ENGINEERING ABSTRACTS

Section 3. SHIPBUILDING AND MARINE ENGINEERING

Compiled and published by The Institute of Marine Engineers in co-operation with The Institution of Naval Architects.

Volume XVII, No. 2, February 1954

Annual Subscription: 25s., post paid

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Electrolytic Corrosion Inhibiting

The authors point out that as far as is known, there has been no application of energized anodes for electrolytic protection of the tanks of petroleum carriers in service. An experimental installation has been made in a tank used as needed for oil cargo, dry cargo and salt water ballast. The installation was made and all operations were conducted by the ship's crew in accordance with instructions furnished with the apparatus. The protection process was applied on the return trips from Europe with sea-water ballast. On the second voyage it was decided to raise the current density for attempted exfoliation of rust from tank surfaces. This decision resulted from the fact that on the previous voyage a considerable amount of tallow had been leached from the rust coating. When the tank was cleaned at the end of the voyage the rust flakes removed filled thirteen 5-gallon paint cans. After several voyages, over a period of about six months, the experiment was discontinued and a close comparison was made with a similar adjacent tank, not protected but otherwise subjected to the same conditions. The last cargo carried was tallow, and after discharge of this cargo both tanks were twice cleaned, first by a ship cleaning gang and then with the "Quin-sinlee" rig. Excerpts from the report of inspection of the two tanks, after cleaning, are quoted as follows: "On examining the processed (port) tank, we were impressed with the cleanliness of the steel surfaces and with the absence of odour in the tank which had been sealed up for some time. The principal surfaces including the top and bottom of the tank were free of rust or trace of tallow and felt clean and dry to the bare hand. On some of the more shielded areas, such as the pipes under the guard angles and in the bosom, there was a thin, soaplike film, which, when rubbed with the fingers, had an odour of tallow. In the untreated (starboard) tank, on the flat surfaces, the rust was about $\frac{1}{8}$-in. thick and was covered by a thin film of tallow which appeared to have come out of the rust. In concealed corners, coves of channels, etc., the rust was thick, rough and porous and contained considerable tallow on the surfaces and, particularly, in pits and pores of the rust. The air in the tank was heavy with the odour of the tallow. It was agreed by all present that the port tank was practically ready for any type of bulk oil or dry cargo, whereas the starboard tank would have to be descaled and wiped out for almost any cargo other than tallow or fuel oil".—*Paper by H. F. Harvey, Jr., and O. J. Streever, read at the Annual Meeting of The Society of Naval Architects and Marine Engineers, New York, 12th-13th November 1953.*

Oil Burner Trials

Details are given of the design and planning of the trials and the techniques of measurements on the flames from several commercial high-pressure blast-type burners, together with tables of the input variables applied to the burners and graphs showing the effects of the variables on several important parameters such as the radiation and emissivity of the flames and the temperatures of the surrounding walls. The results show that the greatest change of flame condition is brought about by a change in the momentum of any given fuel quantity. This change in flame condition from any given burner is greater than the differences in flame produced by different burners when they are operated with the same momentum. Comparing flames from the same type of burner, with the same momentum and fuel quantity, but with air as the atomizing and carrying medium in one case and with steam in the other, the flame with air as the atomizing agent has a higher peak radiation and emissivity but after this the two flames are indistinguishable. In the experiments described, increasing the momentum always decreased the radiation and the emissivity at all points along the flame length, the temperature of the flame in the very early part rising with increasing momentum; the air-atomized flame was also at a higher temperature than the steam-atomized flame. An increase in jet momentum causes an increase in flame temperature, more so for the air-atomized flame than for the steam-atomized flame, whereas in the later development of the flame an increase in momentum decreases the flame temperature, there being little difference between the air- and the steam-atomized flames. From the results it is seen that the size of the oil droplets for the range examined has no appreciable influence on the rate of combustion and subsequent emissivity and flame temperature. The biggest difference in mean particle size was produced by going from a back-end to a front-end mixer (but both of the internal-mixer type) burner and there was very little difference in their performance with regard to the combustion and radiation and there was little difference in efficiency of production of jet thrust for given atomizing agent/fuel ratios used between these two kinds of burner.—*R. Mayorcas and M. Rivière, Journal, Institute of Fuel, October 1953; Vol. 26, pp. 211-224.*

Fuel Burning Heaters for Marine Use

At the recent Engineering and Marine Exhibition at Olympia, Smiths Industrial Instruments, Ltd., showed examples of fuel burning units for producing heat. The problem of producing a heater capable of burning Diesel fuel or paraffin of relatively small heat output has been a difficult one and even in America, where heaters of this type are used on various forms of transportation, such as rail transport and motor coaches, etc., the smallest heater commercially available, capable of operating on Diesel fuel, has a minimum output of 75,000 B.Th.U. per hr. In Germany, the firm of Wilhelm Baier of Stockdorf, Munich, has been successful in developing a Diesel fuel burning heater for capacities from 15,000 B.Th.U. per hr. upwards, and in view of their considerable design and service experience with this type of unit, Smiths have entered into an arrangement to pool developments and share patents, etc., in this field. The immediate result of this technical co-operation is the production of the Smiths Mark IIA oil burning air heater, of which a general technical description is given herewith. This heater burns either Diesel oil or paraffin with a maximum output of approximately 50,000 B.Th.U. per hr., the heat from combustion being applied to a separate current of air circulated by one of the heater fans. Fig. 1 shows a diagrammatic view of a section through this heater, illustrating the general principle of combustion air flow and fuel flow, etc. The heater has two electrically driven air fans, one to provide the air for combustion of the fuel and the other to provide air which, when heated, is distributed into the space which it is required to heat. A special feature of this heater is that the combustion air circuit, including the actual products of combustion and exhaust gases, is quite separate and isolated from the air being heated for supply purposes—as a result the latter air cannot possibly become contaminated by the gases or fumes resulting from combustion. This special feature is obtained not only by the use of a high quality stainless steel barrier between the two air streams but also by the fact that pressure of the air being heated is higher than the pressure inside the combustion circuit, so that any leakage can only take place into and not out of the combustion circuit. It should be mentioned that with the special combustion arrangements in this heater, the exhaust gases are entirely non-toxic, since the carbon monoxide content is well below the safety level. The heater consists essentially of a combustion unit which supplies hot gases to a heat exchanger and an independently powered air fan which propels the air to be heated over the heat exchanger and to the distribution ducting, the whole being contained within a sheet metal cylindrical casing approximately 11 inches diameter and 33 inches long and weighing 65lb. The combustion unit comprises an electric motor which drives, through a flexible coupling, a positive displacement piston type fuel pump. The spindle projecting through the fuel pump housing drives also the combustion air centrifugal impeller, fuel slinger and radiation shield. Air for combustion is drawn by the impeller through an intake tube into the cast housing of the fuel motor past the fuel pump into the impeller inlet. This air is delivered through a diffuser into the combustion chamber past the fuel slinger. Fuel drawn from the supply tank by the piston pump is delivered through a "swan necked" pipe into the fuel slinger, which, rotating at motor speed approximately 3,000 r.p.m., discharges it in a finely divided spray, so as to thoroughly mix with the incoming combustion air. With this arrangement the quantity of fuel delivered is a direct function of motor speed, as is also the quantity of combustion air, so that the heat output may be controlled by the motor speed without affecting the air fuel ratio. Ignition of the fuel air mixture is affected by a glow plug situated in the plane of the fuel slinger. This glow plug is operated only to initiate combustion. During normal operation it is inoperative.—*The Shipping World, 4th November 1953, Vol. 129; pp. 381-382.*

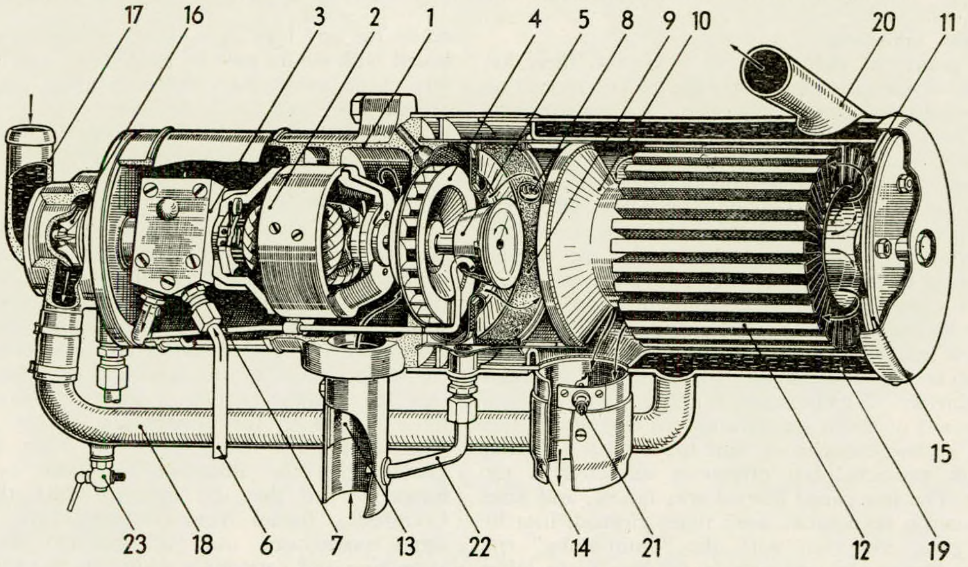


FIG. 1—*Section through fuel burning water heater*

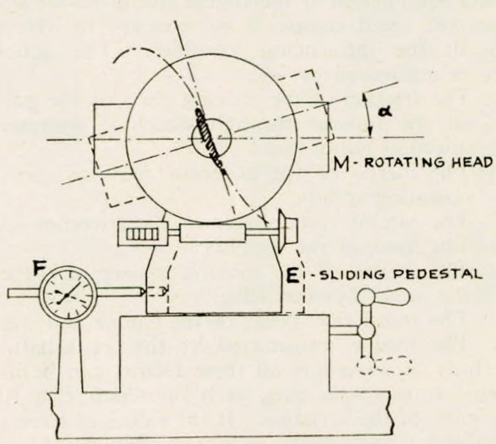
1.—Motor housing body. 2.—Electric driving motor. 3.—Fuel pump. 4.—Combustion air fan. 5.—Fuel distributor. 6.—Fuel inlet pipe. 7.—Fuel pipe—pump to distributor. 8.—Glow plug igniter. 9.—Front combustion chamber. 10.—Flame constrictor. 11.—Rear combustion chamber. 12.—Heat exchanger fins. 13.—Combustion air inlet. 14.—Combustion exhaust. 15.—Inspection and cleaning cover. 16.—Water pump. 17.—Water suction pipe. 18.—Water pipe—pump to heat exchanger. 19.—Water jacket. 20.—Water outlet from heater. 21.—Combustion indicator. 22.—Combustion chamber drain pipe. 23.—Water drain cock.

Fuel Oil Delivery from Tanker through Submarine Pipe

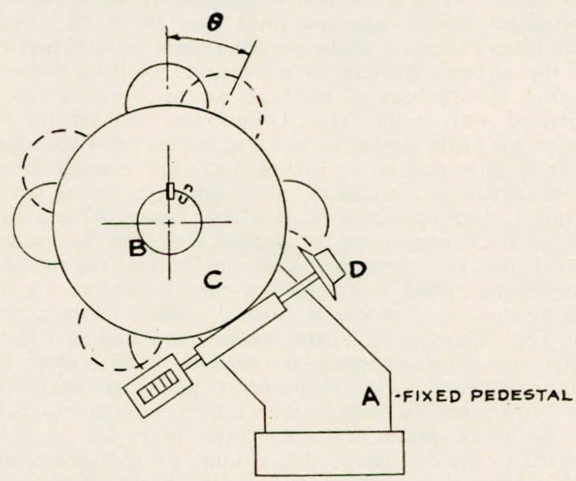
The location and capacity of the Moss landing steam plant made it desirable to provide for deliveries of fuel oil of the highest commercial viscosity from large tankships through a submarine pipe line approximately 1¼ miles long to the storage tanks of the plant. The use of submarine lines is not new, but the type of application involved several unknowns for which there was very little prior experience. In this paper the economic size of the line is discussed in some detail, and experiences in the displacement of cold oil and test data relating to higher viscosity oils are presented. Other features covered are the radio-communication facilities, the cathodic protection of the buried line, and the general maintenance problems. The submarine fuel-unloading facility described combines features of reliability, ease of operation, economy of time and manpower that make it worthy of consideration for oil-fuel plants near deep water, the paper states. Some of the doubtful factors, such as the time to remove cold plugs and the need for recirculation, have proved far less onerous than anticipated. The design used at Moss Landing incorporates margins sufficient to make the unloading time dependent almost entirely on the capacity of the pumps on the ships that so far have delivered oil. The operations corroborate the design estimate that little if anything would be gained by the use of larger-diameter piping. It is the authors' belief that the Moss Landing arrangement can handle fuel oil of viscosity as high as 300 SSF at 122 deg. F. without difficulty.—*Paper by S. F. Johnson and E. A. Salo, A.S.M.E., 1953 Semi-Annual Meeting; Paper No. 53-SA-30.*

New Propeller Measuring Instrument

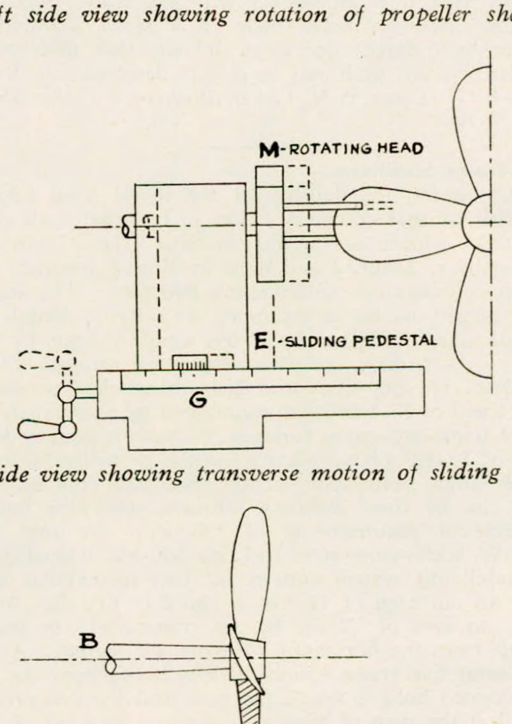
Since the accuracy of the manufacture of small propellers used in model test work is of great importance, the Experimental Towing Tank has had a new instrument built for the accurate measurement of propellers of 2 inches to 12 inches in diameter. Its initial use has been in measuring the propellers used in the self-propulsion scale effect project sponsored by the Office of U.S. Naval Research at the Experimental Towing Tank. The measuring instrument consists of a shaft on which the propeller to be measured is mounted, and a head carrying two probes which can be rotated so that the probes are perpendicular to the designed pitch surface at any desired radius. These probes are connected to dial gauges which are set to read 0 at the pitch surface, and hence when in contact with the face or back of the blade give direct readings of the blade offsets. The instrument is constructed on a small lathe bed. A fixed pedestal (A) carries shaft (B) on which the propeller to be measured is mounted. A single thread worm and 260-tooth gear (C) permits the propeller to be rotated to any desired angular position, a counter indicating the number of degrees and dial (D) the decimal fractions of a degree. The sliding pedestal (E) which carries the rotating head and the probes is arranged to move horizontally in longitudinal and transverse directions, i.e. either parallel to or at right angles to the propeller shaft axis. The gauge for longitudinal motion is shown at (F), and the scale for transverse motion is at (G). The measuring probes are carried by rotating head (M), which may be turned by means of a mechanism similar to that used for shaft (B) to any desired angle *a*, so that measurements of the blades can be made perpendicular to the helical pitch surface. The axis of head (M) intersects the propeller shaft axis at right angles. Probe (H) is fixed to a sliding base, the position of which is shown by dial gauge (J). Probe (K) is directly connected to dial gauge (L) which is fixed to the rotating head



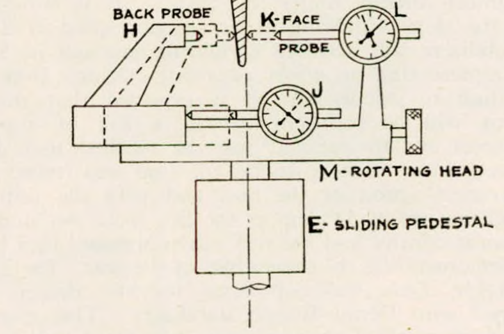
Front view showing longitudinal motion of sliding pedestal and rotation of rotating head.



Left side view showing rotation of propeller shaft.



Right side view showing transverse motion of sliding pedestal



Top view showing motions of probes

(M). The zero positions of gauges (J) and (L) are on the axis of the head, and a circular bar is provided which, when inserted in an axial hole, can be used for adjusting these gauges to zero. The procedure in checking propeller blade section offsets is first to bolt the propeller in place on the shaft and then to adjust the transverse position of the sliding pedestal to give the desired radius on scale (G). The rotating head is then set at the angle *a* corresponding to the design pitch angle at this radius. The horizontal motion of the head parallel to the propeller axis is adjusted until face probe (K) reads 0 at a convenient location on the propeller blade section, preferably where the surface coincides with the reference helix (offset=0). The offset for the back of the blade at this position can then be obtained with probe (H). Offsets are obtained for other points on the blade section by rotating the propeller successively to other angular positions. For each change in angle the head must be advanced parallel to the shaft axis by an amount determined by the designed pitch at that radius. The amounts of the shift corresponding to different angular positions are tabulated for the particular propeller before beginning the measurements. Blade widths at the various sections are determined by the use of probe (H) turned parallel to the propeller axis. The difference in angle between the leading edge and trailing edge touching positions permits blade widths to be readily calculated. For convenience in determining when probe (H) touches the blade, a neon light is arranged to indicate when the probe makes contact. Since probe (K) is directly connected to the dial gauge (L), a light spring loading suffices to ensure that it is in contact with the blade when readings are made. Measurements made on a 3½-in. diameter bronze propeller by different operators indicate that discrepancies of the order of 0·001 inch may be readily detected with the instrument.—*E. V. Lewis, S.N.A.M.E. Bulletin, October 1953; Vol. 8, pp. 17-18.*

Denny-Brown Stabilizers

The *Andes*, the flagship of the Royal Mail Lines, Ltd., is the first of that company's fleet to be fitted with stabilizing equipment, which is, in this instance, the "Denny-Brown" ship stabilizer, designed and built by Brown Brothers and Co., Ltd., in collaboration with Denny Brothers. The ship is one of the largest to be so equipped and has a length between perpendiculars of 630 feet by 83 feet moulded beam by 47ft. 6in. depth to "C" deck, a gross registered tonnage of 25,676, and carries 324 first-class and 204 second-class passengers. A service speed of 21·5 knots is maintained by a two-shaft arrangement of triple-expansion turbines of Parsons design, developing a total of 30,000 s.h.p. and driving the propellers at 140 r.p.m. through single reduction gearing. Steam is supplied at 430lb. per sq. in. by three Babcock-Johnson watertube boilers, and the electrical requirements of the ship are met by three 1,000-kW. turbo-generators and two 500-kW. Diesel generators. The stabilizing system consists of two rectangular fins, each having an outreach of 12 feet, a chord of 6ft. 6in., and representing an area of 78 sq. ft., set transversely in the ship at an angle from the horizontal of about 15 degrees. A compartment about five frame spaces in length has been cut off from a refrigerated hold to house the gear, and the fins project from the hull at the turn of bilge at a position forward of 'midships and almost directly under the bridge. It is anticipated that when the ship is moving at her service speed of 21·5 knots, the stabilizers will produce a righting moment of 5,500 foot-tons, representing an effort of about 70 tons from each fin. According to calculations, it is expected that this righting moment will be sufficient to kill a roll of approximately 22 degrees out to out. It may be recalled that during the stabilizer trials on the *Chusan*, the ship was forced to roll by the reversed action of the fins, and with the ship swinging through an arc of 34 degrees the fins were put under normal gyroscopic control and the roll was suppressed in a half swing, thus demonstrating the capabilities of the gear. Earlier this year the R.S.S. *Leda* was completed for the Bergen Line and equipped with Denny-Brown stabilizers. This ship of about 7,000 tons is the first passenger ship fitted with a stabilizer to operate in the North Sea, and records of performance show that under free rolling condition the ship rolled through a total angle of 27 degrees, which was reduced to 2½ degrees with the stabilizers in action. The fins oscillate round an axis synchronously and in opposite senses, and are balanced so that the torque to produce this angular movement is small and calls for only a small amount of power, each fin being operated by a 50 h.p. motor, while a 15 h.p. motor pump provides the power needed for extension and retraction. Rapid oscillation of the fins is achieved by two electrically driven variable delivery pumps, which provide the fluid for operating the oscillator rams, and the pumps are controlled by the precession effort of two small gyroscopes. The "vertical keeping" gyro comes into action when the ship lists, to angle the fins to resist movement from the vertical when a certain angular velocity is exceeded during the rolling of the ship. The overall control is effected from a small cabinet, installed in the bridge. The captain signals for the motors to be started, after which he switches on "Extend Fins"; then electrical connexion is made with the control gyros and the gear automatically operates to damp out the roll of the ship.—*The Engineer, 16th October 1953; Vol. 196, pp. 500-501.*

Variable Speed Governors

Mirrlees variable-speed marine governors are of the Hartnell spring-loaded type; the variable load on the springs, which determines the governor speed, is applied by an oil-operated piston. A spool valve system, connected to the throttle lever or handwheel, regulates the oil supply to the piston. A typical Mirrlees governor is illustrated in Fig. 1. To secure

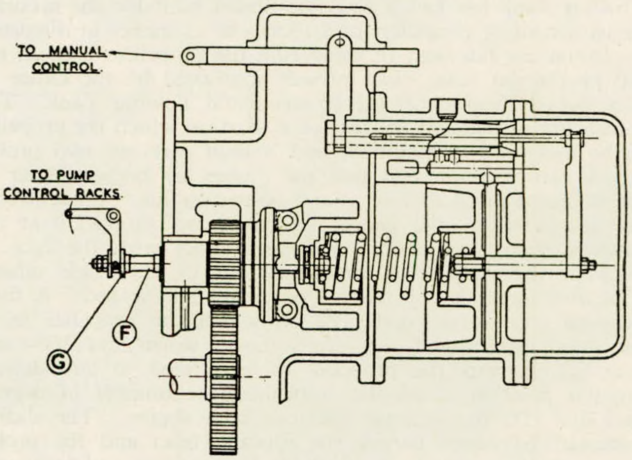


FIG. 1

F.—Governor operating spindle. G.—Spool.

the correct equilibrium of the engine and governor system, even of a constant speed engine, it is necessary to have agreement between all the influencing variables. The action of the governor is influenced by:—

(*i*) The friction of the moving parts of the governor and all the linkage through which it operates the fuel injection pump racks.

(*ii*) The inertia of the governor parts as they move the operating spindle.

(*iii*) The rate of compression of the governor springs.

(*iv*) The mass of the weights.

(*v*) The inertia of the moving masses of the engine, i.e. the total "flywheel effect".

(*vi*) The rotational speeds of the engine and the governor.

(*vii*) The torque transmitted by the crankshaft.

In the study of governors all these factors can be inter-related to provide three coefficients, each of which is a function of three or more of the variables. If the values of these coefficients are calculated it is possible, in certain circumstances, to design a highly sensitive governor, free from any tendency to hunting throughout its range, by observing the following relation-ships: Let the coefficients be known as C_1, C_2 and C_3; it can then be shown that if the product C_1 C_2 is less than C_3, a disturbance in the engine motion would cause the governor to build up a hunting motion that would increase in severity indefinitely unless the engine were shut down. If C_1 C_2 is equal to C_3, the hunting would be of constant ampitude, and would persist until a change in one of the factors occurred. The conditions making for hunting are avoided if C_1 C_2 is greater than C_3. These factors show how much care is necessary in designing a satisfactory governor, particularly one for high speed engines having small flywheel effects. It will be appreciated that the problem is greater with high speed marine propulsion engines, because of the wide speed range. With geared drives, because the inertia of the rotating parts changes when the engine is disconnected from the propeller, it is similarly complex. It is comparatively easy to maintain governor stability within the speed range, provided there is some load on the engine, but unless the designer is prepared to worsen the speed regulation by decreasing the sensitivity, there is a tendency to hunting in the high and low idling speeds when the gearbox is in neutral. At these governor positions, the quantity of fuel injected into the cylinder is so small that the engine becomes more sensitive to a slight unbalance between the load shared by the cylinders. One cylinder might be taking a small percentage overload with a consequent unloading of the other cylinders. This condition can set up an interference in the engine turning moment and, without the restraining influence of the engine load and the added flywheel effect of the driven machinery, it is transmitted to the governor in the form of a periodic disturbance. The governor, stable enough for other duties, cannot control the unloaded engine, and unpleasant hunting is the result. This fault is frequently overcome by incorporating a dashpot and generally, for slow and medium speed engines, this is a satisfactory solution. On higher speed engines, however, instances have occurred where the dashpot has proved unsatisfactory, due to incorrect adjustment or to a change in viscosity in the dashpot fluid. The problem is thus reduced to one of overcoming hunting tendencies under no-load conditions by some means other than the fitting of a dashpot. In certain types of Mirrlees engines this has been done by fitting a snubber spring in the governor. This spring has no influence on the movement of the governor from full load to low idling. It is fitted to the governor operated spindle F in place of the spool G, in Fig. 1. The governor spindle moves in and out of the governor assembly when it is controlling the fuel pump racks. When a governor is hunting, the spindle takes up a regular reciprocating

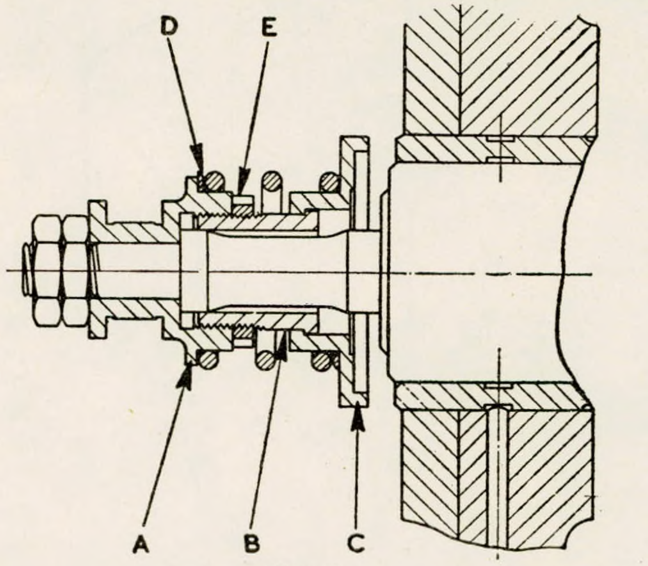


FIG. 2

A.—Spool. B.—Thimble. C.—End cap. D.—Spring. E.—Screwed ring.

motion ranging from full fuel to no fuel. The excessive inwards movement to no fuel starts the hunting, and the snubber spring cushions the inertia effect of the expanding governor weights by abutting against the face of the governor bush when the spindle is approaching the end of its inward stroke. The snubber spring assembly, Fig. 2, is adjustable in length. To set this spring, screw in the assembly to its shortest length; then with the engine running at low idling speed, start the engine hunting by disturbing the fuel pump racks. The snubber spring assembly should then be lengthened until hunting disappears. Under steady conditions of low idling, the snubber should be 0·008 inch clear of the abutting surface of the governor bush. It will then be found possible to observe the full action of the snubber. If the engine is now put on load, and then suddenly relieved of load, there will be a sharp rise in speed which will pull the spindle inwards and compress the snubber spring a small amount. A study of Fig. 2 illustrates the construction of the snubber assembly. The spool A, which is secured to the governor spindle by two lock-nuts, is threaded to receive the thimble B. The shoulder of the thimble retains the end cap C against the thrust of the stiff spring D. The adjustment to the length of the unit is made by turning the thimble in the spool by means of a pin punch inserted in the small barring holes. After adjustment, the snubber assembly is locked by means of the screwed ring E.—*(Technical Journal of the Brush Group, August 1953). The Shipping World, 25th November 1953, Vol. 129; pp. 443-444.*

Australian-built Turbine Ore Carrier

The *Iron Whyalla* was designed and built at the shipyard of the Broken Hill Proprietary Co., Ltd., Whyalla, South Australia, for the ore and finished products trade on the Australian Coast. The designed service speed at load draught is 14 knots. The hull, machinery and equipment are in accordance with the highest requirements of Lloyd's Register. The principal dimensions are as follows:

Length overall	498ft. 0in.
Length b.p.	475ft. 0in.
Moulded breadth	62ft. 0in.
Moulded depth to upper deck	38ft. 6in.
Moulded depth to second deck	30ft. 0in.

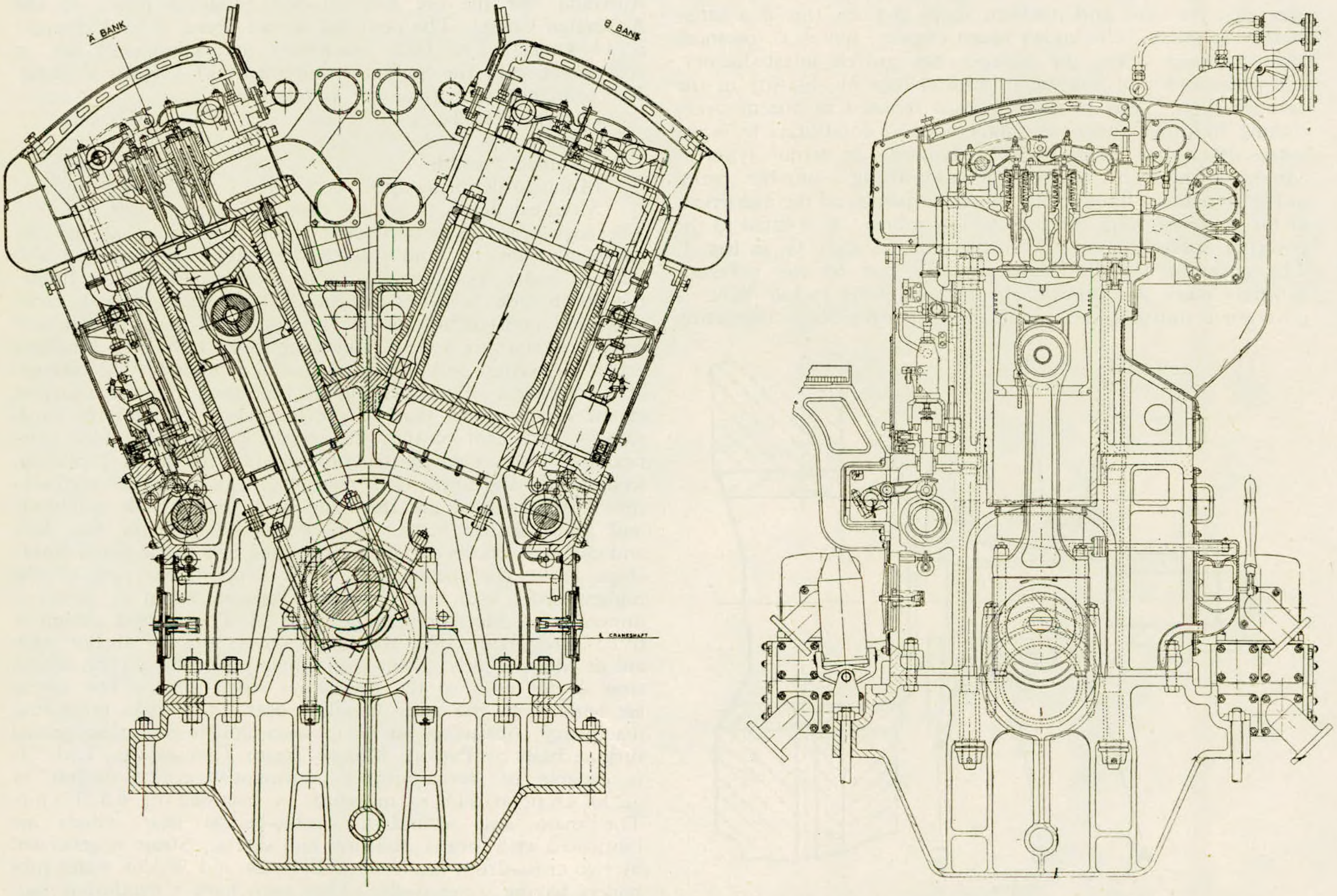
The sequence of compartments from forward to aft is fore peak tank, Nos. 1, 2 and 3 holds, deep tank, self-trimming coal bunker, boiler room, engine room, No. 4 hold above tunnel flat (with side tanks under tunnel flat) and aft peak tank. The hull construction is on the combined longitudinal and transverse framing system, and wing tanks have been arranged under the second deck in Nos. 1, 2 and 3 holds for the carriage of water ballast only. Welding has been used to a much greater extent in this vessel than in previous ships built in the yard, some 43 per cent of the steel weight having all-welded connexions. The welded construction includes the tank top plating, second deck and upper deck plating in way of the accommodation. Also welded are the main and 'tween deck bulkheads and shell plating butts. All accommodation for the deck and engineer officers and crew is situated in a single house amidships above the shelter deck. All appointments are of the highest order, each man having a separate cabin of generous dimensions. Mechanical ventilation of Thermotank design is fitted throughout. All the hatch covers on the shelter deck are of MacGregor's patent steel single-pull rolling type, which stow at the ends of the hatchways when open. The covers are handled by the cargo handling gear. The main propelling machinery consists of an articulated double-reduction geared turbine built by Parsons Marine Steam Turbine Co., Ltd. It is capable of developing a continuous service output of 6,200 s.h.p. at 115 r.p.m. with an overload of 6,820 s.h.p. The main and secondary double-helical gear wheels are fabricated with forged steel rims and shafts. Steam is generated by two cross-drum header-type Babcock and Wilcox water tube boilers having water walls. They each have a maximum continuous rating of 30,000lb. of steam per hour at 430lb. per sq. in. and 730 deg. F. These boilers are mechanically coalfired by B. and W. Detroit Roto-grate type stokers fed from hoppers which lead from the coal bunker bulkhead. These are filled at regular periods from the self-trimming coal bunker by mechanical means. A Babcock and Wilcox Hydro-vac suction type ash handling plant is installed. All the necessary fans, pumps and stoking equipment for the steam generating plant are electrically driven with the exception of the ash disposal ejector pump, which is steam turbine driven.—*The Marine Engineer and Naval Architect, November 1953; Vol. 76, pp. 444-445.*

New Marine Diesel Engine Type

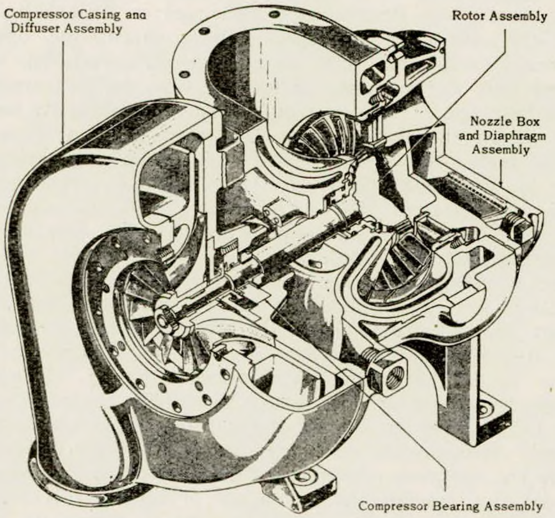
The new West Indies motor coaster *Commandant Milliasseau* has a Mirrlees engine of the new K design. The K range has been designed to replace the 13¾-in. bore by 21-in. stroke, HF series, which found wide application, both ashore and afloat in naturally-aspirated and turbo-charged forms. The latest type is offered in five-, six-, seven- and eight-cylinder in-line form, with turbo-charged versions of the last-named three. A Vee-form 12-cylinder engine in naturally-aspirated, turbo-charged or high-pressure turbo-charged form is also available and this makes use of identical running gear although the frame design is necessarily different. The cylinder bore of 15 inches and stroke of 18 inches conforms with modern trends. With the higher running speeds of 428 r.p.m., the mean piston speed is 1,284ft. per min. The maximum ratings of the naturally-aspirated and turbo-charged engines at 428 r.p.m. are 138 and 207 b.h.p. per cylinder, corresponding to b.m.e.p's of 80 and 120lb. per sq. in. The cast-iron pistons carry three compression and one scraper ring above the pin and one high-pressure scraper ring below. Cooling oil is supplied through drillings in the connecting rods, the return flow being directed by means of a nozzle into an oil catcher arranged beneath each liner, whence it is returned to the sump. In this way, thrashing by the crankwebs with consequent aeration and deterioration of the oil is avoided. Much thought has been given to the efficient cooling of the cylinder heads and the design incorporates a cast intermediate deck which constrains the incoming water to take a path at comparatively high speed over the top of the cylinder head combustion plate and around all the valves and the fuel injection nozzle. It then passes upwards through two cored passages in the deckplate to the upper chamber in the head, where it cools the valve ports and is finally discharged on the exhaust side into the main water outlet pipe. The cylinder head also carries air-starting, relief valves and a maximum pressure fitting.—*The Marine Engineer and Naval Architect, November 1953, Vol. 76; pp. 455-460.*

New Turbocharger

A turbo pressure-charger of conventional design has recently been introduced by the General Electric Company, of America, and is now in production for use on the 300 and 600 b.h.p. lightweight Packard four-stroke marine Diesel engines. The first of the two models, known as the 7SRA type, is suitable for engines of from 200 to 400 b.h.p. and the larger one, the 7SRB2 model, for engines of outputs ranging between 400 and 800 b.h.p. Both are designed for medium-pressure turbo-charging of up to 10lb. per sq. in. The main casing, compressor housing, diffuser and air impeller are of aluminium alloy, anodized to resist corrosion from sea air, except the impeller; such treatment of this important and highly stressed component might cause embrittlement. The main frame is



These sections through the Vee-type and in-line engines show the simple castings which are employed. The running gear is common to both designs water cooled. An interesting feature is the employment of precision-cast blades for the turbine wheel. These blades, which are of heat-resisting steel, are attached to the rotor by fir tree dovetailing. The rotor blank is a high-temperature alloy steel forging to which its shaft is flash-welded. The open-type air impeller, which is keyed to the shaft, is milled from an



Exploded section showing the constructional details of the new turbocharger

aluminium alloy forging. A close degree of dynamic balance is given each rotor and impeller assembly, which is carried in sleeve-type bearings; these, incidentally, are lubricated with ordinary engine lubricating oil and are provided with special seals. The turbine nozzle diaphragm is fabricated from stainless steel, the blades being welded to an outer spacer ring and mounted in free-fitting slots in an inner spacer ring, a form of construction which gives sufficient flexibility to take up expansion. The nozzle diaphragm is in turn welded to the water-cooled turbine nozzle box casting. The compressor diffuser is a one-piece anodized aluminium alloy casting. The main casing which, of course, carries the bearings, is a rigid water-cooled one-piece anodized aluminium casting.—*Diesel Progress, May 1953. Gas and Oil Power, October 1953, Vol. 48; p. 255.*

Fatigue Properties of Cast Iron

It is clear from the literature on the fatigue strength of cast iron that for materials with widely different compositions, an indication of the fatigue strength can be obtained from the tensile strength. The endurance ratio for cast iron in reversed bending fatigue has been reported to vary between 0·33-0·60. Comparison of the results of various workers shows that there is no definite relationship between the endurance ratio and tensile strength. Some workers have reported the endurance ratio to increase with tensile strength while others have shown it to decrease with tensile strength. The endurance ratio does appear to decrease, however, when tensile strengths are increased by alloy additions. Heat treatments which raise tensile strengths also decrease endurance ratios, and acicular cast irons have lower endurance ratios than pearlitic cast irons. Only a few results on the effect of understressing and overstressing on the fatigue properties of cast irons are reported in the literature, owing to the time required for such tests. It has been shown that the endurance limits of weak cast irons with both pearlitic and ferritic structures can be increased to a remarkable degree by previously understressing at or just below the endurance limit. Little change in the fatigue properties of a higher strength pearlitic material was effected by understressing. Weak cast irons also show a remarkable ability to absorb overstress, both in the notched and unnotched condition. The ability to withstand overstress, however, is reduced by alloy additions which increase the tensile strength. From the scanty evidence available, it appears likely that the ability of a material to absorb overstress, and the extent to which the endurance limit of a material is raised by understress, increases as the tensile strength and hardness decreases. Ferritic cast irons would be expected to have better properties in this respect than pearlitic materials. Cast irons in general have low notch sensitivities in fatigue. In some cast irons, notching has no effect on the endurance limit, while in others, usually with higher tensile strengths, the effect is small compared with steels. Cast iron may be considered as an intrinsically strong material weakened by the presence of graphite flakes serving as notches or stress raisers. A deliberately produced notch cannot seriously affect the already notched material. Though static properties have not been discussed in this paper, it is also found, contrary to what has generally been thought in the past, that flake graphite cast irons are also insensitive to notches in the tensile test. It is thought that cast irons, which have low tensile strengths compared with steels, will frequently behave in fatigue better than expected because of their ability to withstand overstress, the tendency of their endurance ratios to increase with understressing, and their low notch sensitivity. The conclusions reached in this paper are based on results reported in the literature. In many cases, particularly on the subjects of overstressing and understressing, few results are available and the conclusions should be treated with caution.—*G. N. J. Gilbert, Journal of Research and Development, The British Cast Iron Research Association, December 1953; Vol. 5, pp. 94-108.*

Protection of Iron and Steel by Metallic and Non-metallic Coatings

This is the second report on a series of tests on the protection of structural steel by metallic coatings and by non-metallic coatings other than oil paints, which was begun in 1940. Specimens were exposed in the open air at various sites in the U.K. and overseas, where they have been kept under observation for up to 12 years; rather more than half of them are still under test. Other specimens were immersed in the sea and were finally removed after six years. The tests have confirmed earlier conclusions that certain metallic coatings give excellent protection to steel exposed to atmospheric corrosion. Zinc and aluminium coatings are most generally useful for heavy structural steel, but good results may also be obtained from lead coatings in industrial atmospheres. Of the metal coatings tested, zinc is the best for protecting steel immersed in sea-water. Vitreous enamel and thick sprayed coatings of cement-asbestos behaved well in the atmospheric tests.—*J. C. Hudson and J. F. Stanners, Journal of the Iron and Steel Institute, December 1953; Vol. 175, pp. 381-390.*

Light Metals in German Vessel

Light metal was used to a considerable extent for rebuilding the German steamboat *Goethe*. Sundeck and upper deck were built of light metal the joints being welded by the argonarc process or riveted with 5-mm. aluminium rivets. Light metal and steel parts were joined by steel rivets. All light metal parts were thoroughly degreased before painting. The total weight of light metals in the construction is about 17,000 kg. This saves a weight of 30 m.t. compared with an all-steel construction and it diminishes the draught 6 cm. The ship is considerably more stable than older constructions of the same width. This construction shows the largest amount of light metal parts ever used on a German ship.—*H. Harder, Aluminium (Germany), July/August 1953; Vol. 29, pp. 317-319. Abstract Bulletin of Aluminium Laboratories Ltd., November 1953; Vol. 24, p. 651.*

Lubrication of High Speed Ball and Roller Bearings

This paper deals with the lubrication aspects governing the performance of rolling bearings operating in the speed range 6,000 to 30,000 r.p.m. and under radial, thrust and combined loading. Most of the work was carried out on standard ball bearings of 2-inch bore. Three test machines are described. Factors investigated included: effects of variation in oil feed, including reduction of oil feed to zero; a comparison of various methods of supply of lubricant; and the influence of viscosity and boundary properties of the lubricant. In general, the criteria of bearing performance were temperature rise and friction. Bearing life was not considered, except where it was determined by instability of thermal conditions, since, under thermal equilibrium, bearing life is largely dependent on factors other than lubrication. It is concluded that, provided the bearing housings are adequately drained, the rate of oil supply and the method of lubricating high-speed rolling bearings are not critical, and that lubricating oil can be used as a very effective coolant with no appreciable increase in friction. Viscosity of lubricants has little direct effect on performance, but there appears to be a case for good boundary lubricating properties, particularly at elevated temperatures and under combined radial and axial loading.—*A. Fogg and J. S. Webber, Journal of the Institute of Petroleum, November 1953; Vol. 39, pp. 743-764.*

Running Götaverken Engines on Heavy Fuel

It is remarked in the specification to which the diagrams shown in Fig. 3 refer, that with the use of fuel having a viscosity of about 400 seconds Redwood I at 100 deg. F., it is necessary to preheat the oil and this applies to the condition when the engine is brought into operation after a prolonged stop, heated fuel having to be circulated through the pipes, pumps and injection nozzles for some time before starting takes place. The injection-pump plunger, with the arrangement illustrated, can be turned through an angular position so as to

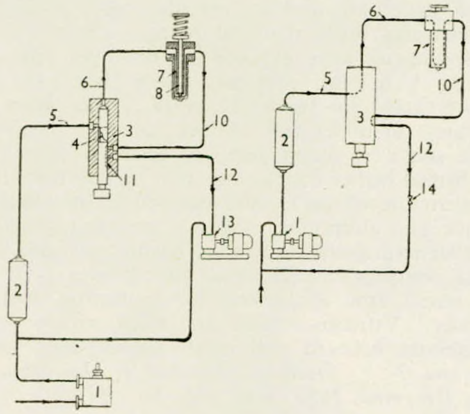


FIG. 3

open a passage for the return flow of oil from the supply, through the injection nozzle, and back into the system when required. Referring to the left-hand diagram, the system includes a supply pump (1) discharging the fuel through a heater (2). The injection pump (3) has a plunger with an oblique edge (4) registering with the inlet from the pipe (5). The discharge pipe (6) from the injection pump communicates with the nozzle (7), which comprises a spring-loaded needle valve (8). When the plunger is turned to the stop position, so that the injection of fuel is cut off, the groove defined by the oblique edge (4) registers with the fuel inlet to the pump. Oil, therefore, passes through the pump body to the nozzle (7), but the pressure is insufficient to open the needle valve (8) so that the fuel passes out through a pipe (10) to the pump (3). In the lower part of the pump plunger there is a groove (11) and the oil is thus enabled to flow through the pipe (12) to a return circulation pump (13). With the alternative system shown in the right-hand diagram, no return circulation pump is embodied and the supply pump (1) has an independent driving motor. A reducing valve (14) is fitted in the pipe (12).—*Patent No. 697,249, A/B Götaverken, Gothenburg. The Motor Ship, December 1953, Vol. 34; p. 397.*

Babbitt Blisters in Thrust Bearing Pads

Babbitt blisters were observed on water-wheel generator-bearing pads. Investigation revealed that these blisters resulted from the accumulation of high pressure hydrogen at the steel-babbitt interface. The hydrogen, it was found, came from the steel and was introduced during the manufacture of the steel. The maximum level of hydrogen content to ensure freedom from blisters was determined. Processes to remove hydrogen from the steel were developed. Investigation and operating experience indicated that such hydrogen-produced babbitt blisters were an important factor affecting the performance of thrust bearings. In the past, such blisters occasionally had been observed on thrust pads that had been held in storage for several years. However, it had never been appreciated that these blisters could result in the failure of thrust bearings. It is now apparent that such failures have occurred in the past, but were never explained because the wiping of the babbitting had obscured the cause of the trouble. As part of the investigation which followed the discovery of blisters, laboratory tests were conducted on steel specimens covered with babbitt. In order to accelerate the process, these specimens were kept at a temperature of 150 deg. C. for a period of time. It was found that blisters were formed rapidly. It was first thought that the hydrogen causing the babbitt blistering resulted from the pickling or fluxing of the pads during the babbitting process. Refinements of the process to eliminate pickling and ensure the complete removal of flux resulted in improvements, but did not prevent the formation of blisters. It was then suggested that the hydrogen causing the blisters could be contained in solution in the steel. This was confirmed by a survey of the literature and additional blistering tests on small samples. It is concluded that hydrogen contained in commercial steels is responsible for the babbitt blisters observed in bearings. A low level of hydrogen content is required to prevent the formation of blisters. This low level is obtained and ensured by suitable heat-treatment and quality control.—*Paper by R. A. Baudry, D. W. Guntler and B. B. Winer, read at the 1953 A.S.M.E. Semi-Annual Meeting. Paper No. 53-SA-23.*

High Temperature Properties of British and American Steels

Some rather substantial differences are found in creep data on similar steels published in various countries. In view of the importance of these differences in relation to design stresses, arrangements were made for an exchange of specimens between The Timken Roller Bearing Co., of Ohio, and The United Steel Companies in order to determine whether the results from the two laboratories, on the same steel, would be in agreement, and to compare the creep resistance of selected steels. Five representative high-temperature steels were chosen comprising three commonly used ferritic steels and two austenitic steels. For each composition, four creep curves were obtained, two by each laboratory. Stress, temperature and time adopted in the tests in the two laboratories were the same and no attempt was made at standardization of test procedure. A high degree of reproducibility was shown by the test results for the two laboratories on the ferritic steels, but those on the austenitic steels showed generally substantial differences. Only two British steels, both of ferritic type, showed similar creep behaviour compared with the corresponding American steels, the remaining three steels showing appreciable differences, which are discussed.—*Paper by W. E. Bardgett and C. L. Clark, read at a General Meeting of the Institution of Mechanical Engineers on 4th December 1953.*

Vibration Tests on Up-river Collier

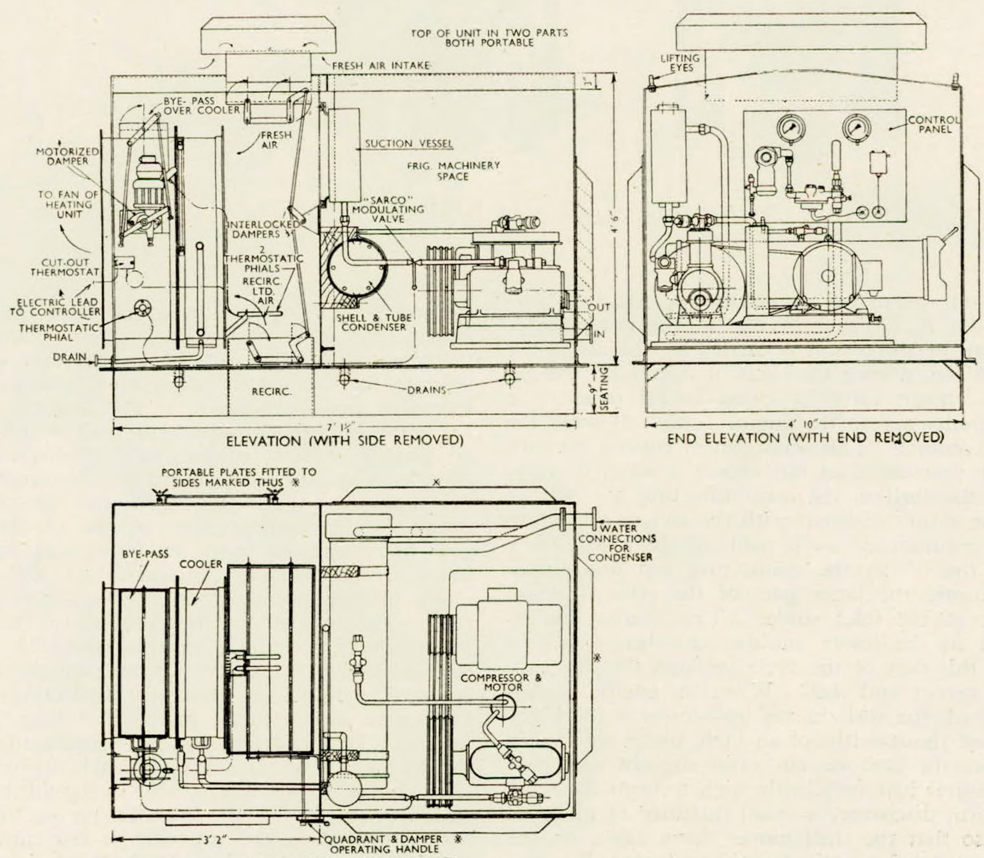
Vibration measurements were made during a routine voyage of an up-river collier. The construction was all-riveted for tank top. The primary object of the measurements was to examine the influence of depth of water upon hull critical frequencies. It is shown that the influence of depth of water upon vertical vibration frequencies may be considered as negligible for h_w/d (depth of water to draught) ratios in excess of 5·0. For values of h_w/d less than 5·0, the frequencies decrease with increasing rapidity as h_w/d diminishes. For transverse modes of vibration the evidence obtained regarding the variation of frequency with depth of water was not so conclusive. It was revealed, however, that horizontal modes of vibration were not influenced to the same extent as the vertical modes. The theoretical treatment of the problem is discussed briefly and attention is drawn in particular to the pubished works of Prohaska and Havelock. Comparisons are made where possible between the experimental results and those of the above-mentioned investigators. Some additional experimental data obtained on a model are given which demonstrate the effect of a plane boundary upon flexural vibration in the vertical and horizontal planes and help to confirm certain conclusions arrived at by Havelock. The hull vibration frequencies in deep water have been calculated by various methods and the results presented in tabular form. Strain measurements made during the loading of the vessel are described, the results being used as a guide in estimating the flexural rigidity of the ship girder. The investigation was undertaken jointly by the staffs of the Ship Division of the National Physical Laboratory, and the British Shipbuilding Research Association.—*Paper by W. J. Marwood and A. J. Johnson, read at a meeting of the North-East Coast Institution of Engineers and Shipbuilders, 11th December 1953.*

Self-contained Air Conditioning Unit

The accompanying illustrations show the specially-designed air conditioning unit for deck mounting, produced by the marine department of the Norris Warming Co., Ltd., Newcastle-upon-Tyne. This can operate in conjunction with existing heating and ventilating installations, or where compartments to house such equipment are not available. The unit is weatherproof and supplied ready for welding to the deck. It contains a compressor, evaporator, condenser, controls, air cooler and reheat bypass, fresh air and recirculating mixing chamber, and thermostatic controls. Only electric power and water have to be connected to the unit. The two sets have ratings sufficient to cover the requirements of existing ventilating plants which may be installed. When required, air filters may be fitted to the air inlet, in which case an inlet tower is directly attached to the set, ensuring that no additional deck space is required. A special deck coaming is provided so that when it is welded in position and should it be required to unship any of the air conditioning equipment at some future date, the bottom flange can be unbolted, allowing the equipment to come away freely.—*The Motor Ship, December 1953, Vol. 34; p. 371.*

Accident

The chief mate of a Company vessel was caught across the face by a mooring wire. His nose and right cheek bone were fractured and deeply cut. He is well on his way to recovery and should be none the worse for the incident. He will, however, have some painful memories to look back on. Here is what happened. The vessel was at the wharf. It became necessary to shift the breastline for use as a spring. The breast was slacked off and the wire fell down around the base of the forward fairlead. An ordinary seaman was told to clear the wire from the lip of the fairlead. The man threw the wire over the fairlead. The wire ran out the chock from its own weight and the bight struck the mate in the face. *Shore Safety Committee Comment:* We see two points here that might be commented on. First, the old one of keeping out of bights. The mate may not have been aware he was in a bight, but he was. Certainly, he was an innocent victim of other people's actions, but then most individuals who get caught in bights fall into the same category. The second point is also an old one, the problem of the transmission of ideas, i.e. the giving of orders. The seaman was a new man; in fact, he had joined the ship, and did not know what was expected of him. He cleared the wire all right. What he



Arrangement plan of a Norris air conditioning unit should have done was to have lifted it clear of the lip and placed it in the fairlead.—*Proceedings of the Merchant Marine Council, United States Coast Guard, October 1953; Vol. 10, p. 14.*

Automatic Compression Ratio Adjustment

The B.I.C.E.R.A. piston automatically adjusts the compression ratio so that a predetermined maximum pressure is not exceeded. The piston, shown in Fig. 3, is composed of two main pieces, the inner portion termed the carrier (A), which is mounted on the gudgeon pin in the conventional manner, and the outer portion (B), called the shell, which slides over the carrier, thereby changing the compression ratio. These two parts are arranged to form an upper chamber (C) and a lower annular chamber (D). These two chambers are kept full of lubricating oil which is supplied to them from the engine lubricating oil system via the connecting rod and non-return valves (E and F). The load due to the gas pressure acting on the crown of the piston is carried by the oil in the upper chamber. When, during the cycle of operation, the gas pressure reaches a chosen value, a spring-loaded relief valve opens and oil is discharged to the engine sump, allowing the upper chamber to contract. This arrangement ensures that the maximum cylinder pressure does not exceed a selected value. In order to move the shell in the other direction so that the highest compression ratio consistent with the chosen maximum pressure is always maintained, use is made of the inertia force of the shell and the oil in the connecting rod and upper chamber, acting during the latter part of the exhaust stroke and the first part of the inlet stroke. This inertia load is carried by the oil in the lower annular chamber, which is forced out during this part of the cycle through the clearance at H, between the carrier and shell. When the engine is running at a steady load, the shell moves up—towards the high-ratio position—a few thousandths of an inch, under the inertia action. This raises the compression ratio slightly and produces cylinder pressures just sufficiently high to open the relief valve. This, in turn, discharges a small quantity of oil from the top chamber, so that the shell moves down again on the carrier, thus the compression ratio remains substantially constant at a value giving the desired cylinder pressure. If now the engine load be raised, there is an increased discharge through the relief valve due to the momentarily increased cylinder pressure and the shell moves down in a few cycles to a new position of equilibrium. If the load is decreased, the cylinder pressures diminish and the relief valve remains closed for a few cycles. The cyclic discharge of oil from the lower chamber continues, raising the compression ratio until cylinder pressures are again sufficiently high to operate the relief valve.—*The Oil Engine and Gas Turbine, November 1953, Vol. 21; pp. 258-259.*

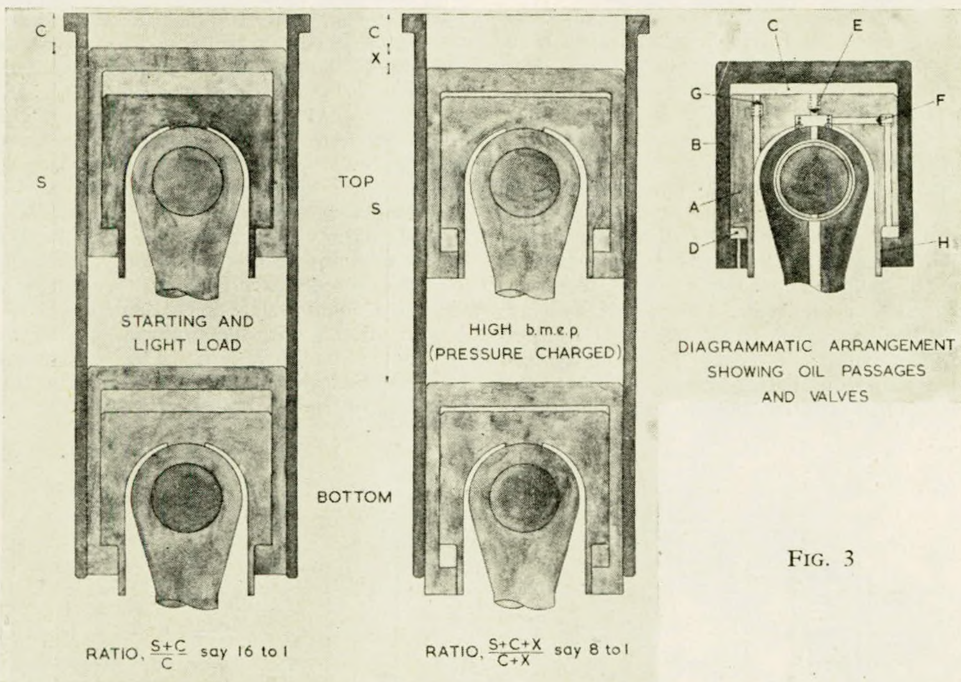


FIG. 3

Experience with Turbocharged Two-stroke Engine

A milestone in the development of the Diesel engine was passed when the 17,000-ton motor tanker *Dorthe Mærsk*—equipped with the first turbocharged two-stroke engine—on 9th October 1952, after the completion of successful sea trials in the Sound, was taken over by the owner A. P. Møller, Copenhagen, and placed in service with the two turbochargers as the only active means for pumping scavenging air. From the trial trip and up to the end of this year the vessel has been at sea for about 9,600 hours, mainly between Western European ports and Sidon or Ras Tanura. The main engine has worked entirely satisfactorily both at full power and during all periods of slow running and manœuvring. The owners' confidence in the new engine is illustrated by the fact that the chain-driven blowers which were retained but never used except on the testbed and on one of the trial trips, have now been removed from the vessel and replaced by a small steam-driven emergency blower which will be employed only in the unlikely event of both turbochargers failing at the same time. The removal of the chain-driven blowers has reduced the weight of the engine plant by about 25 tons, and increased the deadweight capacity correspondingly. The output has averaged about 7,400 i.h.p., corresponding to 6,500 b.h.p. at 112 r.p.m. and with an m.i.p. of 7·2 kg./cm.^{2} (102·5lb. per sq. in.). Designed for a speed corresponding to 5,530 b.h.p. the vessel is not considered capable of utilizing more than 6,500 b.h.p. economically, which should have been obtained with 7·5 kg./cm.^{2} m.i.p. (106·5lb. per sq. in.) and 108 r.p.m., but unfortunately the propeller is not quite suited for these conditions. Compared with the testbed results there is an increase in exhaust temperatures of 10 deg. to 30 deg. C. For the turbocharged engine, the exhaust temperatures follow the scavenging air pressure and temperature (i.e. water temperature and air cooler efficiency) in much the same way as they follow the atmospheric conditions for the non-turbocharged engine. The fuel consumption has been measured frequently during 12- to 24-hours' test runs, and shows a figure of about 134 gr. per i.h.p. hr. with a mean indicated pressure of 7·2 kg. per sq. cm. or 102·2lb. per sq. in. This is equivalent to 152 gr. per b.h.p.-hr. or 0·334lb. per b.h.p.-hr. The mechanical efficiency is 88 per cent. The fuel oil bunkered for the first 5,000 hours has been a light Diesel oil of 0·84 specific gravity with a viscosity of 1·49 deg. E. corresponding to 6 Centistokes at 20 deg. C. On later voyages an oil with 1·79 deg. E. (9 Centistokes or 50 sec. Redwood No. 1) at 20 deg. C. has been bunkered. The pistons have been overhauled after service periods of 2,300 to 6,500 hours and were found in good condition. A few of the narrow upper piston rings were found broken in two pieces of nearly equal size, and on a few rings the tongues in the joints were missing. The average cylinder wear for the four normal cast-iron cylinders was 0·17 mm. per 1,000 hours, and for the two chromium-hardened liners 0·05 mm. per 1,000 hours after 2,300 to 6,500 hours. After about 6,000 and 7,000 hours respectively the turbochargers were dismantled, and the rotors taken out for cleaning and inspection. The dismantling, cleaning, and reassembling of a charger may be done in a normal working day. On the blower side a certain amount of deposit consisting of oil mixed with dust was found. The layer had a maximum thickness of 4 to 6 mm. on the diffuser, but only a thin layer of about 1, perhaps 2 mm., on the blower impellers and on the casing. Apparently, this amount of deposit had not influenced the performance of the chargers as both were running at exactly the same speeds with one charger cleaned and one not cleaned. The turbine sides were practically free from deposits. On the turbine blades some marks were found, indicating that metal pieces had gone through the blading, and in one of the turbine outlet casings a piece of a broken piston ring with dimensions of about 10 by 10 by 18 mm. was found. Apparently, the marks had no influence on the efficiency, speed, or the running of the turbine. It was not thought possible that pieces of broken piston rings could get through the exhaust valve, but apparently this can happen, and precautions have immediately been taken to prevent any damage to the turbines, by building in a grate before the turbine.—*The Motor Ship, January 1954; Vol. 34, pp. 404-405.*

Failure of High Pressure Fuel Pipes and Its Prevention

The breakage of high-pressure fuel pipes has been experienced with many makes of Diesel engines, and has given rise to some investigational work in the past without, however, achieving any fully satisfactory solution to the problem. Failure usually consists of a break, more or less transverse, generally at the nipple; but with some terminations, other types of failure, causing leakage rather than breakage, occur first. Work has been carried out, in the past, on the fatigue strengths of the normal single-taper pipe ends, some other forms of termination, and on pipes made of different materials. Some improvement of fatigue strength was found possible, but uncontrolled trials in service did not appear to show any considerable improvement in service life. While C.A.V., Ltd., of London, have supplied fuel-injection equipment for a very wide range of engines, the high-pressure fuel pipes have been supplied, generally, by the engine-builder. However, it seemed logical for the company to investigate the whole problem, in an endeavour to find a means of eliminating pipe failures; accordingly, a programme of work was initiated. A review of service practice and experience was first made, to discover if there was any fully satisfactory solution in existence. Concurrently, a laboratory investigation was made, the primary cause of failure established, and a number of possible cures examined. As a result, a solution was devised, which appears to be entirely satisfactory and generally applicable. Whatever else is done, it is of the first importance that the mounting of the pump on the engine should be as rigid as possible, since, other things being equal, pipe stresses are proportional to the amplitude

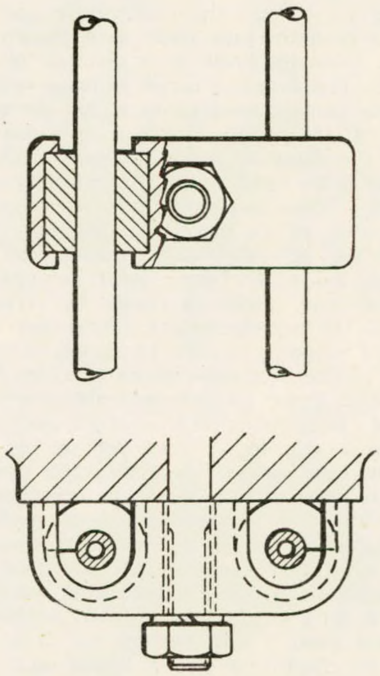


FIG. 3—*Diagrammatic arrangement of C.A.V. synthetic-rubber damper*

of vibration of pump with respect to engine. In practice, however, it would appear to be seldom, if ever, possible to achieve so rigid a system that pipe vibrations do not occur. The possible means of reducing failures are:—(1) By increasing the fatigue limit of the pipe material and/or the termination. (2) By increasing the damping capacity of the pipe material. (3) By imposing restraint on the pipe, e.g. by clamping. The expected fatigue limit of the normal mild-steel pipe itself is

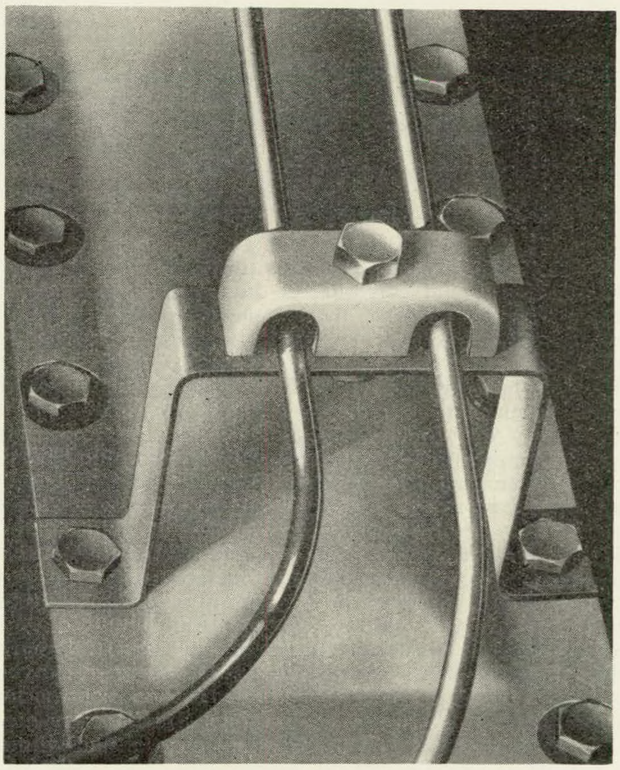


FIG. 4—*Double-pipe damper*

12 tons per sq. in. Thus, the double-cone end is, in fact, slightly stronger than the pipe itself, as is shown by the fact that such ends generally break at a distance of about ½-in. from the nipple. The damping factor increases with stress and with the amount of cold working to which the test-piece has been subjected. If the piece is run at a stress near the fatigue limit, however, the damping factor decreases with time. The relevant quantity is the stable value after running at just below the fatigue limit. Measurements under this condition have been made for most of the promising materials. In order to avoid failure, it was generally found necessary to operate at a stress appreciably below the fatigue limit, probably because of occasional increases of amplitude caused by variations of the mains supply to the test equipment. Attention was given to the reduction of working stresses by means of dampers, the effect of which had been clearly demonstrated by merely laying a finger on a pipe. Strain-gauge measurements were made with various types of clamp on an engine which gave pipe stresses, without clamps, of up to 11 tons per sq. in. It will be appreciated that, in a system where the high stresses are due to resonance, the stresses found may be capricious from one occasion to another. It is therefore necessary that, for any device to be regarded as offering a cure for the breakage trouble, it must give stresses very substantially less than the fatigue limit. The target set in these experiments was one-half the fatigue limit, i.e. with single-cone ends on mild-steel pipes, a stress not greater than 5 tons per sq. in. The arrangement most likely to be effective is one in which each pipe is connected to the engine structure by some material capable of absorbing vibrational energy. It was found that attaching the mid-point of each pipe to the engine by means of a clamp lined with sponge rubber was very effective, provided the clamp was not fully tightened. With the sponge-rubber lined clamp fully tightened, however, or with rigid (steel) clamps to the engine, the particular resonance giving the highest stress, with pipe unclamped, disappeared, but resonances appeared at other engine speeds which were generally almost as serious as the original one. A similar result was obtained by clamping the pipes together, in pairs, with either steel, wood-lined steel, or even sponge-rubber lined clamps not fully tightened. It was concluded that, of these palliatives, the most effective was clamping to the engine, with resilient, energy-absorbing material interposed. The C.A.V. synthetic-rubber damper, illustrated in Figs. 3 and 4, was then designed. With this arrangement, the pipe is supported in a rubber bush, the outside surfaces being held in a metal housing which does not compress the rubber.—*The Shipbuilder and Marine Engine-Builder, December 1953; Vol. 60, pp. 677-680.*

Vanadium Problem in Gas Turbines

The effect of V_2O_5 and mixtures V_2O_5 and Na_2SO_4 on turbine fouling and corrosion is discussed. Laboratory tests have shown that none of the commercially available alloys is immune from attack when these mixtures are molten, that is, at temperatures above 650 deg. C. Nickel base heat resisting alloys of the Nimonic type offer higher resistance to attack than the austenitic steels and some degree of protection is afforded by electro-deposited coatings of chromium. Steels and heat resisting alloys are not attacked to any significant extent at temperatures below the melting point of the ash, but copper base alloys suffer considerable corrosion at temperatures as low as 500 deg. C. A detailed investigation of the effect of additives on the corrosion rate of V_2O_5-Na_2SO_4 ashes indicates that corrosion can be considerably reduced, and that such a method of approach is promising. Suitable additions may be made by a mixture of oil or water soluble materials with the fuel, by suspension in the fuel, or by separate injection into the combustion chamber with the object of reducing corrosion and deposition. Laboratory corrosion tests, using ash additives such as ZnO, Al_2O_3, and kieselguhr, have shown that the rate of attack can be greatly accelerated if the additive is present in certain critical proportions, but when this is exceeded, corrosion is considerably reduced. With MgO, however, no such increase in attack was noted. Rig tests have confirmed some of the beneficial effect of additives.—*Paper by S. H. Frederick and T. F. Eden, submitted to the Institution of Mechanical Engineers for written discussion, 1953.*

Alkaline Batteries for Marine Purposes

Alkaline batteries are in wide and growing use for a large variety of marine purposes, and may range in size from relatively small-capacity and low-voltage batteries—for service as engine-room emergency lighting and ships' telephones—to batteries of large capacity and high voltage—for such purposes as emergency steering. The new regulations governing the safety of life at sea call for additional safety measures, which are most satisfactorily met by the general characteristics of alkaline batteries, and these regulations will undoubtedly give rise to still wider adoption of batteries of this type. As alkaline batteries are relatively expensive items of equipment, and as they are installed by shipowners in order to obtain a long and trouble-free life with the minimum of maintenance, it is essential that certain points should be watched at the installation stage, if the maximum possible economies are to be reaped. The first point is to ensure that the proposed battery is of the correct type technically, and that it is adequate in capacity for the intended duty. With regard to technical type, it should be remembered that there are two basic types of alkaline accumulator, viz. tubular positive nickel-iron and the flat-plate nickel-cadmium, and that each has specific characteristics which give it particular advantages for certain varieties of duty. The tubular iron battery is more expensive than the flat-plate cadmium type, and finds its best application where regular full charges and discharges are called for, and particularly at relatively high temperatures. Regular cycling is a condition not often encountered in liners or cargo vessels, and hence the main marine application of the tubular iron battery has been in the larger private yachts, where regular cycling and persistent deep discharging form the general condition; wherever this condition is met, the tubular iron type of battery should be selected. The flat-plate cadmium battery is the type most generally met with in all commercial marine work; but it is important to appreciate that there are two variations of the basic type, and the selection of the right one is essential to the obtaining of optimum results. One of these variations is a normal resistance type, and should be selected wherever the load emphasis is on moderate currents for considerable periods of time. The other variation is a low-resistance type, and should be selected wherever the load emphasis is on peak currents for short periods of time. Batteries are expensive units, provided to fulfil vital functions; yet it is surprising how often they appear to be overlooked, until everything else has been installed, and they are then crammed into some totally unsuitable quarter, with every condition to produce the shortest possible life. As actual examples met in practice, one may cite a battery fitted against an engine-room bulkhead, where the temperature was so high that the battery needed daily "topping up", which, of course, was not done, and so the battery was ruined. Another battery was installed in a shallow steel tank without drainage, so that, as a result of careless topping-up, water ultimately reached the cell cases and corroded away the bottoms. Yet another battery was installed with so little headroom that the top tier of cells could not be topped-up at all, while a further unit was mounted in the direct flow from a fan blowing salt-laden air straight on to the cell tops. Should it not be possible to provide the most desirable condition of a separate compartment for the battery and control gear—and this may well be impossible in smaller ships—then the foregoing general principles must still be borne in mind, viz. adequate ventilation, adequate drainage, and accessibility for inspection and topping-up. If the place of installation has to be the engine-room, then the coolest and best-ventilated space must be selected, and particular care should be taken to avoid direct heat from exhaust pipes or other hot spots. The ideal working temperature for an alkaline battery is about 70 deg. F., and the maximum safe operating temperature is 115 deg. F. The latter figure is an upper limit which should only be reached for short periods under peak conditions, and continuous operation at or near the maximum temperature may result in reduced battery life.—*G. Scott Atkinson, The Shipbuilder and Marine Engine-Builder, December 1953; Vol. 60, pp. 689-691.*