion motor. When, however, the vessel is on salvage or fire service, one generator supplies current for the salvage pump and the other for both propulsion motors—which can still be operated quite independently of each other.

Such an installation may be somewhat more costly than a steam plant, but its immediate availability, large bunker capacity, and manœuvring flexibility make it a very attractive scheme.

Cross-Channel Passenger Ship.—For this type of craft, engines of the trunk type are almost essential. Latest vessels have been about 345ft. long, 4,400 tons displacement, 17 knots speed, and have set, by common admission, a new standard of comfort in cross-Channel services.

Such vessels are twin-screw. each engine having 10 cylinders, 500 mm. (19·69in.) bore, 900 mm. (35·43in.) stroke, very moderately rated at about 120 r.p.m.

The pumps for ship and engine-room services need no special reference. There is a composite thimble-tube type exhaust-gas and oil-fired boiler raising steam at 80lb. sq. in. pressure. Three 175-kW. auxiliary engines, 6 cylinders, 220 mm. (8·66in.) bore, 370 mm. (14·57in.) stroke, 300 r.p.m., supply current for ship and engine room purposes.

A feature of the latest ships is the mounting of the propelling engines on rubber.

To minimize the transmission of noise and vibration from the machinery to the hull, the two possibilities open to the builder are: (*a*) to produce a quiet-running engine; (*b*) to isolate the machinery.

The reduction of noise at its source, by making the machinery as quiet as possible, is naturally the builder's first concern. But when he feels that he can do no more, it becomes necessary to explore the second possibility, viz.: the insulation of the machinery from the hull.

In departing from the usual system of bolting propelling engines solidly to cast-iron chocks, a number of problems arise, *e.g.* the necessity for preserving crankshaft alignment, also alignment with the tunnel shafting; the prevention of sideways or end-wise movement of the engine on its foundation, also swaying of the engine and ensurance of freedom from trouble with holding-down bolts.

In the vessels referred to, the aggregate surface of the special rubber used is ample for very safe continuous loading; safety strips ensure that, in the unlikely event of the rubber failing, the engine is still carried on a cast-iron surface of normal area. The bedplate longitudinal girders are so designed that in no circumstances will the crankshaft alignment suffer. The thrust block is integral with the bedplate. Hard wood chocks, backed by steel wedges, at the fore and aft ends of the bedplate take the propeller thrust. To maintain alignment with the intermediate shafting, the foremost bearing block is also mounted on rubber.

Simple gauges are provided for the engineers to take periodical readings from the foundations, these being recorded in a special book as required by the Board of Trade in view of the lack of information available at present.

With the experience to date there has not been the slightest difficulty with rubber foundations, the alterations in gauge readings being infinitesimal.

The auxiliary engines in some recent vessels have been carried upon springs. Into all water, fuel, air, etc. pipes about such engines short non-metallic lengths are introduced.

Some years ago rubber-lined stern tubes were fitted to a Belfast-built motor yacht, with satisfactory results. The rubber was cast into stern bushes of cast steel, a copious supply of water being circulated through same in service. It would be interesting to see such stern tubes fitted to a large passenger vessel.

(c) Supercharged Four-cycle Installations.

A remarkable vitality persists in 4-cycle single-acting machinery, continuously supercharged.

This type of prime mover is favoured for tankers and cargo vessels by many shipowners. One example, out of many which could be chosen, is a 460-ft. single-screw tanker, having a 6-cylinder engine, 740 mm. (29·13in.) bore, 1,500 mm. (59·00in.) stroke, 105 r.p.m., 3,000 s.h.p. with under-piston super-charging. The cylinder heads and jackets are cooled by fresh water—sometimes salt water. The pistons are cooled by oil, unless specifically required by an owner to be cooled by fresh or salt water. The pumps for fresh and salt water circulating, lubricating oil, fuel surcharging, etc. are chain-driven from the crankshaft, stand-by pumps being provided. One, or two, single-ended boilers supply steam to pumps, deck machinery, tank steaming, etc. If two boilers, one is heated by the main engine exhaust gases and oil fuel; the other is oil-fired only. If only one boiler, this is oil-fired on natural draught; a thimble-tube boiler of ample size for exhaust gas and oil-firing is then also supplied. The thrust block may be independent, or incorporated in the engine.

II.—MAIN AND AUXILIARY ENGINES.

(a) Double-acting Two-cycle Engines.

Fig. 3 shows, as an example, an engine 530 mm. (20·87in.) bore, 1,250 mm. (49·2in.) stroke. A fore and aft section is given in Fig. 4.

The bedplate is cast iron, with separate cross-girders; this construction allows the components to be restricted to such dimensions that sound and strong castings are ensured. In the largest engines the cross-girders are cast steel; for all but exceptional circumstances cast iron is completely satisfactory.

The crankshaft is fully built. The shafting, framing, chain drive, etc. are carefully calculated to avoid all objectionable vibrational stresses.

The engine has longitudinal scavenging, air being admitted at cylinder mid-length—*i.e.* at the inner end of the respective top and bottom strokes—and exhaust gases discharged at the outer ends of the cylinder. The influence of this system of scavenging upon the engine rating receives attention later.

The scavenge and exhaust ports are respectively controlled by the main and exhaust pistons. The exhaust pistons are a fixed distance apart and reciprocate together. The eccentrics are integral with the crankwebs, the angle of advance being usually $180° + 7°$ ahead, becoming $180° - 7°$ astern. Fig. 6 shows the periods. When going ahead there is a large lag between opening of exhaust ports and scavenge ports. The exhaust area thus becomes so great that the pressure at end of expansion falls to atmospheric, or below; the scavenge air attains a velocity which, with the whirling action, effectively scavenges the cylinder. The astern lag is sufficiently large at the end of the expansion to prevent gases blowing into the scavenge space; a suction effect is not necessary.

The piston rod guides are at the back of the engine, *i.e.* for twin-screws the ahead guide pressure is taken on the side bars, for single-screw engines on the guide plate. The crosshead shoe is lubricated from the cooling oil pipe ascending to the main pistons. By lubricating the shoe and not the guide plate the crosshead is able to work with equal effectiveness on guide bars and guide plate.

Fig. 5A shows a 620 mm. (24·41in.) cylinder.

The piston consists of two end castings of chrome steel and a centre piece of special cast iron, all strongly bolted together. The piston rod flange is sandwiched between the bottom and centre components of the piston. There are six piston rings at

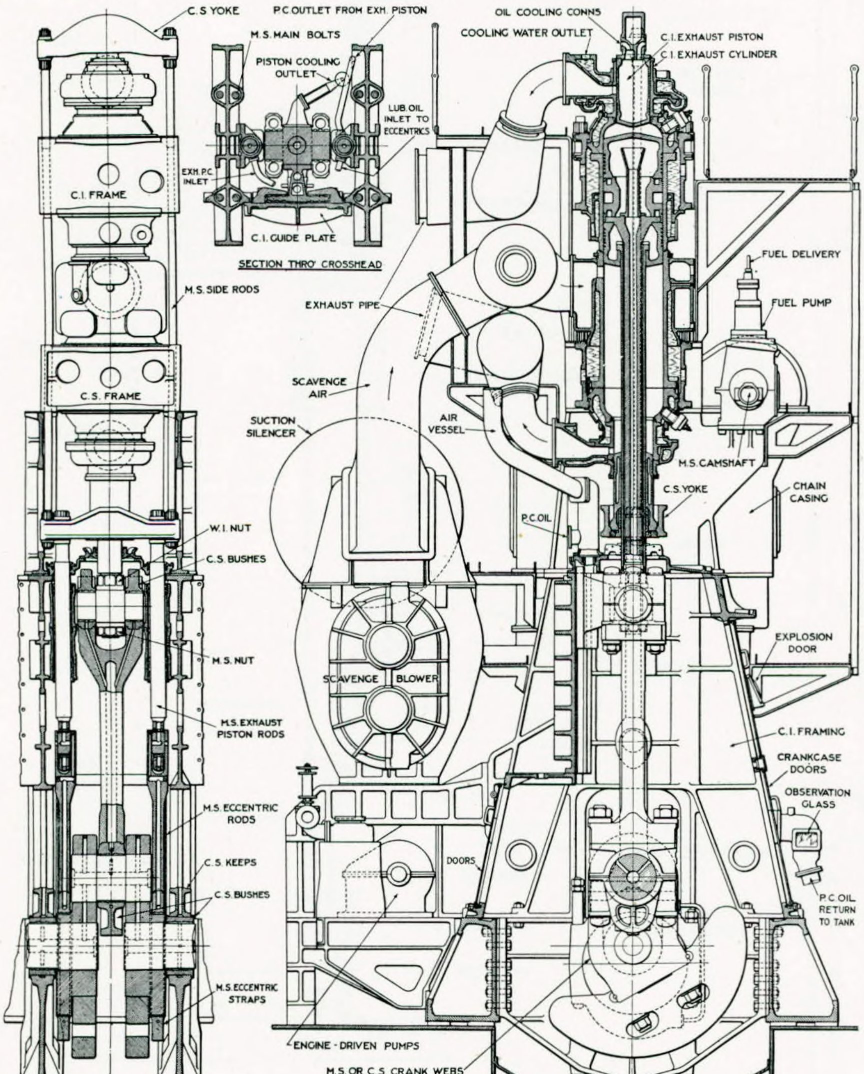


FIG. 3.—Double-acting two-cycle engine.

each end with one scraper ring adjacent to the scavenge ports; cast-iron carrier rings, ground on the faces, are caulked into the piston. The piston rings are located at the maximum distance from the piston crown to minimize the possibility of rings sticking, and to allow the piston crown freedom to expand.

The mild-steel piston rod is protected from the hot gases by a cast-iron sleeve screwed into the lower piston and sliding freely on a bush at the crosshead end of the rod.

The cylinder liner is of special cast iron; it is in two parts which abut above the scavenge ports, *i.e.* at the region of lowest temperature. For a 2-cycle engine the liner is simple. The top and bottom cylinder covers, to which the liner is bolted by long flexible bolts of 3 to 5 per cent. nickel steel having ferrules under the heads, are of chrome steel

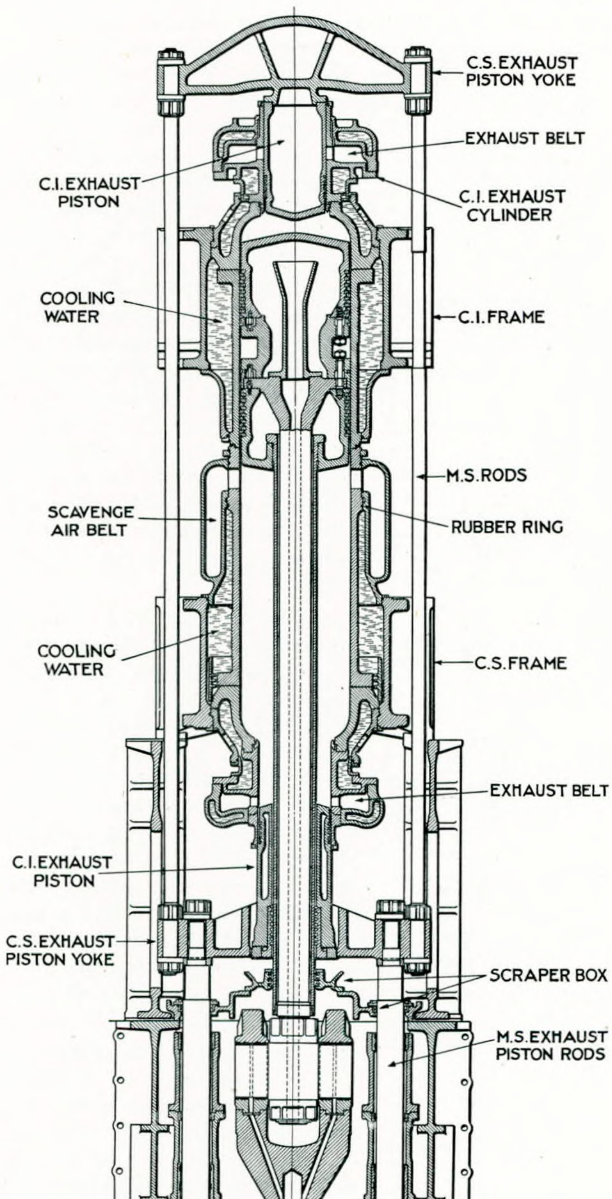


FIG. 4.—Fore and aft section of double-acting two-cycle engine.

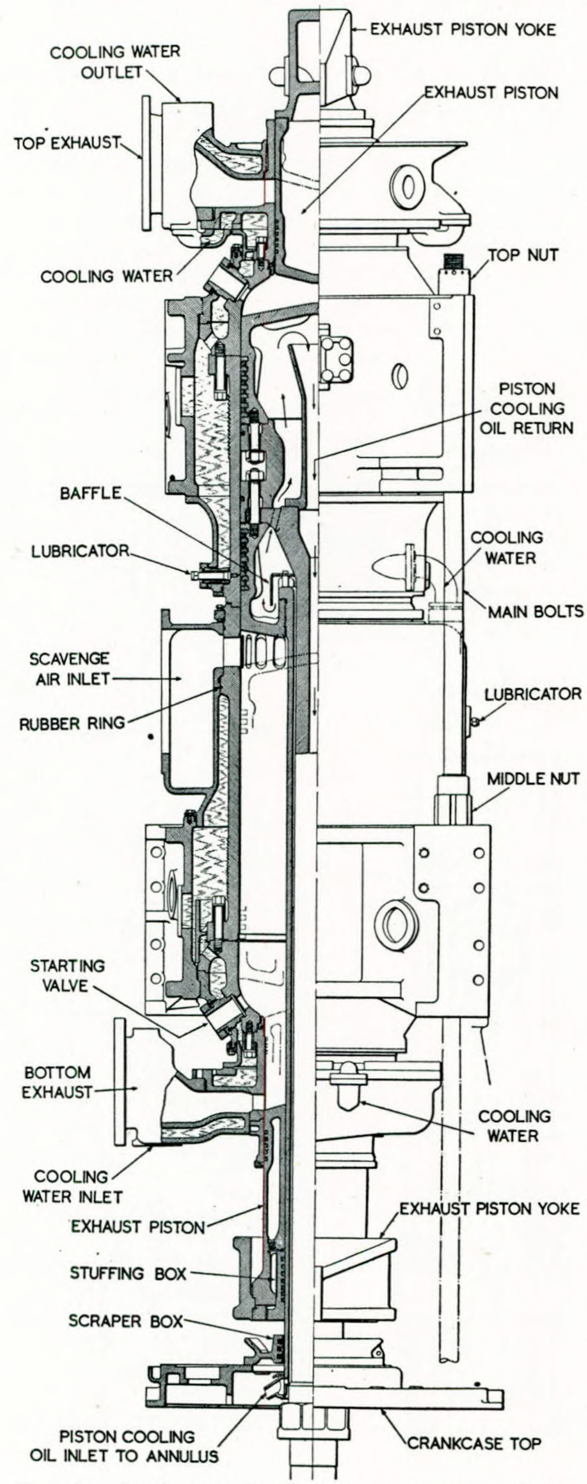


FIG. 5A.—Section of double-acting two-cycle cylinder.

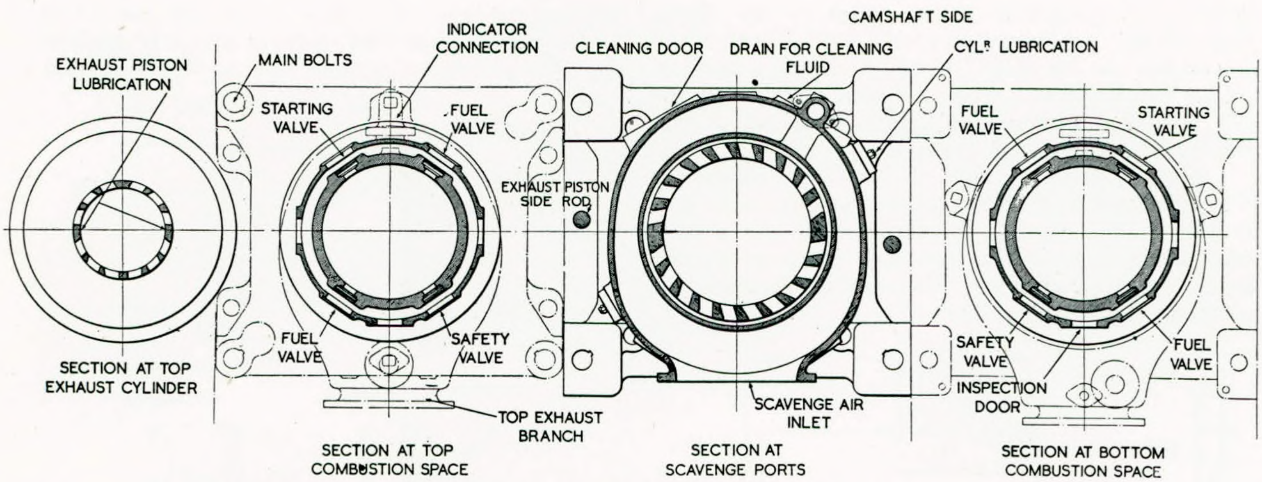


FIG. 5B.—Section through ports, etc., double-acting two-cycle cylinder.

or chrome-molybdenum steel. In Fig. 5A the cover jackets are shown separate, of cast iron, with steel sleeves screwed into the covers and suitably attached to the jackets for taking the fuel, starting and relief valves. Occasionally—if so required—the jacket is cast with the cover. The bottom cover is housed in a cast steel frame which rests upon the main framing; the top cover is surrounded by a frame of cast iron, the forces being smaller than in the bottom frame.

Castings bolted respectively to the top and bottom frames constitute the water jackets and scavenge belt.

Each complete cylinder, consisting of covers, liner and frames, is bound together by four long bolts, which continue through the main framing and terminate in nuts under the bedplate. The elasticity of the main bolts and the compressibility of the framing allow the expansion of the cylinder liner, etc. to be taken safely.

Fig. 5B shows a section at the scavenge air ports. Two wide bars are provided, to cover the piston ring joints. There are lubricating points above and below the scavenge ports; also at the exhaust cylinders.

The cylinder is fresh-water cooled from a closed circuit. The cooling of main and exhaust cylinders extends for the total travel of the respective piston rings.

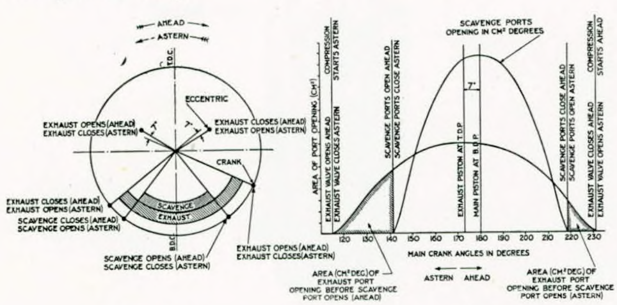


FIG. 6.—Scavenge and exhaust periods.

The main piston is oil-cooled. The oil is taken by telescopic pipe to the annular space between piston rod and sleeve, circulates through the lower piston, flows through holes in the piston rod flange to the upper piston and returns down a longitudinal hole in the piston rod, thence to a spout leading eventually to the suction tank. A baffle in the bottom piston assists cooling of the crown.

The exhaust pistons are also oil-cooled. The oil is fed to a point on the upper part of one of the exhaust piston-rod guides, passing into the rod through two radial holes, flowing up the rod and circulating through the lower exhaust piston. Ascending a vertical pipe clipped to the adjacent side rod it is then led to the top exhaust piston, from which it returns by a pipe clipped to the same side rod and led to the other exhaust piston rod, entering this just below the yoke and descending through a longitudinal hole to the upper half of its guide, passing out through radial holes and returning through a pipe to the suction tank. On the way to the suction tank the cooling oil from main and exhaust pistons is led to a divided observation glass at the foot of each frame.

The cooling oil for pistons and the lubricating oil for bearings is dealt with by a common system, supplied by the same pump, the oil pump discharge being divided in suitable ratio. The control is by one valve.

Lubricating oil is fed from the main pipe to each main bearing. The oil flows through pockets in the bearing sides, inwards through two radial holes in the journals—of such an included angle that one is always abreast of the pockets—to a longitudinal hole in the crankshaft, through holes in the crankweb to the crankpin and side pockets in the bottom end bushes, thence vertically through the connecting rod to the top end bushes, whence it finds its way out into the crankcase.

The principle underlying the design and construction of all the chief bearings is the same.

Pockets are provided at the sides of the bush; longitudinal grooves, scraped wedge-form and spaced not too far apart, divide the bearing surface. Oil is fed by holes, from a common inlet, to the side pockets and the longitudinal grooves. There are no circumferential grooves.

For the eccentric gear, each exhaust piston rod is supplied with oil fed to a point on the lower half of the guide, passing through radial holes to a longitudinal hole in the rod, then down to the top end bearing, continuing downward through the eccentric rod to the straps.

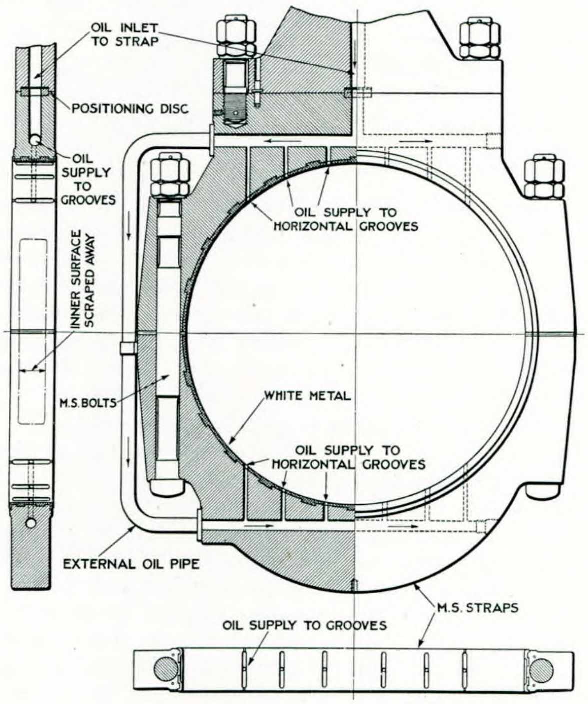


FIG. 7.—Lubrication of eccentrics.

Fig. 7 shows the method of lubricating the eccentric straps. The load on the eccentrics is always opposed to the piston pressure, thus tending to relieve the main bearings. The power obtained from the exhaust pistons is of the order of 10 per cent. The eccentric drive has been conspicuously successful in service.

The thrust block, if incorporated in the engine, is essentially a single, split, hollow casting faced with white-metal on both sides and in the bore, held in the thrust block casing. The shaft has two collars, *i.e.* one on each side of the block. There are numerous helical oil grooves in the bore—fed through holes from the hollow casting which acts as an oil reservoir—leading to radial grooves on the faces, slightly taper-scraped at the sides to induce and maintain a heavy oil film.

Past tendency in marine engineering has undoubtedly been to have too great a thickness of white metal; it had its origin in steam engines. Better adhesion, reduced liability to cracking and lower first cost all follow from the use of thin white metal of suitable mixture properly applied.

The piston-rod scraper box, as shown in Fig. 8,

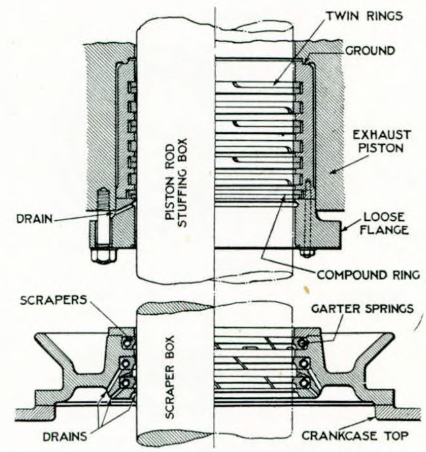


FIG. 8.—Stuffing box and scraper rings.

consists of three cast-iron segmental scraper rings with drains of ample size. This construction maintains a moist rod, without oil being taken up into the stuffing box to cause ring sticking. The piston rod stuffing box has six twin rings of the type used for many years for 4-cycle engines, and a bottom ring—to give the last degree of tightness—of the compound sealing type, *i.e.* a ring of square section inside a gnomon.

Starting valves are fitted to all cylinders, top and bottom, to obtain the quickest possible astern running when the ship is moving ahead. The first few seconds are the most important.

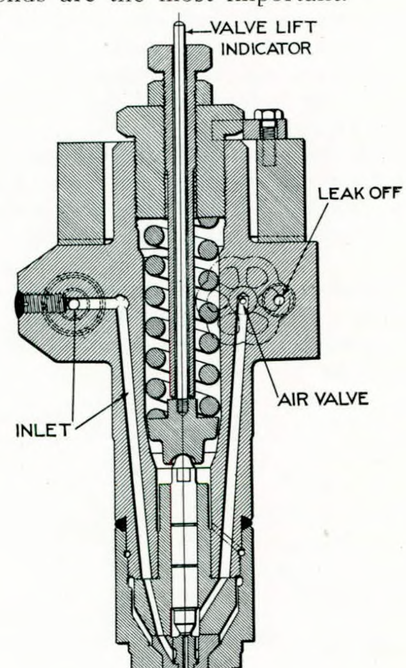


FIG. 9.—Fuel valve.

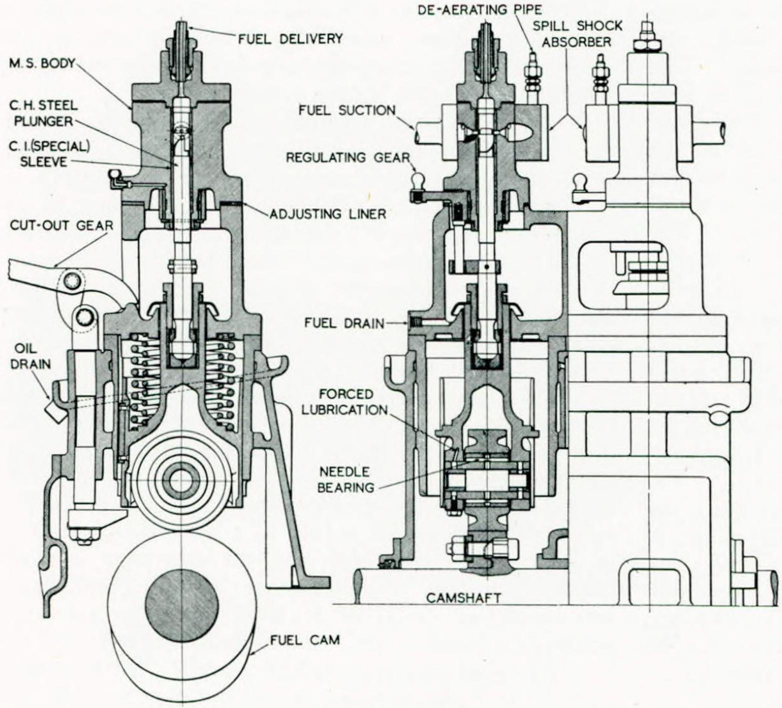


FIG. 10.—Fuel pump.

used for both ahead and astern running, there being a lost-motion clutch on the camshaft.

The fuel quantity is varied by altering the effective pump stroke—by rotating the plunger—the surplus fuel being after-spilled from the helical edge of the groove in the plunger. The regulating gear is hand and governor operated. Any plunger can be cut out while the engine is running. A spring shock-absorber, for levelling-out pulsations in the surplus fuel spill, is fitted opposite to the suction inlet. The pump suction side is automatically vented against air-lock.

Each fuel pump discharges to a distribution block, thence to the fuel valves. The filter for each valve is of the gauze design. There is no non-return valve between fuel pump and fuel valve.

The fuel is injected into the cylinder at high pressure at the appropriate moment, the fuel pump being operated by a cam. Immediately fuel injection has taken place, the fuel line is relieved of pressure until the next injection. A sharp and snappy cut-off of the fuel—which avoids after-burning—is important.

A pair of positive displacement blowers at the back of the engine supply scavenge air; one blower is shown in Fig. 11.

The rotors are of light cast iron, bolted to the shafts. The vane profile is so designed that machining is a minimum, and the fine clearances can be obtained without skilled hand work. The tips are cut away to avoid shock due to trapped air, but a

A fuel valve is shown in Fig. 9. Each top and bottom cylinder has two fuel valves; the sprays are partially directed towards the piston and in the same direction as the swirl of the scavenge air.

The fuel valves are automatic; they open somewhat below the injection pressure, which is 280 to 350 kg.cm.² (4,000 to 5,000lb. sq. in.). The oil speed through the nozzles is about 190 to 210 metres per second (625 to 690ft. per sec.). For a 620/1,400 engine there are three nozzle holes 0·037in. bore (top) and 0·0292in. bore (bottom). The nozzle hole length is about 3 diameters.

The positiveness of the opening and closing of the valve prevents dribbling, also carbon formation on the nozzle end. Further, the nozzle is cooled either by circulating the incoming fuel oil around it—the nozzle being provided with fins—or by circulating fuel oil from an independent circuit comprising pump and tank. The latter arrangement allows a lower grade of fuel oil to be used, a fact which may appeal to shipowners.

Fig. 10 shows a pair of fuel pumps. One pair serves a cylinder, *i.e.* one pump delivers to the two top fuel valves. the other to the two bottom valves. The cams are separately adjustable for injection timing. A single cam is

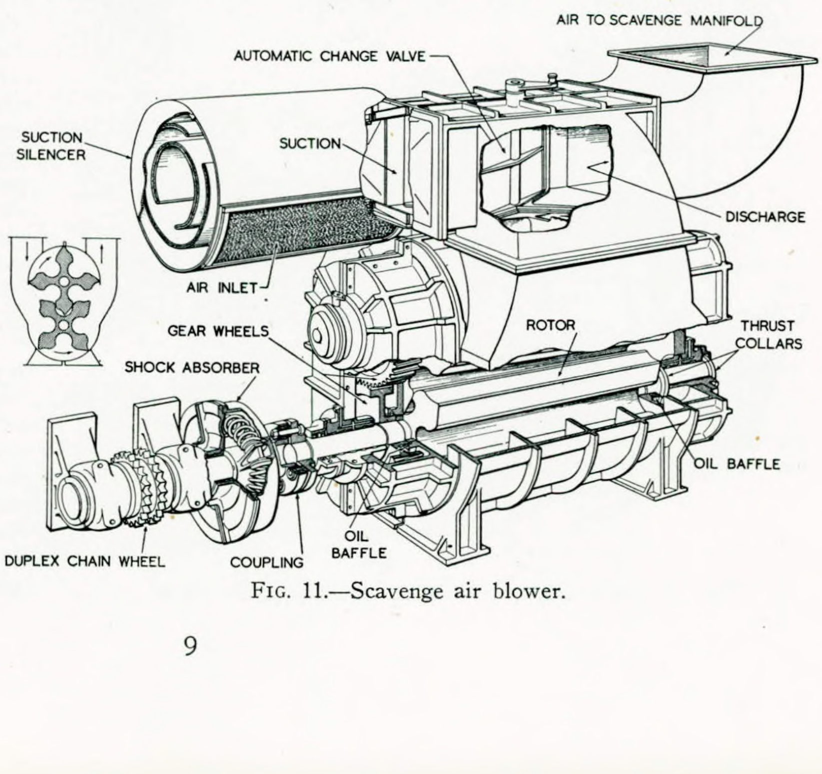


FIG. 11.—Scavenge air blower.

small amount of sealing air is retained between the two rotors. The rotor vanes do not touch; the clearance is about 0·5 mm. (20/1,000th ins.). The tip clearance is 0·75 mm. (30/1,000th ins.) or less, depending upon the rotor size. Meshing takes place at the steel gear wheels, arranged at one end of the rotors. The blower volumetric efficiency is 82 per cent. to 85 per cent. The scavenge pressure is about 0·10 to 0·15 kg.cm.^{2} (1·4 to 2·1lb. sq. in.). The blowers are designed for 0·24 kg.cm.^{2} (3·4lb. sq in.) pressure.

Thrust collars ensure minimum clearance between rotors and covers for avoidance of end leakage. Baffles and oil throwers in the covers prevent leakage of oil into the rotor casing and air into the end cover. The forced-lubrication system serves the blower bearings; there are oil sprayers at the gear wheels.

On reversal, the blower suction becomes the discharge and vice versa, by the automatic movement of a flap valve.

The blower rotates at 2·5 to 4 times engine revolutions and is chain-driven. A shock-absorber, of angular movement 20°, is interposed between the chain drive and the rotors; the rubber ferrules of the blower coupling contribute to further flexibility. On small engines, where the blower is spur-wheel driven from the crankshaft, a flexible gearwheel is substituted for the spring coupling.

The blower casing and covers being joined vertically down the middle, the outer parts may be taken away without disturbing the moving parts.

The galvanised steel, felt-lined, silencer is of the interference type. Air flows through the ducts to the centre, passes to the change-over valve which deflects it to the suction side of the blower casing, thence through the blower to the other side of the change-over valve, and so to the scavenge air manifold and the cylinder.

The fuel pump camshaft, the scavenge air blower and the engine-driven pumps—if any—are driven by separate chains from the crankshaft. For the fuel pump camshaft chain, assuming blow-off pressure on the fuel pump relief valve, *i.e.* 10,000lb. sq. in., the factor of safety is a minimum of 30—in service at least 50: for the scavenge blower chain, assuming severe starting torque conditions, the factor of safety is at least 25. The polygon action of chains is fully taken into account.

The governor is of the Harland-B. & W. twin-pawled acceleration type, capable of adjustment during running, especially of the cut-out pawl for earlier cut-out during heavy seas; it is driven by eccentric from the camshaft chain guide-wheel. From the latter is also driven, by a small chain, the air distributor shaft which in turn is utilised for operating the cylinder lubricators.

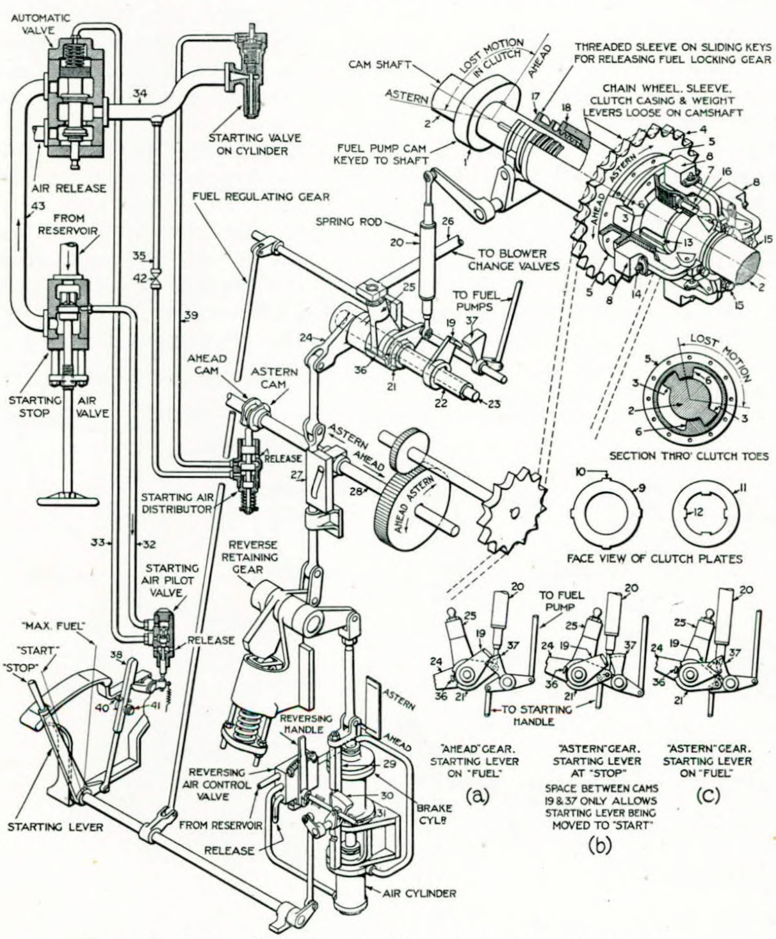


FIG. 12.—Reversing gear, double-acting two-cycle engine.

The starting and reversing gear is shown in Fig. 12. Fuel cam 1 is keyed to camshaft 2, on which there are two clutch toes 3—opposite each other—formed solid with the camshaft. Chainwheel 4, loose on the camshaft, carries a lost-motion clutch ring 5, with two toes 6. These toes are of such size that when one side engages camshaft toe 3. as shown, fuel cam 1 is in phase with the crankshaft for ahead running; when chainwheel 4 and clutch ring 5 are rotated until toes 6 engage the opposite side of toes 3, the fuel cam is in phase for astern running.

To prevent chatter the lost-motion coupling 5 has bolted to it a friction plate arrangement. This comprises casing 7, which carries weight levers 8 and alternate friction plates 9—having projections 10 engaging slots in 7, also plates 11—which have projections 12 engaging slots 13 in 2. The movement of 8 is limited by studs 14, also by springs inside 8 so adjusted that no force is exerted upon the clutch plates when the engine is turned slowly. This allows the chainwheel to move freely on the camshaft, to take up the lost-motion when reversing. As the engine gathers speed, the centrifugal force of 8 overcomes the spring on 14 and exerts pressure on adjusting pins 15 which is transmitted through pins 16 to the clutch plates. The lost-motion rotation of 4 carries with it a nut 18 on the chainwheel sleeve. Rotation of 18 slides threaded sleeve 17 along splined camshaft, thereby moving 20. 21 and sleeve 22—which is free on shaft 23—and operating fuel locking cam 19.

The air cylinder, acting through the reverse retaining gear and 24, 23, 25, 26, operates the blower change valves, held firmly against their faces by spring rods, not shown—and the retaining gear provides the re-action.

A slot in plate 27 moves shaft 28, rotating at engine speed, which carries ahead and astern distributor cams. By reason of differential pistons on the distributor valves, these make contact with the cams only so long as starting air is on the distributor, the springs holding off the valves at other times.

Reversing air and brake cylinder piston rod moves locking bar 29 which has two slots so placed that, when full-up or full-down, one is opposite to cam 30, which is operated by starting lever. This prevents starting air being put on the engine before the air cylinder piston has completed its stroke. The movement of the starting lever which operates cam 30 also operates pin 31; this locks the reversing handle, except when the starting lever is at stop.

Fig. 12 shows the starting lever at "Stop", engine in ahead gear, and fuel locking cam 19 as at (*a*). The starting air stop valve is closed. In opening this valve, before pressure can reach the automatic valve by pipe 43, the air passes through 32 to the pilot valve and 33 to the automatic valve top chamber keeping it closed. Starting air cannot, therefore, reach 34 or 35.

To reverse the engine: bring starting lever to "Stop", move handle on reversing air control valve to "Astern" and hold it there. This supplies air to the air cylinder bottom. When the pointer comes to "Astern" on the index plate, the reversing handle is moved to the vertical or neutral position; otherwise when attempting to bring starting lever to "Start", locking pin 31 will come against reversing handle boss instead of entering the hole. The astern cam is now over the air distributor valve, the blower change valves are reversed, and key 36 on lever 25, acting on lever 21, has extended spring rod 20 somewhat, bringing cam 19 into position shown at (*b*).

The space between regulating lever 37 and cam 19 permits the starting lever being moved to "Start", cam 30 travelling into slot in bar 29, and pin 31 into hole in reversing handle. Draw-bar 38 lifts the air pilot valve spindle, releasing the pressure from the automatic valve top chamber, which automatic valve thereupon opens and starting air enters 34 and 35, forcing the air distributor valves against the cams. The engine crankshaft begins to move astern; chainwheel 4 with its clutch gear, all driven from the crankshaft, takes up the lost motion—about one-third of a revolution—until clutch toes 3 and 6 come into contact for astern

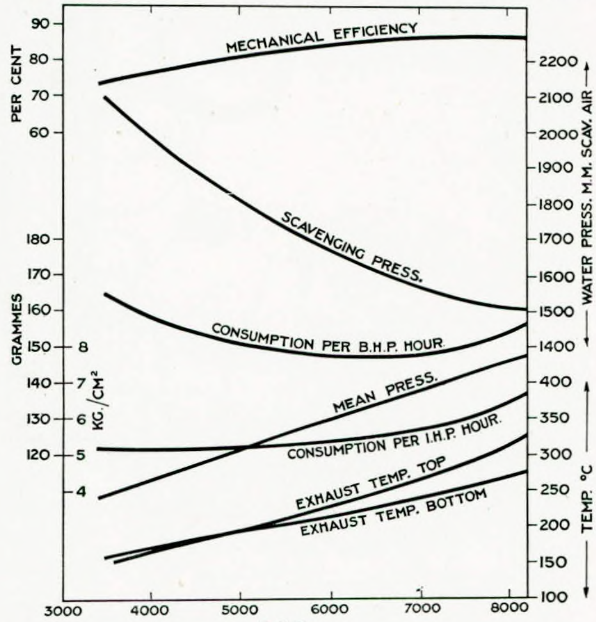


FIG. 13.—Test-bed results, 7,000-b.h.p. double-acting two-cycle engine.

running. During this time 18 moves 17 towards 1, relieving the tension in 20 and moving 19 to position shown at (c), releasing 37 and so allowing starting lever to be moved past "Start" to "Fuel". As soon as starting lever is moved past "Start" pin 40 engages roller 41 and trips the air pilot valve, thus closing the automatic valve and allowing the air in 34 and 35 to escape.

Valve 42 is closed only for testing starting valves.

Fig. 13 shows the test-bed results of a 6-cylinder 620/1,400 engine, obtained in the shops of Messrs. Burmeister & Wain. At the designed rating of 7,000 b.h.p., 105 r.p.m.. the indicated mean pressure is 6·75 kg.cm.² (96lb. sq. in.) based on the total stroke volume of main and exhaust cylinders; the fuel consumption is 0·326lb. per b.h.p. hour (148 grammes). The bottom mean pressure can be expected to be about 0·25 kg.cm.² (3·5lb. sq. in.) less than the top mean pressure. Where mean pressures are mentioned, the mean of the top and bottom mean pressures is intended.

The double-acting engines shown in Figs. 3 *et. seq.* have been fitted in the last few years to a total of 1,000,000 s.h.p. It is only to be expected that so firmly established a design will remain standard for years to come. Whilst most shipowners

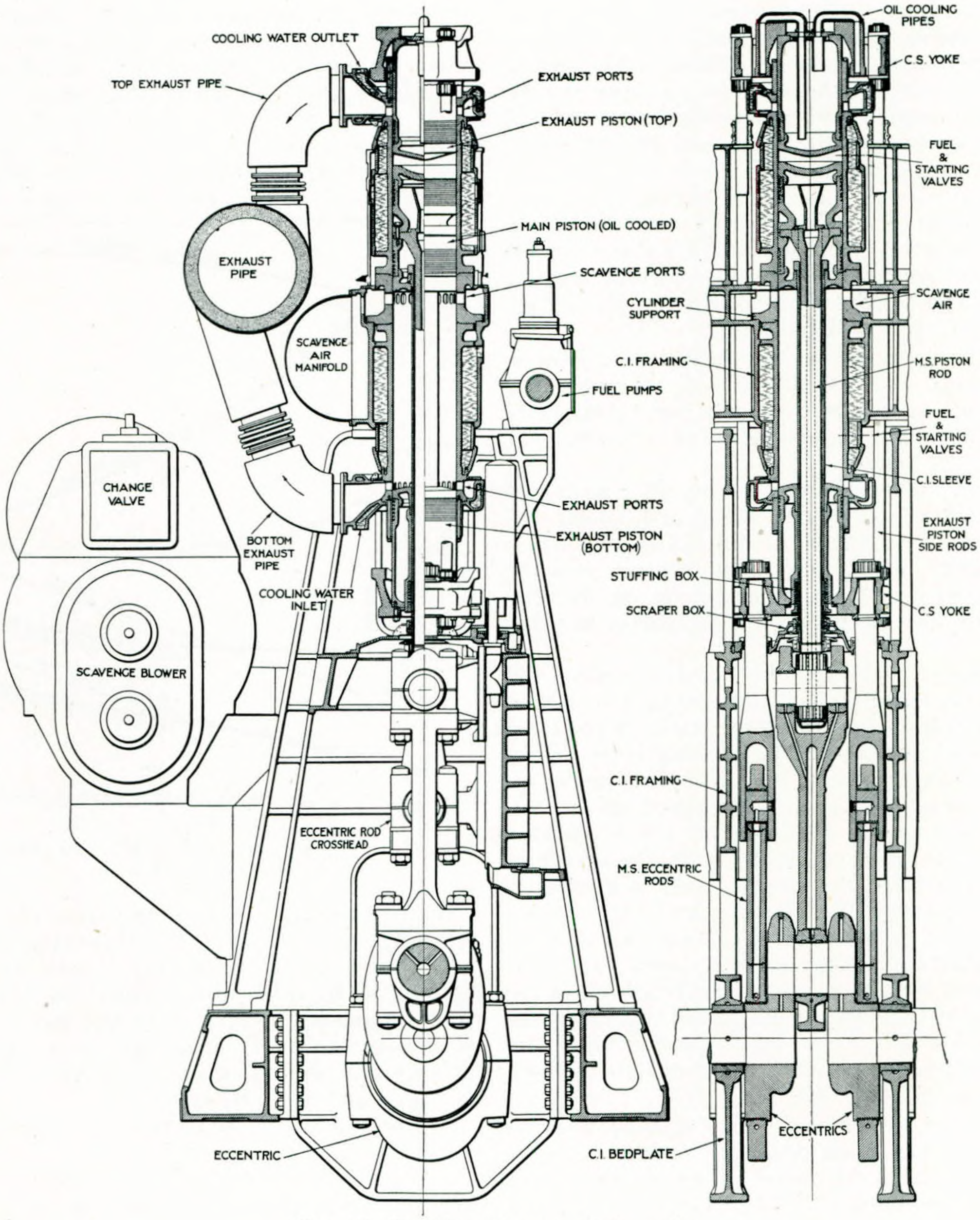


FIG. 14.—Double-acting two-cycle engine.

are always anxious to have machinery that has stood the test of time, there are others who are quite ready to play their part in the utilisation of still more progressive designs. For such there is available the design shown in Fig. 14 in which all the characteristic features of the standard type have been retained. As a result of the very successful experience with the eccentric drive, the exhaust piston valves have been increased in diameter to that of the main piston, the short travel of the valves remaining as before. Such owners as those referred to above will be interested to know that an engine on these lines has been designed and developed by Messrs. Burmeister & Wain, and Messrs. Harland & Wolff, Ltd., in the closest collaboration, and a single-cylinder unit has been built and is running at the works of Burmeister & Wain, Copenhagen. This particular unit develops 1,000 b.h.p. at 125 r.p.m. It is somewhat lighter in weight and occupies rather less space than the present types.

(b) Single-acting Two-cycle Engines.

Fig. 15 shows an engine 500 mm. (19·69in.) bore 900 mm. (35·43in.) stroke. Such an engine may develop up to 400 b.h.p. per cylinder.

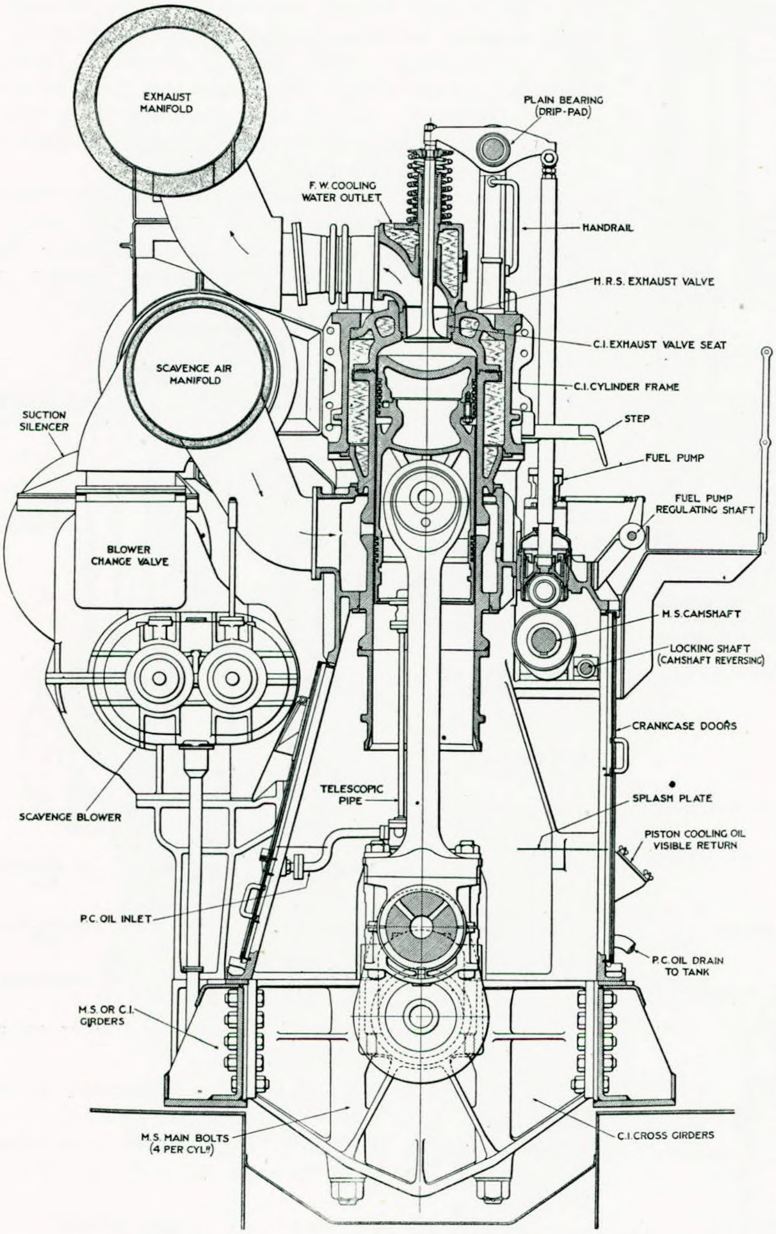


FIG. 15.—Section of single-acting two-cycle engine.

The general design follows the double-acting engine. The crankshaft is fully-built for 5, 6 and 8 cylinders, semi-built for 10 cylinders. The piston is a special steel crown bolted or screwed to a cast-iron skirt. The piston is oil-cooled. The gudgeon pin—shown in Fig. 16—is self-explanatory. The cylinder cover is of special steel, to which the liner is securely bolted. There are two fuel valves per cylinder.

Scavenge air is admitted at the bottom of the stroke by the piston uncovering ports in the liner; the exhaust gases are expelled through a poppet valve centrally arranged in the cylinder cover. The approximate periods are: exhaust opens 70° before bottom centre and closes 45° after; scavenge opens 32° before and closes 32° after.

The camshaft, which operates the fuel and exhaust valves, rotates at crankshaft revolutions and is spur-wheel driven from the crankshaft. There is one fuel pump per cylinder, arranged at the crankcase top.

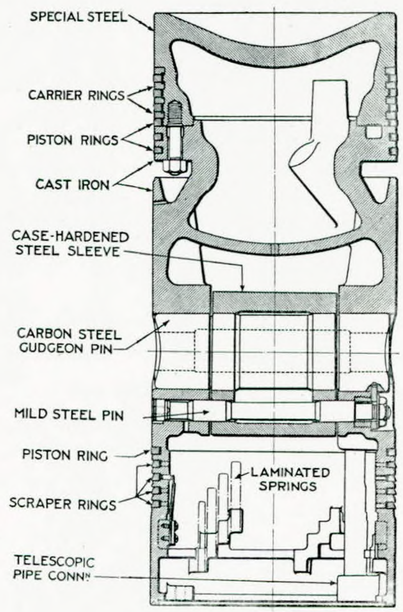


FIG. 16.—Piston.

The cylinder, cover and exhaust valve are fresh-water cooled from a closed circuit. The lubricating oil system is similar to that for the double-acting engine. For single-acting engines the bearing pressure is in one direction; for effective lubrication the requirements are thus not unlike those for thrust blocks. In the double-acting engine the pressure acts alternately upwards and downwards. Some engineers advocate a dual circuit for lubricating oil, *i.e.* the separation of piston cooling and lubricating systems. Years ago this matter was investigated in collaboration with oil experts and the present system was established. More effective centrifuging should be sought where there is any difficulty with asphaltic substances.

There are a certain number of people who dislike any kind of trunk engine for propelling purposes. The dislike seems to be centred not in using the piston skirt as a guide but in the gudgeon pins, the argument being that after gudgeon pins have been removed once or twice they cease to be tight. All this must depend, of course, upon the design and manufacture of the gudgeon pins. There is also the question of lubricating oil consumption.

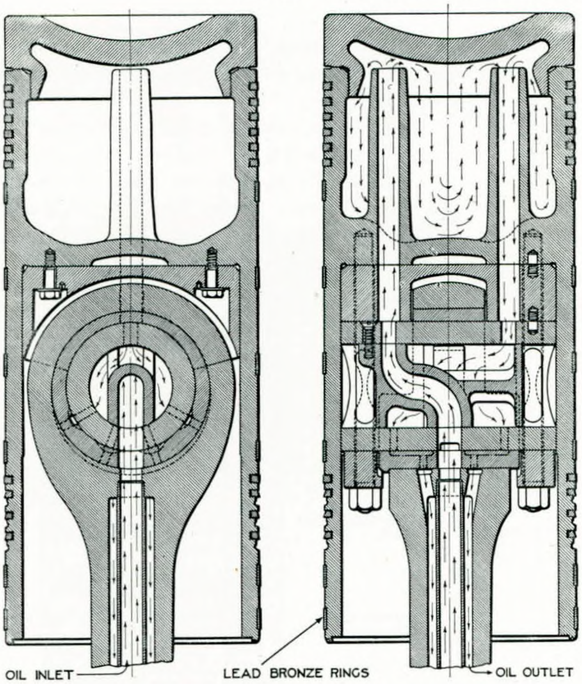


FIG. 17.—Alternative design of piston.

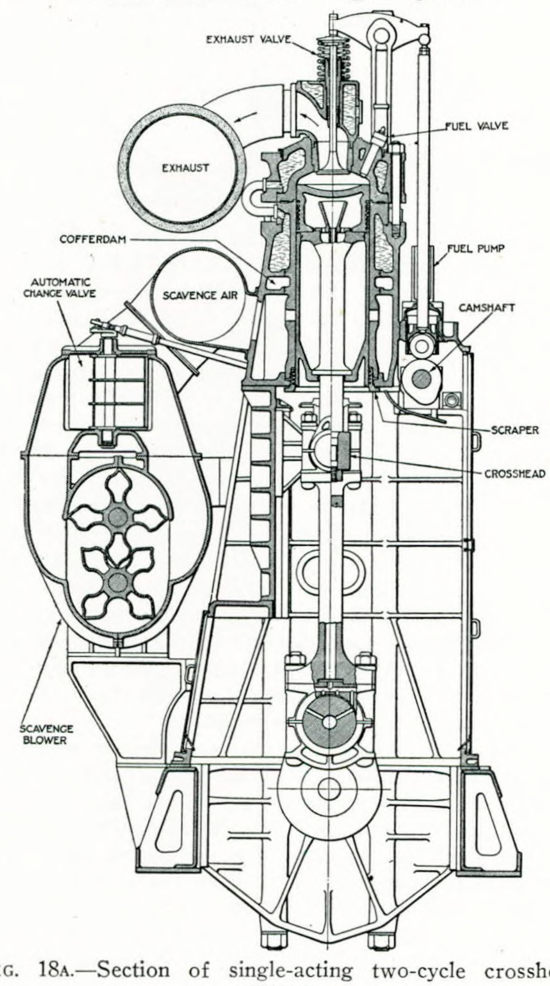


FIG. 18A.—Section of single-acting two-cycle crosshead engine.

Continental owners appear to have favoured trunk engines hitherto, judging by the number put into service every year. Fig. 17 shows a patent oil-cooled piston which overcomes the chief objection to gudgeon pins; it also dispenses with telescopic pipes. The experimental experience with it to date is very satisfactory.

For engineers who dislike gudgeon pins there is available the engine indicated in Fig. 18A, which incorporates a crosshead in a design otherwise the same as Fig. 15. In Fig. 18B another design is shown, in which the crankcase is separated from the cylinder, as in a normal crosshead design. The crosshead engine is slightly higher and somewhat more costly; otherwise there is certainly no objection to it.

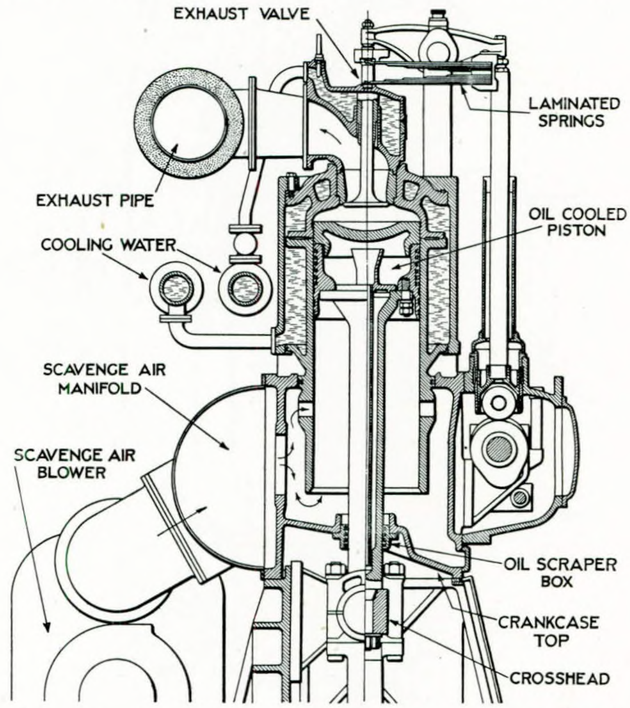


FIG. 18B.—Alternative single-acting two-cycle crosshead engine.

The initial pressure for single-acting and double-acting engines is not more than 49 kg.cm.^{2} (700lb. sq. in.). The relief valves are set to 55 kg.cm.^{2} (780lb. sq. in.). The starting air reservoirs are pumped to 25 kg.cm.^{2} (355lb. sq. in.).

As many people do not seem to be clear regarding the construction of the reversing gear of this type of engine, a fairly full description is given below. The description is somewhat tedious, although the manipulation of the gear is simple enough. See Fig. 19.

Exhaust cams 4 are keyed to camshaft 3 which drives, through a dog clutch, fuel cams 1—for two cylinders together—fixed to sleeve 2. Clutch toes 5 and 6, to avoid chatter, have rockers with fulcra and springs on exhaust cam boss and rollers pressing on inclined faces on sleeve 2. Between ahead and astern positions there is 130° lost motion. Except at (*e*) and (*f*), Fig. 19 shows the gear in ahead position. Split wheel 7, driven from the crankshaft, runs free on yoke 8, keyed to camshaft 3. Two cranks 9 connect 7 and 8 so that 8 always leads 7 by 10°. On 9 are mounted gear wheels 10, meshing with wheels 11 on fuel sleeves 2 and reversing sleeve 12. While 7 moves 110° astern—wheels 11 being held by brake wheel 16—10 moves 180°, which by action of 9 and slide blocks 13 brings 8 in advance of 7 in astern direction by 10°. Toes 5 and 6 arrest further movement of 10. Cranks 9 have now assumed such position that pressure between 8 and 7 produces no turning effort on them.

At (*a*) the full lines show cams relative to crank at top centre, for ahead. If crankshaft be rotated 360° astern, while 7 rotates 110°, 8 moves from 10° behind 7 to 10° in front, *i.e.* 130°; hence exhaust cam becomes as shown dotted for astern. Similarly for fuel cam, while 7 rotates 110°, 8 and 3 rotate 130° before 5 and 6 engage. To complete the revolution, crankshaft also 7 further rotate 250°, fuel cam changing from full line for ahead to dotted line for astern. To prevent rotation of sleeves 2 during 130° lost-motion, each sleeve has spur wheel 11 meshing with pinion 14 keyed to manœuvring shaft 15. Reversing sleeve 12 has keyed to it brake-wheel 16 also spur wheel meshing with 14 on shaft 15. When engine is running, 10 and 11 have no relative movement; 14 and 15 revolve idly.

Brake cylinder 17 is inoperative except during reversal.

There is 130° lost-motion between toes 20 and 21, respectively on 12 and threaded sleeve 22, keyed to 3. Nut 23 in halves engages 22; 23 is bolted to grooved ring 24 sliding on keys 25 in sleeve 12 and operates 26 and 27 through 28, 29 and 30. Blower change valves are held firmly against their faces by spring rods (omitted). Crank 30 comes to dead centres for ahead and astern, locking the spring rods and relieving 22 and 23 of load; 30 also trips brake pilot valve 38. Reversing lever 31, loose on 32, operates 33 through links 34 and lever 35, keyed to 33. Reverse locking sector 36 is keyed to 33. Fuel locking sector 37, loose on 33, is operated both by 31 and 27. See (*d*), (*e*) and (*f*).

The distributor camshaft rotates at engine revolutions. Lever 31 operates 38 and distributor cams 39 through gear shown. Owing to the differential pistons acting against springs, valves 40 only contact the cams when air is on the distributor.

Starting lever 41, keyed to 32, operates valve 43 through draw-bar 42; also rod 44. Double-sided locking sector 45, keyed to 32, moves with 41. The side of 45, remote from 31, is the longer. and acts as a reverse lock, being clear of slots in 36 only when 41 is at stop. Sector 36 prevents 41 being moved, unless 31 is at ahead or astern. The shorter side of 45 engaging fuel locking sector 37, only permits 41 being moved to "Start" —see (*e*)—until the engine has taken up the lost motion on camshaft, when slot in 37 comes opposite 45 and lever can be pushed to fuel—see (*f*). On opening starting air stop valve, before pressure can reach the automatic valve by pipe 46, air passes to the top of the automatic valve by pipe 47, valve 43 and pipe 48, thus preventing air reaching 49 or 50.

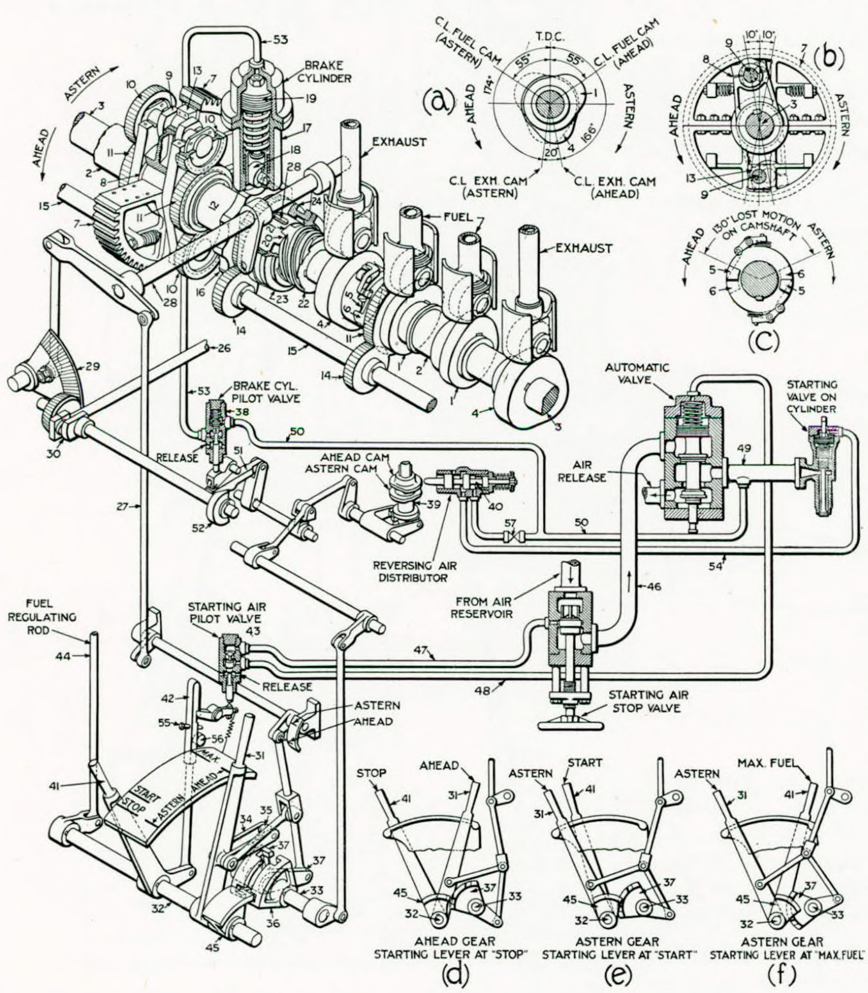


Fig. 19.—Reversing gear, single-acting two-cycle engine.

To reverse the engine: 41 is brought to "Stop" and 31 to "Astern". Sector 36 moves anti-clockwise until slot is opposite 45; sector 37 moves from (*d*) to (*e*)—where there is no slot opposite 45. Movement of 31 astern moves the astern distributor cams opposite to 40; projection on wedge-bar 51 mounts trip disc 52 and opens pilot valve 38.

Lever 41 is now moved to "Start"—the momentary limiting position because of sector 37; see (*e*). Drawbar 42 lifts valve 43, releasing pressure from automatic valve top chamber; the valve opens, air enters 49, 50 and 53, air distributor valves 40 are brought against the cams, and piston 19 in brake cylinder is forced down bringing roller 18 on to brake wheel 16 thus preventing rotation of all parts except those keyed to camshaft 3. Starting air passes by 54 to starting valves opening same to pipe 49; the engine moves astern and takes up lost-motion between clutch toes; screw on sleeve 22 moves nut 23, also 28, 29 and 30, reversing the blower change valves by rod 26. Rod 27 moves up, bringing 37 into position (*f*). With the completion of 130° lost-motion between clutch toes 5—6. 20—21, trip wheel 52 closes valve 38, air above piston 19 escapes and roller 18 is lifted from the brake wheel.

Lever 41 is now moved to "Fuel"; pin 55 contacting with roller 56 trips valve 43, closing automatic valve. Maximum fuel position is shown at (*f*).

Valve 57 is used only for starting valve testing.

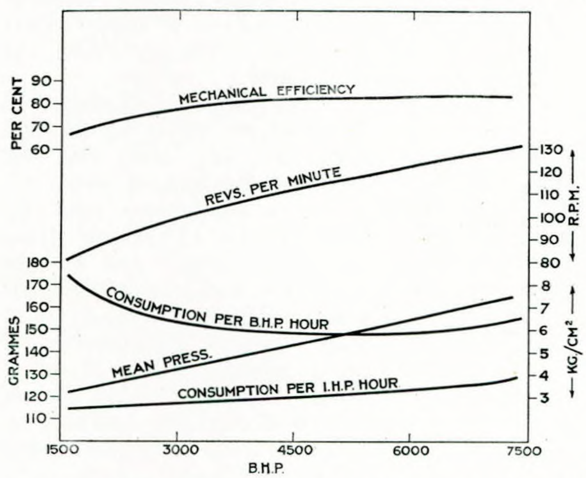


FIG. 20.—Test-bed results, 5,750-b.h.p. single-acting two-cycle engine.

Fig. 20 shows the fuel consumption curves of a 12-cylinder 620 mm. (24·41ins.) bore, 1,150 mm. (45·28ins.) stroke trunk engine, 5,750 b.h.p., 122 r.p.m., recently tested in the shops of Messrs. Burmeister & Wain. The maximum power is 8,100 b.h.p. at 135 r.p.m., 8 kg.cm.² (114lb. sq. in.) m.i.p.

(c) Single-acting Four-cycle Engines, supercharged.

Supercharging increases the cylinder pressure, and therefore the engine power, without increasing the temperature. A safe figure for continuous supercharge is 9 kg.cm.² (128lb. sq. in.) m.i.p.; 8·5 kg.cm.² (120lb. sq. in.) m.i.p. is moderate; these are in comparison with 6·25 kg.cm.² (89lb. sq. in.) for a non-supercharged engine.

Fig. 21 shows an engine 740 mm. (29·13in.) bore, 1,500 mm. (59·00in.) stroke, in 6 and 8 cylinders, with under-piston supercharging. The scheme consists in enclosing the space between cylinder end and crankcase top, and utilising the undersides of the pistons as air pumps. Fig. 21 is self-explanatory.

There is a valve chest abreast of each cylinder

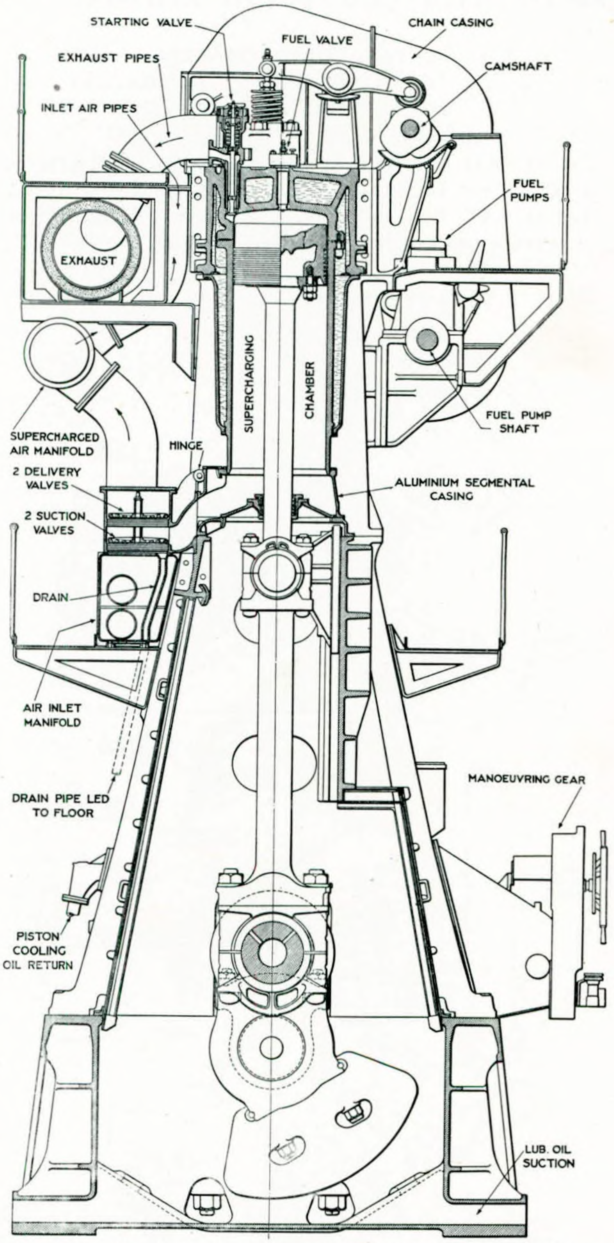


FIG. 21.—Section of single-acting four-cycle supercharged engine.

containing two suction and two discharge valves of Hoerbiger type. The weight of the chest on the inlet manifold satisfactorily maintains tightness. The branch to the supercharging chamber is bolted and hinged for ready overhaul. Internal divergent nozzles quieten the air suction, and baffles ensure that each valve chest receives its quantum of air. Inspection doors are provided, and safety discs are fitted to the supercharging manifold.

The amount of air delivered every downstroke, under the piston, is in excess of the suction requirements every alternate downstroke, above the piston. Adjustable throttle valves between adjacent valve chests enable the excess air to be pumped from one cylinder to another without compression. The supercharged air pressure is usually 0·25 to 0·33 kg.cm.^{2} (3·5 to 4·75lb. sq. in.).

The piston-rod stuffing box consists of a lower box of three segmental cast-iron scraper rings held by garter springs, and an upper box of two scraper rings in four parts, the segments having ground ends and being held by garter springs. The two groups of rings are separated by a small cofferdam having a drain pipe, which indicates leakage of oil or air.

A certain degree of scavenging is obtained by the exhaust and inlet valves being simultaneously open for a period. The scavenging overlap is less than with the Büchi system. Table I is a comparison of engines having connecting rod/crank ratio of 4. All starting valves open about 14° after top centre and close at 138°.

Under-piston supercharging is very simple and works remarkably well. A danger which used to be anticipated was that sparks passing the piston would cause explosions in the presence of such carbonised oil as may be found in the supercharging chamber. These fears have proved groundless.

In the widely-used Büchi pressure-induction system use is made of the heat energy in the exhaust gases. The cylinder exhaust pipes are subdivided and coupled to an exhaust turbo-blower in such a way that the impulses in the gases are used to the greatest advantage. The pressure waves are followed by depressions, which facilitate scavenging. In a 10-cylinder engine, for example, the turbine is served by a group of four exhaust pipes, respectively led from cylinders 1, 2; 9, 10; 3, 6, 7; 4, 5, 8; where the firing order is 1, 4, 3, 2, 5, 10, 7, 8, 9, 6.

Fig. 22 shows a single-stage turbo-blower suitable for a 10-cylinder engine, developing 6,000

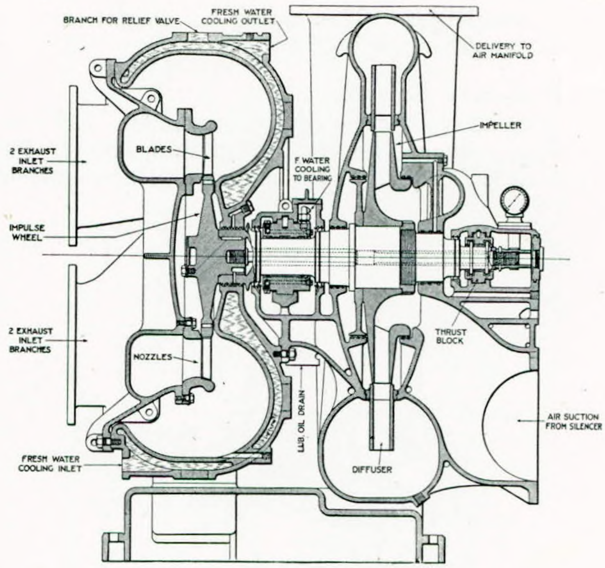


Fig. 22.—Exhaust turbo-blower.

s.h.p. at 115 r.p.m., with 8·25 kg.cm.^{2} (117lb. sq. in.) m.i.p. The exhaust gas quantity is 640 kg. (1,411lb.) per minute at 1·23 kg.cm.^{2} abs. (17·49lb. sq. in.) pressure at turbine inlet, 1·04 kg.cm.^{2} abs. (14·79lb. sq. in.) at turbine outlet; the air quantity is 538m^3 (19,000 cub. ft.) per minute, 1·30 kg.cm.^{2} abs. (18·49lb. sq. in.) pressure. The gas temperature at exhaust valves is 390° C. (734° F.), with 450° C. (842° F.) at turbine inlet. The blower rotates at 4,600 r.p.m.

The turbo-blower delivers air to a common manifold, from which pipes are led to each inlet valve. The inlet and exhaust valve opening periods are arranged to overlap, the combustion chamber thus being completely scavenged, the air pressure exceeding the average exhaust pressure above one-third of full load; about 30 per cent. of the supplied air is utilised for scavenging, which has a marked cooling effect upon the piston crown, cylinder cover and exhaust valves.

Exhaust blowers adapt themselves automatically to changes of load. The fuel consumption and exhaust temperature during overload conditions do not rise to the same extent as in unsupercharged engines, the weight of air increasing with the load.

For a supercharged engine the pressure parts are proportionately stronger, *e.g.* crankshaft, bedplate cross-girder, crosshead pins, long bolts, piston and connecting rods. The respective bearing sur-

TABLE I.

TYPICAL VALVE-OPENING PERIODS.

Engine.	*Inlet valve*			*Exhaust valve*		
	Opens.	*Closes.*	*Lift.*	*Opens.*	*Closes.*	*Lift.*
Unsupercharged	21° B.T.	21° A.B.	62 mm.	41° B.B.	15° A.T.	62 mm.
Büchi supercharged	70° B.T.	25° A.B.	76 mm.	20° B.B.	55° A.T.	76 mm.
Under-piston supercharged	53° B.T.	29° A.B.	76 mm.	45° B.B.	37° A.T.	76 mm.

B.T.=before top. A.T.=after top. B.B.=before bottom. A.B.=after bottom.

faces are larger. The saving in weight with a supercharged engine is 25 per cent.

(d) Auxiliary Engines.

Fig. 23 is a 4-cycle engine, 6 cylinders, 330 mm. (12·99in.) bore, 580 mm. (22·84in.) stroke, 500 b.h.p. at 300 r.p.m. This engine has been widely

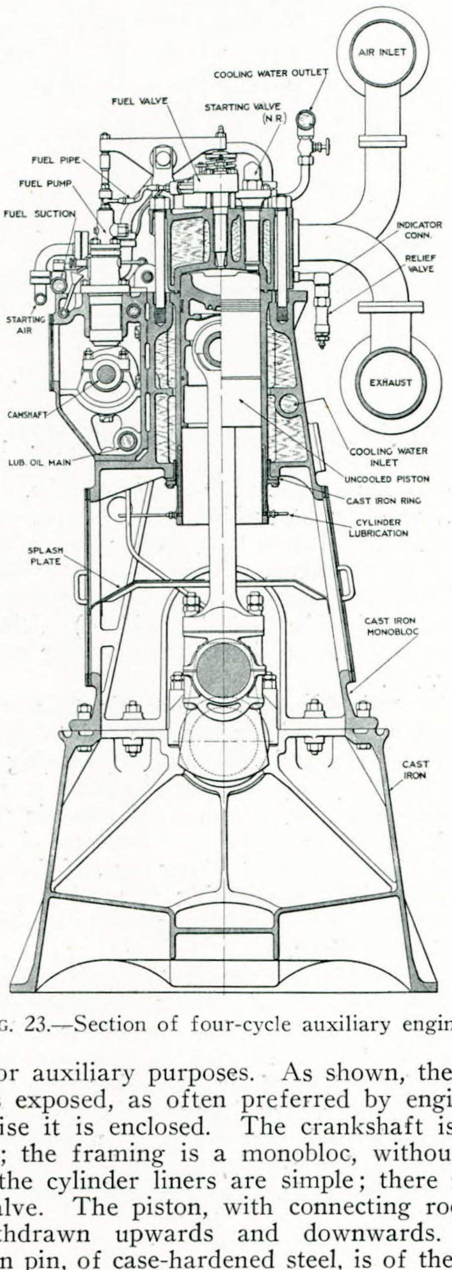


FIG. 23.—Section of four-cycle auxiliary engine.

used for auxiliary purposes. As shown, the valve gear is exposed, as often preferred by engineers; otherwise it is enclosed. The crankshaft is solid forged; the framing is a monobloc, without long bolts; the cylinder liners are simple; there is one fuel valve. The piston, with connecting rod, can be withdrawn upwards and downwards. The gudgeon pin, of case-hardened steel, is of the floating type. The piston rings comprise one simple top ring and three compound sealing rings—see Fig. 27 (d); there is a cast-iron scraper ring below the gudgeon pin. The piston is not cooled.

The camshaft, which is chain and spurwheel-driven from the crankshaft, is housed in a separate casting bolted to the monobloc, an arrangement facilitating economical and accurate manufacture. Fuel pumps and valves follow the principles of design outlined earlier. The inlet valves are generally similar to the exhaust valves but there is no water cooling and the valves are simpler. Starting

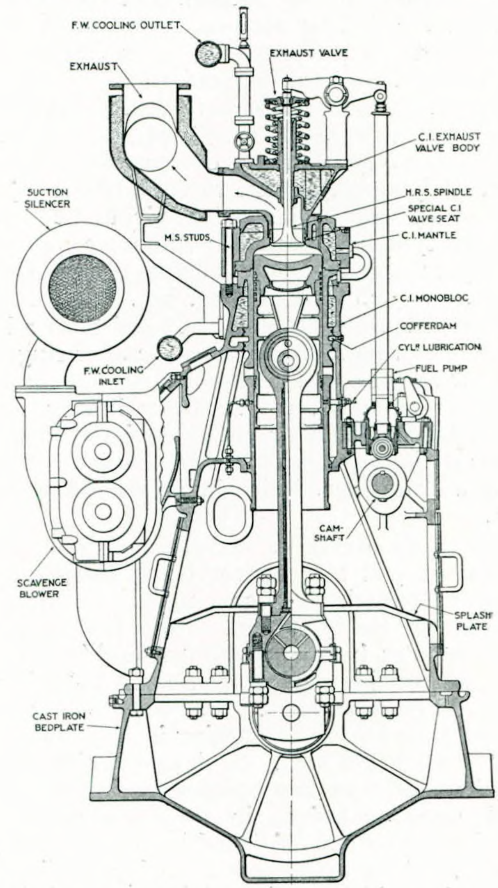


FIG. 24.—Section of two-cycle auxiliary engine.

valves are fitted to all cylinders. The pumps on the engine include forced lubrication and fuel surcharging pumps.

The next examples are two-cycle engines. Fig. 24, a 280 mm. (11·02in.) bore, 500 mm. (19·69in.) stroke engine is typical of the larger and more moderately-rated auxiliaries up to say 1,000 b.h.p. Fig. 25 is representative of smaller, faster-running machines.

The crankshafts are respectively semi-built and solid-forged steel. There are no long bolts. The piston crown is of heat-resisting steel; the pistons are uncooled. For the smaller engines the gudgeon pin is of the floating type, case-hardened nickel steel. The gudgeon pin in Fig. 24 follows the main engine design, *e.g.* Fig. 16.

Piston rings are of anchored ramsbottom type, with diagonal cut; scraper rings are variously of normalised tool steel and slotted cast iron.

For the smaller engines, cylinder liner and cover are of special cast iron, in one piece; for the larger units the cover is separate, of heat-resisting steel. There are two fuel valves per cylinder.

The starting valves—fitted to all cylinders—are non-return, operated by air pressure controlled by cam-operated valves in a distributor.

Scavenge blower and camshaft are spur-wheel driven; in the high-speed engines the gears are hardened and ground. Rotors of small blowers are steel; in larger engines they are cast iron. Vane clearance is about 0·1 mm. (4/1,000th in.); meshing takes place at the gears.

All parts are forced lubricated except the exhaust lever needle bearings, which are grease-packed. The exhaust pipe is of welded steel, lagged, with adequate expansion joints. Each engine is complete with pumps, filters, cooler, etc.

In Fig. 24, which is typical, the exhaust valve opens at 75 per cent. stroke and closes 8 per cent. after bottom dead point; the inlet port opens 18 per cent. later and closes at the same time as the exhaust valve. In Fig. 23, a 4-cycle engine, the exhaust valve opens at 86 per cent. of the firing stroke and remains open until 3 per cent. after top centre; the inlet valve opens 4 per cent. before top centre and closes 2 per cent. after bottom centre.

Bottom end bolts of all auxiliary engines are 3 per cent. nickel steel, 38-42 tons sq. in. ultimate strength.

The governors are of horizontal, centrifugal type, acting upon the fuel pump regulating gear.

III.—SOME GENERAL POINTS.

The rating of engines.

So many differing views on engine rating appear to be held that it seems appropriate to ventilate this complex subject here.

(1) *Maximum Rating.*—The practical limit of output in a Diesel engine may be said to have been reached when one or more of the following factors operate:—

(*a*) the maximum percentage of fuel possible is being burned effectively in the cylinder volume available. (For the fuel to burn most effectively, the combustion must be perfect and completed at the earliest possible moment during the working stroke).

(*b*) the stresses in the component parts of the engine generally, for the mechanical and thermal conditions prevailing, have attained the highest safe level for continuous working.

(*c*) the piston speed and revolutions per minute cannot safely be increased.

For a given cylinder volume, it is possible for one design of engine effectively to burn considerably more fuel than another engine. This may be brought about by more effective scavenging, by a more suitable combustion space, by a more satisfactory method of fuel injection, and so on. Similarly, the endurance limit of the materials of cylinders, pistons and other parts, may be much higher for one engine than for another; this may be achieved by the adoption of more suitable materials, by better detail designing as regards shapes, thicknesses, etc., more satisfactory cooling and so on.

The piston speed is limited by the acceleration stresses in the materials, the speed of combustion and the scavenge efficiency, *i.e.* the ability of the cylinder to become completely rid of its exhaust gases. Cylinder liner wear may possibly be considered in certain circumstances. Within

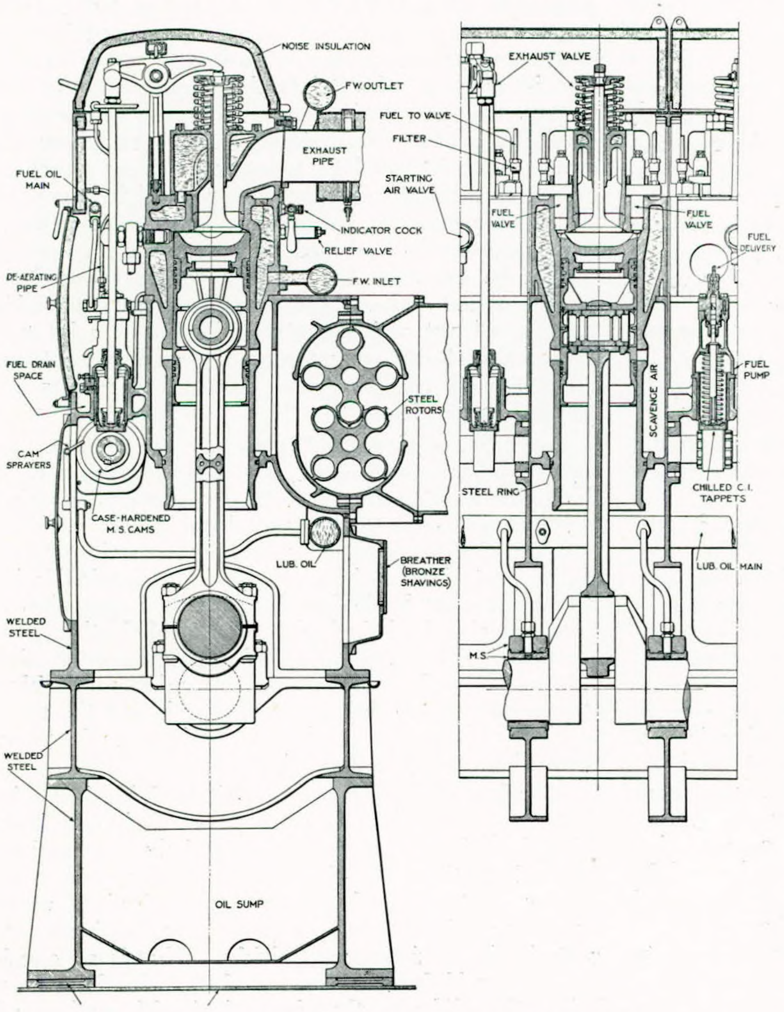


FIG. 25.—Section of small two-cycle auxiliary engine.

limits, so far as combustion is concerned, it is possible sometimes to increase the speed of an engine if the mean pressure is reduced—which may be of importance where generators are concerned.

For each type of engine, therefore, there is a top limit beyond which the engine should not be run continuously. This limit is the maximum rating for that engine type. It is not an easy thing to determine this maximum rating; in fact, it can only satisfactorily be established for each size and type of engine by exhaustive tests.

For the 2-cycle engine types, which form the subject of this paper, the following factors contribute to a high, safe limit of engine rating, viz. :—

(*a*) Longitudinal scavenging, combined with a correctly tuned exhaust system which enables a high mean pressure to be obtained without smoke.

(*b*) Fuel injection system controlling the quantity of oil injected into the cylinder through an automatic fuel valve, which ensures positive and quick cut-off of fuel.

(*c*) The use of special materials for the parts subjected to heat stresses and an effective system of cooling.

The shop tests of these engines consistently show that a mean indicated pressure of well above 8 kg.cm.^{2} (114lb. sq. in.) is fully within the capacity of the cylinders of both double-acting and single-acting two-stroke types as regards perfect combustion and low exhaust temperatures. But to ensure maximum life with present materials, it is preferred to limit the mean indicated pressure to 7 kg.cm.^{2} (100lb. sq. in.) for continuous working.

For example, the ships with 6-cylinder 530/1,250 D.A. 2-C. engines, Fig. 3, when coming home from South America fully laden, run continuously with m.i.p. of 7 kg.cm.^{2} (100lb. sq. in.) as do others with larger engines. The mean pressures are calculated on the total stroke volume of the main and exhaust cylinders.

For single-acting two-stroke engines, it has been found that for the 500/900 size (Fig. 15), running at 140 r.p.m., 8 kg.cm.^{2} (114lb. sq. in.) m.i.p. can be easily obtained so far as smokeless combustion and low exhaust temperatures are concerned. At 160 r.p.m. the combustion may begin to fall off, and 7·7 kg.cm.^{2} (110lb. sq. in.) may be probably not much exceeded for perfect combustion at these revolutions, with the standard cylinder design.

It is not prudent for a superintendent engineer, in connection with new tonnage, to settle upon what appears to be a moderate rating for an engine without having available all relevant comparative data. Thus, 6 kg.cm.^{2} (85lb. sq. in.) m.i.p. might imply a definitely overrated engine for one type and a distinctly underrated engine for another type.

If a cylinder is overloaded by an attempt to burn too much fuel, combustion will continue to the end of the working stroke and perhaps also until after exhaust has begun. Besides suffering an efficiency loss, the engine will become overheated and piston seizures or cracking of engine parts may result—or at least sticking piston rings, dirty and sticking fuel valves will be experienced, with consequent heavier upkeep costs.

While the evils of overrating are well-known, it does not seem to be so well-known that an engine can suffer undue wear and tear because it is consistently too-easily rated. An example of this kind came to the author's notice quite recently.

(2) *Acceptance Ratings.*—It is a frequent procedure for owners to insert in their contracts some such phrase as : "The engines are to be capable of maintaining the designed sea speed of the vessel, fully loaded, when developing not more than 80 per cent. (or some other percentage) of their rated b.h.p.". This type of clause leaves the full-rated power undefined and does not necessarily therefore ensure a moderate rating.

An alternative practice is to agree upon the mean pressures and revolutions which the engines can carry continuously; determine from them the somewhat lower figures to be adopted on trials; then by prescribing a sufficient margin between trial trip power and service power, a moderate rating in service is assured. On trials the engines should not be run one revolution more than is essential to fulfil the contract trial speed. This procedure has much to recommend it, being something which is proper to all high-class machinery until it is run-in.

The acceptance of a ship and its machinery must necessarily be based on a sea trial; a builder cannot be expected to accept a contract based on service speed—where there are so many variable factors, all out of his control. In his own interest the owner should so arrange the requirements of the sea trials that the most economical results are assured in service.

(3) *Some General Comments on Rating.*—In considering the rating of any Diesel machinery a sharp distinction must be drawn between genuine over-rating and some purely local fault, such as may occur at some time or another with almost any engine part, faults which may be fully explicable by a retrogression in design for that part, unsatisfactory material, bad fuel oil, a local lubrication problem, etc. The over-rating of an engine must also not be confused with some propeller difficulty which may simply mean that the propeller is not the most suitable one for the particular conditions of power, revolutions and speed of ship.

Scales of rating are definitely laid down by some superintendent engineers for different types of engine, these being based upon their observations over a long time, showing that beyond certain revolutions, piston speed, power per cylinder, mean pressure, exhaust temperatures, etc., maintenance costs begin to rise more quickly. This is an excellent notion if founded on sound experiential knowledge; the difficulty then is to deal with engines of newer types.

TABLE II.

GENERATOR ENGINES.

Type of Ship.	*No. of Ships.*	*Tonnage.*	*S.h.p.*	*No. of Sets.*	*kW. per Set.*	*Average kW. per Set.*
Single-screw cargo	33	1,000-4,000	1,800-2,500	3	65-75	65
	5	4,000-6,000	2,500-6,000	3	100	100
	6	6,000-8,000	7,000-8,000	3	300	300
Twin-screw cargo	10	1,000-4,000	1,000-2,000	2	55-100	75
	18	4,000-7,000	2,500-5,000	2.3.4	100-200	100
	15	7,000-10,000	5,000-10,000	2.4	100-250	150
	11	10,000-12,000	10,000-14,000	3.4	330	300-330
Single-screw passenger... ...	5	1,500-4,000	1,400-2,400	3	40-65	55
Twin-screw passenger	5	1,000-2,500	1,200-2,000	2	55-60	55
	10	2,500-4,500	3,000-6,000	3	110-220	130
	7	8,000-12,000	5,000-8,500	3	200-250	225
	9	12,000-15,000	8,500-10,000	4	200-350	250
	16	15,000-25,000	10,000-25,000	4.5	350-700	450

(4) *Rating of Auxiliary Diesel Engines.*—The principles indicated above are equally valid for generator engines. There is, however, an additional point, *viz.* the *number* of auxiliary engines necessary.

In a projected ship any proposal to provide other than fully adequate auxiliary power should be regarded with disfavour. This applies to the number no less than to the size of the units.

Table II indicates the number and size of generator engines fitted in 150 typical ships.

(5) *Exhaust Temperatures.*—The exhaust temperature may prove to be a limiting factor for the maximum output of an engine.

An exhaust temperature graph—a diagram in which mean indicated pressures are plotted as abscissæ and exhaust temperatures as ordinates—will generally indicate when the economical combustion limit and sometimes when the safe working limit of an engine has been attained. The economical limit is reached shortly after the exhaust temperature curve begins to bend upwards from what was previously a practically straight-line law. Very often the safe continuous working load limit is also reached at the same time—as the designer naturally strives to make all the parts of an engine suitable for withstanding to an equal degree the respective thermal and mechanical stresses to which its parts are subjected.

Exhaust temperature cannot be taken as proportionate to mean indicated pressure for comparing engine types. Thus in certain 2-cycle engine types the average working mean indicated pressure is about 6 kg.cm.^{2} (85lb. sq. in.) and the exhaust temperature about 350° C. (660° F.). In the 2-cycle, double-acting engines of the type described in this paper the exhaust temperature is often below 300° C. (570° F.) for 7 kg.cm.^{2} (100lb. sq. in.) m.i.p. at normal revolutions. By comparison a 4-cycle non-supercharged airless-injection engine has an allowable exhaust temperature of between 425° C. and 450° C. (say 800° F. and 850° F.).

A small engine can usually work satisfactorily at a higher exhaust temperature than an engine of the same type of larger size.

The exhaust temperature is influenced to a considerable extent by the lead and dimensions of the exhaust piping. The more easily the exhaust gases can flow away the lower their temperature, and vice versa.

Manoeuvrability.

It is commonly said that one of the many advantages of the Diesel engine is its 100 per cent. astern power. For this to be true, it must be instantly available for full use under all conditions of ship speed.

Table III shows the results of typical tests on the engines forming the subject of this paper. The

TABLE III.

TYPICAL REVERSAL TESTS.

Type of Ship.	*Single-screw tanker (fully-loaded).* (1)	*Twin-screw passenger ship, 100% displacement.* (2)	*Single-screw cargo vessel; 70% displacement.* (3)	*Twin-screw passenger ship, 90% displacement.* (4)
Type of engine	4-C S.A.	2-C. S.A.	2-C. D.A.	2-C. D.A.
Ahead speed of ship at time of test, knots ...	11·5	16·75	16·8	13·5
Ahead r.p.m. at time of test	100	112	95	70
Time interval between reversal order on telegraph and engines running astern on fuel, secs.	19	15	20	21
Astern r.p.m	50 for few secs.; then 100	110	50 for few secs.: then 95	50
Time interval between reversal order and ship stopped	4 mins. 25 secs.	1 min. 51 secs.	3 mins. 25 secs.	2 mins. 48 secs.

times are normal manœuvring times, such as can be reproduced at will an indefinite number of times. In example (4) the revolutions are less than service revolutions, due to circumstances existing at the time of the trial.

So far as quick and positive reversal is concerned, the reversing gear will respond as quickly as the engineer cares to operate it, irrespective of ship speed.

Record of engine behaviour.

It not infrequently happens that when a builder is called in to give an opinion on difficulties which have arisen with an engine, there are few, if any, data readily available regarding general behaviour.

It is suggested that on each outward and homeward voyage one set of normal indicator cards and one set of out-of-phase cards, *i.e.* draw cards, be taken, and a simple table compiled as indicated in Table IV. Test-bed and trial trip records, similarly arranged, should be available for comparison.

TABLE IV.

GENERAL DATA.

Cylinder.	1	2	3	4	5	6
m.i.p.						
$p_{max.}$						
$p_{comp.}$						
fuel pump index ...						
exhaust temperature						

Thus if $p_{max.}$ slowly falls, over a succession of tables, the fuel pumps may be worn and leaky; if suddenly, then fuel valves may be choked; if $p_{comp.}$ goes down then piston rings may be sticking or broken, or, much more infrequently, the exhaust valve may be leaking; and so on.

Crankshaft alignment:

Periodical clock-gauge testing of crankshafts is now well-established practice.

For engines with shafts large in diameter relative to cylinder dimensions, it is possible that, when tested by clock gauge alone, comparatively satisfactory readings may be obtained; these may not, however, be true readings, as the crankshaft may not be lying properly on the bottom of all its main bearings. Accordingly it is sometimes desirable to supplement the use of the clock gauge by a long thin feeler, introduced to both ends of the lower main bearing bushes. Alternatively the bottom bushes can be pulled-up by the screw gear provided and the movement measured. Equivalent attention should be given to the top bushes.

Fig. 26 shows a graph which may be helpful in crankshaft alignment problems. It aims at a mean between unnecessary stringency and unduly high crankshaft stresses.

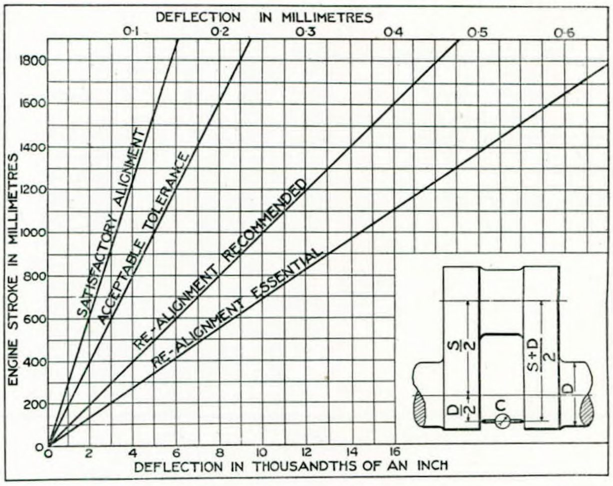


FIG. 26.—Alignment diagram.

Crankshaft stresses: balancing:

A certain amount of confusion seems to persist regarding critical stresses and engine balancing.

In every engine there is a series of torsional critical speeds at which in some instances it is not desirable, and in others dangerous, to run. These critical speeds appear whenever the pulsations in the engine torque coincide with the natural frequency of the shafting system, producing conditions of resonance. There are other critical speeds besides torsional, to consider. For example, it has lately been recognised that there can be whipping in certain circumstances, measurable at the connecting rod foot as very heavy pressure on the crankpin; also there may be critical fore-and-aft vibrations, measurable at the free end of the crankshaft, setting-up bending stresses.

Reduction of stress may be obtained by increasing the shaft stiffness; fitting vibration dampers; sometimes by altering the sequence of cylinder firing; etc. In the D.A. 2-C. engines described in this paper the ratio of cylinder centres to cylinder bore is usually 2·35 and never more than 2·5, and in the S.A. 2-C. engines 1·75 to 1·9. This ensures a very stiff shaft. For only very few crankshafts are dampers necessary. The type of damper then fitted reduces the critical stresses very effectively and the engines can run with impunity on any critical point.

The sequence of firing is bound up with the constants by which crankshaft diameters are determined.

Engine balance is concerned only with so arranging the reciprocating and rotating weights of an engine that the forces and couples which they create cancel out, so far as possible. Balancing problems have no concern with cylinder pressures and crankshaft torque. (Sometimes the levelling-up of i.h.p. amongst the cylinders is termed by marine engineers "balancing an engine"—but this is something quite different.)

In engines of six or more cylinders there is complete balance if the cranks are symmetrical about the crankshaft mid-length; examples are 6 and 8 S.A. 4-C. cylinders and 6 and 10 D.A. 2-C. cylinders. Where the cranks have star formation there may be a residual couple—usually a secondary couple; this is so small as to be completely negligible. In two-cycle engines with symmetrical cranks, resulting in two cylinders firing at the same time, the guide-reaction couples are taken by the hull; star-formation of the cranks reduces these couples.

Crankcase ventilation

From such evidence as is available, it would appear more prudent to provide positive crankcase ventilation—*i.e.* a fan vapour extraction system—than to omit it. Fan ventilation has the added advantage of reducing oil seepage at crankcase doors, etc.

Cooling water systems, etc:

It has been found that the most satisfactory scheme for cooling the combustion chamber walls and reducing liner wear to a minimum is to ensure that a large quantity of air-free, fresh cooling water at a comparatively high inlet temperature and small temperature increase flows at high velocity past the hottest parts of the combustion chamber space. By preventing the temperature inside the cylinder, including piston rod sleeve, from falling below the dewpoint at any point of the cycle, wear is reduced and temperature stresses minimized. It is essential that there should be no pockets in the cooling spaces around the combustion chamber, whereby cooling fluid may be prevented from taking proper part in the circulation. The difference between effective and ineffective cooling is often the difference between materials continuing to stand up to their work and not continuing to do so; also the more effective the cylinder cooling, the more satisfactory the lubricating oil film on the liner, and therefore the less the wear.

Before starting up, the circulating water should have a temperature not less than, say, 50° C. (120° F.).

The fresh-water circuit, closed and air free, consists of pumps, coolers, make-up head tank, and de-aerating line at highest point of system. Distilled water is not required.

By cooling the main and exhaust pistons with lubricating oil, these are maintained at as high a temperature as practicable, thus assisting good combustion. Leakage from joints is harmless.

Lubricating oil system:

Where difficulty occurs in the cleansing of coolers, etc. of carbonised oil, experience shows that it is because carbonisation has been allowed to proceed too far. There appears to be no solvent for carbon. But there is no difficulty in removing partially solidified oil and thus maintaining coolers in such a condition that their efficiency is unimpaired. Trichlorethylene, used as vapour, has proved its value in the cleansing of coolers. Carbontetrachloride, equally suitable, is more costly. The degree of success attending the use of these substances is necessarily bound up with the method of application. Very good reports are given of preparations consisting chiefly of sodium metasilicate and trisodium phosphate. A very simple means is the use of plain fuel oil, if applied early.

Cylinder covers and pistons:

The average physical properties of the chrome steel used are: tensile strength 38-47 tons sq. in., elongation 30 per cent. to 18 per cent., yield point 20 tons sq. in., bend test 120°. The technique of manufacture and heat treatment is rather sensitive. Correct heat treatment is essential if grain size is to be kept within reasonable limits. The percentage content of certain constituents affects the heat treatment range.

An alternative material is chrome-molybdenum steel, which is less costly and the coefficient of heat transmission of which is about twice that of chrome steel.

It is noteworthy that the cast-iron covers of a D.A. 2-C. engine running continuously since 1931, are still sound save for a few small cracks. Something just a little better than cast iron is probably all that is required.

The life of pistons is undoubtedly affected by frequency of starting and stopping, and the conditions under which manœuvring is carried out. It is doubtful if Diesel engines receive as kindly usage at the hands of ship officers as did the reciprocating steam engine.

Piston rings:

It is easy to state that the materials, design and manufacture of piston rings and cylinder liners should be such that there is neither blow-past nor sticking, that the friction and the wear are the least possible, and so on; it is much less easy to satisfy these desiderata.

Very conflicting results can be obtained with rings of equal hardness when applied to different liners. This is mentioned because it is sometimes the practice of superintendents to require a specific hardness for the materials of rings.

The edges of the working face of piston rings should be bevelled, or rounded, to assist in retaining the lubricating oil film on the cylinder wall until the rings and liners are worn smooth. For the 620 mm. (24·41in.) piston of Fig. 5A a 2 mm. (0·08in.) radius would be suitable.

Regarding the ends, some engineers favour overlapping ends as at (*a*) and (*b*), Fig. 27, to obtain a seal at the joint; others prefer the diagonal cut of Fig. 27 (*c*). Overlapping ends are liable to break off, especially if pinned there; an objection some-

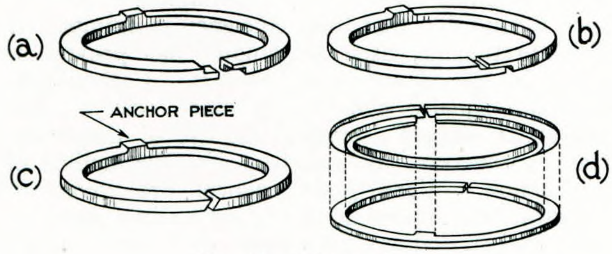


FIG. 27.—Piston rings.

times raised against the diagonal cut is that the gap tends to increase rapidly, being a multiple of wear of ring plus liner. In the author's opinion the balance of advantages lies with the diagonal cut.

Rings should be anchored (Fig. 27) diametrically opposite the ends, whatever the form adopted, to minimize vertical wear and to enable the rings to mate perfectly with the cylinder. No cylinder completely retains its form cold and hot, and an un-anchored ring is liable to rotate somewhat and show error. The ends coincide with the broad bar at the scavenge ports.

A correctly made ring, when closed-in to the predetermined amount at the ends, should present a truly circular working surface. To reduce costs, however, makers' processes sometimes are such that rings are only approximately circular; such rings should not be used.

The finish of the groove is important. It is of little avail to have excellently finished rings working in grooves having a rough surface; blowing-through with new rings is often attributable to this cause. Caulked carrier rings, being carefully ground, are free from this difficulty.

It is important that the maximum amount of heat from the combustion space should be conveyed away before the top piston ring is reached, otherwise this ring is likely to stick due to heat expansion. For this reason therefore the first groove is as distant as practicable from the piston crown. The first and second rings should always be made to float more easily in their grooves than those further down. These points are of enhanced importance in small engines having uncooled pistons.

Liner surface should be considered in conjunction with piston ring problems. It is often believed that the best finish is that imparted by grinding; this, however, is not altogether proven. If a liner is turned to a reasonably good finish, the innumerable tiny hollows in the surface may serve as lubricating oil receptacles tending to reinforce the oil film which is so important.

To reduce the likelihood of seizures during the running-in period, lead-bronze rings fitted to machined grooves in the piston circumference are very useful, the face of the rings being about 0·1 mm. (0·004in.) above the piston surface. Piston clearances may thus be somewhat reduced.

With tight piston rings having small gaps, scavenge fires are unknown. If there is much piston ring wear, with corresponding gap enlargement, permitting blow-past, scavenge fires will appear. So long as lubricating oil can be drained from the scavenge belt there will be no fires, but if, due to blow-past, the lubricating oil becomes carbonised, then the walls around the scavenge ports will gradually become dry and danger of fire will arise.

For small auxiliary engines, 4-cycle as well as 2-cycle, ring-sticking is sometimes a problem. The evidence on the subject is surprisingly tangled and contradictory. The factors involved may be: liner wear, by producing metallic particles; design of piston, as regards crown thickness, location of top and spacing of other rings; proportions, shape and elasticity of rings; revolutions—not always due to being too high; quantity of lubricating oil—usually an excess—and method of filtration; presence of water in the lubricating oil; quality of the fuel oil—chiefly as regards its coking tendencies.

Main bolt tightening:

As questions regarding pre-tightening of main bolts are frequently raised, the subject is very briefly touched upon.

In single-acting 4-cycle and 2-cycle engines there is a nut on the cylinder top, also one on the framing top. The latter is pulled hand-tight—after being sledged and slackened, to bring all faces together—and then rotated by tup the amount the builders prescribe. The top nut is then pulled hand-tight and rotated the prescribed amount, which just lifts the intermediate nut clear. Thus, for a 500/900 engine the intermediate nut is rotated 226° from hand-tight and then the top nut 49°. The intermediate nut holds all the engine parts together when the cylinder cover is dismantled; it also simplifies the tightening-up of the top nut.

For D.A. 2-C. engines the tie bolts are in two pieces, the junction being at the middle nut (see Fig. 5A). The latter is pulled hand-tight and the upper bolt is pulled into the middle nut; then all the main bolts are tightened hydraulically by a patented system consisting of a hand pump coupled to portable rams on the top nuts. The pump pressure ensures a definite and equal tensile stress in all bolts, usually about 650 kg.cm.^{2} (9,250lb. sq. in.). The top nuts are then run-down, by means of a toggle bar, until they are hand-tight on the cylinder cover. The hydraulic pressure is re-applied for a check reading, then released and the portable rams taken away. The space under the middle nut is filled by a split steel washer, which slides into place.

Fig. 28 shows the arrangement. At (*b*) the screwed part of the bolt is made longer to take the hydraulic ram. In existing engines, with bolts having screwed portions of normal length. a tap bolt can be screwed into the end of the tie bolt, holding the ram, as at (*c*). The ram diameter varies with different engine sizes, the hydraulic pressure being constant, usually about 211 kg.cm.^{2}

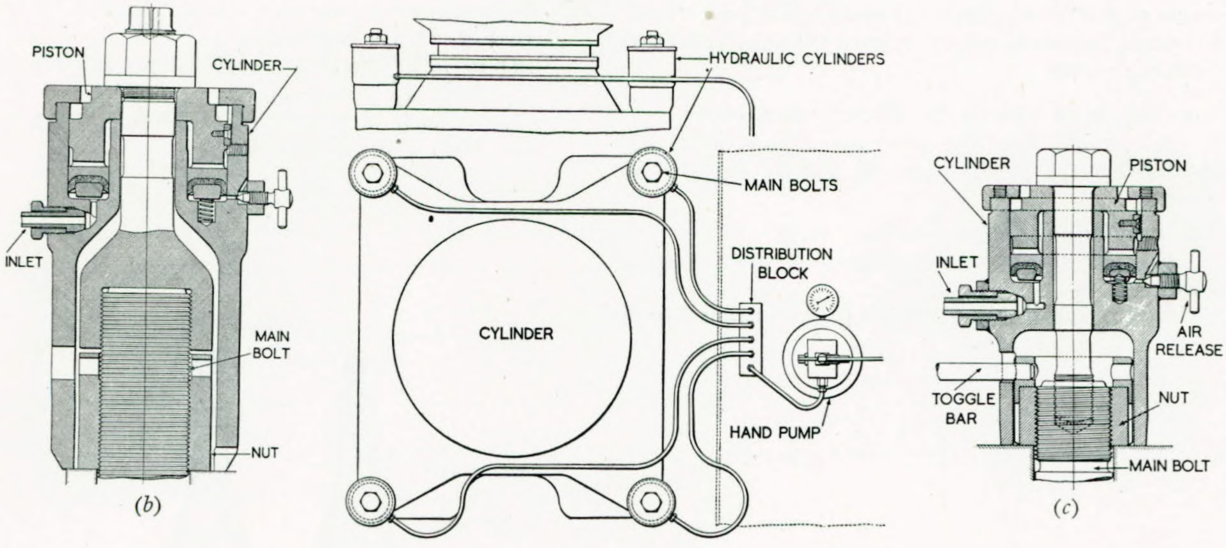


FIG. 28.—Main bolt tightening.

(3,000lb. sq. in.), except in special circumstances.

The present tendency is to extend the principle

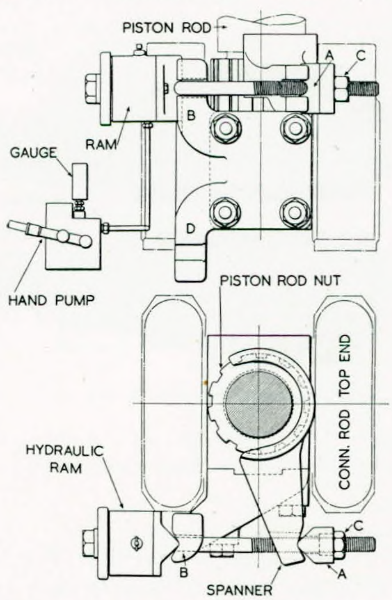


FIG. 29.—Crosshead tightening gear.

of hydraulic or pneumatic tightening for important nuts, *e.g.* Fig. 29 shows a method of hydraulically tightening or untightening crosshead nuts. A screw-jack can be substituted for the hydraulic cylinder.

Tuned exhaust systems:

Any kind of interference with the free escape of exhaust gases results not only in increased temperature but also in augmented back pressure. Increase in back pressure means a proportionate increase in scavenge pressure; this in turn implies a corresponding loss in mechanical efficiency of the engine.

The tuning of an exhaust system, in principle, consists of so arranging—and if necessary subdividing—the exhaust pipes, and determining the length of pipe to silencer, that the waves in the exhaust gases are of such period that the maximum suction effect is obtained. By this means thorough scavenging is obtained with reduced back pressure and least expenditure of blower power. Between a tuned and an untuned exhaust system there may be a difference of about 1·0lb. sq. in. (0·07 kg.cm.^{2}) at the blower.

Translated into practice, analysis of different numbers of cylinders may show that for one number a large single exhaust manifold with appropriate length of pipe to silencer, provides optimum conditions; for another, grouping of exhaust pipes into 2, 3 or 4 pipes may be necessary. The saving in blower power makes the arrangement well worth while.

Silencing:

There is not space to do more than refer in broadest terms to this important subject.

The ideal exhaust arrangement would be a straight pipe, with silencers of very large volume, led to the top of a high funnel; but conditions on shipboard often make effective silencing difficult to realize.

There are also degrees of silencing; the level of silencing regarded as satisfactory for a tanker is something very different from that required for a high-class liner, as some silencer specialists have learned to their cost. Also, auxiliary engines may be worse offenders than propelling engines.

For many installations it is essential to have two, occasionally three, silencers, arranged serially. The isolation of exhaust pipes, silencers and their hangers in a way which prevents noise and vibration being transmitted to accommodation spaces is a study in itself.

Comparison of spaces for Diesel machinery:

As an index to the progress achieved during, say, the last 15 years, Figs. 30 and 31 may be of interest.

(1) *Installations Having Same Power.*

Fig. 30 shows two installations of the same power, viz. :—

(*a*) to full lines; 4,500 s.h.p., 115 r.p.m.; built 15 years ago.

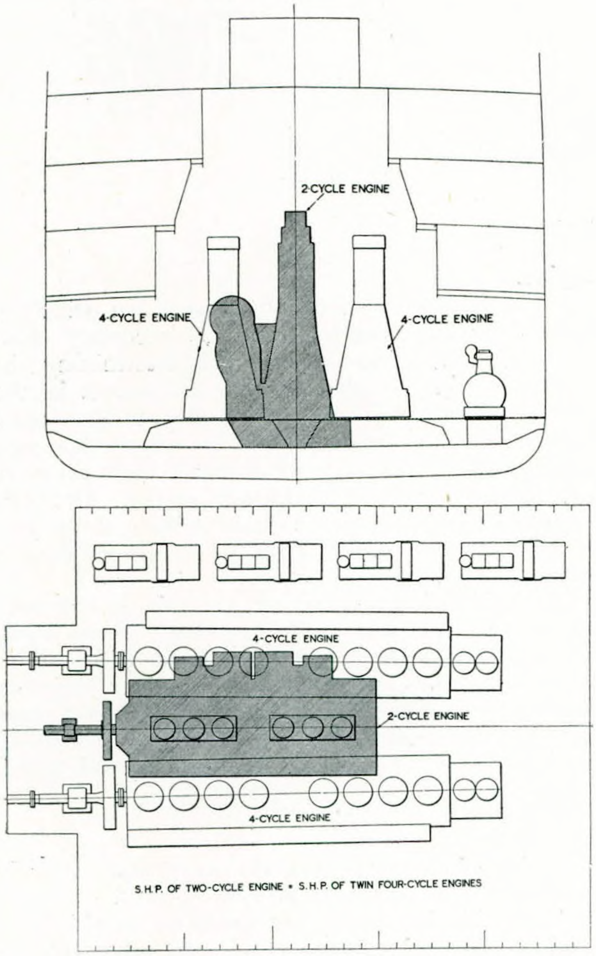


FIG. 30.—Diesel engine comparisons.

(*b*) to shaded lines; 4,400 s.h.p., 110 r.p.m.; built few months ago.

Decrease in engine weight for same power= 74 per cent.

(2) *Installations Occupying Same Space.*

In Fig. 31 are indicated :—

(*a*) to light shading; 4,500 s.h.p., 115 r.p.m.; built 15 years ago.

(*b*) to dark shading; 9,200 s.h.p., 98 r.p.m.; built few months ago.

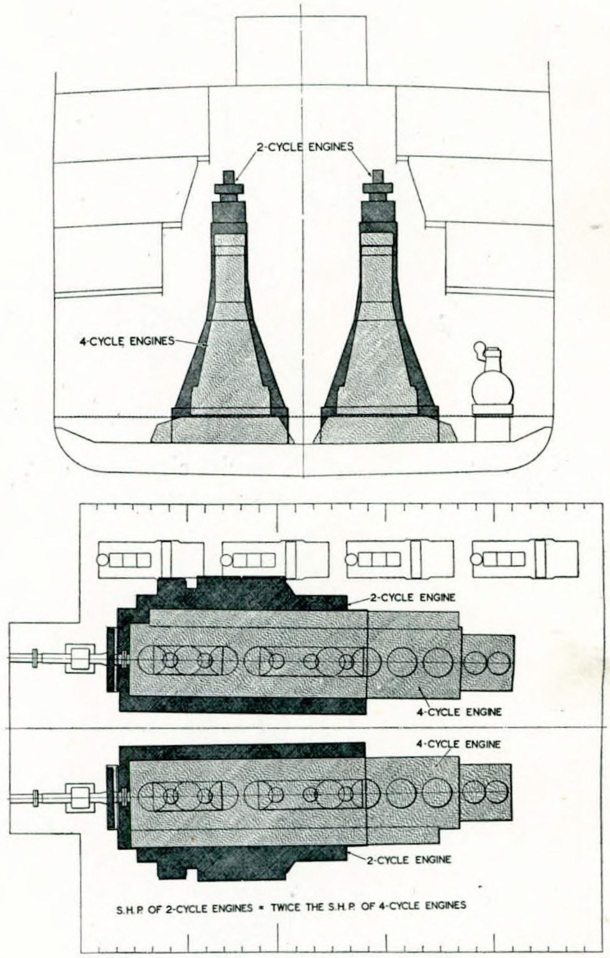


FIG. 31.—Diesel engine comparisons.

Increase in power	104%
Decrease in space for (*b*) ...	15%
Decrease in engine weight for (*b*)	47%

The progress made during the last 15 or even 10 years, is clearly exemplified by three large motor liners which were re-powered in Belfast in 1938 to meet a greatly accelerated service. The new 2-cycle machinery has capacity more than twice that of the original 4-cycle engines, on an equivalent rating basis, with a considerable saving in weight and much engine room space to spare.

Discussion.

Mr. W. S. Burn, M.Sc. (Member), opening the discussion, said that the paper was filled with valuable information concerning some of the most interesting types of oil engines of the present day, and the fact that these were very different from the type with which the speaker had been chiefly associated made the paper all the more interesting to him.

Reference was made to engine-driven pumps being duplicated as independent units. Did this imply that they should be of similar size and construction but driven by an electric motor or steam engine? Such an arrangement had definite advantages.

The arrangement of combining the deckhouses and the funnel was an excellent one and, as the author remarked, would doubtless be developed to make the modern ship a thing of some beauty, as well as being functionally correct and facilitating a measure of streamlining.

Whilst the saving in engine-room length given on page 2 was striking, it was not perhaps as great as that possible with an orthodox type of double-acting two-stroke which had cylinder centres of only 2·1 instead of 2·36 cylinder diameter, nor was the saving in weight so striking, as figures of 120lb. per b.h.p. could readily enough be obtained in an orthodox double-acting engine of 5,000 b.h.p. with a welded structure.

Indeed, the B. & W. engine had perhaps the unique distinction of having by far the greatest weight of moving parts, compared with the total weight, of any double-acting type, and it would appear that only at the expense of mechanical complexity of these parts had the undoubtedly high combustion efficiency been obtained. At the same time there seemed little doubt that the intrinsic cost per ton of the engine must be relatively high, although again this might be balanced by the relatively high specific power output and the excellent production facilities available.

The pillarless construction of engine room was undoubtedly attractive, but in connection with the relative design of the tank top and bedplate, one would have thought that an extremely stiff flat tank top type bedplate to give, with the engine cylinder entablature, an extremely stiff engine in itself would be preferable from the point of view of vibration and noise elimination. Especially would this be the case if the engine was mounted on some form of rubber chocks. This was certainly the usual line of development when mounting auxiliary engines or automotive engines on rubber.

The idea of insulating the engine from the hull was sound and logical, and the work of the author in this connection was most valuable. The design of the rubber chock described was most reasonable for the initial development, although gradually one would imagine a type of chock with greater "flowing" capacity would be developed, as if rubber was too much restricted it became virtually solid and the vibration damping properties were not fully realized. The use of synthetic rubber would have further advantages due to its oil and chemical resistance and its greater damping properties compared with natural rubber. An alternative chock would be a plain sandwich of rubber bonded to two light steel plates. There was also a possibility of using a granulated cork impregnated with synthetic rubber solution, but this implied a much reduced unit loading and it was most suitable for flat-bottomed bedplates.

In connection with the section of the paper giving particulars of the double-acting designs, seeing that there were so many B. & W. two-stroke engines running satisfactorily, it was almost presumptuous to criticize the basic design, but the speaker could not help wondering what would have happened had B. & W. and the author's firm developed an ordinary plain double-flow type double-acting two-stroke, and it would be interesting to know if such engines were designed and built and discarded in favour of the reciprocating cover type of engine.

From the speaker's own experience very high outputs could be obtained, with a clean exhaust, with the double-flow fixed-cover type of engine, and the power limit was usually imposed by the piston design which was a problem common to all types.

There was no doubt, furthermore, that the success of an engine was chiefly due to the gradual perfection of details in service to give the correct balance between the various component parts, and this unfortunately required both time in service and money.

When the single-acting engine was considered, the uniflow principle became extremely attractive, especially when the upper piston did a fair share of the work as in the Doxford engine, the success of which type had been so complete. Eventually, it was to be expected that the stroke of the upper piston would tend to increase to justify the cost, weight and complication of the operating mechanism, as well as giving superior port opening time area properties.

As a designer, the speaker would have been pleased to have seen the equivalent of Fig. 6 for the exhaust-piston type of engine.

With regard to the piston design, the relatively small amount of guiding of the cooling oil was noticeable, and it would be interesting to have the author's views on the relative merits of splash cooling or leading the oil rapidly over the heated surface by means of baffle plates.

It was also noticed that no special endeavour seemed to be made to avoid stress-raising oil holes in the various crank journals and pins. Such holes had been eliminated in the engine for which the speaker was responsible and certain other marine engine builders were developing on similar lines.

It was noted that the cast-iron sleeve on the piston rod was retained, but there was no doubt that completely satisfactory results could be obtained with a mild steel rod. To-day there need not be a gland problem. An important feature of an efficient gland seemed to be to have a set of labyrinth packing (as in Fig. 8) and then to have a really good gastight seal at the bottom of the gland. This preferably should be segmental and readily renewable. Provided the gland operated at a fairly high temperature but was kept well away from the maximum temperatures and pressures, the modern gland gave very little trouble and very moderate wear.

The speaker was interested to note that two fuel valves could be efficiently operated from one fuel pump and this confirmed his own experience, although there were certain engine builders who had had trouble and preferred to use one pump per valve.

It was noticed from the illustrations that the scavenge pump had tended to get rather more complicated and it would be interesting to know whether, before deciding on the Roote's blower type of pump, B. & W. had tested out an ordinary piston type pump with mechanically-operated valves. Whilst the rotary blower looked attractive diagrammatically, in practice a number of complications seemed to develop, particularly in the drive and in the reversing. Even on the score of reduction in the size of the scavenge air receivers, there was much to be said for a timed reciprocating scavenge pump, so that the discharges from the pump filled in the pressure depressions when the scavenge air ports opened. This tended to reduce the size of the scavenge air piping generally and made the engine neater.

The spring shock absorber in the fuel pump suction line was definitely a worthwhile refinement, and the elimination of the fuel pump delivery valve was interesting but might cause trouble if the fuel valve seats were not just perfect during critical periods such as manœuvring. Because of this the speaker had developed a special type of fuel valve to leave a low residual pressure in the fuel system.

The author mentioned the factor of safety of 50 for chains. The speaker thought this should not be less.

The fuel consumptions of the B. & W. engine were remarkably good, but were the figures given based on metric or British horse power? In this connection it would be interesting to know the difference in performance between the standard and the latest types of double-acting engines, as the combustion spaces were quite different.

It would also be interesting to know whether the new cylinder liner was intended primarily as a structural liner improvement, or to obtain an increase in power.

Coming to the smaller sizes of engines, it was noticed from Fig. 25 that sound insulation had been adopted to reduce the noise from the valve gear, which was a defect of these valve-in-head engines, and one wondered why the camshaft was not near the valve as in the older four-stroke design, as in Fig. 21.

The speaker had recently had experience in designing and developing a small single-acting two-stroke with a single exhaust valve, and the problem of getting adequate time area without undue stroke loss entailed an exceedingly rapid opening and closing of the exhaust valve, which could only be obtained by making the weight of the mechanism between the valve and the cam an absolute minimum.

On the subject of crankcase ventilation, the conclusions the speaker had reached were that large engines of the crosshead type might be fitted with vapour extractors but should not be ventilated; this would give a reduced oil seepage without much greater risk of explosion; small trunk-piston engines, on the other hand, should be ventilated to prevent oil contamination and consequent bearing and liner corrosion, as the chance of an explosion was more remote.

Coming to the materials used in cylinders and pistons, it was doubtful if the "fancy" cast steels had been worthwhile, and there was much in the author's remark that something a little better than cast iron was required in this connection. Certain special cast irons were now available which had a higher fatigue impact value than mild steel; certainly the simplicity of a plain cast-iron piston had much to commend it, and the design was worth developing to make this possible.

It was particularly interesting to note the author's statement that the piston rings should be anchored. Logically there was much to support this, but on the other hand very few two-stroke makers did nowadays actually anchor the piston rings or find it practically desirable. It would be interesting to know whether the anchoring was required to stop ring breakages at the ports or to reduce blow past and consequent ring sticking. Curiously enough, only some weeks ago the speaker was interested in a design of a hydraulic spanner for removing the piston-rod nuts in R.W. oil engines, but in addition to the hydraulic force a percussion element was introduced to reduce momentarily any adhesion between the threads; the results in actual service were not yet known.

Mr. G. E. Carter (Visitor) observed that Great Britain introduced the steam engine, and the steam engine, especially on the marine side, was a simple, easily constructed and easily repaired machine. He thought he was not overstating it when he said that simplicity was characteristic of British design; on the contrary, Continental design seemed to revel in complications and gadgets and as the marine Diesel had been influenced greatly by Continental designers it had tended to get away from the simplicity that was essential. The consequence was that sea-going engineers, especially in the tramp class, did not realise that they had to deal with a machine which required a proper routine of examination. They still were in the position of the old coachman who got promoted to chauffeur when the motor-car replaced the horse. They could work the levers but let the machine alone until something went wrong.

It would appear, therefore, that one of two things must happen. The characteristics of the man must change or the characteristics of the machine could be changed. The latter, he ventured to say, after a long experience of engineers and engines, was the simpler plan and he would, therefore, suggest to the author that of all the designs shown the engine having a piston valve of the same size as the main piston approached the ideal of simplicity much nearer than those having poppet valves. He thought that this design of engine both in single- and double-acting types would be the engine of the future. He saw at Copenhagen recently the double-acting engine referred to (which he understood Messrs. Harland & Wolff, Ltd. sponsored), and he thought that it had qualities which would appeal to the sea-going engineer as well as to the engine-builder.

For the low-powered tramp these engines gave rather too much power and for round about 2,000 b.h.p. he expected the pressure charged four-stroke engine would still be popular, especially as with the Buchi system of charging brake mean effective pressures of 7·75 kg.cm.^{2} or 110 lb.in.^{2} could be carried as normal. This engine, like the steam engine, could run under the most extraordinary conditions due to lack of attention, and this was important in a class of ship where the engineers were not of high quality and very frequently changed.

The author stated on page 5 that in a vessel with steam auxiliaries having only one boiler he would recommend in addition a thimble-tube boiler to take exhaust gases. This was not necessary as there were a number of ships in service having one boiler which could take exhaust gas on any furnace at the same time as the other furnaces were on oil fuel.

The author covered in a few words the fitting of rubber chocks under the engines of some cross-Channel boats, but the speaker considered a lot of praise was due to the firm which had the courage to take a step of this type into the unknown. It was pleasant to know that our engineers could tackle and overcome a problem successfully when called upon.

Mr. A. C. Hardy, B.Sc. (Associate Member of Council) observed that the paper seemed to him to be divided into Three Main Sections and to present Two Main Thoughts.

The three sections were each valuable to a different section of the shipping community, and the first he thought would be of most value to the shipowner. Mr. Burn dismissed this section somewhat summarily, but this was natural as he was a designer and more interested in the details of the engine than its application. Those who had at heart the furthering of Diesel progress, as well as many shipowners, would be pleased to have and to study the type of information in the first section.

The second section was undoubtedly one much more for the designer. Here again the points were handled in a manner of frankness which he (the speaker) had seldom seen in a paper of this kind.

In the third section the author appeared in part to be indulging in a characteristic which he had of "leg-pulling". The speaker liked the way he dealt with superintendent engineers and the subject of rating. The subject of rating of engines was one which had caused more trouble to the Diesel engine than any other difficulty which it had encountered. Many engines had been spoilt because they had been underrated, overrated, or, as the author suggested, not rated at all.

So much for the three sections. The two Thoughts were even more important. The first was that the Diesel engine was not after all, as its detractors would have them believe, a foreign contraption. They had been fortunate to have Mr. Burn, who had done as much as anyone to develop an all-British oil engine, to open the discussion. As far as the author was concerned, Belfast had been the home of all British engines, and he thought it could be said that the internal-combustion engine to-day was really a British product in every sense of the word. It owed to Belfast no little of its present success.

The second thought the author left with them was the way in which the Diesel engine had developed in the last 15 years as regards power and size. The last diagrams in the paper were most illuminating in this respect. To-day 9,000 h.p. could be accommodated in less space than was required for 4,000 h.p. 15 years ago. That was an indication of how things were going.

The author might have said some harsh things on the subject of current tonnage measurements. It would have been valuable from the point of view of The Institute if the author had told them how tonnage measurement could be altered to enable advantage to be taken of this compression of power which the engineer had given them.

He would draw wrath down on his head by suggesting that the author should stress the section of the paper of interest to the shipowner rather than the Diesel engine design aspect. It might be said that one could get anyone to design a fuel valve —in this the owner was not interested, "tons per 24 hours" being his criterion as to the success of the design—but one could not get everyone to put a Diesel engine into a ship even to-day.

Mr. H. J. Wheadon (Member) considered the author's remarks on the rating of the marine Diesel engine the most valuable part of the paper. The tendency in the past had frequently been to be over-optimistic with regard to the continuous power output of Diesel machinery, encouraged no doubt by the lack of precise definition of full power rating in the owner's building contract, which allowed engine builders to reduce their tendering costs by overrating the propelling machinery. The author's observations, if adopted, should benefit both owner and builder alike, for not only would the former be assured of obtaining engines of suitable service rating, but builders would in future quote on an equal basis.

Although the speaker was in general agreement with the author when he stated that "the acceptance of a ship and her machinery must necessarily be based on a sea trial", there was, however, the difficulty, particularly in the case of cargo vessels, of the allowance to be made for the amount by which they were light of their loaded displacement under trial conditions, when frequently the propeller was only half immersed.

On page 12 the author briefly introduced his firm's latest development. Many of them had hoped that he would have dealt more fully with this new departure, particularly as regards the advantages to be gained by its adoption. It would add to the interest of the paper if he would provide more definite details of the reduction in weight and space and information as to whether production costs would benefit.

It appeared from Fig. 14 that the long stay bolts, which in the case of the previous designs of double-acting B. & W. engines extended from the top cylinder cover casing down to the underside of the bedplate cross girders, had been eliminated. Any saving in weight in this direction, however, might be negatived by the heavier scantlings of the driving parts of the exhaust pistons. After comparing the relative sizes of the eccentric and main crossheads, and also the eccentric rods with the connecting rods, the speaker wondered whether this engine should not properly be regarded as an opposed-piston engine with, he would imagine, rather inefficient power transmission from the side rods through the eccentrics to the crankshaft. In this connection it would be interesting to learn the percentage proportion of the total power developed by the exhaust piston. In the case of the present double-acting two-stroke B. & W. engine with exhaust pistons half the cylinder diameter, he believed the proportion was 10 per cent.

So far as his own experience was concerned, the only weakness with the double-acting two-stroke engine was the difficulty in maintaining gastight piston-rod stuffing boxes, although he could see no reason why they should be any different in this respect than was the double-acting four-stroke engine, which was much more free from this trouble. The author might have some views on this point and might also inform them whether his firm had considered the possibility of designing a stuffing box with removable segmental rings after the fashion of the United States packing, so that it would be possible to renew the packing rings without having to disconnect the piston rod from the crosshead as at present.

Finally, on page 9 the author stated that "a lower grade of fuel oil can be used if the fuel is circulated through the fuel valve from an independent circuit". The possibility of using a cheaper fuel was of course attractive, provided it did not lead to increased maintenance costs owing to cracked pistons and covers or accelerated liner wear. Would the author explain more fully how this system enabled the engine to use a lower grade fuel?

Mr. E. G. Warne (Member) said that the main point he proposed to raise had already been partly dealt with by the previous speaker. It was in connection with the piston valves for the exhaust. The earlier designs of single-acting and double-acting engines of the B. & W. type used—and in the case of the double-acting engine still used—the piston exhaust valve. Then a development of the single-acting engine was to use a poppet valve, thereby losing the 10 per cent. gain attributed to the piston valve, but presumably obtaining an improved mechanical efficiency. He believed he was right in stating that there had been a proposal to convert the double-acting two-stroke piston exhaust valve engine to a poppet valve engine. In that case there would again be a loss of 10 per cent. and the presumed increase in mechanical efficiency. Now the latest development was to drop the poppet valve idea for the double-acting engine and not only retain the piston exhaust valves but increase their diameter to that of the main piston. It seemed to him that a deadlock had been reached where no gain was made by using poppet valves nor was the mechanical efficiency improved, due to the increased diameter of the piston valves.

On the proposal of **Mr. J. Calderwood, M.Sc.** (Member of Council) a most cordial vote of thanks was warmly accorded to the author.

By Correspondence.

Mr. W. E. McConnell (Vice-President) wrote that parts I and II, which were mainly descriptive, were chiefly noteworthy for the excellent illustrations and the amount of detailed information they contained; the drawings showed how marked was the advance in design during the past twelve years, although there had been no important reduction of fuel consumption. The complicated and delicately designed engines of to-day had a greatly improved power-weight ratio, but they needed, of course, very skilled supervision if their performance was to be maintained. Even the full description given could not entirely convey a proper sense of the extreme precision needed for the satisfactory working of these efficient and powerful engines.

Part III introduced matter which offered more scope for comment, and students would note with profit the remarks on rating and materials. The subject of rating was one of the most controversial topics; this would be evident from the author's definition of what he truly described as a complex subject. It seemed that it should not be difficult to determine a rational basis of comparison, but in fact the matter was complicated by the claims of different builders and the requirements of different buyers; the desire to maintain sales in a highly competitive market had sometimes led to claims for performance which had not been sustained in service, and the attempt to obtain the promised results had on occasion been disastrous. The author touched the core of this problem in his remarks that, although shop tests of an engine showed that an m.e.p. of 114lb. per square inch was quite possible, it was better to limit the m.e.p. to 100lb. per square inch to ensure maximum life with the materials now available. This was especially true of pistons for high-speed engines, which in some cases ran at a rate so near their maximum possibility that a small increase in loading was sufficient to melt them. In this, as in so many other features of oil engine development, designers awaited the metallurgist, who had solved many of the earlier problems, but had yet a few to deal with. Certainly it would not improve the reputation of the high-speed oil engine to market it with a higher rating than could be fully guaranteed under the severest conditions of service, which might range from arctic to tropic temperatures.

The author's remarks on this matter were worthy of consideration and so also was his reference to the desirability of a fan vapour-extraction system for the crank cases of oil engines. He pointed out one advantage, the reduction of oil seepage at the doors, etc., but it was suggested that even greater benefit was the improvement in the engine-room atmosphere and consequent benefit to the health of the staff; the inhalation of fumes had been regarded as the cause of much sickness in the past.

It was noted that no reference was made to the abolition of dowel pins in crankwebs; this was probably now regarded as general practice, but it might be profitable to mention the accuracy required in assembling shafts, the after unit of which transmitted 14,000 h.p. on a journal which depended entirely on the effectiveness of a shrunk fit. Two recent accidents involving injury to main crankshafts showed partial shearing of the dowels; if none had been fitted the results would doubtless have been more serious, but with the improved method of preparing the journal and web surfaces it seemed to be generally agreed that careful calibration would ensure freedom from failure in built crankshafts

Sea-going members especially would be grateful to the author for a very informative paper, and it was to be hoped that further information as to the performance of the designs dealt with might be available after they had been a little longer in service.

Mr. G. Jacobsen (East Asiatic Co. Ltd.) wrote that on page 7 the author referred to steel covers and stated that "occasionally—if so required—the jacket is cast with the cover". The writer would point out that it should not be "occasionally" that the jacket was cast together with the cover, but that the cover should always be cast in this way, thus eliminating the screwed-in type of steel sleeves for the valves which in many cases had shown itself to be an impracticable construction owing to the difficulty of removing parts of the sleeve in case of repacking. It had also been a common practice in later years to cast the cover and jacket in one piece.

Referring to page 20 where the author referred to the indicated pressure, he was in full agreement with him that a fine combustion could be obtained under service conditions with this type of engine working at a mean indicated pressure of 7 kg./cm^2 and also at higher pressure. This was one of the big advantages of this type of engine, but on the other hand, with regard to cylinder wear, it would in some cases show a better result commercially not to run the engines with such a high mean pressure, as under such conditions experience showed that a cylinder wear of 0·3 to 0·4 mm. per 1,000 hours could be expected.

On page 23, under the heading of "cylinder covers and pistons", the author referred to a case where cast-iron covers had been in service over an extended period. The writer thought that this had reference to the m.v. "Erria", which was now entering her 8th year in service, and in which the top covers were still the original cast iron ones. A year after the ship had entered into service, it was proposed to change all the covers (bottom as well as top), but as the top-end covers at that time were all in fine condition it was preferred not to change them and to continue running with the cast-iron covers, which had the cover and jacket cast in one piece. The covers were still in good condition, and therefore it would appear that it *was* possible to make cast-iron covers which could last for a long period when cast in the right way and of a suitable material. However, until metallurgists produced a metal of the qualities indicated, the writer considered it a very sound policy to have cylinder covers made of high-quality heat-resisting material, such as chrome-molybdenum.

The experience of the writer's firm with Diesel engines had been a continuous one going back 27 years to the old "Selandia". Recently this well-known vessel has been replaced by a new ship bearing the same name, in which two-cycle engines of the Burmeister & Wain type were fitted. It would be difficult, he thought, to find a more impressive testimony to the progress made in a quarter of a century than a comparison of the machinery of these two vessels.

Mr. R. Wright (Member), in a written contribution. referred to the description of resilient foundations for cross-Channel passenger vessels mentioned on page 4. The two vessels to which these resilient foundations were fitted, although not mentioned in the paper, were the "Leinster" and "Munster" of the British & Irish Steam Packet Co., Ltd., engaged on the cross-Channel service between Dublin and Liverpool. The "Leinster" had been in service now about 15 months and the "Munster" for a shorter period.

The foundations were very carefully watched and gaugings taken, as mentioned in the paper, and up to the present the results had been entirely satisfactory.

These ships generally, from the point of view of the comfort of the travelling public, represented a very distinct advance on earlier vessels. So far as the machinery was concerned, the chief contributory factors had been the complete and carefully thought-out system of sound insulation and the steps taken to isolate the machinery from the hull, not only at its foundation but at all the metallic connections. The painstaking care and thought which was given to the system of rubber chocks in every detail on the part of the author of the paper and the writer had so far been well rewarded—the hard metallic sound usually heard in the passenger accommodation near the engine room in this type of motor ship had been almost entirely eliminated.

A point which required a little more stress was the statement on page 23 under the heading of "cooling-water systems". It was there stated that before starting up, the circulation water should have a temperature not less than say 50° C. (120° F.). In the writer's opinion this heating up of the circulating water was of very considerable importance, not only in starting up to ensure a quick and easy start but also to spare the cylinder parts from undue heat stresses. There seemed to him to be an undoubted connection between cracking of covers and cold circulating water, and the hot circulating water was so easily obtained before starting by being used to circulate the generator engines.

Mr. F. Frenay (Soc. Anon. John Cockerill) wrote that on page 20, after referring to the evils of overrating an engine, the author mentioned the possibility of an engine suffering undue wear and tear because of underrating. There was undoubtedly a basis of truth in this remark. Not long ago the writer had a very unusual experience. One of his double-acting two-cycle engines, after a very brief time in service, showed signs of extensive corrosion at the piston rod sleeve. As all the collective experience over many years, both with two-cycle and four-cycle machinery, had shown a conspicuous absence of trouble with sleeves, it was something of a mystery. After intensive investigation into questions regarding quality of material, characteristics of the lubricating oil, fuel oil, etc., the writer came to the conclusion that the sleeve corrosion was specially strong when the engine was underrated. In this case the temperatures of cooling oil for pistons and circulating water were lower, and as was known with fast-running Diesel engines, moisture was at the root of the trouble. Opportunity was taken of modifying the lubricating oil circuits, so as to let the engine work "warmer" and the difficulty disappeared.

The design of stuffing box and scraper rings shown in Fig. 8 was interesting. It would appear that the scraper box design was a very good one. Regarding the stuffing box, the practice of the writer's firm was to use a built-up construction, each element of which carried a compound sealing ring. If carefully manufactured, experience showed that this design gave very good results.

With reference to the special materials used in the two-cycle engines, such as the heat-resisting steel cylinder covers, these gave no trouble whatever. The castings were made in their own foundry, where everything was under control, and the results to date had been all that could be desired.

The writer noticed that some of the engines illustrated in the paper had forged steel crankwebs, while others had cast steel webs. This would appear to be due to the preferences of individual clients. In the writer's opinion, between a cast steel web made in accordance with the latest technique, such as was adopted by his firm, and a forging, there was nothing to choose as regards reliability. It was only a matter of time before crankpin and crankwebs cast in one piece become standard practice. Such a design had much to recommend it—from the point of view of torsional vibration stresses, for example.

The remarks in the paper on the subject of the rating of the auxiliary Diesel engines in ships, and especially the number to be fitted, were very sound. From time to time one met ships in which the main machinery functioned perfectly but in which the generating engines were a source of continual anxiety to the engine-room staff, due either to their being too small or too few in number.

Mr. Wm. McArthur Morison wrote that because of its extent certain features of the paper evidently had had to be dealt with somewhat briefly. In particular, the system of pillarless construction seemed to offer considerable possibilities, and from a strength point of view it would be interesting to have the author's opinion as to its suitability for small craft. The writer had in mind the raised quarter deck vessel of about 170 to 220 feet with two holds, which with the present system of construction was inclined to be weak between the after end of the engine room and the after hatch. Subject to the extra weight, if any, not being excessive the pillarless system seemed to offer a desirable alternative.

The recommendation to have the pumps and other auxiliary services separate from the main propelling motors was sound, and, despite the first cost, it was suggested that it was also desirable for any vessel of the coasting type which had to be continually in and out of port; especially if the pistons were water or oil cooled, the circulating pump ought to be separate. In this connection the closed fresh-water system of cooling had much to recommend it, and no doubt as its advantages were appreciated it would in time become universal. As many owners and superintendents were not acquainted with it. a diagrammatic sketch of the system might be of interest.

The author's remarks regarding noise, especially in passenger vessels, were interesting. The generator motors were most often the worst offenders in this respect. The clatter from the cams and the whine of the blowers could travel far and be very distressing to sensitive ears, despite insulation of decks and casings, and one wondered if all that was possible had been done mechanically to eradicate this nuisance.

On page 9, the author mentioned the omission of non-return valves between the fuel pump and fuel valve. This was a moot point and the practice was presumably adopted to avoid the risk of excess oil being pumped into the cylinder. Against this there was the danger of an air lock occurring in the fuel line and the engine stopping at a critical time. It seemed to the writer that such a defect would show up quickly, and if the fuel valve lift rod was missed the exhaust thermometer either at the cylinder top or at the starting levers should show any unusual change of temperature in the cylinder. so that the lesser evil, the non-return valve, had its points.

The scavenge pressures given on page 10 were interesting, but it was doubtful in smaller engines if such pressures were always attained or, indeed, necessary. Possibly the tuned exhaust assisted and the writer had knowledge of several motors running satisfactorily with considerably less. Indeed in the case of a 60 kW. generating set, the engineer in charge experimented by taking a canvas trunk from a down take ventilating fan and was able thereby to get quite a satisfactory output.

The author's remarks on rating of engines were valuable to those who had to advise owners on the most suitable type of motor. Since m.e.p. and exhaust temperatures were apparently not to be regarded as constant equivalents, and the latter was a measure of the dangerous heat stresses which might cause trouble, it might be desirable to specify an exhaust temperature, compatible with good combustion of the fuel, which would ensure safe working conditions.

The advice regarding sea trial limitations of power was also sound. There was no doubt that much harm might be done by trying to "break things up" or take the maximum power out of a motor during sea trials and before it had been properly run-in. Indeed motor-car practice in this respect could be adopted with advantage.

Mr. Alfred J. Büchi (Member), in a written contribution, referred to a number of points in connection with supercharged engines, which the wide scope of the paper had prevented the author from dealing with in detail.

Because of the continually increasing severity of competition. all shipowners had to endeavour to equip their vessels with machinery that would prove to be the most advantageous in all respects. In this connection, in addition to dimensions, weights and first cost, an important part was played by service costs, including the cost of replacements, and also by reliability. If the time during which the services of a ship could be utilised was reduced, for instance through breakdowns in the machinery or due to the time taken for overhauls which had become necessary, which of course also entailed certain direct expenses, the vessel might prove not to be paying, even though the purchase price had not been high.

It was evident that double- and single-acting two-stroke engines were certainly the most suitable types for the very great outputs required to-day for big vessels. This could be readily understood, since four-stroke engines could not well be built for very great outputs without their dimensions becoming too great and their individual parts too heavy. But now, since the introduction of supercharging for this type of engine. it was possible that in many cases, where small and medium outputs came into question, there might often be good reasons for preferring supercharged four-stroke engines. When deciding which kind of engine was the most advantageous, taking everything into consideration, comparison must be made in such a way that the parts or features compared were really comparable. The four-stroke engine must not be built simply like the old heavy types of such engines, but according to modern principles and in a manner ensuring the greatest reliability which present experience showed to be possible. The engine must also be worked with such a mean service pressure that the chosen system of supercharging was fully utilised. With regard to this. it could be said that all systems in which the supercharging blower was mechanically driven by the engine itself were naturally more unfavourable than systems working with a blower driven by an exhaust-gas turbine, where the greater part of the supercharged output was obtained gratis from the energy that would otherwise be lost in the exhaust.

From experience gained with nearly 1,000 turbo-charged Diesel engines, many of which had been working satisfactorily for years, it might be stated that a mean effective service pressure of 110lb. per sq. in. might be maintained continuously in the engine cylinders without in any way adversely affecting the reliability of the engines in continuous service. It had been proved time and again that mechanical drive of a supercharging blower of any type whatever, in contrast to the drive by exhaust turbo-charging, must result in a reduction of the output of the engines at given average combustion temperatures. But this meant also an increase in fuel consumption by at least 5 to 10 per cent. at normal load, and still more at lower loads.

Unfortunately, when fitting marine four-stroke engines with exhaust turbo-charging up to now, full use had not been made of the increase of output permissible with this system, except in a few cases and these generally outside Great Britain. This was, however, to be recommended in the manner hinted at above. On the other hand, in stationary installations and in locomotives and railcars, it would be found that in first-class engines mean effective piston pressures of 115 to 120lb. per sq. in. were almost everywhere employed to-day. It was undeniable that the stresses in a four-stroke engine, whether caused by heat or by pressure, were smaller and/or also occurred less frequently, thus giving these engines an advantageous position in respect of reliability in service and minimum of wear. Such an engine might consequently also be run at a higher piston speed without the slightest danger In many cases also the plunger-piston type should be used to a much greater extent than hitherto. This type could certainly not be used for under-piston supercharging, where closing of the lower end of the cylinder was necessary; but this, unfortunately, made it impossible to have ready and convenient inspection and control at any time of the running surfaces of the cylinder and the piston, and it served also as an undesirable trap for collecting all sorts of fuel and oil residues coming from the combustion end of the cylinder.

Further, exhaust turbo-charging had also the inestimable advantage of automatically suiting the quantity of air to the quantity of fuel introduced. This distinctive quality arose from the fact that the exhaust turbo-blower was driven absolutely independently of the engine. If the output of an engine was raised by increasing the quantity of fuel introduced, there was also automatically an increase in the energy contained in the exhaust gases. In consequence of the higher pressure and of the higher temperature of the exhaust gases at the turbine inlet, the supercharging blower rotated more quickly, with the result that the quantity of charging air increased with increasing output; this increase in air quantity was to be ascribed partly to the increase in pressure of the charging air and, in the system with which the writer was associated, particularly to the scavenging being then much more efficient. Therefore, in contrast with all other engines including those working with mechanically-driven charging pumps or blowers, there was the phenomenon (advantageous for internal-combustion engines that had to contend mainly with heat problems) that. as the author had already mentioned, the temperatures during the working process and in the exhaust did not increase nearly so much at higher loads.

If, for example, the quantity of fuel injected was increased by 20 per cent., the quantity of air introduced with exhaust turbo-charging also increased by at least 12 per cent., so that, with increasing load, the fuel/air ratio, and consequently also the temperatures during the working process, did not increase nearly so much as in ordinary engines where the fuel/air ratio increased at least in proportion to the increase in the quantity of fuel. It was known that in a given engine the fuel/air ratio might not be increased beyond a certain value without adversely affecting the combustion, causing the engine to smoke, and getting into a region where the temperatures of the working process increased to an abnormally high and dangerous extent.

To-day, turbo-blower builders had got as far as to be able to build exhaust turbo-blowers of high efficiency for Diesel engines of 150 kW. and less. Consequently, the provision of exhaust turbo-charging for larger marine auxiliary engines, which were always plunger-piston engines, could also be recommended. In this way savings could be effected in dimensions and weights, as well as in fuel and lubricating oil.

It should further be noted that much less compressed air was required to start engines with exhaust turbo-charging than any other kind of four-stroke engine. Since the charging blower was not mechanically coupled to the engine and the cylinder dimensions were reduced. the required starting force was less; this was also expressed in the fact that the mechanical efficiency of such engines under continuous loading amounted to 86/87 per cent., in contrast to that in engines with mechanically-driven charging, where the efficiency was generally less than 80 per cent. In consequence of the exhaust turbo-blower being completely independent of the Diesel engine, the manœuvring qualities of the engine were also excellent. The exhaust turbo-blowers of large engines continued to run for about 10 to 15 minutes after the engine had stopped. Therefore, if the engine had to be started again within that time, it would start rotating easily in either direction in consequence of charging and scavenging air being still delivered by the blower.

With regard to the damping of noise, it might be pointed out that the damping of noise at the blower inlet, as well as at the exhaust turbine outlet. could be effected easily in these machines, since they only rotated and consequently worked uniformly. The silencer at the exhaust side need be only a fraction of the size of the silencer required for engines working without exhaust turbo-charging, when the gases left the engine in intermittent puffs. Hitherto, far too little use had been made of this advantage.

Mr. W. A. Tookey wrote that on the subject of engine rating the author had mentioned the fact that while the practical limit of output depended primarily upon (*a*) the quantity of fuel effectively burned, (*b*) mechanical and thermal stresses, and (*c*) engine speed, it was difficult to find a basis commanding universal acceptation. The author proceeded to comment upon the permissible mean indicated pressure which varied with different types of construction, upon exhaust temperatures, etc., and showed why scales of ratings based thereon were not entirely satisfactory as "yardsticks" for Diesel engine rating. With his remarks there would be general agreement, but the subject was so important that one wondered if there might yet be some other standard upon which engine rating could be agreed.

In a paper entitled the "Rating of Diesel and Other Engines", read and discussed at a meeting of the Diesel Engine Users' Association in November, 1932, it was suggested by the writer that possibly the composition of exhaust gas would be found useful as a basis for agreement upon "maximum" and "rated" outputs, and a chart was then given correlating exhaust temperatures with oxygen content in exhaust gas analyses for typical engines. At the time the suggestion was deemed likely to be of service, but as the exhaust temperature even when corrected to give temperature difference between inlet and outlet manifolds depended so much upon site conditions and upon design characteristics this, as the author had stated, was an unreliable factor for the purpose. From a study of heat balance sheets, however, it was known that while the quantity of heat passing away to the cooling system, by radiation and with the exhaust gases, might vary proportionately one with the other, the amount of heat converted into work was fairly constant in well-designed and properly-tuned compression-ignition engines, and conformed to the lower curve in the chart, Fig. 32, prepared from test results of both atmospherically- and pressure-charged cylinders. It would be noted that the mean curve of "work" passing through the plotted points having reference to pressure-charged engines was slightly higher than the "atmospheric" performances and that near the atmospherically-charged limits of mixture strength, say 30 to 35 B.Th.U. per cubic foot, this discrepancy was more pronounced—in other words the thermal efficiency per horse-power fell off at the higher ratings. From this it would appear that a mixture strength of 30 to 33 Qt. as plotted could well be accepted as the equivalent of rated output for atmospherically-charged engines. For pressure-charged engines the efficiency curve for "work" was greatly extended according to the degree of boost—a matter decided by the designer according to circumstances.

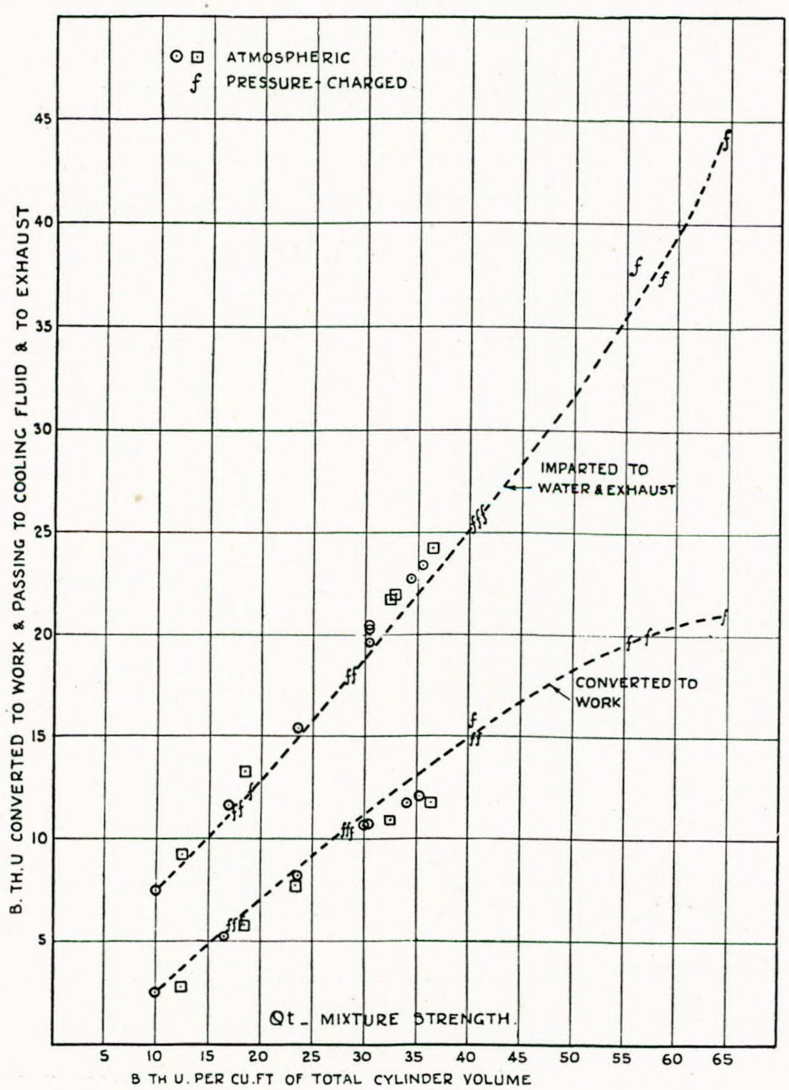


Fig. 32.

Fig. 33 had been prepared from data obtained from test-bed performances and it showed the relation between mixture strength and CO_2, H_2O and O_2 in exhaust gas analyses, with proportionate increase of the first two as the engine output increased and, of course, proportionate decrease of O_2. Experience had shown that trouble-free conditions of working were obtained so long as the limit of 8 per cent. CO_2 in exhaust gas was not exceeded, and, allowing 3 per cent. for water vapour, the oxygen content should be $21-11=10$ per cent. in round figures. To allow a margin for "rating" purposes the CO_2 content should not exceed 6·0 per cent.—equivalent to about 12·5 per cent. oxygen as indicated in Fig. 33 for atmospherically-charged engines and 7 per cent., equivalent to about 11 per cent. oxygen, for medium-boost pressure-charged engines. The reason for this would be evident from Fig. 34, for this illustrated that the effect of a pressure charge in engines equal to 8·25in. mercury above barometer was to permit the increase in the amount of fuel from 31·5 to 41 B.Th.U. or, say, 1·30 times with proportionate increase in output for the same CO_2 content—6 per cent.

In a paper "Combustion Efficiencies of Gas and Oil Engines" read before the Institution of Civil Engineers in November, 1937, it was pointed out by the writer that the performance of an engine based on mixture strength was fairly well established, and in Table XV of that paper the Tookey factor for a mixture strength of 30 Qt. (B.Th.U. per cubic foot of piston displacement plus clearance) was 2·72 and for 35 Qt. = 2·63; therefore, the mean indicator pressure to be expected for a mixture strength of say 32·5 Qt. would be $32{\cdot}5 \times \frac{2{\cdot}72 + 2{\cdot}63}{2} = 87$lb. per sq. in for atmospherically-charged engines. For 41 Qt. (pressure-charged) the Tookey factor of 2·55 would promise a mean indicator pressure of 104·5lb. per sq. in., an increase of 20 per cent. for the same CO_2 percentage (6 per cent.) in exhaust gas content, and as much as 47 Qt. Tookey factor of 2·45 = 115lb. mean indicator pressure or over 32 per cent. increase in power for a CO_2 content of 7 per cent. when compared with an atmospheric rating based on 6 per cent. CO_2 as before. With heavier boosting, when permissible, still higher

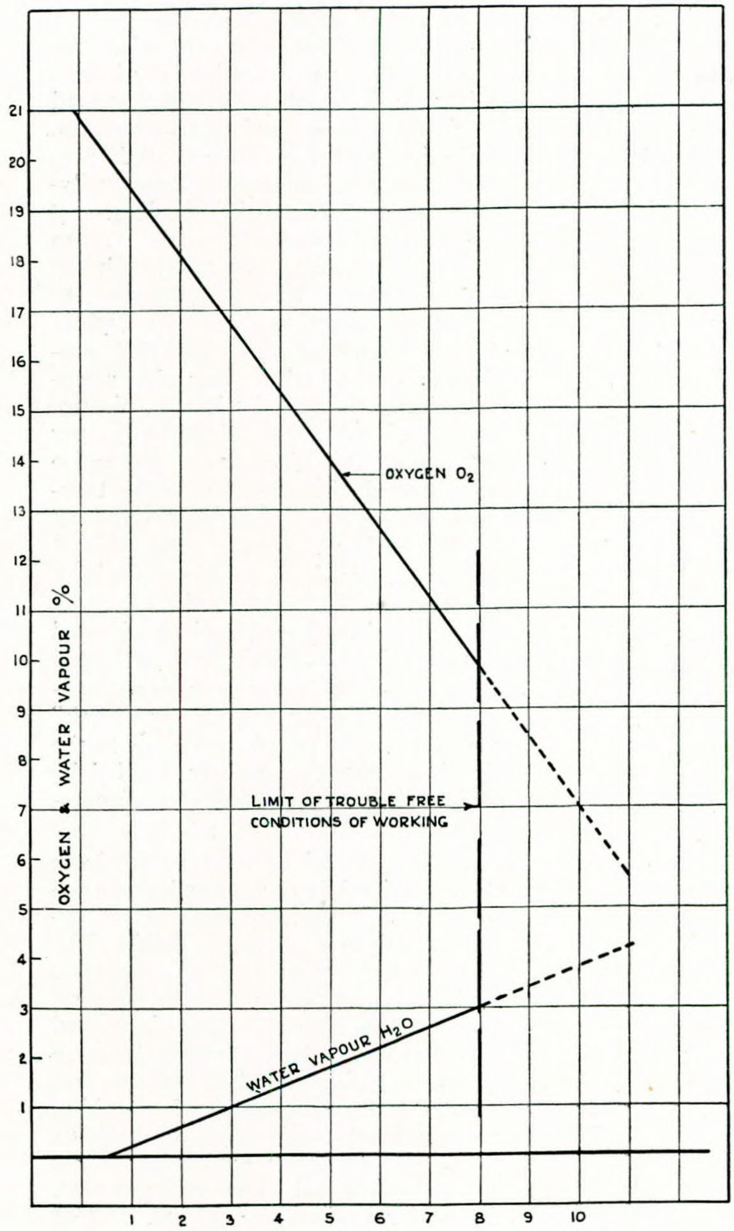


Fig. 33.

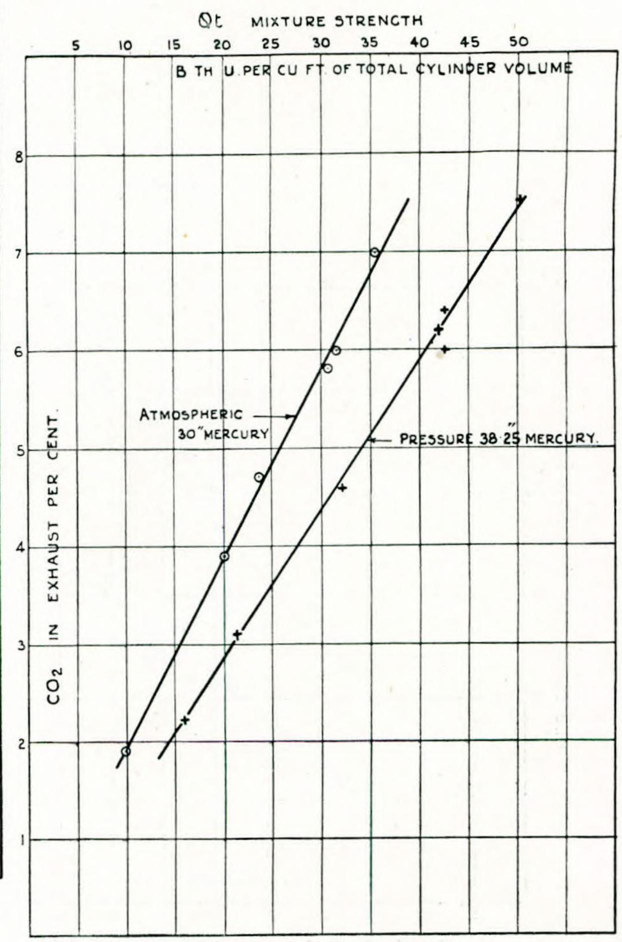


Fig. 34.

performances would be reached without the CO_2 per cent. limit being exceeded, and in this connection it might be mentioned that aeroplane engines with 6½lb. per sq. in. of boost were capable of giving on long continued flights mean *brake* pressures of the order of 170lb. per sq. in.—or over 200lb. mean indicator pressure.

It might be considered that the CO_2 basis of Diesel engine rating was not convenient because it could not readily be appreciated by prospective purchasers, but it seemed to possess some merit and might well be included as a contribution to the discussion on Mr. Pounder's excellent paper for his consideration and comment.

Mr. H. W. van Tijen (Koninklijke Mij "De Schelde") wrote that it might be remarked that the paper dealt exclusively with installations of which the main engines were directly connected to the propeller shafts. No reference had been made either to geared or Diesel-electric installations, although both systems were finding increasing application with Continental builders, even for large powers, on account of the saving in space and weight which could often be effected.

The author's remarks concerning the rating of engines were worth careful consideration. With a number of reliable types of marine Diesel engines now on the market, interest to-day chiefly centred on the question of what horse power could be got (safely and economically) out of a certain engine per unit of weight and space. Unfortunately, up to now, there was no generally accepted method of rating marine Diesel engines. Makers of the smaller internal-combustion engines often expressed the load as "horse power per litre of piston—swept volume". This figure, however, did not give a proper impression of the actual load and unduly credited the smaller dimensions.

Instead of expressing the load as $N_c \div Q_c$, it would therefore seem preferable to use as a rating formula :—

$$\text{Specific load, } N_s = N_c \div Q_c^{\frac{2}{3}},$$

where N_c=horse power developed by one working side of one piston, and

Q_c=volume swept by one working side of one piston ($\frac{\pi}{4}d^2s$) in litres.

As for a given engine this could also be written :—

$$N_s = \text{const.} \times pm \times v,$$

(pm=mean effective pressure, v=mean piston speed)

the above "specific load" reckoned with both m.e.p. and piston speed, and could be expected to be independent of the absolute dimensions of the engine.

Further, this rating formula was in agreement with two fundamental factors in connection with the load. Firstly, the value $Q_c^{\frac{2}{3}}$ was, for similar engines, proportional to the area of the scavenging and exhaust ports (or of the inlet and exhaust valves), and it was on these areas that the supply of the combustion air to the cylinder depended. Secondly, the value $Q_c^{\frac{2}{3}}$ was, for similar engines, proportional to the area of the cooled cylinder surfaces, and it was through these surfaces that a certain amount of heat must pass to the cooling medium.

Where H_c=the heat passed to the cooling water, per cylinder, in calories per hour, the value $H_c \div Q_c^{\frac{2}{3}}$ would be an expression for the mean heat flow per unit of cylinder wall area, and consequently for the temperature of the cylinder walls.

In Fig. 35 values of $H_c \div Q_c^{\frac{2}{3}}$ were plotted against N_s for two engines, of which results were published. Curve *A* represented a two-cycle engine, 5,500 b.h.p. at 125 r.p.m., whereas *B* represented a four-cycle engine of 1,200 b.h.p. at 700 r.p.m., supercharged on the Büchi principle. Further, curve *C* belonged to a two-cycle submarine engine developing 2,600 s.h.p. at 450 r.p.m. The continuity of the curves for three modern yet so very different types of engine was certainly striking, and would indicate a linear relation between the thermal load of the engine and the value of N_s.

When calculating the specific load, N_s, for marine engines of types and dimensions different from the published data, two-cycle and four-cycle engines appeared to fall well in line, provided the latter were supercharged and not restricted in revolutions by propeller considerations.

Apparently a moderate rating for continuous use at sea was of the order of :—

Installation.	Description.	No. of cyls. × cyl. diam. × stroke, mm.	Total b.h.p.	N_c	Q_c	$Q_c^{\frac{2}{3}}$	N_s
Author's figures.	Single-screw cargo, d.a. 2 c.	6×530×1,250	4,500	375	276	42·4	8 85
" "	Twin-screw cargo, d.a. 2 c.	12×620×1,400	14,000	583	423	55·5	10·45
" "	Twin-screw passenger, d.a. 2 c.	20×660×1,500	24,000	600	512	64 0	9 4
"Bessarabia"	Twin-screw passenger, s.a. 2 c.	24×620×1,150	11,500	479	347	49 5	9·7
"Brastagi"	Twin-screw cargo, s.a. 2 c.	16×650×1,200	8,000	500	400	54·1	9 25
"Oslofjord"	Twin-screw passenger, d.a. 2 c. (geared)	28×530×760	15,800	282	168	30·5	9 25
"Zwarte Zee"	Ocean-going tug, s.a. 4 c. supercharged (geared)	12×500×650	3,360	280	128	25·4	11·0

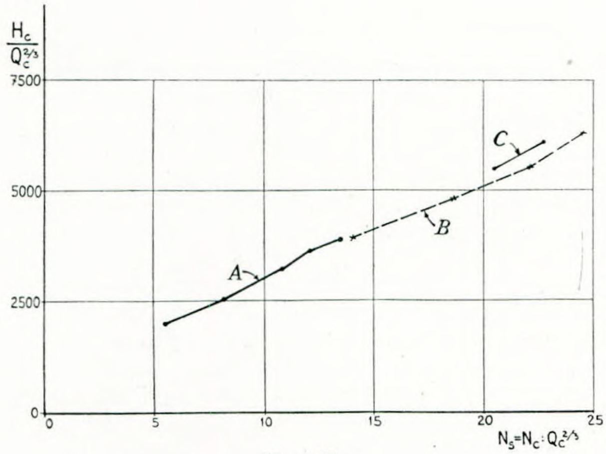


FIG. 35.

$N_s = 9/10$ s.h.p. per sq. decimetre.

To explain this more clearly the table on page 37 had been compiled, partly from figures given by the author and partly from technical data as published in the press.

Engines for fast cross-Channel vessels were loaded on trials with N_s equivalent to approximately 18, *i.e.* 18 or a little above or a little below, and submarine engines up to N_s approximately 22 s.h.p. per sq. decimetre.

It might be mentioned that the rating formula outline above could be used with advantage over a much wider range of internal-combustion engines than used for ship propulsion.

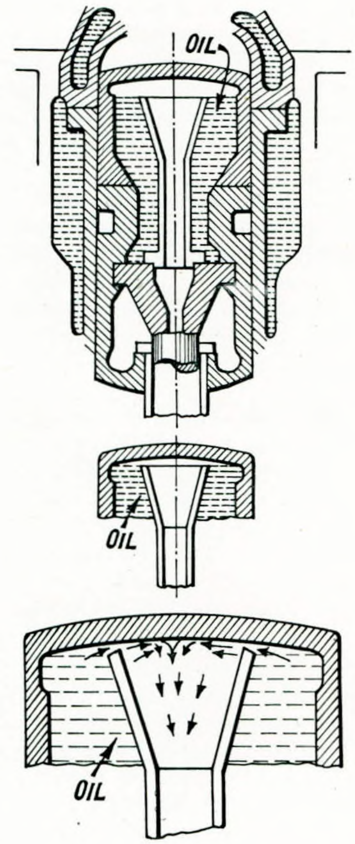


FIG. 36.

Mr. R. Oxburgh (Member), referring to Fig. 4, wrote that the large number of separate water pockets made one fear the formation of air locks, particularly after dismantling the parts for overhauls. Perhaps the author would express his views with regard to the fitting of air cocks to the water jackets, with internal pipes led to the top of annular spaces most likely to trap air.

In Fig. 4 the internal pipe fitted in the top piston appeared to be a considerable distance from the crown of the piston. When an engine came to rest after a long run, with one or more pistons on the top centre, and with combustion-chamber metals in hot condition, all were cooled by direct contact with cooling media except the crown of the piston. Other pistons at lower positions in their respective cylinders had a better chance of cooling off, as the heat radiated from the cylinder head was not so confined and there was a larger annular space above with larger cooling area. So the walls of the liner conducted the heat to the surrounding fresh water; further, the oil cooling the lower pistons was likely to be more effective, as it would surge over the internal stand pipes more freely than in pistons at or near their top centres.

The higher the internal stand pipe was to the crown of the top piston, the more oil would be in contact with the piston crown and so cool the whole piston uniformly; the returning oil temperature was then a truer guide to the piston temperatures. An arrangement as shown in Fig. 4 would permit the shutting down of the piston cooling oil, while the piston crown was still very hot. If the area of opening between the top of the internal stand pipe and piston crown was slightly less than the total area of inlet between the piston rod and sleeve, the top edge of the stand pipe might be chamfered, in order to direct the oil to the centre of the piston crown.

With reference to the stuffing box shown in Fig. 8, whilst these had been very satisfactory in certain sizes of engines, they had been troublesome in other sizes. To overhaul a defective gland entailed considerable time and cost; would it be possible to develop a gastight gland by using segmental rings held in place round the piston rod with garter springs? The conditions in a motor room would be greatly improved if the stuffing box could be quickly opened up and overhauled, particularly in trades where times in port were very limited throughout a voyage.

The author stated that the life of pistons was affected by frequency of starting and stopping and also by manœuvring; he questioned if the ship's officers were as kind to the Diesel engine as they were to the reciprocating engine. Assuming the statement to be correct, how did the builders help the ship's officers to be kind to the Diesel engine, not only in the past but even to-day?

Most of them remembered cold, frosty mornings in the workshops when the point of the chisel was broken under the first hammer blow, and a journeyman of wider experience said, "Always place the chisel in the gas or candle flame to drive the frost out and take the chill off before you use it". There was a parallel with Diesel engine pistons, but how did the builders take the chill off the pistons? The circulating water round the jackets received attention and, if there was sufficient time, reasonable heat was conducted to the pistons, but the piston-cooling oil from the chilly double-bottom lubricating oil tank maintained the pistons in a cold condition ready for starting the engine. The preheating of the lubricating oil would help the ship's officers.

With fresh-water cooled pistons it was a simple matter to blow steam into the system whilst circulating the pistons with water gradually rising in temperature. This had been done with very satisfactory results, resulting in easier starting and less frequent piston failures. The heating of oil was not quite so simple or inexpensive.

In the early Diesel ships the speed of engine rotation on starting air was so slow, the pistons so cold and compression temperatures so low, that the ship's officers had of necessity to push the starting lever well over on to fuel to avoid a false start and consequent loss of blast air. The boiler was not lit except on very cold occasions, as additional units required extra attention and the Diesel engine room had not the personnel of a steam ship. The preignitions that occurred with heavy charges of fuel were not beneficial to pistons.

The ship's officers would handle the engines just as they handled reciprocating engines if they had:—

(*a*) A Diesel engine quick turning on starting air giving high compression temperatures, thus needing a small charge of fuel per injection to continue the rotary movement of the shaft;

(*b*) A simple means of removing the chill from pistons, by circulating pre-heated oil through the pistons before starting the engines.

The latter point was most important and should receive special consideration as it had a direct bearing on point (*a*).

The engines of a steamship were warmed through with very little steam relative to the amount available from boilers, with their special attendants. The position was completely reversed in a Diesel ship.

To overcome this difficulty a waste-heat boiler heated by exhaust gases from the generators would supply sufficient steam to heat the enclosed freshwater cooling system and a quantity of lubricating oil for warming the pistons, without using the oil fuel installation on the boiler.

With lubricating oil stored in double-bottom tanks it would be impossible to heat the total volume. and the diagrammatic sketch (Fig. 37) briefly illustrated a suitable arrangement. During

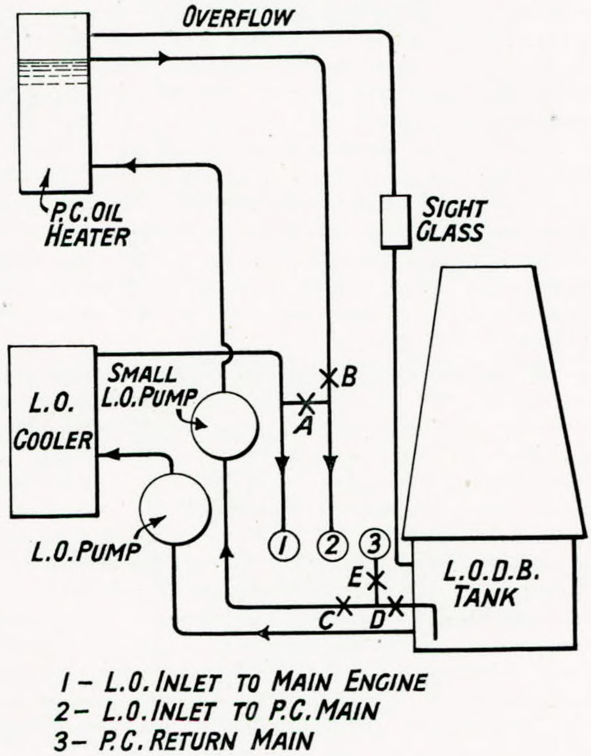


FIG. 37.

the warming-up process the small lubricating oil pump would draw from the double-bottom tank with valves D and C open and valves A, B and E closed; as soon as the heater overflowed to the double-bottom tank valves B and E should be opened and D closed. The small pump would then circulate a minimum supply of lubricating oil through the heater and engine pistons.

Prior to the manœuvring of main engines the large lubricating oil pumps were in service for the bearings; when the oil was reasonably warm, the oil heater might be shut off, valve D opened and valve C closed, valve A opened and valve B closed and the small lubricating oil pump stopped.

Dr. H. H. Blache wrote that the installations described in the paper were of great interest; they included the largest engines yet made for marine work of the two-stroke double-acting type with which the paper was concerned; further, mention was made of the pioneer work of the author's firm to obtain improved hull design and improved fitting of the engines to the hull structure to prevent vibration and noise, all of which were matters of general interest to all British and Continental builders of motor vessels. The subject was dealt with mainly from the builders' point of view, based upon many years of experience in the building and running of Diesel engines for British owners.

The author had necessarily had to restrict the scope of some of his remarks in view of the large number of engineering questions dealt with in the paper. Accordingly, the writer thought it would be of interest to make further comment upon some of the points, especially in view of the developments since his own paper* was read before The Institute in 1936. Such further remarks concerned the experimental design of piston shown in Fig. 17, the question of crosshead and gudgeon pins, anchored piston rings, caulked carrier rings, liner surface, etc. The design and manufacture of these important parts of a Diesel engine were all bound up with modern views on such subjects as lubrication and lubricating oil films.

It had been fully ascertained that sliding surfaces could rest on an oil film under great pressure without touching and be able to slide with a very minimum of floating friction. The thickness of such oil films might be as small as say 10 oil molecules—equal to about 1/1,000,000th in.—without the surfaces coming into contact. This fact proved the importance of having the highest possible degree of exactness, also the finest polish of the surfaces. To meet this demand for exactness in a practical way, one of the sliding surfaces could be made of a soft material such as white metal or lead bronze. the soft rough surface being able to yield to the other hard exact surface under the pressure of the oil film without coming into metallic contact.

Cylinder liners of the dimensions dealt with in the paper were expanded at the top end by the heat and even in new, unworn liners, the piston rings had to work in and out each stroke, pressed hard against the lower surface of the grooves by the compression and combustion pressure. It was obvious that to stop this movement by a pin in the cut or close to the cut, forcing the ring to a circumferential movement, was a mistake which would result in broken piston rings. Accordingly the use of unpinned piston rings, or piston rings with stoppers opposite the cut were essential. the last-mentioned to have a full, *i.e.*, unweakened, section such as the anchored rings shown in Fig. 27.

To reduce the wear of the lower surface of the piston ring grooves all piston rings were ground on the lower surface. This, however, could not be done at the surface of the usual grooves and it was peculiar to find—although nevertheless a fact—that the lower surface of the grooves never was worn plane by the ground lower surface of the piston ring but conical. *i.e.* wider at the circumference of the piston. When the piston was fitted with ground carrier rings the plane smooth ground surfaces of piston rings and carrier rings were properly resting on each other from the beginning, separated by a thin oil film and no vertical wear at all took place in service.

The purpose of grinding a cylinder was mainly to obtain exact cylindrical bore, important for cylinders of small diameter. As mentioned above, cylinder liners of the dimensions dealt with in the paper expanded due to the heat and accordingly might change the exact cylindrical shape attained by grinding.

The piston rings, having only a small wearing surface compared with the liner, were the most-worn parts even when manufactured of a quality of cast iron harder and more wear-resisting than the liner. Anchored piston rings gave superior results as they were able to wear themselves to the shape of the liner in running condition. To keep the oil film on the liner and between the surface of the piston rings and liner in order to prevent wear and reduce friction loss the finest possible polish of the cylinder liner surface was essential, but not more so than for piston rings and grooves, the cylindrical and plane surfaces of which should be exact.

Cylinder liners of the right quality of cast iron, turned to a good finish, after a short period of running-in on the test bed showed such a high degree of polish that no further improvement could be expected by the use of ground or polished cylinder liners.

Gudgeon pins did not distort and accordingly the most exact cylindrical shape was essential as well as the highest polish of the surface, both to be attained by case-hardened steel—which was superior even to high carbon steel as used in crossheads of the usual marine type (Fig. 18A) which was too soft to be ground.

Preference was usually given by marine engineers to crossheads of the above-named type as being easily overhauled and re-aligned for wear, which re-alignment had in any case to be very carefully executed, especially for two-stroke single-acting engines.

It was better to have a design of crosshead not exposed to wear at all and not to require any re-alignment, such as the crosshead shown in Fig. 17, in which there was a single white-metal lined steel or bronze bearing fitted in the closed head of the connecting rod and a case-hardened ground and highly polished gudgeon pin fitted as shown in Fig. 17 for a trunk piston engine of the type shown in Fig. 15. A corresponding design could be fitted to the crosshead slippers and lower end of the piston of a single-acting engine of the type shown in Fig. 18A, or any usual type of double-acting engine. The engines for m.v. "Selandia" built 28 years ago had a system not unlike that of Fig. 17 for supplying lubricating oil to the gudgeon pin and the cooling oil to the piston, through common internal holes in the connecting rods and crankshaft for the main bearing, but having a single free outlet from the piston. The resistance to the passage of the oil up to the piston cooling space was found to be too

* "The Burmeister & Wain Two-Stroke Cycle Engine". Vol. XLVIII of Trans., Part 10, November, 1936.

great and telescopic pipes were adopted in later designed engines. Although telescopic pipes were fully reliable and satisfactory they were a complication to the engine, and it was an interesting speculation that if the original design adopted in the "Selandia" had been developed further the whole telescopic gear could have been spared in all the trunk piston, as well as crosshead, engines built since the time of the "Selandia". The special design used over a number of years in two-stroke engines wherein the main bearings and bottom end bearings for oil holes bored in the crankshaft at approximately 90°, as shown, for example, in Fig. 15, allowed for the full and free passage of the oil, an arrangement which was adequate for the design of piston in Fig. 17.

As the above questions raised by the writer concerned engine parts common to all types of marine Diesel engines, it was hoped that the above views might be of some general interest.

Mr. B. Giljam, Jun'r. (P. Smit, Jun'r., Rotterdam) wrote that the author had given a very detailed description of such engine types as had been constructed at Belfast and installed in practically every type of vessel. By giving a description of several installations, the author did not emphasize the merits of single-acting two-stroke machinery of a larger size. As the writer had had an opportunity of watching more closely the performance of these engines he would say a few words on the trunk-piston design. The engine had been constructed specially on the Continent in large size units, up to 12 cylinders, and had done exceedingly well. The bore being 620 mm. with a stroke of 1150 mm. it developed 500 b.h.p. per cylinder at 125 revs. The engine showed the same simple construction as that illustrated by Fig. 15, which drawing made clear that it represented a unit capable of cheap manufacture.

No troubles had been encountered either with the exhaust valve operation or with the gudgeon-pin construction. After four months' service an inspection showed that the white metal in the small-end bearings was in perfect condition. These engines were now uniformly made with the lead-bronze rings fitted to the pistons as mentioned by the author on page 24. These rings contributed greatly to shorten the running-in period. When initially the rings were touching the liner surface, the space between the rings would prevent a certain amount of lubricating oil being scraped away by every stroke of the piston. In addition, any small particles of iron or dirt were caught by the lead-bronze rings and might be found subsequently embedded in the material of the ring after running-in.

In the case of the large trunk-piston engine, the cam-shaft was chain driven and the reversing gear was of another construction than that mentioned by the author on page 15. The exhaust valve opened at 68° before bottom dead centre and closed 56° after bottom dead centre, the piston being 6° before bottom dead centre for maximum lift of exhaust valve (centre line of cam) instead of 10° as with the type described by the author. This opened up the possibility of having a reversing gear with a fixed position of crankshaft and exhaust cams relatively, and the gear was simplified insofar that only the position of the fuel cams was to be changed when reversing. This was done in a manner similar to that shown for the double-acting type described on page 10 of the paper.

For large outputs it was understood that the author preferred the double-acting type of engines as described on page 2 under (*a*) and illustrated by three typical examples. There were utilized three types of engines with respectively an increasing amount of s.h.p. per cylinder.

It was stated on page 12 that the new type of double-acting engine developed 1,000 b.h.p. per cylinder, *i.e.,* the size was between those of the first two examples, respectively 750 and 1,150 s.h.p. per cylinder. Was the author in a position to state whether the new type of engine was meant to be applied in both the above-mentioned examples, *i.e.* with corresponding number of cylinders, or was it the intention to develop finally one or more standard types of larger size to cover the various cases of shaft horse power required? It would be specially interesting to know whether the author had in mind a large type suitable for the vessel mentioned in the third example.

Mr. Harry Hunter (Member) wrote that it was noted that all three typical double-acting installations had double-acting engines with an even number of cylinders, and therefore he presumed would have pairs of cranks opposite. If so, then the engines would have simultaneous firing of two ends, a feature which some would prefer to avoid by having an odd number of cylinders; indeed the author mentioned the very point on page 23. Probably there was some other factor or factors, not immediately apparent, which had led to the adoption of even numbers of cylinders, and which perhaps the author might be able to explain.

In installation (*a*) it was noted that such auxiliary steam as was required was obtained from exhaust gases; presumably the only steam required at sea was for the dynamo, boiler auxiliaries, steering gear and some domestic purposes. Hitherto it had been generally found that the heat content of exhaust gases from 2 S.C. engines barely sufficed for this purpose; presumably, however, the comparatively high rating of the author's engines would ensure a greater steam generation than usual.

On this subject of steam generation from exhaust gases, it might be of interest to state the following information in connection with an installation by the writer's firm of 4 S.C. Büchi engines, and otherwise generally the same as the author's examples (*a*) except that the power was 2,100 b.h.p.

Particular care was taken with the steam side so as to ensure adequate steam generation at reduced powers, and as a result it had been found possible in service, for example, to maintain steam at 70 per cent. of full power.

Great interest was attached to the author's information on rubber mounting of two-stroke engines in the 17-knot cross-Channel vessels. Presumably the vessels were now in service and the author might be able to express an opinion on the efficacy of the arrangement.

The writer would like to put forward a suggestion in this connection which might be of interest from the point of view of preserving alignment. In a reciprocating engine the dynamic balance was usually obtained by arranging for opposite couples and forces to cancel out. This they did through the framework which in consequence bent—the lighter the framework the more flexion and *vice versa*, and with moderate or fast running engines it was impossible to avoid some flexion. In the simple case of a six-cylinder engine with symmetrical cranks the flexion occurred about two nodes—approximately under Nos. 2 and 5 cylinders respectively. Solid chocks could be fitted in these node localities without any dynamic flexion reaching the hull; elsewhere, chocking might be dispensed with or flexible chocking used. The proposal would ensure alignment and at the same time permit the engine to flex freely and independently of the hull.

Table III in the paper contained some information seldom given but often sought. Could the author oblige by adding displacement and powers, ahead and astern.

Mr. H. P. Christensen (Elsinore Shipbuilding & Engineering Co.) wrote that for passenger-carrying ships the ideal of complete absence of

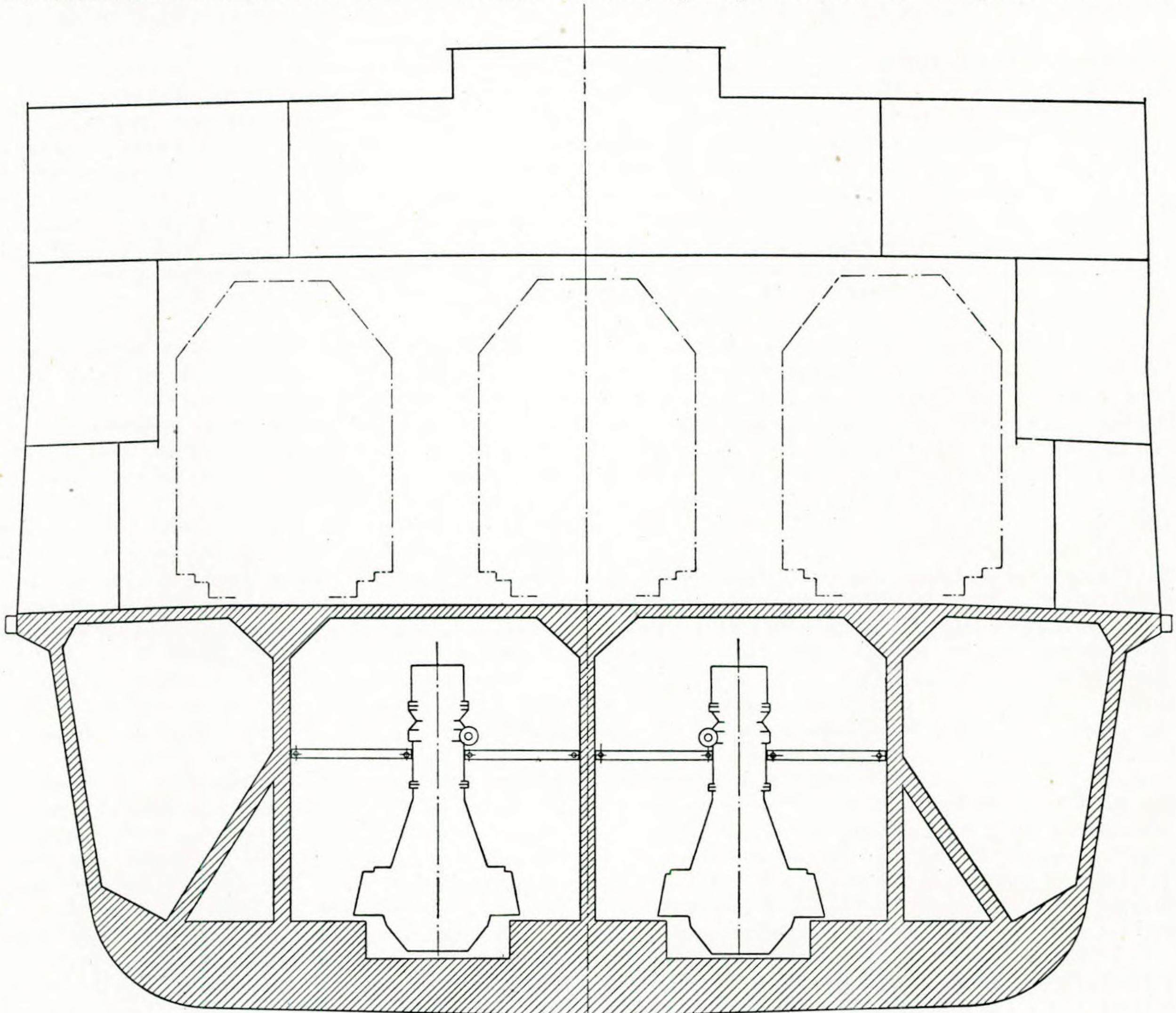


FIG. 38. Section through engine room—Diesel ferry.

noise and vibration should be kept continually before the minds of naval architects and engine designers. It must be admitted that until recent years it was almost axiomatic to regard a certain amount of vibration and noise as inevitable for motor vessels. The time had fully come when this mental attitude should be realized to be not in accordance with fact.

Great progress had been made by means of collaboration between naval architects and engine designers, the frequency of vibration for passenger decks and the scantlings for these now being determined theoretically in advance, so that the frequency relative to that of the engine was suitably established, and it might be said that modern passenger motor vessels, correctly designed, were now on a level with steamships.

The present paper was valuable for the mention made of some of the successful methods adopted for engine-room construction, machinery foundations, etc. Further to the subject of the design of pillarless engine room illustrated by the author, Fig. 38 showed a design which was being used in the broad, low ferries of the Danish State Railways, where the double bottom was connected with pillars and diagonals to the main wagon deck, so that in way of the engine room the vessel formed a rigid framework on top of which deckhouses and passenger saloons had been fitted practically free from vibration. One felt that much more could have been written on this subject.

In Part III of the paper many things were stated which had required saying for a long time. The leading problem which arose with any Diesel machinery undoubtedly was the margin to be allowed between the builders' top continuous rating and what was expedient for any particular shipping schedule. This must be decided upon by taking into consideration the service conditions of the ship, especially with regard to the time in port, the facilities available for maintaining the machinery and the skilled assistance available.

The writer agreed with the mean pressures stated by the author to be attainable with the engines forming the subject of the paper. But whether, say, a single-acting two-cycle engine should be run with an m.p. of 7, 6·75 or 6·5 kg.cm.^{2} depended upon such factors as those mentioned.

Several of the passenger and fruit vessels built by the writer's firm in recent years and fitted with the 500×900 mm. single-acting two-stroke engine shown in Fig. 15, had in regular daily service sailed with a mean pressure of 6·75 at about 155/160 r.p.m., corresponding to 410/420 i.h.p. per cylinder. The results had given every satisfaction.

Mr. I. Yamashita (The Tama Shipbuilding Co., Japan) wrote that on page 7 it was stated that the cylinder cover and the cover jackets were normally built separate, but that occasionally the jacket was cast with the cover. What would be the benefit of making it in one block in spite of its supposed high expense, difficulty of casting and also high repair expense?

On page 9 the author suggested, for cooling the fuel valve, having an independent circuit comprising pump and tank, which would enable lower grade fuel oil to be used. Would it be possible to use the same fuel oil primary pump for cooling purposes by increasing its capacity?

For the newly-developed engine shown in Fig. 14, should nearly the same difference of mean pressure be expected between the top and bottom cylinder due to the slower piston speed in dead centre and also the lower cooling water temperature in the bottom cylinder, as in the case of the 620/1400 mm. engine where the diameters of the top and bottom exhaust piston were different?

With regard to the remarks on page 8 about the scraper box and stuffing box of the piston rod, might it be assumed that the thin film of ordinary lubricating oil was quite sufficient to protect the piston rod from wear?

On page 13 it was stated that the piston had a special steel crown bolted or screwed to the cast-iron skirt. An engineer at sea faced with replacement would prefer the bolted-on type of crown, as the change could easily be made. Against this, the screwed-on design was naturally what a designer would prefer because it ensured tightness of the piston against lubricating oil leakage under the high temperature prevailing at the piston crown. Faced with these two points of view, what a designer should do was an interesting problem.

In Fig. 21 showing a supercharged four-stroke engine the piston was shown with a plug. It would be interesting to have some information on the latest practice regarding the design and fitting of plugs to such engines, which form of thread should be adopted, the material of the plug, etc.

On the supercharged engine in Fig. 21 automatic non-return valves, presumably of a very light construction, were shown. Would not the engine have a somewhat higher mechanical efficiency if such valves were mechanically operated?

On the eccentric system generally adopted for two-cycle double-acting engines, could this width of eccentrics not be made less in view of the excellent system of lubrication adopted, for example as shown in Fig. 14?

Mr. S. B. Freeman, C.B.E., M.Eng. (Vice-President), in a written contribution, asked whether the pillarless engine room construction was as free from danger of vibration as the previous type with pillars between the tank top and the upper structure.

One of the features of the engine which the author described as built in some works was the shallow bedplate sitting on deep floors. It was suggested that the cost of the deep floors, which had to be riveted in a manner equal to the highest class of boiler work, was greater, and that a deeper cast-iron bedplate with a shallower floor would result in a more economically built ship. It would also do away with the difference in level between the engine-room floor and the tunnel floor, which was awkward to manage. If a forward tunnel was fitted, the difficulty was increased.

The rubber filled chocks appeared to be satisfactory, and in the writer's own experience the fitting of rubber sleeve holding-down bolts to auxiliary engines running at high speed had resulted in a great diminution of vibration.

There was a large number of small points, which, taken together, made for the satisfactory working mentioned in the paper, as, for example, the fact that the crosshead shoe was lubricated and not the guide plates, so that the crosshead was equally effective on the guide plate and guide bars.

The change from nickel-chrome steel to chrome-molybdenum steel for cylinder covers appeared to be a step in the right direction, and the author might perhaps tell them what the statistics showed in the comparative immunity from cracking in these two types.

It was noted that the power obtained from the exhaust pistons was of the order of 10 per cent. The new type of engine which had exhaust pistons of the same diameter as the cylinder (it would be an advantage if the bottom exhaust piston were smaller, and the top exhaust piston were larger than the main piston, so that all three could be withdrawn with ease from the top of the engine) would presumably return more than 10 per cent. of the power exerted in moving them to the crankshaft. It would be of much interest to know what this percentage of power would be.

It was found that among engineers there was a hesitancy to accept the proposition that power could be transmitted through an eccentric drive as readily as by a crank. Would the author please deal with this matter authoritatively and show what were the differences between the two applications of force to the crankshaft.

The cooling of the fuel nozzle by circulating oil either from the incoming fuel or from an independent circuit was undoubtedly good, especially in burning the lower classes of oil.

The overhauling heights of this type of engine were too great to be popular with the shipowner and naval architect. It was understood that with the new 550×1200 engine, although the overall height was 3ft. less than the 620×1400 engine which preceded it, the overhauling height was no less than 3ft. more. This was a very serious matter.

The Author's Reply to the Discussion.

The author had been surprised by the international nature of the discussion, the contributors to which included so many eminent Continental engineers—from Switzerland, Belgium, Holland and Denmark. Apart from this its representative nature was also noteworthy. It had been a revelation to the author to observe the grip which the work of The Institute had upon so many people. Amongst the younger contributors might be mentioned an engineer from the Tama Dockyard at present resident in Europe, and an enterprising young man who came from Rotterdam to be present at the meeting.

Many of the contributions were in the nature of brief theses, to comment upon which would be to detract from their value. Accordingly the author confined his remarks almost exclusively to replying to specific questions raised.

It was a personal pleasure to the author that the discussion should have been opened by Mr. Burn. With reference to the single-screw cargo ship mentioned on page 2, where it was stated that the engine-driven pumps were duplicated as independent units; this had reference only to capacity. It was sometimes possible to adopt the same pump-ends for engine-driven and independent pumps if the latter were electrically-driven. In the example mentioned in the paper the independent pumps were of the steam-driven, direct-acting type.

Regarding the saving in engine room length mentioned on page 2, where two machinery sets for large refrigerated cargo vessels were compared, one comprising two-cycle double-acting engines and the other four-stroke single-acting supercharged engines: the comparison had reference to respective sets of machinery of the same relative rating, the same robustness of design, etc., as was the owners' custom to receive. There was no question of introducing welded structures. Mr. Burn stated that figures of 120lb. per b.h.p. could readily be obtained in an orthodox double-acting engine of 5,000 b.h.p. with a welded structure. For the double-acting engines mentioned in the paper, with welded construction, a specific weight well below the figure named by Mr. Burn could be obtained without difficulty on a conservative rating basis. While low ratio of cylinder centres to cylinder diameter might be important for crankshaft stiffness—which was the reason for referring to it in the paper—it was no criterion of power developed. An engine such as Fig. 3 having ratio 2·36 could be expected to be a smaller engine for the same power than the orthodox engine mentioned by Mr. Burn having a ratio of 2·0.

The double-flow type of double-acting two-stroke engine had never seemed sufficiently attractive to the author's firm to be considered seriously. Contrary to what Mr. Burn stated was his experience, and perhaps that of other builders, the piston of the two-cycle double-acting engine had never given the slightest difficulty.

Mr. Burn stated that the Burmeister & Wain engine had perhaps the unique distinction of having by far the greatest weight of moving parts, compared with the total weight, of any double-acting type, and it would appear that only at the expense of mechanical complexity of these parts had the undoubtedly high combustion efficiency been obtained. This remark, in the author's opinion, erred on the side of overstatement. It should be remembered that the total weight was small, and in surveying the design as a whole and bearing in mind its very high, safe output, it could not be said to be complex; rather the contrary. One important point was that those parts with which the owner was concerned, and which in some designs involved him in considerable expense, were in the designs with which the paper was concerned made simple. The cylinder liner was an example.

In considering questions of noise and vibration, it should be pointed out that however stiff the tank top and bedplate might be made, when pillars were fitted they would always be a means of sound transmission into the regions of the ship outside the engine room and omission was desirable wherever possible. It was unsound, in the author's opinion, to reckon on the engine as assisting the hull. The ship should be designed to do what was required of it and the engine to fulfil its function as a propelling machine without being called upon to reinforce the ship. Fig. 39 showed an arrangement which superficially would appear to be attractive. The diagonal struts, the stays between the engines, the engines themselves, and the double bottom of the ship constituted a girder. But sooner or later such

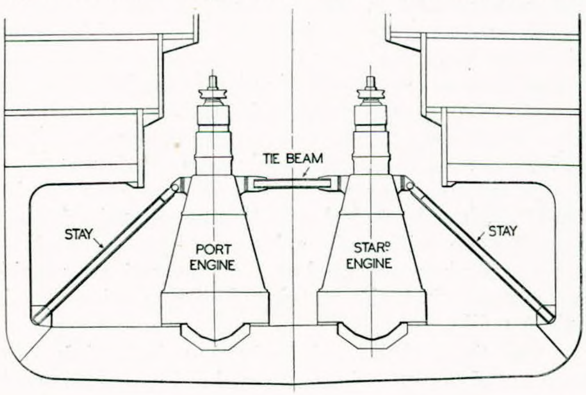


FIG. 39.

a proposal, or others of the same broad category, could be expected to involve the engine in some form of trouble. It might be—and often was—very sound to adopt a flat-bottomed bedplate as in Fig. 21, when questions of individual preference, etc. arose, but in such circumstances the double-bottom stiffness should be independently sufficient.

Regarding the best design of rubber chocks, there was probably not as much scope as might at first be thought. As Mr. Burn correctly stated, if rubber was too much restricted it became virtually solid and then its vibration-damping properties were not fully realized. On the other hand, if the rubber was not restricted and was allowed to flow, the engine gradually settled, the holding-down bolts required more attention, and alignment difficulties might appear. A plain sandwich of rubber bound to two light steel plates might give satisfaction if the vibration frequency was low, but for higher frequencies the author had seen arrangements of this kind where there was no damping effect.

Mr. Burn referred to single-acting engines constructed on the uniflow principle, in which a piston was used, and stated that eventually it was to be expected that the stroke of the upper piston would tend to increase. By so doing, however, the height of the engine was increased, which was a serious disadvantage in passenger ships constructed in accordance with the ideas of the most progressive owners and builders.

On the subject of the relative merits of splash cooling and leading the oil rapidly over a heated surface by means of baffle plates, it was more than twenty years ago since this matter was fully investigated by Messrs. Burmeister & Wain, and the free unguided oil cooling of pistons was found to be definitely superior by reason of the intensive whirling motion of the oil set up by the acceleration of the piston.

With reference to radial oil holes in crank journals and pins, if other things were equal it was always advisable, as a matter of sound principle, to avoid holes. But if, as in the present instances, careful experimental investigation showed that the effect of such holes under working conditions was fully negligible there was then no virtue in foregoing the valuable function fulfilled by them for the sake of a technical truism.

Mr. Burn's remark that the scavenge pump had tended to become rather more complicated was not understood. Since the early engines of the type were made years ago a number of simplifications had been introduced. On the general question of rotary blowers versus reciprocating blowers the author had no prejudices. Analysis of all the issues involved had seemed to show that the rotary blower fulfilled the greatest number of requirements. For example, there were rotary blowers in service now for almost ten years without a penny being spent on them. There was nothing to examine, nothing to overhaul. Incidentally, it was observed that certain competitive makers of good-class engines were leaving the reciprocating for the rotary blower. There was certainly no objection to supplying reciprocating blowers to clients who desired them.

In Figs. 13 and 20 the horse-powers were metric; in other places in the paper British horse-powers were used.

It was rather premature to state the difference in performance between the standard and the latest type of double-acting engine. The standard type of engine had a combustion efficiency which was so good that it was hardly practicable to expect anything better. The present indications were that the new engine had a combustion efficiency at least as good as the standard engine. This probably met Mr. Burn's point arising from the fact that the combustion spaces were quite different.

Mr. Burn asked why the camshaft in the two-stroke auxiliary engine, Fig. 25, was not near the exhaust valve as in the four-stroke design of Fig. 21. Fig. 21 was a slow-moving main engine in which the camshaft could be safely exposed, and therefore the cheaper arrangement was adopted, but in Fig. 25 where it was necessary that the camshaft and adjacent parts should be in the crankcase the general compactness of the design had to be considered. The difficulty which Mr. Burn mentioned had not hitherto been experienced. It was noted that his high-speed engine had a single exhaust valve.

The essential purpose of anchoring piston rings was to avoid wear of the rings and maintain gas tightness; only incidentally did it obviate any possibility of the cut ends coming into line. The piston rings near the top of the piston, which worked over the upper part of the cylinder liner, were increased in diameter by the temperature and the wear of the liner, and accordingly at every stroke there tended to be a slight movement around the circumference of the rings. By adopting a substantial anchorage diameterically opposite to the cut, sliding could only only take place on one-half of the circumference at each side. The anchored rings assumed, exactly, the shape of the cylinder liner which always changed its shape in service by reason of heat stresses. The customary method of pinning a ring was to have a small round pin in the neighbourhood of the cut. This arrangement could be a source of endless trouble and no doubt was the reason why many makers of two-stroke engines had discarded it. Anchoring of rings in the manner described in the paper was, however, a very different proposition.

Mr. Burn's proposal for a percussion element in a hydraulic spanner for removing piston rod nuts was very sound. Presumably the device was used only for slackening, and not also for tightening. as for the latter it would tend to abrogate the notion of tightening to a well-defined limit.

The author agreed with Mr. Carter that the four-cycle engine, supercharged on the Büchi and on the under-piston principle, would be likely to remain popular for years to come. For robustness and ability to withstand very severe conditions for prolonged periods it was hard to conceive a more suitable propelling engine. The qualities which it possessed were in most instances well worth any extra weight which was put into it.

On the subject of simplicity in design: simplicity, it seemed to the author, was the essential characteristic of anything—or anybody—that was really first-class. Unfortunately to make a thing simple—or to get other people to make it simple—was probably the hardest task in life. A country where design was really understood and practised and which had a long praiseworthy tradition was Switzerland. There was no mystery about why it should be so. The explanation was very clear to those who had studied closely the subject of machine design.

The author was much interested in the remarks of Mr. Carter regarding the engine shown in Fig. 14.

On the subject of current tonnage measurement mentioned by Mr. Hardy, it was true that a good deal might be said concerning it, but in dealing with this subject the author would be tempted to extend the discussion much beyond the limits intended by the paper. Incidentally, however, it might be pointed out that, apart from tonnage measurements, any additional space allotted to cargo-carrying as a result of economy in engine room space meant additional revenue-earning power for the vessel. Mr. Hardy's recognition that the marine Diesel engine owed to Belfast no small amount of its present success was appreciated.

In the first paragraph of his remarks Mr. Wheadon summarized very clearly the points which it was the author's desire to drive home on the subject of rating.

It was natural that Mr. Wheadon should have expected that the engine described on page 12 should have been dealt with at greater length. The subject matter of the paper, however, was a description of installations which had recently been put into service and to that extent it was desirable that the reference should be brief. The chief point about the new double-acting, coverless, engine was that it probably represented quite a considerable step towards the ultimate type of engine. At present there were many different types of marine Diesel engine, as long ago there were many different types of marine steam engine. But, in time, the marine steam engine settled down to one, clearly-defined, general type with only sub-variations of detail as between maker and maker. It seemed reasonable to suppose that the marine Diesel engine would follow a like course.

On the subject of reduction in weight and space, and as to whether production costs would benefit, the author would prefer for the moment to do no more than state that the reduction of weight should be of the order of 15 per cent. with a reduction also in overall length.

The differences between the double-acting, coverless, engine of Fig. 14 and the standard engine of Fig. 3 were more extensive than indicated by Mr. Wheadon. By transferring the piston-load reaction from the cylinder cover to the exhaust piston and thus back to the crankshaft, it also became possible to reduce the weight of frames, bedplate, etc. very considerably.

Mr. Wheadon wondered whether the engine shown in Fig. 14 should not properly be regarded as an opposed-piston engine with, he would imagine, rather inefficient power transmission from the side rods through the eccentrics to the crankshaft. The engine was not intended, in any sense, to be an opposed-piston engine, the exhaust pistons having a travel not greater than necessary for them to function effectively as exhaust valves. By so doing the engine height was further reduced from its present moderate dimension. Regarding power transmission through eccentrics, the author dissented from the view that such power transmission was inefficient. This point was reverted to later. The percentage of the total power developed by the full-sized pistons was of the order of 25 to 30 per cent.

Fig. 40 showed, in diagrammatic form, a system of circulating fuel oil through the fuel valve from

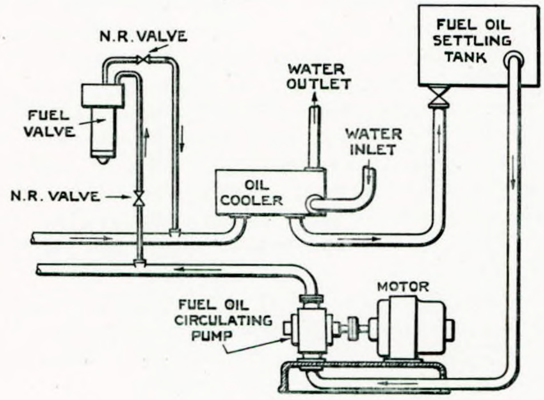


Fig. 40.

an independent circuit. It was not always necessary to have an independent pump; the fuel surcharging pump on the engine could be utilised. The cooler could also be dispensed with, if desired. With heavy, viscous fuel oils there was a strong tendency to coke heavily on the end of the warm nozzle. By cooling the nozzle the sticking and carbonising disappeared. With a low grade fuel increased liner wear could be expected, and it became a matter of equating fuel cost savings against heavier replacement costs—as mentioned by Mr. Wheadon.

On the subject of stuffing boxes the remarks of Mr. Burn, Mr. Wheadon, M. Frenay and Mr. Oxburgh could profitably be studied together. The construction of a stuffing box with removable segmental rings after the fashion of United States packing would appear to be attractive but in practice the components seemed to be more troublesome than the simpler construction.

The reason for changing over from piston valves to poppet valves for single-acting engines was not that suggested by Mr. Warne. It was purely a matter of obtaining a cheaper engine for the same output, the reduced cost with the poppet valve more than outweighing the loss of work from the piston valve. On one occasion, some years ago, to suit a certain client, a proposal was made for a double-acting two-stroke engine with poppet valves, but the engine was not built. Such an engine would give a large output on remarkably small dimensions and weight. Since then it had been possible to compromise the cheapness of the poppet valve engine with the additional work obtained from the piston valve. Mechanical efficiency in relation to valve power had never been a factor. The increase in diameter of the exhaust piston valve for double-acting engines was logical, the earlier arrangement being necessarily an intermediate step for consolidating experience with eccentrics.

Mr. McConnell's remarks on the subject of rating and especially his reference to the commercial considerations which were so often at the root of technical difficulties were very sound. The author was particularly interested in Mr. McConnell's comments on the subject of crankcase ventilation and the general improvement of the engine room atmosphere.

Mr. McConnell was correct in stating that no reference was made to the abolition of dowel pins in crankshafts because this was now fairly general practice. Whether it was better partially to vitiate web shrinkage by fitting dowels for the sake of the direct shear strength which they provided or whether it was advisable to rely exclusively on the shrinkage was a matter on which there was unlikely to be an unanimous opinion for a long time. Regarding the standard of accuracy required in assembling shafts, a crankshaft of 600mm. (23·62in.) diameter, to transmit 14,000 s.h.p. had a plus-minus tolerance of 0·05mm. (2/1000th in.).

With reference to the last paragraph of Mr. McConnell's contribution, further information as to the performance of the designs dealt with in the paper would be given in due course.

The author was very interested in the remarks of Mr. Jacobsen whose company had had so long and unrivalled an experience in Diesel propelling machinery, and had pioneered to success so many new propositions. Mr. Jacobsen was correct in surmising that on page 23 of the paper—where it was mentioned that certain covers of cast iron for two-stroke double-acting engines had been in service over an extended period—the covers of m.v. "Erria" were in the author's mind. It was valuable to have the comments of Mr. Jacobsen on this point. The opinion expressed in his first paragraph was also interesting.

Mr. Wright's remarks on the subject of the two cross-channel passenger ships "Leinster" and "Munster", the engines of which were fitted with resilient foundations, confirmed very clearly the excellent results attained to date. Without the co-operation and enthusiasm of Mr. Wright this experiment would not have been possible. Mr. Wright was correct in stressing the necessity for maintaining the circulating water temperature before starting-up.

M. Fernand Frenay described an unusual experience which he had had, due to the underrating of a Diesel engine. The connection between low temperatures and wear of liners, rings, pistons, etc. did not appear to be widely known. It was possible, for example, for liners to show serious wear on one machine while others of the same design and same material remained free from wear. In cases known to the author of machines made by another firm, careful and costly metallurgical research with a view to obtaining more suitable metals had shown results which were practically barren because, notwithstanding changes in material, the trouble persisted. Wear could definitely be expected when temperatures fell below the dew point. In such circumstances a cast-iron surface became superficially oxidised at every stroke in one direction, the oxidised film being in turn removed on the return stroke for the surface to become oxidised afresh. The solution, which M. Frenay had independently obtained, was to ensure that at every point in the circuit the temperature was always maintained above the dew point. Very often it was only a few degrees change which was necessary, when the raising of the temperature from just below to just above the dew point was to make the difficulty disappear as if by magic. It might also be proved that there was a definite connection between high inlet air temperature and reduced cylinder liner wear.

M. Frenay's remarks upon stuffing boxes were interesting. In principle, the built-up construction of stuffing box was described in a paper read before The Institute in October, 1936, by Dr. Blache. Equally of value was his reference to cast steel versus forged steel crankshafts. Regarding the casting of cast steel cranks, cylinder covers, etc., M. Frenay was in the fortunate position of possessing a steel foundry second to none.

Regarding Mr. McArthur Morison's remarks on the subject of pillarless engine room construction, there was certainly no objection to the use of the pillarless construction for small craft such as described by Mr. Morison. The particular application of it would naturally depend upon individual circumstances and such extra weight as might be involved would not be excessive. The author agreed with Mr. Morison regarding the separation of pumps from the main engine for coastal vessels, which were continually entering and leaving port.

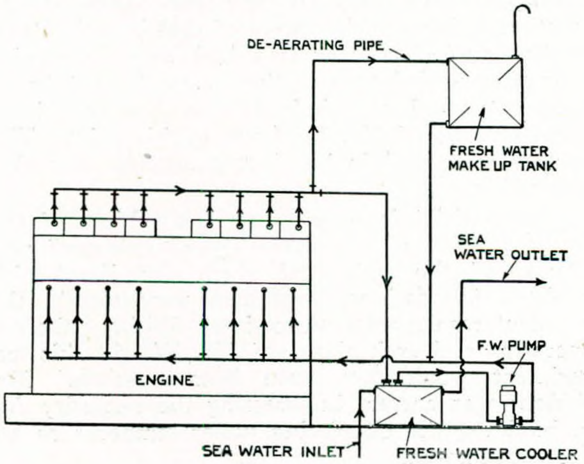


Fig. 41.

As suggested by Mr. Morison, a diagrammatic sketch of the closed fresh water system of cooling was given in Fig. 41. The diagram was self-explanatory. The purpose of the elevated tank was to ensure that losses were automatically made up, the tank being replenished at intervals from the ship's general system.

Referring to the reduction of noise from auxiliary engines, designs were available which were likely to eradicate the difficulties which Mr. Morison mentioned.

The author concurred in Mr. Morison's remark that in smaller engines scavenge pressures of the order mentioned on page 10 were not necessary.

Regarding the description on page 9 of a double-acting engine, Mr. Morison mentioned the omission of non-return valves between fuel pump and fuel valve. This remark might be supplemented by stating that in engines for small craft, such as those used for river and coastal work, non-return valves were usually provided between the fuel pump and fuel valve for the reason which Mr. Morison mentioned.

Mr. Büchi covered the points in connection with supercharged engines so thoroughly that there was little left to be said. He correctly stated that the four-stroke supercharged engines must not be built simply like the old heavy types of such engines, but according to modern principles and in a manner ensuring the greatest reliability which present experience showed to be possible; also that the engine must be worked with such a mean service pressure that the chosen system of supercharging was fully utilised. The author fully agreed with Mr. Büchi that it was unfortunate when fitting marine four-stroke engines with exhaust turbo-charging up to now, full use had not been made of the increase of output permissible with this system, except in a few cases and these generally outside Great Britain. Mr. Büchi understated rather than overstated the position when he mentioned that in first-class engines mean effective piston pressures of 115 to 120lb./sq. in. were almost everywhere employed to-day in stationary installations, etc. Mr. Büchi's remarks on mechanical efficiency, starting, manœuvring and silencing were noteworthy.

It was refreshing that Mr. Tookey, whose name was a household word, especially as an authority on gas engines, should have made so interesting a contribution to a paper for marine engineers.

The author recommended a critical study of the two papers written by Mr. Tookey—mentioned in the contribution—together with the discussions which followed. For controlling the combustion of Diesel engines on test beds and in power houses the author was in general agreement with Mr. Tookey.

The latter was not unmindful of the difficulties associated with the application of the method to marine engines at sea. The author was of the opinion that Mr. Tookey's proposal merited careful consideration.

Mr. van Tijen approached, from a different angle, the search for some equitable basis for assessing engine rating. It was a subject to which he had given long and painstaking effort and it was to be expected that at a convenient opportunity he would enter into more detail. Mr. van Tijen's remarks might be supplemented by reference to a paper entitled "Improvements in the Economy and Output of Heavy Oil Engines" read before the Diesel Engine Users Association, London, on March 10th, 1937, by Mr. J. Calderwood.

Mr. Oxburgh raised some very useful and pertinent points. To take these in turn : by reason of the limited size of Fig. 4, in common with the other illustrations, the connections between the cooling pockets were not shown, a fact which no doubt gave rise to Mr. Oxburgh's remark that air locks might be likely to form. Actually there were no air pockets. Fig. 5A might possibly make this point a little clearer. It might be mentioned that vent pipes were fitted where required. As the fresh-water cooling system was a closed circuit, air in the fresh water was very quickly got rid of by repeated heating and cooling as the water passed through the circuit.

Experience showed that the distance between the top of the internal pipe and the piston crown could be varied considerably without appreciable influence on the cooling of the crown. If the oil outlet from the piston was placed too near the crown there was a danger of the space becoming choked with carbon if certain kinds of lubricating oil were used. Actually no instance of a cracked piston was known to have occurred where the arrangement was as shown in the drawing. In the lower illustration of Mr. Oxburgh's sketch it did not seem very probable that the oil would affect to any appreciable extent the centre of the piston; rather would it tend to lip-over the perimeter of the funnel and flow down the inside. The outlet pipe nozzle dimensions were determined during the shop test, being restricted to ensure a full piston. In the top position of the piston the cover, being cooled by water, might be expected to cool quickly.

The closed fresh-water system should surmount the trouble on cold mornings mentioned by Mr. Oxburgh, the arrangement being very much superior to the old-time salt-water cooled, blast air engines. The solid-injection system had dispensed with the necessity of overloading the engine when cold-starting. The present two-stroke airless-injection engine could start cold with the handle at 'slow'. When the engine was started at 'slow' the combustion temperature was low and the temperature of the fresh water was gradually raised, the cooler being by-passed so far as salt water circulating was concerned.

The kind of thing which the author had in mind when he doubted if the Diesel engine always received kindly treatment from the bridge was exemplified by a report made by an engineer sent abroad to a Belfast-built vessel. The obvious roughness of the handling of the engines in approaching the pilot station and when in the hands of the pilot, and the insensate rushes from full ahead to full astern, caused the engineer to remonstrate with the people on the bridge. The reply received was that it was a Burmeister type Diesel engine and would stand up to anything!

Regarding piston rod stuffing boxes, Mr. Oxburgh touched the salient points: segmental rings and garter springs had been used with full satisfaction but as they consisted of a greater number of parts the time for overhaul was somewhat increased and the work must be done very carefully. Chiefly for this reason the simpler form of stuffing box seemed to be preferred. Divided stuffing box rings had been fitted from time to time, chiefly as a temporary measure, but experience had not shown them to be as satisfactory as solid rings.

The scheme proposed by Mr. Oxburgh for warming-up lubricating oil prior to starting on cold days was very sound and practical. The low heat transmission of oil cooling of pistons compared with water cooling, might be expected to render heating unnecessary, except in unusual circumstances. One might expect the effect of cold oil to be less serious than, say, that of cold starting air, if for no other reason than it probably did not occur more than perhaps once every voyage, while cold starting air effect took place every time the engine was manœuvred.

The remarks of Dr. Blache, having behind them a great wealth of experience garnered over a long period of close association with marine Diesel machinery development, were worthy of close attention. The author expressed his pleasure at having been associated with Dr. Blache for a considerable number of years.

The description which Mr. Giljam, Junr. gave of single-acting two-stroke machinery of large size was very interesting. Presumably one of the ships he had in mind was the m.v. "Noordam". He touched upon the points which were of chief concern in such an engine, especially referring to the benefits obtained by lead bronze bands on trunk pistons.

Generally speaking it might be said that a cylinder of the size indicated in Fig. 14 could replace one of the size mentioned in the second example on page 2, so far as power was concerned. Two cylinder sizes should be sufficient to embrace a very wide range of powers.

Regarding the rubber mounting of two-stroke propelling engines, Mr. Hunter would no doubt be interested in the remarks of Mr. Wright who had also contributed to the discussion.

Mr. Hunter suggested the possibility of fitting cast-iron chocks approximately under cylinders 2 and 5 of a six-cylinder engine having symmetrical cranks. Chocks so arranged would no doubt ensure that dynamic flexion did not reach the hull, but the arrangement would not overcome the other essential difficulty, *i.e.* of noise transmission, which was so subtle a problem. If rubber-lined chocks and solid cast-iron chocks were used together the difficulty would arise of suitably distributing the load. In the engines mentioned in the paper, which were carried exclusively on rubber-lined chocks, it was rather a striking fact that both on the test bed and in the ship the crankshaft alignment was better than it was in a normal engine carried on cast iron chocks. The resilient material seemed to facilitate the crankshaft taking up a mean position to a greater degree than was possible with cast-iron chocks, where the slight differences between adjacent chocks tended to cause some small degree of local mal-alignment.

The information requested by Mr. Hunter was given below :—

	(1)	(2)	(3)	(4)
Displacement (tons)	17,360	4,200	8,300	22,000
S.h.p. ahead ...	2,600	5,000	6,200	9,500
S.h.p. astern ...		not available.		

The author concurred in the remarks given in Mr. Hunter's second paragraph.

The fact that the three typical double-acting installations all had an even number of cylinders was simply coincidence. It might be stated that for 5, 7, 8 and 9 cylinders the cranks were of star formation, this being the preferred arrangement as giving a smoother turning moment and usually a smaller shaft. For 6 and 10 cylinders the cranks were symmetrical, the couples being rather too great for star formation. For single-acting engines all combinations of cylinders had star formation.

Mr. Christensen summed up very fairly the present position on the subject of noise and vibration as it affected naval architects and engine designers. The section which he gave of a Diesel-engined ferry was very interesting. The author had had experience of quite a number of these train-ferries and the steadiness in trains and saloons was remarkable.

Mr. Christensen coupled the question of rating with the service conditions of a ship and the facilities available for maintaining the machinery, remarks which were very relevant to the matter under discussion.

Mr. Yamashita's question on the subject of cylinder cover design was partly answered by Mr. Jacobsen's first paragraph. The only thing that could possibly be advanced against the casting of the cover in one with the jacket was the chance of local distortion. This again depended partly upon the material used.

Regarding the fuel valve circulating system, the fuel oil primary pump was quite suitable for this purpose and was usually of ample capacity as normally provided.

The same difference in mean pressure between top and bottom cylinders could be expected in the engine of Fig. 14 as with the standard type of engine.

With reference to Fig. 8, it could be assumed that the thin film of ordinary lubricating oil was quite sufficient to protect the piston rod sleeve from wear.

For the central plug of a piston such as that shown in Fig. 21, nickel cast iron was very suitable. Years ago it used to be the practice of some builders to use square threads for the plug, the idea being that the flat face of the thread would allow for expansion and yet maintain tightness. An alternative form of thread was the buttress thread, which was expected to combine the strength of the V-thread with the characteristic of the square thread just mentioned. But in point of fact the V-thread had been found to be the most suitable. The design shown in Fig. 21 had been fitted to hundreds of pistons, a great number having been in service five years to date without any difficulty.

The supercharged engine shown in Fig. 21, if fitted with mechanically-operated suction and discharge valves, would no doubt have a somewhat higher mechanical efficiency, but such an arrangement would depart from the essential simplicity of the Hoerbiger valves.

Mr. Yamashita concluded by suggesting that the width of the eccentrics shown in Fig. 14 could be reduced in view of the excellent system of lubrication adopted. This was very probably so, but the author was of opinion that the breadth of the eccentrics should not be reduced until lingering prejudices against the eccentric had disappeared.

In reply to Mr. Freeman it could be stated that the pillarless engine room construction reduced the transmission of noise and vibration to the accommodation spaces.

The maximum depth of double bottom within convenient working limits was desirable in order to distribute the stiffness over the structure of the ship. The point really came to this : what was the figure of value placed by the shipowner upon minimum vibration?

The remarks in Mr. Freeman's third paragraph were interesting and the author concurred in what was stated in the next paragraph. No statistics were available regarding chrome steel versus chrome molybdenum steel cylinder covers.

Regarding Fig. 14, Mr. Freeman's suggestion of stepping the exhaust and main cylinder diameters to enable all pistons to be withdrawn upward was very sound. The power obtained by the exhaust pistons was 25 to 30 per cent.

The author did not quite understand Mr. Freeman's last paragraph: the overhauling height of Fig. 14 compared with the standard type of engine was from 3 feet to 5 feet *less*.

In his reference to the working of an eccentric compared with a crank Mr. Freeman put his finger on a difficulty which the author was glad he had raised. A well-known engineer, a vice-president of The Institute, who did not desire to be quoted, recently mentioned to the author his difficulty in comprehending how an eccentric could be an effective means of transmitting power. His query took this form: "I can understand force transmitted through an eccentric assisting an adjacent crank but I cannot see how an eccentric can function as a crank, assuming no friction". It might therefore be desirable to close this reply with a few remarks upon the eccentric.

In the first place it was necessary to define an eccentric. To the author, an eccentric was simply a crank with an abnormally large crankpin. Fig. 42 showed the evolution of the eccentric. At *A* there was a normal crank. The top end of the connecting rod reciprocated up and down and the bottom end swept through a circle whose radius was the crank-throw. At *B* the crank was shown with

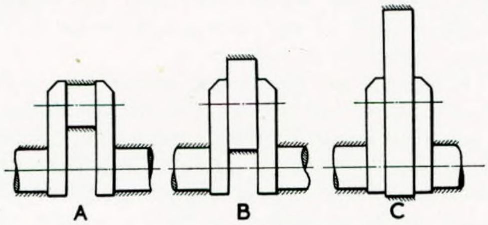


FIG. 42.

a considerably larger crankpin. Apart from the fact that the bottom end was larger, everything functioned as before. At *C* the crankpin had grown larger still, otherwise nothing was changed in the manner of working. The connecting rod still actuated the bottom end on the crankpin circle, and the direction of force was still from the crosshead, —reciprocating on the vertical centre-line—to the crankpin centre, for all positions of the crank. The crank *C* was now called an eccentric. Granted no friction, how could it fail to work as a crank?

But, it might be objected, friction made all the difference. The answer was that this was not so; frictional loss was remarkably small. Painstaking investigation showed that the coefficient of friction was of the order of 0·0035. The limits were 0·0035 to 0·0075, depending upon viscosity, temperature, circumferential velocity, pressure, etc. The lower value was the more usual. These values were for surface pressures up to 120 and 125 kg.cm^2 (approximately 1,700 to 1,800 lb. sq. in.), which were much higher than the surface pressures actually adopted. The eccentric drive was fully equal to the crankpin drive as regards efficiency, superior as regards strength and stiffness and cheaper in first cost.

To the British marine engineer the word eccentric was inseparably associated with the old steam engine eccentric which lumbered round in a trough of water, fed by an oil wick, the lubricating oil grooves being arranged in accordance with notions which were incorrect. To think of an eccentric was to be reminded of M'Andrew's Hymn:

> "The crank-throws give the double-bass,
> the feed-pump sobs an' heaves,
> An' now the main eccentrics start their
> quarrel on the sheaves. . . . . .
>
> *Below there! Oiler! what's your mark?*
> *Ye find it runnin' hard?*
> *Ye needn't swill the cap wi' oil—*
> *This isn't the Cunard!"*

Actually the Diesel engine eccentric bore no more relation to the eccentric of the steam engine than the latest high-class motor car bore to the deacon's famous one-hoss shay.

'Loded' Cast Irons—Further Discussion.

The following further contribution to the discussion on Mr. H. J. Young's paper "'Loded' Cast Irons", published in the January, 1939, Transactions, has been received from Mr. J. G. Pearce of the British Cast Iron Research Association. Mr. Pearce wrote:—"I have read Mr. Young's paper with interest and it is gratifying to find that he has thought it desirable to examine the high-silicon high-chromium cast irons. The idea of utilizing for wear resistance purposes an iron of such composition that it is all-pearlitic with a small and well-distributed amount of free carbide has been made use of in the production of engineering castings for many years. This reference is not to automobile but to steam engine and Diesel engine castings. Such castings were obtained not by high-silicon, high-chromium but by low silicon with the manganese content low enough to be out of balance with the sulphur content, well known to be carbide stabilising.

Mr. Young will be aware of the Silal irons developed by the British Cast Iron Research Association. covered by British Patent 323076 (1929), in which the use of 4 per cent. to 10 per cent. silicon is covered for the production of heat-resisting cast irons. For this particular purpose fine graphite is preferred, although in commercial practice the graphite is frequently of the flake type and the matrix generally ferritic. Chromium additions were studied at the time and although it was not practicable on account of prior specifications to include chromium, this element has been and is frequently recommended by us in the Silal type irons. That all-pearlitic grey irons have been and can be obtained with high silicon contents will be evident from the photomicrographs contributed by Norbury to the Journal of the Iron and Steel Institute 1936/1, page 336, in the discussion on Tapsell, Becker and Conway's paper. This micro shows pearlite in a 6 per cent. silicon iron produced by the use of antimony. Further published evidence on pearlitic irons high in silicon is given in a paper by Norbury and Morgan to the Iron and Steel Institute, Volume 1, 1932, which shows a complete series of structures for irons up to 10 per cent. silicon and 14 per cent. manganese, giving the austenitic, martensitic, pearlitic and ferritic ranges, as well as the physical properties. In the nickel-silicon series in the same paper up to 32 per cent. of nickel was studied. The photomicrographs clearly show the pearlitic nature of the 5 per cent. silicon high-manganese materials. This paper embodied Research Reports 96 and 98 issued by the British Cast Iron Research Association. In a similar Research Report, No. 84, issued in February 1930, the constitution, heat-resisting and other properties of the 5 per cent. silicon irons containing various added elements, no less than sixteen in number, were disclosed, including chromium, molybdenum, manganese, nickel, copper. The curves in this report clearly show that in 5 per cent. silicon irons a pearlitic structure is obtained with about 4 per cent. chromium or 7 per cent. molybdenum, and the impact strength is unusually high with chromium, copper and molybdenum additions. A photomicrograph shows 1 per cent. combined carbon, *i.e.* all-pearlitic with a little free carbide, with 3·5 per cent. chromium. In Research Report 93, issued in 1931, a complete study of the chromium-silicon irons is given, up to 10 per cent. silicon and 7 per cent. chromium. A 4·9 per cent. silicon cast iron, which without chromium is completely ferritic, shows with 0·5 per cent. chromium a mixed ferrite-pearlite structure, and with 1·6 per cent. chromium a completely pearlitic structure.

"If experience supports the wear-resisting properties of these materials the result from the engine makers and automobile points of view will be important, but it is hoped that Mr. Young's further work will in addition show not only that the improvement in wear-resistance is more than counterbalanced by the handicap of brittleness brought about by the use of 5 per cent. silicon (a property which will show itself not only in the reduced tensile and impact strengths but in reduced deflection in transverse), but also that these materials have a definite advantage over the irons of the type on which Mr. Young has previously done so much, an all-pearlitic structure with, if required, free carbide brought about by the use of a silicon content lower than normal".

Mr. H. J. Young, in reply, wrote:—"Mr. Pearce's contribution is very welcome although there are many differences of opinion. His experience that *all-pearlitic iron obtained by low silicon with the manganese content low enough to be out of balance with the sulphur content has been made use of in the production of steam engine and Diesel engine castings for many years,* is opposite to that of the author, who has never heard of anyone here or abroad intentionally making a casting of this type in iron of the above-described quality. In 1921, before The Institute of Marine Engineers, the author introduced the question of, what he named, sulphur-manganese balance, as follows:—

> 'Foundries have ever been prone to sudden mysterious spasms of bad metal or bad castings . . . in every case in the author's experience the iron has always suffered from inadequately-balanced sulphur . . . there does exist a very vital equilibrium point between the sulphur and the manganese . . . With sulphur-control, low sulphur for one type of work and higher sulphur for another, these difficulties can be got over and the condition and amount and disposition of the carbon and of the eutectics can be controlled in the casting itself—*the sine qua non being that no matter what the quality of sulphur that of the manganese must be more than sufficient to balance it*'.

"On almost all points the author has different experience from that of Mr. Pearce; it would take much space to argue fruitfully each one. Although Mr. Pearce is pessimistic about 'Loded' irons, the process functions on a practical scale; moreover, *all* the range of irons cannot suffer by reason of this 'handicap of hardness' because two irons of the series are working well as cylinder liners, while another has given improved results in drawing dies of large size, a terrific service where it would be unreasonable to expect other than very quick failure if the brittleness was of that order suggested by the word 'handicap'. Further, Mr. Pearce states that brittleness is a property which will show itself *not only in the tensile;* but 5 per. cent.-silicon 'Loded' iron, as only one example, has never given less than 17 tons per square inch tensile, namely, a figure, the author suggests, far removed from that which Mr. Pearce could have had in mind when making the remark. What the author feels about all this is that consideration merely of the results already obtained on cylinder liners, drawing dies and on tensile proves that we know nothing about these irons and that our theories do not apply because they are based on irons quite dissimilar to these. Mr. Pearce's remarks cannot be explained otherwise; were he right this paper would never have been offered and, very certainly, the author would not have placed liners into service on many engines including marine units, whilst drawing dies would have been to court ignominious disaster. Nothing could be better evidence of the newness of these irons than the extremes thus presented between present fact and past theory. Despite these contentious points the author believes the new irons will have valuable uses in British engineering and that Mr. Pearce, and the Association which he has brought to active being, will find in them an attractive and hitherto untrodden field. He hopes, indeed, to collaborate with Mr. Pearce whose knowledge. criticism and help will be invaluable".

ELECTION OF MEMBERS.

List of those elected at Council Meeting held on Monday, 6th February, 1939.

Members.

Alex. M. Barclay, 272A, Earls Court Road, S.W.5.

John Spence Blair, 19, Broomhill Terrace, Glasgow, W.1.

John Martin Binmore, Widewell, 18. Riddlesdown Avenue, Purley, Surrey.

Philip Henry Dallison, 17, Church Hill, Loughton, Essex.

William Edwin Hunter, 5, Selwyn Avenue, N. Ferriby, E. Yorks.

William John Padbury, Rydal, Hillside Drive, Old Road East, Gravesend, Kent.

Frederick William Smith, Auldgirth, River Drive, Upminster, Essex.

George Hudswell Stallworthy, 37, Park Road. Gosport, Hants.

Thomas Walkington, Greenways, Hunsdon, Ware, Herts.

Edgar James Warder, 72, Parkside Avenue, Romford, Essex.

Associates.

Alexander MacQuarrie, 302 Dumbarton Road, Glasgow, W.1.

Henry Basil Hadow Maundrell, 47, De Lisle Road, Winton, Bournemouth.

Bert Stubbs, 45, Bells Road, Gorleston, Great Yarmouth.

David George Webster, 4, Holly Drive, Maidenhead.

ADDITIONS TO THE LIBRARY.

Purchased.

The Engineers' Who's Who. Compiled, edited and published by M. E. Day, Glenwood, Dorking, 20s. post free.

Air Conditioning. By Moyer and Fittz. McGraw-Hill Publishing Co., 24s. net.

Presented (on permanent loan) by Mr. E. W. Green, O.B.E. (Member).

Transactions of the Second World Power Conference, Berlin, 1930. Vols. 1 to 20.

Presented by the Publisher.

The National Physical Laboratory: Abstracts of Papers published during 1937. H.M. Stationery Office, 1s. net.

Structural Analysis Laboratory Research, 1937-38. Department of Civil and Sanitary Engineering, Massachusetts Institute of Technology.

Handbook of British Refrigeration Material, 1939. The Cold Storage and Produce Review, Empire House, St. Martin's-le-Grand, London, E.C.1, 2nd edn., 196 pp., 5s. net.

The following British Standard specifications:—

No. 18-1938. Tensile Testing of Metals.

No. 823-1938. Density-Composition Tables for Aqueous Solutions of Sodium Chloride and of Calcium Chloride for use in conjunction with British Standard Density Hydrometers.

No. 824-1938. Density Composition Tables for Aqueous Solutions of Caustic Soda for use in conjunction with British Standard Density Hydrometers.

Electric Arc Welding in Shipbuilding. By A. M. Carrick. The Draughtsman Publishing Co., Ltd., 48 pp., illus., 2s. net.

Since welding is a growing practice the author has deemed it unwise, in a pamphlet of this nature, to spend much time on a description of work completed, but rather has directed his attention to giving an indication of the principles involved. The illustrations and examples have been chosen with this end in view, rather than as being descriptive of practice. Some idea of the scope of the work may be gathered from the following selection from the headings:—Description of process, types of joints, indication of welds on drawings, checking welds, strength of welds, butt welds, fillet welds, fillet welds in bending, costs, design, bulkheads, contraction problems and all-welded vessels.

The work concludes with two appendices (one on geometrical properties of sections suitable for welding and the other on electrodes) and a useful bibliography.

The Journal of Commerce Annual Review. Charles Birchall & Sons, Ltd., 312 pp., copiously illus., 2s. net.

The publishers of this informative annual which records the development of shipbuilding, marine engineering and shipping from year to year, have again maintained that high standard to which we have grown accustomed. Besides many of general interest, it contains the following articles having a particular appeal to marine engineers:— "Progress in Hull Design and Deck Equipment", "Steam Propulsion of Ships", "Marine Electricity", "Progress of Motorshipping", and "Electric Arc Welding in Shipbuilding".

The publication is excellently produced, copiously illustrated, and for a work of such bulk the price of 2s. seems extraordinarily low.

Handbook for Welded Structural Steelwork. The Institute of Welding, 220 pp., copiously illus., 5s. 6d. post free.

This handbook has been produced to meet the need for a standard book of reference on welded construction, and to provide the technical data necessary for the design of welded structures. The first part of the work contains tables of properties of standard sections and arrangements commonly used in welding construction. The second part contains calculations and details for typical examples of welded construction.

The preparation of the handbook has been carried out under the direction of the Handbook Committee of the Institute of Welding. The work contains previously unpublished data and information provided by a number of the welding companies and by the Department of Scientific and Industrial Research.

The book is copiously illustrated with clear line drawings and excellent photographs, and is altogether a creditable production.

Electrical Installations. By J. W. Sims. John Murray, 191 pp., 110 illus., 6s. net.

This little book is, according to the preface, designed primarily to cover the syllabus recommended by the Advisory Committee of Electrical Installations of the E.C.A. and the City and Guilds of London Institute. It will be clear that a book of this small compass covering such a wide field, must of necessity be very brief; hence, whilst the book is probably a satisfactory class-book used in conjunction with lectures and demonstrations, the general reader may, in some cases, find himself left in doubt, as naturally where matter is dealt with so briefly, important qualifications are sometimes omitted.

The first 50 pages deal with the fundamentals of electrical engineering, and a further 67 pages describe electrical equipment, machines, instruments, transformers and lamps. In these two sections a very large amount of useful data is included. The remaining 68 pages cover the application of fundamental principles to the practice of electrical installations. As the book has been written primarily for students of electrical contracting, it will be obvious that only a very brief outline can be given in the space available of the many problems to-day confronting the electrical contractor and upon which guidance is frequently urgently needed. From this point of view, therefore, it is somewhat disappointing to note that there is only one page in the book dealing with the dangers of shock, and nothing in regard to the fire risks. If the installer's aim is to supply adequate electrical facilities with the maximum safety from fire and shock reasonably obtainable, the lack of treatment of these two aspects can only be regarded as an unfortunate shortcoming.

The book contains less than two pages dealing with fuses, based upon an obsolete B.S.S. Specification. The new British Standards Specification No. 88 issued in July, 1937, alters fundamentally our ideas on the selection of fuses, and is a subject on which the reviewer is sure contractors would value very much further enlightenment, as the question of safety is intimately bound up with the satisfactory operation of fuses. Similarly, in regard to earthing, which often is the second factor in ensuring safety, the author, in drawing attention to the necessity for an earth leakage trip in some cases, appears to have overlooked the fact that even an earth resistance of 1 ohm is not necessarily sufficient to ensure safety.

Figure 82 on page 121 is apt to be misleading, as it implies that a rising main may be graded according to the amount of current which the different sections have to carry. This is only true if the fuses at the point of supply afford adequate protection of the smallest section, a condition very rarely obtaining. The diagrams on the two following pages, showing different methods of control for lighting circuits, will be welcomed by many wiremen. On page 154 it is stated that a diversity factor must not be applied to a final sub-circuit. According to the I.E.E. Rules in the amendment issued in June, 1935, diversity may be applied to final sub-circuits where such diversity is justified. Again, on page 157, the author has noted the limit of voltage drop imposed by the I.E.E. Regulations, but has not noted that according to the current edition of the regulations this only applies to lighting wiring, and not to power and heating circuits.

The Performance and Design of Direct Current Machines. By A. E. Clayton, D.Sc. Sir Isaac Pitman & Sons, Ltd., 2nd edn., 445 pp., 276 illus., 16s. net.

This is a second edition of the author's work, first published in 1927. During the intervening period, little has been added to our knowledge of the theory and performance of d.c. machines. Developments have, however, taken place in constructional details and materials, hence the author has replaced most of his original illustrations of machine construction by modern examples and extended considerably the section of the book dealing with electrical design.

The book has been specially written as a textbook for students at universities and technical colleges, and adequately covers the syllabuses of the B.Sc. (Eng.), the I.E.E. examinations, and Higher National Certificates in the subject. Especially written as it is for examination purposes, the author has linked the subject of design with that of performance, and has endeavoured to strike a correct balance between the requirements of these two related branches of the subject. About two-thirds of the book is devoted to the study of performance and the remainder to the principles of design. The contents of each part appeal to one as well chosen, and the author achieves his aim. In the first section he has covered all the important principles of d.c. machine performance and characteristics. The following headings give an idea of the portions of the subject which are thus thoroughly considered :—General principles, armature windings and reaction, commutation, characteristics of d.c. motors and generators, speed control, efficiency testing and special machines. The reviewer has nothing but praise for the treatment in this section of the book. Its scope, illustrations, etc., are all that can be desired.

The latter part dealing with design is an excellent introduction to the subject. It treats of the constructional and mechanical details of d.c. machines, and the fundamental principles governing the design of the various parts and types. Excellent and numerous drawings provide a large amount of information which it is not possible to include in the descriptive matter in the space alloted to this part of the subject. Examples of the calculations involved and their application in the construction of various machines are given in detail, together with design data for several types of generators and motors.

At the end of the book is given a collection of exercises and questions based on the matter of each chapter, with answers to the numerical examples, which will be useful to student and teacher alike.

The book is an excellent combined treatise on d.c. machine design and performance, and can be recommended to all electrical engineering students, as well as to teachers of the subject.

Mechanical Drawing. By Edwin S. Youngberg. Sir Isaac Pitman & Sons, Ltd., 96 pp., 84 illus., 3s. 6d. net.

This is an American publication in which the author aims at providing a "two-year course for the Junior High School" as a preparation for the more advanced work in machine, architectural or sheet metal drawing which the pupil may be called upon to follow in subsequent years. It barely covers the work of the first and second years of the courses provided in junior technical and other post-primary schools in this country.

It has been said that knowledge is of two kinds; that passed on by others like the ancient languages and that gained through personal experience. Technical drawing provides much scope for personal experience. The author of this book has, however, given so much thought to the planning and grading of the numerous problems, and provided such clear and precise information and instructions regarding each exercise, that the pupil should find little difficulty in following the author's footsteps. It would seem that a stronger challenge to the pupil's mental capacity, energy and initiative might occasionally have been made and the pupil left with occasional opportunity for the development of judgment and personal experience, which, of the two roads, is the high road to knowledge. Herein, of course, may lie the function of the teacher who otherwise might conceivably feel himself "supernumerary to the establishment". On the other hand the author may have had prominently in mind the needs of those who may not have the opportunity for personal contact with a teacher.

The work is divided into eleven "units", commencing with drawings in two dimensions of length and width requiring only one view and arranged in the following sequence, viz. :—outlines, lettering and dimensions, working drawings from pictorial views of simple objects in three dimensions with the aid of the glass box method of observing three views, geometrical constructions, sectional views, lettering exercises, revolution of objects into different positions, development of surfaces, auxiliary views of objects with rectangular and oblique surfaces, elementary machine drawing-conventions, American screws, bolts, nuts, fastenings, tool post and bench grinder, and isometric, oblique, and perspective drawing. The diagrams are well drawn, and the arrangement has the great merit in that the text, which is brief, is printed below the drawings on the same page.

American (3rd angle) projection is used throughout the book. This, together with the likelihood of educationists in this country frowning upon such examples of spelling as *center, color,* and *vise,* seems likely to preclude its use as a standard work for class use. The price is favourable.

A Course in Practical Drawing, Book 2. By H. H. Crump. Sir Isaac Pitman & Sons, Ltd., 74 pp., 125 illus., 1s. 2d. net.

This handy little volume is designed for use with young students of 12 to 13 years of age who are not necessarily intended for engineers. In the main the subject matter is geometrical, though frequent reference is made to engineering practice in the examples. A section dealing with borders and repeated patterns should give considerable practice in necessarily accurate work, while at the same time possessing a fascination of its own for most youngsters. Isometric drawing is given due but not undue prominence, while the developments of the surfaces of various solids are a useful feature. In such an elementary work, perhaps the bold but inaccurate statements that the "width of one face of a nut is equal to the diameter of the bolt on which it fits" (page 37) and "the width across the corners of the nut is equal to twice that diameter" may be allowed, though it might be better to admit that these measurements are only approximately correct. A "fact" of this kind instilled at the age of 12 may need to be "unlearned" at a later stage when a greater degree of engineering accuracy is required.

This is, however, a small point in an otherwise praiseworthy book. In Section 11 several of the sketches are unavoidably remote from the letterpress referring to them, but in general the lay-out has that clarity which one associates with the house of Pitman.

Trade Marks. By R. Haddan. Sir Isaac Pitman & Sons, Ltd., 128 pp., 5s. net.

As a result of an agitation extending over some years, a considerable alteration of the law has now (apart from certain features of common law applicable to the subject) been embodied in a new consolidated law, the Trade Marks Act, 1938. The principal alterations concern the connection of the mark with the goodwill of the business, and the use of the mark by others under permission of the proprietor, but there are many other minor amendments, including a new classification of goods for trade mark registration purposes. The author considers that the compilers of the new Act appear to have some doubts as to the use that might be made of the new facilities and the Act therefore bristles with provisos, exceptions, etc., intended to prevent possible abuses, which, with the rules applicable to them, render the practice under the Act somewhat difficult. It seems certain that the importance of the new provisions is sufficient to warrant the issue of this book, which is intended to explain the new law and practice to trade mark owners and others concerned. The seven chapters comprising this work deal with the nature of trade marks, the ownership of trade marks, the registration of trade marks, with the subject of appeals, oppositions, amendments, assignments and licensed use of trade marks, with remedies for infringement, with sundry provisions, and finally with protection in British dependencies and colonies and in foreign countries.

English for Students in Applied Sciences. By S. A. Harbarger, W. R. Dumble, W. H. Hildreth, and B. Emsley. McGraw-Hill Publishing Co., 260 pp., illus., 12s. net.

This book would prove of some value to teachers in charge of students who are spending most of their course in the laboratory, workshop or machine shop. The students themselves would not profit much from a study of this volume, since it is written in a somewhat difficult and unattractive style. To students accustomed to American methods of detailed instruction, the book might have a greater appeal.

The chief value of the work lies in its obvious usefulness to advanced students doing post-graduate work. Much good original research work done for M.Sc. and Ph.D. loses some of its effectiveness for practical purposes through faulty presentation of the findings in the theses or dissertations offered as evidence of two years' work in the laboratory. Many students would gain considerably by reading those portions of this book which explain in detail the best methods for assembling, arranging and presenting research findings.

Part 1 deals with vocabulary, grammatical usage, the different kinds of functional writing, the use of figures and tables, etc.

Part 2 deals with the application of the mechanical principles of composition to the requirements of scientific work. This, the more important and useful section of the book, describes the different kinds of scientific reports, and how to set about their preparation, numerous and useful examples being given for the students' guidance. The authors' directions to students include hints on the proper use of the library, preparation of the bibliography, the recording of the material and its subsequent summarizing. One of the appendices, a general reading list in modern English and American literature, is an admirable guide to both teacher and student. A particularly useful section in this appendix contains a list of books in which science is linked to literature, *e.g.,* Sir James Jeans' "The Stars in their Courses".

Both parts of the book are full of carefully chosen examples and each chapter concludes with a number of graded problems.

JUNIOR SECTION.

Electricity Applied to Marine Engineering.

By W. LAWS, B.Sc.

What is Electricity.

There must be very few people indeed in the civilised world to-day over the age of ten, who have not at some time uttered the words electricity, or electric current, with some idea in their own minds as to what they meant. Not a single one of those persons, however, has ever seen an electric current. All that they have ever seen is some effect which they have been taught to attribute either to electricity, or to the passage of an electric current. Lord Russell, a philosopher and mathematician of international reputation, after discussing at some length the modern theories as to the nature of electricity, summed it up by describing electricity as "a way in which things behave". If he is satisfied to leave it at that, then perhaps the marine engineer who is concerned rather with practical things than with philosophical abstractions need not concern himself overmuch with the ultimate nature of electricity, but rather with how the effects of electricity may be applied to his work, and how the electric current may be produced and controlled.

The Electric Current.

Since we can never see electricity, we must form some kind of mental picture as to what might be described as the mechanism of it at work. One man's picture may not be the same as another's. That does not matter in the least. The acid test is, whether with the aid of his picture he can prophesy correctly what effects will follow certain causes. If he can, then he has a good picture.

The word current implies movement, and not only movement but a rate of movement. A pond or lake is not a current of water. Neither is a canal, or water in a pipe when the tap is closed. As soon as one hears the word current one thinks of a fluid on the move. If one stands beside a swiftly flowing stream and says to a companion "the current is very swift, I should say it is quite 100,000 gallons", he will most probably answer "per what, second, minute, or hour?" That is, the word current has called up the idea of the rate of movement of a quantity. In precisely the same way we form a mental picture of an electric current as being a time rate of movement of a quantity of electricity. Just as the unit of quantity of water is the gallon, so the unit of quantity of electricity is an amount which is called a coulomb; and just as we might take a rate of flow of water of one gallon per second past any given point in a pipe as the unit of hydraulic current, we do actually take a rate of flow of electricity of one coulomb per second as the unit of electric current. Only we call it an ampere for short, just as nautical people speak of one knot, when they mean one nautical mile per hour. The time element is implicit in the word knot, or ampere, as the case may be.

Electrical Pressure Difference.

Before a current of water can be caused to flow in a channel or pipe, there must be a difference of pressure between any two points in the pipe between which water is to flow. This difference of pressure may be produced

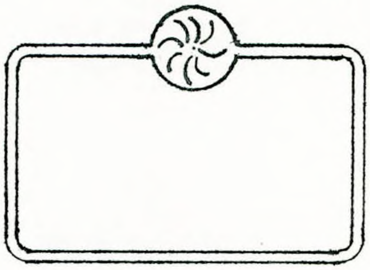


FIG. 1.

in various ways. Let us consider some simple illustrations. We might have a simple pipe line connected to a pump. The pump produces a difference of pressure between its inlet valve and its delivery valve which keeps the water circulating round the pipe. The arrangement is illustrated in Fig. 1. Or we might have an arrangement as illustrated in Fig. 2. If the pump were working,

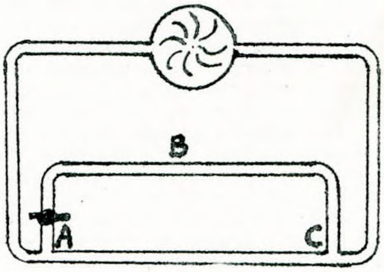


FIG. 2.

and the stop-cock at A were open then water would flow along the branch A B C. But there is no pump in the branch A B C. The water flows along it because there is a difference of pressure between the points A and C, and this exists because of conditions obtaining in the circuit as a whole, and would not be there were the pump not working. There is a fundamental point to be observed at this stage, namely that so long as the pressure difference exists, work is being done or has been done somewhere. In this case it is being done by the prime mover driving the pump.

It is possible to visualise another way in which a difference of pressure might be produced. A large tank, mounted on high pedestals, might be kept filled by a succession of men running up a ladder carrying buckets of water and pouring them in. A water turbine at a much lower level could be kept working by a current flowing from the tank because of its pressure head.

and this turbine could discharge into a reservoir from which the men filled their buckets. The head is maintained only so long as the men go on doing foot-pounds of work in carrying the buckets up the ladder. Should they cease work, then the current would go on flowing only so long as there remained any pressure difference, and this would exist only until the work done by the men had been all used up in driving the turbine, and supplying the various frictional losses.

All of this no doubt seems very elementary, but the point is that we can draw almost precise analogies between these hydraulic happenings and what happens in an electric circuit, and in passing it may be remarked that a sound understanding of fundamental mechanical principles is of the greatest possible help in understanding electrical happenings. It is possible to draw very close analogies, because after all, they all ultimately come back to the universal conception of energy.

Let us apply some of these ideas to the electrical circuit. In Fig. 3, D represents an electrical pump, commonly called a dynamo or generator. S1 is the delivery valve, and S2 the inlet valve. It will be

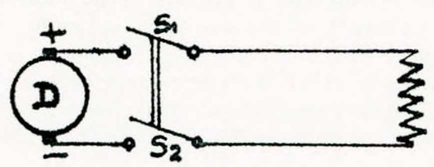


FIG. 3.

observed that they are both coupled together and form what is called a double-pole switch. When the dynamo is driven by a prime mover a difference of pressure is induced in it whether the switch be open or closed (*i.e.* the valves shut or open). If the switch be closed, then a current will flow round the circuit from the higher pressure outlet, called the positive terminal, to the lower pressure inlet, called the negative terminal. The whole purpose of a dynamo is to produce a *difference* of electrical pressure—not an absolute pressure. The current will flow round the circuit passing through the

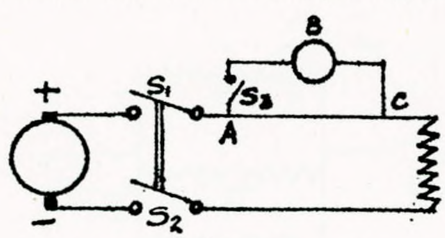


FIG. 4.

dynamo as well as the external circuit. To force the current through the dynamo will require a certain pressure, so that once the current starts to flow some of its total induced pressure will be used up inside itself, leaving less available for the external circuit. Another point worth noting is that the dynamo does not make electricity any more than the pump makes water. It simply sets in motion something which is already there.

Fig. 4 represents the electrical variation of Fig. 2. If between the two points A and C we connect another circuit containing a lamp B, and a stop-cock or switch S3, then when S3 is closed a current will flow round the Branch A B C and, given suitable conditions, *i.e.* the right amount of pressure difference, will cause the lamp to light. Yet there is no dynamo in the branch A B C; there is, however, a difference of pressure between A and C while the dynamo is working. Were it to stop, this pressure difference would cease to exist. Note also that it exists only so long as work is being done by the prime mover driving the dynamo.

With regard to the analogy of the men filling the tank, this might be applied very roughly to the case of an electric cell inasmuch as at some previous date a certain transference of electricity took place which resulted in a difference of electrical pressure being established between the plates of the cell. This difference of pressure persists so long as the plates are not connected. Once they are connected outside the cell a current flows between them. For a certain time the electrical condition of the chemicals making the cell maintains this pressure difference, but in course of time the cell is completely discharged. In the case of the primary cell its usefulness is then finished. In the case of the secondary cell or accumulator it can be re-charged, i.e. work can be done on it, and the pressure difference between the plates once again established.

Electrical Resistance.

So far we have assumed without question that to set either water or electricity in motion and keep it moving requires a certain pressure difference. In the case of water we know that first of all its inertia must be overcome to set it moving from rest, and that to overcome the inertia requires force, quite apart from whatever force is required to overcome any frictional resistances there may be. An exactly similar state of things obtains in the electrical circuit, but for the moment we will consider only the case of a steady flow of electricity so that only frictional resistances are involved.

Just as a pressure difference is required to maintain a current of water at a uniform velocity because of frictional resistance, so a pressure difference is required to maintain electricity moving at a steady speed because of electrical resistance, which there is no harm in thinking of as being of the nature of a frictional resistance.

Now it seems to be a not unreasonable assumption that provided the electrical resistance of a circuit remains unchanged, then if the electrical pressure acting on the circuit is increased the electricity will move more quickly, that is the current will be increased. It seems an equally reasonable assumption that if the pressure remains unaltered but the resistance is increased, then the current will get less. Alternatively, if the resistance is diminished the current will become greater for the same pressure difference; in short:—

Ohm's Law.

The current in an electric circuit varies directly as the pressure difference applied to the ends of the circuit, and inversely as the resistance of the circuit.

This law was first enunciated in 1827 by Georg

Simon Ohm (1787-1854), though not precisely in that form, though it amounted to the same thing, and he had a very uncomfortable time over it. He was ridiculed by his fellow scientists, and it was very many years before it was generally accepted, and yet it is one of the foundation stones of all applied electrical theory.

Let us put it another way which, if it does not seem quite so obvious, is more easily capable of experimental check.

The pressure difference between the ends of a circuit is directly proportional to the current flowing in the circuit.

In making this statement it is assumed that the temperature of the circuit remains constant. Also the statement is true either for the whole circuit or for any part of the circuit.

Therefore,

$$\frac{\text{Pressure Difference}}{\text{Current}} = \text{a constant}$$

Putting it in symbols,

$$\frac{E}{I} = R$$

where E = pressure difference,
I = current,
R = the constant relationship between, them.

This constant R we call the resistance of the circuit. Now if this relationship is to be literally and arithmetically true we must establish some units of measurement for the quantities involved for which it will be true. For example, when we say that the density or specific gravity of some substance is some number, we automatically assume that the volume has been measured in certain units of volume, and the weight in certain units of weight.

The units of current, pressure difference and resistance are as follows :—

Unit Current.—The Ampere.
Unit Pressure Difference.—The Volt.
Unit Resistance.—The Ohm.

All civilised peoples use electricity to-day. It was therefore decided that if electrical engineers of different nationalities were discussing electrical matters they should all mean the same thing when they referred to the same unit. These units were therefore defined by international agreement in terms of practical things which could be handled. For example :—

The International Ampere is that steady current which deposits silver from a solution of silver nitrate at the rate of 0·001118 gm. per second under certain specified conditions.

The International Ohm is the resistance to a steady current of a column of mercury at the temperature of melting ice, its mass being 14·4521 gm. and its length 106·3 cm., the cross-sectional area being uniform (1 square millimetre).

The International Volt is the steady pressure difference which when applied to the ends of a resistance of one international ohm produces a current of one international ampere through it.

Why should the dimensions be given so precisely in defining the ohm? To appreciate this we must study how they affect resistance.

Laws of Resistance.

It can be shown experimentally that the resistance of a conductor varies directly as its length and inversely as its cross-sectional area. It also depends upon its material.

Suppose then we wish to compare the electrical resistances of two conductors of different materials. It is obviously not of the slightest use taking 20 yards of one with a cross-sectional area of 0·01 sq. in. and attempting to compare its resistance with that of 50 yards of the other having a cross-sectional area of 0·008 sq. in. To get a fair comparison we must take specimens of the two conductors having precisely the same dimensions. The most obvious dimensions to take are unit length and unit cross-sectional area, from which we arrive at :—

Resistivity, formerly called "specific resistance", which is the resistance of a unit cube (either a cm. cube, or an inch cube) of the material taken between opposite faces.

A word of warning is needed here. The resistivity is *not* the resistance of a cubic inch, or cubic centimetre of a material. An inch cube of a material is always a cubic inch, but a cubic inch need not necessarily be an inch cube. Confusing these two ideas is the most common mistake made in doing numerical problems on resistance.

The resistance is also affected by change in temperature. This will be treated more fully later.

Grouping of Resistances.

In practice an electrical circuit usually consists of a number of parts, wires, pieces of equipment, etc., connected together in various ways. Suppose we have a number of conductors having resistances r_1, r_2, r_3, etc. Then they might be connected in :—

Series connection, thus

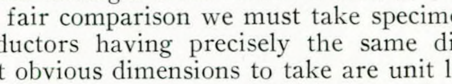


Fig. 5.

or in

Parallel connection, thus

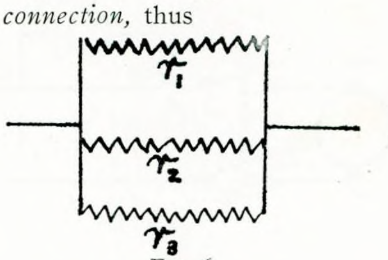


Fig. 6.

or in a mixture of the two, thus

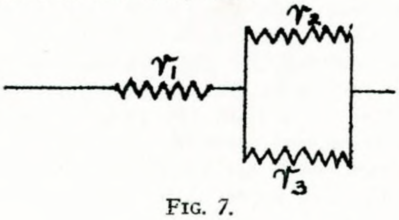


Fig. 7.

If we wish to carry out an electrical calculation for any circuit, it is most convenient to find out its equivalent resistance, that is the resistance of the single conductor which could be substituted for the composite circuit with the same electrical effect, that is, permitting the same current to flow for the same electrical pressure difference. To arrive at the appropriate expressions to use, we make use of two conceptions which are regarded as fundamental. Firstly, the current at every point of a series circuit is the same, just as in any continuous hydraulic circuit the current at every point must be the same. Secondly, when there are several circuits in parallel, although the currents in the various branches may be different, the pressure difference between the common ends must be the same. This is really another way of saying that one particular point in an electrical circuit cannot be at two different pressures at the same time.

Series Circuit.

Suppose a pressure difference of E volts is applied between the points A and B, Fig. 8, and causes a current

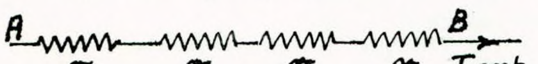


FIG. 8.

of I amperes to flow. Then this current is the same all along the circuit and from Ohm's Law

Pressure drop down $r_1 = I \times r_1$

" " " $r_2 = I \times r_2$

" " " $r_3 = I \times r_3$

" " " $r_4 = I \times r_4$

therefore total pressure drop $I(r_1+r_2+r_3+r_4) = E$.

Let the equivalent resistance of the circuit be R ohms; then this permits the same current to flow, namely I amperes, therefore

$$I = \frac{E}{R} \text{ or } E = I \times R,$$

therefore $I \times R = I(r_1+r_2+r_3+r_4)$ since they are both equal to E; dividing both sides by I,

$$R = (r_1+r_2+r_3+r_4)$$

so that we find the equivalent resistance of a number of resistances in series by the simple expedient of adding them together. This lines up with the fact that the resistance of a wire varies directly as its length because we can think of any length of uniform wire as a number of shorter lengths joined together in series.

Parallel Circuit.

Suppose a pressure difference of E volts is maintained between A and B, Fig. 9. Then, by Ohm's Law

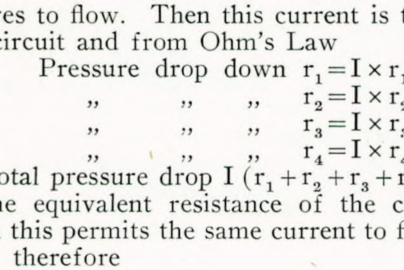


FIG. 9.

which applies to a whole circuit or to any part of a circuit,

$$\text{current in branch a} = \frac{E}{r_1}$$

$$\text{current in branch b} = \frac{E}{r_2}$$

$$\text{current in branch c} = \frac{E}{r_3}$$

then total current through the whole system

$$I = \frac{E}{r_1} + \frac{E}{r_2} + \frac{E}{r_3} = E\left(\frac{1}{r_1}+\frac{1}{r_2}+\frac{1}{r_3}\right)$$

Let R be the equivalent resistance

$$\text{then } I = \frac{E}{R}$$

dividing both sides by E

$$\frac{1}{R} = \frac{1}{r_1} + \frac{1}{r_2} + \frac{1}{r_3}$$

$$\text{or } R = \frac{1}{\frac{1}{r_1} + \frac{1}{r_2} + \frac{1}{r_3}}$$

Conductance.

In direct current work, the reciprocal of the resistance is called the conductance of the circuit, so that whereas in a series circuit the sum of the individual resistances gives the total resistance of the circuit, in a parallel circuit the sum of the conductances of the various branches gives the total conductance of the circuit.

There is another point worth noting here. If there are three equal resistances in parallel, the combined resistance is one-third of any one of them; if there are four, it is one-quarter, and in short if there are "n", then the combined resistance is 1/n of any one.

Let the equal resistances be r ohms each, then

$$\frac{1}{\frac{1}{r} + \frac{1}{r} + \frac{1}{r} + \text{ to n terms}} = \frac{1}{n/r}$$

that is, combined resistance $= \frac{r}{n}$.

This also links up with the fact that the resistance of a conductor varies inversely as its cross-sectional area, because if we have a conductor of a certain length and say 0·1 sq. in. in cross-section, and another of the same length and material of 0·2 sq. in. in cross section, we can think of the second as consisting of two of the first joined together in parallel.

There is another point which frequently presents some difficulty. The more resistances are added in parallel to a circuit, the more this *lowers* the resistance of the circuit, and the more it *increases* the load on the circuit, that is the load on the dynamo which supplies the circuit, and therefore the load on the prime mover which drives the dynamo. The difficulty appears to lie in appreciating why a *lowering* of resistance should *increase* the load. This point can perhaps be better understood by means of another hydraulic analogy. Suppose there is a large tank with four delivery pipes,

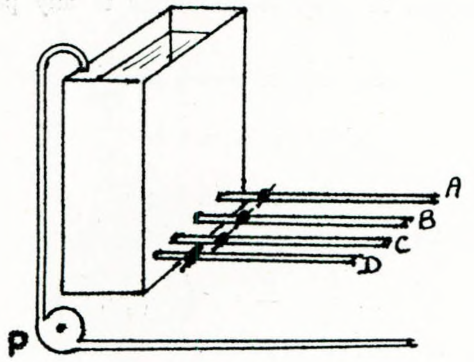


FIG. 10.

A, B, C and D, Fig. 10, the water in the tank being maintained at a certain level by means of pump P. If the delivery valve of pipe A only be open, then the pump will have to work at a certain rate in order to maintain the necessary pressure head. If the delivery valve of pipe B is also open, then the tank will tend to empty more quickly, and the pump will have to work still harder to maintain the same head, and, the more valves are open, the harder will the pump have to work in order to maintain the head. But in opening the valves we are *reducing* the resistance offered to water flowing from the tank, and by so doing making it necessary for the pump and its prime mover to work harder if the head is to be maintained, that is we have increased the load on the prime mover.

Electrical Work and Power.

Whenever a current passes through a conductor, heat is generated. Joule demonstrated that heat is directly expressible in terms of work, so that work must have been done during the passage of the electricity. This gives us one way of defining pressure difference.

The unit of mechanical work is the foot-pound. The corresponding unit of electrical work is the *joule*

$$1 \text{ joule} = 0{\cdot}7373 \text{ft./lb.}$$

A pressure difference of one volt exists between two points in a circuit when one joule of work is done in conveying unit quantity of electricity, *i.e.* one coulomb, between the two points.

Electrical Power.

Electrical work in joules = volts × amperes × seconds.

Electrical power = rate of doing electrical work

$$= \frac{\text{volts} \times \text{amperes} \times \text{seconds}}{\text{seconds}}$$

$$= \text{volts} \times \text{amperes} = \text{Watts.}$$

The Watt is the unit of electrical power and is the power expended in a circuit when a current of 1 ampere flows between two points the pressure difference between which is 1 volt.

746 Watts = 1 horse power,
just as 1 h.p. = 550 ft.-lb. per second,
so 1 Watt = 1 joule per second.

A very common mistake made by students is to describe 1 joule as being 1 Watt per second. This has just as much meaning as to describe 550 ft.-lb. as being 1 horse power per second, and that is, none at all, because 1 horse power per second strictly interpreted is 550 ft.-lb. per second per second, whatever that may mean. A joule is 1 Watt-second, *not* 1 Watt *per* second.

(To be continued).

Abstracts of the Technical Press

Superheated Hot-water Heating.

The superheated hot-water system of heating is often called high-pressure hot-water heating, but the former expression is to be preferred, as the heat transmission depends not on pressure but upon temperature, or, more precisely, upon the temperature difference between the heating and heating medium. The superheated hot-water system is characterised by flow temperatures above the evaporation point at atmospheric pressure, and these temperatures can be obtained only by a corresponding increase of pressure. The term "superheated" therefore indicates the rise of the water temperature above the atmospheric conditions of 212° F. by reception of more heat at increased pressure, contrary to the superheating of steam which takes place at constant pressure. The maximum temperature of the flow water is naturally slightly lower than that of steam at the same pressure, and the return temperature is even considerably lower. As the heating effect depends upon the average temperature, it might be supposed that the boiler pressure must be higher than in the case of a steam distribution system, but the required excess of pressure which varies with the temperature drop is actually almost of the same order as the pressure drop of a steam distribution system. The greatest advantage of the usual low pressure hot-water heating system with forced circulation is its excellent adaptability to all kinds of local conditions and its greater simplicity as compared to a steam-heating system. The superheated hot-water system has the same features, but with a far wider temperature range which admits of many variations to suit local conditions. The higher temperatures also allow of a bigger temperature drop than in the usual low-pressure hot-water heating system and the pipe sizes can be correspondingly reduced, the heat being carried to all points of the system by circulating pumps, while unit heaters, coils, or heating batteries can be used for the transmission of heat. A properly designed and constructed superheated hot-water heating system is fool-proof, and gives an excellent heat distribution and an effective heat supply to suit all weather conditions. All the unavoidable losses of a steam system due to condensate, drainage and venting, traps, by-passes, etc., are eliminated, while losses by neglect, often considerable in the case of high-pressure steam, are simply impossible. The design of a superheated hot-water system requires, however, more consideration and thoroughness than that of a steam system owing to the number of possible variations with regard to the method of heating the water, in boilers, calorifiers or cascade heaters, to the arrangement of boiler connections, circulating pumps and expansion space and to the distribution and transmission of heat. It is sometimes useful to arrange a double distribution system of superheated low-water and steam. The supply of both systems can be arranged from the same boilers, as indicated in Fig. 2, where the Rud. Otto Meyer System is shown in which the mixing of the flow water with a part of the return water takes place inside the boiler, thus ensuring quiet circulation and freedom from water-hammer under all circumstances. On the right side of the figure is shown the superheated hot-water system combined with low-pressure heating, and on the left the steam heating. Only one connection is required for the flow and return water, with a combined down pipe for each boiler. The superheated hot-water is taken from the boiler below water level, and the steam, as usual, from the steam space. Both services can be run quite independently of each other, and the steam space, if large enough, can serve as the expansion vessel. The use of the same boiler for steam and superheated hot-water supply enables first costs to be reduced and offers the advantage of lower running expenses, but special attention must be paid to the equalisation of water level to permit two or more boilers to be run in parallel for both services. The superheated hot-water system of heating is also very suitable for the utilisation of waste heat from different sources, as, for instance, the cooling water or exhaust gases of Diesel engines. The return water is more or less preheated by the waste heat and is then delivered to the boilers or calorifiers for the final temperature rise.—*A. Margolis, Dipl.-Ing., "The Industrial Heat-*

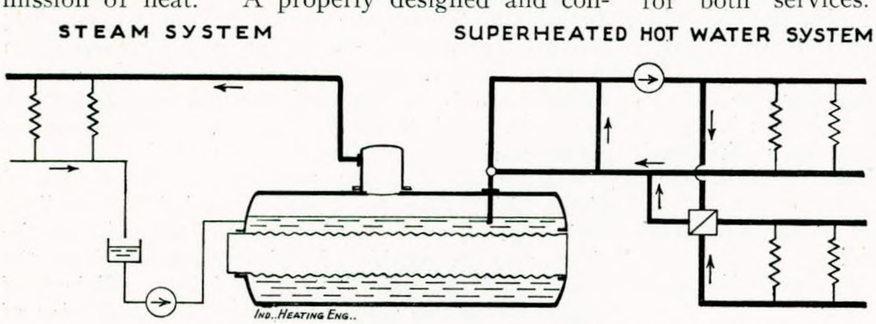


FIG. 2.—Steam and superheated hot-water supply from a steam boiler.

ing Engineer", Vol. 1, No. 1, January, 1939, pp. 3-4.

Reciprocating Steam Engines Without Cylinder Lubrication.

In spite of the advances made in the use of the steam turbine and the internal-combustion engine, the steam reciprocating engine is still a popular prime mover. If the exhaust steam from the steam reciprocator could be completely free

FIG. 1.—Types of unlubricated non-contact packings.

from oil, this type of engine would then be relieved of one of the criticisms levelled against it. Experiments to do away with the need for cylinder lubrication are very old (*e.g.*, the Perkins high-pressure unit), but no entirely satisfactory results have accrued. Fundamentally, there are two methods of achieving the desired end :—

1. Piston rings made of material capable of working without lubrication or of working with such substances as are not a disadvantage when mixed in the exhaust steam.

2. Avoidance of contact between the moving and stationary portion of the cylinder and piston and piston rod.

Under 1, there is first of all the ring made of carbon, if we except the newer unproved bakelised fabric materials. Although the carbon ring is

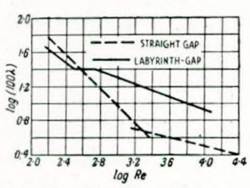


FIG. 2.—Flow resistance of straight gap and labyrinth gap packings.

successfully used in turbine construction, the conditions appertaining to a reciprocating ring are entirely different and the mechanical properties of the carbon ring seem scarcely adequate for the steam reciprocator. Under 2, we can place "gap" and labyrinth packings (see Fig. 1).

The properties of a "gap" packing are :

(*a*) Important effect of gap width on leakage; with streamline flow the leakage varies as the third power of the width. but with turbulent flow it varies as the 1·5 power of the width.

(*b*) Importance of the piston being maintained central, *i.e.*, uniform gap all round.

(*c*) Working capabilities are poor because seizure takes place easily.

The labyrinth-gap packing lies, in form, between the simple gap and the full labyrinth

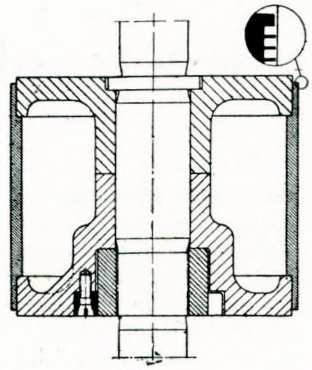


FIG. 3.—Labyrinth gap for piston.

packing. It is less sensitive to gap width and centring, and is therefore better suited to pack a reciprocating piston. Moreover, the working capabilities are much better, because the splitting up of the surface into thin strips which might come into contact with the stationary part permits of deformation and prevents seizure. Fig. 2 shows how the resistance varies with both gap and labyrinth-gap packings. As will be seen, it is only when the Reynolds number is very small that the straight gap is tighter. So far as reciprocating engines are concerned, most cases come into the range where the labyrinth-gap is the better packing, other things being equal.

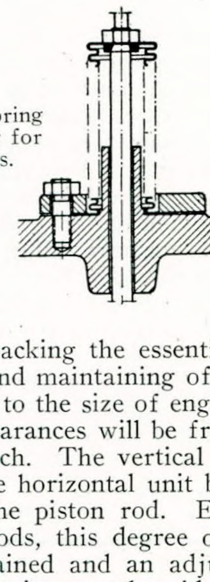


FIG. 4.—Spring tube packing for valve rods.

With simple gap packing the essential requirement is the securing and maintaining of a uniform small gap. According to the size of engine and the permissible leakage clearances will be from four to 20 thousands of an inch. The vertical engine has the advantage over the horizontal unit by avoiding possible bending of the piston rod. Except with mass-production methods, this degree of accuracy cannot usually be attained and an adjustment on the upper and lower piston rod guides, such as an eccentric ring method, is desirable.

With labyrinth-gap packing a special design of piston skirt (Fig. 3) is recommended for medium and large pistons. In this case the material of the skirt must be capable of resisting erosion from the high speed through the gap and its coefficient of expansion be the same as that of the remainder of the piston and of the cylinder; the material should also be rustless.

Glancing at the requirements for the cylinder itself, it will be seen that wearing properties need not be considered. To fulfil the same conditions as that for the piston skirt material, however, the best design would involve a separate sleeve insert with the space between cylinder and liner heated so as to be able to vary at will the mean wall temperature for thermal efficiency reasons and in order to maintain the exact clearance. The centring of the piston can be checked by three measurements on top and bottom dead centres taken through test holes drilled in position.

Piston rod glands can be run without oil by treating them in the same general way as the pistons, but a wider choice of solutions is possible. Carbon rings can be used without oil, or cast iron rings with graphite as a lubricant, or bakelised fabric used with water. Valves, etc., which must be run unlubricated by oil, can also be treated as above or with the spring tube shown in Fig. 4.

Tests carried out have given the leakages to be met under various conditions and on the basis of these tests it can be decided whether an expensive engine with small clearances and a minimum steam consumption or a cheaper, less economical machine would best suit the needs of a particular case.

The adoption of "gap" packing may have a far-reaching effect on the design of reciprocating engines. The absence of oil avoids the limits of temperature set by the properties of lubricating oil. The heat flow conditions are quite different when there are no piston rings to conduct heat direct to the cylinder walls. With no friction between piston and cylinder one of the principal limits to piston speed (or revolution speed) is removed. The use of gap packing is not, of course, limited to steam reciprocators; it can be adopted with equal advantages for compressors and gas engines, while its adoption for the coal dust engine would seem to warrant investigation from the point of view of cylinder wear.—*Dr. Ing. K. Trutnovsky, "Die Wärme", Sept. 17th, 1938; "The Marine Engineer", Nov., 1938, p. 345.*

Cellulose Sponges in Feed Water Filters.

Eight months ago the excelsior or burlap used in the feed-water filters of the Delaware-New Jersey Ferry Company's ferry-boats was replaced by cellulose sponges, and it is now claimed that the cost of filtering the feed water has thereby been reduced by 50 per cent. The feed filters in the ferryboats are 5ft. by 3ft. and the cellulose sponges are placed on baffle plates set up at a 45° angle. About 100 sponges are placed in each filter and used for one week, when they are removed and washed in steam vats containing a solution of soap dust. If the sponges are very oily, sal-soda is added to the solution. The life of the cellulose sponges may be as long as three months.—*"Marine Engineering and Shipping Review", Vol. XLIII, No. 12. December, 1938, p. 565.*

New Electrode for Depositing Machinable Alloy on Cast Iron.

A new electrode is being marketed in this country and the U.S.A. for depositing a soft machinable alloy on cast iron. It operates best with d.c. negative polarity, although a.c. may be used. While a single layer deposit will be machinable in the weld metal, at least two layers should be deposited in order to obtain a soft fusion zone, which may then be machined, sawn, drilled, and tapped without difficulty. Current should be just high enough to obtain the necessary bond, the arc being about ⅛in., and the electrode moved slightly from side to side. An average of 90 amps. and 18 volts at the arc, will generally prove satisfactory. This new electrode—designated "Softweld"—is primarily suitable for small fillings, as where a large or deep area is to be filled, or where a strength weld is required, it is more economical to use another type of electrode—like the "Ferroweld"—to within ⅛in. of the machined surface, and then finish off with a couple of layers of "Softweld".—*"Metallurgia", Vol. 19, No. 110, December, 1938, p. 17.*

Spot Light for Welding Work.

To obviate the difficulty frequently experienced in locating the exact spot at which the weld is to be made and to save time in doing so, a leading Dutch firm of incandescent lamp manufacturers have introduced a welding spot light consisting of a bottle-shaped metal housing containing a step-down transformer and a lamp and lens system, giving an intense beam of light which can be directed so as to form a brilliant spot on the work. The primary winding of the transformer is made to suit all standard a.c. voltages, and the lamp, supplied from the secondary, takes 1 amp. at 2·5 volts. If the housing is mounted about 10in. from the work, the spot produced is approximately ⅛in. in diameter and sufficiently bright to be easily visible in daylight. A universal mounting bracket enables the housing to be secured to a welding machine in such a position that the spot of light falls exactly where the electrode will make contact with the work and the operator has only to place the latter so that the spot falls on marks previously made to ensure that the welds will be made in the correct positions. For work requiring a long row of spot welds in a line, two spot lights will materially increase the speed of operation, as it is then only necessary to mark out the work with the line on which the welds are to be made and set the two spot lights to the required spacing between the welds. By placing the work so that one spot falls in the centre of the weld just made and the other spot appears on the marked line, the exact position for the next weld will be indicated.—*"Engineering", Vol. CXLVI, No. 3807, 30th December, 1938, p. 761.*

New Soviet Tonnage.

A number of new vessels have been acquired by Soviet shipping organisations during the past year, among them being the timber and oil-carrier "Collectivisatsia", which has been specially designed for the transportation of oil and coal and for mechanical loading and unloading. Two powerful oil carriers—the "Donbas" and "Azerbaijan"—have been acquired by the Black Sea Shipping Company, fifty steam tugs have been built during the year, and a considerable number of motor ships, steam cutters, barges and dredgers have been put into commission. Among other vessels to be completed by the end of 1938 are five two-deck steamers each of 3,000 tons' carrying capacity, timber and oil-carriers, cargo and passenger steamers and tugs. Three vessels built for the Soviet Government in Holland comprise the seagoing steam tugs "Squall" and "Respublica" of 900 i.h.p. and a dredger with a capacity of 26,500 cu. ft. per hour. Two 500 i.h.p. steam tugs—the "Zaria" and "Truzhennik"—built in Abo (Finland) are ready for delivery to the Central Ports Administration of the Commissariat of Water Transport and a fire-fighting cutter (with five Diesel engines) built in the U.S.A. is also to be delivered shortly.—*"The Syren", Vol. CLXIX, No. 2207, 14th December, 1938, p. 385.*

Mechanical Stokers in Coal-fired German Liners.

As a result of eight months' experience with the North German Lloyd liner "Aachen", fitted with Steinmüller automatic furnaces to her three boilers, it has been decided to convert five more coal-fired vessels of that company to mechanical firing, which is also to be installed in a new steamer now under construction. Although considerations of space precluded the use of automatic coal supply from the bunkers of the "Aachen", an all-round boiler efficiency of 85 per cent. was maintained with the Steinmüller furnace, which is considered to be very satisfactory for cylindrical boilers. The steam pressure in the h.p. boilers of the five N.D.L. vessels to be converted to mechanical firing, is from 500 to 750lb./in.^{2}.—*Shiffbau", Vol. 39, No. 24, 15th December, 1938, pp. 489-490.*

High Speed Coastal Liner of Novel Design.

The twin-screw m.v. "Cubahama", recently built by Henry Robb, Ltd., Leith, for the Bahama Line, of Nassau, is one of the very few ships built for American owners by a British yard, during the past 20 years. She is a vessel of 932 gross tons with cruiser stern and machinery aft, possessing unusually graceful lines and the relatively high speed of 15 knots for a ship of her class. The length b.p. is 250ft., the moulded breadth 38ft., and the depth moulded, to shelter deck 21ft., with two decks and two cargo holds. The propelling machinery consists of two 2-stroke cycle, single-acting Polar Diesel engines, each engine having seven cylinders, 340 mm. diameter by 570 mm. stroke and developing 1,120 b.h.p. at 250 r.p.m. on an estimated fuel consumption of 8·8 tons per day. The auxiliary machinery is electrically-driven by current supplied by three 240 kW. 220 volt Diesel-driven generators, with engines of 4-cylinder type, each developing 62 b.h.p. at 900 r.p.m., and there are two electrically-driven Reavell air compressors each of 41 cu. ft. capacity, six 3-ton electric winches, an electric capstan and windlass and electro-hydraulic steering gear. The capacities of the lower holds and 'tween decks total 97,000 cu. ft. grain of 90,000 cu. ft. bale and the double bottom is designed to carry 291 tons of water and 84 tons of oil-fuel. —*"Shipping World", Vol. XCIX, No. 2375, 21st December, 1938, pp. 680-681.*

Tests for Cylinder Wear in Powdered Fuel Diesel Engines.

A series of experiments undertaken at the Fuel Research Station with three single-cylinder J.A.P. overhead valve, water-cooled engines included tests for wear of cylinder liners of aluminium alloys, unhardened cast irons, hardened and tempered cast irons, nitrided cast iron and steel, anodised aluminium and chromium-plated cast irons. Tests were also made with ordinary cast iron pistons and rings and with specially tough, close grained cast iron pistons and rings. The effect of anodising, tinning, nitriding and chromium-plating these components was likewise investigated.

It was found that liner wear is almost wholly confined to the ring track and reaches a maximum at a point slightly below the top of the latter. Piston wear is confined largely to the thrust faces and is greatest at the top or first ring land, decreasing progressively towards the skirt. Ring wear is always most severe in the top ring and is usually least in the bottom ring.

In general, the harder the liner, the smaller is the rate of wear of liner, piston and rings, the lowest rates of wear being obtained with chromium-plated and nitrided liners and the highest rates with aluminium and unhardened cast iron liners. The former show a considerable advantage over ordinary cast iron.

The substitution of hardened rings for plain cast iron rings, in conjunction with a hard liner, usually reduces the rate of ring wear at the expense of a slightly increased rate of liner wear.

The best results to date have been obtained with certain types of chromium-plated liner in conjunction with ordinary cast iron piston and rings, the rate of liner wear being about one-seventieth, the top ring wear one-sixteenth and the piston wear about one-thirtieth of that obtained with ordinary cast iron liner, piston and rings. The actual rates of wear are still high, but the abrasive conditions applied during the tests were more severe than those met with in actual powdered fuel engines and it is considered that the rates of wear in practice, with suitable coal, should probably be nearer one-tenth of those found in these tests using similar combinations of materials.

Comparative tests to determine the influence which the fineness of the ash exerts upon wear rates indicated that the effect of increasing the fineness of the original powdered coal from 58 per cent. to 92 per cent. through 240 B.S. sieve, is to reduce the wear of liner, piston and rings by about 25 per cent. The finer of these two samples of ash approximates closely to the ash actually liberated in the cylinder of a powdered coal engine and gives rates of wear equal to about half of that obtained with the ash used in the main series of experiments.

Comparative tests were also made to compare the abrasive action of ashes from the bright and the dull fractions of a South Yorkshire medium caking coal, these ashes being prepared from coal samples ground to a fineness of 81 per cent. through a 240-mesh B.S. sieve and used in conjunction with ordinary cast iron liners, pistons and rings. It was found that the wear rates with the dull fraction of this coal were more than double those with the bright fractions.

Further tests will be conducted with a 2-cylinder Diesel engine, with a cylinder bore of 12in. and stroke of 18in., designed to run at 230 r.p.m., now being installed for conversion to powdered coal in one cylinder.—*"Report of the Fuel Research Board for 1938", pp. 220-224.*

Grooved Boiler Tubes.

A well-known tube manufacturing concern in Germany has developed a new type of grooved tube, which yields an increased heat transference by having a spiral groove cold rolled on the outside surface to a depth of about $\frac{3}{16}$in., the spiral having a pitch of about $\frac{1}{4}$in., and the tube being subsequently heat-treated to remove any residual stresses. It is claimed that actual tests have shown these tubes to possess a heat transfer capacity about 30 per cent. greater than that of plain tubes of the same diameter, while the mechanical properties of the grooved tubes are stated to be equal to those of the plain ones. The grooved tubes can be bent, expanded or welded in the same way as the latter.—*"Shipbuilding and Shipping Record", Vol. LII, No. 26, 29th December, 1938, pp. 795-796.*

New Japanese Cable Ship.

The submarine cable ship "Toyo Maru", built by the Kawasaki Shipbuilding Company, Kobe, for the Japanese Ministry of Communicatons, is the largest vessel of her type yet built in Japan. Of a gross tonnage of 3,718, she is 349ft. long, with a beam of 50ft. 9in. and depth of 27ft. 3in. Her propelling machinery consists of a steam turbine of 2,900 h.p., wth double reduction gears, giving the ship a maximum speed of 15 knots. Steam is supplied by three Brudhon Capus boilers, with superheaters. There are three decks, of which the shelter deck extends the full length of the vessel to facilitate cable work on board. Besides a screw propeller of the ordinary type, the "Toyo Maru" has a Voith Schneider propeller, driven by a 100 h.p. motor installed on the stern bottom deck, which enables the bow of the vessel to be readily turned in any direction. Cables, which are usually of the lead-sheathed multicore type, are paid out from, or picked up to, the main drum of 10ft. diameter, capable of handling 22½ tons of cable at a speed of one knot. The ship is provided with the latest equipment for laying and repairing cables, and her mechanical instruments for navigation include a Leoard electric steering engine, a Sperry gyro-compass, a Zigspie electric sound recording machine, a Barr & Stroud range finder, and a Telefunken radio direction finder. There are four electric generators, two of them being 60 kW. turbo-generators.—*"The Shipping World", Vol. XCIX, No. 2376, 28th December, 1938, p. 706.*

Silent Electric Motors for the "Queen Elizabeth".

Silent electric motors will be used for the ventilation system of the "Queen Elizabeth", there being 263 such motors, varying in power from ½ to 3½ h.p. for cabin ventilation. Metrovick motors of from 2 to 19 h.p. will be used for engine-room ventilation, 8 of the 21 motors being of the totally enclosed type for use in the air stream and driving propeller type fans, while the remaining 13 motors will be of the enclosed ventilated type. There will also be five totally enclosed deck-mounted motors for driving winches, six for use with separators and several small motors for the special gymnasium appliances.—*"The Imperial Review", Vol. V., No. 11, 28th November, 1938, p. 455.*

Special Type Cargo Vessels for New Zealand Coasting Trade.

The steamships "Komata" and "Kurow", built by Alexander Stephen & Sons, Ltd., Glasgow, for the Union Steam Ship Company of New Zealand, are specially designed for service on the New Zealand coast, where flood water sometimes runs at 14 knots past ships moored at quays. Both vessels are practically identical, but the following particulars apply to the "Komata" now on her maiden voyage to New Zealand.

The ship has a length b.p. of 330ft., a moulded breadth of 52ft. 6in., a moulded depth of 28ft., and a gross tonnage of 3,850 tons. Constructed on the Isherwood Arcform system, the hull is divided into six compartments by watertight bulkheads extending to the upper deck, the ship's keel is fitted with a steel rubbing piece for crossing river bars and the sides are protected by a heavy steel fender fore and aft, in addition to the usual wood fenders at the stern. The sternpost is streamlined, as is also the rudder. The ship is provided with extra large hatches of the Macgregor steel type, with twelve derricks over the three hatches in addition to a heavy derrick to lift 25 tons. The poop and forecastle are fitted for light cargo and the poop is made fireproof for a benzine cargo. There are also a number of horse stalls in the poop. The cargo winches and windlass are of the latest Clarke, Chapman design and are steam driven.

The propelling machinery, constructed by the shipbuilders, consists of a triple-expansion engine with cylinders of 24½in., 41in. and 68in. diameter, with a 48in. stroke worked by superheated steam at 200lb. pressure and 600° F. temperature supplied by three oil-fired cylindrical boilers with Me-Le-Sco superheaters and Howden's forced draught. These boilers are arranged for rapid conversion to coal firing and the oil fuel tanks can readily be adapted for use as coal bunkers if required, the 'tween deck coal bunker not being used for oil fuel and forming an extra cargo space worked by side doors in the shell. All engine-room auxiliaries take steam at full boiler pressure and to secure maximum economy of fuel, the installation includes a regenerative condenser, air pre-heating and two-stage feed-heating, the final feed temperature being about 300° F. Forced lubrication has been adopted for all main engine and shaft bearings. There is a Falco-Perfecto oil-fired range in the galley using gravity-fed oil without blowers and to provide lighting in port when cargo is not being worked, a large electric battery is installed, which comes into action automatically when the dynamo is shut down and is also connected to emergency lights in the engine-room and elsewhere. A cold provision store is provided, cooled by an automatic Hallmark No. 3½ W.H. machine. The accommodation for officers and crew is on a luxurious scale rarely found in British ships.

One of the minor differences between the "Komata" and "Kurow" is that one ship has the butts and laps of the shell-plating faired off by Aranbee composition, so that a comparison of their performances during service should be interesting. —*"Shipbuilding and Shipping Record", Vol. LII, No. 25, 22nd December, 1938, pp. 772-774.*

Uses for Plastic Materials on Board Ship.

Pressings made of synthetic resinous materials of great strength and elaborate form have now been developed to such an extent, that they can be utilised for numerous ship's fittings, including the following:—

Cabin lamp fixtures,
Electric lampholders,
Cabin door furniture,
Fiddles for securing to tables,
Wing nuts for scuttle fittings,
Handwheels for ventilating valves, etc.
Companionway handrails, etc. (in lieu of brass rails),
Window and door curtain rails,
Window frames for inboard spaces,
Casings of heating apparatus fixtures,
Covers and casings of all nautical instruments,
Mouthpieces of voicepipes.

The pressings used for these purposes are fire-resisting and made in a variety of finishes and colourings. Synthetic resinous materials require no varnish or other external finish and are admirably suited for use in connection with the manufacture of ship's furniture and interior decorations. They are not only chemically and mechanically stronger than wood, but far less inflammable and capable of withstanding temperatures of up to 480° F. for a short time without ill effect.—*"Shiffbau", Vol. 39, No. 24, 15th December, 1938, pp. 485-486.*

American Marine Developments.

The launch of the passenger and cargo steamer "Panama" for the Panama Railroad service between New York and the Canal Zone, is to be followed within a few months by that of her sister ships "Cristobal" and "Ancon". They are twin-screw, geared turbine ships 495ft. long, of 64ft. beam and 39ft. depth, having all-electric auxiliaries, raked bow, flush deck and a single funnel. Luxurious accommodation for 200 passengers is provided, with air-conditioning and other elaborate provisions for comfort. The equipment of each ship includes a stabiliser which continually indicates the metacentric height when the ship is at sea. The liners "Argentina", "Brazil", and "Uruguay", formerly on this service have now been reconditioned for a new service between New York and the east coast ports of South America. To determine the possibilities of aluminium and its alloys in shipbuilding, a 15ft. model was built three years ago and moored continuously in Chesapeake Bay. At its last inspection it was in good condition, thereby indicating that aluminium is satisfactory if properly protected by paint and electrolytic corrosion is controlled. The plates are of aluminium-manganese-chromium alloy, on frames of aluminium-manganese-chromium-silicon. Castings are of aluminium-manganese alloy and bolts and nuts of cadmium-plated steel. The deck plating is covered with heavy paper and teak plank, the wood being coated with aluminium paint. It is hoped to develop such construction for high-speed harbour police and patrol boats and similar small craft.—*"The Engineer", Vol. CLXVI, No. 4328, 23rd December, 1938, p. 714.*

Propelling Machinery of First Passenger Steamer with "Velox" Boilers.

The passenger steamer "Bore II" completed this autumn by the O.Y. Crichton-Vulcan A.B. of Abo (Finland) is the first vessel to be designed for propulsion by steam supplied by "Velox" forced-circulation boilers.

The "Bore II" is a single screw steamer of about 250ft. in length, with a beam of 39ft. and draught of 15ft. 6in., built for passenger service between Abo and Stockholm. Her propelling machinery consists of a double-compound engine of the Christiansen and Meyer type of 2,500 i.h.p. at 120 r.p.m., provided with steam at a pressure of

225lb./in.^{2} and temperature of 608° F. by two oil-fired "Velox" boilers supplied by Messrs. Brown, Boveri & Co., of Baden (Switzerland), who have developed this type of boiler. The "Velox" is a forced circulation boiler with an extremely high rate of evaporation, automatic regulation and pre-heating of the air used for combustion by means of exhaust steam. The total space occupied by the entire machinery and boiler installation in the "Bore II" is only 45ft. long and there is an auxiliary oil-fired boiler generating saturated steam at a pressure of 160lb./in.^{2} for harbour use and for starting up the "Velox" boilers. A Bühring oil separator and Lurgi feed-water filter are fitted to ensure the purity of the feed water which is heated by auxiliary exhaust steam. A special "wave" condenser with curved tubes rolled into the tube plates by a process is provided.

The advantages claimed for "Velox" boilers are saving of space (enabling the "Bore II" to accommodate 12 per cent. more passengers than if she were equipped with oil-fired cylindrical boilers), a considerable saving in weight, and high efficiency (91 per cent.) under all condtions between one-third load and overload, in addition to which the time required to raise steam from cold is only 20 minutes.

The performance of the German naval experimental vessel "Arkona" equipped with a "Velox" boiler has proved extremely satisfactory. After overcoming a certain amount of minor initial trouble, the boiler has been worked for nearly 6,000 hours without developing any defects.—*"Schiffbau", Vol. 39, No. 24, 15th December, 1938, p. 483.*

The "Hydrogap" Rudder.

Recent experiments at the National Physical Laboratory with a model of a 410ft. cargo vessel with a speed of 10·8 knots, have been succeeded by service experiments, the results of which should be available very shortly.

The "Hydrogap" rudder consists of a streamlined fixed portion—the rudder post in the case of a single screw ship—immediately astern of which (with only ½in. clearance) is the movable portion, so that when the rudder is put over, the gap between the fixed and movable parts increases. The construction is simple and claimed to give an increased turning moment per unit of rudder area, maximum propulsion efficiency, smaller rudder head and steering gear, and cheaper maintenance. The principle employed is the well-tried one used for slotted aeroplane wings. As the rudder is put over, the water pressure on one side increases and a narrow column of water is forced through the gap referred to above at accelerated speed, thereby inducing a speedier flow of the water on the outer side and increasing the relative pressure between the two sides and consequently the turning moment and manœuvrability of the ship.—*"The Shipping World", Vol. XCIX, No. 2375, 21st December, 1938, p. 677.*

Cone Propellers for R.N.L.I. Boats.

By a special design arranged by the Royal National Lifeboat Institution's engineers, the Hotchkiss cone propellers fitted in the new surf-type lifeboats now completing for six of their stations, can have the gearbox unit removed from between the cones without disturbing the seating of the latter. The six blades are slotted into the hubs and secured with brass bolts and nuts, each blade being separately removable through the handholes provided for the purpose. One of the safety features lies in making the blades of thin material, which although adequate for the stress of propulsion, bends should a solid object, such as a projecting spar or other obstruction, be forced up into the casing, thereby obviating damage to the engine or transmission gear and causing little loss of speed to the boat, notwithstanding the bending of the blades. Ball and roller bearings are used throughout, lubricated from the oil in the gearbox. The discharge portion of the open side of the cone is not covered with grids, but the inlet side (about two-thirds of the total area) is covered with self-clearing grids to guard against damage to the impellers from any objects which might be drawn into the cones with the entering water. Tests carried out with a 36ft. boat fitted with a Hotchkiss propeller, proved that the boat could be run through beds of rushes and manœuvred in a narrow creek strewn with logs and other floating rubbish. In Burma it has been found that the arrangement enables the beds of water hyacinth which abound in the Irrawaddy River to be negotiated satisfactorily. Prior to the adoption of these intake grids as the results of experiments carried out, at Wareham, no perfect method of taking in weedy water without fouling some part of the gear employed, had ever been devised.—*"The Motor Boat", Vol. LXIX, No. 1797, 30th December, 1938, p. 695.*

Use of Blow Lamps.

A memorandum on the use of blow lamps has just been published as Form 819 by H.M. Stationery Office (price 2*d*). It includes descriptions of typical blow lamps for use with petrol and paraffin, with a diagram of an excellent design of safety valve. The following points are emphasised: Petrol and paraffin blow-lamps are constructed for burning their respective liquids, and petrol should never be used in a lamp with a pump fitting. A safety valve is essential on both kinds of lamp and should be of such design as to make it difficult for the user to put it out of action. The paraffin lamp should have an air relief valve in addition to the safety valve, to enable the lamp to be quickly extinguished. In each type of lamp the filler tubes should extend into the container far enough to allow of an air or vapour space of one-quarter of the container's capacity. Lamps should be lighted in accordance with their maker's instructions and should not be heated on stoves or by another lamp.—*"The*

Foundry Trade Journal", Vol. 59, No. 1167, 29th December, 1938, p. 488.

Safety Petrol Tank for Aircraft.

A frequent cause of fire in aircraft crashes is contact between the fuel from the tank and oil ignited by its contact with hot metal, since oil will flash under these conditions, whereas petrol may merely vaporise. If fuel tanks can be constructed to withstand impact and distortion in the event of a crash without bursting, the risk of a major conflagration is reduced to a minimum. Such a tank has now been developed by a London firm after seven years' research work in co-operation with the Air Ministry, the results being embodied in various types of fuel tanks with features protected by British and foreign patents. The walls of the Henderson Safety Tank are built up in three layers to a thickness of $\frac{3}{16}$in., the inner wall, of copper sheet with soldered joints, being coated outside with a thin film of "Hencorite", a substance having many of the properties of rubber. This coating of "Hencorite" is in turn covered with an outer skin of copper sheeting, soldered at all joints like the inner wall. Each layer of the tank is a complete, leakproof unit. Baffles attached to the inner wall of the tank are secured by a special self-releasing device by which they become detached in the event of a crash, thereby enabling the pressure inside the tank to be evened up. There are no rivets, soft soldering being employed throughout and the tanks are rustproof and do not require normalising. It is claimed that they are leakproof and fire-proof when fired at by incendiary, tracer or ordinary ammunition. Damaged tanks are repairable and impervious to climatic conditions. In spite of the triple skin, the weight of the tank is very low—between 1lb. and $1\frac{1}{4}$lb. per gallon for capacities of 50 gallons, which is little more than that for tanks used on modern commercial aircraft. The Henderson Safety Tank has successfully passed the official drop, firing and flying tests and has been approved for all civil types of aircraft.—*"Flight", Vol. XXXIV, No. 1566, 29th December, 1938, p. 605.*

Saving of Weight in All-welded Tankers.

In the case of a 21,340 tons deadweight tanker recently ordered from the Chantiers Penhoët, the adoption of the all-welded method of hull construction will enable a total saving of 400 tons weight to be effected. Two 16,000 tons deadweight 14-knot tankers just ordered in Sweden will also have all-welded hulls which will enable a still greater saving in weight—800 tons per ship—to be achieved. A considerable reduction in cost accompanies this saving in weight and the owners will possess vessels of 16,000 tons deadweight instead of 15,200 tons as would have been the case had their hulls been riveted.—*"Journal de la Marine Marchande", Vol. 20, No. 1029, 22nd December, 1938, p. 1901.*

New La Mont Boiler on Test.

The 1,000lb./in.^{2} La Mont boiler installed at the works of G. & J. Weir, Ltd., at Cathcart, has now been tested at full normal load. The actual pressure maintained was only a little over 900lb./in.^{2} as a matter of practical operating expediency, and no difficulty whatever was experienced in obtaining the full 1,000lb./in.^{2} if required for test shop purposes. The reduction in the working pressure was adopted to save the faces of the boiler safety valves, as the turbine plant which it supplied takes steam at 850lb./in.^{2}. In considering the overall efficiency figure of 86·46 per cent., it should be remembered that this is achieved with considerable saving of space.

Abridged Results.

Evaporation	40·748lb. per hr. N.R.
Steam pressure	907·3lb./in.^{2}
Steam temperature ...	877·2° F.
Boiler exit gas temperature	766·6° F.
Economiser gas outlet temperature	295·6° F.
CO_2 economiser outlet ...	15·057 per cent.
Gross C.V. of fuel as fired...	10,865 B.T.U. per lb.
Moisture	16·77 per cent.
Overall efficiency of boiler plant	86·46 ,,
Heat lost in dry flue gases...	4·82 ,,
Heat lost by moisture in flue gases	5·50 ,,
Heat lost due to incomplete combustion	0·39 ,,
Heat lost due to combustible in ash	0·736 ,,
Unaccounted for and radiation losses	2·094 ,,
	100·000
Power absorbed by auxiliaries	1·25 per cent
Net overall efficiency on gross C.V.	85·21 ,,
Power absorbed by circulating pumps	0·55 ,,
Heat releases in combustion chamber	39,760 B.T.U. per cu. ft./hr.
Guaranteed overall efficiency	81·0 per cent
Guaranteed efficiency exceeded by	5·46 ,,

—*"Industrial Power", Vol. XIV, No. 159, December, 1938, p. 432.*

New Troopships.

The newly-commissioned P. & O. troopship "Ettrick" and the recently-launched Bibby Line troopship "Devonshire", are both twin-screw vessels of 11,279 tons gross, 490ft. × 63ft. × 35ft., sub-divided into 10 watertight compartments by bulkheads reaching to the upper deck and with a double bottom extending all fore and aft, arranged to carry oil fuel, fresh water and water ballast. There are five complete decks. Both ships can carry 1,150 troops, with separate accommodation for N.C.O.'s and families, in addition to 104 first-class and 90 second-class passengers. The propelling machinery is intended to give the ships a speed of about 14 knots at 120 r.p.m. and the designed b.h.p. is 6,300. The propelling machinery of the "Ettrick" comprises a pair of Barclay Curle-Doxford opposed piston airless-injection oil engines of 5 cylinders each, with a diameter of 560mm. and stroke of 1,680mm., crank driven scavenge pumps and Bibby-Doxford detuners for eliminating torsional vibration stresses at all speeds. The "Devonshire" has two sets of 2-stroke, single-acting Sulzer type engines of 8 cylinders each, with a diameter of 660 mm., and stroke of 1,040mm. All auxiliaries in both ships are duplicated and electrically-driven, current being supplied by four Atlas Diesel airless-injection 4-cylinder engines directly coupled to B.T.H. generators. Two composite exhaust heat and oil-fired cylindrical boilers in a compartment at the forward end of the machinery space, supply steam for hotel purposes.—*"The Shipping World", Vol. XCIX, No. 2376, 28th December, 1938, p. 708.*

Gas in Containers as Fuel for Small Craft on Inland Waterways.

The annual increase in the number of motor boats and motor vehicles presents a big problem as regards the supply of light oil fuel, owing to certain districts being situated at a great distance from the oil fields. One of the principal local fuels for inland waterway craft is gas of a condensable or compressible nature, which together with solid and liquid fuel could be utilized for internal combustion engines. This gas is procurable in sufficient quantities in certain regions of the U.S.S.R. and may be divided into two groups :—the first group, condensable gas and the second group, compressible gas. To the first group belong the large caloric hydrocarbons—propane (C_3H_8), butane (C_4H_{10}) and their compounds. They are obtained by re-working naphtha and coal and are found in the form of naphtha gas in the various oil fields. These gases as regards their calorific and octane values. are in no way inferior to petrol and paraffin. They may easily be condensed at a low pressure and comparatively low temperature. The advantage of these gases is the fact that they are capable of being condensed, therefore they are suitable for stowing in containers at a pressure of 15-20 atm., which is very convenient for transport. As these gases possess good caloric and anti-detonation qualities, they undoubtedly occupy the first place for gas engine fuel. To the second group belong the compressible gases, among which are the natural and industrial gases, also producer gas. Natural gases are obtained from gas beds in oil fields, the basic components of these being methane (CH_4). The industrial gases consist of illuminating and coal gas, these being by-products of different manufacturing processes. Producer gas although possessing moderately low caloric values as compared with the other gases, plays an important part in districts with large supplies of solid fuel. This gas is obtained from semi-stationary producer gas plants, which possess a number of advantages as compared with craft fitted with producer gas installations, *i.e.,* a supply of perfectly purified and cooled gas, increased calories up to 2,000 cal/m³ instead of 800 to 1,000 cal/m³. The use of compressed gas for motor craft engines is considerably more convenient than the installation producer gas plants. These gases are difficult to compress, the compression pressure being usually about 200 atm., and limited by the strength of the material from which the containers are made. The conversion of engines of small craft from liquid fuel to gas, presents no difficulty. The installation of an engine worked by gas is not complicated and inexpensive. There is no necessity to carry out alterations in the engine and the high anti-detonation qualities of compressed gas, permit an increase of power at the expense of an increase in the compression stage. For instance, in the case of a carburettor engine working on butane gas with a compression ratio of 4·38 the engine developed a power of 66 h.p. and with a compression ratio of 6·75, it developed 86·8 h.p. From a practical aspect, however, in order to retain a uniform working of the motor, it is advisable when switching over from liquid fuel to gas. to keep the compression when working with butane within the limit of 5 to 5·5; this will make it possible to work without any alteration, on either liquid fuel or gas. Tests carried out on an engine when working with butane and petrol with a compression ratio of 4·38, developed 66·0 h.p. when working with butane gas, and 63·4 h.p. with petrol. This increase in power may be explained by the increase in the caloric effect or the thermic content of the gas mixture and may be determined by the formula :—

$$N = \frac{H}{1 + a\ L_o}$$

Where N = Thermic content of the working mixture cal/m³

H = the calorific value of the fuel (cal/m)

a = coefficient of surplus air

L_o = quantity of air theoretically required for combustion, in cubic metres

The specific consumption of gas producing fuel expressed in grammes per h.p./hour, decreases in comparison with petrol. When working with butane the consumption of fuel was 208 gr./h.p./hour, and with petrol it was 305 gr./h.p./hour. Tests with engines using gas producing fuel (butane mixture) showed a decrease in the specific consumption of 10·1 per cent. as compared with petrol. Tests carried out in the National Research Department of the U.S.S.R. established the following advantages for engines when working with gas producing fuel :—(1) In all conditions of temperatures a gas engine may be started more quickly than a petrol engine. (2) A gas engine works steadily not only at a high number of revolutions and with heavy loads, but also at slow speeds and when running light; smoothness in running, complete combustion, and a smokeless exhaust are also characteristics of this type of engine. (3) Feeding the engine by gas has a beneficial effect on the motor, the lubricating oil is less rarefied and there is also less deposit of carbon on the inside of the cylinders, which ensures a longer life for the engine. An installation for a motor boat using condensed gas is very simple and consists of one or more gas containers, an evaporator, reduction device for reducing the gas pressure, and a mixer for mixing the gas with air, which then forms an explosive mixture and is thence led into the cylinders. Detailed descriptions and sketches are given for each of the foregoing fittings; a sketch and full description of a universal mixer which could be attached to any carburettor, or direct to the engine are also included. The successful running of gas-driven vessels with gas stowed in containers depends on a good gas supply. In addition to supply stations on shore, special floating stations may be arranged with gas stowal in suitable tanks; these floating supply bases could be supplied from a central station situated near the gas source. As regards the supply of compressed gas, it would be necessary to organize a series of gas distributing stations similar to those arranged for motor vehicles. These stations should be fitted with compressors for forcing the gas into receivers made of special steel of 600-750 litres capacity, with a pressure of about 350 atm. The gas is delivered from these receivers into the containers, the time necessary to charge a container being about 5 minutes. A station with a capacity of about 160 to 200 m^3 per hour would supply about 35 gas-driven motor craft for 24 hours. Gas installations as described occupy much less space than producer gas plants and this should be borne in mind when considering the possibility of their use for passenger vessels. The weight of an installation with gas containers is approximately a third of that of a producer gas installation.—*"Soudostroienie", No. 4, 1938.*

Progress of the Maierform Type of Hull.

Although only introduced 10 years ago, there are now 481 ships with Maierform hulls in commission or under construction, including passenger and cargo vessels. tankers, coasters and trawlers. The majority of the vessels are German (245), but 55 are British. The most recent Maierform passenger ship to be put into service is the Norwegian-American transatlantic liner "Oslofjord". The Maierform vessels at present under construction include several 15,000-ton oil-tank motorships of 15 knots service speed and 37 motor fishing vessels and trawlers. Exhaustive tank tests indicate that the Maierform type of hull in comparison with the most fully-developed conventionally-designed models of motor tankers, constitutes an improvement of 12 to 17 per cent. The various advantages claimed for the Maierform design may be summarised as follows:—

(1) Eight to ten per cent. saving of power effected, equivalent to a corresponding saving in fuel in calm weather.
(2) An average of some 25 per cent. less power and fuel consumption in service and adverse-weather conditions as a result of the diminished pitching movements and racing of screws.
(3) Two to three per cent. more cubic capacity in the holds forward.
(4) Two per cent. more deck space forward.
(5) Improved stability, due to the sloped sections and the larger water-plane areas.
(6) With the severely raked stem, there is far less danger in collision for the rammed or ramming vessel.

The adoption of the Maierform type of hull for trawlers has been particularly widespread, no fewer than 139 having been built to this form—including 50 for a single owner. This type of hull is now used almost exclusively in Germany for trawlers.—*"The Shipbuilder", Vol. XLVI, No. 351, January, 1939, pp. 48-49.*

Ice Temperatures.

Progress Report No. 35 of the Pacific Fisheries Experiment Station declares that "green ice", *i.e.*, ice just taken from the freezing tanks, is not ideal for the preservation of freshly-caught fish in the holds of fishing vessels. Blocks of ice straight from the freezing can must be wet, due to the thawing necessary to extract the blocks from the cans, and when such blocks are immediately crushed and discharged into the hold of a vessel. wet particles of ice become intimately mixed with particles at a temperature well below freezing point, the result being the formation of a solid frozen mass which makes it almost impossible to ice down the catch. Although fishermen have come to the conclusion that ice is improved by long storage, the period of storage has actually no effect upon the cooling capacity of the ice. What is important, however, is the temperature of the ice as it enters the hold, and the lower the temperature, the greater the cooling capacity and the greater the chances of the ice remaining free-running in the hold, as except for a frozen layer on the top and around the edges of the pile where it comes into contact with warm air. the ice will be more powdery and dry. It is, therefore, unnecessary to store newly-made ice longer than the time required to refreeze the wet outer surface of the blocks and let the mass come to the temperature of the storage room. There is some advantage to the fisherman if the temperature of storage is maintained fairly low, as not only does the ice then remain free-running and more convenient to handle, but also because each 3° F. lowering of the storage temperature below freezing point adds approximately 1 per cent. to the cooling capacity of the ice, so that ice stored and entering a hold at 10° F. has 6·9 per cent. refrigeration added. Experiments were made with ten different kinds of ice to determine the comparative cooling capacities of natural ice, snow, very finely crushed ice. coarse ice, and benzoic acid ice. The results showed practically no difference in the latent heat of fusion or "cooling capacity" of the ordinary water ices commercially used. Further investigations of the relative temperature effects induced by "chunk" ice and finely crushed ice, disproved the popular belief that because coarsely crushed ice lasts longer in the hold than finely crushed ice, it is better for cooling purposes. If the finely crushed ice disappears faster in the hold, it indicates that it is absorbing heat more rapidly and, therefore, must be inducing a lower temperature, thereby serving its purpose better than the coarse ice. Since the whole object of taking in ice is to preserve the fish and not the ice, and it is recognized that the lower the temperature the better the fish keeps so long as freezing does not take place, the foregoing results show that finely crushed ice is better than coarse ice for preserving fish.—*"Ice and Cold Storage", Vol. XLII, No. 490, January, 1939, p. 2.*

New Motorships for German East-African Line.

The motorships "Kamerun" and "Togo", recently put into commission for the above service, are vessels of some 5,000 tons gross, with a cargo capacity of 388,000 cu. ft., including six tanks for the carriage of palm oil, of which large quantities are transported by the company. The propelling machinery of both ships consists of a M.A.N. Diesel engine of 5,100 h.p.. giving a service speed of 15 knots, all the auxiliaries being electrically driven. The cargo handling equipment includes 22 three-ton derricks and a fifty-ton heavy-duty derrick. Two new motorships, each of 6,900 gross tons with a speed of 17 knots and a cargo capacity of 530,000 cu. ft., have just been ordered by the company and should be ready for service by next year.—*"Journal de la Marine Marchande", Vol. 21, No. 1032, 12th January, 1939, p. 47.*

By-pass Valves for Steam Traps.

The author deprecates the neglect to which steam traps are subjected after installation which is often encouraged by their inaccessible position. In view of expense and space occupied by a by-pass, a new design incorporates it directly in the trap. The combined by-pass and shut-off valve is double-conical closing on an upper or lower seating. To operate the by-pass the valve is raised half-way and the water accumulated is blown to the outlet pipe, the rapid flow carrying away any dirt, which is prevented from reaching the trap discs by a screen fitted below them. and by an automatic valve in the passage to the outlet pipe. The valve consists of a sleeve with closed bottom, spring-loaded and with a port in one side; in the by-passing position the flow lifts the valve and closes this port cutting the connection to the upper side of the discs. For inspection, the by-pass valve is raised into contact with the upper seating, the water drains away and the cover and discs can be removed. The device is also applicable to a floating trap.—*H. Richter, "Engineering", Vol. 146, 18th November, 1938, p. 604.*

The Marine Propeller.

The conditions affecting propeller performance are numerous, and to obtain optimum performance would be laborious even if they were known accurately and remained reasonably constant. Fundamentals were laid down by Rankine and Froude over 70 years ago, but only with the general establishment of tanks and sensitive recording apparatus has rapid progress been made in correlation of model and full-size work. The editorial points out the increase in the number of papers dealing with propellers which has taken place in recent years, and that "Queen Mary" won the Atlantic blue riband with screws weighing 10 tons less than those originally fitted. Further, the rapid development of theoretical treatment and of the speed, size and power of ships, requires extrapolation from existing data or extension of previous researches, hence the lag in direct application of results and consolidation of progress made. The 10 papers presented to the N.E.C. Institution of Engineers and Shipbuilders in their 1938 Symposium have now been reprinted in book form at 21s. (*E. & F. N. Spon, London*). Design, once the province of the engineer, now the responsibility of the naval architect, is rapidly passing into a preserve of the mathematician, at least on paper; probably three-quarters of the 30,000 screws now at sea could be improved by tank tests, but on the other hand many satisfactory screws have been designed by trial and error. Abell and Dyson have both remarked on the difficulty that they would find in designing a really *bad* propeller, while considerable research is required to improve efficiency by 5 per cent. Finally the editorial pleads for a pooling of the knowledge accumulated by owners, firms and tank staffs, now regarded as private property, in the public interest.—*Leader, "Engineering", Vol. 144, 25th November, 1938, pp. 619-620.*

The Hagan Automatic Boiler Control System.

In the Hagan system of pneumatic control a master regulator reacts to changes in boiler load as expressed by variations of the main steam pressure and controls the heat input to the furnace to suit the load change by simultaneously varying the supply of oil fuel and combustion air in measured quantities and in the correct ratio. This is effected by generating in the master sender a pneumatic loading pressure in accordance with the prevailing steam pressure. A 3-way tranfer valve in the master loading pressure line between the master regulator and pilot valve of the forced draught regulator permits a change-over from automatic to manual control to be carried out and adjustment of the forced draught damper by merely turning a handwheel on the control panel. The fuel oil supply to the burners is controlled by measuring the gas flow through the boiler and using the gas flow as a measure of the air flow by registering the draught loss through the boiler in an air metering regulator which, by means of a double throttle device, converts the registered draught loss into a loading pressure. This loading pressure is transmitted to the outside of the bellows of the fuel oil control valve which serves to maintain an accurate balance between the loading pressure and oil pressure at the burners. The oil flow through the burner nozzles varies as the square of the oil-pressure and as the loading pressure is proportional to the draught loss and therefore varies with the square of the flow of combustion air, a linear relationship between air flow and fuel oil discharge from the burner is achieved. As the proportioning beam of the oil flow control valve rests on a moveable fulcrum, the fuel-air ratio can easily be adjusted by turning the handwheel of the screw wihch shifts the fulcrum. A by-pass connection is also provided to allow manual control of the fuel oil supply to the burner. The Hagan automatic control system is fitted to the Babcock & Wilcox boilers of the American tanker "W. H. Berg", and over a score of other American ships with both oil-fired and coal-burning boilers.—*D. W. Rudorff, Dipl.-Ing., "The Marine Engineer", Vol. 62, No. 737, January, 1939, pp. 12-14.*

Low Pressure Turbine Blades.

In the house journal of a leading British firm of turbine manufacturers a description is given of an improved type of low-pressure blade which enables an improvement in efficiency at the exhaust end of the turbine to be secured, or, alternatively, greater outputs at higher speeds can be obtained without sacrificing efficiency or reliability. The form of blade is a modification of the spiral or twisted form hitherto used at the low-pressure stages, the change being based upon a recognition of the fact that the pressure of steam varies along the length of the blade, being low at the root and high at the top owing to its reactive effect, and this is controlled to balance the effect of centrifugal force, thereby avoiding the radical flow.—*"Shipbuilding and Shipping Record", Vol. LIII, No. 1, 1st January, 1939, pp. 3 & 4.*

Electrically-operated Siren for Large Ships.

An electrical siren constructed by a German firm requires only 2 per cent. of the amount of compressed air used in the ordinary type of compressed air siren when giving a 7-second fog signal at 60-second intervals. In the new siren the usual diaphragm is replaced by a piston, which can be made of any desired diameter, according to the volume of sound required. This piston is moved backwards and forwards in the cylinder by a motor-driven crank mechanism by means of a friction clutch, at a speed in resonance with the oscillating air column at the base of the sound pipe, thereby converting the kinetic energy of the motor armature into sound which is transmitted by the sound pipe. The crankshaft is carried in heavy double ball-bearings lubricated by oil forced through the hollow shaft under pressure. All moving parts are of high-grade hardened chrome-nickel steel and are made as light as possible, while the cranks themselves are of stainless steel. The sound is produced by running the motor up to such a speed that resonance occurs in the sound pipe and then switching off, so that a signal of a number of notes is obtainable simply by reducing the speed of the motor. The running of the motor is controlled by an air-damped contactor which cuts out the starting resistance in two stages within one second, and the resistance is re-inserted when a period of silence is required between two notes, so that the motor is slowed down. A further note can be produced at full strength and without any delay by cutting out the resistance. This method of operation, which is controlled by push-bottons on the bridge, is possible because resonance and thus sound production only occurs when the speed of the motor reaches a certain value. At sea under good acoustic conditions it is stated that a range of 20 nautical miles has been obtained and that the note emitted is pure and clear, being unaffected by secondary noises such as those due to steam or air. A further advantage of the electrical siren is that it is not subject to freezing, so that its position is not dependent on proximity to the boiler or engine rooms. By placing the siren in the bows, for instance, the fog signals of the ship itself are inaudible in the passenger cabins and public rooms. Most recently-built German liners are fitted with this type of siren, which has proved most satisfactory in service.—*"Engineering", Vol. CXLVII, No. 3808, 6th January, 1939, p. 25.*

'Loded' Cast Irons.

'Loded' irons may be produced in any iron-foundry, subject to proper metallographic control and are in effect an extension of the grey cast irons made in refractory or metal moulds. They are all-pearlitic in structure and usable without heat-treatment. Iron castings of the all-pearlitic class are low in silicon-content, this ranging from about 0·95 to 2 per cent. Sometimes a little chromium is present, usually accompanied by about double the amount of nickel, so that an all-pearlitic iron for heavy work may contain 1 per cent. silicon, 1 per cent. nickel and 0·5 per cent. chromium; for lighter work the silicon is increased to about 1·5 to 1·75 per cent., the chromium content rarely being as high as 0·75 per cent. The 'Loded' iron process departs entirely from the low-silicon range, employing any silicon content from 2·5 to 7 per cent., many castings being made with 4·5 and 5·5 per cent. of silicon. These contents are stabilized by a high chromium content—often from 1 to 4 per cent. more. 'Loded' cast iron of high silicon-content can to-day be substituted for any all-pearlitic irons, its high chromum and high silicon content being conducive to great stability under heat, while the material retains the bearing metal properties common to grey cast iron and the wear-resisting properties peculiar to the all-pearlitic structure. 'Loded' cast irons may also contain small amounts of tungsten, vanadium, titanium, and other rare elements which may be fancied, while from 0·5 to 1·5 per cent. of nickel may replace a little of the silicon, if desired. 'Loded' cast iron may be cast centrifugally and such castings are being tried as cylinder liners in commercial vehicles, motor boats, auxiliary engines, etc., the results to date being encouraging and suggesting that the working surface of the metal is uncommonly resistant to wear. The Brinell of 'Loded' irons goes up with the silicon content without affecting their machinability and in large cylinder liners of 'Loded' iron ,the Brinells have been as high as 400 and more. A tensile of 17 tons/in.^{2} is obtainable on a 'Loded' iron of 5 per cent. silicon content and the castings are unusually free from defects like slag inclusions, blow holes, etc. The machined surfaces resemble steel.—*H. J. Young, "Transactions of the Institute of Marine Engineers", January, 1939.*

Ship Built in Disused Canal Lock.

The American all-welded motorship "Dolomite 4" now trading out of New York, was designed and built by Messrs. John H. Odenbach in a disused lock of the Erie Canal at Rochester. The ship's hull is composed of 18in. structural channel sections extending round the vessel in a transverse direction and welded together, the flanges of the channels being on the inside to form the frames and the channels being bent so that the deck edge and bilge have the same radius. The main cargo tanks are lined with pure sheet-nickel (27 tons of this material being used for this purpose) and steam tank-cleaning apparatus is installed by which the tanks can be completely cleaned in six hours. The ship can, therefore, carry any type of cargo, ranging from crude oil to wheat or potatoes.—*"Fairplay", Vol. CL., No. 2904, 5th January, 1939, p. 4.*

Protecting Wooden Fans.

Although wooden ventilating fan blades are light and able to withstand the effects of vibration, they are not hard enough to withstand erosion and abrasion, in addition to which they have the disadvantage of being hygroscopic and therefore liable to distortion and consequent loss of efficiency. To overcome these drawbacks, a well-known British firm specialising in the manufacture of this type of fan, has perfected a process for protecting wooden blades with a plastic finish, which, it is claimed, not only seals the wood and thereby excludes moisture from the atmosphere, but yields a surface which is as resistant as metal to the effects of erosion and abrasion. The finish is actually superior to metal, since the plastic material is non-corrosive as regards the effects of moisture. It is also claimed that hard wooden blades with a plastic covering are fire-resisting, although for fans subjected to high temperatures in actual service, metal blades are to be preferred.—*"Shipbuilding & Shipping Record", Vol. LIII, No. 1, 1st January, 1939, p. 3.*

High Speed Roumanian Liners.

The new motorship "Basarabia", like her sister ship the "Transilvania", has been built and engined by Burmeister & Wain, Ltd., for the Roumanian Maritime Company's bi-weekly service between Constantza and Alexandria. The ship is a twin-screw vessel of 6,672 tons gross, with a length B.P. of 405ft., a breadth of 57ft. 9in., and a depth of 30ft. 3in. Theer are nine watertight compartments and the ship is able to float with any two of them flooded. The cellular double bottom extends from the collision bulkhead to the after-peak tank. There are two complete decks, with superimposed shelter promenade and boat decks. There is accommodation for 80 first-, 100 second-, and about 230 third-class passengers. The four general-cargo holds and one hold space devoted to insulated chambers, have a total capacity of 100,000 cu. ft. The propelling machinery consists of a pair of B. & W. single-acting, 2-stroke Diesel engines of 12 cylinders with a diameter of 620mm. and stroke of 1,150mm., developing a total of 14,400 i.h.p. at 120 r.p.m., and designed to give the ship a sea speed of 22½ knots. Two rotary blowers at the back of each engine are chain-driven from the camshaft. The pistons are oil-cooled, while the cylinders and covers are fresh-water cooled, each main engine having its own cooling-water and lubricating-oil systems. The auxiliary machinery is in a separate W.T. compartment forward of the main engine-room and includes 3 B. & W. single-acting, 2-stroke, six-cylinder Diesels each driving a 240 kW. 220-volt dynamo. The cylinders of these engines are also fresh-water cooled. There are two electrically-driven air compressors with a capacity of 530 cu. ft. each, one 50-ton fuel transfer pump and one 15-ton fuel-oil service pump. There are also two 20-ton bilge and sanitary pumps, one 200-ton ballast pump, four 10-ton fresh-water pumps and one stand-by bilge pump, all the pumps being electrically driven. The two oil-fired and two exhaust-gas fired auxiliary boilers are all connected to a common steam receiver. The total capacity of the three insulated cargo spaces and the various provision rooms amounts to 15,125 cu. ft. In addition to refrigeration, the plant is also used for air-cooling in connection with the air-conditioning equipment. There are 3 compressors of equal size—one for the cargo chambers, one for the provision rooms, and one for air-conditioning. Each compressor is of the vertical double-acting type direct-driven by a 15 h.p. electric motor running at 270 r.p.m. The motor for air-conditioning can, however, be run at between 270 and 320 r.p.m. There are two CO_2 condensers. The four cargo winches and windlass are all-electric and there are also four electric cargo cranes to supplement the derricks. The steering gear is all-electric. There is an emergency dynamo on the boat deck and the navigating equipment includes a gyro-compass, radio direction-finder and Husun echo sounder. Among the ten boats is a 19ft. motor lauch driven by a Penta engine. Fire protection equipment includes a carbonic-acid plant, asbestos-covered bulkheads, smoke detectors and foam extinguishers. During her sea trials in December, 1938, the "Basarabia" averaged a speed of 25·5 knots over the measured mile.—*"The Shipbuilder", Vol. XLVI, No. 351, January, 1939, pp. 29-33.*

New Motorships for Harwich—Flushing Service.

The "Koningin Emma", the first of two vessels for the Harwich-Flushing service of the Zeeland Steamship Company run in connection with the L.N.E.R. boat train from Liverpool Street Station, is due to be launched from the de Schelde Shipbuilding Yard on the 14th January, while her sister ship "Prinses Beatrix" is expected to be launched in February. The vessels will be of 3,500 tons. with a speed of 23 knots and the propelling machinery will consist of 2-stroke Sulzer Diesels of 13,000 h.p. The carrying capacity will be 1,900 passengers, 60,000 cu. ft. of cargo and 24 motor-cars between decks.—*"Shipbuilding & Shipping Record", Vol. LIII, No. 2, 12th January, 1939, p. 61.*

Bread Freezing.

In a paper read at a recent meeting of the American Chemical Society, the preservation of bread by freezing was exhaustively discussed. According to physical and chemical tests, bread stales very rapidly at temperatures as low as −8° F. and is nearly always stale after 24 hours in the freezer, but at temperatures of about −30° F. bread retains its original fresh condition for about four days and takes eight to ten days to become nearly stale. When the tests at −30° F. were continued for a period of 60 to 70 days, a refreshing of the bread was observed and the results returned to values within the region of those for unfrozen bread eight hours out of the oven. According to aroma and flavour tests by consumer-judges, bread frozen at −8° F. remained "good" for a period of 20 days and saleable for about 40 days, but it had a much better keeping quality at the lower temperature and remained saleable for longer periods of time. Although bread does not keep indefinitely at such low temperatures, freezing offers an excellent means of keeping commercial bread in a saleable condition for forty days or longer.—*"Ice and Cold Storage", Vol. XLII, No. 490, January, 1939, p. 6.*

Marine Propeller Blade Vibrations.

The paper deals with the form of propeller blade vibration known as "singing" which, in most cases, is a note of fixed pitch which may be continuous or intermittent. The geometrical and mechanical characteristics of propeller blades and of their operating conditions are described and reference is made to the existence of various periodic forces exciting vibration, varying in nature and magnitude. The modes of vibration are examined, with calculations of their natural frequencies, tending to show that singing is caused by torsional vibration of the blades. The variable wake causes alterations in the angle of incidence and consequent changes in the resultant hydrodynamic force on each blade in regard to magnitude, direction and location; in particular, the centre of pressure may oscillate in position about the flexural centre, thereby causing changes in sign of the twisting moments and thus exciting torsional vibration. Possible resonances are discussed and methods of obviating singing are examined. The adoption of blade centres having fixed centres of pressure is advocated, since the phenomena involved affect both the hull and propeller, and the existence of a variable wake cannot be avoided.—*Paper read by J. F. C. Conn, B.Sc., before the Institution of Engineers and Shipbuilders of Scotland, 10th January, 1939.*

The Electric Propulsion of Ships.

Since Jakobi's experiment in 1839, the electrical propulsion of ships has gradually developed both as regards d.c. and a.c. The first electrically propelled submarine was built in 1886, the U.S.N. collier "Jupiter" had induction-motor drive in 1913 and the first passenger ship with electric drive—the "Cuba"—appeared in 1920. The first large vessel to have Diesel-electric drive with a.c. transmission was the "Wuppertal" built in 1936. The proportion of new vessels with turbo-electric drive is small and in the latest returns it is recorded that only 41 ships of 468,583 tons are so propelled out of a total of 1,264 vessels of 9,439,884 tons driven by turbines. The successful progress of electric drive in the case of small and moderate-sized vessels is more definite, more especially as regards Diesel-electric drive which is being increasingly used in inland waters and for small vessels of under 100 tons gross which are not included in classification returns, since the figures recorded—69 vessels of 176,407 tons—are numerically far short of the total. Probably 250 vessels are driven by Diesel-electric systems and their total s.h.p. is of the order of 250,000. It is estimated that 42 per cent. of new ships under 500 h.p., 37 per cent. of from 500 to 1,000 h.p. and 21 per cent. of over 1,000 h.p. are equipped with Diesel-electric propelling machinery. The established success of electric tugs promises a further increase in the number of small vessels so driven, and the advent of Diesel-electric drive with a.c. transmission seems likely to increase the proportion of the larger category, even if it would be unwise to suggest that steam has shot its bolt in the case of moderate powers.—*Paper read by L. R. Horne before N.E. Coast Institute of Engineers & Shipbuilders, 15th December, 1938.*

Fast Tankers Needed for the British Navy.

In the Royal Navy there are no large tankers with a speed of over 12 knots—a position which appears to indicate a lamentable lack of initiative on the part of the authorities concerned. The Japanese Navy has a number of 18-knot motor tank ships and two 16,300 18-knot tankers completing for the Standard Oil Company, have been purchased by the United States Navy. There are altogether 12 such tankers under construction and no doubt, if necessary, the U.S. Navy could take them all. Two 19-knot tankers have just been ordered for the French Navy. Plans are stated to have been prepared for building 18-knot tankers for private owners in this country, but apparently it has not been found possible to arrange with the Admiralty for such tonnage, if built, to become available for service with the Fleet in time of emergency. This matter should receive immediate attention, as the cost involved would be negligible if tanker owning companies would build vessels of this class under terms favourable to the Admiralty. This they are prepared to do.—*"The Motor Ship", Vol. XIX, No. 228, January, 1939, p. 348.*

Refrigerating Plant of French Fruit Carrier.

The refrigerating machinery of the new motorship "Maurienne" recently completed by Burmeister & Wain, for the Cie Générale d'Armements Maritimes, Paris, was supplied by a leading Danish firm of refrigerating engineers. The refrigerated space totals about 170,760 cu. ft., divided into 12 compartments cooled by eight air cooler batteries on the new Sabroe patent direct multi-temperature system with direct expansion of ammonia, which enables the various cold chambers to be maintained at the correct temperature without restricting the refrigerating surface. The plant includes two vertical 3-cylinder NH_3 compressors, two multitube condensers, two sea-water pumps, four air cooler batteries and two electric distance thermometer installations.—*"Lloyd's List & Shipping Gazette", No. 38, 744, 4th January, 1939, p. 18.*

Tank Strapping.

Tanks may be calibrated in several ways. The method discussed here is that of "strapping" or measuring apparent circumferences around the tank and making the necessary corrections later for all factors affecting the final tables. The most important corrections involved are those to apparent external circumferences required to obtain the mean horizontal cross-sectional area for each course. The effects of temperature, tension or errors of graduation of the tape, and of displacements of the tape from the tank surface by vertical seam edges or other obstacles, or through local distortion of the tank, plates must be considered. The effects of slight conicity and elipticity of the courses must likewise be taken into account, as must also the tilt of the tank as a whole. Corrections depending on the mean temperature of use of the tank and its expansion under the head of oil in it may also have to be made, while the effect of plate and paint thickness and internal fittings must not be overlooked. The tables may have to be adjusted to allow for irregularities in the shape of the tank bottom.—*Paper read by P. Kerr, M.A., B.Sc., at a Meeting of the Institute of Petroleum on 10th January, 1939.*

New Swedish Destroyer.

The Swedish destroyer "Malmö", launched last September, is now nearing completion. The vessel is 312ft. long overall, with a beam of 29ft. 6in., a moulded depth of 17ft. 6in., and a mean draught of 8ft. 5in. The displacement is 1,080 (metric) tons and the designed speed 39 knots. The propelling machinery consists of de Laval geared turbines of 32,000 s.h.p. driving twin screws, supplied with steam at 367lb./in.^{2} by three Penhoët boilers. The ship's armament comprises three 4·7in. guns, six 1in. pom-poms in twin mountings, six 21in. triple torpedo tubes, twenty mines and two depth charge throwers. Electric welding and light alloys have been used to a far greater extent in the construction of the hull than in the case of any other ship yet built for the Swedish Navy. The living accommodation for the 120 officers and men is on a much improved scale. The "Malmö" has been built by Eriksbergs Verksted in Gothenburg and a sister ship (the "Karlskrona") is under construction at the Government Dockyard at Karlskrona.—*"Schiffbau", Vol. 40, No. 1, 1st January, 1938, p. 14.*

Re-engining and Reconstruction of N.D.L. Liner.

The North German Lloyd liner "Nienburg" has recently been lengthened by 18ft., and has undergone partial reconstruction and re-engining at the Deutsche Werft in Hamburg. A new poop has been built onto the vessel. The new propelling machinery consists of an A.E.G. geared turbine of 4,200 h.p., supplied with steam by new La Mont boilers with Steinmüller mechanical firing, giving the ship a service speed of 14 knots (four knots higher than her original speed). The living accommodation for the ship's company of 42 has undergone a marked improvement. After successfully completing trials on the 13th December, 1938, the ship left Hamburg for Antwerp and South America, four days later.—*"Schiffbau", Vol. 40, No. 1, 1st January, 1939, p. 16.*

Methods of Starting Oil Engines.

Papers contributed by leading oil engineers were discussed at a meeting of the Diesel Engine Users' Association recently held in London. It was stated that with the exception of small high speed engines provided with electric starting and built for special requirements, compressed air starting has been standardised. A large percentage of the various mishaps has been associated with the starting period in oil engines, although modern engines are easier to start than the earlier air-injection engines. The instantaneous pick-up of the airless-injection engine and the absence of misfiring as soon as up to speed are due to improvements over earlier types where consideration had not been given to air-blast control at the flame plate. With airless injection engines priming of the fuel-injection system when starting is eliminated if the pumps and pipe are arranged to prevent air-locking or loss of fuel in the system when standing. Severe strains can be imposed upon an engine by overload fuel charges during the first few revolutions if the pump setting given by a governor on its maximum stop is not controlled. Relatively warm circulating water is preferable to an icy cold supply and it is important to keep an engine warm while standing by choking the water supply to guard against the risk of piston seizure. Piston skirt lubrication is generally advantageous. A popular type of starting equipment used with modern airless-injection engines employs pressures of from 250 to 450lb./in.^{2} with a riveted type air receiver, master starting valve of double-beat design at the engine control station, pilot valves driven off the cam-shaft and non-return valves in the cylinder heads (of 4-stroke engines). In 2-stroke (Fullagar) engines, the non-return valves are in the cylinder liners themselves at the mid-point of the combustion space. In the case of marine engines starting is employed on all cylinders and for the largest size of Fullagars, four starting lines are used. An essential condition of good starting when compressed air is admitted into the engine cylinder, is a finely divided fuel jet injected into compressed air or against a hot surface capable of spontaneously igniting the fuel.—*Abstract in "The Journal of Commerce" (Shipbuilding & Engineering Edition), No. 34, 600, 22nd December, 1938.*

New Stroboscope.

From a rapid succession of short glimpses of a moving object the viewer sees a continuous picture, due to persistence of vision; if frequences are identical the object appears stationary, if they differ slightly a slow-motion effect is obtained. The apparatus is therefore adapted to measure rotary, linear, or reciprocating velocities without direct contact. A small metal cylinder with slots for the eyes is rotated through gearing by a constant speed electric motor with hand control. In each a diagonally set blade divides the slot into two expanding apertures so that two glimpses per revolution are obtained; it is however possible to adjust to only one glimpse, and there are three degrees of definition. In this way a range of 150-4,800 r.p.m. is obtained, or alternatively 1,500-12,000. Power is derived from an internal 4-volt battery. In contrast to other instruments not only in semi-darkness of the observed object unnecessary but bright illumination is advantageous.—*"The Oil Engine", Vol. 6, November, 1938, p. 211.*

New Oil Engine Atomiser.

The Ruston Mark 37 atomiser for airless-injection engines, recently placed on the market, is characterised by a simplicity of construction unusual in devices of this nature. It can be taken apart and reconditioned by any intelligent person and the only working parts are the needle valve and small stop pad limiting the lift of the needle. The atomiser nozzle is self-centring and adjusts itself automatically on assembling the atomiser. The fuel to be injected passes through channels formed by a groove in the outside of the atomiser valve and as there are no fine holes in the body of the atomiser, no special probes or pickers are required for cleaning purposes. The fuel channels can be cleaned by stripping down the atomiser and washing the guide and body in petrol or paraffin. The parts are completely interchangeable and the nozzle, needle valve, needle guide, spring or any other component can be changed singly. All the wearing parts are made of hardened steel and all the contact points are metal-to-metal. The lift of the valve is fixed and cannot alter, while all the fuel connections are easily accessible, as the points can be broken and remade with an ordinary spanner, when the atomiser is in use. The reconditioning equipment consists of a testing pump and a reconditioning set, the pump being used to clean the atomiser as required by pumping light clean oil through it and to periodically check the atomisation. The reconditioning set comprises two double-ended laps for reconditioning the valve seating in the nozzle, a lap for correcting the face of the needle valve, a tool for correctng the angle of the needle-valve lap, a pricker for cleaning out the holes in the nozzle, and a supply of a suitable fine abrasive compound. The tools are so designed that the parts referred to can be readily lapped by hand with the retention of perfect alignment. The atomisers are supplied with the fuel inlet parallel to the body, or inclined to it. —*"Engineering", Vol. CXLVII, No. 3810, 20th January, 1939, pp. 64-65.*

Turbine De-scaling.

The operating efficiency of two multistage steam turbines was found to have dropped by more than 50 per cent. from their original output of 350 h.p. each at 3,500 r.p.m. They had been in service for several years, taking steam at 185lb./sq. in. from four boilers in which feed water was very occasionally treated with alum. The turbines became badly clogged with scale and handscrapping was rejected as uneconomic in view of the hight cost and the age of the turbines. Examinations and tests carried out when a third turbine rotor had been scaled and placed on one side, showed that there was about 25 per cent. of silica in the scale. Sections of this rotor were removed and the blades treated with various strengths and temperatures of hydrochloric acid which was inhibited with less than 1 per cent. of certain organic chemicals. These tests indicated that the scale was about one-fourth silica and three-fourths carbonates. The boiling acid did not affect the rotor or the brass blades, but the scale was either loosened or dissolved by lower concentrations of the acid. The two turbines were then acidized, the inhibited acid being introduced with enough steam to rotate the rotor at about half speed, no attempt being made to control the dilution of the acid with steam, but 2 gallons of acid being admitted each hour and a total of 10 gallons being used on the first turbine. Heads of very wet steam were sent through the turbine every 15 minutes in order to try and dislodge any scale that had been loosened but not dissolved and a subsequent examination showed that about one-third of the scale had been entirely removed and about two-thirds had been loosened, but could not be dislodged with streams of high-pressure water. In treating the second turbine, about 50 gallons of inhibited acid were employed over a period of eight hours and examination showed that much of the scale between the small blades had been loosened but not removed, although all the rust and scale from the shell had been dislodged. In neither case was there any appreciable effect on the metals of the turbine. Subsequently both rotors were sand-blasted for about three hours each and this removed the softened scale without any erosion of the blades or rotor. Possibly the sand-blasting would have removed the scale without the acid treatment, but the softening of the scale made the removal easier. Both turbines were then kept practically scale-free for 12 months by thoroughly conditioning the feed water, the total alkalinity of the latter being kept below 200 p.p.m. by adding 2·5lb. of sulphuric acid to every 1,000 gallons of water. A continuous blow-down system was also installed and the condensate checked by conductivity methods for the production of nearly solid-free steam. It was concluded from this more or less experimental work, that inhibited hydrochloric acid could be employed on steam turbines without damage to the metal parts and that if the scale in a turbine is nearly free of silica, satisfactory cleaning can be done by the use of such acid process, but that if there is a carbonaceous scale containing as much as one-quarter silica, then acidizing alone is not enough. The acid softens the scale if it contains carbonates and thus makes it easier to remove by sand-blasting. In the above tests the blasting was carried out with 40-mesh hard sand and a pressure of 125lb./sq. in. The two turbines generated full power again, after de-scaling.—*"Metallurgia" Vol. 19, No. 111, January, 1939, p. 110.*

The Maintenance of Depths Alongside Wharves by Dredging.

In the excavation of bed material alongside a wharf, a dredger must fulfil most, if not all, of the following requirements :—

- (*a*) Rapid working, so as to minimise re-accretion and reduce the time in which berths are out of use.
- (*b*) Minimum mooring space, to avoid interference with traffic and use of adjacent berths.
- (*c*) Easy removal to and from site.
- (*d*) Ability to work close up to vertical face.
- (*e*) Accurate depth and position control, so as to avoid over-dredging and irregular cutting.
- (*f*) Level cutting at the bottom.
- (*g*) Simple expeditious arrangements for removing spoil.

Some of these conditions conflict with one another and compel a certain amount of compromise. Suction types are rarely satisfactory for wharf dredging and unless traffic conditions allow of a pipe-line to an adjacent shore basin or dumping pool, they are usually out of the question. Except in hard bottoms, the usual practice is to employ grab dredgers with attendant barges, the cycle of operations (cut, lift, swing, drop in barge and swing back) occupying usually less than one minute, so that the full dredging rate is usually about 70 times the mean effective bucket load per hour, although rates of up to 120 cycles per hour are possible with favourable materials and highly efficient operators. The mean dredging is, however, reduced from this by the time required to change barges, waiting for empty barges, breakdowns, interference by ship traffic and shifting the dredger moorings, so that it is actually rarely possible to exceed 50 bucket loads per hour. In general a 1½ cu. yd. bucket capacity is convenient, allowing a mean output (including normal delays) of, perhaps, 75 cu. yds. per hour, which is fairly satisfactory. The hull of such a vessel may be 80ft. long and in, say, 30ft. of water, the six moorings will extend some 200ft. above, below and sidewards from it, when in the middle position. If the barges have a capacity of 300 cu. yds., they will take three hours to fill (at the 70 cycles rate), and unless the distance to the disposal point is great or the time of emptying prolonged, two such barges and one tow-boat will suffice. Owing to the great variations in inertial and induced stresses on the various parts, dredging plant is very subject to breakages (notwithstanding all assurances to the contrary by builders), consequently, ample supplies of spare parts and the best possible facilities for repair must be available. Welding is now used extensively for dredger repairs. The very best lubricants and ropes should be used.

—H. Chatley, D.Sc., "The Dock and Harbour Authority", Vol. XIX, No. 219, January, 1939, pp. 79-82.

German Exports of Internal Combustion Engines.

The total value of exports of internal combustion engines from Germany (including spare parts, but excluding motor vehicle or aircraft engines) for the period July to September, 1938, was 17·5 million Rm., the total exports of all internal combustion engines in the first nine months of 1938 being 8 per cent. higher than for the same period of the preceding year. Diesel engines made up 60 per cent. of the value of exports of internal combustion engines for the third quarte rof 1938, the most important being stationary Diesel engines, followed by marine Diesels, motor-vehicle engines constituting the smallest group. Exports of marine Diesel engines were lower than in the second quarter of 1938, but higher than in the first quarter, and it is noteworthy that exports of marine engines have risen year by year since 1935. The value of these exports from January to September, 1938 is greater than that for the whole of 1937 and marine Diesel engines had about an equal share in this movement. *—"Gas and Oil Power", Vol. XXXIV, No. 400, January, 1939, p. 28.*

New Belgian Motor Liners for South Atlantic Trade.

Five new motor liners for the South American services of the S.A. Compagnie Maritime Belge, of Antwerp, were ordered from the S. A. John Cockerill at Hoboken in 1937, and three of them —the "Copacabana", "Piriapolis" and "Mar del Plata"—have now entered service. The first-named vessel is typical of the others, being a single-screw cargo and passenger ship of 7,246 tons gross, with a length b.p. of 435ft., a moulded breadth of 61ft. 4in., and moulded depth (to shelter deck) of 39ft. 2½in. The general-cargo capacity (bale) is 455,550 cu. ft., and insulated-cargo capacity (net) 56,800 cu. ft. There is accommodation for 20 first-class passengers in a deck-house on the bridge deck, and for 116 tourist-class passengers on the shelter deck. The ship has two complete steel decks and a partial lower deck and there are eight transverse watertight bulkheads carried to the main deck, giving three cargo holds forward of the engine-room and three aft. The cellular double bottom extends all fore and aft between the peak bulkheads and is arranged for the carriage of oil fuel or water ballast, fresh water and lubricating oil. The main propelling unit is a five-cylinder B. & W. double-acting two-stroke engine, with cylinder diameters of 620 mm. and a stroke of 1,400 mm., developing 6,100 i.h.p. at 102 r.p.m. The pistons are oil-cooled, while the cylinders are fresh-water cooled. Electrical current is supplied at 220 volts by four 165 kW. Diesel-driven generators, the Diesels being direct-coupled and of the B. & W. single-acting, two-stroke type, with four cylinders of 220 mm. bore and a stroke of 370 mm., running at 400 r.p.m. There is also a 24 kW. emergency dynamo on D deck, driven by a single-acting four-stroke Walschaerts Diesel engine. An air compressor is coupled to each of the Diesel engines of the two forward main generators. The various fuel, lubricating-oil, fresh-water and salt-water pumps are electrically driven, as are also the deck auxiliaries comprising 14 cargo winches, heavy-duty anchor windlass and after warping winch. The steering gear is electrically-operated. There are two 5-ton derricks at each of the six hatchways, together with an additional 30-ton heavy derrick at the fore mast and a 15-ton derrick at the main mast. The refrigerating plant consists of three NH_3 compressors, each with a capacity of 70,000 calories per hour when working between $-5°$ C. and $+38°$ C. There are three multi-tubular condensers and three evaporators, with electrically-driven brine and circulating pumps, the brine distribution being on the three temperature system. Electric fans are used for circulating air in the refrigerated spaces which are arranged in two groups, one forward and one abaft the engine-room, each group consisting of three superimposed rooms. Steam for hotel purposes is supplied by a Clarkson thimble-tube boiler equipped with a Laidlaw-Drew oil-burner on the medium-pressure system, at the forward end of the main propelling engine. The loaded service speed of the "Copacabana" and her sister ships is 14 knots. *—"The Shipbuilder", Vol. XLIV, No. 351, January, 1939, pp. 25-29.*

Quadruple-screw Motor Tug for Service on Danube.

The peculiar conditions in regard to navigation on the Danube—such as the strong current and lack of deep water—have, in the past, necessitated the use of paddle tugs for heavy towing work in preference to screw tugs. Modern deevlopments in shallow-draught oil-engined vessels have, however, made it possible to evolve a satisfactory design of quadruple-screw motor-tug, for which purpose a series of exhaustive experiments were carried out at the Hamburg Experimental Station. The results have been embodied in the motor tug "Bern" built at the Hitzler Yard in Regensburg, Bavaria. She is a vessel with a length of 165ft. overall, a moulded beam of 24ft., and moulded depth of 8ft. 10in. The maximum draught, with 80 tons of oil fuel in the tanks, is 4ft. 7in., and the underwater section of the hull is tapered throughout. The internal arrangements and fittings of the tug follow the usual practice for heavy-duty tugs on the Danube and the accommodation for the crew of 18 is in accordance with present-day requirements. The windlass and towing winch are electrically driven and the tug is provided with a triple rudder of "Hitzler" pattern. Electric welding has been extensively used in the construction of the hull. The propelling machinery, supplied by the Güldner Works of Aschaffenburg, consists of two reversible four-cycle 6-cylinder oil engines, each developing 630 h.p. at 325 r.p.m., and equipped with chain-driven superchargers. The whole of the compressed air required for starting and manœuvring the main engines is supplied by two independent compressors driven by Diesels of 14 h.p. and 28 h.p. respectively. These Diesels are also direct-coupled to electric generators, a third generator being driven off one of the main propeller shafts. The latter supplies the whole of the current required for lighting purposes, whereas the Diesel-driven generators are used to supply current for both lighting and power circuits, either directly or through a 110 amp.-hr. battery which cuts in and out automatically. Under normal towing conditions none of the independently-driven auxiliary machinery will be in use. The auxiliaries include two plunger pumps, as well as stand-by circulating water pumps, an additional centrifugal pump for the same service driven by the smaller of the two auxiliary Diesels, an electrically-driven lubricating-oil pump, an electrically driven oil-fuel transfer pump and an oil-fired boiler for supplying steam for cooking and heating purposes. The triple rudder is actuated by hand. There is an oil purifier of special (non-centrifugal) design. Each of the main engines drives two propellers through Vulcan hydraulic gearing, for which an efficiency of 97 per cent. is claimed. The performance of the "Bern" since her entry into service has been highly satisfactory and the tug has proved capable of dealing with tows normally handled by steam tugs of far greater power.—*"Werft, Reederei, Hafen", Vol. 20, No. 1, 1st January, 1939, pp. 1-5.*

Petrol-carrying Coastal Motor Ship.

The new single-screw petrol-carrier "Shell Spirit II" built by L. Smit & Son, of Kinderdijk, Holland, has now been delivered to the Union Lighterage Co., Ltd., of London. She is a vessel with a length of 157ft. b.p., a moulded breadth of 32ft., and moulded depth of 10ft. 3in., driven by a 4-stroke, eight-cylinder Deutz-Diesel engine of 400 b.h.p. The auxiliary machinery includes a 25 b.h.p. 4-stroke Diesel driving a 16-kW. dynamo and auxiliary air compressor, and two 8 b.h.p. 4-stroke Diesels for the auxiliary fresh-water and salt-water circulating pumps, and a 3-kW. auxiliary generating set and general service pump. The vessel attained a mean speed of 9·5 knots on her official trials.—*"Fairplay", Vol. CL., No. 2906, 19th January, 1939, p. 190.*

Pipe Threads.

As it is now about 30 years since the publication of the current edition of the British Standard No. 21 relating to British Standard Pipe Threads, and there have, during this time, been considerable developments in the technique of screw thread production and gauging, the British Standards Institution has undertaken a revision of this standard and proposes to make certain recommendations in the system of gauging. Some of the decisions concerning fundamental definitions and basic sizes of threads and tolerances to be allowed have already been taken, but no changes will be recommended in the pitches or gauge diameters of the threads, although it is recognised that in certain cases, notably in the ½in., 1in. and 1¼in. sizes, there are reasons why a change might have appeared desirable. In a few cases, however, the gauge lengths have been slightly altered in order to obtain more satisfactory lengths of engagement. It has been recognised that both taper-to-taper joints and parallel-to-taper joints are in common use, and provision has accordingly been made for both parallel and taper threads. The present revised edition of this standard is limited to definitions of the form of thread and the terms used, and to the basic sizes and tolerances, the latter being given alternatively in turns of thread and actual linear dimensions. The supplementary publication dealing with the subject of gauging is now in course of preparation and should shortly be available for issue.—*"The Contract Journal", Vol. CXX, No. 3110, 18th January, 1939, p. 235.*

Marine Diesel Engines at the 1939 Leipzig Fair.

A number of German shipbuilding and engineering firms will be exhibiting marine oil engines at the Leipzig Fair, open from the 5th to the 13th March, 1939. The sizes of the various exhibits range from 20 to 1,500 h.p. and include engines suitable for the propulsion of small craft of all kinds and vessels of moderate size, as well as Diesels adapted for various auxiliary purposes in ships. Among the larger marine oil engines shown will be three manufactured and exhibited by Fried. Krupp A.G., one of these being a direct reversible, 4-cycle Diesel engine of six cylinders, developing 1,050 h.p. at 350 r.p.m., having pressure-fed fuel injection and a camshaft driven by skew-cut gears from the crankshaft, operating the valves in the cylinder heads and the fuel pumps by means of pushrods and rockers. The bearings of the camshaft and fuel pump are of special design, which is the subject of a German patent, providing for a uniform amount of play between the cams and the rollers at all loads, thereby ensuring accurate timing of the valves and fuel pump injection. A second marine engine will be shown coupled to a hydraulic brake. It is a four-cylinder Diesel with a cylinder bore of 250 mm. and stroke of 350 mm., having an output of 185 h.p. at 450 r.p.m. with airless fuel injection. The fuel for each cylinder is pressure-fed by a separate fuel pump to the fuel needle valves and sprayed through a multiple jet in a finely-divided state direct into the combustion chamber. The third marine engine shown on the same stand will be a six-cylinder 4-

cycle engine of 85 h.p. at 850 r.p.m. with airless injection, which does not require auxiliary ignition for starting. A cylinder head with Archaouloff injection and power valve, will also be on view. Under Prof. Archaouloff's system, the compression and combustion pressures in the working cylinder are used to drive the fuel pumps, and a fuel injection pump constructed in accordance with this principle, after exhaustive experiments and tests, has given excellent results in practice. The Archaouloff method of fuel injection in 2-cycle engines enables the camshaft to be dispensed with entirely, thereby simplifying the engine to a considerable extent.—*"Shipbuilding & Shipping Record", Vol. LIII., No. 3, 19th January, 1939, pp. 82-84.*

Isherwood Construction in 1938.

In the course of the twelve months ending on the 31st December, 1938, a total of 39 vessels of varying types of Isherwood construction and aggregating a dead-weight carrying capacity about 450,000 tons, were ordered to be built. Of these 49 vessels, 8 are on the pure Isherwood system of longitudinal framing, 12 are on the "Bracketless System" (an improvements on the old Isherwood system applicable to tanker construction), and 19 are with Isherwood combination framing and "Arcform" design of hull. There were 31 vessels of Isherwood construction delivered in 1938, 14 of them being to the "Arcform" design—some incorporating the Isherwood combination system of framing and the "Bracketless System"—and the remaining 17 on the pure Isherwood system.—*"The Shipping World", Vol. C., No. 2379, 18th January, 1939, p. 159.*

Doubled Power in the Same Engine Room.

The Swedish East Asiatic Co.'s motorship "Formosa" was delivered by Burmeister & Wain in 1921, being then equipped with two 4-stroke single-acting engines with a total power of some 2,300 b.h.p., which gave her an original trial speed (in ballast) of 11·85 knots. The ship has a cargo capacity of about 10,000 tons. In order to increase her speed, she was recently re-engined by the Kockums Mek. Verk. with four 7-cylinder Polar Diesel engines, coupled in pairs to A.S.E.A. electro-magnetic couplings and reduction gear. Each engine develops 1,280 b.h.p. and has a cylinder diameter of 340 mm. and stroke of 570 mm., the speed of the engines being 300 r.p.m. and that of the propeller shafts 130 r.p.m. The power consumption of the couplings is stated to be 8·1 kW., and the shaft slip at full load 4·3 r.p.m. Although the power has been more than doubled, the new machinery has been installed in the same engine-room and on the measured mile, with only three engines in use, the "Formosa" attained a speed of 14½ knots.—*"The Motor Ship", Vol. XIX, No. 228, January, 1939, p. 371.*

A Simple Method for Observing the Combustion Timing in Compression-ignition Engines.

This method of finding the commencement of combustion in an engine consists of observing the flame directly through a stroboscopic device, whereby a 5 mm. diameter quartz window is fitted in the indicator passage of the engine and the light passing through this window is thrown onto the rim of the flywheel by a small mirror. Another mirror is fixed on the flywheel and passes a fixed scale marked in degrees of crank angle in the neighbourhood of the dead centre. The flywheel mirror is a piece of highly polished stainless steel of semi-circular cross-section, so that a light from any particular source appears in it as a line which can be observed from a wide range of angles. The light from the combustion can be observed through this mirror, and by virtue of its being cylindrical, the impression is obtained of a source of light travelling bodily round with the flywheel. The beginning of the band of light, which is observed, indicates the beginning of the combustion on the fixed scale. A very open scale is available on a large flywheel, which greatly assists accurate measurement. In actual practice, a black paper funnel is placed between the window and the flywheel in order to shield off disturbing extraneous light. As an ordinary window in the combustion chamber wall fouls very quickly and is, therefore, of little use for prolonged observation, a special fixture to overcome this defect is used, in which the air trapped in the bore behind the window forms a protective cushion, which is only very rarely penetrated by an oil drop, and which keeps the window clean for several hours running. This simple method for observing the combustion timing in an accurate manner while adjustments are made on the engine, has proved very convenient in use.—*S. G. Bauer, Dipl.-Ing., Ph.D., "Engineering", Vol. CXLVIII, No. 3810, 20th January, 1939, p. 79.*

A Surface Roughness Meter.

An instrument for measuring surface roughness of materials, known as the "Profilometer" and made in the U.S.A., is now being marketed in this country. The instrument has been designed to measure a running average of the height of irregularities on a surface, irrespective of its contour and the portable type, weighing about 50lb., fits in a case 10in. deep by 22in. long with a panel carrying the various meters, switches and terminals. The main instrument assembly and batteries are behind the panel, the small non-spillable storage battery being capable of operating for about 8 hours and the dry batteries having sufficient capacity for several months. A small tracer head is carried in a separate section at the end of the case and in the bottom of the tracer head is a small diamond point, which is supported by pilot balls, and is rigidly coupled to a small coil situated in a magnetic field. The coil and tracer point are held in position by a small spring, and as the head is moved over a surface, the point moves vertically up and down according to the roughness, its motion causing a minute electrical voltage to be produced in the coil, and these electrical impulses are amplified and measured by the instrument. The electric meter which automatically determines the average height of the irregularities is calibrated directly in inches and gives a root mean square average, which is the average commonly used for oscillating quantities. For a simple curve the root mean square value is 35 per cent. of the peak to valley height. The diamond point is sharp enough to bottom the irregularities, but is not loaded heavily enough to scratch or damage the specimen. In use the tracer is moved across the surface to be tested and the meter reads a continuous running average of the height of surface irregularities. The scale is calibrated in micro-inches or millionths of an inch. It is unnecessary to press the tracing head hard against the surface, as its own weight is sufficient, the readings being independent of the pressure on the specimen. A speed of trace of about 1in. per second is suitable, but the readings obtained are independent of the speed of trace over wide limits. To adjust the tracer for curved surfaces, the point may be raised or lowered, to ensure proper contact, by means of a small knob. The sensitivity of the instrument can be set for full scale values of 3, 10, 30, 100, 300, and 1,000 micro-inches, a range which has been found sufficient for moist surfaces. The standard tracer head can be used on external surfaces larger than ⅛in. in diameter, on flat surfaces, and internal surfaces which will admit the tracer, or 2⅝in. diameter. On smaller diameter holes it is necessary to cut the specimen to admit the tracer, and then diameters down to ½in. can be examined.—*"The Engineer", Vol. CLXVII, No. 4332, 20th January, 1939, p. 100.*

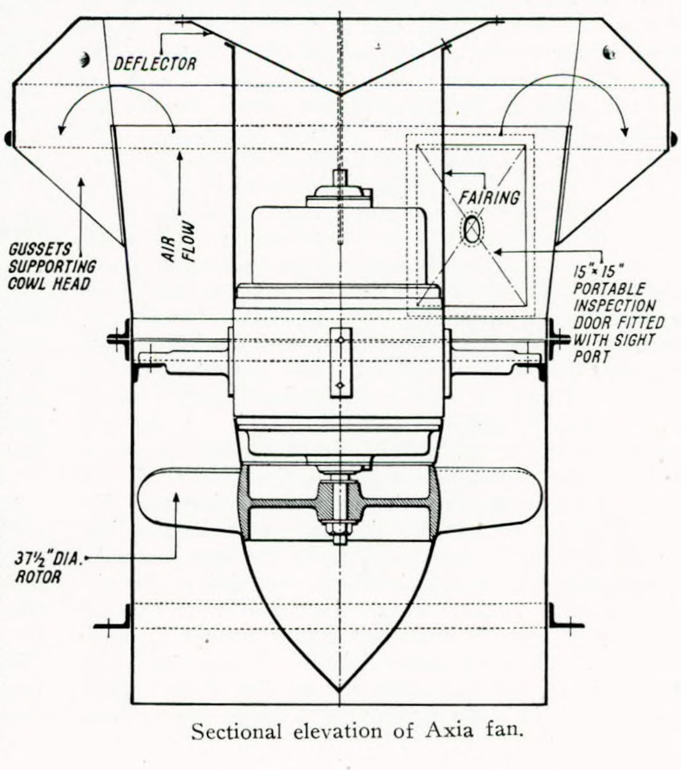
Performance of Separately-fired Superheaters.

The Sun Shipbuilding Company recently carried out an unusual compromise for the boiler plant of a U.S. tanker of 17,320 tons displacement. The ship has a set of geared turbines of 3,500 h.p., giving her a sea speed of 13 knots and supplied with steam by two oil-fired Foster Wheeler "A" type boilers with three drums, the generator tubes being on one side of the furnace space and the superheater tubes on the other. These boilers have a maximum capacity of 50,000lb./hr., including 35,000lb. of superheated steam, and the generation of the steam and its superheating are carried out in two separate and distinct furnaces. The steam-generating furnace discharges its gases through the boiler heating surface and economiser to the funnel, while the gases from the superheater furnace flow into the main furnace. The superheater furnace is built with radiant heat superheating tubes in all the walls except the end ones, which are refractory and this auxiliary furnace is wholly enclosed within the main boiler setting. The superheater tubes are not nested and there are only two staggered rows through which the gases pass, so all these tubes are easily accessible for cleaning. On the ship's maiden voyage the equipment proved highly satisfactory in every way, the steam temperature being maintained between 720° F. and 725° F., even during a six-hour run on one boiler.—*"The Journal of Commerce and Shipping", No. 34,629, 26th January, 1939.*

Improved Engine Room Ventilation.

Among the improvements made in the equipment of the steamship "Rimutaka" (ex "Mongolia") which recently left for New Zealand after an extensive refit, was a comprehensive new system of mechanical ventilation for the engine-room. This was achieved by installing four 37½in. diameter Axia axial flow type reversible fans, with a total capacity of 45,000 cu. ft. per minute, delivering air through a system of ducts to about 90 supply points in the engine-room. At the starter platform a battery of Axia directional and shut-off louvres were installed to enable the individual supply to these points to be controlled. A similar exhaust system with three 27½in. and one 37½in. fans was also fitted to supplement the natural exhaust through the upper casing and skylight, the total capacity of the exhaust fan system being 36,000 cu. ft. per minute.

Reversible control enables all eight fans to be used either to supply or exhaust, as necessary, and this coupled with the fact that each fan has a speed



Sectional elevation of Axia fan.

reduction of 25 per cent., gives an unlimited flexibility to the system.

The fans themselves occupy far less space than the centrifugal type, as may be judged from the accompanying illustration. The exhaust fans were supplied complete with a mushroom-type cowl, all fans being integral parts of their respective ducts. *—"Shipbuilding and Shipping Record", Vol. LII, No. 25, December, 1938, pp. 776-777.*

U.S. Tender for Transatlantic Seaplanes.

A tender of special design for servicing the Pan American Company's transatlantic seaplanes at a refuelling base to be established in the Atlantic, was recently delivered to the owners by the Jakobson and Peterson Yard, of Brooklyn. The little boat, named "Panair V-A" is 52ft. long with a beam of 13ft. 7in., and when fully loaded, has a maximum draught of 6ft. 9in. Her duties will include the supply of petrol, lubricating oil and water to the seaplanes, and the pumping of conditioned and cooled air through them while they are being serviced, the planes being attached to the tender during these operations, for which purpose an overhanging ramp is provided at the stern. During the servicing operations the bow of the plane is pulled up on this ramp, which is hinged to the boat's hull and can be raised or lowered by two hydraulic rams fitted one on each side of the upper deck aft, and operated by a single lever control from the side of the deckhouse. Two detachable steel drums on the under side of the ramp serve as floats to support it. Three covered wells sunk into the aft deck contain the petrol filling hose, a meter for measuring the petrol discharged, and a lubricating-oil filling hose, respectively. Forward of these wells is the low deckhouse over the engine-room, with an electric capstan for hauling the planes up on the ramp, on each aft corner of this deckhouse. The pilot house over the forward end of the engine-room casing contains the navigational and W/T equipment and forward of the engine-room are four tanks built into the hull, occupying the full width of the boat and each having an expansion trunk above it. These tanks contain the petrol and have a capacity of 8,300 Imperial gallons. By the expansion trunks are the suction and discharge connections and a set of petrol filters for trapping the water. There is also a 166-gallon lubricating-oil tank and an 83-gallon fresh-water tank. A recessed panel, closed by watertight doors, on the starboard side of the deckhouse, contains the petrol, oil and water manifolds, with the various suction and discharge pipes to control the flow and distribution. In the forecastle, separated from the tank space by a cofferdam, is a compartment containing the air-conditioning plant, which includes compressors, blowers, coolers, circulating water pumps, humidifiers, etc., all electrically driven. From this compartment an air duct runs aft to the engine-room, terminating in a connection for a large canvas hose through which air is forced into the cabins of the planes. Over the forecastle is a small electric anchor windlass. No living accommodation for the crew is provided on board, as they will be quartered at the shore base. The tender is propelled by a 150 h.p. Cummins Diesel engine driving a 3-bladed bronze propeller through a Twin Disc reduction and reverse gear having a 3-to-1 ratio, giving the vessel a speed of about 8½ knots. The jackets are fresh-water cooled and at the forward end of the engine there is a shaft-driven fire pump of 207 gall./min. capacity. Electric current is supplied by a Cummins-Westinghouse Diesel-generator set of 20 kW. capacity and a 32-volt battery provides stand-by current. Special precautions to prevent sparks from being thrown out by the engine exhausts are provided, each engine exhausting into a Maxim spark-arrestor silencer of the S.C. type, both of these being connected to a third Maxim silencer of the W type containing a 2in. pack of steel wool to intercept any sparks that might get by. At the after end of the engine-room is a motor-driven pump and accumulator for operating the hydraulic rams for the stern ramp. There is also a small pump for pumping fresh water to the planes. Protection against petrol fires includes a very complete CO_2 system with discharge nozzles in every compartment. The control valves for the various outlets are mounted on the port side of the deckhouse where they are readily accessible. The battery for supplying power to the W/T equipment is located in the watertight compartment which houses the air-conditioning plant. Night work will be facilitated by a large searchlight of the standard airport beacon type, and a Portable Light One-Mile-Ray searchlight, both mounted on top of the pilot house. As soon as the vessel's trials and tests are completed, she will be placed on board a ship for transport to her future destination.*—"Motorship and Diesel Boating", Vol. XXIV, No. 1, January, 1939, pp. 28-29.*

First Foreign-built Vessel for N.Z. Shipping Company.

The motorship "Kaituna", built and engined by the Eriksberg yard, Gothenburg for a Swedish firm, was taken over by the New Zealand Shipping Co., Ltd., immediately after her trial trip. She is a passenger and cargo vessel of the shelter-deck type and a carrying capacity of 9,165 tons. The ship has a length of 432ft. 3in. overall, a moulded beam of 56ft. 6in., and depth to shelter-deck of 38ft. 9in. There are four main cargo holds and deep tanks with a capacity of about 1,000 tons of oil. The holds are served by five hatches with twelve 5-ton derricks on two pole masts and a pair of derrick posts. The propelling machinery consists of a single double-acting 2-stroke airless-injection B. & W. Diesel engine rated at about 5,200 i.h.p., and there are two auxiliary Diesels of 172 and 115 b.h.p. direct-coupled to 114-kW. and 76-kW. generators respectively.—*"Lloyd's List & Shipping Gazette", No. 38, 756, 18th January, 1939, p. 19.*

The New Zeppelin.

The giant airship "Graf Zeppelin" has already made several trial flights in preparation for her entry into the Transatlantic service. The airship—like her predecessor the "Hindenburg"—is equipped with four Mercedes-Benz 1,200 h.p. propulsion engines, each contained in a power car suspended from the main structure, with the water cooling radiator for each 16-cylinder V-type engine immediately in front of the 4-bladed propeller at the back of the torpedo-shaped car. The maximum speed of the airship is 84 miles per hour, and in addition to the four propulsion engines there is a Diesel installation of Junkers manufacture for the various auxiliary services. The heat of the exhaust gases is utilised for cooking, so that a considerable amount of fuel is saved, and the water vapour in the exhaust gases of the main engines is condensed and stored, thus obviating the necessity of discharging hydrogen from the airship's gas chambers to maintain constant height a sfuel is used up.—*"The Oil Engine", Vol. VI, No. 69, January, 1939, p. 292.*

Intercrystalline Cracking in Boiler Plates.

The paper deals with the type of cracking in steam boilers, characterised by the spreading of cracks in riveted seams in and around the rivets, which was shown by Parr and Straub to be associated with the use of alkaline water, either natural or rendered so by additions. Unlike cracks caused by fatigue or by corrosion, these cracks are intercrystalline and only occur where it is possible for the alkali to reach a high concentration in capillary spaces. They are not found in seamless drums. The paper reviews the literature of the subject and describes the experiments carried out at the National Physical Laboratory, where the whole of the possible factors were separately investigated under various conditons of prolonged and alternating stress, due account being taken of the condition of steel, intermittent loading and the nature of the solution. The assumption that caustic soda alone is without effect, and that the attack can only occur when sodium silicate is present, has not been confirmed. Experimental methods are described, and the possible mechanism of the attack is discussed. Two parallel cases of attack by solutions of nitrates, in both of which intercrystalline cracking occurs in the absence of external stress, are also discussed. The paper concludes by stating that: (1) Intercrystalline cracking in steam boilers is always associated with a high alkalinity of the water and where this is unavoidable, the best method of protection is to maintain a ratio of sulphate to alkali above a certain value, depending on the working pressure; (2) The steel must be in a condition of stress, either through an external constraint, as in a riveted joint which has been forcibly bent, or by internal stress as in a rivet or in a plate which has been subjected to a high riveting pressure; (3) There must be opportunities for the concentration of the solution in capillary spaces, as in the case of riveted joints, whereas the absence of such cavities in seamless drums renders these immune from caustic cracking; (4) A high temperature must be reached, depending on the composition of the solution, although cracking may sometimes occur at or near 100° C. It should be borne in mind that temperatures much in excess of the working temperature of the boiler may be reached in a riveted butt-strap joint exposed to the flame; (5) The cracks are not to be confounded with those produced by corrosion fatigue, which generally pass through the crystals and not around them and may be distinguished as a rule by simple inspection, although a microscopical examination enables the two kinds of defects to be distinguished in a decisive manner; (6) Both hydrogen and the deposit of iron oxide which is formed by the reaction play a part in the process, but the mechanism remains to be elucidated. —*Paper read by C. H. Desch, D.Sc., at a meeting of the N.E. Coast Institution of Engineers and Shipbuilders, on the 20th January, 1939.*

Shortage of Marine Engineers.

A serious shortage of marine engineers is being felt in South Wales and it is thought that if the good prospects for youths qualifying as such were more generally realised by parents, there would be more applications for apprenticeship. It is also considered that if boys received a preliminary training before entering the profession, there would be greater opportunities for their acquisition of the necessary certificates of qualification. At a recent meeting of the Cardiff and Bristol Channel Shipowners' Association a scheme to provide pre-apprenticeship training for boys destined for the marine engineering profession was discussed, such training to be given through technical colleges, or, alternatively, by other training establishments. The meeting decided to request the Shipping Federation to consider the possibilities of the scheme with a view to its national application and immediate adoption.—*"The Journal of Commerce and Shipping", No. 34, 619, 14th January, 1939, p. 7.*

Danger of Annealing Connecting-rod Bolts.

An interesting case of a Diesel engine breakdown is recorded in a recent issue of *Vulcan.* The unit in question was of the blast injection type, developing 330 h.p. at 220 r.p.m., and the breakdown was due to the fracture of a crankhead bolt of a connecting-rod. The crankshaft, bedplate and one cylinder and column casting were broken and the engine was so badly damaged that it had to be scrapped. A short time before the accident all the connecting-rod bolts had been annealed, and on examining the fractured bolt it was found that the appearance of the fractured surface was unusual, four-fifths of the whole area being covered with a blue-grey oxide film, which the remaining fifth presented a clean metallic surface. It seemed that iron oxide had formed during the annealing process, and its presence proved that there was a crack in the bolt when it was put back. Although mild-steel bolts may be annealed to relieve them of strain under ordinary working conditions, annealing should not be resorted to in the case of connecting-rod bolts made of heat-treated steel, such as are usually found in modern engines, even if mild steel bolts are still met with in the older types of engine.—*"The Oil Engine", Vol. VI, No. 69, January, 1939, p. 261.*

Some Recent Diesel Installations and their Characteristics.

The paper is divided into three parts, the first giving brief outlines of a number of representative installations built during the last two years, the second describing the propelling and auxiliary engines, and the third dealing with a number of points of general interest. The typical installations described are double-acting 2-cycle engines in a single-screw cargo vessel, a large twin-screw refrigerated cargo ship, and a high-powered passenger liner. Single-acting 2-cycle installations in various cargo vessels and small craft are likewise referred to, while brief descriptions of the machinery of a Diesel-electric salvage tender and of cross-Channel passenger ships, are also included. Reference is made to the supercharged 4-cycle installations used in many tankers and cargo vessels. The portion of the paper devoted to main and auxiliary engines includes detailed descriptions of typical double-acting 2-cycle engines, single-acting 2-cycle engines and supercharged single-acting 4-cycle engines, as well as of auxiliary engines of similar type. The last part of the paper deals with the rating of main and auxiliary engines, manœuvrability, records of engine behaviour, crankshaft alignment, crankshaft stresses and balancing, crankcase ventilation, cooling water systems, lubricating oil systems, cylinder covers and pistons, piston rings, main bolt tightening, tuned exhaust systems, and silencing. The paper concludes with comparisons of the space for Diesel machinery required in typical installations completed during the past 15 years, with special reference to the progress made during that time in regard to the saving of weight and space and increase of engine power.—*C. C. Pounder, "Transactions of the Institute of Marine Engineers", February, 1939.*

Orders for Twelve 10,000-ton Tramps in Italy.

Orders for twelve 10,000-ton motorships for Italian owners have just been placed in various Italian shipyards, for delivery in 1940. They will have an overall length of 486ft., a moulded breadth of 61ft., and depth, to main deck, of 29ft., and will all be of the shelter deck type with one deck and shelter-deck. It is proposed to carry heavy cargoes —such as coal—in the lower holds and to load light cargo in the shelter decks, in order to utilise the ship's total carrying capacity without the expense of cargo trimming. Ten of the vessels will be equipped with Fiat Diesels of about 4,000 h.p. to give a service speed of 12 knots fully loaded, the remaining two motorships having Tosi Diesels of 4,500 h.p. each. The Fiat Diesels will be practically identical with those of the motor tankers "Fede" and "Lavora" built by the Cantieri Riuniti d'ell Adriatico, Trieste, except for certain minor alterations in the drive of the circulating pumps and the substitution of fresh-water cooling for sea-water cooling. In several cases the auxiliaries will be steam driven. The initial cost of the machinery will be kept to a minimum by standardisation. Six of the ten Fiat engines will be constructed by the Fiat Works at Turin, two by the Ansaldo Company in Genoa and two by the Cantieri Riuniti in Trieste. The Tosi engines will be built by the Franco Tosi Company of Taranto, who are also designing the hulls of all twelve vessels.—*"Shipbuilding & Shipping Record", Vol. LIII., No. 4, 26th January, 1939, p. 124.*

A Ground Finish Accurate to ±0·00008in.

Among the measuring appliances to be exhibited at the Leipzig Fair in March, will be a form-grinder of such accuracy that it can work to a limit of ±0·00008in., and can produce limit gauges ("go" and "no go" gauges), stamping dies, open or split matrices, relieved tools of special form, round patterns with radial and axial profiles and special cams for automatics. The work is machined by a 5in. wheel running at a peripheral speed of 50ft./sec., and the machine is controlled from a platform from which each working operation can be seen projected on a screen and enlarged 50 times, every movement of the work and the grinding wheel being able to be accurately followed on this screen and correctly adjusted as well by two control wheels, so that, although the machine's accuracy is guaranteed to ±0·0002in., this can easily be brought down to ±0·00008in. A vibrationless dust extractor is fitted inside the machine and the electrical switchgear and controlling instruments for operating all the motors are worked from the control platform, together with the lighting device. Photographic equipment is also provided, so that once any type of gauge has been made, it can be recorded in photographic form.—*"Mechanical World", Vol. CV, No. 2716, 20th January, 1939, p. 51.*

EXTRACTS.

The Council are indebted to the respective Journals for permission to reprint the following extracts and for the loan of the various blocks.

Breakdown of a Diesel Engine Piston.

By L. J. Holman, M.I.Mar.E.

"Shipbuilding and Shipping Record", 1st September, 1938.

The importance of adequately securing internal cooling water pipes of Diesel engine pistons is shown by a breakdown which recently occurred in the piston of a two-stroke, single-acting Diesel engine, the cooling water circulation being interrupted to such an extent that fuel had to be cut off from the cylinder.

As will be seen from Fig. 1 below, the piston is constructed in three parts, the material being cast iron in each case. The body A has an internal flange B, to which is secured the palm of the piston rod, and also the casting C, by means of the studs D. The latter are elongated and fitted with distance pieces, in order to reduce the strain per unit of length. Casting C is further secured to the palm of the piston rod by means of three $\frac{5}{8}$-in. collar studs and nuts, not shown in the sketch. The internal pipe E is screwed into the casting C, and locked by means of a plain pin F, the joint being made by a composition ring, as indicated.

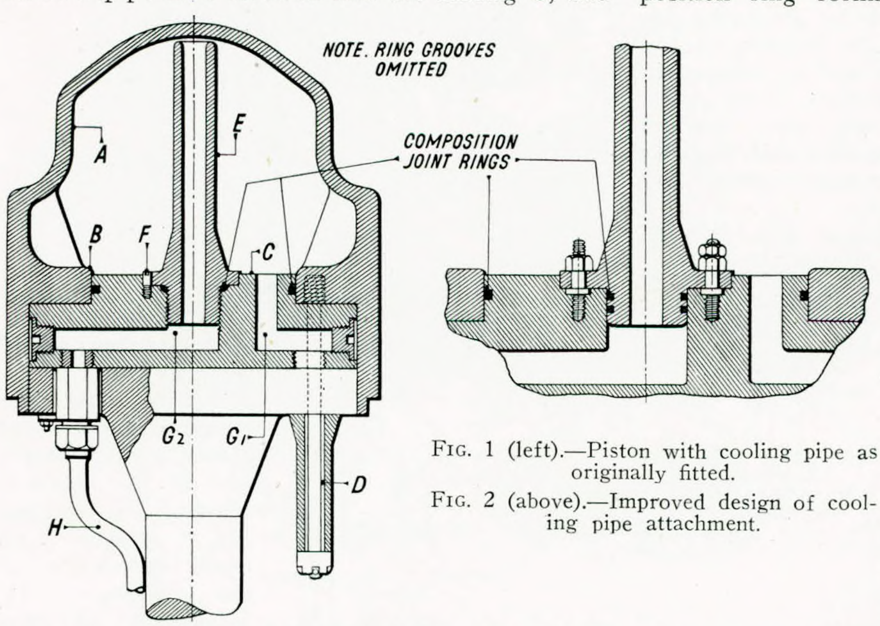


Fig. 1 (left).—Piston with cooling pipe as originally fitted.

Fig. 2 (above).—Improved design of cooling pipe attachment.

Salt-water cooling is employed, the water entering and leaving the piston by way of passages G_1 and G_2 in the casting C. H is the outlet pipe, and there is, of course, a similar inlet pipe diametrically opposite, but which is omitted in the sketch, in order to show the stud D.

On dismantling the piston, it was found that both screw threads securing the internal pipe E in the casting C had almost entirely disappeared. Thus, the type of locking pin F was rendered useless, the effect being that on the down, or power stroke of this particular piston, the inertia of the internal pipe caused it to lag behind, and strike the crown of the piston. In this manner it acted as a valve, and interrupted the circulation of the cooling water.

From examination of the damaged screw threads, it was concluded that water had found its way into them and set up corrosion. This condition may have been aggravated by initial slackness between the threads themselves. As the breakdown occurred when only a day or two from a port, where a stay of some three weeks' duration was made, the corrosion theory appeared a feasible conclusion.

The exact manner in which the water entered between the threads was a little difficult to determine. It may, for example, have worked its way upwards through the threads, owing to a slight initial slackness therein. Or, alternatively, the composition ring forming the joint may have been damaged during assembly, allowing leakage to take place from the top.

This latter observation points to the weakness in the design of the internal arrangements of the piston. The fitting of a rubber or composition ring tends to be unsatisfactory in a screwed joint of any type. In this case, it will be evident that the operation of screwing up the internal pipe is very liable to damage the joint ring.

An improvement in the design is indicated in the enlarged view, Fig. 2. It will be noted that the screw thread is dispensed with, and the internal pipe secured by means of four $\frac{5}{8}$-in. collar studs and nuts. The joint is made by the two composition rings indicated, and the extra length of stud above the nut is required for drawing the internal pipe into place. The fitting of locking washers and/or split pins to prevent the nuts slacking back would be adequate protection. However, in view of the extra length of studs referred to, it would perhaps be advisable to fit lock nuts, as shown on the right-hand stud. These would protect the exposed threads from the action of salt water, and facilitate the operation of dismantling the internal pipe.

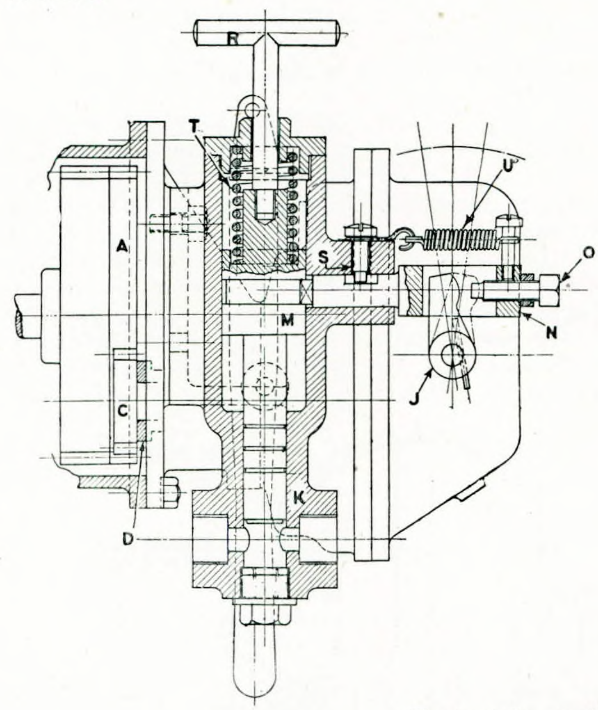
Safety in Operation.

Over-speed Governors, Water and Oil-pressure Safeguards. "The Oil Engine", September and October, 1938.

With the exception of large marine engines, driving propellers through direct couplings, and aero engines, every Diesel unit needs a governor to ensure close control of the engine speed under conditions of fluctuating load. Governor failure is a rare occurrence, for it is customary to employ a large factor of safety in the design, but the risk of failure is always present, and it is unwise to depend with an easy mind on correct functioning at all times by refusing to consider the possibility and the serious consequences of breakdown if an alert attendant is not at hand.

Realizing this, many manufacturers in this country and abroad fit over-speed governors on their engines, either as standard practice or as extras. In some cases a combination unit to stop the engine in the event of over speed or failure of the lubricating oil or water pressure is employed, whilst in others separate devices are used for these functions. Over speed, loss of oil pressure and loss of water pressure are almost equally to be feared, for each of them can be disastrous in its consequences if not detected at once. It is a wise course, therefore, to nullify these evils by arranging for automatic stopping of the engine immediately on their appearance, and in this and a subsequent article the practice of British makers in the provision of safety devices for this work will be described. The Simms and Evershed and Vignoles proprietary governors for over-speed control will also be described and illustrated.

The arrangement illustrated in Fig. 1 is an example of the latest Davey Paxman practice, and embodies a simple governor unit. This takes its drive through a shaft (C) with pinion and spur gear (A) attached to the end of the pump spindle. The operating shaft of the governors is fitted with a sleeved lever (J), which is free to move within a slot in a shaft (N) over the normal speed range of the engine. An adjustable stop screw (O) is provided, with which the lever (J) makes contact when a predetermined increase in speed has been attained.



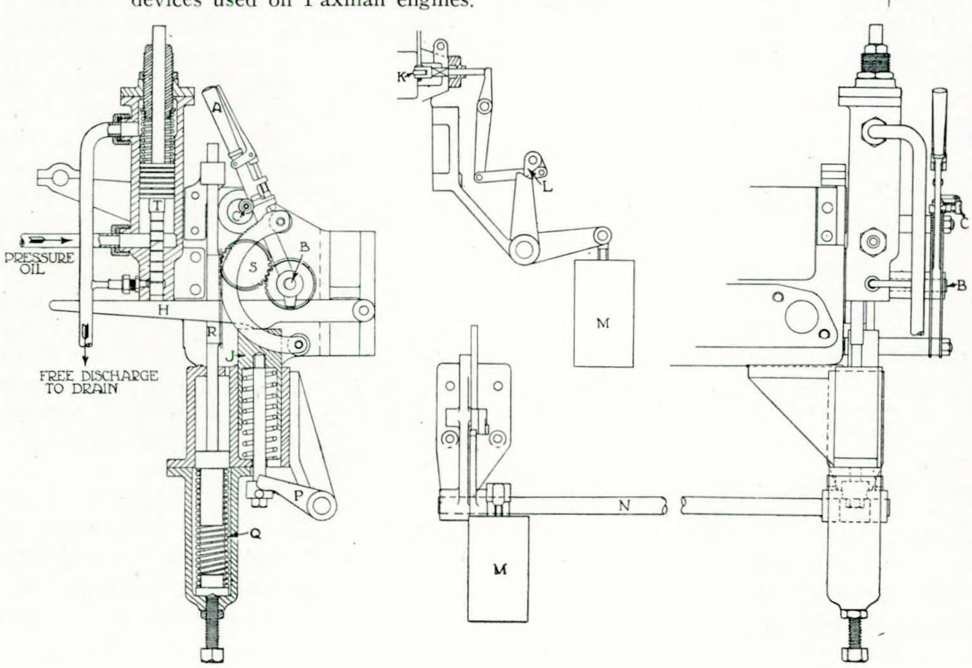


FIG. 1 (right) and FIG. 2 (below) show over-speed safety devices used on Paxman engines.

Contact is made at approximately 10 per cent. over-speed, and any further increase in engine speed causes a catch at the end of the shaft to release the plunger (M), which falls under the action of the spring (T). Movement of this plunger cuts off the fuel supply to the injection pumps, and brings the engine to rest. The mechanism is reset by lifting the handle (R), when the shaft (N) is returned by the action of a spring (U) to retain plunger (M) in the running position.

The system shown in Fig. 2 is also controlled by a single governor (not illustrated), of the vertical double-weight type, so designed as to be unstable. Advantage is taken of this instability to allow the weights to fly open suddenly when the speed rises beyond the set value.

This governor terminates in the link (K), which operates through the medium of a rocking lever and links a catch (L). The catch, when moved, allows a weight (M) to fall and impart a torque to a spindle (N), operating another lever (P), which draws down a plunger (J), normally spring-supported. The control lever (A) is held in the normal running position by a projection on its lower end, engaging in a notch cut in lever (H). This lever is in turn supported by the plunger (J), to which reference has just been made.

When the over-speed governor releases the weight (M) the plunger (J) is lowered in the manner described, and allows lever (H) to fall, due to its own weight. The control lever is then released, and is returned to the stop position by the action of a spring (Q), rack (R) and pinion (S) operating on the spindle (B), on which the lever is mounted. A device for stopping the engine in the event of the lubricating oil pressure falling below the safe limit is incorporated in the system. Plunger (T) is supported against the load of a spring by the normal oil pressure, and should this pressure fail, the spring depresses the plunger, thus releasing the control lever. The engine is then brought to rest, as it would be had the over-speed generator acted.

These two systems, with certain variations, have been used throughout the range of engines up to 1,200 b.h.p. manufactured by the company. The point in the fuel line at which the cut-off is effected depends upon the arrangement of the fuel pumps.

On the large Fraser and Chalmers air-injection engines an emergency governor is driven from the end of the engine camshaft. This is of the spring-loaded centrifugal type, and operates a plunger which controls three ports. The complete system is shown in Fig. 3. The first port is connected to the engine lubricating oil supply. The second port connects to an oil trip and whistle alarm, whilst the third is a drain connection. When the engine is running at its normal working speed the governor weights are closed and the plunger is in the "in" position, enabling the engine lubricating oil to pass from the first to the second port, and so on to the oil trip.

At 5 per cent. above normal running speed the governor weights commence to open and move out the plunger, until at 10 per cent. above the running

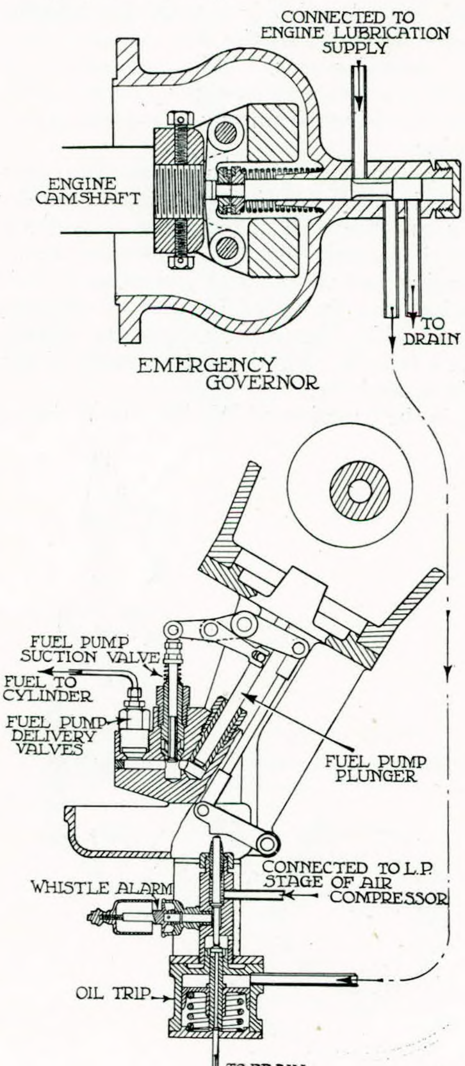


FIG. 3.—The Fraser and Chalmers emergency governing and lubricating-oil relay arrangement is shown above.

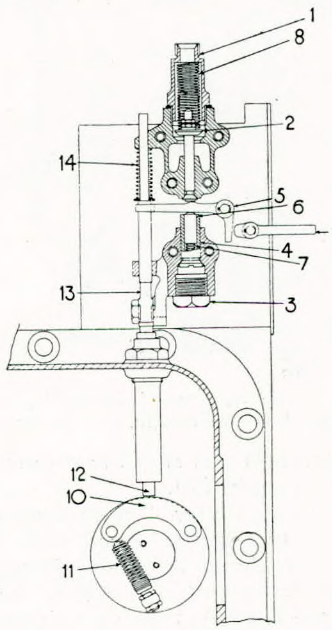


FIG. 4.—The Petter over-speed and oil-pressure safety device.

speed the plunger is in its "out" position. In this position the first port is closed, whilst the second and third ports are interconnected, thereby cutting off the oil supply to the trip mechanism, and at the same time relieving the oil pressure in the mechanism.

As the oil pressure in the trip drops, a spring under the trip piston pushes up a central spindle, which operates first a valve admitting air under pressure to a whistle to give audible warning to the engine-room staff; and secondly, its further movement by means of levers opens the fuel pump suction valves. The fuel oil supply is thus by-passed and the engine speed drops. From this description it will be clear that the trip mechanism will also

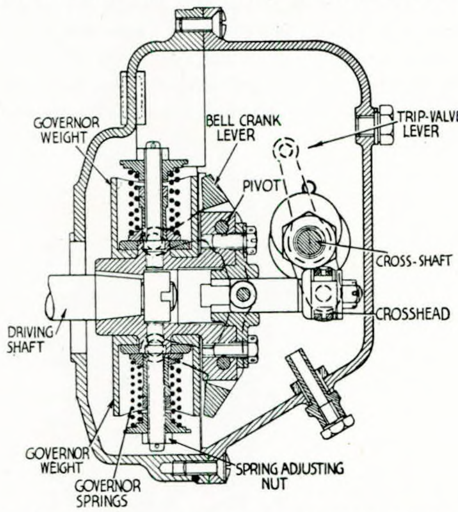


FIG. 5.—The Simms over-speed governor, which can be fitted as an extra to any engine.

come into operation and stop the engine, should the lubricating oil supply fail.

On the new Petter "superscavenge" engine safety devices for engine speed and oil-pressure failure are fitted; the arrangement adopted is shown in Fig. 4. Referring to this it will be seen that both safety stops are combined in one unit, the operation of which is as follows: Fuel is introduced through the union (1) and passes the mushroom valve (2) on its way to the fuel pumps. Lubricating oil under pressure is supplied through the bush (3) and holds up a plunger (4); this has the effect of lifting the fulcrum valve (5) through a tappet and spring (6 and 7) and thus opening the valve (2). Should the oil pressure fail, the valve is closed by the return spring (8) and the fuel supply is interrupted. As is usual with such devices it is impossible to restart the engine without resetting, and if the fault is left uncorrected the device will continue to function. For ordinary starting purposes a catch (9) is provided to hold the valve open until the oil pressure builds up; the catch is then released automatically.

If the engine speed exceeds a predetermined value the weight (10) overcomes the tension of spring (11) and flying out in its circular path lifts the push rod (12). This in turn releases a bell-crank catch (13) and allows a powerful spring (14) to depress the fulcrum lever (5) against the oil pressure. The fuel valve is then closed by spring (8) and the engine stops.

Many of the stationary engines and all the locomotive engines manufactured by Harland & Wolff, Ltd., are fitted with over-speed governors which act by cutting off the supply of fuel to the injection pumps.

An over-speed governor manufactured by Simms Motor Units, Ltd., is illustrated in Fig. 5. This governor is of the spring-loaded centrifugal type and was developed from the maker's fuel-pump governor. The springs retaining the governor weights are initially compressed and the weights are held in position until the centrifugal force overcomes the spring load; movement of the weights is transmitted to a central spindle by bell-crank levers and thence to a cross-shaft on the external end of which is a lever for coupling to a trip valve arranged in the fuel supply pipe by the engine manufacturer. The governor can be arranged to operate at any speed up to 1,000 r.p.m. and may be driven from any convenient position on the engine.

A completely different type of over-speed governor is made by Evershed and Vignoles, Ltd. This is an electrical tachometer, consisting of a permanent magnet direct-current generator and a dial-reading indicator. The generator is driven by the engine, and the speed in r.p.m. is shown on the indicator. The instrument can be arranged to operate a relay to shut down the engine if the speed exceeds a predetermined figure.

It is the practice of W. H. Allen, Sons & Co.,

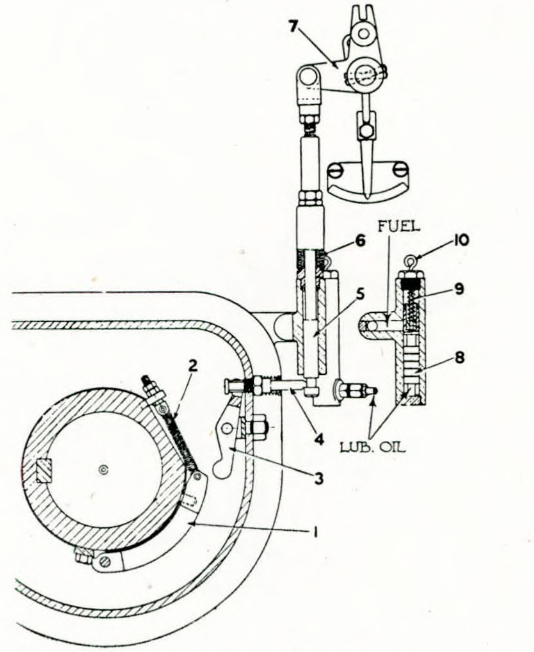


FIG. 6 shows the Blackstone controls for excessive speed and failure of lubricating oil pressure.

to supply engines with special safety devices only when the fitment of these is specified by the customer for the Allen engineers are more in favour of the use of alarm mechanisms which by means of signal lights or hooters call attention to low lubricating oil pressure or high cooling water temperature. They consider that faults in these systems seldom develop suddenly and, when they do appear, can usually be rectified without stopping the engine. This policy is perhaps the best when engines are engaged in important services and involuntary shut-downs are to be avoided at almost all costs. Over-speed governors, also, are not incorporated as standard features. Sudden failure of the engine governor is not feared, for undue friction in the mechanism, an indication of future trouble, is usually reflected in the governing of the engine and, it is claimed, can be corrected before any risk of damage due to over-speeding arises.

Safety controls of the type shown in Fig. 6 are fitted to Blackstone medium-speed vertical engines. In the speed control a weight (1), rotating with the crankshaft, is restrained by the spring (2). If the speed of the engine rises above the predetermined figure (700 r.p.m.), a plunger (5) is released by the weight striking a lever (3) that withdraws a trip-slide (4). The plunger is at once pushed upwards by spring (6) and throws over the trip-lever (7). This action pushes the fuel pump control rod to the "no-fuel" position and the engine stops. The hood enclosing spring (6) is forced down by hand and the device reset by pushing the trip-slide into the groove in the plunger.

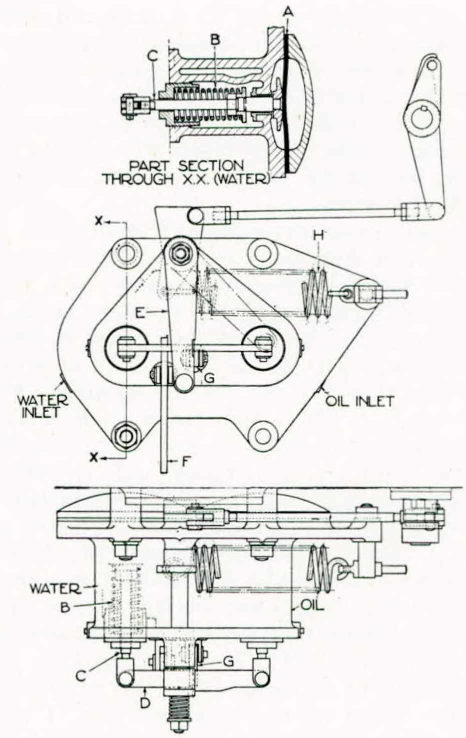


FIG. 8.—The English Electric control for lubricating oil and water pressures.

Should the lubricating oil pressure fail, the plunger (8) is driven downwards by the spring (9), thus blocking the fuel passages and thereby interrupting the fuel supply to the injection pumps. When priming the fuel system, plunger (8) may be lifted by hand or the lubricating system may be primed first, in which case the pressure, being maintained for some time, will support the plunger sufficiently long for the engine to be started.

The arrangement used on Brush engines for automatic stopping should the lubricating oil pressure fail makes use of a heavy weighted lever which is released and swings round on its pivot under gravity when the oil pressure drops below a safe limit. The energy of the lever is used to move the fuel-pump control rods to the "no-fuel" position. This mechanism was described and illustrated on page 45 of our June issue.

On large Crossley-Premier engines an over-speed governor is employed. The arrangement, which is illustrated in Fig. 7, consists of a high-speed centrifugal governor, provided with one compression spring, a compressed-air valve operated by trip-gear and pneumatically operated decompression gear. The last is not shown in the illustration. In the event of the engine over-speeding, the fly-weights of the governor (A) open out and strike the spindle (E) of the trip-gear. Lever (B) then

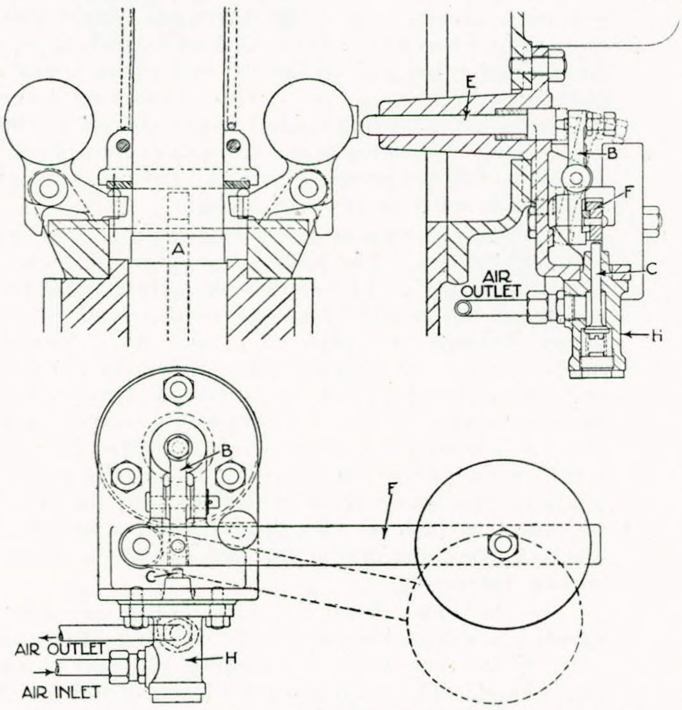


FIG. 7 illustrates the arrangement of the Crossley-Premier over-speed governor.

allows the weighted arm (F) to fall and open the compressed-air valve (C) contained in the case (H) and the opening of this valve admits air to a cylinder that controls the decompression gear. All the cylinders are decompressed simultaneously and the engine stops. This method of over-speed governing, it will be noted, is entirely independent of the fuel system.

Oil and water pressure safety devices of the type shown in Fig. 8 have been fitted on various ranges of English Electric engines and quite a number have been in service for many years. The two diaphragms (A) are compressed against the force of springs (B) by the oil and water pressures respectively, thus forcing out the spindles (C) and the floating plate (D) which holds a trip-lever (E). If either pressure drops below a predetermined figure the trip-lever is released and moved round under the action of a spring (H). The trip-lever is connected by the linkage shown to a lever on the fuel pump control shaft and through this the engine is shut down. When the engine is at rest the gear is in the tripped position owing to the lack of pressure, and must be reset before a start is made. To do this the plate (D) is moved out by a handle (F) and held by a catch (G). After the engine starts the oil and water pressures build up and the plate is pushed out a little farther so that the catch is released and falls clear.

An over-speed governor designed by the English Electric Co. for fitment to shafts rotating in a horizontal plane is shown in Fig. 9. The device consists

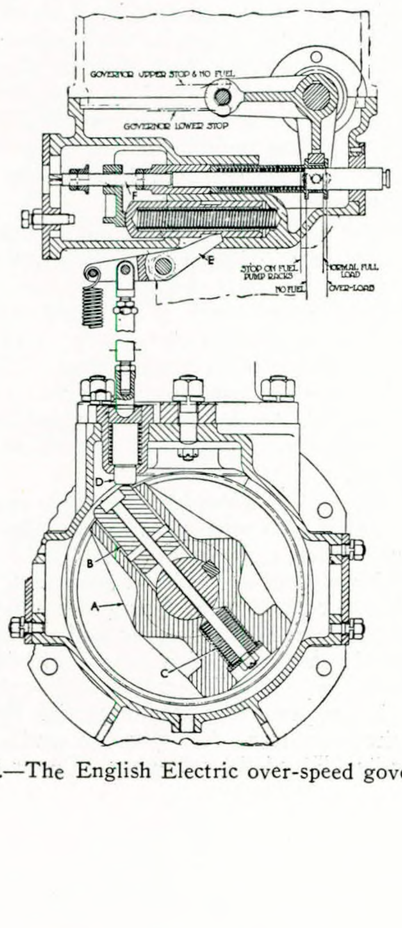


Fig. 9.—The English Electric over-speed governor.

essentially of a cage (A) mounted on the camshaft and carrying a spring-restrained weight (B). If the engine's speed becomes excessive the centrifugal force on the weight overcomes the tension of the spring (C) and the weight moves out to strike plunger (D). The last releases a catch (E) connected in the manner shown to the fuel pump control rod (F) through which the engine is shut down.

On Mirrlees high-speed engines a device, the arrangement of which is shown in Fig. 10, is fitted for enforcing a stop should the oil pressure fail. Oil under pressure is introduced above a spring-loaded piston, the latter being provided with a rod on the end of which is a tripping bar. The fulcrum pin through the fork of the governor lever operates in a specially shaped slot in an extension bar fitted to the fuel pump control rod; if the slotted lever be raised, the fixed connection between it and the fulcrum pin is severed so that the fuel control rod is free to move, under the action of a spring, to the "no-fuel" position, the relative movement between the extension bar and the fulcrum pin being accommodated by the slot in the former. Thus the engine is stopped through the spring-loaded piston lifting the trip bracket and releasing the pump control rod when the oil pressure fails.

The system is reset by means of a self-disengaging handle that depresses the piston initially and a push at the end of the slotted bar for re-engaging the fulcrum pin in its normal position.

An emergency governor is fitted to the same range of engines and its system of operation resembles that of the oil-pressure safeguard. The governor consists of a spring-restrained weight that moves out when its speed of rotation exceeds a predetermined value and strikes the end of an operating plunger. This, in turn, moves a bell-crank lever that releases a spring-loaded trip-rod below the slotted lever connected to the fuel-pump control rod. As with the oil-pressure device, the over-speed mechanism must be reset by hand.

Electro-mechanical safety devices are used on National engines. The lubricating oil pressure control, shown in Fig. 11 has two plungers (C and D) which are supported by the pressure of oil introduced through the pipes (A and B). Should the pressure drop, a spring forces down the plunger and an electrical circuit is completed through the terminal switch (E). An emergency governor (see Fig. 12) embodying two revolving weights (XX) is fitted on the end of the camshaft and if the engine tends to run away these weights strike the lever (A) and trip the lever (B), thus allowing a weighted lever (C) to come into action and operate a switch outside the casing.

In the case of oil pressure failure or over-speeding a solenoid is energized through a relay and cuts off the fuel supply by tripping the fuel pump control rod. The arrangement allows of additional safeguards being used; for instance, the engine can be stopped if the temperature of any main bearing

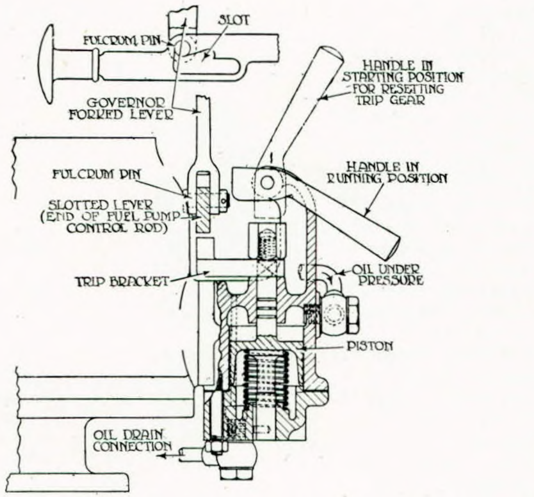


FIG. 10.—The Mirrlees safeguard for lubricating oil pressure.

becomes excessive, if the water outlet temperature exceeds a safe value or if the engine speed becomes abnormally low

A Ruston-Hornsby mechanism for guarding against damage from failure of lubricating oil or cooling water pressure acts by cutting off the fuel supply to the injection pumps. It incorporates a spring-loaded plunger contained in a cylinder through which a transverse passage is bored. This passage is part of the supply line to the fuel injection pumps and is open when a groove in the plunger coincides with the entry and exit holes in the cylinder. Below the plunger is a cross-bar,

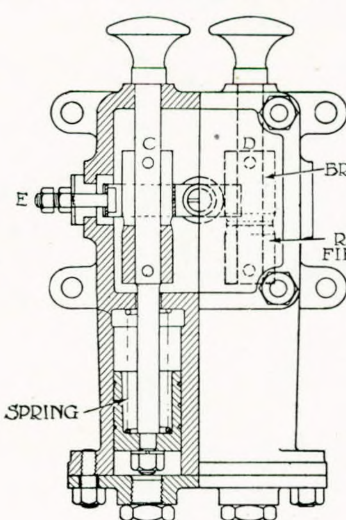


FIG. 11.—This National device is used for indicating failure of the lubricating oil pressure.

pivoted to it so that both form an inverted "T". Oil-operated and water-operated pistons press on rollers at the ends of the cross-bar. and hold the fuel-control plunger in the open position against the thrust of the spring. Should either pressure fail, the cross-bar pivots about the roller remote from the piston of the system at fault and the plunger moves sufficiently to interrupt the fuel supply. A cam operated by a self-disengaging hand-lever and positioned directly below the plunger, is employed to support it and allow the passage of fuel at starting.

The arrangement provides also for the use of electrical contacts to enable alarms to be operated instead of the engine being shut down or before this occurs. The system was illustrated on page 134 of the August issue of "The Oil Engine".

Causes of Oil-failure and Over-speed.

Ordinarily, rapid failure of lubricating oil pressure is due either to the oil filters becoming clogged through neglect or to a breakdown of the

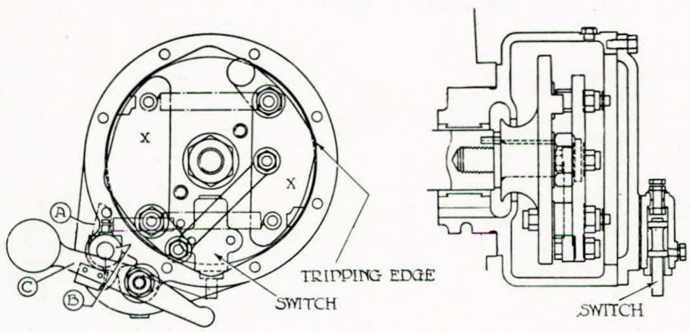


FIG. 12.—The over-speed governor used on National engines.

anti-friction lining of an important bearing whereby oil is allowed to escape freely into the crankcase. Filters are most apt to choke during the first few hours of running of a new engine and especial care is necessary at this time. Gauges and oil pumps, including relief valves, rarely cause trouble, but an oil pipe may break and provide a free outlet. When lubricating oil coolers are fitted, loss of pressure may be due to the accumulation of sediment therein.

Each engine maker recommends a minimum value of oil pressure, generally about 10lb. per sq. in., and in no case should the engine be run on load with the oil at less than the value advised as there would be actual danger of some parts working without lubrication under these conditions.

From a perusal of this and the previous articles it will be noticed the two most popular methods of effecting emergency stops in the event of over-speed are (1) to bring the fuel pumps to the "no-fuel" position and (2) to interrupt the supply of fuel to the injection pumps. The former would be preferable to the latter if the fuel pumps and linkage could be relied on implicitly, as it is generally considered bad practice to allow the fuel pumps to run themselves dry.

The Crossley-Premier method of decompressing all cylinders is perhaps the best of all, for overspeed may be due not to the governor itself but to a fault developing in the fuel-pumps or linkage and making it impossible to return the pumps to the "no-fuel" position.

Another method of stopping a Diesel engine that has been overlooked, or at any rate has not been adopted by any of the makers whose systems are described, is to interrupt the air supply to the engine by means of a throttle valve in the air-intake manifold. Such a valve can be made naturally unstable in its open position so that very little energy is required to start it closing after its release mechanism has come into operation. In this case the intake manifold must be strong and air-tight.

BOARD OF TRADE EXAMINATIONS.

List of Candidates who are reported as having passed examinations for certificates of competency as Sea-Going Engineers under the provisions of the Merchant Shipping Acts.

For week ended 5th January, 1939:—

Name.	Grade.	Port of Examination.
Warren, Christopher B.	2.C.	Liverpool
Stephenson, James V.	2.C.M.	"
Clements, Christopher	2.C.	Newcastle
Collier, George A.	2.C.	"
Harbottle, Robert	2.C.	"
Lawson, George V.	2.C.	"
Nixon, Thomas P.	2.C.	"
Richardson, Ernest	2.C.	"
Thompson, Thomas F.	2.C.	"
Allen, Charles W. G.	2.C.M.	"
Barnes, William F.	2.C.M.	"
Herriotts, George	2.C.M.	"
Jones, Robert G.	2.C.M.	"
Pigott, Albert	2.C.M.	"
Irvine, Alexander P.	2.C.	Leith
Mulgrew, Thomas	2.C.	"
Pearse, David	2.C.	"
Peterson, Andrew S.	2.C.	"
Duncan, Charles	2.C.M.	"
Evans, John N.	2.C.	Cardiff
Jones, Stuart M.	2.C.	"
Shepherd, Philip J.	2.C.	"
Cutlack, Valentine N.	1.C.	London
Naylor, Leslie F.	1.C.	"
Piercy, James C.	1.C.	"
Sunners, Brian P.	1.C.	"
Walsh, Frederick C.	1.C.	"
Richmond, Norman W.	1.C.S.M.	"
Robinson, John	1.C.S.M.	"
Weller, Herbert C. G.	1.C.S.M.	"
Muirhead, John S.	1.C.M.E.	"
Murphy, Evan C.	1.C.	Liverpool
Bowie, Peter J.	1.C.M.	"
Bell, Robert C.	1.C.M.E.	"
Blood, William H.	1.C.M.E.	"
Perks, Eric A.	1.C.M.E.	"
Bourhill, Hugh G.	1.C.	Glasgow
Dryden, Alan D.	1.C.	"
Kennedy, Allan	1.C.S.E.	"
Sinclair, William A.	1.C.S.E.	"
Douglas, William	1.C.M.E.	"
Fraser, James L.	1.C.M.E.	"
Graham, Alexander A.	1.C.M.E.	"
Hay, John C.	1.C.M.E.	"
McMeckan, Hugh	1.C.M.E.	"
Simm, William B.	1.C.M.E.	"
Smith, James A. B.	1.C.M.E.	"
Walkinshaw, Gordon	1.C.M.E.	"
Shearer, Daniel	1.C.M.E.	"
Barton, Edward F.	2.C.	London
Croft, Charles H.	2.C.	London
Hocquard, Carl P.	2.C.	"
Holyoake, Cedric J.	2.C.	"
Marrs, Hugh R.	2.C.	"
Collings, Gerald R.	2.C.M.	"
Johnston, Albert	2.C.	Glasgow
Macdonald, Ronald	2.C.	"
Cadenhead, Ronald S.	2.C.M.	"
Douglas, Donald J.	2.C.M.	"
Lochhead, Archibald	2.C.M.	"
Olsson, Cyril F.	2.C.	Hull
Schoch, Frederick W.	2.C.M.	"
Bramald, Alfred N.	2.C.	Liverpool
Grieve, John D.	2.C.	"
Latto, Arthur A. M.	2.C.	"

For week ended 12th January, 1939:—

Name.	Grade.	Port of Examination.
Hayne, David C.	1.C.	Leith
Davison, Joseph H.	1.C.	Newcastle
Mack, John J.	1.C.	"
Metcalfe, John C.	1.C.	"
Tate, Arthur B.	1.C.	"
Wales, Vincent	1.C.	"
Nash, George H.	1.C.M.	"
Ramsay, James B.	1.C.M.	"
Rush, Thomas	1.C.M.	"
Sinclair, Arthur	1.C.M.	"
Thompson, Wilfred	1.C.S.E.	"
Hughes, William L.	1.C.M.E.	"
Moar, John	1.C.M.E.	"
Baker, Stanley C. J.	2.C.	"
Merwood, William C. C.	2.C.	London
Reynolds, Austin C.	2.C.	"
Niblock, Stewart J.	2.C.	Liverpool
Bell, Archibald	2.C.	Glasgow
Beveridge, Malcolm	2.C.	"
Lyall, Duncan S.	2.C.	"
McKay, James A.	2.C.	"
Watt, John	2.C.	"
MacQuarrie, Alexander	2.C.M.	"

For week ended 19th January, 1939:—

Name.	Grade.	Port of Examination.
Brash, James	1.C.	Glasgow
Gillespie, James	1.C.	"
Raine, Reginald V.	1.C.	"
Souter, James L.	1.C.	"
Anderson, David C.	1.C.M.	"
Cockle, John B.	1.C.M.	"
Wilkieson, Angus	1.C.S.E.	"
Gilfillan, Robert G.	1.C.M.E.	"
Marshall, William J. C.	1.C.M.E.	"
Brown, James McA.	1.C.	Liverpool
Ford, Cecil D. W.	1.C.	"
Clark, Robert H.	1.C.M.	"
Willett, Herbert A. D.	1.C.	London
Smith, George	1.C.M.E.	"
Dale, Ernest	1.C.	Newcastle
Gatenby, Ronald	1.C.	"

For week ended 26th January, 1939:—

Name.	Grade.	Port of Examination.
Bisset, Ernest E. W.	2.C.	London
Arakie, Yorke	2.C.M.	"
Broom, Alfred J. J.	2.C.M.	"
Holmes, Richard T. W.	2.C.M.	"
McGuiness, Robert W.	2.C.	Hull
Potter, Charles J.	2.C.	"
Boyd, William C.	2.C.M.	"
Carstairs, Robert A. R.	2.C.	Leith
Porter, Peter M.	2.C.	"
Murray, Francis A.	2.C.	Glasgow
Short, Wilfrid G.	2.C.	Cardiff
Ardern, William H.	2.C.	Liverpool
Kameen, Thomas	2.C.	"
Howarth, John	2.C.M.	"
Hughes, Hugh	2.C.S.M.	"
Boag, Robert	2.C.	Newcastle
Korn, Henry E.	2.C.	"
Morgan, George	2.C.	"
Wood, George	2.C.	"
Burdon, John E.	2.C.M.	"
Burke, Edmund	2.C.M.	"