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Problems in Marine Diesel Design

The most compact types of multi-cylinder engines are shown in outline in Fig. 1, where the upper diagram represents the polygonal opposed piston type cylinder arrangement, while the lower left-hand diagram shows the multi-row radial type, and the lower right-hand diagram depicts the multi-cylinder swash-plate transmission type. Owing to the complications involved in the swash-plate drive, only the polygonal and the multi-row radial types are considered by the author. In order to obtain comparable data, designs were prepared of a 24-cylinder two-stroke cycle three-row radial Diesel engine of 650 h.p. at 1,200 r.p.m. and an 18-cylinder two-stroke cycle polygonal engine having three hexagon rows with 800 h.p. output. Both designs were based on identical cylinder diameters, and the stroke of each of the two pistons in each cylinder of the polygonal design was made one-half the stroke of the radial design. Because of the smaller stroke of the polygonal design this can be operated at higher speed; also its m.e.p. may be made 5·5 kg. per sq. cm. as compared with 4·5 kg. per sq. cm. for the radial type. Even after deduction of the higher transmission losses in the polygonal design, this has a specific output per cylinder volume which is 65 per cent larger than that of the radial engine. The specific power developed by the radial engine is 183 h.p. per cu. m. of space occupied, and the corresponding figure for the polygonal design is 196 h.p. per cu. m. While in the case of the radial design the generator is placed below the cylinder rows, the polygonal design makes it possible to place the generator within the cylinder polygons. The polygonal engine is therefore considered to provide the most compact engine design available.—*A. Jante, Schiffbautechnik, February 1954; Vol. 4, pp. 53-58.*

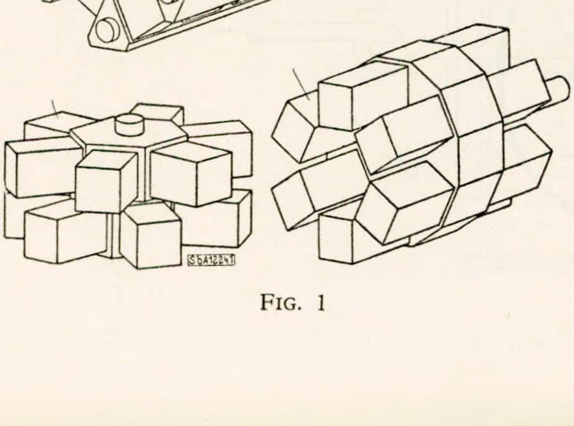


FIG. 1

Peening Last Pass of Welds

Peening of welds to control distortion is essential to the solution of many fabrication problems. The various codes which govern welding procedures permit such peening with the exception of the last pass. The restriction is based on the general opinion of the various code authorities that the cold work associated with peening is detrimental to weld ductility. In the case of the intermediate passes the damage is considered to be eliminated by the heating effects of the subsequent pass and accordingly is permitted. While the restriction to peening the last pass has been in effect for many years there has been considerable contention by fabricators, desirous of using peening for the last pass, that the regulation is both unnecessary and arbitrarily restrictive. This contention is based on the fact that data showing positive damage to the performance of weldments due to peening of the last pass were not available. Thus, the subject and problem of peening last passes has remained a controversial one. In order to resolve this question the Welding Research Council, through the Peening Committee, sponsored an extensive investigation concerned with the evaluation of the notch ductility of as-deposited and peened weld metal. The results of this investigation which was conducted at the American Bureau of Shipping Laboratory, have been made available in a previous report, where it was demonstrated by modified Charpy tests that the effects of peening were detrimental to weld notch ductility. The development of the Explosion Bulge Test for the evaluation of full-size weldments provided a means for direct comparison of the performance of peened and as-deposited weldments. Inasmuch as the problem of peening last passes is of interest to the (U.S.) Navy, the tests were conducted at the Naval Research Laboratory in co-operation with the research programme of the Welding Research Council. The American Bureau of Shipping provided the weldments. The weldments used in this investigation were prepared from 1-in. thick ABS-B ship plate (0·18 per cent C, 0·75 per cent Mn, 0·03 per cent Si, 0·033 per cent S, 0·013 per cent P). Two 12-in. × 24-in. plates were joined by a 60-deg. VV butt weld to form a 24-in. × 24-in. weldment. The weldments were prepared by a commercial fabricator under the direction of the American Bureau of Shipping. Peening was performed by the American Bureau of Shipping using an automatic machine which provided close control and reproducibility of the peening conditions. The explosion tests were made according to the standard practice developed for weldment evaluation. The test results showed that heavily peened weld is a highly efficient crack starter and will serve to introduce a profusion of cracklike notches into the weldment. Whether or not the presence of such severe notches results in the collapse of the weldment is determined by the properties of the base steel. Unfortunately, present structural steels are highly sensitive to the presence of notch defects, particularly at temperatures of "cold" service (below 60 deg. F.). Thus, peening of the last layer of weld metal should be considered a dangerous practice which greatly increases the hazard of the initiation of brittle fractures in the base steel.—*W. S. Pellini and E. W. Eschbacher, The Welding Journal, February 1954; Vol. 33, pp. 71s-76s.*

Air Conditioning Afloat

In the search for the most suitable type of refrigerating machinery for ship air conditioning plant, an absorption system as used for commercial air conditioning appeared to be the answer due to its lack of moving parts. A marine type lithium bromide plant was developed, tested and installed on the U.S.S. *Northampton* as an experimental installation. The lithium bromide cycle, shown schematically in Fig. 4, uses water as the refrigerant and lithium bromide solution as the absorbent. The water to be chilled is sprayed into the evaporator chamber which is maintained at a low absolute pressure approximating the vapour pressure corresponding to the desired chilled water temperature. A portion of the sprayed water flashes, cooling the remainder which drains to the chilled water pump for circulation through the cooling coils. The flashed vapour is absorbed by the sprayed lithium bromide solution which in turn is partially delivered to the generator where the vapour is boiled off by a steam heat exchanger. This vapour is then condensed on sea water coils and returned as water to the water side of the system. The strong solution returns to a heat exchanger and from the heat exchanger returns to the evaporator. The installation on the *Northampton* has a capacity of 100 tons and supplements 5-50 ton freon high-speed reciprocating compressors which operate on water chiller systems. The evaluation of this system is now under way and at this writing conclusions cannot be made. The U.S. Navy's biggest shipboard air conditioning installation planned so far will be installed on the aircraft carrier *Forrestal*. Every berthing and messing space, all vital manned control spaces (except machinery spaces), the

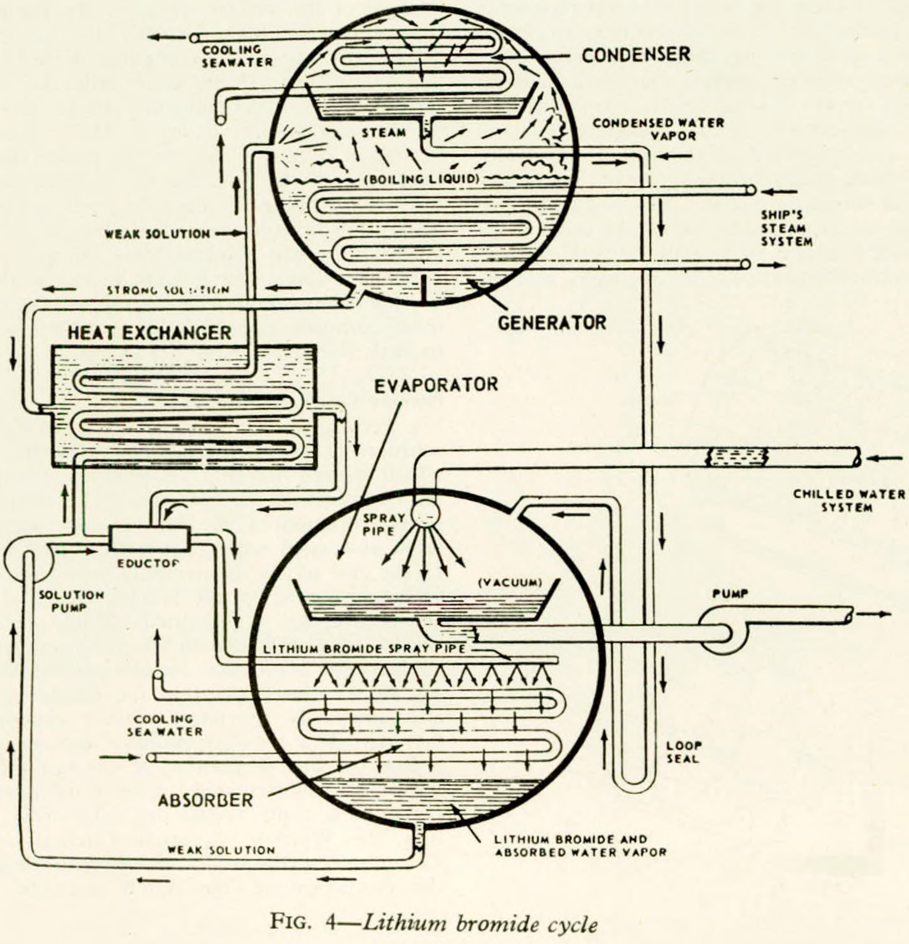


FIG. 4—*Lithium bromide cycle*

hospital spaces, every office, and many ammunition spaces will be air conditioned. In the machinery spaces, air conditioned refuge cubicles will be installed for use by the engineers in the event of smoke. Certain "open" bridges in the island must be glass-enclosed to reduce noise from jet planes; these bridges will be air conditioned and electronic coil controls, selected to compensate for the effect of solar radiation, will be used to maintain satisfactory conditions. About 400 fans, 300 cooling coils, 400 duct type heaters, and 130 convectors will be required in the ventilation, heating, and air conditioning systems on *Forrestal*. When contract plans and calculations were completed the connected coil tonnage was in excess of 1,100 tons. The coils, using chilled water as a refrigerant, will be served from seven air conditioning machinery plants. Each plant will consist of a 150-ton centrifugal compressor with its associated condenser and freon to water chiller. Each centrifugal is a two-stage compressor operating at approximately 7,000 r.p.m. through a speed increasing gear from electric motor drive. The plant is designed for use with freon-11 refrigerant. This type of compressor is new for naval air conditioning use. Similar but larger units have been operating successfully on the AF48 Class for refrigeration application.—*A. S. Panella and E. H. Honegger, Journal of the American Society of Naval Engineers, February 1954; Vol. 66, pp. 63-75.*

Nuclear Power for Submarine

The two submarine nuclear power plants now being built have been designated: STR—Submarine Thermal Reactor, water coolant; and SIR—Submarine Intermediate Reactor, sodium coolant. The thermal reactor effects most of the fissions with neutrons at slow or thermal speeds; the intermediate at intermediate speeds. In both types of reactors, the primary coolant—water or sodium—is pumped through the reactor and boiler in series to transfer the heat to the steam. The steam drives the main turbine and the auxiliaries necessary to run the plant. The region outside the reactor is protected from the neutrons in the reactor. Neutrons tend to make radio-active anything they strike. Accordingly, the reactor itself is surrounded by a primary shield to stop the neutrons. Light elements, such as the hydrogen in water, are effective for this purpose when combined with nuclear poisons. The coolant becomes gamma radioactive when it passes through the reactor. To protect personnel from the dangerous gamma rays, a shield, preferably of high density, surrounds all components that contain the primary coolant. The problems associated with both power plants are similar, and for general unclassified discussion they can be treated together. For both STR and SIR, the power plant designers have relied directly on the Electric Boat Division of the General Dynamics Corporation for the solution of the ship problem. In designing a nuclear power plant for a submarine (Fig. 1) four factors dominate all design considerations within the secondary shield: 1. Because the secondary shield is thick and heavy, it must be kept as small as possible, and the components within it must be compact and closely packed. 2. The equipment within must be located for easy servicing. 3. For all practical purposes, a reactor, once in operation, never stops producing heat. (Actually, its rated output falls off rapidly at first, then slowly over a long period of time.) Therefore, adequate cooling must be provided at all times, even in the event of a power failure. 4. The systems can be designed as hermetically sealed units that will not require any additional coolant to be added during a voyage. Or they can be designed for very low leakage—small amounts of coolant would then be added during the voyage. Because the primary coolant becomes radioactive, and also to avoid pumping excessive leakages back into the system, it is important that the valves do not permit leakage from the system. If water is used as a coolant, it involves high-pressure valves with sealed-in electric actuators, or piston-operated valves with the necessary auxiliary systems. It is easy enough to write the specifications, but it is difficult to design and develop reliable valves and equipment for this type of service. If liquid metal is the coolant, the valves must be sealed against lower pressures. Even so, the hazards associated with a small leak of sodium are probably greater than with water. In addition to the various paths of development open for water valves, a bellows-sealed stem appears practicable at the lower pressures. Again, the problem is to assure reliability. The boilers for nuclear power plants using water as a coolant have an inherently low temperature drop for heat transfer. The average temperature of the water must be sufficiently below the boiling point at the pressure used to avoid local boiling in the reactor. This in turn leaves little temperature margin for the boiler when even moderate pressures are used in the steam system. The end result is boilers of large surface area that must be shielded by the secondary shield to protect the crew from the induced radioactivity in the coolant. With sodium as a coolant, much larger temperature drops are available (boiling point of sodium at one atmosphere is 1,621 deg. F.). But using such high drops and temperatures brings up the serious problems of thermal dislocations and shock. In addition, because of the violent chemical reaction between sodium and water, it is necessary to provide a double barrier between the two fluids. This further hampers the designer because it requires construction of devices that are unusual and that tend to be awkward to manufacture and maintain.—*W. W. Kuyper, Journal of the American Society of Naval Engineers, February 1954; Vol. 66, pp. 211-214.*

Gas Turbine on U.S. Naval Ships

The gas turbine offers a number of outstanding features that make it well suited for driving auxiliary units. As a unit for intermittent operation, the gas turbine is generally a highly competitive prime mover even when matched against the gasoline or Diesel engine. For intermittent operation, over short periods of time, such as would be encountered in emergency generator sets or certain pumps, the relatively high specific fuel consumption is of little concern. The advantages of light weight, dependability, and easy starting may more than offset the feature of high fuel consumption. With the continuing demands to make more efficient use of available space and to pack more power into a given volume, investigations are continuing to find new applications for the gas turbine engine. Numerous uses such as drives for forced draught blowers, emergency generators, pumps, ship service generators and others

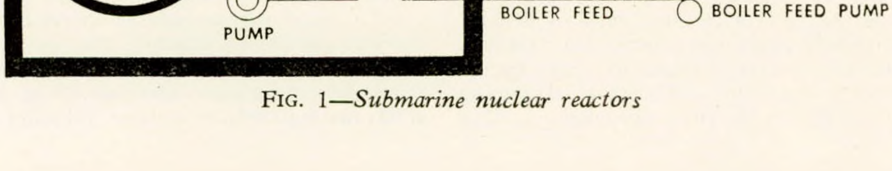
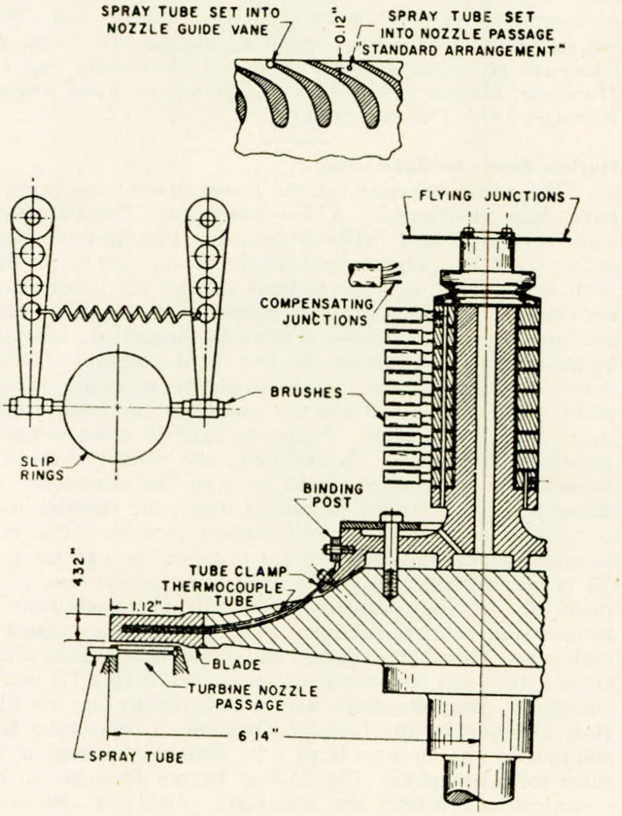


FIG. 1—*Submarine nuclear reactors*

have been or are under consideration. The auxiliary power applications that are being developed by the Bureau of Ships for the gas turbine include the following engines: Solar T-45, Boeing 502-6, Solar T-400 (completed), Solar T-520. With regard to Solar T-45, this engine has been given extensive laboratory tests by the Bureau of Ships. Light weight and ease of starting have been extremely attractive for use as an emergency water pump. Tests have been conducted to determine performance under controlled conditions with temperature as low as 60 deg. F. below zero. The unit as a piece of fire-fighting equipment has been operated under "field" conditions by the U.S. Naval Fire Fighting School at the Philadelphia Naval Base. As a result of the work done at the Bureau of Ships' laboratories and at the Fire Fighter School, some further modifications appear desirable. At the present, a programme is under way to develop a flexible exhaust hose that will permit operators to use the unit in a compartment and direct the exhaust gases to a location well removed from the operating area. This hose should be able to withstand exhaust temperatures up to 900 deg. F. continuously and 1,200 deg. F. momentarily. It should also be of such design as to permit collapsing for compact storage. Seven of the pump units have been delivered to the Bureau of Ships. It is planned to have two of these evaluated by various Naval Fire Fighter Schools. Two may be utilized for investigating the possibilities of a unique application in removing ice from ships' structures that operate in extreme cold. The exhaust from the turbine will be directed against ice coated structures and the pump discharge will be used to wash down slush resulting from the melting ice. The remaining units will be tested for other applications under shipboard conditions. Generators on the minesweeping boat MSB5 class are driven by the Boeing 502-6 gas turbine. The Boeing 502-6 engine was selected for this application due to the stringent requirements for weight and size. Ventilation of the generator compartment presented a major problem which was solved by developing a stack eductor. In this way, ventilating fans and motors were eliminated and a saving in electrical load and machinery weight was effected. Airborne noise from the compressor was reduced to an acceptable level by treating the air inlet chamber with a low density fibrous glass blanket. The exhaust noise was reduced by installing a silencer, designed and tested by the U.S. Naval Engineering Experiment Station. Operating experience on these units has not revealed any machinery difficulties other than those of a very minor nature. Some consideration is being given to use of this engine for removal of ice from machinery and structures on vessels operating in extreme cold. It is planned to modify two engines by removing the power turbines and the output reduction gear and utilize the compressor and connected turbine as a heating unit. Exhaust gases from the set would be directed by flexible hose to iced-over areas. Work is now under way on this project and tests may be conducted during 1954. With respect to the Solar T-400, the first application of a gas turbine engine for an emergency electric generator drive was made by the U.S. Navy. The Solar T-400 was installed to drive a 250-kW generator set in the destroyer *EDD828*. The engine is presently operating. Some difficulties in associated controls and starting equipment have been encountered. As to the Solar T-520, this engine is being produced for use as a drive for an auxiliary electric generator. Another gas turbine propulsion engine which the Navy has under development is the Allison 510-B1. This engine was originally conceived as a modification to the T-38 turbo-prop engine. By splitting the turbine between the second and third stages, it was found that the required performance and flexibility for marine application could be obtained. In its marine form, the engine consists of a twelve-stage axial flow compressor rotating at a speed of 13,800 r.p.m., air throughput of 28·8lb. per sec. and a pressure ratio of 6·41:1. The air leaving the compressor enters the combustion chambers at 533 deg. F. where the temperature is raised to 1,500 deg. F. The power turbine rotates at a nominal 7,500 r.p.m. developing 2,000 horsepower (based on 80 deg. F. air to compressor). The exhaust gas temperature is about 890 deg. F.—*W. M. M. Fowden, R. R. Peterson, and J. W. Sawyer, Journal of the American Society of Naval Engineers, February 1954; Vol. 66, pp. 109-124*

Novel Cooling Method for Turbines

The paper discusses the cooling of turbine blades by direct water spray. This method was first studied at the Westinghouse Research Laboratories in 1947. The research originally was done under contract with the U.S. Navy, Bureau of Ships, and was classified until 1952. Work along similar lines has been conducted by the British National Gas Turbine Establishment since 1950. The desirability of high inlet temperatures for gas turbines, as a means of increasing both the efficiency and the output per pound of working fluid, is well known. Blade cooling has appeared to be an attractive means of operating turbines at gas temperatures which would otherwise be excessive for the blade materials. Water (or possibly some other liquid coolant)

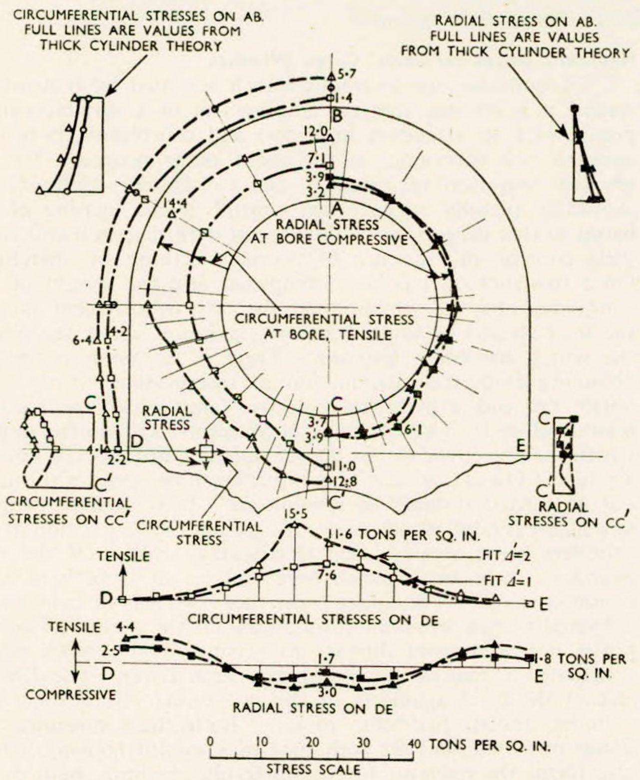


Detail of thermocouple and spray-tube installation.

is sprayed directly on to the rotor blades of a gas turbine from orifices in tubes located near the trailing edge of the turbine-nozzle guide vanes. The liquid impinges on the rotor blades forming a layer which serves both to insulate the blade from the hot-gas stream and to extract heat from the blade. Results of experiments on a supercharger, with gas temperatures between 1,150 and 2,350 deg. F., and theoretical studies of the effect of spray cooling on cycle efficiency indicate that spray cooling, by permitting higher gas temperatures, should increase both output and efficiency of a gas turbine power plant. Heat transfer in a spray-cooled blade is discussed.—*E. Burke and G. A. Kemeny, A.S.M.E. 1953 Annual Meeting; Paper No. 53-A-180.*

Shrink Fit Investigations

In large marine engine crankshafts the crankpins and journals are shrink fitted in the webs, and the torque which the shrink fit can transmit depends on a variety of factors. The interference fit allowed must set up an adequate radial pressure at the mating surfaces without inducing excessive stresses in the web. For a given interface pressure the torsional strength of the grip will vary with the coefficient of friction which, in turn, depends on the degree of lubrication and surface finish of the mating surfaces. In view of the scarcity of data on these effects, the British Shipbuilding Research Association instituted a research investigation on the strength of shrink fits and the results obtained to date are presented in the paper. New, and also used, cranks with a 29½ inch throw and pins 21½-inches in diameter were made available for experiment. Since the range of variables which could be studied on this scale was limited, ancillary studies were made of the behaviour of small shrink-fitted rings and plugs and of model crank-webs. These small-scale tests covered the main variables of mating surface conditions and fit allowance. A special preliminary experiment was also carried out on the measurement of bore strains in a thick cylinder carrying an internal fluid pressure. Five tests on the full-scale webs are reported separately. These include two web slices which were annealed and rebored to allow pre-service stress conditions to be obtained. The fit allowances are limited and vary between 1·0 and 2·0 mils per inch diameter. By means of electrical resistance strain gauges patterned over the surface of the web, the relief of stress was measured when the pins were cut out and again when the web was cut through. The variations of radial and cricumferential stress are given and compared for different fits with certain theoretical considerations. The results from these pre-service webs are shown in Fig. 1. Dismantling of selected "after-service" webs by axial



Fit of Δ′ = 1; circumferential □ radial ■.

Fit of Δ′ = 2; circumferential Δ radial ▲.

FIG. 1—*Comparison of stress patterns for fit allowances* Δ′ = 1 and 2

force up to 2,000 tons provides information on coefficients of friction and loss of fit due to plastic flow. In pre-service webs having a fit allowance of 2·0 mils per inch diameter, the loss of fit is over 30 per cent. In post-service webs the average loss is 40 per cent. The main findings from the experimental work are summarized and reviewed. Experimental results are found to agree with theory in all cases where the ring or web was stressed within the elastic range. Where overstraining has occurred, however, experiment and available theory are, in general, not in agreement; with rings, unexpectedly high permanent strains are revealed. The stress patterns obtained for overstrained webs are complex; yielding in the bridge piece between the pins results in uneven radial bore pressures which in turn are low relative to the bore circumferential stresses; loss of fit is again excessive. There are definite indications that one explanation of the divergence is the inadequacy of overstrained-thick-cylinder theory as at present established. In the large webs a further complication is provided by the evidence of an additional stress system due, presumably, to uneven flame heating of the webs in preparation for shrinking. The investigation has been confined to the static effects of shrink fitting, but evidence of bellmouthing in the bores of the used crank-webs indicates that the dynamic effects of service stresses also require examination. It is considered that one cause of this bellmouthing is the excessive fit allowances used in current marine engineering practice.—*Paper by A. S. T. Thomson, A. W. Scott, and C. M. Moir, read at a meeting of the Institution of Mechanical Engineers, 2nd April 1954.*

Model and Ship Trials of River Vessel

In order to judge accurately the resistance and propulsion qualities of a river vessel by means of a model test, it is necessary, first of all, to make a study of the allowances required to bring trial-trip and service results into agreement with the results of the model test. The allowances on the model-test results of river vessels are of another kind than those of sea-going veessels. On the one hand, there will be no resistance increase in the case of river vessels due to sea-way, and on the other hand, owing to the flow phenomena of the river, acceleration and deceleration forces will cause extra resistance. Moreover, these non-stationary phenomena present an extra difficulty in the correlation of the model-test results and the results of the actual ship. A satisfactory reproduction in a flow tank of the flow phenomena of a river will in many cases be hampered by great difficulties. Besides this, the definition and the determination of the velocity of the model in a flow tank form a problem in model investigations of river ships. The Netherlands Ship Model Basin thought it advisable, therefore, when investigating the service and trial-trip allowances of river ships to start with model tests in undisturbed water. The tests have shown that it is impossible to determine the speed of the ship by direct measurements in a restricted depth of water on the river. This ship is constantly in accelerated or decelerated motion as a result of the irregular flow-phenomena of the river. This circumstance does not simplify the determination of the allowances necessary to bring the service results into good agreement with results of the model tests. In this paper a method is given to determine these allowances, taking as starting point the results of the model test and the revolution-power relation measured on the ship. In model tests the influence of the tank-width increases as the depth of water decreases. The fact, however, that the resistance curves for one depth of water and varying tank widths, when plotted against the Boussinesq number coincide, provides a simple yet valuable basis for the elimination of the wall-effect in model tests.—*Paper by W. P. A. van Lammeren and J. D. van Manen, read at a meeting of the North East Coast Institution of Engineers and Shipbuilders, 26th February 1954.*

Hydraulic Deck Machinery

In this paper the author describes certain types of hydraulic deck machinery for a number of marine applications. Referring to anchor windlasses the author points out that the type and size of a windlass required for handling the anchors is based on the size of anchor cable which, in turn, is usually specified by the shipbuilder, and is calculated from a numeral based on the length, beam, and draught of the vessel, in rules laid down by one of the classification societies. The haulage effort at the cable lifters on the windlass is usually calculated from the weights of two anchors and 60 fathoms of cable, so that the

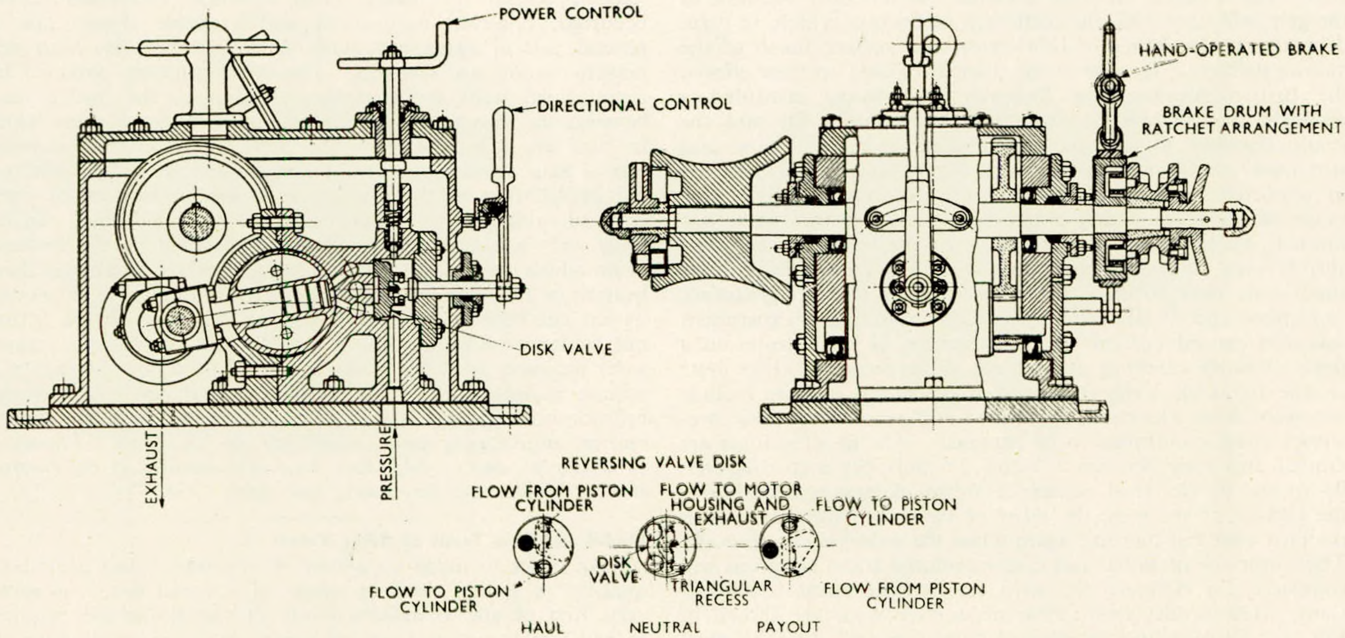


FIG. 7—*Hydraulic anchor windlass. Small-diameter cable*

windlass will be capable of hauling-in both anchors simultaneously, each with 30 fathoms of cable. For example, a windlass suitable for 1⅜-inch stud-link of wrought-iron cable would need to have a haulage effort of 92 cwt. for 60 fathoms of cable, plus 71 cwt. for the two anchors, giving a total required effort of 163 cwt. The speed of haulage is usually in the region of 30-40ft. per min. unless special speeds are called for. Hydraulic fluid is fed from a constant-delivery pump by way of a directional-control valve to the hydraulic motor, which in this case is a three-cylinder oscillating-type motor developing 15 b.h.p. at a speed of 271 r.p.m. Windlasses for hauling larger sizes of anchor cable are fitted with the piston-valve-type hydraulic motor. The total gear ratio of this particular windlass is 32·3 : 1, giving the main shaft on which the cable lifters are mounted a speed of 8·4 r.p.m. Each cable lifter has five pockets to take the cable, so that ten links of cable are hauled in in one revolution, this being equivalent to 4·17 feet of cable, giving a haulage speed of 35 feet per minute. Incorporated in the gearbox are two mechanical clutches, which give independent operation of each cable lifter, and which also allow operation of the two warping drums when the cable lifters are stationary. The general procedure when dropping the anchors is for the directional-control valve to be placed in neutral position, the two hand-operated brakes, of the screw type, which are fitted to a brake drum on each cable lifter, is applied and then the clutches are released. Each anchor is usually dropped separately, and checked by means of the brake. To haul-in the anchors the clutches are engaged, the brakes released, and the directional-control valve lever is moved into the hauling position. In certain installations when cargo winches have not been conveniently placed, the warping drums of the windlass have been used for handling the hatch covers of the cargo holds. When this operation of the windlass is required, a hydraulically-operated automatic brake has been arranged in addition to the hand-brakes on the cable lifters. The coupling between the hydraulic motor and the gearbox serves as the brake drum and does not prevent the windlass being operated in the normal way. The addition of this automatic brake does enable the warping drums to lower the hatch covers smoothly, as with standard cargo winches. Hydraulic windlasses are also used on small types of craft. An illustration of one of these small windlasses is given in Fig. 7.—*Paper by J. R. Fairs, read at the Conference on Hydraulic Mechanisms, The Institution of Mechanical Engineers, 26th March 1954.*

Hydraulic Drives for Ships' Cargo Winches

To consider one hydraulic winch actuated by hydrostatic means, it is obvious that the employment of a variable-output pump, with its attendant high cost and complexity, is neither desirable nor economic, as the speed range necessary for the efficient operation of cargo is not too closely confined. It should be possible to have fine control of the inching of the barrel so that cargo may be broken out with despatch and complete control and absence of excessive jerking or snatching. Once this inching has been completed and the weight of the sling taken by the winch, then the next requirement is that the load should be hoisted at the maximum speed for which the winch has been designed. There is no point or benefit obtaining from the introduction of intermediate speeds. To obtain this end, a hydrostatic electric cargo winch has recently been developed. This comprises a constant-speed electric motor, which may be either a.c. or d.c. or, in some instances, a suitable air-cooled Diesel engine. This input machine operates throughout at a fixed number of revolutions. It is directly coupled to a nine-cylinder, radial, hydraulic pump, the disposition of the cylinders being about a central eccentric shaft. Of the nine cylinders, three are of small bore and six of large bore, with common stroke. The outlet from these two sets of cylinders is delivered to two separate outlet points on the pump. The two outlet points connect directly to a control-valve block which is actuated by manual control of the winch driver. The delivery end of the block connects to the two outlet points on a six-cylinder, radial, hydraulic motor. Both these machines are flange mounted, and are contained in a welded housing, which also forms the reservoir for the hydraulic medium, both pump and motor being completely submerged by the fluid. The output shaft of the hydraulic motor is connected to the winch barrel by way of a two-speed spur gear, the low speed being exactly half that of the high speed. When the winch itself is at rest, the input electric motor is running at its constant speed, which may be suitably selected from the power cable rating of the ship. Whilst the winch is at rest the fluid delivered by the pump is simply circulated back to the tank against no pressure. The controls comprise two hand levers, one being a direction lever which controls the direction of the hydraulic motor. It may be interpolated here that full torque is available in either direction from this machine. The first operation of the winch man is to move the direction lever into the hoist position. The second lever is then moved half way along its quadrant to a notch, which is the creep position. With this lever in the creep position the control valve is so arranged that the delivery from the three small rams only is passed to the hydraulic motor; the delivery from the six large rams continues to circulate to the tank against no pressure.—*Paper by M. G. R. Petty, read at the Conference on Hydraulic Mechanisms, The Institution of Mechanical Engineers, 26th March 1954.*

Emergency Stopping Gear for H. and W. Engines

In Fig. 4 are shown four diagrams relating to the emergency stopping gear of Diesel engines. The first position of the gear (A) is that for normal running. All the numerals in the diagrams are for the same parts throughout. If the lubricating oil pressure fails, the mechanism assumes the position shown in diagram (B). The oil pressure switch (2) is open and current flows through the coil (3) to the relay (4), which loses energy and releases the switch (23). Energizing the coil (3) operates the indicator (5) which moves to the "fault" position and remains there until it is reset by the coil (6), which is energized when the pressure switch (2) is closed on restarting the

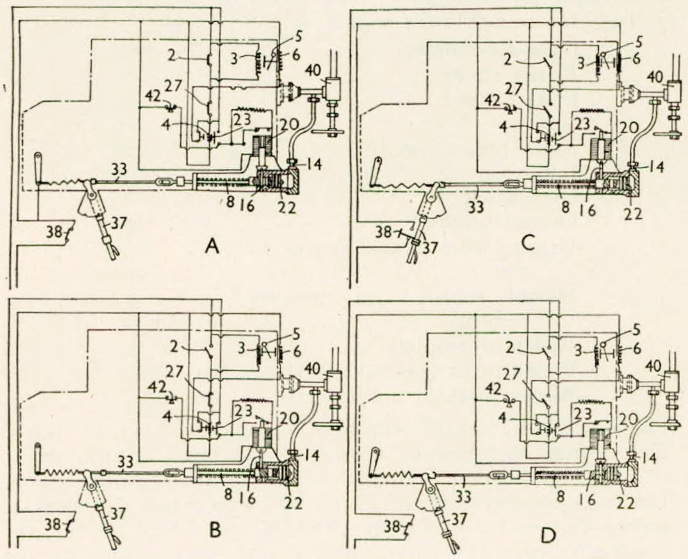


FIG. 4

engine. Should the cooling water supply fail, the flow-operated switch (27) opens and the switch (23) closes on being released by the relay (4). No current flows through the coil (3) and the lubricating oil indicator (5) is unaffected, thus showing that the fault must be in the cooling water system. When the engine has stopped (C) no current passes to the stopping mechanism or to the alarm (42), the switch (38) being open. When the engine is to be restarted (D), the air valve (40) is opened and the piston (12) moves the spindle (8) through its maximum travel. The switch (38) passes no current through the solenoid (20) and the lock (16) engages the spindle (8). The fuel regulating rod (33) returns to the "fuel on" position as soon as the starting handle (37) is moved from "stop" and the switch (38) is closed. When the starting air is shut off, the non-return valve (14) closes and air leaking past the piston (22) allows the spindle (8) to move back until it is arrested by the lock (16).—*Patent No. 702,508. A. J. Rodick. Harland and Wolff, Ltd., Belfast. The Motor Ship, March 1954; Vol. 34, p. 555.*

Detecting and Recording of Vibration

One of the biggest enemies of continuously sound working of oil engines, or any machinery for that matter, is vibration. Sometimes it is due to defects in bearings, crankshafts, valves, or to fuel-injection faults. To enable records of behaviour to be available on site immedately after a test a Cambridge Universal vibrograph affords many advantages. This instrument provides on a celluloid surface, lines so fine and smooth that, when optically magnified, it is possible to measure with ease to 0·01 mm. To read to an accuracy of one per cent, therefore, the amplitude of the record can be less than 1 mm., so that the moving parts of the instrument can be kept light and small, with negligible inertia, enabling a high frequency response to be obtained. Records taken over long periods occupy little space; they are immediately available for inspection, and measurement is by the use of a simple form of microscope or projector, whilst they may be photographically enlarged to a considerable magnification. The records are durable and impervious to oil and water. Normally suitable for the investigation of horizontal or vertical vibration problems, the instrument can, by a simple interchange or addition of parts, be adapted for rotational tests, or arranged as a portable unit in which the vibrations are communicated to the recording mechanism by bringing a projecting toe into contact with the vibrating body. It may also be adapted for use as a deflectograph for testing the deflection of structural parts.—*The Oil Engine and Gas Turbine, March 1954; Vol. 21, pp. 405–406.*

Cargo Winch

The General Electric Company has supplied Maxspeed electric cargo winch systems for twenty-five of the thirty-five Mariner-class vessels. The system is a simple modification of the conventional variable-voltage system, but its inherent characteristics are superior to those of the more common direct current (d.c.) constant potential system or any straight alternating current (a.c.) motor system. The other ten C4-S-1a vessels are equipped with different variable-voltage cargo-winch systems supplied by other manufacturers. The fundamental principles of the various systems are the same, but the operation and performance of each system differs. In addition to the twenty-five Mariner vessels, the Maxspeed system has been installed on the new passenger liner *United States,* and on the *Angele Higgins*; the latter has been in operation between New Orleans and Central America since 1945. Several industrial-crane installations have employed the Maxspeed system, especially where heavy loads have had to be handled and spotted accurately. When the Maritime Administration began to formulate the plans for the Mariner class of vessels, it was desired to utilize an a.c. auxiliary power supply for these vessels. For years, the advantages of using a.c. for most ship's auxiliaries has been recognized. The fact that d.c. motors provided the best means of speed control required for the cargo-winch application has resulted in d.c. being used as the power supply of general cargo vessels until recently. Other ships, such as tankers, naval vessels, and bulk-ore carriers have been able to take advantage of a.c. power systems. Three possible cargo-winch arrangements were considered. First, a large a.c. to d.c. motor-generator set could be installed in the engine room to supply the 1,200 h.p. of constant-potential cargo winches; secondly, some form of an a.c. motor cargo-winch system could be used; and thirdly, individual motor-generator sets for a pair of winches could be installed in the deck houses for a variable voltage cargo-winch system. The General Electric Company's experience with the Maxspeed system and the better characteristics of this system were important factors in deciding to use the third arrangement for the C4-S-1a vessels.—*F. A. Shean, Marine Engineering, January 1954; Vol. 59, pp. 59–66.*

Periphery Pump

The periphery pump, also designated as the tangential, turbine-vane, regenerative, turbulence, or friction pump, produces pumping action by the motion of a rough surface in a channel-containing fluid. The fluid is dragged along by the rough surface and, with suitable restraints in the channel, the fluid head is increased in the direction of the flow. Essential elements of the periphery pump are as follows: The impeller is a disc with vanes on the periphery. The casing contains the fluid passage, or raceway. Inlet and discharge ports connect the external-system piping to the raceway. Between the discharge and inlet, the casing clearance is reduced to block the high-pressure inlet. Clearances between the impeller disc and the casing are kept to a minimum to prevent leakage from the high-pressure side of the pump back to the low-pressure side. The periphery-pump performance is analysed in terms of shear stresses imparted to the fluid by the impeller. The resulting expressions include two shear coefficients and an average impeller velocity which are determined experimentally. The analysis predicts the shapes of the performance curves of head, power and efficiency as functions of the flow rate. The shear coefficients, and hence the pump performance, are shown to be functions of the impeller roughness, and of the flow channel area.—*Paper by H. W. Iverson, 1953 A.S.M.E. Annual Meeting; Paper No. 53-A-102.*

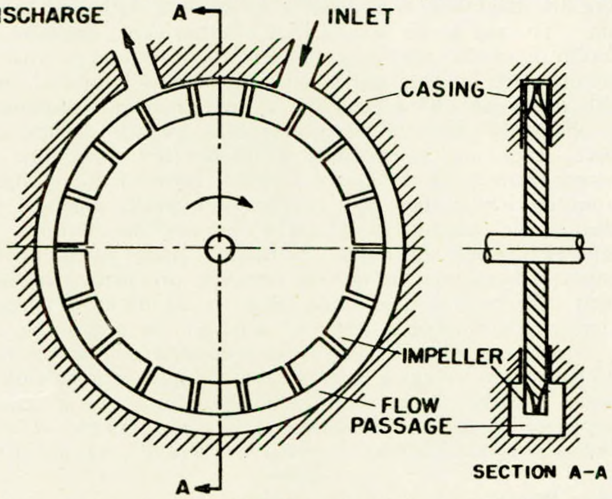


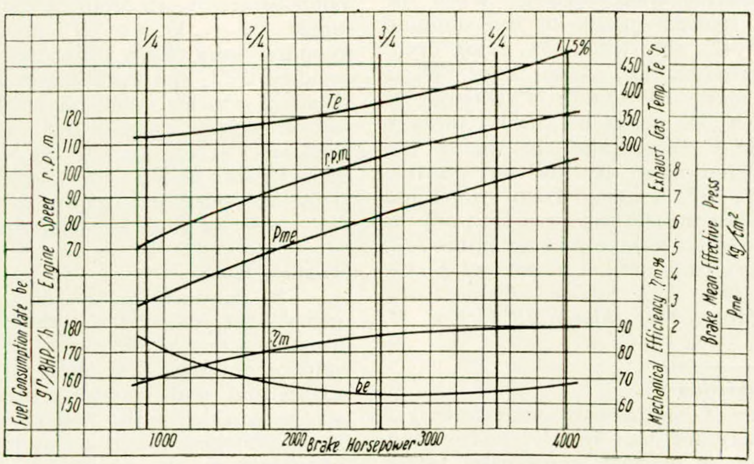
Diagram showing essential elements of the periphery pump

Japanese Turbocharged Engine

The Mitsubishi Shipbuilding and Engineering Co., at their Nagasaki Works, in 1932, developed their two-stroke single-acting M.S. engine, of which type eighty-five have been built, forty-five before the war and forty since its conclusion. They have now applied turbocharging to this design and after experimenting with a large unit are building a number of six- and nine-cylinder engines with outputs of 7,500 to 8,000 b.h.p. and 11,250 to 12,000 b.h.p. respectively. The first are expected to be completed in August of this year. The experimental engine has three cylinders, 720 mm. in diameter with a piston stroke of 1,500 mm., the speed being 115-120 r.p.m. and the normal output 3,500 to 3,700 b.h.p. The corresponding brake mean effective pressure is stated to be 7·47 kg. per sq. cm. to 7·58 kg. per sq. cm. (106-108lb. per sq. in.) and the mean piston speed 5·75 m. per sec. to 6 m. per sec. The engine has a centrally placed fuel valve with three exhaust valves actuated from a camshaft located at a level of about half the height of the engine. In the first place the blower was driven by an electric motor and this was followed by the employment of an exhaust gas-driven turbo-blower. No engine-driven blower is required. Progressive trials were carried out up to an overload of 15 per cent and a maximum figure of 4,455 b.h.p. was attained, with the engine running at 120 r.p.m., the brake mean effective pressure being given as 9·1 kg. per sq. cm. (130lb. per sq. in.). Trials were also made with the engine operating on boiler oil and these proved satisfactory so far as it was possible to ascertain during the limited period. It is arranged that the temperature of the exhaust gas leaving the turbine is higher than when the engine is unsupercharged and an increased utilization of the exhaust gases in waste heat boilers is rendered possible. Compared with the normally aspirated unit, the output is raised about 35 per cent. The engines now being built for installation in ships have an increased diameter, namely 750 mm., with the same stroke of 1,500 mm. The following are the main particulars of a nine-cylinder engine:—

Cylinder diameter	750 mm.
Piston stroke	1,500 mm.
Engine speed	115 r.p.m.
Output	11,250 b.h.p.
Mean piston speed	5·75 m. per sec.
B.m.e.p.	7·38 kg. per sq. cm.
Overall length	16·8 m.
Overall height	9·325 m.
Height from crank centre to top...	7·825 m.
Height required for drawing pistons	10·050 m.
Width of bedplate	3·600 m.
Weight, cast construction ...	510 tons
Weight, welded construction...	400 tons

The 12-cylinder engine will have an output of 15,000 b.h.p. The general performance results of the trials on the three-cylinder experimental engine are shown in the adjoining graph. The fuel consumption at 3,750 b.h.p., which may be taken as normal full load, was 155 gr. (0·341lb.) per b.h.p. per hr., the minimum being 153 gr. at three-quarter load, whilst for the range between half load and full load the consumption did not exceed 160 gr. (0·35lb.) per b.h.p. per hr. The mechanical efficiency of the engine was 80 per cent at half load, 88 per cent at three-quarter load and 90 per cent at full load. The exhaust gas temperature at three-quarter load was 375 deg. C. and at full load approximately 430 deg. C. The weight of the

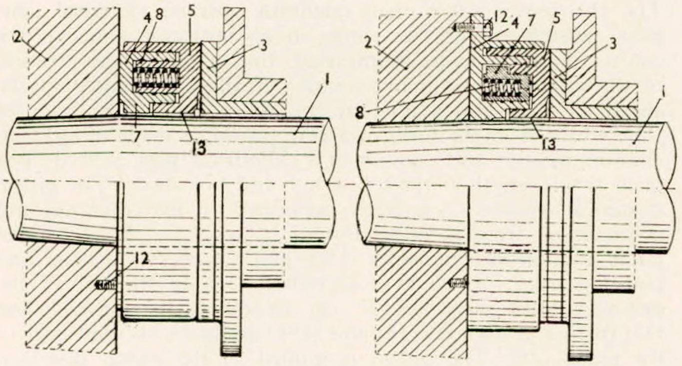


Graph showing test results on an experimental 3,750 b.h.p. Mitsubishi two-stroke turbocharged engine

engine of welded construction is equivalent to about 80lb. per b.h.p. If the cast-iron construction be employed, the specific weight is 102lb. per b.h.p.—*The Motor Ship, April 1954; Vol. 35, pp. 32-33.*

New Stern Tube Oil Gland

Oil-retaining glands for stern tubes have been in use on board various types of vessels for many years. With a view to improving the conditions at the outboard end of the stern tube, the Lips Propeller Works, Ltd., Drunen, Holland, are now manufacturing a device known as the Lips-Vandam stern tube gland and the accompanying sectional diagrams show two different types. The sealing portion of the gland is arranged



Two designs of the Lips-Vandam stern tube oil gland

between the stern bush face (3) and the propeller boss (2). The oil gland is entirely of bronze and comprises two cylinders (4, 5) enclosing the exposed part of the propeller shaft (1), a small clearance being allowed. The cylinders float on an oil or grease film and are machined with a number of grooves. Between the cylinders is a space incorporating a central spring holding-ring. There are two rows of spring chambers (8) bored alternately from the forward or after ends and containing stout helical springs. These springs exert a pressure, so that the cylinders bear continuously against the face of the propeller boss (2) and the stern bush (3). Along the cylindrical outer face of the spring ring is the required axial seal of the cylinder (5) and the same applies to the inner face of the ring. The cylinders (4, 5) are filled with marine lubricant from the stern tube and oil traverses the grooves (13), the helical springs being in an oil bath. The gland need not be attached in any way to the propeller, the stern tube or the shaft and it can, therefore, be entirely free to revolve, remain stationary, or "creep". However, if it is desired to employ a fixed type, the flange of the cylinder (4) is provided with holes to receive setbolts (12) which attach it to the propeller boss (2). The outer face of the spring ring has a number of rectangular projections (7) fitted with a small clearance in corresponding slots in the adjacent sealing face of the cylinder (5) and a similar arrangement is adopted with reference to the cylinder (4). These sealing faces are lined with white metal and are provided with an oil groove which lubricates the working surface. There are also several axial grooves (13) extending towards the tapered part of the propeller shaft (1).—*The Motor Ship, April 1954; Vol. 35, p. 35.*

Prevention of Growths on Hulls

Organic growths are vulnerable to vibration and are reluctant to settle on an unsuitable surface. The proposition outlined in Fig. 5 is to fit a number of magneto-striction transducers (1) on the band of the hull particularly susceptible to algæ fouling. The engine (2) drives a 230-volt alternator (4) and a step-up transformer (5) is fitted. A device (6) rectifies the current derived from the secondary winding of the transformer (5). An oscillator (13) generates alternating current at frequencies in the ultrasonic range and supplies the windings (21) of the transducers (1). One end of each core (22) is

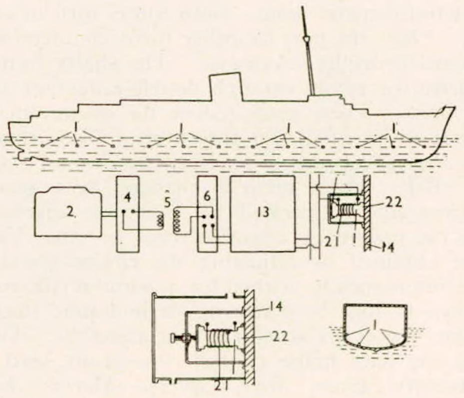


FIG. 5

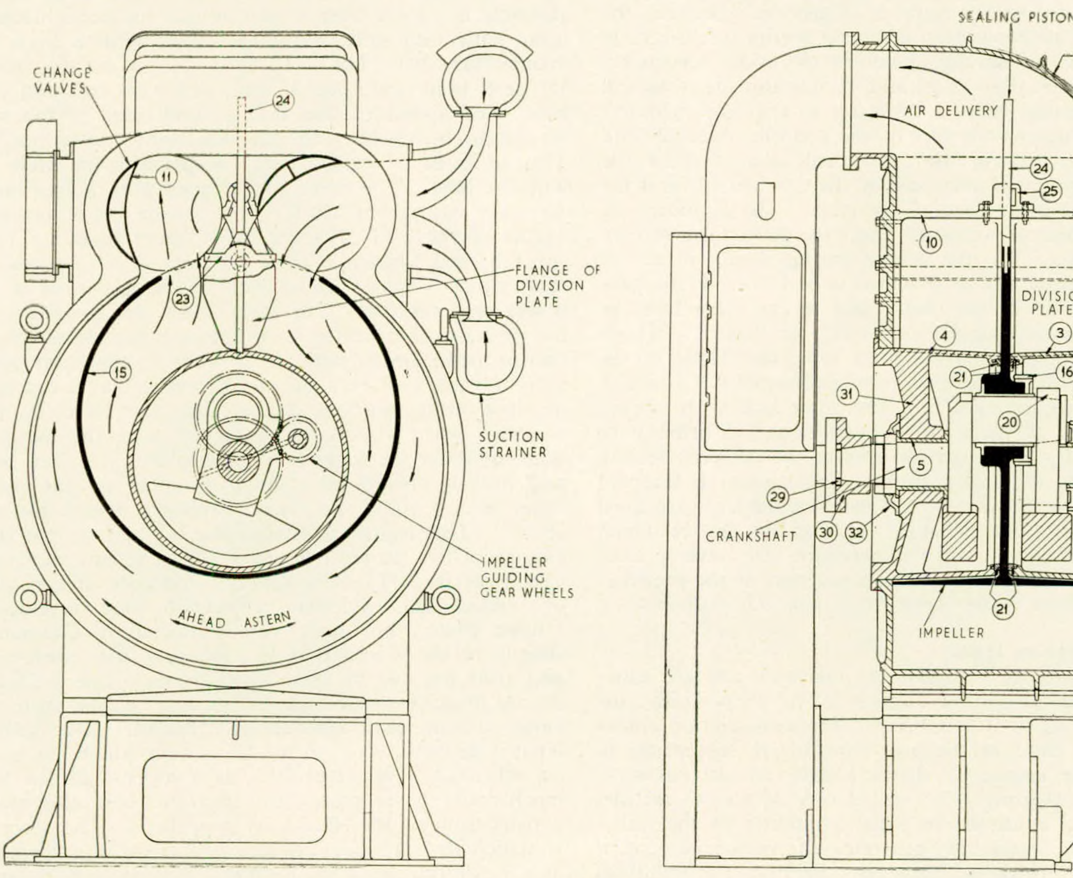
fixed to the hull plating (14). According to the specification, it has been found that magneto-striction transducers rated at 100 watts, arranged at intervals of 20 feet and preferably operated intermittently, are suitable for the purpose in view, so that presumably the invention has been put into practice, in some form. The vibration set up is transmitted from molecule to molecule of the hull plating, in contradistinction to the type of vibration which is caused by bulk movement.—*Patent No. 703,158. Postans, Ltd., and C. O. Morley, Birmingham. The Motor Ship, April 1954; Vol. 35, p. 40.*

Towboat with Sinusoidal Propulsion

The first U.S. river towboat with vertical axis propellers, the LTI-2194 employs a type of propulsion relatively new to the U.S.A. and radically different from the conventional screw-type propeller. Two vertical axis propellers are mounted in separate wells in the stern of the boat. At the base of each assembly is a rotor from which project six metal blades. These units both steer and propel the vessel with a high degree of manœuvrability. The LTI-2194 has a length moulded of 150 feet, beam moulded 32 feet, depth 10 feet and draught 7 feet. The all-welded steel hull is subdivided by five transverse, watertight bulkheads and one non-tight centre-line bulkhead. Towing knees of welded steel construction are built integrally with the bow. The vessel also is fitted with power capstan and accessory equipment aft for pull towing on a hawser. Main engine of the LTI-2194 are two Cooper-Bessemer 12-cylinder supercharged Diesels. They develop 1,000 h.p. each at 900 shaft r.p.m. Two Diesel-driven 60-kW generator sets provide power and lighting. The main shafts are direct-connected to the vertical axis propeller units through flexible couplings. The two propeller units, installed in wells in the stern, are covered by removable steel deck plates. Propeller blades can be changed in these units, without dry-docking the boat, by filling the forepeak with 50 tons of water to raise the stern propeller housing above the water-line. The blades then can be removed and pulled inboard by a crane mounted on the stern. In a series of test runs, the vessel displayed remarkable manœuvrability. This feature is attributable to the fact that the vertical axis propellers deliver maximum thrust in any direction. This also gives the LTI-2194 superior stopping ability. Principles of vertical axis cycloidal propulsion were developed in the United States, but later were acquired by German interests shortly before World War II. Research was continued during and after the war by the United States. The LTI-2194 units do not produce a true cycloidal motion because their pitch ratio varies during each revolution. Instead, these units produce what is described as a sinusoidal motion which has less theoretical efficiency. However, this is compensated for by simpler mechanism, fewer parts, less frictional loss and more rugged construction of the sinusoidal propellers. The rotor assembly, to which the six blades are attached, turns in a horizontal plane at a maximum speed of 76 r.p.m. The rotor is 11 feet 4 inches in overall diameter. The blades, fabricated of manganese bronze, are each 4 feet 6 inches long. Both rotors turn in an outboard direction. Thus, the port propeller turns counterclockwise and the starboard propeller, clockwise. The shafts from the main engines drive the rotors through double-reduction gears in the propeller units. These gears reduce the maximum 900 r.p.m. of the shaft to 76 r.p.m. for the rotors. Blade roots are connected by individual rods to an eccentric ring around the vertical rotor shaft. Blade pitch is changed by a gear rack and pinion arrangement operated by hydraulic mechanism connected to the pneumatic engine-control system. Variations in thrust are obtained by adjusting the engine speed while the blades are automatically pitched for maximum efficiency. Steering the boat is done by actuating an hydraulic steering piston which turns the entire eccentric ring assembly. Two steering wheels on the pilot house control console are used instead of the customary levers for rudders.—*Marine Engineering, February 1954; Vol. 59, pp. 36-38.*

New Scavenge Pump Design

Scavenging air for the latest type MT Gray-Polar engine, as installed in the new ore carrier *Orelia,* is supplied by a double-acting ring piston blower. It replaces the rotary fan type blower fitted previously to Gray-Polar engines, is positive in action, and gives 50 per cent excess air based on cylinder swept volume. Known as type RK-V, it is driven through gearing and a hydraulic coupling from the main engine crankshaft. The unit has a low mass of inertia and is isolated from the main engine torsional system through the hydraulic coupling. It operates on the Straatveit system and is manufactured at the Central Marine Engine Works, West Hartlepool, where the Gray-Polar engine is built under licence from A/B Nydqvist and Holm, Trollhättan, Sweden. *Description of Scavenge Pump:* The RK-V blower consists of a cast-iron outer housing (1) with a detachable end cover (2) and an inner cylinder in two parts (3) and (4). (The numbers refer to the diagram.) The inner cylinder (4), contains a guide or thrust bearing (5), and cylinder (3) contains a detachable end bearing (6) with cover (7) for the crankshaft which is in two parts. The end part (20) is secured to the main part (8) on a taper by means of a locking nut. The pressure side is separated from the suction side by a division plate which is in two parts (9) and (10), and two changeover valves (11) for ahead or astern running. These valves are operated by quadrants which are in turn operated by an air cylinder, which receives air at a pressure of 100lb. per sq. in. being inter-connected with the manœuvring gear for operation. The air is reduced from 350lb. per sq. in. through a reducing valve and reservoir tank and a delay stop valve, and is directed by a manœuvring valve which is operated by a fibre friction clutch from the end of the blower crankshaft. The changeover valve main quadrant operates electrical contacts causing appropriate lamps in an indicator panel at the control platform to be illuminated, thus showing the position of the valves during manœuvring. A silencer is fitted at the ahead air inlet and a filter at the astern side. The impeller (15), which is in the form of a ring piston, is made of elektron aluminium alloy and consists of a cylindrical part, slotted open in its whole length at the top, a hub and hub disc. The piston is statically balanced, is freely supported on the crankpin, and is prevented from rotating around its own axis by a parallel guide motion mechanism. This guide consists of sun and planet gearing formed by a gearwheel (16) on the hub of the piston, another gearwheel (17) on the journal bearing end, and two further gearwheels (18) and (19) carried by the end part of the crank (20). The piston is guided in the lateral direction by two spring-loaded sealing rings (21) placed at the ends of the inner cylinder and supported by the hub disc of the piston. These rings also act as scraper rings and prevent the lubricating oil from penetrating into the working cylinders along with the scavenging air. A balance weight (22) with a detachable keep (23) is riveted to the upper part of the hub disc of the piston.



Arrangement of RK-V ring piston blower

This balance weight carries a sealing piston (24) forming an elongation of the hub disc towards a bearing (25) which is attached to the top of both dividing plates. The sealing piston running in its bearing and sliding freely between the division plates, separates the pressure side from the suction side during the rotation of the piston. The crankshaft and its assembly is statically balanced by weights which are bolted to the webs. Lubricating oil is supplied to the bearings through the crankshaft, which is bored for this purpose, and the crank end can be dismantled for fitting the piston.—*The Shipping World, 24th March 1954; Vol. 130, pp. 328-330.*

Noise Reduction for Diesel Engines

A practical noise reduction treatment for Diesel engines is described, which, though not the ideal approach to the problem is, besides being practical, economical and applicable to both new engines and those already in service in the field. This treatment was designed making use of the results of a survey of the relative importance of the various noise sources on the Diesel engine. The treatment consisted of a set of sheet metal covers, lined internally with acoustical absorbing material, which were mounted directly over the noisy areas of the engine by means of built-in vibration isolation. Noise reductions for the direct sound were at least 8 to 12 db, which were observed at a distance of one foot from the engine. The total loudness of the engine noise at 6 feet was reduced from 320 to 220 sones. This is a substantial improvement with regard to voice communication in the immediate vicinity of the engine. The Speech Interference Level of the engine noise at 6 feet was reduced from 87 to 82 db.—*J. E. Ancell, Journal of the Acoustic Society of America, November 1953; Vol. 25, pp. 1163-1166.*

Stability of Single-step Boats when Planing

This paper describes some systematic model experiments which have been carried out at the Swedish State Shipbuilding Experimental Tank on a fast, planing, single-step boat having a displacement of about 35 m.^{3}. It was intended, in the first place, to investigate the influence of breadth variation both on the transverse statical stability under planing conditions and on the resistance. The aim was to arrive at a relation between the breadth and the other dimensions which would give good transverse stability together with acceptable resistance. In addition, the extents to which these qualities are affected by the height of the step and by the angle between the fore- and after-body keel lines were investigated by systematic variation of each of these characteristics. Similarly the deadrise was also varied in two stages.—*R. Röderström, H. Eostrand, and H. Bratt, Swedish State Shipbuilding Experimental Tank; Report No. 25, 1953.*

Fatigue Strength of Steel Pieces with Stress Concentrations

In previous investigations under the same general research programme, determinations were made of the intrinsic and vee-notched fatigue strengths of the first seven steels investigated, under various combination of bending and torsion. The intrinsic fatigue limits were found to fit reasonably to an ellipse quadrant, and the notched fatigue limits to a two-constant displaced ellipse or "ellipse arc". The bulk of the present paper concerns the behaviour under combinations of bending and torsional stresses of test pieces with transverse holes, for the same seven steels. The fatigue limits under the various stress combinations have been found to fit reasonably to an ellipse arc of the same form as for the vee-notched specimens. The similarity in form of the ellipse arcs for the seven materials supports the theoretical conclusion that the shape of the curve is dependent only on the shape of the hole, and is independent of the nature of the material. The information given on strength reduction factors under various combinations of bending and torsion, together with the tables of results of supplementary static tests, should be of use in such problems as crankshaft design. The paper also includes the results of flexural and torsional fatigue tests on an aircraft crankshaft steel, with other forms of stress concentration. These results are compared with those on two other steels of similar composition.—*Paper by R. C. A. Thurston, submitted to the Institution of Mechanical Engineers for written discussion, 1954.*

Marine Shafting Alignment

In Part 1 a brief account is given, of a method in general use, of installing propeller shafting and setting an engine bedplate to align the crankshaft with propeller shafting. This is followed by a discussion of the events which led to the need for modification of the above procedure and goes on to describe a simple technique which enables alignment to be checked during (*a*) construction; (*b*) installation on board ship; and (*c*) in service under loaded and light ship conditions. Alignment readings obtained during the above stages are given together with readings taken during the investigation of service difficulties in which malalignment had been suspected. In Part 2 the results are given of calculation made by the staff of the British Shipbuilding Research Association of the variation in the loading of the shaft bearings in a 17,000 tons d.w. Diesel-engined tanker, with wear-down of the stern bush. The significance of the results are discussed with regard to such factors as the effect of propeller forces, the allowable wear-down of the stern bush and the possibility of lateral vibration of the shafting occurring in service. It is concluded that the aim in design should be to provide sufficient flexibility so that the bearing loads on the after plummer blocks do not change appreciably with wear-down of the stern bush.—*Paper by G. Yellowley, and J. E. Richards, read at a meeting of the North East Coast Institution of Engineers and Shipbuilders on 12th March 1954.*

Oil Tank Motorship

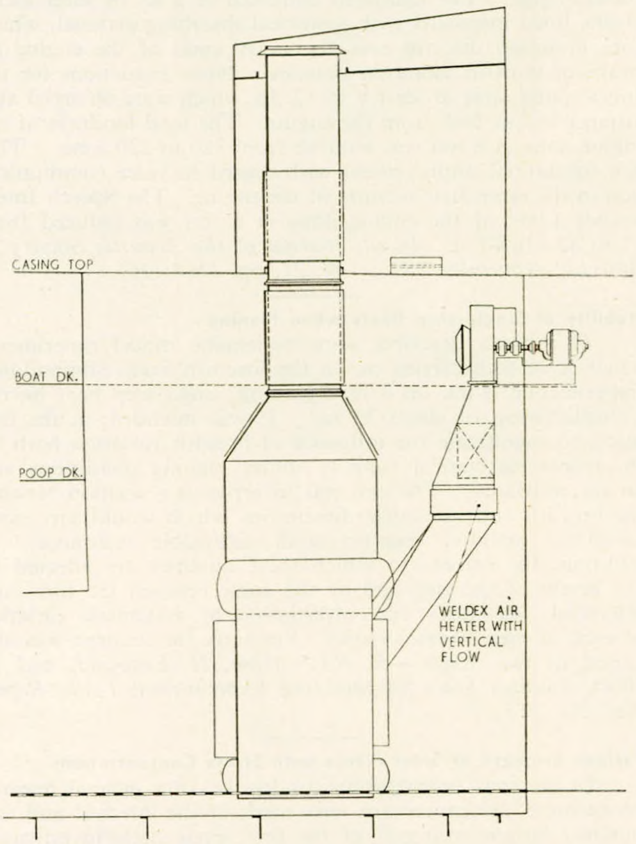
After completing successful trials, the oil-tank motorship *Hallingdal*, built by Messrs. Burmeister and Wain, of Copenhagen, for Messrs. Boe and Pedersen, of Oslo, has entered her owners' service. The *Hallingdal* has been built to the requirements of Det Norske Veritas for the classification +1A.1, and to the regulations laid down by Norske Skipskontroll. The principal dimensions and other leading characteristics are given as follows: —

Length b.p.	465ft. 0in.
Breadth moulded...	62ft. 10in.
Depth moulded to main deck ...	34ft. 7¾in.
Load draught	27ft. 6¾in.
Corresponding deadweight, tons	13,400
Capacity of cargo tanks, cu. ft. ...	630,000
B.H.P.	4,600
Corresponding r.p.m.	115
Speed on loaded trials, knots ...	14

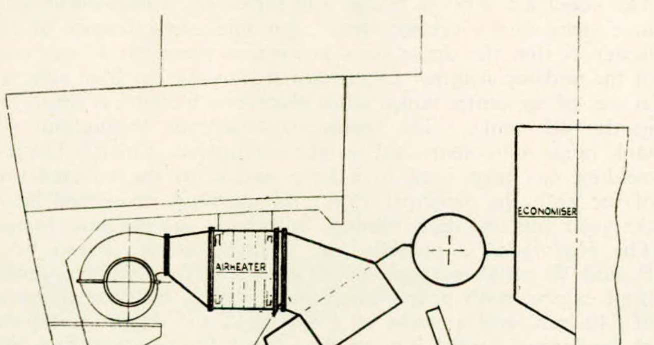
The vessel has a poop, bridge and forecastle, a well-raked softnose stem, and a cruiser stern. An interesting feature of the design is that the upper deck is without sheer for 45 per cent of the midship length. Longitudinal framing has been adopted in way of the centre tanks, while transverse framing is employed in the side tanks. The transverse bulkheads throughout the tank range are constructed on the corrugated system. Electric welding has been used to a large extent in the construction of the hull, the principal structural members concerned being the shell plating, deck plating, bulkheads, frames and beams. The *Hallingdal* is propelled by a single screw, driven by a B. and W. single-acting, two-stroke cycle, five-cylinder, crosshead engine, with airless injection. With a cylinder diameter of 740 mm. and a stroke of 1,600 mm., the engine is capable of developing 4,600 b.h.p. at 115 r.p.m. Electricity is provided by two 200-kW Diesel-driven generators, which supply direct current at a tension of 220 volts. The prime movers for these sets consist of B. and W. single-acting, four-stroke cycle, fivecylinder, trunk-piston Diesel engines. In addition, a 33-kW generating set is installed, and this unit is driven by a B. and W. four-stroke cycle, two-cylinder, trunk-piston engine. There is also installed a 30-kW steam-driven dynamo. Steam for operating the 30-kW dynamo, the windlass and the winches is generated in two oil-fired boilers, with a heating surface of about 1,940 sq. ft., and an exhaust-gas boiler with a heating surface of about 1,500 sq. ft. The boilers are all designed for a working pressure of 180lb. per sq. in.—*The Shipbuilder and Marine Engine-Builder, March 1954; Vol. 61, pp. 170-171.*

Bled Steam Marine Air Preheaters

There is a definite trend for higher steam conditions in vessels being built today, calling for higher boiler efficiencies than are possibly obtainable with a gas air preheater and three-stage feed heating. In his paper "Steam Air Heaters for Marine Water Tube Boilers", read before the Institute of Marine Engineers on 11th November 1952, the author, W. J. S. Glass, comments on the difficulties experienced with gas air heaters and their attendant troubles. Five American tankers with which the author has been concerned are fitted with steam air heaters which have given trouble-free service for three years. These ships are powered with geared turbines of 13,000 s.h.p., taking steam at 600lb. per sq. in. and 850 deg. F. superheat from two integral furnace boilers having steam air preheaters and economizers. The combustion flow to the boilers is 73,000lb. per hr. at 17,100 cu. ft. per min. at 100 deg. F. The final air temperature of 279 degrees is achieved by bleeding steam from the seventh stage of the h.p. turbine at 63lb. per sq. in. absolute and 405 deg. F., which is approximately 100 degrees superheat. The steam flow is 3,110lb. per hr. and the air pressure drop 2·6in. water glass. When manœuvring, or steaming in port, steam from the exhaust range at 15lb. per sq. in. (gauge) is used, giving an air temperature of 240 deg. F., an important point on a ship using a boiler or boilers when the main machinery is shut down, as it gives a better port operation with resultant fuel economy. The application of steam heating to marine boilers has resulted from chokage troubles in gas air heaters. This trouble is thought to be due to the increased sulphur content of modern fuel oils, apparently a result of changes in the source of crude oils and developments in refinery methods. Steam air heaters, when properly designed are, of course, entirely free from the need for maintenance or cleaning, and trials figures confirm that the overall cycle efficiency is reduced by about ½ per cent only. The following variants are in use, or have been considered, for applying steam air heating to marine boilers. In new designs steam bled from the main turbines is supplied to a steam air heater, thus eliminating the need for a gas air heater. When the main propulsion machinery is not operating, auxiliary steam is supplied to the air heater. In some cases it may be advantageous to subdivide the heater and use more than one stage of steam bleed; this usually results in a smaller heater for a given temperature rise. The heater can also be fitted with a desuperheating stage operating on the main steam supply, thus contributing some additional air heating at high temperatures, while also enabling a close control to be maintained on superheat temperature. On existing ships which have experienced severe heater trouble, it is possible to modify drastically the gas air heater supplies by the removal of tubes and substituting a steam air heater. Many of the troubles of the gas air heater, it is thought, derive from condensation of sulphuric vapours on the tube walls. Existing heaters can be fitted with a steam air preheater on the air intake side so that, although with some loss of efficiency, the tube wall is maintained always above the dew point. The size and weight of a steam air heater depends on the weight of air to be handled, but this is not variable on any particular installation. The most important feature that can be varied is the outlet air temperature in relationship to the steam temperature available. In this connexion it is worth noting that the relevant temperature is the saturation temperature of the steam, since moderate degrees of superheat—say, up to about 150 deg. F.—do not materially affect the performance of a gilled tube battery. A typical example of a bled steam air heater installation in a passenger vessel is to be found in the Orient liner *Orsova,* which is now undergoing her trials. This vessel is equipped with three watertube boilers of Foster Wheeler controlled superheat design, arranged to give a superheat control from 850 deg. F. down to 600 deg. F. when manœuvring, the steam pressure at the superheater outlet being 525lb. per sq. in. The air heaters give an air temperature of 210 deg. F. with bled steam at 35lb. per sq. in. absolute. A temperature difference of 10 to 12½ per cent can be achieved without difficulty when required, below which there are limitations in the heater unit. Another which is equipped with air heaters is the *World Enterprise,* a turbine tanker with Foster Wheeler boilers supplying steam at 850lb. per sq. in. pressure and a temperature of 850 deg. F. at the superheater outlet. The *World Concord* and *World Unity* were both converted from gas air heating to



Typical arrangement of bled steam air preheater in a large tanker



Arrangement of Weldex air heater in Foster Wheeler boiler bled steam heating with satisfactory results. Eleven of the 32,000-ton tankers owned by British Tanker Co., Ltd., are also being fitted with Weldex air heaters.—*The Shipping World, 10th March 1954; Vol. 130, pp. 282-283.*

Ultrasonic Level Indicator

A novel liquid level indicator system, Model SL-101, is an instrument using ultrasonic pulse ranging techniques. The system, which uses no moving parts such as floats, synchros, potentiometers and switches, has been designed to indicate true liquid levels in petroleum, chemical and pharmaceutical processing and storage tanks within plus or minus 0·01 foot. The equipment is also suited to shipboard ballast and fuel tank installations as well as to applications requiring the precise location of the demarcation line existing between immiscible liquids. A direct level reading in tens, units, tenths and hundredths of a foot is presented on illuminated decimal counters. Level readings are always correct due to the action of an automatic calibration system that compensates for any changes in the velocity of sound propagation that may be caused by varying temperatures, specific gravities and similar differences between liquids being gauged.—*Design News, 1st February 1954; Vol. 9, p. 102.*

Protection of Gearboxes Against Corrosion

Rusting and corrosion are particularly destructive in the field of power transmission, as particles of rust have a tendency to become detached and, when mixed with oil, cause abrasion and the risk of bad damage to gears. This is particularly prevalent in ships, docks, harbours and similar marine installations, where sand is liable to get into the boxes, and traces of salt in the atmosphere magnify the effect of corrosion on steel. One interesting method of protection is the use of Detel chlorinated-rubber paints on the interior surface of the gearbox. Detel coatings consist of chlorinated-rubber and other synthetic paints, and have the attractive appearance of a lacquer. Most pigments can be ground in Detel medium, as it is very inert and will protect sensitive pigments from the action of acids or alkalis. The best protective compositions, however, are made with highly resistant pigments, such as titanium dioxide. For protecting iron and steel surfaces, a modification, known as Detel metal undercoat (D.M.U.), is used, which depends chiefly on its high metallic zinc content (over 96 per cent), for protecting these metals. This method is particularly suitable for the protection of marine gearboxes, and extensive tests have shown that after twelve months' running, their condition is unchanged, with the surface untouched. Another important advantage is that the film is impervious to lubricating and mineral oils.—*The Shipbuilder and Marine Engine-Builder, April 1954; Vol. 61, p. 224.*

Ductility Transition of Weld Metal

It has been recognized that development of sharp, crack-like flaws during fabrication or in the course of service of welded structures provides the "trigger" elements required for the initiation of brittle failures. Whether or not the presence of such trigger elements actually results in the initiation of a brittle failure is determined by the properties of the material in which the notch is located, the temperature range of service and the loading conditions. If the material retains the ability to deform in the presence of the notch flaws at the lowest service temperature, fractures will not initiate in structures which are loaded within the elastic range. That is to say, the stress level is of no importance as long as it is kept below the range of plastic deformation, which is the usual case. On the other hand, if the material containing the notch loses its ability to deform in the presence of the sharp notch flaws at service temperatures, fracture initiation becomes possible and design factors related to levels of nominal elastic loading then become important. The development of cracks in arc craters, heat-affected zones and/or welds may be considered to represent a process of notching in the same sense that a notch is imposed by machining or other means. Unfortunately, the natural notching process produces the sharpest possible types of notches (cleavage cracks) and, therefore, the most critical with respect to service performance. In laboratory investigations it is important to determine the effects of such sharp notches by methods which actually produce the natural type. Previous investigations have demonstrated the effects of crack-like notches on the performance of structural and alloy steels. These studies were concerned with fracture initiation under conditions such that a crack fault projected into the base metal. The present investigation is concerned with fracture initiation in welds resulting from the presence of crack flaws located entirely in weld metal. The relative resistance to initiation of brittle fractures for various types of weld metal is determined and compared with the previously established characteristics of structural steels. It should be emphasized that this and the previous investigation are not concerned with the actual level of elastic loading required to initiate fracture but rather with determining the "boundary" temperature above which plastic deformation is required for the initiation of fractures. At temperatures above the boundary temperature the fracture initiation problem for structures which are loaded only within the elastic range is considered solved by material properties. At temperatures below the boundary temperature design factors related to the actual level of elastic loading become important.—*W. S. Pellini and E. W. Eschbacher, The Welding Journal, January 1954; Vol. 33, pp. 16s-20s.*

Accident to Steam Turbine

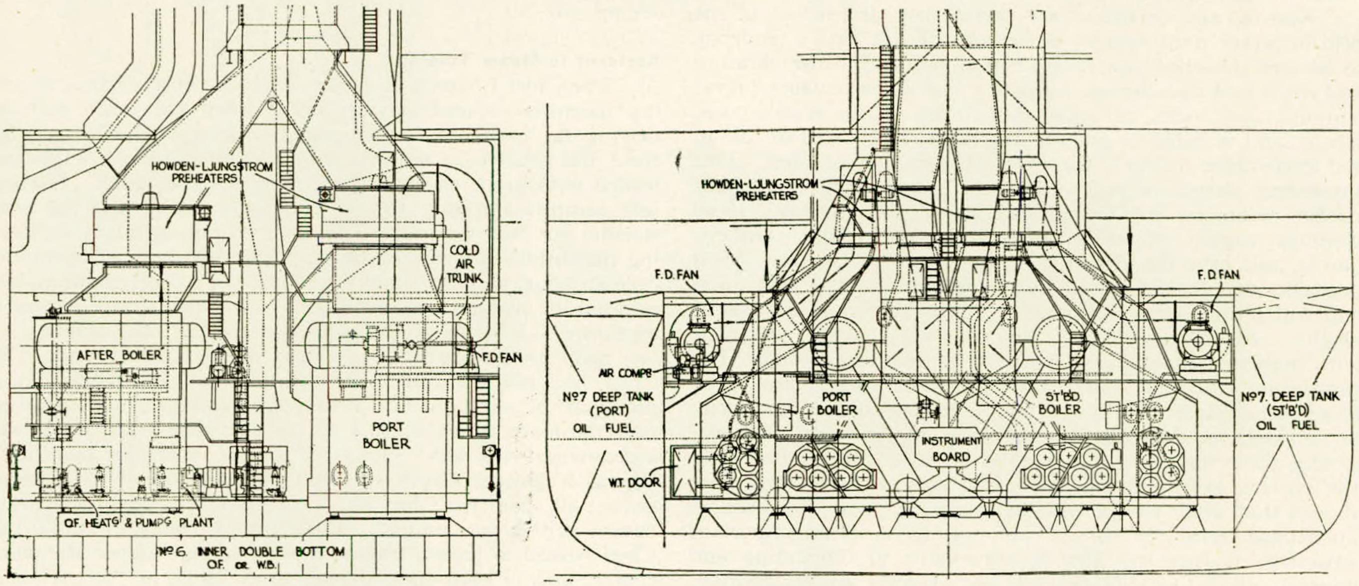
The Chief Engineer and First Assistant of a seagoing steam tug narrowly escaped serious injury when the steam turbine driving the main circulating pump disintegrated with terrific force and splattered the lower engine room as though a cannon loaded with grape shot had been fired at close range. Having just completed repairs to the evaporator plant, the tug was starting out on a sea trial. The main propulsion plant, including the turbine-driven circulating pump, had been in operation over an hour, and the Chief was on watch in the engine room, engaged in making an inspection of all main and auxiliary machinery. The First Assistant was checking the operation of two main fire pumps at the after end of the engine room. The Chief had just completed his examination of the circulating pump on the starboard side of the engine room and was crossing over the main thrust bearing to the port side when there was a detonation—"a heavy explosion, as of a steam line bursting". Flying fragments struck the hull and machinery with lethal impact all about the Chief and First Assistant, but, very fortunately, neither was injured. With great presence of mind, the Chief rushed to the top engine room grating, stopped the main engine, and ordered the Fireman to cut fires in the port and starboard boilers. Saturated steam at 250lb. per sq. in. was blowing freely from the fractured end of the supply line to the turbine. The Chief closed the main steam stop to the turbine as quickly as he could get to it and neither he nor the First Assistant suffered any steam burns. Inspection showed that the turbine casing was ruptured; the rotor had torn loose from the shaft and disintegrated into several parts, the poppet valve and overspeed trip valve were hanging adrift with their mounting bolts broken from the head of the turbine; the main steam valve was punctured; and the steam line strainer was ruptured. Examination of the rotor for signs of any old fractures or cracks was futile as the parts were too badly damaged in the "explosion". Close scrutiny of the steam strainer to see if any foreign objects could have caused the rupture was inconclusive, as the rupture seemed to be more a failure of the crimped joint in the screen due to steam pressure. Since the Chief Engineer had just checked the water level in both boilers, which were operating normally a minute or two before the casualty, the possibility of boiler priming and the turbine being damaged by a slug of water was ruled out. There was no evidence that the turbine had "run away" as the governor was checked for proper operation only an hour before the casualty, and the Chief would have detected any rapid increase in turbine speed as he was in the immediate vicinity. The most logical conclusion was that the rotor had suffered a structural failure.—*Proceedings of the Merchant Marine Council, U.S. Coast Guard, January 1954; Vol. 11, pp. 12-13.*

New Passenger Steamship

The P. and O. passenger liner *Arcadia* has been constructed to the highest class of Lloyd's Register of Shipping and also complies with the requirements of the Ministry of Transport for foreign-going passenger vessels. The principal particulars are as follows:—

Overall length	720ft. 0in.
Length between perpendiculars...	668ft. 0in.
Moulded breadth...	90ft. 6in.
Depth to "C" deck	49ft. 9in.
General cargo capacity (bale) ...	211,000 cu. ft.
Insulated cargo capacity... ...	158,500 cu. ft.
Gross tonnage	29,734 tons
Service speed (34,000 s.h.p.) ...	$22\frac{1}{2}$ knots
Passengers (First class)	679
Passengers (Tourist)	735
Crew	711

The propelling machinery installed in the *Arcadia* is generally similar to that which the P. and O. Co. have found successful in the *Himalaya*, of 1949, and the *Chusan*, of 1950. There are, however, important differences, notably in the steam generating plant which now consists of three equal-sized Foster Wheeler two-furnace controlled-superheat boilers instead of the two large and two small ones of the earlier ships. Three 1,200 kW generators have replaced the former four 850 kW sets and the distilling plant is on a greatly increased scale. The main boilers are of the two-furnace Foster Wheeler controlled-superheat type, built under licence by the shipbuilders. The designed steam conditions at the superheater outlet are 530lb. per sq. in. and 850 deg. F. The temperature can be reduced down to 600 deg. F. for manœuvring. Each boiler has a generating tube surface of 20,355 sq. ft. and a superheater surface of 2,525 sq. ft. The normal rating is 105,000lb. per hr. and the maximum rating is 165,000lb. per hr. There are five Wallsend burners in each saturated furnace which has a single-row water-wall, floor and roof of 2in. tubes. This furnace is separated from the superheated furnace by an intermediate bank of seven rows of $1\frac{1}{4}$in. tubes and four rows of 2in. tubes. The superheated furnace has seven Wallsend burners and a single-row water-wall, and floor of 2in. tubes. The superheater nest is carried on Cronite supports and consists of a bank of five loops of $1\frac{1}{4}$in. tubes. It is shielded from the furnace by three rows of 2in. generator tubes. The main bank of generator tubes is on the far side of the superheater and has twenty-one rows of $1\frac{1}{4}$in. tubes. Each boiler has a feed water economizer supplied by E. Green and Son, Ltd., of Wakefield. After leaving the economizer the flue gases pass through a Howden-Ljungstrom CXG rotary generative air preheater. Combustion air is supplied to each boiler by a $43\frac{1}{2}$in. diameter Howden SV7 double-inlet fan, driven through flexible couplings by a 151 h.p. pipe-ventilated Allen motor. The fan motors have a wide degree of speed variation (about $4\frac{1}{2}$ to 1), by means of Allen contactor-type controllers. The capacity is 60,000 cu. ft. of air per min. each at 80 deg. F. when running at 1,230 r.p.m. The induced draught fans are installed at the base of the funnel and are of Howden TV2 single-inlet, $81\frac{1}{2}$in. diameter type, with a capacity of 92,000 cu. ft. of gas per min. at 340 deg. F. against a combined suction and discharge pressure of $15\frac{1}{2}$in. water glass, when running at 830 r.p.m. The induced draught fans are driven by 400 h.p. Allen motors having control from 900 r.p.m. down to 415 r.p.m. by shunt regulation and a further reduction to 150 r.p.m. by series regulation. The Allen controllers for both the forced and induced draught fans are of automatic contactor type having remote control units at a central position in the firing aisle. Three independent Wallsend oil fuel pumping and heating installations have been provided; each has a duty of 30,500lb. of fuel per hr. at 300lb. per sq. in. and consists of a motor-driven Weir reciprocating pump and a heater. Two of these sets are capable of supplying all three boilers at full power. The third is installed for standby purposes.—*The Marine Engineer and Naval Architect, March 1954; Vol. 77, pp. 85-99.*



Elevation looking to port and section looking forward through the boiler room

Welds in Cr - Mo Steels

Steels used for high-pressure steam lines of power plants are primarily the Cr - Mo types which were developed for resistance to graphitization at high temperatures. In welding these steels it is desirable that the heat-affected zones transform primarily to ductile, high-temperature reaction products. In order to prevent the development of high hardness reaction products resulting from transformations at low temperatures the generally recommended practice involves both preheat and postheat treatments, usually in the order of 400-600 deg. F. preheat, followed by a 1,250-1,350 deg. F. postheat. Such heat treatments are expensive and difficult to apply to practical pipeline systems. The extent to which the various thermal treatments applied in practice actually produce the desired metallurgical effect has been subject to considerable debate. This investigation was aimed at studying the transformation behaviour of these steels so as to provide needed basic information. Material from a 1·25 per cent Cr - 0·5 per cent Mo steel pipe (8⅝-in. outside diameter by 0·837-in. wall thickness) and a 2·25 per cent Cr - 1·0 per cent Mo steel pipe (6⅝-in. outside diameter by 0·750-in. wall thickness) was used for this investigation. It is shown that the Cr - Mo steels investigated undergo primarily a bainitic reaction of relatively high hardness. Therefore, if preheat is to have a beneficial effect, it must prolong the time the steel has to transform at temperatures above the bainitic range so as to promote ferrite and pearlite transformation. The extent to which preheating as high as 600 deg. F. can be beneficial in eleminating the bainite reaction is shown. Even a 600 deg. F. preheat is not effective in that it will drop the maximum hardness in the heat-affected zone less than 40 Vickers hardness numbers for both steels investigated. From consideration of the time-temperature relationships of the continuous cooling transformation diagrams it appears that preheats of approximately 1,000 deg. F. would be required. This, in effect, represents holding the metal at a temperature such that isothermal transformation to ferrite and pearlite structures will occur. Such preheat conditions do not appear to be practical. This should not be interpreted to mean that preheats of 600 deg. F. or less are not useful; certainly under high restraint conditions preheat quite frequently will eliminate cracking of root passes. However, it is concluded that preheating of these materials provides no beneficial metallurgical effects.—*W. R. Apblett, R. P. Dunphy, and W. S. Pellini, The Welding Journal, January 1954; Vol. 33, pp. 57s-64s.*

Steam Power Installation

The invention consists in a steam power installation wherein the condenser for the exhaust steam from a steam turbine power plant is interposed in a closed cooling water circulating system incorporating heat exchangers which are also incorporated in series in a closed gas circulating system. Fig. 4 diagrammatically illustrates a steam power installation according to the invention, where in the steam generator (1) supplies steam to a turbine (2), the exhaust steam from which is condensed in a water tube condenser (3). The water circulated through the condenser tubes is delivered by a circulating water pump (4) in a closed water circulating system including the condenser (3), a primary heat exchanger (5) and secondary heat exchanger (10), the water flowing in the direction of the arrows in the illustration. The primary heat exchanger (5) is interposed in a closed gas circulating system and the gas flowing through and heated by the water circulated through the heat exchanger (5) passes to a compressor (6) wherein the gas is raised to the desired temperature and pressure. It is then delivered under pressure to a re-heat boiler (7) where it gives up heat to water of condensation drawn from the condenser (3) by a feed pump (11) and circulated by way of a third heat exchanger (13) through the re-heat boiler (7), wherein saturated steam at the desired pressure is thus generated and supplied to the steam generator (1). After the gas has given up heat in the re-heat boiler, it is supplied to an engine (9) wherein the energy remaining in the gas is utilized to assist the work done by the compressor (6). The expansion of the gas in the cylinder of the engine (9) lowers the temperature of the gas, depending upon the degree of expansion, and the gas exhausted from the engine (9) is utilized to cool the water circulating in the closed water circulating system. For this purpose the exhaust gas from the engine (9) is admitted to the secondary heat exchanger (10) and thereby cools the water passing through the heat exchanger (10) to the water circulating pump (4) which delivers the water to the condenser (3). The advantages of the present invention are (*a*) a large proportion of the low potential heat normally wasted in the condenser water is collected by the gas in the primary heat exchanger and raised by the compressor to such a temperature that it can be usefully employed in regeneration of steam; (*b*) of the power plant.—*British Patent No. 703,979, issued to H. G. Turnell. Complete specification published 10th February 1954. Engineering and Boiler House Review, May 1954; Vol. 69, pp. 156-157.*

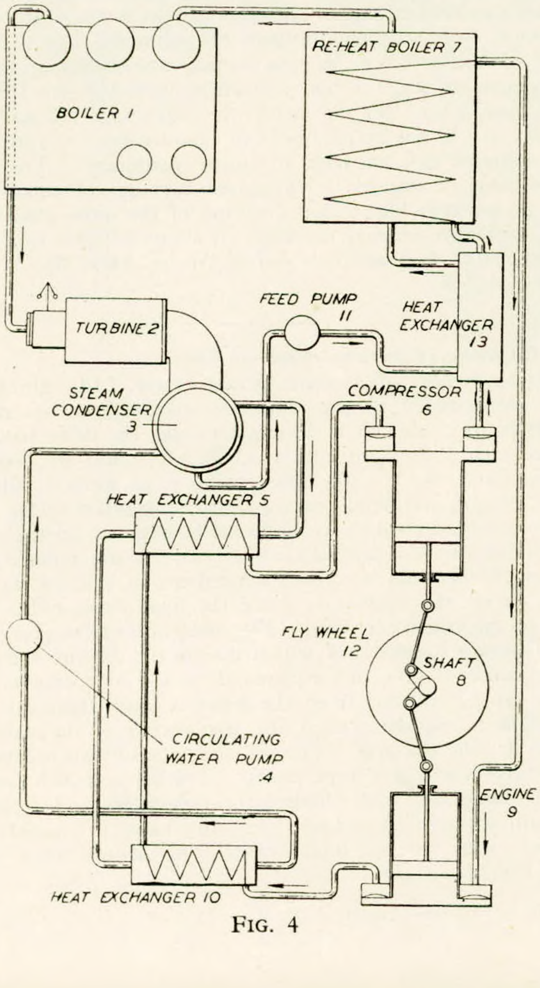


FIG. 4

Chemical Tank Cleaning System

Many tankers have now been cleaned with the use of Gamlen chemicals; of these twenty-one are British owned, and they include a number of Admiralty vessels. The principal advantage claimed for this process is that the time spent on tank cleaning is appreciably reduced—in some cases by as much as 50 per cent over the previous time. Furthermore, the heavy wax and heavy sludge deposits encountered on the tank surfaces and bottoms after carrying crude oil need no longer be removed manually—a lengthy, dirty and expensive method. The chemical used, a strong emulsifying solvent, on striking the tank surfaces has the effect of breaking up the sludge and wax and bringing down the oil adhering to the tank sides. This method can be undertaken on ships with or without Butterworth equipment. It can, of course, also be used for the cleaning of bunker and settling tanks on cargo ships using heavy oil. The general practice is to introduce the liquid chemical into the suction side of the cleaning water by means of a small air-driven pump. The treated water is then pumped to the tanks to be cleaned, the ship's Butterworth system being used, if fitted. The tank being treated is pumped out in the normal manner, this ensuring that the pipe lines are also thoroughly cleaned. It is claimed that the chemical has no adverse effect on the pump, heater or lines through which it passes. Tank cleaning has recently been carried out at sea by ships' crews. Naturally, the cost of cleaning varies according to the type of cargo carried, some crudes having a higher wax content than others. A reasonable average figure would be 9d. per ton of cargo capacity. Approximate figures show that the cost of chemicals for cleaning a T2 tanker would be in the region of £730, while for a 32,000-ton tanker it would be about £1,200. Cleaning tanks for a change of cargo is, of course, more expensive; in such circumstances, the chemical is injected into the preheated tank by a steam injector. By this means, the vapour carries the solvent to every portion of the tank, ensuring that the deck heads, sides, cross-members and crevices are cleaned as thoroughly as the tank bottom. This treatment is followed by a wash down with either fresh or salt water. The cost for such cleaning, again depending on the state of the tank prior to cleaning, would be in the region of 2s. 6d. per ton of cargo capacity.—*The Motor Ship, March 1954; Vol. 34, p. 529.*

Inert-arc Welding of Pressure Piping

Since the advent and development of the inert-gas-shielded arc-welding process, it definitely appears that through its use it is possible to progress a step closer to the perfection desired for root conditions of circumferential butt-welded joints in pressure piping systems. The following conclusions may be made as to the adaptability of the inert-gas-shielded arc-welding process: 1. An improved and more desirable root bead condition can readily and consistently be produced. 2. Although sufficient cost figures have not been accumulated as yet, there is every indication that the cost for the quality produced will be favourable. 3. The method of accomplishing the results reported herein is adaptable to both shop fabrication and field erection of pressure piping. 4. The method used to inspect the inside root bead condition is easily performed and very satisfactory. 5. By retaining the backing gas for the first two or three beads, it is possible to obtain a more desirable inside condition than if the backing gas is removed after the completion of the first bead. 6. Pipe-to-pipe joints within reasonable tolerances (misalignment of $\frac{1}{16}$th to $\frac{3}{32}$-in.) can be welded together with satisfactory results. However, when joining valves or fittings to pipe, it may be necessary in some sizes and schedules to bore the pipe to within the limits specified herein. 7. When using the method employed herein for piping that does not require preheating and/or stress relieving, it would be necessary to heat the joint uniformly to 600 deg. F. in order to burn up the paper discs and tape. In general, this would not be an adverse requirement.—*R. T. Pursell, The Welding Journal, January 1954; Vol. 33, pp. 41-46.*

Active Rudder in New Cable Ship

The first British ship to be fitted with the German-developed active rudder will be a cable ship, which is to be converted from a cargo ship for Submarine Cables, Ltd., a firm which is owned jointly by Siemens Brothers and Co., Ltd., and the Telegraph Construction and Maintenance Co., Ltd. The purchase of this ship marks the return to the owning of cable ships of the parent concerns after more than a decade: the predecessor of the present ship was the *Faraday*, owned by Siemens, which was lost through enemy action in 1941. The new ship, which is at present named *Empire Frome*, is a vessel of 5,360 tons deadweight, built in Germany in 1948. After completion of the conversion, which will take about twelve months, she will be renamed *Ocean Layer*, and will be available both for the firm's own work and for charter by cable operating companies. Experience is already available of the use of the active rudder for cable-laying work, as the first ship to be fitted with it, the coaster *Irmgard Pleuger*, was specially fitted with bow sheaves and time chartered by a German cable company. It proved extremely useful for this type of work.—*The Shipping World, 10th March 1954; Vol. 130, p. 271.*

Higher Powered Marine Diesels

The changes in tonnage regulations which, as a result of a Bill now before Parliament, are likely to be made in the near future, will encourage manufacturers of propelling machinery to take the fullest advantage of modern design in the building of engines of smaller size and reduced weight. The most promising development in this field is the evolution of two-stroke turbo-charged Diesel machinery and a further stage was reached by the completion of the highest-powered engine of the type yet built—an 11,200 b.h.p. Burmeister and Wain unit—and by the decision to fit the first Doxford pressure-charged engine in a British tanker. Sea trials were also carried out of a 17½-knot cargo ship with a Japanese-built 11,200 b.h.p. B. and W. installation. There are now about sixty-five B. and W. turbo-charged engines, of 550,000 b.h.p., in service or on order, eight of the Werkspoor-Lugt type of 36,000 b.h.p., and five of the Stork single-acting design of 30,000 b.h.p., the total of all such engines being nearly eighty of approximately 620,000 b.h.p. Experimental work is being carried out on Sulzer and M.A.N. engines with turbo-charging, and in the lower-powered range a Sirron pressure-charged unit of 1,680 b.h.p. has just been completed, whilst tests are being made on engines of the Polar type. Eight pressure-charged high-powered engines of two designs have now been in service for periods up to more than a year and the results are uniformly satisfactory. The great advantages are unquestionable and there is, indeed, some reason for the claim that the large turbo-charged two-stroke Diesel engine represents the greatest advance made in this class of machinery during the past forty years. The increase in output for given cylinder dimensions and engine speed is not less than 33⅓ per cent and it is believed that with further experience it may rise in some cases to 50 per cent. This would follow the line of development of the four-stroke pressure-charged engine, which commenced by adding 33⅓ per cent to the output with normal aspiration, and now as much as 75 per cent can be obtained. The 11,200 b.h.p. turbo-charged engine mentioned previously has nine cylinders and develops slightly more power than the corresponding 12-cylinder engine of equal cylinder dimensions, which is 35 per cent heavier and 10 feet 10 inches longer. Builders will naturally proceed by stages but the ultimate possibilities are clear, and there is no room for doubt that they will finally be achieved. This development puts a new complexion on the question of competition between steam turbines and Diesel engines for high-powered single-screw ships. A 10-cylinder standard engine of the class of which many are already in service with smaller numbers of cylinders, has a normal output of 12,500 b.h.p. (normal rating) and, based on certified performance at sea, the fuel consumption is between 0·33 and 0·34lb. per b.h.p. per hr. with the advantage of additional economy due to the employment of exhaust gases to raise steam —an economy not available to steam machinery. The saving in fuel when compared with geared turbines of equal power and a 60 per cent higher consumption of the same grade of oil, would represent a yearly economy of about £50,000 in a tanker operating 300 days annually.—*The Motor Ship, March 1954; Vol. 34, p. 514.*

Large Oil-operated Reverse-reduction Gear

At the works of Modern Wheel Drive, Ltd., Slough, the largest oil-operated reverse-reduction marine gear has recently completed its trials. It is designed to take the drive from four engines with a total output of 8,000 b.h.p. and the reduction ratio is about 4½ : 1. The gear casing is of welded mild steel plate, carrying gunmetal bearings lined with white metal. Each engine operates an oil-controlled clutch, gear-cut on the outside and meshing with a second clutch, both driving pinions which mesh with the final wheel. Engagement of one or the other clutch causes the engines to drive the final wheel either in the ahead or the astern direction. The operation of the gear is controlled from a master cock which directs the oil into the appropriate clutch to give the required direction of rotation. Each engine can be isolated from the gearbox, and ahead, stop and astern orders may be carried out, irrespective of the number of engines driving the gear. The oil is supplied by an independent electrically-driven gear-type pump. The oil is drawn from the gearbox sump and is discharged through filters and a cooler to the main supply. A pressure-reducing valve is located in the oil spray pipe for the final drive.—*The Motor Ship, March 1954; Vol. 34, p. 551.*