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Marine Engineering and Shipbuilding Abstracts

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Lead Plating of Aluminium

The direct lead electroplating of aluminium and aluminium alloys in a conventional electroplating bath cannot be carried out successfully, as the resultant plating contains blisters and is not sufficiently adherent. Unfortunately, the various forms of treatment, such as zinc or iron plating, which are usually employed before the electroplating of aluminium with other metals, are not suitable for use prior to lead plating. The reason for this is that zinc and iron are attacked by the usual lead plating baths and, therefore, do not protect the aluminium from the corrosive action of the electrolyte. This results in the formation of blisters in the lead plating. It has therefore been proposed to carry out the electrodeposition of lead upon a nickel plating previously formed on the aluminium by electrolysis; however, in the most recent modification, the layer of nickel is chemically deposited. Contrary to expectation, the lead deposit is perfectly adherent and is free from blisters, while its thickness may vary from a few microns up to several millimeters. Before treatment, the aluminium or aluminium alloy articles to be lead plated are pickled, in order to form on the surfaces of the article roughnesses which, in the plating stage, facilitate the adherence of the lead. The articles are then treated with an acid solution containing a nickel salt capable of giving a deposit of nickel through chemical reaction with the aluminium. Electroplating is then effected in a bath of the usual type, containing a lead salt, such as lead perchlorate, lead fluoborate, or lead fluosilicate.—*The Engineers' Digest, September 1954; Vol. 15, p. 353.*

Fatigue Strength of Gear Teeth

Fatigue tests on many machine elements have shown an endurance limit for steel at a stress level corresponding to a life of ten million cycles. To run tests for more than ten million cycles is expensive, and few laboratories have carried out large-scale test programmes involving such long runs on gears. For this reason, data relating to the endurance limit for gears are sparse. Much of the fatigue data published on machine elements other than gears show a straight-line relationship on a log-log scale, and gear test data have likewise been expressed by a straight line in the SN diagrams. It has, however, been found that tests of spur-gear teeth show that the true diagram follows a characteristic pattern which is divided into three regions. The first of these, covering the range from 1 to 1,000 cycles, shows a very slight slope, while in the second region, from 1,000 to 300,000 cycles, a steeply sloping characteristic is observed; finally, in the third region, from 300,000 to 1,000 million cycles, the slope is again very slight. In the steep-slope region, a tenfold increase in cycles will drop the load capacity by about 35 per cent. Beyond one million cycles, the drop in capacity is less than 10 per cent when the cycles are increased tenfold. Recent tests have shown that in a pinion-tooth test, comparing shot-peened teeth with teeth not thus treated, extending over 200 million cycles, there was no evidence of a steep slope in the region beyond one million cycles. Shot-peening of some of the teeth of a pinion showed those peened to be about 35 per cent stronger. In this case, shot-peening was done by hand. Pinions peened by machine have shown gains in strength as high as 40 per cent.—*The Engineers' Digest, September 1954; Vol. 15, pp. 353-354.*

Nuclear Power Plants for Ship Propulsion

Two prototype submarine power plants are being developed in the United States, namely the Submarine Thermal Reactor (STR), which is already in operation, and the Submarine Intermediate Reactor (SIR) now under construction. Although these particular power plants are for submarine application, much of the experience accumulated during their development is applicable to ship-propulsion plants in general. Both the STR and SIR plants include the same basic components. Heat is generated in the reactor, and transferred to a heat-transfer fluid which carried the heat to a steam generator. The steam raised is then used to drive turbo-machinery in the conventional way. The basic difference between the STR and the SIR lies in the reactor coolant or heat-transfer fluid, which is a major factor in establishing the characteristics of all the equipment through which it flows. In the STR the liquid is pressurized water, whereas in the SIR it is liquid sodium. The nuclear, thermodynamic, and mechanical characteristics of the two systems are discussed, and the differences between them are summarized. The discussion of details is limited owing to security restrictions. Preliminary studies have been made of another type of nuclear propulsion plant comprising a gas-cooled reactor coupled to a closed-cycle gas-turbine system. Helium was selected as the working fluid mainly because it does not become radioactive; thus, the heat-transfer system requires no shielding and the total weight of the plant is thereby appreciably reduced. Absence or reduction of this shielding also leads to greater flexibility in the machinery arrangement. Gases other than helium have also been considered, and their properties for closed-cycle gas-turbine systems are shown in a table. Another table compares the main mechanical characteristics of nuclear plant cooled by water, sodium, and helium. A diagram of a gas-cooled reactor plant is given. It is pointed out that the preliminary study of the helium-cooled plant is based on certain assumptions which need experimental verification.—*Paper by F. E. Crever and T. Trocki, read at Mid-Winter Meeting of the American Institute of Electrical Engineers, 19th January 1954. Journal, The British Shipbuilding Research Association, August 1954; Vol. 9, Abstract No. 9229.*

New German Waste Heat Boiler

A recent article in *Hansa* describes how the firm of R. Noske Nachfolger has developed a waste heat boiler installation in which the boiler is built directly into the exhaust stream, a method which dispenses with flap valves of any kind, and saves weight and space on board, owing to the elimination of the exhaust branch mentioned above. Furthermore, these installations are self-regulating, all that is necessary being to set them to the required heat-supply temperature. In addition to hot water heating and supply, these installations can also be used for air heating. Two German motor vessels have already been equipped with installations of this type, and four other vessels are now being fitted with this equipment. The installation, shown schematically in Fig. 1, consists of a heat exchanger or waste heat boiler (*A*), which operates as a low pressure boiler, i.e. up to a steam pressure of 0·5 atmosphere gauge maximum (7lb. per sq. in. gauge approximately). It is supplied with the necessary heat from the exhaust gases of a Diesel engine. The steam generated flows via a manually operated control valve (*B*), and a thermostatically controlled regulator (*C*), to a heat exchanger (*D*) operating on the counter-flow principle. This heat exchanger delivers the extracted heat to a heating installation. The condensate return pipe to the waste heat boiler is connected to the balance tank (*E*), located above the boiler. This tank also serves as a stand-pipe. Below the whole system a boiler water receiver (*F*) is located, which holds boiler water in reserve and which also is capable of receiving the operational boiler water of the system when it becomes necessary to drain the installation, as, for instance, during the summer. This receiver is fitted with a water level indicator and an overflow pipe. A hand pump (*G*), with three-way cock and the necessary pipe connexions, serves to fill and drain the heating system. The balance tank (*E*) is connected to the receiver (*F*) via an overflow pipe (*H*). The receiver (*F*) is provided with a vent pipe (*J*). The waste heat boiler (*A*) is fitted with a water level indicator and a pressure gauge. If the exhaust heat installation is to be used exclusively for hot water supply, control valve (*B*) can be omitted, as the thermostatic regulator (*C*) is sufficient to ensure a constant temperature of the hot water supply. The low pressure steam generated in the waste heat boiler (*A*) is led through control valve (*B*) to the heat exchanger (*D*), valve (*B*) passing only that amount of steam required to satisfy heat demands, depending upon the outside temperature. The boiler heating surface is designed for maximum heat requirements and, with control valve (*B*) fully open, the water level comes to rest at the corresponding mark on the graduated scale, i.e. at about − 15 deg. C. (+ 5 deg. F.). The installation operates with barely measurable steam pressure and attains its highest output when the entire heating surface is effective. If less heat is required, as, for instance, when the outside temperature is − 5 deg. C. (+ 23 deg. F.), the necessary regulation is effected by throttling the steam in control valve (*B*) to such a degree that the water level falls to the corresponding mark on the scale. The resulting reduction in heating surface determines the lower output of the whole installation. Although steam pressure is somewhat higher, the quantity of steam is reduced. The water displaced from the boiler flows to the balance tank (*E*), and the increased height of the water column produces a pressure which is equal to that of the steam. The water level and hence the boiler output are self-regulating, and while this regulation is not very sensitive with regard to the desired temperature of the heating medium, it is usually quite satisfactory for the purpose. Fine regulation, i.e. control of the heating medium to within ± 1 deg. C. (± 1·8 deg. F.) is provided by thermostatic regulator (*C*), which is fitted with a scale and which, in addition, safeguards the installation against overheating in the case of low heat extraction or shut-down. A breather provides the necessary air balance when the installation is shut down or put into operation.—*The Shipping World, 29th September 1954; Vol. 131, p. 354.*

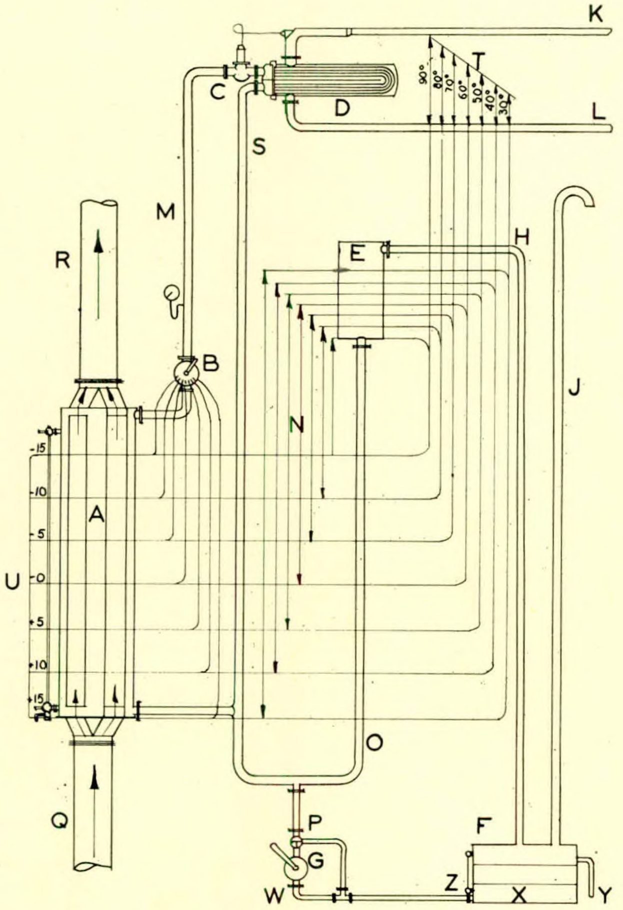


FIG. 1—*Diagram of installation*

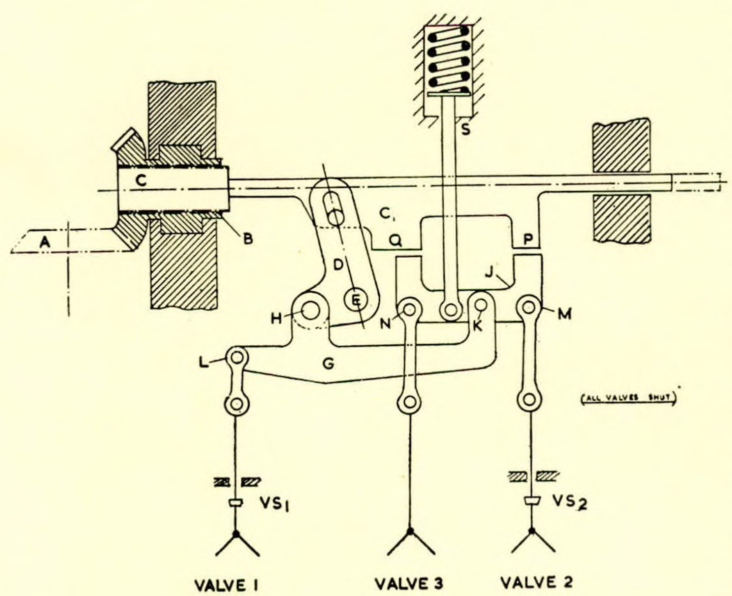
A.—Waste-heat boiler; B.—Steam control valve; C.—Thermostat control; D.—Heat exchanger; E.—Balance tank; F.—Receiver; G.—Hand pump; H.—Overflow pipe; J.—Vent pipe; K.—Supply pipe; L.—Return pipe; M.—Steam pipe; N.—Steam pressures corresponding to water levels; O.—Balance pipe; P.—Three-way cock; Q.—Exhaust gas inlet; R.—Exhaust gas outlet; S.—Condensate return pipe; T.—Supply temperature; U.—Heating surface at *x* deg. outside temperature; W.—Pipe for filling, drainage and make-up water; Y.—Overflow; Z.—Water level.

***Nestor* Class Ships**

In a paper presented to the Student Section of the Liverpool Engineering Society, Mr. G. M. Jordan and Mr. R. P. Williams gave additional details of the machinery installed in the three *Nestor* class ships of the Blue Funnel Line. In these ships, the propulsion machinery consists of a set of three-cylinder steam turbines developing 8,000 s.h.p. maximum at 112 r.p.m. and supplied with steam at 600lb. per sq. in. pressure and 950 deg. F. total temperature by two Foster-Wheeler D-type watertube boilers. The recent adoption of the burning of boiler oil and the supercharging of marine Diesel engines now make Diesel propulsion more economic than steam for powers in the region of 8,000 s.h.p. However, at the time of conception of the *Nestor* design, boiler oil burning and supercharging were not in use. By careful design to reduce space, a saving of about 400 tons of cargo space was effected as compared with the equivalent Diesel machinery. The resulting increase in earning capacity coupled with the advantages of ease of maintenance, reduction in lubricating oil consumption and reduction in the carriage of spares were sufficient to outweigh the loss due to an increased fuel bill. The main boiler steam drum, single water drum and waterwall header are joined by tubes forming a single furnace fired by four oil-burners. The boilers are operated under balanced draught. Some design particulars are given below: —

Pressure at superheater outlet...	625lb. per sq. in.
Temperature at superheater outlet	950 deg. F.
Normal evaporation (per boiler)	25,000lb. per hr.
Maximum evaporation (per boiler)	30,000lb. per hr.
Temperature of feed at economizer inlet (normal output)	245 deg. F.
Temperature of feed at economizer inlet (maximum output)	230 deg. F.
Efficiency at maximum output	88·5 per cent

It was a requirement that the steam temperature at the boiler outlet should not exceed 750 deg. F. for astern operation and 950 deg. F. for ahead. The final steam temperature is controlled by dampers operated by push-button control at the manœuvring platform and which can also be operated by local hand control. The dampers are arranged to allow the gases to bypass the final bank of a three-pass superheater at the same time operating an air damper to allow combustion air to pass over the outside of the desuperheater tubes. During ahead operation the furnace gases are allowed to pass over the third pass of the superheater and the combustion air bypasses the desuperheater. Steam is led from the intermediate superheater bank to the desuperheater and thence to the final pass of the superheater, finally emerging at a temperature of 950 deg. F. For astern operation at reduced superheat the furnace gases are allowed to bypass the final superheater and air is allowed to flow over the desuperheater and cool the steam to 750 deg. F. The turbine employs a novel form of control for sequential operation of nozzle groups in the h.p. turbine: the valves open in the sequence 1-2-3; valve 1 is sufficient for powers up to 5,000 s.h.p., valves 1 and 2 for 7,250 s.h.p. and valves 1, 2 and 3 for 8,000 s.h.p. The manœuvring valve handwheel is geared to rotate the block *B* and in the direction to open moves *C* to the right and in so doing, turns the lever *D* about *E* thereby lifting the lever *G* which is pivoted at *H*. As the point *L* is nearest to *H*, valve No. 1 lifts first and, when fully open, engages a valve seal stop Vs_1. The first $\frac{1}{4}$-in. of travel of valve 1 opens a bypass valve before commencing to open the main valve. In the event of excessive friction in the No. 1 valve movement, there would be a tendency to lift lever *J* to which Nos. 2 and 3 valves are coupled. This action is prevented by the interference of faces *P* and *Q* which are normally separated from the lever by a small clearance. In normal condition, without such excessive friction, further movement of the handwheel causes lever *J* to be lifted, and when this point has been reached, the interfering face *P* has moved clear of *J* and valve No. 2 opens. Valve No. 3 cannot move out of sequence as the face *Q* has not yet moved clear of *J*. Finally, movement of the handwheel, having already opened valves 1 and 2 to the limit imposed by the stops Vs_1 and Vs_2 begins to open valve No. 3 as the face moves clear. Lever *J* is spring-loaded to ensure smooth running and to overcome slight frictional changes which might otherwise alter the opening sequence.—*The Marine Engineer and Naval Architect, September 1954; Vol. 77, pp. 322-324.*



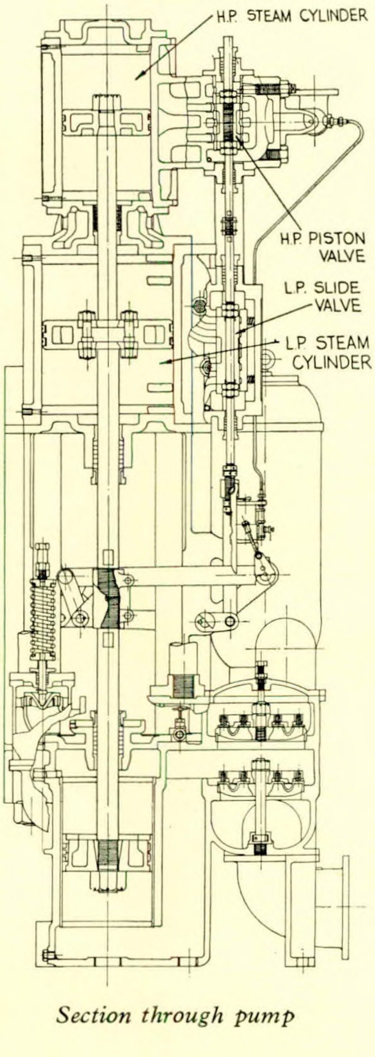
Sketch showing mechanism of the sequential-opening nozzle control valves which admit steam to the h.p. turbine

Cavitation Erosion of Diesel Cylinder Liners

For the past few years, large ocean-going motorships have been troubled with a peculiar form of corrosion occurring on the water side of their Diesel-engine cylinder liners. In practically all the cases, nature of the damage is as follows: 1. Corroded metal has a honeycombed appearance, but the surface remains substantially free of corrosion products. 2. There is usually an irregular but sharply defined boundary between affected and unaffected metal. 3. Centrelines of areas of greatest attack are always at right angles to the centreline of the crankshaft, and attack on one side is usually more severe than on the other. Also, the attack is deeper at the bottom of the liner—the looser end. It is often completely absent in the centre section, but occurs again, although to a lesser degree, at the top. 4. Water-inlet and outlet passages and so-called "hot spots" do not seem to be controlling factors in this particular form of corrosion. Such forms of attack have been known to occur in areas of unusually high water velocity or turbulence, but improvements in flow arrangements have either eliminated or effectively reduced them. 5. Rate of corrosion has been amazingly high, $\frac{1}{16}$in. in 900 hours, or the equivalent of 3 inches per year. This rate rules out ordinary forms of corrosion resulting from stray electrical currents, oxygen depolarization or hydrogen evolution. 6. Damage is not confined to any one design or make of engine. It may appear in only a few engines of a particular type and sometimes only in certain cylinders of a particular engine. Cavitation erosion is a complex effect produced by the simple mechanical pounding of the collapsing vacuum bubbles combined with chemical or galvanic action—the ordinary forms of corrosion—of the liquid involved. Perhaps the mechanical effect is all-important, but it is more probable that both are contributing factors with cavitation preventing formation of a protective film over the metal surface. From the practical point of view, it would be easier to stop corrosion than to stop cavitation. The latter step requires elimination of mechanical vibration, a job that is often difficult where ship's Diesels are concerned. Some engineers have tried to arrest cavitation erosion attack, by using protective coatings such as baked phenolics. These have not performed well and have "blasted off" in as little as three months. In such cases, the exposed bright metal started pitting. Since corrosive quality of the water could not affect a phenolic coating, this leads to the belief that vibration is a definite factor. Considering cavitation erosion as a combination of several forms of attack seems to be the best method of arriving at preventive measures. Practical experience has shown that the following steps will help to reduce damage to a great extent: 1. Reduce vibration by any practical means, such as re-designing liner supports to introduce vibration-damping effects. 2. Treat water with the proper corrosion inhibitors in the right concentrations and adjust *p*H to recommended value. Begin this as soon as engine is put into service. Remember, any type of corrosion is easier to prevent than it is to cure. 3. If possible, use more resistant liner metal. Hardened cast iron liners appear to respond better than soft plain cast iron to common corrosion inhibitors. 4. Reduce opportunities for vacuum-cavity formation: (*a*) keeping water temperature as high as practical; (*b*) keeping water pressure as high as possible; (*c*) introducing air bubbles to serve as cushions against vacuum-bubble collapse. This is common practice in hydraulic turbines and is accompanied, of course, by the use of corrosion inhibitors.—*F. J. McCloud, Marine Engineering, August 1954; Vol. 59, pp. 50-51, 80.*

Large Cargo Pumps

The large supertankers now being delivered must have pumping capacity commensurate with their size, if extended turn-round periods are not to occur. In many cases pumps of up to 750 tons per hour water rate are called for, with a consequent heavy demand for steam. Any economies in steam consumption such as can be offered by tandem compound operation and improved valve gear are therefore likely to be welcomed by owners and charterers alike. The pump is seated upon lugs cast on the valve and cylinder housings. Each valve box is a separate close-grained nickel iron casting fitted with a renewable gun-metal liner. The valve lids and seats are of bronze, with phosphor-bronze springs, and the cast iron plungers have gun-metal rings, fitted with laminated plastic rings for ease in overhauling. The pump rods and valve spindles are of cold rolled manganese bronze. As the illustration shows, a Y-piece between the valve boxes connects the suction branches to the suction crossover line in the vessel, while the discharge valve boxes are inter-connected by a loop, and the discharge line can be attached to either side of the pump. The steel piston rods are connected at the low pressure piston, the latter taking the form of a sandwich plate between the bolted flanges. Both rods may be withdrawn between the l.p. and pump cylinders after removing the high pressure piston. Self-adjusting metallic packing is fitted in internal boxes. Steam is admitted to, and released from the h.p. cylinders through piston valves, whence it travels by external transfer pipes to the l.p. slide valve casings. The pumps are supplied with all necessary drain and air cocks, mechanical sight feed lubricators, pressure and vacuum gauges, and a cast iron air bottle. The oil relief valve discharges internally to the suction side of the pump. These pumps are made in five sizes, covering a range of capacities from 340 tons of water per hour at 35 double strokes per minute, to 750 tons per hour at 35 to 29 double strokes per minute.—*The Marine Engineer and Naval Architect, September 1954; Vol. 77, pp. 329-330.*



Section through pump

Salvage of Sunken Barge

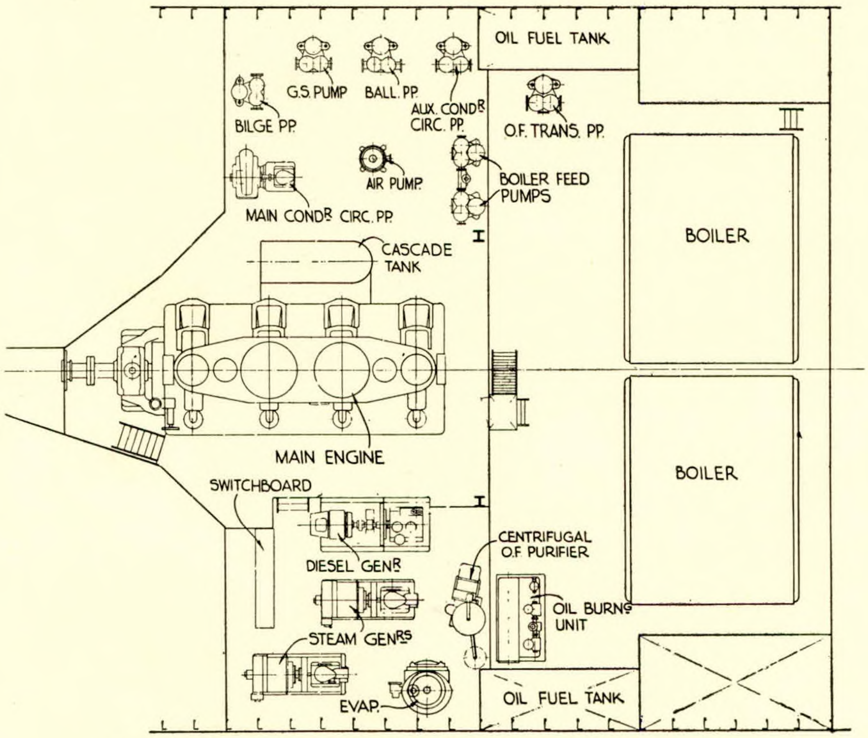
A unique instrument known as the "Guillotine" was employed by the Corps of Engineers, U.S. Army personnel in removing and salvaging the wreck of a 1,735 net ton seagoing barge *Rockville* which sank in 1952. The *Rockville* crashed into a pier opposite the United Nations Building and sank in approximately 65 feet of water at its bow and 28 feet at the stern. The use of conventional salvage methods such as divers and cutting torches was found to be exceedingly difficult because of the velocity of the current and turbidity of the East River at this point. Therefore, the "Guillotine" was employed to facilitate the removal of the wreck. The "Guillotine" is a 44-ft. length of the heaviest standard structural steel "H" beam available and weighs some 9 tons. The beam is tapered at one end and wings have been attached to give a cutting edge with a "bat wing" appearance. This guillotine-like arrangement swings from the crane of the derrick barge *Babcock* alongside the sunken barge and is dropped in pile-driver fashion a height of 5 to 8 feet. The "bat-wing" shaped cutting edge creates an effect similar to a giant can opener which is used to sectionalize the wreck. These sections are then raised to the river's surface by means of a bucket and cable sling.—*Marine News, September 1954; Vol. 41, p. 57.*

Turkish Steamship

The Moss Vaerft and Dokk A/S are completing a series of handy-sized 4,050 tons d.w. cargo vessels, the latest of which they have delivered to the Turkish State Steamship Company. This ship, named *Eskisehir,* was originally laid down as the *Bjorgsund* for Hakedal and Bjorge of Oslo, but was transferred while under construction to her new owners. She is a shelter-decker with a half-raised forecastle and poop with deckhouse. The principal dimensions are as follows:—

Length overall	337ft. 3in.
Length between perpendiculars...	314ft. 0in.
Moulded breadth...	48ft. 6in.
Moulded depth to main deck ...	21ft. 6in.
Deadweight capacity on 20ft. 9in. draught	4,050 tons
Cargo capacity (bales)	238,241 cu. ft.
Cargo capacity (grain)	266,784 cu. ft.
Gross tonnage	2,456 tons

The double bottom space extends the whole length of the ship and is used for fuel and water ballast, while the deep tanks for fuel are arranged in the wings of the tunnel beneath No. 5 hold. There are five holds each served by two 5-ton derricks with an additional 25-ton derrick for No. 2 hold. The main engine is a size 8 Fredriksstad Steam Motor, built under licence by the Moss Vaerft and Dokk Company. It is of the open double compound type, the h.p. cylinders have a bore of 425 mm., the l.p. cylinders are 1,015 mm. in bore and the stroke is 930 mm. The designed output is 1,900 i.h.p. at 103 r.p.m. but 2,600 i.h.p. can be obtained. Two sets of Stephenson link gear are required for the four cylinders. Each compound unit has one piston valve which admits and releases steam to and from the h.p. cylinder. As the corresponding l.p. crank is arranged at 180 degrees, and the cylinder is of true uniflow type, no l.p. valve is necessary. The two halves of the four-cylinder engine are, of course, arranged at 90 degrees to one another to ensure even balance and rapid manœuvring. This particular unit was built in the engine shops at Moss and installed in one piece by floating crane. Steam is generated at 220lb. per sq. in. and 585 deg. F. in two oil-fired single-ended Scotch boilers, each having three furnaces and a heating surface of 2,650 sq. ft. They are fitted with superheaters and desuperheaters, the latter for supplying the steam-driven auxiliaries with steam at 430 deg. F. Most of the auxiliary pumps are steam-driven and electrical power is supplied at 110 volts by two 25 kW steam generators and a 20 kW Diesel generator.—*The Marine Engineer and Naval Architect, September 1954; Vol. 77, pp. 319-320.*



The plan of the engine room is of unusual shape and the boilers are fired from the after side

Manually Guided Submerged-arc Welding

Submerged-arc or submerged-melt welding is the process of electric welding beneath a blanket of granular welding composition. The granular welding composition provides the special operating characteristics that distinguish the process. It permits the use of higher current densities on welding electrodes than could be used successfully in earlier methods and performs the function of cleansing the weld puddle and protecting it from atmospheric contamination. The melted base metal and welding electrode metal are completely submerged beneath the melted welding composition; hence the name "submerged melt welding". A manual submerged-melt welding installation has the same elements found in a completely mechanized installation except that a travel carriage is taken over by the operator whose function during welding is to guide the welding head over the seam at a reasonably constant speed. Manual submerged-melt welding equipment combines the flexibility of hand operation with the well-known advantages of quality and economy of submerged-melt welding. It provides the manual welding arc with a tool capable of producing high quality welds at low cost, with greater operator comfort, and generally with less skill than required with coated electrode open-arc welding. Manual submerged-melt welding apparatus consists basically of a portable drive unit and flexible wire conduit terminating in a holder or guide at the welding end. With this machine, which is the most common type, the wire feeding unit includes a wire feed control, drive motor and wire reel. The welding composition is held in a small container or hopper which is attached to the wire feed nozzle at the end of the flexible conduit. This nozzle and hopper assembly is held in the operator's hand and the granular material flows by gravity from the hopper end to the weld zone. The welding wire is driven through the nozzle which is insulated from the hopper, and the granular material surrounds it as it emerges from the tip. The hopper holds 3 to 5lb. of welding composition. In some apparatus, the flexible conduit with its nozzle and hopper assembly may be used separately from the portable drive unit by attaching it to a machine welding head. Such an arrangement provides for both fully mechanized and manually guided semi-automatic welding with minimum investment. Another type of machine contains approximately the same components but in addition incorporates a large pressurized hopper, mounted with the wire feeding unit, which feeds the welding composition through a flexible tube to the weld zone. This hopper holds 75lb. of material and, therefore, requires only infrequent filling. The rated capacity for all those commercially available machines is in the neighbourhood of 800 amp.—*R. A. Kubli, The Welding Journal, September 1954; Vol. 33, pp. 835-841.*

East German Shipbuilding

Shipbuilding in the Soviet Zone of Germany has been promoted so heavily by the Russian Occupation Authorities that the present construction capacity of the area has reached 800 per cent of the pre-war level. Capacity is still growing and there are no hints that the trend will reverse in the near future. Most of the gain must be credited to the building capacity for greater cargo ships and some special types of vessels such as patrol boats and fishing craft. There are now sixteen important plants in the Soviet Zone of Germany. They have been developed from a greater number of smaller shipyards, none of them having been of international importance before the war. Almost all of the plants are so-called "People's Enterprises", which means that they are owned and operated by the government—which in turn acts under tight control exercised by Russian officers. Private enterprise shipbuilding in the area is now almost zero. Like all industrial activities in the Soviet Zone of Germany, shipbuilding, too, is following a master plan. The main features of the plan have never been published but it is now evident that it has been designed for the following purposes: 1. To build and organize an East German Merchant Marine which would mainly operate on a China run and between East German and other East Bloc ports. 2. To supply patrol boats and other smaller vessels for the "Sea Police", which is the nucleus of a small East German Navy. 3. To build repair facilities into a capacity big enough for repair orders from East Germany, Soviet Russia and other East Bloc countries. There are no plans to launch any passenger ships or tankers. And there are no plans to export any vessels into the West, according to the latest information which is available.—*G. Genschow, Marine News, September 1954; Vol. 41, pp. 14-15*

Shipboard Maintenance Aboard Great Lakes Vessels

As a group, the steamers that span the Great Lakes with coal, iron ore, grain and limestone are exceptionally well maintained. While a bit unusual in appearance to deep water men, the low, flat, extremely long-decked (sometimes two city blocks long) Lake steamers present the usual shipboard maintenance problems. Above deck there is an abundance of paint to be kept clean and periodically stripped. Below, there are engines and engine rooms to be cleaned and pre-heaters and similar water circulating equipment to be descaled. These ships carry solution-lifting guns for cleaning decks, fore and aft cabins, engine rooms and deck machinery. Another maintenance job handled on many of the Great Lakes steamers with Solution-lifting Steam Guns is paint stripping. Those Lake boats that carry steam guns for cleaning find the guns ideally suited for applying specialized paint stripping solutions over paint to be removed from decks, bulkheads, anchor chain compartments, etc. In the past, stripping hull paint from Lake carriers was of necessity confined to the lay-up season. On the Lakes this means the months between November and March. Even the milder of these months are so cold that it is extremely difficult to keep a stripping solution hot enough to strip a 500-ft. hull. The metal is cold and as soon as the solution hits it the solution in turn begins cooling rapidly—with an accompanying loss in stripping effectiveness. Stripping during the warm months is not feasible, because every available boat must earn its keep during the time the Lakes are free of ice. This is the background for a decision to attempt to strip during its trip a Lake carrier—532ft. long, 32ft. deep, and with a 58ft. beam—carrying about forty coats of red lead. Actual procedure involved suspending over the side an applicator made of pipe in the form of a "rake". The pipe section that formed the cross-piece of the "rake" was perforated to flow stripping solution over a wide horizontal area. A hose connected to the handle of the "rake" carried the stripping solution from the solution drum and doubled as a means of controlling the "rake" suspended over the side. The stripping solution was kept hot in the drum by a Solution-lifting Steam Gun. Hot solution was fed by gravity through the hose to the rake, from which it flowed down the sides of the hull. All stripping was done from the deck and at no time during the operation was it necessary for personnel to go over the side. This paint stripping method, known as Hot-Flow-On Stripping, proved to be very successful in its initial trial for removing paint while the vessel was under way. The paint was effectively removed and the cost of achieving its removal was less than any previous attempts at chemical stripping.—*F. Schnitzler, Marine News, September 1954; Vol. 41, pp. 21, 22, 46.*

Steam-heated Pipe

For handling of tar, pitch and other highly viscous substances an extruded aluminium tube is now available which is equipped with a built-in steam line which prevents certain products from turning to solids or crystallizing in the pipeline. Normally, it is necessary to use external steam jackets or lines to supply this heat. The tube has a main circular bore below which is a crescent-shaped passageway for steam which heats the solution travelling in the passage above.—*Aluminium News, August 1954; p. 1.*

Pleuger Rudder for British Cable Ship

The *Ocean Layer* ex *Empire Frome,* which is at present in the hands of R. S. Hayes (Pembroke Dock), Ltd., for conversion to a cable ship for her owners, Submarine Cables, Ltd., London, is likely to be the first British vessel to be fitted with a Pleuger active rudder. Built by Flensburger Schiffsbau-Gesellschaft, in 1948, the vessel has a length of 343 feet, a breadth of 50ft. 9in., and a depth of 18ft. 7in. Her gross tonnage is 2,774 tons. It is understood that the conversion to a cable ship will take about twelve months. During her conversion, the *Ocean Layer* will be fitted with a Pleuger rudder which will provide the vessel with additional manœuvring qualities. Electrical power for the submersible motor driving the three-bladed propeller built into the centre of the rudder blade will be provided by a 500 kVA. alternator coupled to a 600 h.p. Lister-Blackstone ERSMA8 vertical Diesel engine. Manœuvring of the rudder motor will be by lever remote control coupled to the engine—as a servo variable speed governor. Two 187 kVA. alternators driven by four-cylinder Lister-Blackstone type ERSMA4 vertical Diesel engines will provide current for general requirements. Pistons, bearings, valves, and all parts of the Lister-Blackstone engines will be interchangeable on all three sets.—*Shipbuilding and Shipping Record, 2nd September 1954; Vol. 84, p. 310.*

Separator for Producing Dry Steam

This invention relates to a separator for producing dry steam from a mixture of steam and entrained water, and in particular to a separator for use with evaporators used for the distillation of sea water for the production of drinking water. In this case, any water entrained in the steam passing from the evaporator to the condenser will carry with it dissolved salts which decrease the potability of the condensate. The combined evaporator and separator shown in Figs. 7-9 comprises a generally cylindrical casing (10) with top and bottom end-plates (11) and (12) and an intermediate plate (13) which divides the interior of the casing into heating section (14) and a separating section (15). The heating section delivers steam through the outlet pipe (16) to the top tangential inlet connexion (17) of the separator section (15). Depending from the top end-plate (11) is a steam outlet conduit (18) having a laterally-disposed window-like opening (19) adjacent to its lower end. The conduit (18) is closed off at its bottom end by a plate (20), which is of substantially greater diameter than the conduit, and which also forms the bottom of the window-like opening (19). This opening is streamlined by an apron (21), as in the previously mentioned construction, the apron serving to reduce entrances and to afford some measure of protection against water creeping into the opening. Suspended immediately below the plate (20) by a short collar (22) is a conical anti-flash plate (23), extending nearly to the cylindrical casing wall but spaced from it to leave an annular gap (24), through which water may pass into the receiving pool below. Below the plate

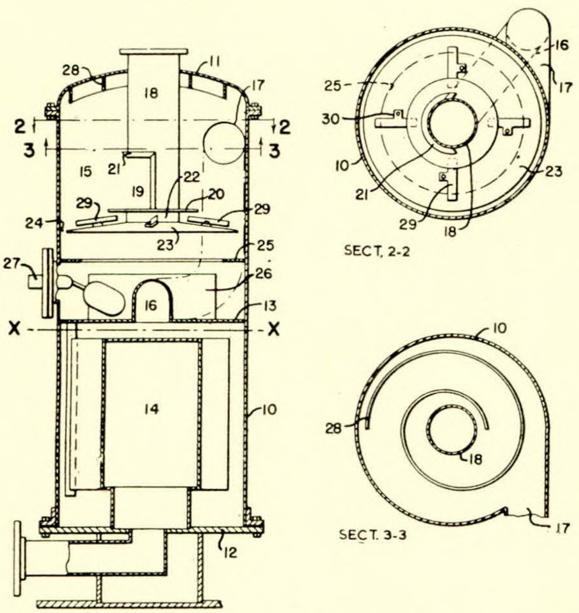


FIG. 7 *(left)*; FIG. 8 *(top right)*; FIG. 9 *(bottom right)*

(23) is a flat ring (25) secured to the cylindrical wall of the separator chamber. This ring prevents water from being thrown upwards through the gap (24). The chamber or receiving pool between the ring (25) and the transverse plate (13) has anti-whirl vanes (26) in crossed formation and a float valve (27). Secured to the under side of the top end-plate (11) is a fin (28), which projects downwards and is formed as a spiral progressing outwards in a direction away from the tangential inlet (17). The purpose of this fin is to prevent water from reaching the outside of the outlet conduit (18). It accomplishes this result by leading out towards the casing wall (10) such water drops as may be splashed against the top end-plate (11) from the inlet (17). In order to increase the efficiency of the separator, it is necessary to keep all surfaces directly associated with the window (19) as nearly completely dry as possible. The fin (28) accomplishes this purpose in the upper part of the separator. To keep the bottom part of the outlet conduit (18), the plate (20), and the top of the anti-flash plate (23) essentially dry, it is necessary to reduce, but without turbulence, the angular velocity of the steam in the lower part of the separator. This is accomplished by four anti-whirl blades. The four vanes (29) also reduce the back pressure of the device by reducing the whirling of the steam before it passes into the outlet conduit. Additionally, the slant given to the vanes creates a down draught over the surface of the plate (23), so that any water droplet entrained in the whirling steam will be thrown down upon the plate and thus prevented from reaching the window (19).—*British Patent No. 714,953, issued to The Maxim Silencer Company. Application in U.S.A. made 20th March 1952. Complete specification published 8th September 1954. Engineering and Boiler House Review, November 1954; Vol. 69, p. 359.*

Radiography in the Shipyard

The number of shipbuilders applying radiography as a means of inspecting welded construction has grown appreciably in recent years, and the biggest increase in the use of this method of non-destructive testing was, of course, due to the rapid expansion of shipbuilding and the greater use of welding in the early years of the last war. It is well known that radiography as a means of weld examination is accepted by all inspecting authorities, and it is of interest to recall that the Admiralty were one of the pioneers in the use of this method. The manufacturers of X-ray equipments have also contributed to general progress by building a number of robust, compact and manœuvrable X-ray units for the examination of welded butts and seams in ships. The use of radioactive isotopes has also greatly increased since the war and certain shipyards have favoured the use of this method of radiography, mainly due to the convenience and cost factors. Obviously, however, the particular method to be employed depends on the circumstances, and both X-ray equipment and radioactive isotopes can be used as complementary methods. It would obviously be impracticable to carry out a 100 per cent radiographic examination of a ship's welded butts and seams. Therefore it will be appreciated that only welds in areas to be subjected to maximum stress and strain can be included in the syllabus of inspection. In an Admiralty ship to be radiographically examined, for example, depending on the tonnage of the ship under surveillance, a predetermined number of radiographs are taken of the welding in the stressed areas in accordance with a set ratio. These radiographs are distributed between the upper deck, shell, lower decks, bulkheads, etc. The areas of primary importance for radiographic examination are:—*(a)* Flat, vertical and rider keels throughout length of ship; *(b)* sheerstrake, including break of forecastle; *(c)* stringer plates; *(d)* main and weather decks; *(e)* turn of bilge; *(f)* transom. These areas are subjected to the following stresses:—*(1)* Hogging and sagging; *(2)* panting stress; *(3)* rack stress (transverse distortion); *(4)* propulsion stress: *(5)* stresses due to excessive weights; *(6)* longitudinal bending stress in still water.—*C. J. Storey, The Welder, No. 118, 1954; Vol. 23, pp. 146-149.*

Small Boiler for Shipboard Use

The smallest 250 h.p. fire-tube steam generator ever put into production has been built by the Cyclotherm Division United States Radiator Corporation for shipboard use. First installation of this new 250 h.p. boiler has been made aboard the *Sinclair Chicago,* a Diesel towboat owned and operated by the Sinclair Refining Company. The Cyclotherm generator requires an installation space only 9ft. 5in. long, 6 feet wide, and 6ft. 8in. high—about one-fifth the cubic volume needed to install competitive boilers of equal rated h.p. A typical competitive boiler is 20ft. 4in. long, 7ft. 3in. wide, and 9ft. 9in. high. Aboard the *Sinclair Chicago* the Sinclair engineers use steam to gas-free barges, heat Bunker C oil, steam-out and wash tanks. Like other Cyclotherm units, ranging in size from 18 to 500 h.p., the MC 250 burns oil or gas or a combination of oil and gas. For marine use, it can be adapted to burn the same Diesel fuel that propels the vessel. The MC 250 is designed for 15 to 200lb. per sq. in. pressure. It delivers a maximum guarantee of 7,500 pounds of steam per hour with a normal operating capacity of 6,250 pounds per hour. The boiler is equipped with a modulating unit that operates from 30 to 100 per cent of capacity. Cyclotherm has packed all this power into the small unit through its patented Cyclonic Combustion principle, coupled with a two-pass design. A revolving spiral vortex of flame in the combustion chamber travels at a rate of approximately 290ft. per sec. All fuel is consumed in the combustion chamber.—*The Motor Ship, October 1954; Vol. 35, pp. 293-297.*

Multiple-engine Diesel-electric Ship-propulsion

The use of electric equipment in a Diesel propulsion system for ship drive serves as a non-mechanical means of connecting one or more relatively high-speed engines to an inherently slow-speed propeller. Of the many multiple-engine D.C. Diesel-electric-propelled ships now in operation, almost all use the series loop system wherein the generator armatures are connected in series; likewise, if there are multiple motors driving the propeller in tandem or through a gear unit, the motor armatures are also in series in the loop circuit and are interspersed between generator armatures in order to keep the ground potential to a minimum. In most cases the selection of the series connexion is the proper one and can be justified by a simple analysis. However, there is a strong tendency to propose the series loop system without careful study. If such a study were made, it would in some cases lead to the conclusion that the generator armatures should be connected in parallel. The series loop circuit has many desirable features. One of the most important is that, up to a limiting value of motor field weakening, it is possible to utilize the full rated horsepower of all available engines regardless of the number of engines in operation; another is that load division between the various engines is satisfactory regardless of minor differences in engine speeds or generator voltages. With the series system, the only effect of such differences is a proportional difference in the load carried by the generators and their respective engines, whereas parallel operation of generators with the variable voltage system requires much more precise automatic engine governing, matching of generator load characteristics, and manual vernier adjustment of engine speed or generator voltage to ensure equal load division. With generators connected in series, a reduction in the number of engines in use brings with it a proportional reduction in voltage applied to the motor terminals and this in turn, if uncompensated for, results in a proportional reduction in motor speed. Since the power required to drive most propellers varies very nearly as the cube of the speed, it is desirable to weaken the motor field to such an extent that the motor will return to a speed which will utilize the full available engine horsepower. Field weakening of the motor is effective over a wide range; however, reduction to less than about 40 per cent field flux is usually not permissible since it can result in bad commutation and poor motor performance. In the example chosen it will be seen that the minimum flux condition will actually limit the available shaft horsepower when less than 50 per cent of the engines are in use. In contrast with the series case, for generators connected in parallel a reduction in the number of engines in use does not reduce the voltage applied to the motor terminals and, if uncompensated for, the motor will attempt to maintain full speed. This, of course, will result in overloading of the connected generators and engines. Therefore it is necessary to increase motor field flux sufficiently to slow down the motor and thus match required propeller horsepower with available engine horsepower or, if the motor field cannot be further increased, it becomes necessary to decrease the generator voltage. Generator voltage can be decreased in one of two ways. Engine speed can be reduced while maintaining full generator field flux. However, this results in less available engine horsepower output and, therefore, less shaft horsepower output than a direct proportion to the number of engines in operation. Whether or not this decrease in shaft horsepower is of prime importance is a debatable question. In any case, the reduction can be readily calculated. Generator voltage can also be decreased, while maintaining full engine speed, by reducing the generator

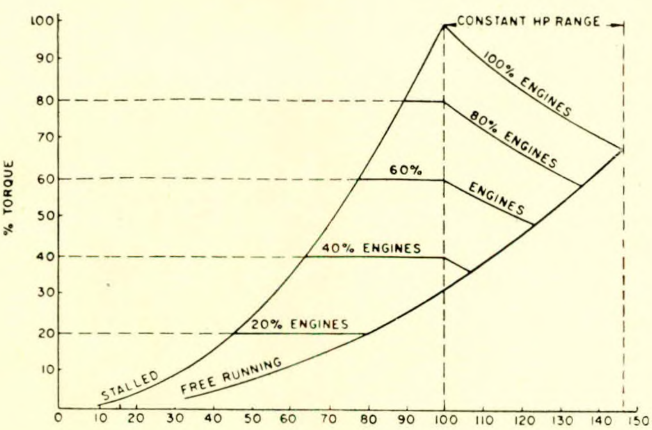


FIG. 2—*Propeller speed-torque curves from free-running to stalled ship conditions based on an arrangement of five generators and one motor; a constant horsepower motor speed ratio of 1·46 to 1, and a motor field weakening limit of 40 per cent flux*

field flux. This method has the advantage of utilizing the full engine horsepower available. However, to take advantage of the full engine rating and to maintain full generator output, it is necessary for the generator to carry a current overload to compensate for the reduction in voltage. If this current overload is not carried to extremes and is only for short-time operation, the generator can carry the increased current at a moderately increased temperature rise without damage. In most cases where direct current is selected for ship drive, one of the reasons for the selection is its ability to provide increased torque at low speed as illustrated in Fig. 2. For this reason, most D.C. ship propulsion motors are designed to have a considerable range over which the speed can be varied by field control. For purposes of comparison, an example has been chosen which represents an installation consisting of five generators driving a propulsion motor. The weak field motor speed is 46 per cent greater than the full field or base speed. If a vessel with parallel-connected generators is running in open water without a tow, it is possible to reduce the motor speed by field strengthening and to utilize full engine output with as few as two of the engines in operation without any reduction in generator voltage. With only one of the engines in operation and generator voltage reduced to 85·5 per cent, full output of the engines can be utilized by carrying a 17 per cent generator overload current. If the vessel were to have series-connected generators with the motor flux necessarily limited to a minimum of 40 per cent, then only 81·3 per cent of the connected engine rating can be utilized with two of the engines in operation; with one engine in operation, only 20 per cent of its rating can be utilized. Thus it can be seen that, for free running, the series-connected plant operates at a disadvantage when under one- and two-engine operation. Actually this limitation could be considered academic and of no practical importance.—*J. A. Wasmund, Applications and Industry (A.I.E.E.), July 1954; No. 13, pp. 135-138.*

Strain Gauges on Dutch Slipways

Automatic potentiometer-apparatus are suited for the measuring and recording of surface stresses and strains arising in conjunction with mechanical loading of structures, tools or other auxiliary installations such as large steel structures, rollers, presses, cranes and dredging plant, and for ascertaining the load on axles, winches, etc. A description follows of an application of such apparatus at the shipyard "Oranjewerf" of Verschuren and Co's Scheepswerf en Machinefabriek N.V., in Amsterdam. At this shipyard an automatic potentiometer is used in a safety device against chain-failure on a new slipway recently put into service. Strain gauges are employed as measuring-elements to determine the mechanical surface stresses and strains in the tension rods of chain-wheels. The values found in this way are a measure for the forces arising in the chain. The slipways have been designed for the sideways launching of ships, and consist of seventeen double concrete strips 65 m. long, of which 27 m. are above water. The strips are about 5 m. apart and provide building capacity for ships about 95 m. long and with a maximum weight of approximately 2,500 tons. The slipway carriages run on two sets of three wheel-frames with two wheels each, mounted in hinged bogies (see Fig. 2). This type of construction has been adopted to ensure maximum uniformity in the way the load on the carriages is distributed over the foundation of the concrete strips (timber piles), which makes for a higher permissible load. The carriages run on crane tracks; the space between track and concrete has been filled with hardwood. In the construction utmost care has been devoted to the elimination of differences in height between the strips, because such differences may cause the load on ship and slipways to assume inadmissible proportions. The carriages are drawn by means of a Gall's chain moving over a chain disc which freely rotates on a shaft. The final gearwheel of the winch is firmly connected to the chain disc. The chain moves at a maximum rate of approximately 0·4 m. per min. There are various

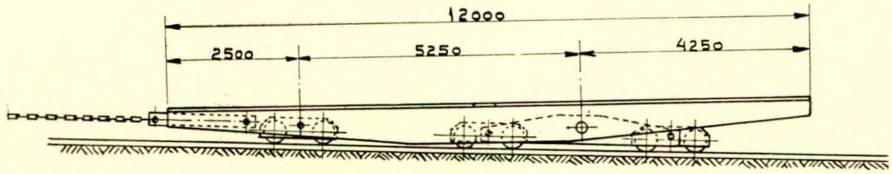


FIG. 2—*Slipway carriage*

reasons which call for ascertaining the pulling force acting on the chains, viz.; 1. To be kept informed on the way in which the load is distributed over the various winches and chains. 2. To prevent overloading of carriage or concrete strip. 3. To avoid undue loads on the ship. In fact, if the load on the various carriages is exactly known, any irregularities can be easily corrected by manipulating the movable pinions of the slipway winches. The company then decided to use Philips' strain gauges type in combination with the direct-reading self-balancing measuring bridge and a selector switch, which connects the seventeen winches entirely automatically to the measuring instrument. Thus a system was obtained which, by using only one reading instrument, requires little attention from the operator and at the same time gives a survey of the situation all over the slipway within the space of a minute. The force acting in the chain is transferred to the stationary shaft via the chain disc. One end of the shaft is supported by the winch frame, the other end by a yoke. As the point of application of chain and shaft is always the same, it is sufficient to measure the force acting on the yoke, because the latter is proportional to the pull of the chain. For this purpose the force on the yoke is transferred to the winch frame by means of two connecting rods. Several strain gauges are cemented on one of these rods and the elongation is measured with the help of the instrument in the slipway control cabin. As it is essential for the operator to be able to read the forces acting on all the chains, an automatic switch has been provided, which every three seconds connects the rod of another winch to the instrument. The scale of the meter has been calibrated in tons of chain pull. An interesting feature in the design of the instrument is that when the force in one of the chains exceeds a specified value, the automatic switch is stopped, permitting the operator to ascertain the number of the overloaded chain.—*Philips Serving Science and Industry, 1954; Vol. 1, No. 5, pp. 5-11.*

German Research Vessel

The 800 gross ton fishery research vessel, *Anton Dohrn,* has been launched at the Mützelfeldtwerft GmbH, Cuxhaven, for the Federal German Ministry of Food, Agriculture and Forests. The *Anton Dohrn,* which is due for delivery at the end of the year, is 190 feet in length. Triple-expansion steam machinery, with Bauer-Wach exhaust-steam turbine, will be fitted instead of a Diesel, to preclude the possibility of vibration from an engine of the latter type affecting the scientist's work. In addition to a normal fish hold the vessel will have refrigerated and deep-freeze compartments.—*World Fishing, October 1954; Vol. 3, p. 372.*

Care and Handling of Fibre Rope

Constant attention must be focused on the proper use, care, and maintenance of fibre ropes used on board ship if they are to be relied upon for any length of time. Fibre rope can be damaged by excessive hot and cold temperatures. For example, a line which is stowed near a steam pipe will dry out too fast and become brittle, having lost all the oils intended for lubrication purposes. On the other hand, a line which has become frozen will break more easily under such conditions, and should therefore be thawed and dried out before it is placed in service. Abuse of rope probably causes more rope to be replaced aboard ship than normal wear. All of the causes of rapid deterioration of rope aboard ship cannot be listed here; there are, however, several others which warrant mention. For example, a hawser should not be used on one side of a winch or windlass continuously, but rather should be shifted, port and starboard, frequently. In this way, the construction balance of the rope is maintained and the ship's lines have a longer life with greatest efficiency. Another poor practice when working with winches is permitting a rope to slip on a winch drum. This slipping causes sudden jerks which strain the fibres and shorten the rope's life. A rope should not be permitted to lie idle against a moving drum or capstan either. The friction created chafes the fibres and may generate enough heat to burn the rope. Fibre rope which has been subjected to any of the abuses described cannot be relied upon for its maximum strain and must be used with caution. It is, therefore, essential that all rope be carefully examined before it is placed in service.—*Proceedings of the Merchant Marine Council, U.S. Coast Guard, September 1954; Vol. 11, p. 143.*

Some Practical Experiences of Sea Clutter

It is widely known that whilst radar is almost unaffected by the densest fogs, some reduction of its performance must be expected in rain, and sometimes considerable reductions under conditions of very rough sea. Little practical data have been available to the mariner on the extent of reduction which may be expected in rough sea conditions and this short paper attempts to provide some practical information on the subject. These notes have been compiled on the basis of experience gained during a voyage from the United Kingdom to Japan and back; they are intended to give some assistance to the seaman in estimating the degree of protection which his radar may give under different conditions of wind and sea. The control usually labelled "Clutter" on British radars and S.T.C. (sensitivity time control) on American radars regulates a swept gain circuit. Waves on the surface of the sea return echoes to the radar which appear on the screen as a solid mass through which it is impossible to discern the echoes from any other targets. It should be realized that even if the echo from a large target is very much stronger than the clutter echoes, it will still not be seen on the plan position indicator since the clutter echoes are already showing at full brightness. The strength of the clutter echoes is greatest at close range and decreases gradually to a strength at which they no longer paint on the screen, the range to which they extend depending on the state of the sea. The purpose of the swept gain circuit is to reduce the sensitivity of the receiver at close range where the clutter echoes are strongest and slowly and automatically to restore sensitivity of the receiver to normal at the range where the clutter strength is effectively zero. In this manner it is possible to eliminate the clutter echoes from the centre of the tube without decreasing the performance of the radar at longer range. It must, however, be remembered that the sensitivity of the receiver in the clutter region has been reduced and that it will be only those echoes which are stronger than the clutter which will be able to paint; for this reason, small and even medium size targets which would be readily detected under calm sea conditions may not be detected when swept gain is in use, the size of the target likely to be lost clearly depending on the amount of swept gain employed. It follows from this that it is most important to use only sufficient swept gain to reduce the mass of clutter to the point where individual wave echoes are occasionally speckling on the screen; under these conditions a steady target just stronger than the clutter will be observed. To use so much swept gain as to clear the screen completely of clutter is to reduce the sensitivity unnecessarily and to run the risk of eliminating targets which otherwise might have been detected. A table is given which relates the types of target likely to be detected to the state of the sea. The assumption is made that just enough swept gain is applied to reduce the wave clutter to a mild nuisance level.—*E. M. Robb, The Journal of The Institute of Navigation, October 1954; Vol. 7, pp. 362-364.*

New Large Tanker

The Cities Service Oil Company's new tanker, *W. Alton Jones,* has a summer loaded displacement of 48,660 long tons, and a winter loaded displacement of 48,510 tons. Her light displacement is 10,749 tons. The *W. Alton Jones* has a deadweight capacity of 36,876 tons on her summer loading draught, and a speed of 17½ knots when bunkered with sufficient fuel for a round voyage from the Persian Gulf to Philadelphia, Pa. Under such conditions her pay-cargo capacity is 35,876 long tons. Her principal dimension are as follows:—

Length overall	707ft. 0in.
Length b.p.	677ft. 0in.
Length, w.l.	688ft. 4½in.
Breadth (extreme)	93ft. 0in.
Depth (moulded to upper deck)	48ft. 6in.
Draught (summer loaded) ...	36ft. 8⅝in.
Draught (light)	9ft. 3in.
Length of engine room... ...	96ft. 0in.

For propulsion power she has high and low pressure steam turbines driving a single-screw through reduction gears, the propeller having a diameter of 22 feet and a pitch of 20·4 inches. It turns at 103 r.p.m., when the turbines are developing 20,000 s.h.p. This gives the ship an operating speed of 17½ knots, but on sea trials a speed of 18·75 knots was reached. The fuel consumption economy on the trial run was excellent for steam power, being 0·496lb. per s.h.p. hr. Steam pressure in the twin boilers is 600lb. per sq. in. at 875 deg. F. temperature. Under average sea-going conditions her daily fuel consumption will probably be 106·3 long tons, with a power output of 20,000 s.h.p. Fuel capacity is 1,623·64 tons, and the deep tanks are capable of holding an additional 1,858·63 tons, but, of course, if used for fuel cut down on the pay-cargo capacity. There is a capacity for 77·83 tons of distilled water for the boilers, and 102·07 tons of fresh water for the crew's use.—*The Oil Forum, September 1954; Vol. 8, p. 310.*

Welding of Large Tanker

The *World Enterprise* (32,500 tons) is the largest tanker afloat in the world at present, and the advancement in welding is clearly indicated by the fact that the whole of this vessel was welded, with the exception of the gunnel bar and bilge strake, which were riveted. Machine-welding was employed for 80 per cent of the seams and butts; both visible-arc and submerged-arc processes were used in combination to full advantage. The shell bottom was welded in the fabrication shed, using one-third/two-thirds preparation, with a nose of one-sixteenth/one-eighth. The first pass was put in on the two-thirds side, using the visible-arc process. The panel was then turned over and the one-third side was completed in one pass, using the submerged-arc process. The panel was then again turned over for the final pass. This pass was put in simply to control distortion. It was possible to make the weld rise or fall as required. On completion, X-ray spot checks were taken, and the results of these were definitely up to Class 1 standard. It may be asked why, with the submerged-arc process, the weld was not completed with a single pass from either side. The answer is that, in shipbuilding, the ideal arrangement is not always possible, especially on plates 40 feet long. It is in such instances that the two processes combine so well. By using the visible-arc process on the first side, any inaccuracies in the plate preparation can be detected and adjustments can be made accordingly. Although preparation should be perfect, it seldom is so, and it is then that the visible-arc process is so valuable in shipbuilding. All bulkheads were welded with a square-edge preparation, using visible-arc on one side and submerged-arc on the other. The visible-arc machine welded in the trough and the submerged-arc on the top. Spot X-rays were again taken and the results were perfect. Staggered seams and straight butts were used on the shell bottom. The panels were joined up on the ship by welding first the seams and then the butt. Two machines were always employed so as to keep the welding even. The overhead on the butt was done by four welders, working from the centre outwards. For the top of the butt, two visible-arc machines were used, again starting from the centre and working outwards. One side of the side shell, which was shaped, could be welded by machine, using telescopic skids. The overhead welding was done first and then the machine was brought into use. At times a man had to walk alongside the machine to ensure that it did not fall over. The rolled sections of the bilge were also machine-welded. These sections were fitted into a special jig; the inside was welded by visible-arc in two passes and the outside by submerged-arc in a single pass. Both upper and lower struts in the wing tanks were welded by a special mast-welding machine. This is a very useful machine in a shipyard and has various other applications in addition to the welding of masts and derrick posts. The main deck work was carried out in the fabrication shed, as was the shell bottom. The panels were joined up on board by welding the seams and then the butt. Two visible-arc machines were employed. The 1,000-amp. generators for these machines were placed amidships on the deck, giving each machine a working distance of 600ft.—*D. E. Baty, British Welding Journal, September 1954; Vol. 1, pp. 392-396.*

Arc-Air Process

The Arc-Air Process is essentially the melting of metal, any metal, with an electric arc and simultaneously mechanically removing the molten metal by means of a high velocity air jet external and parallel to the electrode. This process is not dependent on oxidation, and for this reason works on metals which do not readily oxidize as well as those which do. The equipment consists of a torch with a concentric cable which carries both air and current. The electrode is usually carbon graphite though coated metal may also be used. An air line from an ordinary air compressor and the cable from a D.C. welding machine are both attached to the end of a concentric cable which carries both the air and the current to the torch. The lever at the bottom of the torch controls air flow. The electrode is held in a rotating head which allows it to be set at any angle, but maintains the air stream always directed at the proper location. The electrode usually used is a combination of carbon and graphite either plain or copper coated. An ordinary D.C. welding machine with reverse polarity is used; straight polarity is not effective with this process. The current depends on the size of the electrode and varies from 70 amp. on $\frac{5}{32}$in. to 600 amp. on ½in. electrodes. Although the higher the current density, the more efficient the process becomes as shown in Fig. 2 (not reproduced) which was made up for ⅜in. plain electrodes. The necessary air is obtained from an ordinary compressor. The torch is designed to operate at 90 to 100lb. pressure, which is the usual line pressure in most shops, and since this is not a critical value, no regulator is needed. The torch is used by holding the electrode at a leading angle and striking an arc between the electrode and the material to be cut. The air blast is directed immediately behind the point of arcing, and the electrode is pushed forward at a rapid rate with the air jet on continuously. The depth of the groove is determined by the angle of the electrode and the speed of travel. Because of the small area the metal being cut is instantly brought to the molten stage and speed of travel is very rapid, the surrounding metal does not reach a very high temperature and there is little distortion or crack propagation during the operation. In order to compare the temperature rise with that caused by oxy-acetylene gouging, a groove was cut in a 6in. round steel plate, ⅜in. thick, with a pyrometer attached to the back. After a one-minute gouging interval, the plate was still only 250 deg. F. when the Arc-Air torch was used. On the other hand, a similar plate was 600 deg. F. when oxy-acetylene gouging was used. After two minutes the temperature was 450 deg. F. with Arc-Air, but 1,000 deg. F. with the oxy-acetylene gouging.—*M. D. Stepath, The Welding Journal, September 1954; Vol. 33, pp. 860-864.*

Eliminating Cargo Pump Leakage

Service in an oil-tank vessel can certainly be a hazardous occupation; despite all modern safety measures, there can be no doubt that danger still exists. Adherence to stringent regulations has met with a certain amount of success in minimizing the three principal hazards—fire, explosion and asphyxiation. Nevertheless, the risks will remain, so long as the methods used for handling fluids are liable to leakage through mechanical fault or poor design. It is conceded generally that, during the discharging operation, the pump room is the most hazardous space in an oil-tank vessel. Leakage along the shafts of rotary and centrifugal pumps is the most common source of danger. There is doubt as to whether the older type of conventionally packed stuffing box is adequate for handling, with one and the same grade of packing, sea water and such low-boiling, low-flash materials as, say, gasolene, kerosene, fuel oil, etc. Apart from the danger of such materials leaking into the bilges, and there vaporizing to form explosive mixtures with air, there is the additional possibility that the liquid, leaking along the pump shaft, can very easily wash away all lubricants from the shaft bearings. This, of course, will lead to overheating, so that the three essentials for fire, viz., fuel, oxygen and a temperature sufficient to cause ignition, are thereby brought together. It has been shown that something like 250 cu. ft. of air can be rendered explosive by the evaporation of one pint of petrol. A good ventilation system will, of course, minimize the risk of the formation of explosive accumulations of vapour. However, the only sure way to solve the problem is to remove the cause, and this can only be accomplished by designing an efficient means of eliminating gland leakage. Soft packings require frequent renewal and, moreover, they are fundamentally incapable of eliminating gland leakage. So far as rotary and centrifugal pumps are concerned, the problem can be best dealt with by fitting mechanical seals. A drawing of a typical seal of this type—the Flexibox Type RC—is illustrated in Fig. 1. It consists, essentially, of two seal rings, one stationary and the other rotating with the shaft. Intimate contact between the faces of these seals prevents leakage from the pump. The stationary seal ring (1) is made from compounded carbon and is, of course, self-lubricating. The rotary seal ring (5) is manufactured from stainless steel (martensitic). Contact faces of both seal rings are works-lapped to within two or three tight bands. Resiliently supported by a cover seal (2), the stationary seal ring is located in the cover (3), which is bolted to the stuffing-box face. A stainless steel spring (7) furnishes the necessary pressure between the seal faces. It also provides for the positive drive of the rotary seal ring (5). Allowance is made in the design for coping with shaft misalignment, eccentricity, whip, thermal expansion and end float within wide limits. In order to remove frictional heat, at its place of generation, arrangements are made for the forced flow of some of the pumped liquid around the stationary and rotary seal rings. This also prevents the accumulation of harmful sediments.—*The Shipbuilder and Marine Engine-Builder, September 1954; Vol. 61, pp. 552-553.*

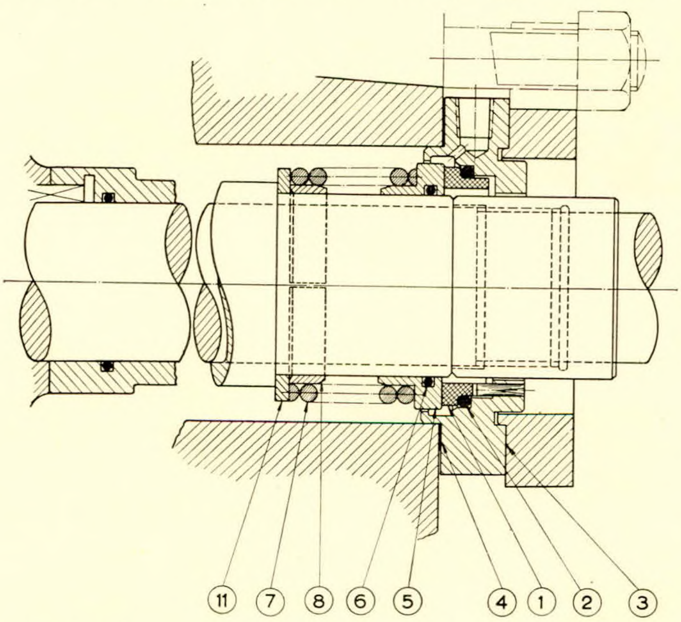


FIG. 1—*Diagrammatic drawing of Flexibox mechanical seal, type RC, for fitting to shafts of rotary pumps*

Provision is made for positive circulation of the product from the pump discharge. This serves to prevent the accumulation of sediments, to remove frictional heat, and to cool the seal. (1) Stationary seal ring; (2) cover seal; (3) cover; (4) static seal; (5) rotary seal ring; (6) static seal; (7) spring; (8) spring sleeve; (11) backing washer.

Response of the Sea Surface to Winds

The basic problem is to find how the momentum transferred from the wind to the sea is shared among ordinary waves, long waves and drift currents, and to study the related problems of generation, propagation and decay. Wave recorders have been developed to meet most requirements and since, right from the start, it was found necessary to treat the development of the wave pattern as the growth of a spectrum of component wave trains, a good deal of effort has been used in making simple and effective apparatus for analysing wave records. By selecting well-defined storms at various distances from recording stations on the west coast of England, methods have been devised for predicting the wave spectrum and some theoretical basis has been found for the results. The work is now being extended to deep water, well away from a coast and its associated disturbing factors, with the help of routine recordings from one of the weather ships. Wave generation is also being studied theoretically and experimentally with the help of fine measurements of waves made under a great variety of conditions of wind strength and air stability on a reservoir not far from the laboratory. A theoretical and practical study has been made of the relation between microseismic oscillations and waves, which may be useful in obtaining information about wave action. A start has been made on an investigation of the sources of energy that are responsible for exciting the natural oscillation of bays and harbours. The possible effects of groups of high and low waves, oscillations associated with the onshore and offshore movement of water by wind, deflection of tidal streams, and propagation of long waves from a storm area are being borne in mind.—*G. E. R. Deacon, The Journal of the Institute of Navigation, July 1954; Vol. 7, pp. 252-261.*

New Passenger Liner

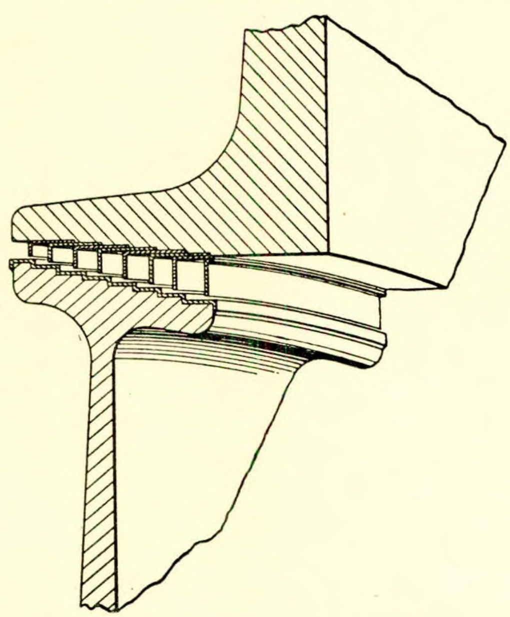
The Queen named the new Shaw Savill liner *Southern Cross* at the Belfast shipyard of Harland and Wolff, Ltd., on 17th August. This was the first occasion on which a reigning monarch has launched a merchant ship. The principal particulars of the *Southern Cross* are as follows:—

Length o.a.	600ft. 0in.
Length b.p.	560ft. 0in.
Breadth moulded...	78ft. 0in.
Depth	45ft. 3in.
Draught	24ft. 10in.
Gross tonnage (about)	20,000 tons
Shaft horsepower...	20,000 s.h.p.
Speed in service	20 knots

The machinery installation will consist of a twin-screw arrangement of geared turbines with high pressure boilers, the electrical power being provided by Diesel-driven generators. The installation, situated as far aft as the hull permits, is arranged in three watertight compartments—boiler room, engine room and generator room. The turbines are capable of developing a continuous power of 20,000 s.h.p. with propeller revolutions of 120 per minute, sufficient to maintain the vessel at a speed of 20 knots. The main boiler system consists of three Yarrow-type boilers designed for a pressure of 550lb. per sq. in. at the steam drum with an aggregate maximum continuous evaporation of 180,000lb. steam per hr., and a controlled superheat temperature of 800 deg. F. The superheaters are of the MeLeSco damper controlled type, which can maintain the designed final steam temperature over a range of evaporations from 35,000 to 80,000lb. per hr. Boiler oil will be burnt in the boilers under balanced draught conditions and air for the consumption of the fuel heated by steam bled from the turbines and not, as is usual, by the exhaust gases. Hull space for fresh water has been reduced to a minimum and all fresh water consumed on board for all purposes will be made on board. Owing to the concentration of machinery weight aft special attention has to be paid to trim, and certain oil tanks as they become empty will need to be filled immediately for that purpose. To avoid the contamination of the boiler oil by sea water, all ballasting will be with fresh water which will be made by the ship's evaporating plant. To cover the extensive duty of providing fresh water for domestic use, boiler feed and ballasting, five Weir evaporators will be arranged to operate either as single or triple effect units and these will be capable of maintaining an output of 300 tons of distilled water per twenty-four hours from sea water. For boiler feed, double-distilled water only will be used. Two electronically controlled fully automatic Victor oily-water separators will be installed.—*The Shipping World, 25th August 1954; Vol. 131, pp. 207-209.*

Labyrinth Seal

The difficulty of maintaining a good seal between shaft and rotor casing in steam or gas turbines and similar high temperature rotating machinery by the use of labyrinth seals is mainly due to the fact that considerable clearance has to be allowed between the moving parts to take care of possible deformation. This invention describes a means of overcoming the trouble and working with extremely small clearances. A hard abrading surface is provided on the shaft in the region of the labyrinth rings and the seal assembled so that there is a small interference between the labyrinth rings and the shaft.



With this type of labyrinth seal, the shaft is provided with a hard abrading surface so that when it is rotated the edges of the rings are lapped. In this illustration the clearance is exaggerated

When the shaft is rotated the edges of the rings are lapped by the abrading surface and the interfering metal rubbed away. Subsequent distortion of the rotor due to severe operating conditions causes further lapping of the rings; thus the minimum clearance required is automatically obtained in the first place and this clearance is increased only when working conditions demand it. The small particles of metal ground away in the lapping process are removed by the stream of vapour or gas flowing through the seal. The hard abrading surface may be formed by metal spraying a steel tungsten-molybdenum alloy, by electro-plating with chromium on a grit blasted surface or by electro-plating with chromium on a smooth surface followed by reversed electrolytic etch.—*British Patent No. 660,396. N.R.D.C. Bulletin No. 2, July 1954; pp. 6-7.*

Italian Marine Diesel

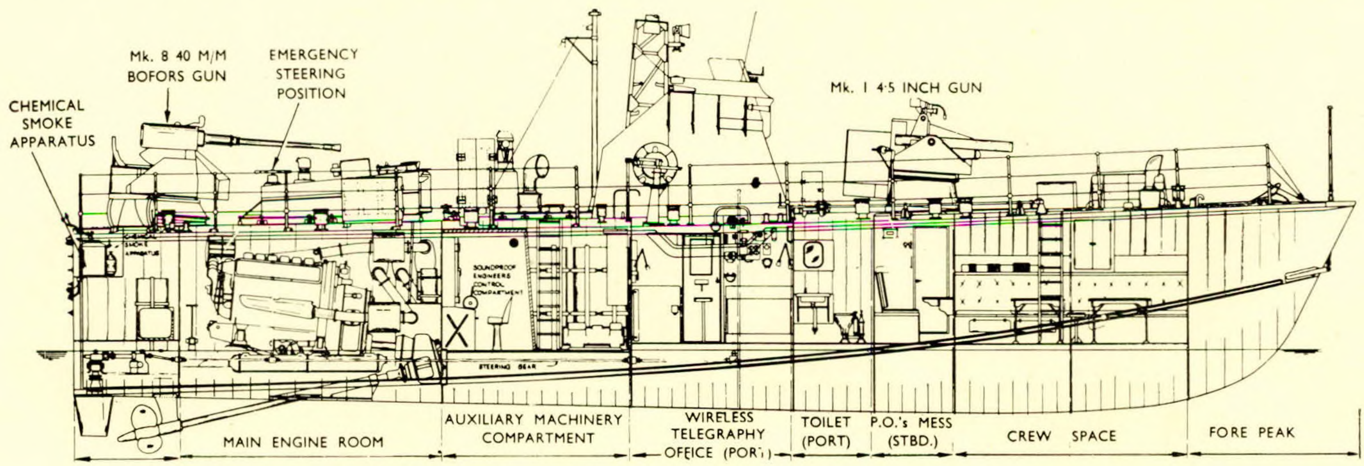
The Italian company of S.A. Motori Marini G. Carraro have recently released details of their latest Diesel engine design, a 12-cylinder Vee-type unit for rail traction and marine uses. Normally-aspirated and pressure-charged versions are available and the marine design is entirely self-contained with integral oil and water pumps, coolers, heat exchanger and reverse-reduction gearbox. The two banks of cylinders are arranged at an angle of 60 degrees with pre-combustion chamber type cylinder heads cast in pairs. The inlet manifolds are on the outside of the engine and the exhaust manifolds within the Vee. The normally-aspirated engine develops 350 b.h.p. at 1,500 r.p.m., but when fitted with twin Brown Boveri pressure-chargers, the output is increased to 500 b.h.p. A heat-treated light alloy, cast cylinder block carries centrifugally cast liners, while the statically and dynamically balanced crankshaft is carried in lead-bronze main bearings. The connecting rods are H-section stampings with fully floating gudgeon pins. The two rods for each crank throw bear on rings lined inside and out with a copper-lead alloy. Aluminium alloy pistons having four pressure rings (of which the top one is chromium plated) and two scraper rings are employed. The fuel injectors for each bank of cylinders are supplied by a six-unit multiple pump arranged across the front end of the engine. Engine starting can be either 6 h.p. 24-volt electric, assisted by heater plugs, or by compressed air. A dry sump lubrication system with twin gear-type pumps is employed. An S.A.E. grade 40 is recommended in winter and S.A.E. 50 in summer. The reverse-reduction gearbox is available in a number of ratios varying between 1·4 and 2·1 to 1. It is oil-operated and controlled by a quadrant lever.—*Gas and Oil Power, October 1954; Vol. 49, pp. 239-240.*

Effect of Welding Variables on Hard-facing Deposits

Alloy steels are special-purpose steels for high-tensile strength and other special requirements. They differ from mild steel in that they have higher carbon and alloy content. The higher carbon and alloy content confer the special properties to these steels. The special properties, however, are only available for use through proper heat treatment of the steel. Heat treatment determines the structure of the steel which, in turn, determines properties. Hard-facing with the electric arc essentially duplicates the electric furnace process of making alloy steels. Hard-facing welding electrodes contain carbon and other alloying materials which are introduced into the "furnace" of the welding arc so as to produce deposits of alloy steels and irons. Simultaneously, the heat cycle of the welding process produces a heat treatment. Hard-facing is thus a process for simultaneously alloying and heat treating. In hard-facing with the electric arc, as with all steel making, the process of alloying and heat treating must be precisely controlled to produce satisfactory structures for abrasion and impact resistance. The electrode manufacturer must correctly balance his electrode formulation to provide for variables that enter the process, and the consumer must control the welding procedures to produce a satisfactory deposit. The rapid heat input and fast cooling of the weld deposit can materially alter the structure and, therefore, the properties of the deposit. The control of welding variables in hard-facing determines the results achieved. Frequently, insufficient attention is given to the control of these variables, and it is the purpose of this paper to indicate what variables are significant and in general how they should be controlled. The following conclusions were arrived at: 1. Hardness is not necessarily a measure of wear resistance. 2. As alloy content increases, abrasion resistance increases. 3. As base metal dilution increases, abrasion resistance decreases. 4. As alloy content increases, cooling rate can be decreased. 5. The results intended by the manufacturer for a given type electrode will not be obtained unless proper procedures are used. 6. Semi-austenitic, low-alloy electrodes are used to resist both abrasion and impact. Their deposits are affected by cooling rate. For abrasion values of 3-4 use two-layer deposits. Preheat, higher currents and high interpass temperatures improve impact resistance, but decrease abrasion resistance. For thin sections that provide slow cooling rates, use low currents. For heavy sections that provide a fast cooling rate, use higher currents. 7. Chromium-carbide-type electrodes resist severe abrasive wear. Their deposits of two or more layers are not materially affected by cooling rate. 8. By correlating the control of factors that determine deposit analysis and cooling rate a wide variety of service requirements can be met.—*A. Svanut and V. Peters, The Welding Journal, August 1954; Vol. 33, pp. 778-782.*

Naval Craft with Napier Deltic Engine

The high-speed Napier-Deltic-type engine is to be installed in a number of naval craft now under construction, including a coastal minesweeper being built by Camper and Nicholsons and a similar vessel by J. I. Thornycroft and Co., Ltd. It is also to be fitted in fast patrol boats of Type "A", five of which have been ordered from Saunders-Roe (Anglesey), Ltd., who are acting as the parent firm, responsible for working drawings for six other companies building these vessels in different parts of the country. They are 71ft. 4in. in length, with a beam of 19ft. 10in. and a draught of 6ft. 1in., the displacement being 64 tons, the complement twelve officers and men. The framework is of riveted aluminium alloy, the main component parts being manufactured in special jigs. The deck is of aluminium alloy and the hull is completed with double diagonal planking or, alternatively, a riveted aluminium skin. The Napier-Deltic Diesel machinery develops 2,500 b.h.p. and the drive is taken forward to a gearbox as the machinery installation is at the after end of the vessel.—*The Motor Ship, October 1954; Vol. 35, p. 287.*



British fast patrol boat type "A" fitted with Napier-Deltic machinery

Marine Fuel Blending

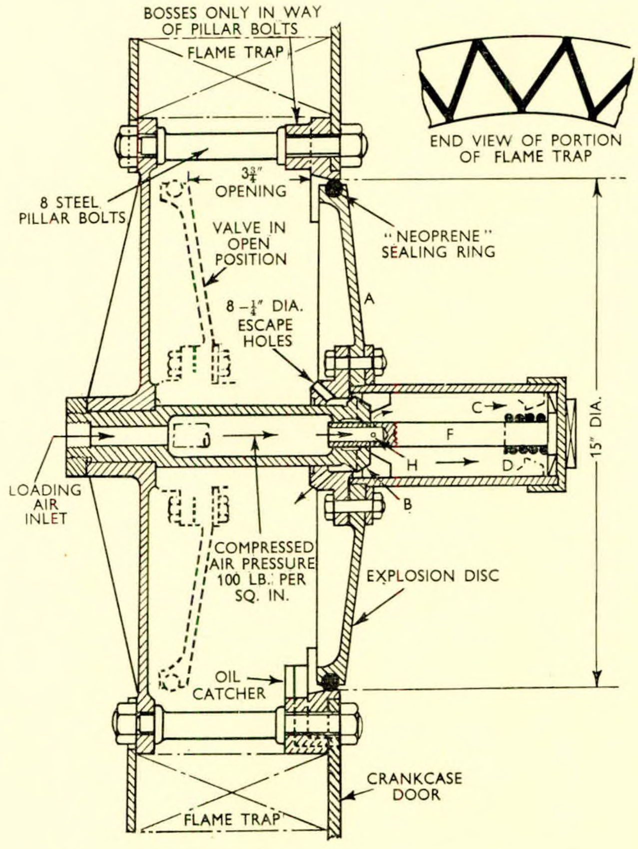
Ultrasonic energy and electronics have been harnessed to develop a special monitoring system to automatically blend fuel oil for marine use. This system is claimed to have proved superior to conventional mixing procedures by accurately blending light and heavy fuel oil to conform to varying burner characteristics in different ships. By conventional methods the blending procedures are worked out in advance by mathematical calculation. Chief disadvantage of this so-called "paper blending" was that it failed to take into account varying factors which might throw off the measurements. For example, errors in the pumps or meters feeding oil into a ship, or possible stratification in the storage tanks, are two conditions which might disturb the advance figuring and result in a poor blending of the oils which itself contributes to lower combustion efficiency. Under the new technique, the viscosity of the fuel oil is measured by an Ultraviscoson unit which utilizes ultrasonic energy. A probe, attached to the instrument, is inserted into the line carrying the final blend. The amount of energy absorbed by the fluid from the probe as it oscillates at high frequency indicates the viscosity. Since viscosity varies with temperature change, a compensating unit is incorporated to adjust the measurement for varying temperatures. As the viscosity varies—and is sensed by the probe—the information is fed to an electronic control unit which automatically controls the flow of light fuel oil to the mixture (by opening or closing a valve) to bring the viscosity to the predetermined desired value. The new system will not only assure more accurate blending of fuel oil, but it is also expected to result in substantial savings by conserving the amount of light fuel oil used. —*Marine News, September 1954; Vol. 41, p. 78.*

Scandinavian Ice-breakers

The Finnish icebreaker, *Voima,* is equipped with four propellers, two aft and two forward, being the first icebreaker to have two bow propellers. The main advantage of this is that the continuous wash round both sides of the bow prevents the vessel from being jammed in the crushed ice and decreases the pressure of the ice against the hull. In addition, the four-propeller system affords excellent manœuvrability, which is of great importance to an icebreaker. This advantage is further increased by using Diesel-electric machinery, thus enabling the propelling motors to be directly operated from the bridge. The primary power is supplied by six Diesel engines, each directly connected to its generator. The generators deliver current to the four propelling motors through a modified Ward-Leonard system. The Diesel engines maintain the same r.p.m. and direction of rotation and are able to deliver full power irrespective of the r.p.m. of the propelling motors. The 160-tons heeling tanks are situated amidships between the main and upper deck. By means of a propeller pump, 160 tons of ballast water can be transferred in ninety seconds from one side to the other. When jammed in the ice the vessel is thus able to loosen the grip by heeling from one side to the other while running the engines astern. The main engines consist of eight-cylinder single-acting two-stroke Nohab-Polar K58M-type Diesels, having a bore of 340 mm. and a stroke of 570 mm. The engines are normally capable of delivering 1,625 b.h.p. at 325 r.p.m., but can be continuously overloaded for six hours, when they deliver 2,000 b.h.p. at 400 r.p.m. The four propelling motors are of double rotor type. The after motors have a weight of 64 tons each, the forward motors being somewhat smaller. Each motor is rated to develop a continuous output of 2 × 1,280 kW = 3,500 b.h.p. at 120 r.p.m. for the after motors and 180 r.p.m. for the forward motors. The Swedish icebreaker, *Thule,* completed earlier this year for the Swedish Government by the Swedish naval dockyard at Karlskrona, is a much smaller ship than the *Voima,* having a displacement of only 1,930 tons as against the 4,350 tons of the Finnish vessel. She is intended to assist shipping primarily in the narrow and shallow waters of the Southern Oresund. Despite this addition, the Swedish icebreaker fleet is still not up to the size considered necessary by the Government committee concerned. The two large icebreakers, *Atle* and *Ymer,* are twenty-seven and twenty years old respectively, and the construction of a new large icebreaker is likely to be undertaken soon. As with the *Voima,* the Diesel engines driving the main generators are of Nohab-Polar-type. Each of the three engines develops 1,600 b.h.p. at 335 r.p.m. On overload the three engines can develop a total of 6,000 b.h.p. The main generator units are arranged abreast across the ship. The maximum output of each of the three propeller motors is 1,850 b.h.p.; this output is developed by the forward motor at 180 r.p.m. and by the two after motors at 145 r.p.m. The efficiency of the motors, from one-quarter to full power, is stated to be 94 per cent. The three auxiliary generators are also driven by Nohab-Polar Diesel engines, each of 345 b.h.p., while a small harbour auxiliary set is driven by a Nohab-Polar engine of 110 b.h.p. The Danish icebreaker, *Elbjorn,* of 1,400 tons displacement was built for the Danish Ministry of Commerce by Frederikshavns Værft and Flydedok A/S, and is intended for service in Danish waters. The principal dimensions of the *Elbjorn* are as follows:— Length o.a., 167ft. 6in.; breadth, 39ft. 4in.; draught when ice-breaking, 16ft. Like the *Thule* she has Diesel-electric drive but with only two propellers, one forward and one aft. The machinery occupies the four main compartments below main deck level, the arrangement being very much like that of the *Thule.* The two centre compartments house the main and auxiliary Diesel generators. The former consist of three turbo-charged four-stroke single-acting Diesel engines by A/S Frichs, of Aarhus, each developing 1,200 b.h.p. at 425 r.p.m. and directly coupled to an 800-kW 440-volt D.C. generator by Thomas B. Thrige, of Odense.—*The Shipping World, 30th June 1954; Vol. 130, pp. 655-657.*

Crankcase Explosion Research

At the Thornton Research Centre some forty crankcase explosions have been artificially created in the eight-cylinder 1,100 b.h.p. Sulzer trunk-piston engine removed from the Shell Diesel-electric tanker *Auris* and replaced by a B.T.-H. gas turbine. This engine is now installed in what remains of a specially-built shed and has been subject to tests over a period of about six months. The tests have now been discontinued. The total crank-chamber volume is about 220 cu. ft. in eight compartments and seventeen Lightning valves have been fitted to the crankcase doors, there being eight at each side and one at the forward end. These are loaded with compressed air at 50lb. per sq. in. and lift at about 0·5lb. per sq. in. Operation of the valves is practically instantaneous, taking about 1/50th second to open (thus avoiding a serious build-up in pressure) and about 2/50th second to close, this preventing the ingress of air into the crankchamber which generally causes a second and more violent explosion. No tests have been made with the engine running and the method adopted to create an explosion has been to pass a calculated amount of lubricating oil through an electric heater, mix it with air to the proportion of 6oz. of oil to 5½ cu. ft. of air and pump the mist into the crankcase, where it is electrically ignited. The maximum pressure created in the crankcase is in the region of 40-50lb. per sq. in. and it is apparent that few engines, if any, are able to withstand such a pressure and to redesign them to do so would be economically impossible. A number of motor ships in the Shell and other fleets have already been equipped with Lightning relief valves, which have been approved by Lloyd's Register of Shipping.—*The Motor Ship, October 1954; Vol. 35, p. 298.*



Section through Lightning crankcase relief valve

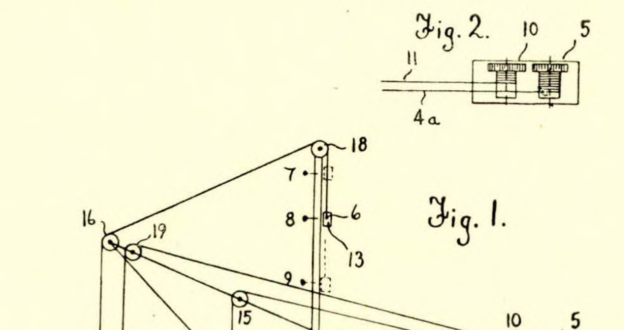
Marine Pilotage

In the port of Southampton, where pilots are called upon to handle any type of vessel ranging from a small yacht to the world's largest liners, many pilots have expressed surprise and consternation on finding that so little attention appears to have been paid to the siting and layout of the navigating bridge. In many cases it would almost seem that the bridge had been an afterthought. To con a liner efficiently, it is necessary for the pilot to place himself in a position which is on the fore-and-aft line of the ship, i.e. amidships, in order that he may judge more accurately the angles of approach; but, frequently, he finds that his vision is obscured for several degrees on either bow by sampson posts and/or other obstructions, and in some cases excessive streamlining makes the accurate judgment of distances whilst docking wellnigh impossible. The efficiency of a pilot is, to a large extent, based upon the accuracy of his judgment and on his ability to anticipate the vessel's reactions to alterations of helm and engine movements. It is, therefore, essential that the pilot should be able to observe the necessary instruments or indicators continuously without leaving the conning position. This could be accomplished by the provision of an instrument dashboard or panel fitted with suitable lighting and dimming controls.—*The Journal of the Institute of Navigation, July 1954; Vol. 7, pp. 280-289.*

Patent Specifications

Adjustable Head of Floating Suction Dredger

The present invention is concerned with floating suction dredgers of the type comprising a suction tube coupled at one end to the dredger and having at the other end a suction head for traversing the surface to be dredged. The satisfactory functioning of such a suction dredger depends essentially on the fact that the suction head during dredging is constantly on the bottom. The steady support of the suction head during dredging, however, is jeopardised firstly by the motions of the ship due to the swell and secondly by the changing depths of the surface to be dredged. The invention has for its object a means which renders it possible for the dredger to be under way with the suction head resting constantly on the bottom independently of the swell and the changing depth of the bottom. The suction tube in Fig. 1 comprises the upper section 1, which is rotatable on the vessel about a horizontal axis and perhaps also about a vertical axis. Located at the bottom end of the section 1 of the suction tube is the universal joint 14 for the movable connexion of the lower section 2 of the suction tube, including the suction head proper 3. The universal joint 14 renders it possible for the suction tube section 2 to execute a movement within certain limits, for example, a movement of 15 degrees upwardly and downwardly (positions 3b and 3c) with respect to the axis of the suction tube section 1. The universal joint 14 is suspended from the single cable 11, which passes to the drum 10 of the electrically driven winch over the roller 15 mounted in a jib-like frame. The suction head 3 itself is suspended from a double cable 4, 4a which is passed around the balancing pulley 17 connected to the suction head 3. The one run 4a of the cable passes over the pulley 19 arranged on a fixed derrick secured to the ship's hull and to the winding drum 5 of the same winch, while the other run 4 passes over the pulleys 16 and 18 to the frame 12. The end point 6 of the run 4 is secured to the counterweight 13, which in turn can carry out a limited upward and downward movement in the frame 12 about a central position.—*British Patent No. 717,862, issued to A. G. Weser. Complete specification published 3rd November 1954.*



FIGS. 1 and 2—*(Patent 717,862)*

Damper Controlled Marine Boiler

In one form of marine boiler one gas pass leading from a combustion chamber includes a steam generating tube bank, a second or superheater pass leading from the combustion chamber includes another steam generating tube bank, within which a superheater is arranged; dampers are provided for controlling the gases of the respective passes, and the gases from the two passes re-unite and flow together over an economizer. In a marine boiler, normally a high steam temperature is required, so that the dampers of the pass in parallel with the superheater pass are nearly closed. On the other hand, when manœuvring, a low steam temperature is required, so that the dampers of the superheater pass are nearly closed. In both cases, therefore, a resistance to gas flow is set up by the dampers. The object of the invention is the provision of improved means for controlling the distribution of gas flow between the parallel gas passes of a tubulous vapour generating and superheating unit. The invention provides for bringing into play, when it is desired to restrict the gas flow in either of the parallel connected passes, an increased pressure drop across one pass, and simultaneously increasing the cross-sectional area of path through the other gas pass. In Fig. 1 combustion gases are shown as flowing from a combustion chamber 1 to deliver heat to heat exchange surfaces in two gas passes 2 and 3 and then to deliver heat to an economizer 4 in a subsequent gas pass 5. The heat exchange surfaces in the first parallel pass 2 consist of a steam generating screen section 6 and, in the gas flow path behind the screen section, a steam generating section 7, and the heat exchange surfaces in the second parallel pass 3 consist of a steam generating screen section 8, a superheater 9 in the gas flow path behind the screen sector 8, and a steam generating section 10 in the gas flow path behind the superheater 9. The gases from the first and second parallel passes 2 and 3 are led to respective spaces 11 and 12 in front, as regards the direction of gas flow, of the economizer 4 and on opposite sides of a swinging damper 13, which can be moved to one side or other of a neutral position so as to restrict to a greater or lesser extent the gas flow into the economizer 4 from either the space 11 or the space 12. In the gas pass 5 a number of baffles 14 to 18 are provided which extend through

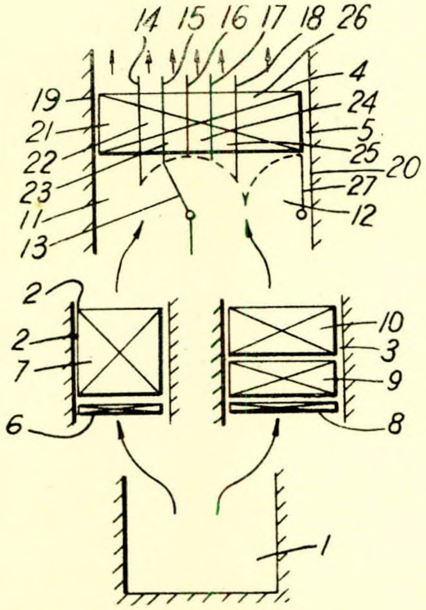


FIG. 1—*(Patent 717,420)*

the economizer 4, and extend parallel to opposite walls 19 and 20 of the gas pass 5 and to the axis of the damper 13 and subdivide the gas pass 5 into a number of separate passages 21 to 26. The damper 13 and the baffles 14 to 18 are so arranged that the damper can determine by its position whether any of the passages 22 to 25 lying between the two outermost passages 21 and 26, which latter are respectively adjacent the gas pass walls 19 and 20, is included in the gas path from the space 11 or the gas path from the space 12. As shown, the edge of the damper is adjacent the baffle 15, and a gas path consisting of the passages 21 and 22 arranged in parallel is placed in series with the gas pass 2 containing the screen section 6 and steam generating section 7, and a gas path consisting of the passages 23 to 26 arranged in parallel is placed in series with the gas pass 3 containing the screen section 8, superheater 9 and steam generating section 10. The relative quantities of gas flowing in the gas passes 2 and 3 depend upon the respective flow areas of the paths consisting of the passages 21 and 22 and of the passages 23 to 26. A further damper 27 which in the open position lies against the wall 20 but which closes towards the damper 13 is provided for independently varying the relative gas quantities flowing through the passes 2 and 3. In the operation of the unit, by varying the position of the damper 13 so that the gas passage previously in one gas path in the pass 5 is instead included in the other gas path in the pass 5 and adjusting so far as necessary the firing rate of the combustion chamber so that the same quantity of steam is produced by the unit at the same pressure, the superheat temperature can be regulated over a range of loads. If a superheat temperature is required which is between that obtained with the damper edge positioned adjacent one baffle and that obtained with the damper edge positioned adjacent the next baffle, then the damper may be set to that of the two positions yielding the higher superheat temperature and the gas flow through the gas path 3 including the superheater 9 is reduced by operating the further damper 27 to a suitable extent from the open towards the closing position. —*British Patent No. 717,420, issued to Babcock and Wilcox, Ltd. Complete specification published 27th October 1954.*

Fenders for Ships and Docks

This invention relates to fenders for ships, docks or the like, and has for its object to devise an inexpensive form of fender that can be used as a floating fender or as part of a dock, harbour or like installation. In Fig. 1 is shown a fender intended to float in the water so that it may be conveniently interposed between a ship's hull and some other object with which the ship contacts. The fender essentially comprises a cylindrical air-tight body (1) which may be fabricated by welding from steel sheet. Bollards (2) at either end of the body (1) permit the fender to be controlled with the aid of ropes or

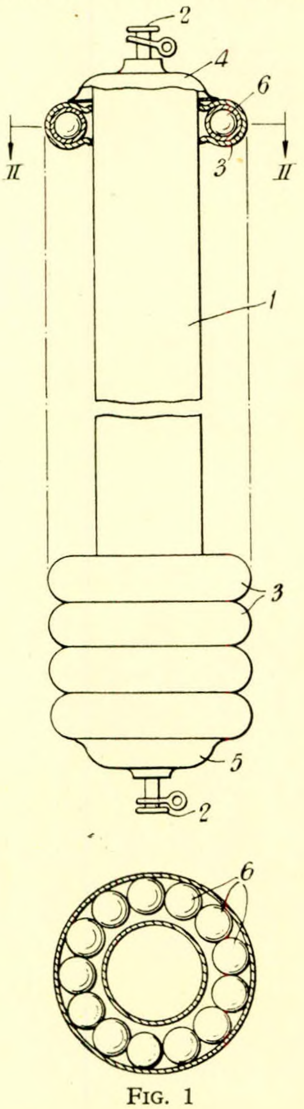


FIG. 1

the like when it is floating in the water. Tyre outer covers (3) are disposed over the cylindrical body (1), and are retained between end-plates (4 and 5), made easily removable so as to permit replacement of the tyre outer covers when desired. Preferably the tyre outer covers are worn heavy duty outer covers which possess considerable resistance to deterioration in salt water. Before being placed in position each outer cover is packed with hollow rubber balls (6).—*British Patent No. 718,624, issued to F. W. Allen. Complete specification published 17th November 1954.*

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Marine Engineering and Shipbuilding Abstracts

Volume XVIII, No. 2, February 1955

Ship Repairs in South Africa

The high cost of work in Europe, together with other difficulties, has brought many repair contracts to South Africa in the past ten years, the vast majority of the ships going to Durban. Strengthening of the American-built oil-tank vessels proved a useful source of work for Durban repairers, and one such vessel was handled at Cape Town. Major repair contracts in the Union are not sufficient to keep the industry fully employed, and some difficulty is experienced in Cape Town during the spring and summer months in maintaining full employment. Whereas all the major shipping lines, operating on the South African trade, prefer to do all but essential repairs in Britain or Europe, there is a natural desire on their part for a quick turn-round when contracts are placed locally. But in an industry which lives from hand to mouth, it is not always possible to maintain large staffs on the off-chance of a casualty. Until recently, British and European shipyards had all the advantages over the South African industry, but experience is showing that, both for standard of work and price, the pendulum is on the point of swinging in favour of work in the Union. Instructions have been given by the headquarters of major shipping lines that only essential work is to be done in South Africa. This essential work has increased in the past few years. It is not possible for large liners, operating on fixed schedules, to undergo major surveys in South African waters; but the suggestion has been made that more freighters, with terminals in South Africa, should patronize local facilities, so that, in time of a major casualty, the facilities will be equipped with both the manpower and the machinery with which to give a quick job.—*The Shipbuilder and Marine Engine-Builder, October 1954; Vol. 61, pp. 600-601.*

The Seaguard Automatic Alarm Receiver

A new type of automatic alarm receiver, complying fully with the revised Post Office specification for the reception of the international radio alarm signal, has been designed by the Marconi International Marine Communication Co., Ltd., of London, and is now in production. This new equipment—appropriately named Seaguard—is switched on when the radio officer goes off watch. It then maintains a reception watch on the distress frequency and, if the international alarm signal of twelve four-second dashes at one-second intervals is transmitted within range, the Seaguard responds before the end of the fourth dash by actuating an alarm-bell system. The Seaguard consists of two main units, the receiver and selector, both of which are housed in one cabinet. The receiver is of an optimum sensitivity, carefully designed to ensure reception of alarm signals within a reasonable range, without responding to transmissions from ships at a too-great distance. The receiver is pre-tuned on a frequency band of 490 to 510 kc/s, which allows for adequate coverage on the distress frequency of 500 kc/s, while still permitting the reception of alarm signals from a transmitter which may be slightly off-tune in an emergency. Due to its comparatively high sensitivity, the unit has been designed with an elaborate automatic dual gain control system, which provides circuit precautions against response to heavy traffic in congested radio conditions, or to prolonged static. The selector consists of an electro-mechanical timing system, built to very stringent physical tolerances. A highly stable electronic oscillator provides the audio-frequency power for the phonic motor, which is the driving unit of the selector, and a system of cams, pawls and relays lock in progression on the receipt of each correctly timed and spaced dash. An incoming signal, passed from the receiver to the selector, energizes the input relay which, in turn, closes a clutch and brings the cam and pawl system into operation. If the signal is correct, the first dash pawl engages at the end of 3·45 seconds, and at the end of 4 seconds the space cam begins to rotate. This cam, after a lapse of 1·55 seconds, closes a pair of contacts which, unless the next acceptable dash has begun, will operate a trip relay returning the mechanism to its datum position. In this way, the required one-second interval between dashes, within reasonable limits of accuracy, will not operate the trip. The second dash, commencing before the end of 1·55 seconds, actuates its pawl, which engages and holds if this dash is also correctly timed. The mechanism continues to operate in a similar way through the reception of the third dash, and when the fourth dash has lasted 3·45 seconds, the alarm relay is closed, completing the bell circuit. Rejection of false signals, caused by two or more stations transmitting simultaneously on the same level of volume and, therefore, producing between them long dashes, is guarded against by extremely accurate timing. The Seaguard is capable of registering very short gaps in transmission down to less than 10 milliseconds and can "see through" prolonged but slightly broken interference, which would otherwise simulate, and be accepted as, one continuous dash. The dash mechanism rejects any signal lasting less than 3·45 seconds or more than 6·1 seconds, while the space cam trips the mechanism if any space is longer than 1·55 seconds.—*The Shipbuilder and Marine Engine-Builder, October 1954; Vol. 61, p. 603.*

New Ore Discharging Equipment

Two electrically-operated transporters, each weighing 600 tons and of unique design, have recently been put into operation at Bidston Dock, Birkenhead. Constructed by Joseph Booth and Bros., the transporters are used for unloading iron ore from ships at the dockside into waiting road, rail or barge transport for dispatch to the Hawarden Bridge Steel Works of John Summers and Sons, Ltd. Each grab has a capacity of six tons and the time taken to pick up and discharge a grab load is sixty seconds, giving each transporter a capacity of over 350 tons an hour. The 140-ft. high transporters have a number of unique features. They are the only two in Great Britain which can unload either on to land or into barges, the length of the transverse boom—219ft.—allowing the grab to discharge into barges lying on the offshore side of the ships. The transporters themselves can move along the quay on rails for a distance of 1,000 feet. The current collection equipment, for moving the transporters along their railway, is believed to be unique in Great Britain. An underground duct runs behind the transporters parallel with the rails, and the collectors of each transporter, together with the T-bars carrying the current, are, therefore, completely enclosed. Installed in the duct are three 1,000-ft. copper T-bars along which run the three current collectors fitted to each transporter. These are provided with 1,000-amp. twin-head carbon inserts and the T-bars are fixed to the side of the underground duct by means of steel brackets and insulated supports. The current collection equipment for the cross-travelling cranes with their huge grabs presented special difficulties. The 219-ft. booms along which the cranes move are hinged so that the 93-ft. end section can be raised clear of the superstructures of ships to allow the whole transporter to move along from hold to hold and, as a result, special T-bar and trolley wire fittings had to be installed at the hinge. —*The Shipping World, 29th September 1954; Vol. 131, p. 350.*

Inflatable Liferafts

The Ministry of Transport has approved the use of inflatable liferafts in place of buoyant apparatus in certain types of merchant ships, and in addition to buoyant apparatus in others. The principal types of ship in which the new rafts can replace buoyant apparatus are passenger vessels not engaged on international voyages, fishing boats, and pleasure yachts exceeding 15 tons burden. They may be used in addition to buoyant apparatus in passenger vessels engaged on international voyages, non-passenger ships, tugs, tenders and dredgers. This approval follows exhaustive tests carried out by the Ministry. These inflatable liferafts have been developed since the war by the Admiralty, in conjunction with the various manufacturers concerned. Warships are, of course, incapable of carrying sufficient boats to provide the necessary lifesaving capacity for the whole of the crew, and Carley floats have been used for a long time to supplement the comparatively few boats carried. Carley floats offer a poor chance of survival in anything but temperate weather unless rescue is almost immediate, while modern warships carry even fewer boats than their predecessors. For this reason the Admiralty determined to develop an inflatable raft, based on the R.A.F. inflatable dinghies, which would stow in a smaller space than Carley floats and would at the same time provide better conditions for survivors. This has been achieved with considerable success in the liferafts now developed. If thrown over the side they automatically inflate themselves right way up, and when inflated provide insulation (by means of air between the two skins) both above and below. Sufficient water and glucose sweets are normally provided for a minimum period of three days. From the point of view of merchant service use, the obvious disadvantages of inflatable liferafts are the necessity to jump overboard or climb down a ladder to board them, and the fact that their life under service conditions is likely to be shorter than that of either lifeboats or buoyant apparatus. Against this must be set the fact that they can be launched under any conditions of sea or list of the ship, that they need little or no maintenance, and that their weight is much less than that of lifeboats and davits or buoyant apparatus.—*The Shipping World, 6th October 1954; Vol. 131, p. 370.*

Corrosion in Aluminium Lifeboat Shell

On an ocean-going passenger vessel, a number of aluminium lifeboats failed in pressure tests on the built-in buoyancy tanks after about two years' service. Examination showed that perforation owing to corrosion of the 10 s.w.g. sheeting had occurred in way of the galvanized steel hawsers used to secure the boats. Corrosion of the hull sheeting was confined to areas in close contact with the galvanized hawser, which was itself in an advanced state of rusting. Other galvanized steel hawsers on board were in good condition, the comparison suggesting that the zinc coating of the hawser securing the boat was unduly thin, although no estimation of the original thickness of the zinc coating was possible on the rusted hawser. This failure seems to have been owing to chafing of the paint on the lifeboat with simultaneous breakdown of the galvanized coating on the steel hawser; in time, therefore, a steel-aluminium couple came into play and caused rapid corrosion of the aluminium alloy sheet when exposed to direct salt spray. In this case, chafing could have been averted, either by "serving" the hawsers or by placing wooden distance pieces between them and the shell of the boat.—*D. C. G. Lees, The Shipping World, 6th October 1954; Vol. 131, p. 377.*

Single Screw Steamship for Wool Trade

The single screw steamship *Ballarat*, built by Messrs. Alexander Stephen and Sons, Ltd., of Linthouse, Govan, Glasgow, for the Peninsular and Oriental Steam Navigation Company, of London, has recently completed successful trials. The vessel will be employed in her owners' Australian service, her speed fitting her admirably for the wool trade, for which she and her sister ship *Bendigo* (launched on the 12th August) have been designed. The principal dimensions and other leading characteristics of the *Ballarat* are as follows: —

Length b.p.	490ft. 0in.
Breadth moulded	69ft. 0in.
Depth to shelter deck	43ft. 0in.
Load draught	29ft. 9in.
Deadweight, tons	12,790
Gross tonnage	9,300
Service s.h.p.	11,000
Maximum s.h.p.	13,000
Service speed, knots	18

The main propelling machinery, which consists of a set of steam turbines geared to one propeller shaft, develops 11,000 s.h.p. in service, with a maximum power of 13,000 s.h.p. and an astern power of 7,800 s.h.p., i.e. 60 per cent of the maximum load power. There are three turbines, viz. h.p., i.p. and l.p. In the case of the h.p. turbine, there are two reductions, but the i.p. and l.p. turbines are geared directly to the main gear wheel. Steam is supplied to the turbines from two Foster Wheeler boilers, at a pressure of 500lb. per sq. in. and a temperature of 800 to 850 deg. F.—*The Shipbuilder and Marine Engine-Builder, September 1954; Vol. 61, p. 529.*

Rudder Position Indicator

The Desynn rudder position indicator gives accurate and instantaneous information of any movement of the rudder. The equipment consists of a transmitter, positioned by the movement of the rudder, and one or more indicators, sited on the bridge, in the wheelhouse, or any other place where desired. Electric transmission between the transmitter and the receiver is by means of the Desynn system. This system is self-synchronous, an important factor in the matter of rudder indication, because, in the event of a momentary failure of the power supply, the system will automatically re-align itself when power is restored. The accuracy is unaffected by large variation of supply voltage since the rotor of the receiver takes up a position depending only upon the direction of the stator field. Care and maintenance are negligible because there are no contacts which, due to excessive sparking, need cleaning. Working from a direct current supply, the Desynn system is designed for a potential of 24 v., so that it is possible to feed the rudder indicator from the ship's D.C. main through a small resistance, or from a stand-by battery. When only A.C. mains are available, the supply may be obtained by using metal rectifiers. The current consumption is low, being in the region of 0·28 amps. The transmitter and receiver are housed in watertight casings. The actuating lever shaft fitted to the transmitter is bored to form a reservoir for grease which is injected through the Tecalemit nipple to pack the annular grooves in the bearings and form a seal against the ingress of water. The receiver is supplied in gunmetal or aluminium alloy. In certain of the models a special feature is dial illumination, and the intensity of the light can be varied to suit all conditions of service. The indicators are arranged so that the pointers move into blank sectors when a power failure occurs.—*Shipbuilding and Shipping Record, 30th September 1954; Vol. 84; pp. 443-444.*

High Silicon Cast Iron Anodes

An investigation has been in progress since early 1952 to determine the merits of high silicon cast iron as an impressed current anode material in cathodic protection installation. The material is a 14·5 per cent silicon cast iron which possesses excellent resistance to a variety of corrosive media. Like other metals and alloys it has a dense, homogeneous structure which assures uniform properties throughout. It will neither absorb moisture nor flake and cannot become soft or lose electrical properties. Although unlike some other metals its tensile strength is relatively low and it can be broken by mechanical shock, it is still much superior to graphite in this respect. It can be produced only in castings which somewhat restricts the anode length that can be produced. Numerous tests in barrel cells have been conducted in synthetic sea water at different current densities.—*W. A. Luce, Corrosion, September 1954; Vol. 10, pp. 267-268.*

High-powered Tug

A shipyard in Holland is building for the Suez Canal Company a ship-handling tug stated to be the most powerful ever constructed, as 4,500 h.p. will be developed in a hull 150 feet by 39·25 feet by 18·1 feet; fire pump capacity will be 32,800 gallons per hour. This tug has twin screws direct-coupled to four-cycle engines, the screws being of variable-pitch type, presumably with an elaborate system of bridge control.—*A. C. Hardy, The Journal of Commerce, Shipbuilding and Engineering Editions, 30th September 1954; p. 5.*

Thrust Deduction

In the comprehensive historical survey with which this report begins, the authors state that it was only as recently as 1938 that sound hydrodynamical approach was devised to the problem of interaction of ship and screw propeller. This introduced the concept of singularities in a uniform flow as substitutes for the hull and propeller moving at a constant speed. Its application resulted in the first successful attempt to find a sound analogy between a propeller and a sink distribution. This novel idea in dealing with wake and thrust deduction was due to J. E. Dickmann. The present report discusses some of the remaining questions not dealt with by Dickmann, and presents a more specific approach to the problem of ship-propeller interaction. An analysis is formulated leading to a system of algebraic equations for the determination of certain body-shape and propeller characteristics of a self-propelled body of revolution, moving axially at constant speed at a maximum useful power for a given constant power input and prescribed displacement volume. The equations give a relationship between the body-shape parameters and all other parameters defining the body, propeller, relative position of body and propeller, and flow. By means of the so-called sphere theorem and Dickmann's concept, employing a single sink as a substitute for a propeller, an analogy has been established between the system sphere-sink and body-sink based on the equivalents of the interaction forces. The frictional interaction force is then approximately derived by the use of Hoerner's resistance formula and the propeller flow as given by Dickmann. The diameter of the frictional belt which is needed for determining the potential interaction force has been found by qualitative comparison of the turbulent boundary-layer flow for a plate with that for the body. The report concludes with some general recommendations for further investigation on simplification of lines, effective-wake measurements, propeller-location tests, empirical formulæ, wave component of wake, and a statistical approach. *J. Martinek, G. C. K. Yeh, and L. Crawford. Office of Naval Research, U.S. Dept. of the Navy. Journal, The British Shipbuilding Research Association, September 1954; Vol. 9, Abstract No. 9,296.*

Experiments with Model Cargo Vessels

Small coastal freighters differ from larger ships in several respects, one of the most marked being the low length-beam ratio compared with other vessels. In length, they are comparable to fishing vessels and trawlers, but the displacement-length ratio is usually higher. Also, a comparatively high speed is generally considered advantageous for such craft. Four models were used, representing small coasters to the scale 1/7·5; for the same length and draught two length-beam ratios were taken, and two displacement-length ratios for each of the two beams. The lines were devised for steel vessels, as it seemed unlikely that the freighters envisaged would be made of wood. Towing tests were carried out in the usual manner, with rudder, propeller boss, and trip wire. From the results, diagrams were prepared from which the effective horsepower can be obtained for any size of vessel of the same geometric form as the models, within the practical range for these vessels. From these diagrams, vessels of different form can be compared on a basis of displacement. Self-propulsion tests were made with all the models; three model propellers designed for some 250, 325, and 400 r.p.m. being tested in the open condition and behind one of the models. The propellers were of three-bladed, variable-pitch design. Diagrams are given of body plans and stem and stern profiles, sectional-area curves, profile for stability calculations, Chebycheff's sections, stability curves, and various experimental results.—*A. Skulberg, Norwegian Univ. Tech. Ship Model Tank Report No. 30. Journal, The British Shipbuilding Research Association, September 1954; Vol. 9, Abstract No. 9,288.*

Fretting Corrosion

According to recent investigations, the mechanism of fretting corrosion includes a chemical as well as a mechanical factor. Asperity rubbing on a metal surface is considered to produce a track of clean metal, which immediately oxidizes or upon which gas rapidly adsorbs. The next asperity wipes off the oxide or initiates a reaction of the metal with the adsorbed gas, to produce oxide. This constitutes the so-called chemical factor of fretting. In addition, asperities dig below the surface, causing a certain amount of wear by welding or shearing action, in which metal particles are dislodged. This is the mechanical factor of fretting. It is now believed that the metallic debris produced by fretting does not oxidize spontaneously, as suggested by the molecular-attrition theory, but is partially converted to iron oxide by the secondary fretting action of particles rubbing against each other or against adjacent surfaces.

This is claimed to account for the fact that ferric oxide is found by X-ray examination to be the major corrosion product, while metallic iron is present in only very small amounts. When investigating the fretting corrosion of mild steel, the humidity of the ambient air was found to be an important factor. This fact was first discovered through discrepancies in weight-loss data obtained in winter and summer respectively. It was ascertained that the fretting corrosion of mild steel in moist air is only 55 to 65 per cent of the weight loss in dry air, depending upon the length of the test. At the same time, it was observed that fretting corrosion is appreciably greater with lower temperature, the loss at 0 deg. C. being twice that incurred at 50 deg. C.—*The Engineers' Digest, October 1954; Vol. 15, p. 413.*

Stability of Whale-catchers

Comparatively small vessels with relatively high speed-length ratios have occasionally capsized in an unaccountable manner when under way in moderate weather. In discussing possible causes for these accidents, it has been suggested that the transverse stability of a ship under way is different from that when the ship is stationary. The influence of the bilge keels has been particularly mentioned in this connexion. To investigate this problem, model tests on modern whale-catcher forms have been carried out in the Trondheim experimental tank. The models represented ships 100 feet long; one of them was tested with and without bilge keels, while the other was tested with bilge keels only. This report describes these experiments, from which it was concluded that:—The "dynamic" and "static" transverse-stability curves have the same character for the two models tested, and the difference between them is insignificant. The stability increases with speed for small angles of heel (less than 10 degrees). At greater angles, a critical speed seems to appear at about 9½ knots (full-scale) where the stability is least. This minimum may be slightly smaller than the corresponding statical value. For all speeds, the bilge keels have very little influence upon the stability of these vessels.—*O. Stenvaag and A. Garberg, Norwegian University Tech. Ship Model Tank, Report No. 26, December 1953. Journal, The British Shipbuilding Research Association, October 1954; Vol. 9, Abstract No. 9437.*

Cast Nickel-alloy Steel Marine Propellers

Propellers operating on inland waterways are subject to many underwater hazards from submerged débris, which may cause breakage or distortion. The material selected must, therefore, be one which will provide an optimum combination of strength and toughness, at reasonable cost: a further requirement is that it should be capable of undergoing satisfactory repair by welding. Propellers for this type of service vary up to about 2½ tons in weight and 10 feet in diameter: a five-blade design typical of those used in tow-boats on the Mississippi River is illustrated in this article. It is a nickel-alloy steel casting of the composition shown below, which is reported as giving particularly good service in this type of work.

Carbon	0·12-0·15 per cent
Silicon	0·50-0·60 per cent
Manganese	0·45-0·65 per cent
Nickel	2·00-2·50 per cent
Chromium	0·90-1·10 per cent
Tensile strength	82,000lb. per sq. in. 36·5 tons per sq. in.
Yield strength	55,000lb. per sq. in. 24·5 tons per sq. in.
Elongation, in 2in.	25 per cent
Reduction in area	45 per cent
Bend test	120 degrees

—*Nickel Topics, 1954; Vol. 7, No. 6, pp. 1, 5. The Nickel Bulletin, September 1954; Vol. 27, pp. 170, 171.*

Cargo Vessel with 1,100 b.h.p. Doxford Engine

In taking delivery of the m.s. *Rambler Rose,* Hughes Holden Shipping, Ltd., have placed in service a cargo vessel propelled by one of the lowest-powered Doxford-type opposed-piston engines installed in a ship of this particular type. The hull and machinery were constructed by J. Lewis and Sons, Ltd., Aberdeen, and the vessel loads 1,650 tons on a mean draught of 15ft. 3in. The length is 225ft. b.p., the breadth 36ft. 6in. and the depth 15ft. 9in. to the upper deck. The engine develops 1,100 b.h.p. at 145 r.p.m., giving the ship a speed of 11½ knots. It has three cylinders with a diameter of 440 mm., the combined stroke being 1,440 mm. There are three holds forward of the engine room, with a water ballast tank below the bridge deckhouse. All the auxiliary machinery is electrically-operated, current being supplied by three McLaren Diesel-engined generating sets, two with an output of 80 kW at 220 volts, the third being a 40 kW set. The holds are served by three 3-ton winches, and all the hatches are fitted with McGregor steel hatch covers. A Cochran composite boiler is fitted at the after end of the engine room, taking the exhaust gas from the main engine and fitted with oil-firing equipment. Among the auxiliaries are two Sharples centrifugal purifiers, one for fuel and the other for lubricating oil. Starting air is supplied by two electrically driven Hamworthy compressors, scavenging air for the main engine is delivered from a reciprocating pump operated by levers from the centre cylinder crosshead.—*The Motor Ship, October 1954; Vol. 35, p. 282.*

Automatic Tensioning Winches

A mooring winch which automatically maintains a given tension on the wire which it is tending is now being manufactured in Germany. The operation of the automatic winch can best be understood by reference to the accompanying diagram. The winch itself is of the normal two-cylinder steam-driven type, equipped with two warping heads. These are at first disengaged, and sufficient wire rope for mooring purposes is paid out. The pawls on the automatic control device are then engaged, and the control lever set to wind in. The winch will now take in the wire until the pull on the wire is just balanced by the steam pressure in the cylinders. If the equilibrium is upset, say, by the tanker being lifted by a wave, the tension on the wire will increase and the wire will pay out. When this takes

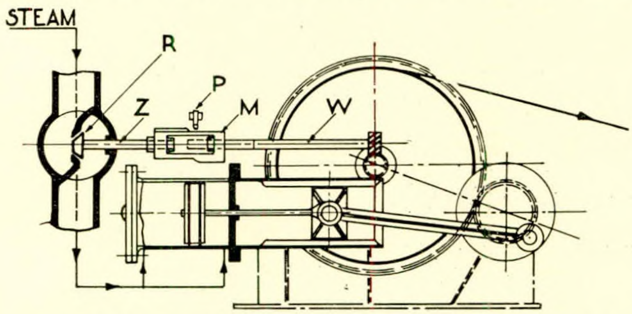


Diagram of the winch with automatic control device

place the automatic control device comes into effect. While the drum is paying out wire a shaft (*W*) connected to it is being turned through a helical gear. The other end of this shaft is threaded, and on it works a threaded sleeve (*M*), which will move up or down the shaft as it turns if the sleeve itself is prevented from turning by engaging the pawl (*P*). At the other end of the sleeve, and free to rotate in it, is a connecting rod (*Z*), to which is coupled the steam valve (*R*) controlling the admission of steam to the winch cylinders. The mechanism is arranged so that as wire pays out the steam valve is opened and more steam admitted to the cylinders, until finally the process is reversed and the wire recovered. The steam valve will by then also have returned to its original position. As far as the general design of the winches is concerned, they are of welded steel construction. The cylinders have a 14 inch stroke and 12 inch piston diameter. With a steam pressure of 100lb. per sq. in., cut-off at 90 per cent of stroke and a piston speed of 400ft. per min., the steam consumption is

145lb. per min. with a 20,000lb. pull on the wire and a wire speed of 100ft. per min.—*The Shipping World, 8th September 1954; Vol. 131, pp. 277-278.*

Mercedes-Benz Highly Supercharged 1,350-b.h.p. Engine

A Mercedes-Benz four-stroke highly supercharged engine has been developed with a continuous rating of 1,350 b.h.p. at 1,500 r.p.m., a total weight of 2,900 kg. (2·1 kg. per b.h.p.), the corresponding b.m.e.p. being 205lb. per sq. in. This is a continuous rating and on trials the engine developed a maximum of 1,625 b.h.p., the air temperature being 40 deg. C. and the air pressure 2·35 atmospheres, or 34lb. per sq. in. The engine has 12 cylinders in Vee formation with a diameter of 175 mm. and a piston stroke of 205 mm., the output at 1,350 b.h.p. being 22·4 b.h.p. per litre. There are two exhaust-gas blowers with intercooling. The fuel consumption is stated to be 152 gr. per b.h.p.-hr., or 0·335lb.—*The Motor Ship, October 1954; Vol. 35, p. 301.*

British-engined German-built Tug

Completed recently by the A.G. Weser, Werk Seebeck for the Société Chérifienne de Remorquage et d'Assistance, Casablanca (French Morocco), the salvage tug, *El Kebir*, is the largest vessel of this type built in Germany since the war. She is propelled by a 1,250 b.h.p. British Polar engine, the speed being 250 r.p.m. As auxiliary engines, two 50-kW Diesel generators and a 15-kW Diesel generator have been installed, driven by Deutz engines. The displacement is 540 tons. The vessel is 126ft. 7½in. in length, has a beam of 27ft. 11in. and a draught of 12ft. 6½in. The speed is 13 knots. The rudder is electro-hydraulically operated. The nautical equipment includes radar, a wireless station, echo-sounder, a direction-finder and gyro-compass.—*The Motor Ship, October 1954; Vol. 35, p. 302.*

Swedish Survey Ship with Geared Diesel Drive

In the survey ship *Eidern*, operated on behalf of the Swedish Government, new machinery comprising two Penta Diesel engines has been installed. The drive is taken through gearing to a single propeller shaft, running at 450 r.p.m. The engine speed is 1,800 r.p.m. and the total output 250 b.h.p. The gearing was supplied by Modern Wheel Drive, Ltd., and is of the oil-operated reverse-reduction type, allowing the propeller to be stopped, when required, with the engine still turning. The machinery is provided with remote control from the bridge, permitting rapid manœuvring to be effected. Current for lighting and running the various instruments used during survey operations is supplied by a 100 volt 8 kW generator driven from one of the main engines. The *Eidern* is 73ft. in length and has a displacement of about 95 tons. The cruising speed of the vessel is 10 knots.—*The Motor Ship, October 1954; Vol. 35, p. 303.*

Velox Boilers in French Vessels

The Velox boilers in the *Ville-de-Tunis* and the *Cambodge* are of more or less similar characteristics, and were designed at approximately the same time in accordance with similar ideas. However, as the boilers in the *Cambodge* were installed at a later date than those in the *Ville-de-Tunis,* they have benefited from the experience acquired with the latter. The *Ville-de-Tunis* entered into service at the beginning of 1952, and the *Cambodge* in July 1953. Experience over only one or two-and-a-half years can hardly be called decisive. It does, however, permit forecasts to be made of the probable future behaviour of an installation. Most of the elements of this study have been provided by the Compagnie Générale Transatlantique and by the Compagnie des Messageries Maritimes, owners of these ships. To recall briefly the principle applied in these boilers, combustion is effected in a furnace with an effective internal pressure of 2 kg. per sq. cm. (28·4lb. per sq. in.). Combustion air is raised to this pressure by means of a twenty-stage axial compressor, driven by a four-stage gas turbine using outlet gases from the superheater, and by a steam turbine supplying auxiliary power to facilitate regulation. Fuel oil is introduced into the furnace by a burner fed at constant pressure. Increas-

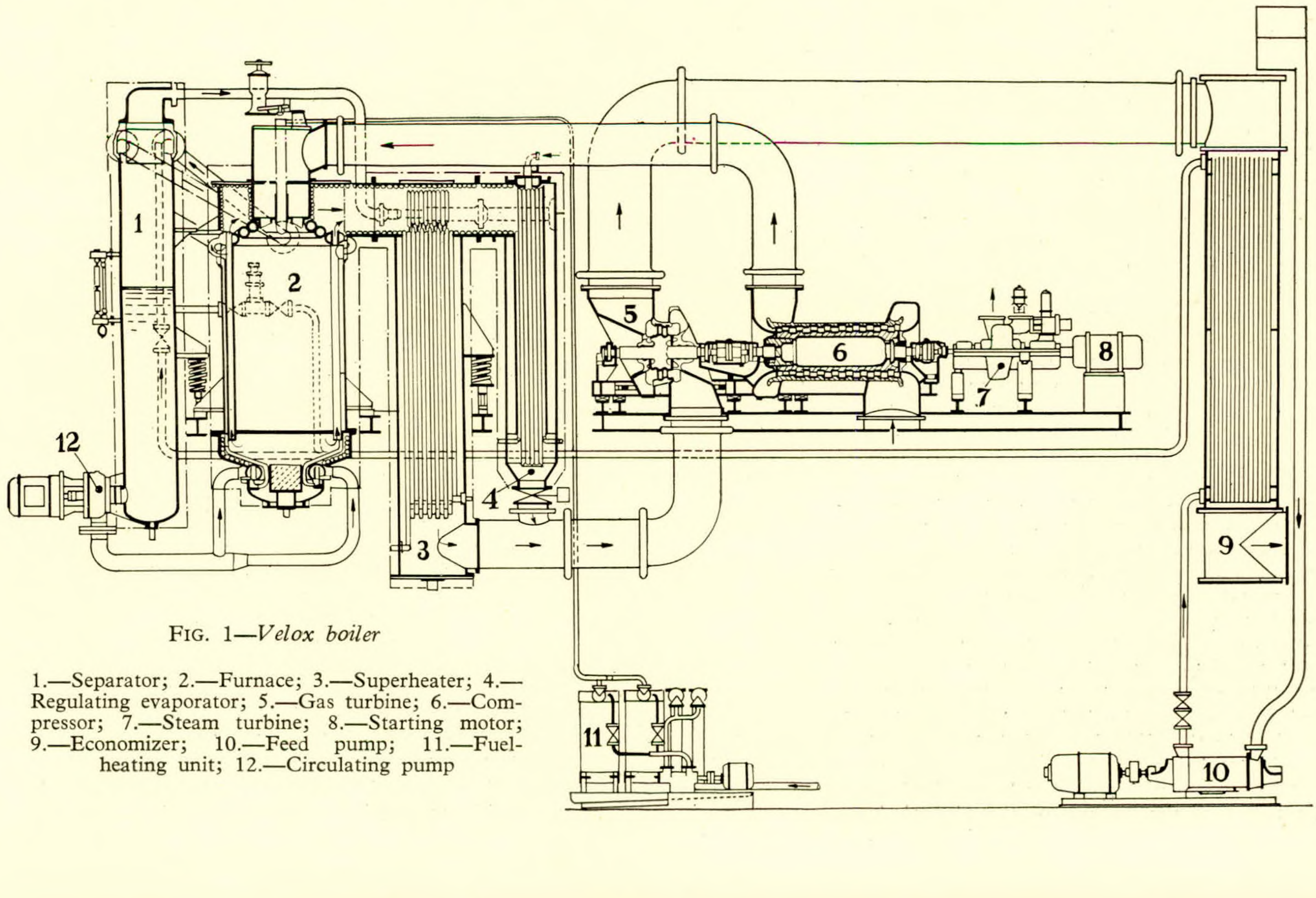


FIG. 1—*Velox boiler*

1.—Separator; 2.—Furnace; 3.—Superheater; 4.—Regulating evaporator; 5.—Gas turbine; 6.—Compressor; 7.—Steam turbine; 8.—Starting motor; 9.—Economizer; 10.—Feed pump; 11.—Fuel-heating unit; 12.—Circulating pump ing or decreasing the amount of fuel injected is accomplished by adjustment of the nozzle aperture. The flue gases pass into tubes, each of which is located at the centre of a water tube subjected, over half its exterior surface, to the radiation of the furnace. Evaporation of the boiler water takes place in the double envelopes thus formed, circulation of water through these double envelopes being assured for each boiler by an electric circulating pump. These tubes line the interior wall of the furnace. The steam-water mixture leaving the evaporator tubes is led to a cylindrical drum or separator, where water is separated from the steam by means of the turbulence produced by conveniently located baffles. From here, steam is led to the superheater coils, arranged in a third cylindrical section which receives the outlet gases from the inner tubes of the furnace wall. A fourth cylindrical section is fed with gas in parallel with the superheater; this section contains the evaporator coils, which are fed with water in parallel with the evaporator tubes of the furnace. The admission of gas into this fourth section can be controlled by means of a damper, and this control permits adjustment of the superheater temperature to the desired value. A fifth section, in this case rectangular, contains the economizer bundle and immediately precedes the funnel uptakes in the gas circuit. The gas turbine used to drive the compressor is included in the gas circuit between the superheater and the economizer. The gases flow through the circuit at high speeds; hence the coefficient of heat exchange is very high, this being due also to the high specific weight of the gases, with the result that the heat exchange surfaces can be reduced to a minimum. The walls of the superheater and the regulating evaporator, and also those of the communicating ducts between the diffierent sections, are lined with tubes in which water or saturated steam circulates. There is very little brickwork. The five sections in each boiler are arranged vertically and are aligned parallel to the axis of the ship. The charging unit is located at the side, on the level of the firing floor. The operation of these boilers is entirely automatic. Pressure in the steam separators is maintained at a constant level by means of a pressure regulator which controls (1) the amount of fuel oil supplied by the furnace by adjusting the nozzle aperture, as previously mentioned, and (2) by the speed of the compressor unit and hence the amount of air supplied. The particular cases of the *Ville-de-Tunis* and the *Cambodge* serve to define tangibly the advantages offered by the Velox boiler, which may be listed as follows:—(1) The adoption of combustion under pressure and high gas speeds leads to reduced weight and floor space. The weight of the steam-generating plant in the *Ville-de-Tunis,* and also in the *Cambodge,* is 330 tons. In the *Ville-de-Tunis* it is installed in a compartment measuring 33 feet by 62 feet by 30 feet to 49 feet, of which the upper part is particularly spacious. In the *Cambodge* the compartment measures 33 feet by 62 feet by 33 feet, arranged in front of the usual compartment for the propulsive machinery. An exact idea of the efficiency with which material and space are employed in Velox boilers is afforded by the heat release in the furnace, which here exceeds 670,000 B.Th.U. per cu. ft. per hr. (2) The separation of the tubes into different sections enables them to be given simple and identical shapes in each bundle, thus facilitating the manufacture and maintenance of replacement stocks. Above all, the tubes are assembled in compact bundles, which make the greatest possible use of available space, and through which gas flow is perfectly uniform. There are no stagnation zones where deposits of soot might have a tendency to build up. The separation of the units also ensures a certain flexibility in installation. (3) A layout parallel to the centre line of the ship can facilitate the passage of the propeller shafts through the boiler room. (4) Finally, dismantling of the bundles can be carried out in each section quite independently of the others. Conditions of operation in the *Ville-de-Tunis* and the *Cambodge* are not exactly similar. The quantity of steam normally required in the *Ville-de-Tunis* is less than that required in the *Cambodge,* being two by twenty-two tons per hour, as against two by twenty-nine tons per hour. This difference is due to the fact that the engines of the *Ville-de-Tunis* are normally intended to develop 13,000 s.h.p., their maximum rating being 14,300 s.h.p., while those of the *Cambodge* normally develop 16,300 s.h.p., with a maximum rating of 23,700 s.h.p. In both cases, two boilers are sufficient to satisfy steam requirements, but the shipowners have provided for a third reserve boiler in each ship. The operational cycle also differs. The *Ville-de-Tunis* does the Marseilles-North Africa crossing in twenty-two hours, stops for either seven or twenty-four hours, and then returns to Marseilles, where it normally remains twenty-four hours. In the case of the *Cambodge,* the round trip takes from fifty to sixty days. In each of the two ships, it has been found in practice that the need to have three boilers simultaneously in operation has never arisen, although in the case of the *Cambodge* the maximum power, i.e. 23,700 s.h.p., was designed to be achieved with three boilers, which would, therefore, operate under a smaller load. On the other hand, one boiler only is practically never left in operation.—*J. Morelle, The Shipping World, 6th October 1954; Vol. 131, pp. 390-391.*

Chilean Cargo Liner Built in France

The Ateliers et Chantiers de la Loire have floated from their dry dock construction platform at Saint Nazaire, the insulated turbine cargo liner *Lebu* for the Compania Sud Americana de Vapores. This is a vessel of full scantling type, having five holds, three forward and two aft (of which two, consisting of four chambers are for the carriage of perishable cargo). The remaining three holds have partial centre-line bulkheads for grain transport. The dimensions are as follows:—

Length overall	488ft. 9in.
Moulded breadth...	60ft. 8in.
Cargo capacity (grain)	468,000 cu. ft.
Insulated capacity (included in above)	42,700 cu. ft.
Deadweight capacity on 26ft. 10in. draught	9,250 tons

The hull design was originally based on that of the Dreyfus Line motorships delivered by this yard last year. Numerous alterations have since been made, including an increase in breadth of two feet. The cargo-handling equipment will consist of one 30-ton, four 10-ton, and twelve 5-ton derricks, served by fourteen 5-ton and two 8-ton electrically operated winches. The machinery consists of a single-screw set of 7,500 s.h.p. at 118 r.p.m., two-casing C.E.M.-Loire-Parsons double-reduction geared turbines taking steam at 510lb. per sq. in. and 750 deg. F. from two Babcock and Wilcox oil-fired boilers to give a service speed of 15½ knots. Three 300 kW Sulzer engine Diesel generators provide all auxiliary power.—*The Marine Engineer and Naval Architect, October 1954; Vol. 77, p. 377.*

Largest Deutsche Werft Tankers Built in Halves

The first of a series of 32,500 ton d.w. tankers, the *Cabimas,* was launched in halves by the Deutsche Werft recently. The largest vessel to be built at the yard, she is for the Gulf Oil Corporation of New York. Speaking after the launching, Dr. Scholz, chairman of the builders, said that he considered launching in halves subjected the hull to fewer launching stresses, and did not involve the lengthening of any building berths. A further advantage was that if the after part was launched first, as in this case, the engine department could begin their installation work several weeks sooner than if they had to wait for the whole hull. The afterpart of the *Cabimas* was 417 feet long and weighed 5,500 tons; the forepart, 213 feet long and weighing 2,650 tons, followed eight weeks later. The joining together of the two parts in this welding age presented no problems and could be carried out in two or three days. The new No. 1 floating dock at the Reiherstieg repair establishment was used for this work. The *Cabimas,* when completed, will be 630 feet in length b.p., 87ft. 3in. in breadth, and will carry her deadweight on a draught of 34 feet. The propelling machinery consists of a 12,500 s.h.p. two-casing set of A.E.G. double-reduction geared turbines taking steam from two Combustion Engineering two-drum watertube boilers. The boilers each have four Todd oil-burners and operate with automatic combustion control. Bled steam air heaters are fitted in the air supply trunks.—*The Marine Engineer and Naval Architect, October 1954; Vol. 77, p. 375.*

Naval Gas-turbine-powered Landing Craft

Since 1951 the U.S. Navy has been testing a 160-h.p. 220-lb. gas turbine in a standard landing-craft hull. This paper describes the three configurations developed in two boat hulls. Installation problems and operation data, as well as results of tests conducted at a Naval Amphibious Training Base, are revealed. The three configurations were as follows: (1) The basic installation to determine the practicability of the drive; (2) modifications to incorporate silencing and provide weather protection; and (3) installation in a watertight compartment with side exhaust in a second LCVP hull. The turbine comprises a single-stage centrifugal compressor, direct-connected to a single-stage axial-flow turbine. Exhaust from the first stage drives a second or power-stage axial-flow turbine comparable in principle to a fluid transmission. The power rotor is contained in a gear case which reduces the speed to the output shaft from 8·62 to 1. Diesel fuel is burned in two diametrically opposite mounted combustors that connect the compressor-discharge scroll to the first-stage nozzle box. Integral with the engine frame are all of the necessary auxiliaries for starting and sustaining operation. External requirements are fuel, lubricating-oil cooler and filter, and a 24-volt d.c. supply. The first stage operates at 36,000 r.p.m. at rated speed and the maximum output shaft speed is 2,750 r.p.m. The turbine is 40 inches long, 23 inches wide and 23 inches high and weighs about 220lb.—*Paper by J. O. White and J. S. Pasman, presented at the 1954 A.S.M.E. Semi-Annual Meeting; Paper No. 54-SA-43.*

Preventing Corrosion in Cargo Vessels

It is often thought that paint failures are due primarily to impact damage, abrasion or environmental attack—i.e. through external influences. Indeed, paint failures partly from this source are common in the case of ships' holds, where all three of these corrosive agents may be present. For instance, three tricky types of cargo are grain in bulk, fertilizers and timber, particularly if they are loaded straight into the hold. Most types of grain tend to collect and preserve moisture, which will accelerate the breakdown of painting systems which are not completely sound. Many fertilizers contain corrosive ingredients which are hostile to paint. While timber in log form is likely to roll in heavy seas, causing extensive abrasion of the painting system. Now, while these and other kinds of cargo may assist in establishing corrosion they are usually, in fact, only secondary causes. A far more important element in the breakdown of a given system is the presence of rust even before the paint was applied. For it cannot be over-emphasized that unless the metal surface is chemically clean —that is, free of the slightest evidence of corrosive media— any subsequently applied paint coatings are doomed to early failure irrespective of any other corrosive agents which may be present. Experts on the complex nature of metallic corrosion have agreed that the use of non-corrosive acids for rust removal, and the production of water-insoluble phosphate coatings, will greatly increase the life of painting systems as a whole. The effective action of these processes, such as Jenolizing, may be described briefly as follows: Rust removal is obtained through reaction of the acid on the corrosive media, with the result that the rust is dissolved. Since the processes are in liquid form they are able to penetrate into the pits of the metal and obtain the effective dissolution of every particle of rust. Simultaneously, the chemical reacts with the basic metal to form a water-insoluble, non-reactive, corrosion resistant phosphate coating. The effect of the physical nature of this coating (characterized by a matt finish) is to anchor the sealing coats of paint to the metal, making them resistant to impact, abrasion or attack from the atmosphere. An interesting and severely practical demonstration of the properties of pre-treatments of this kind was found recently in the case of the Norwood Steamship Company's s.s. *Lord Gladstone.* This ship, 431 feet long and with a displacement of 10,280 tons, is operated by the Ships Finance and Management Co., Ltd., London, on cargo trips between North Sea ports and the Far East. The latter company decided to treat one-fifth of the total cargo space with the jenolizing process, the remainder to be painted in the normal way and to be used for purposes of comparison. Accordingly, in the period from July to September 1952 all internal parts of the ship's hull in No. 6 'tween deck and lower hold were brush-treated with Jenolite RRN (rust remover and neutralizer), followed by a coat of zinc sacrificial primer, with a final coat of grey bitumastic paint, at Rotterdam, Holland. The remaining cargo areas were repainted at the same time. In September 1952 the ship sailed from Rotterdam to pick up cargo at Antwerp; and on its return from the Far East was given a thorough examination. Though at this early stage the relative efficiency of the various protective schemes could not reasonably be weighed one against the others, it was clear that the paintwork over the pre-treated sections was in perfect condition. On a subsequent voyage the cargo included grain in bulk and sawn timber. It was with this type of cargo in particular that the value of the process was most apparent, for such critical areas as the toe and heel of the side frames to the hull retained a completely sound painting system in contrast to the serious and widespread paint failure suffered in other, unprocessed sections of the holds.—*Cargo Handling, September 1954; Vol. 2, pp. 146-147.*

Mechanics of Cavitation

It is now generally recognized that cavitation is a dynamic process involving the formation and collapse of vapour-filled cavities in a flowing liquid. With normal liquids, these cavities form where the local pressure would drop below vapour pressure if the cavity did not form. Conversely, collapse commences when a vapour cavity is transported to a region where the pressure is above vapour pressure. There are several characteristic types of cavitation. In one common type the cavity remains fixed with respect to the boundary surface. The liquid-vapour interface is very active, with material interchange between the two phases, presumably with vaporization in the upstream regions and condensation toward the downstream end. There is physical entrainment varying from a smooth, fine-grained process to a violent large-scale process in which a major fraction of the cavity may be suddenly swept away, followed by regrowth and resumption of entrainment. There is a relatively rapid circulation within the cavity, induced by the flowing liquid. In another distinctive type of cavitation, discrete travelling cavities are formed. They move with the local liquid velocity. Such cavities have been studied in detail for cases in which the pressure distribution in the non-cavitating flow is known. The cavity first appears where the pressure drops to vapour pressure. It grows as long as the pressure is at or below vapour pressure, and starts to collapse as the pressure becomes greater than vapour pressure. Experimental measurements of the time rates of growth and collapse of such isolated spherical cavities agree well with the behaviour predicted by Rayleigh. Deviations are in the direction to be expected from the idealized boundary condition used by Rayleigh, e.g. non-viscous incompressible liquid, completely empty cavity. In this type of cavitation, as the intensity is increased, the number of cavities per unit area increases rapidly until there is interference between them. During the growth period two or more may coalesce, and in the collapse zone they often collapse in tandem; i.e. the first in a series collapses into the second, and so on. Although such interference makes analysis impossible, it is obvious that the general mechanics is the same as that for the solitary travelling spherical void. Another distinctive type of cavitation has been observed so frequently at the tips of the propeller blades that it is called "tip" cavitation. The cavity forms in the core of a vortex and appears as a long small-diameter tube. On revolving propellers these tubes appear as helices. They trace the path of the propeller tips relative to the liquid and hence do not conform with the flow lines of the liquid. Occasionally other apparent types of cavitation have been observed. Usually a little study shows that they are combinations of the three types just described.—*R. T. Knapp, Mechanical Engineering, September 1954; Vol. 76, pp. 731-734.*

Instrumentation in Engineering Research

The efficiency of modern steam turbine machinery depends, among other things, on its ability to run with small clearances between the stationary and rotating elements. Various events can occur which tend to reduce these clearances. The machinery may be mishandled in service, and manœuvred at rates for which it was never designed. The finest clearances in a turbine are in the gland packings, nowadays almost universally of the labyrinth type, and when an accidental touch occurs, with the high rubbing velocities involved, intense local heating takes place. In the case of the rotating shaft, the effect of this local heating is to make it bow towards the rub, thus aggravating the situation, and creating large out-of-balance forces—e.g. a shaft of weight 1 ton running at 6,000 r.p.m. produces an out-of-balance force of 2,290lb. per 0·001 inch displacement of the centroid from the centre of rotation. Thus serious vibration begins, the rubbing heat increases yet again, and the turbine has to shut down, the rotor removed, trued up and rebalanced—a very expensive proceeding. There is, therefore, a strong incentive for the design of gland packings which will tolerate rubbing without the generation of intense heat and which will discriminate between the amounts of heat put into the stator and rotor respectively. It was decided that Pametrada should look into this matter. Various types of gland packing were to be investigated, involving the consideration of gland strip material and the configuration of gland strips, and a rig was required in which the various configurations and materials could be proved experimentally under rubbing conditions. The first and obvious approach was to make up a great many glands according to the various ideas put forward, put them into a suitable number of full-scale turbines, arrange for these turbines to be subjected to conditions under which gland rubs were likely to take place, and see what happened. Apart from the obvious objection to this method, that it would be rather expensive in turbines, was the fact that the important quantity would be difficult to measure, this quantity of course being the rate of heat flow into the shaft surface for a given severity of rub. This would have entailed delicate and complicated measurements of temperature distribution in the material of a shaft running at 6,000 r.p.m. or so, not an easy problem. It might also have entailed the precise movement of the centre of a heavy rapidly rotating body or of a very heavy solidly mounted casing to cause similarly severe interference in each case—again not an easy problem. The method was not adopted. Instead, the experiment was related, as it should be, to the measurements which were required. The important thing was the rate of heat flow into the shaft at the beginning of the rub—the fact that it was subsequently self-aggravating was not important; this was consequential, not radical. Therefore there was no need for the shaft to rotate, so long as the required rubbing velocities were attained, and it would be much easier to make precise heat-flow measurements on a stationary shaft. After much consideration, the rig finally crystallized in the form shown in Fig. 2. The rotor is simulated by a short section of shaft about 2 inches long, kinematically mounted, and arranged to be moved into contact with the rotating gland housing element for a pre-determined time with a pre-determined amount of interference by means of an electrically driven cam mechanism. The shaft carrying the gland housing also carries a heavy flywheel, to put plenty of energy into the rub, and is driven by an electric motor through a belt increasing drive at speeds up to 6,000 r.p.m. Thermocouples are embedded in the shaft element. Time intervals are too short to allow of manual reading of the transient temperatures, so the couples are scanned by a switch rotating at 250 r.p.m., the voltages being displayed on a cathode-ray oscilloscope, which can be photographed each revolution.—*Paper by B. J. Terrell, read at a meeting of the North-East Coast Institution of Engineers and Shipbuilders, 12th November 1954.*

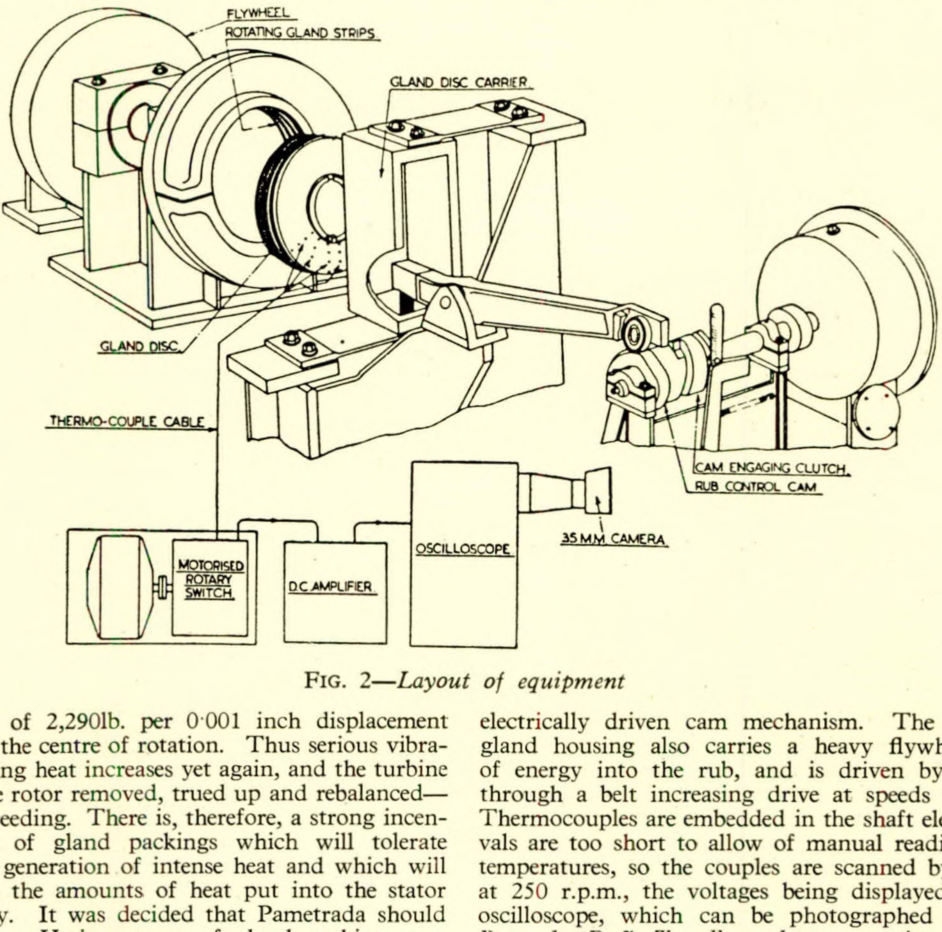
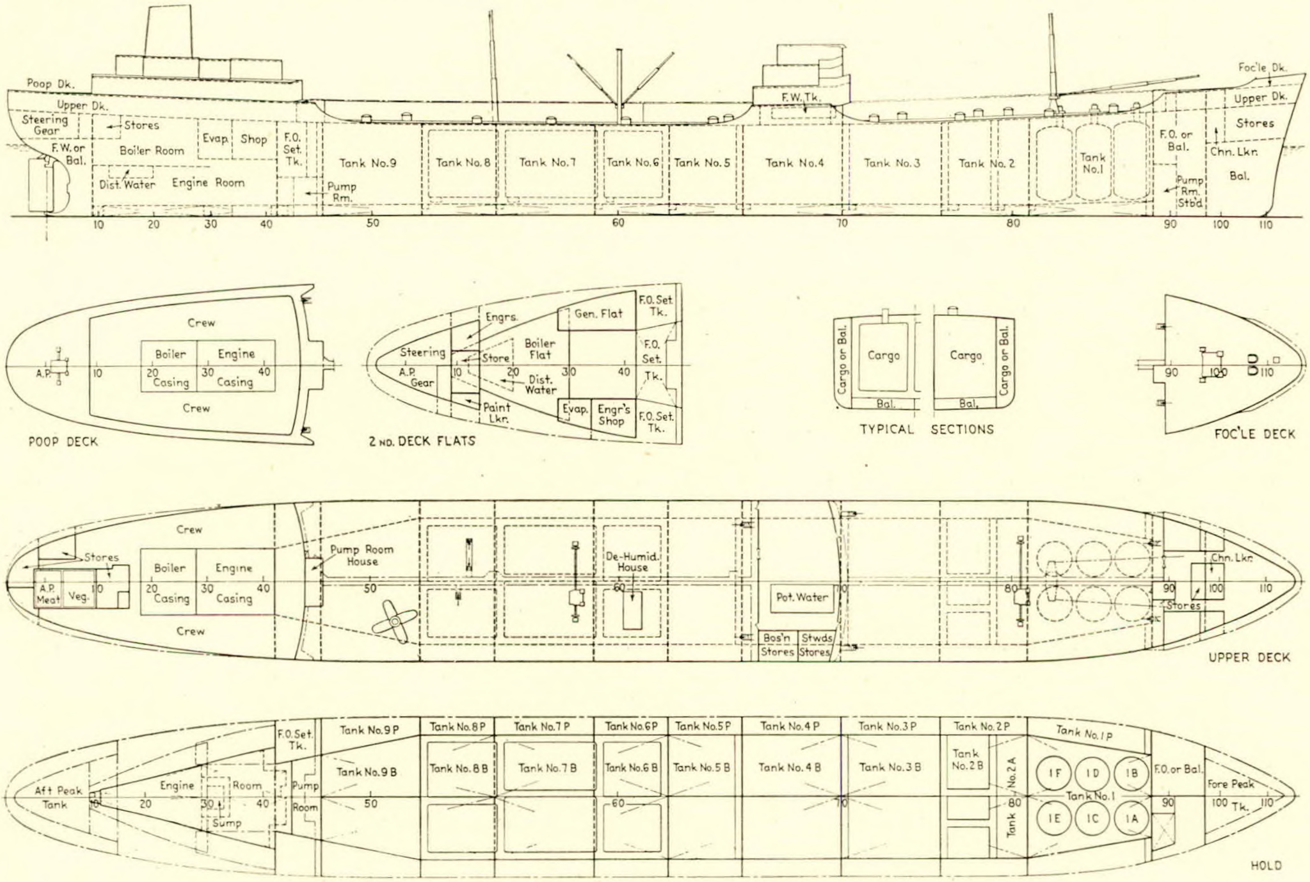


FIG. 2—*Layout of equipment*

Tanker for Chemicals

The *Marine Dow-Chem* is the first ocean-going tankship especially designed and built to transport a diversity of high-density liquid chemicals in bulk. The hull form and machinery chosen was that of an 18,000-deadweight-ton Bethlehem tanker, a number of which are in operation for several different owners. This unusual vessel, which has a capacity of 3,500,000 U.S. gal., unlike any tanker ever built, was designed, built and equipped for one role—the economical maker-to-user movement of large tonnages of liquid chemicals. To accomplish this, many structural features representing first-time engineering by both the chemical and shipbuilding industries were incorporated in the vessel. One principal difference between the *Marine Dow-Chem* and other tankers is in the protective arrangement of cargo tanks. In movement of chemicals great care must be taken to prevent contamination since quality of these commodities is measured in "parts per million". It further was required that the cargo tanks be isolated entirely from sea water—that through which the ship moved and also that used for ballasting purposes. The desired isolation was accomplished by double bulkheads, which put two thicknesses of steel between any two cargo tanks and between cargo tanks and the sea. Other notable features include cathodic protection of the hull, and extensive use of nickel and rubber linings in the cargo systems. The vessel is of the three-island type with flared stem and cruiser stern. The hull, in general, is framed longitudinally and of welded construction. Special consideration was given to the structure in way of the independent cargo tanks and to take care of the unusual loading conditions. Propulsion is by a single 19¾-ft. diameter propeller driven by Bethlehem 7,000-s.h.p. steam turbines with Falk double-reduction gearing. The high-pressure turbine is of the impulse-reaction type with a two-row impulse wheel followed by several expansions of reaction blading totalling twenty-two moving rows. The low-pressure ahead turbine is of the all-reaction type, with a hollow, conical rotor carrying twenty-five moving rows. The astern turbine in the low-pressure turbine casing is of the impulse type with one three-row wheel and one two-row wheel, separated by a diaphragm. The two Foster Wheeler boilers are designed to deliver steam at 725 deg. F. at 425lb. per sq. in. gauge throttle pressure. They are water-tube units with water walls, economizers, superheaters and internal desuperheaters in the steam drums for handling the auxiliary steam. The *Marine Dow-Chem* has the distinction of being the first American-built vessel to have cathodic protection installed when the vessel was on the building ways. One hundred and eighty anodes of Dow DX-8 magnesium weighing 60lb. each were attached to the hull by steel studs. The anodes were arranged in groups of thirty located forward, amidships and aft, on both sides of the ship. They were placed along flow lines to minimize resistance to the vessel's movement through the water. The specific cargoes which the *Marine Dow-Chem* is designed to handle are as follows: Hydrochloric acid, a widely used chemical; percloroethylene, a cleaning agent; methylene chloride, used in the manufacture of photographic film and paint removers; chloroform, a widely used raw material and solvent; carbon tetrachloride, a familiar solvent and raw material; ethylene dichloride, a widely used solvent and raw material; styrene, an ingredient of many plastics and other synthetic materials; glycols, the ingredient of many commercial products; 73 per cent caustic soda, one of the most widely used basic chemicals. Of the foregoing materials, hydrochloric acid and caustic soda presented the greatest design problem since both chemicals attack steel and both are made unacceptable for most commercial uses by even small amounts of iron-salt contamination. Each of the six huge (15ft. diam. by 41ft. deep) tanks, located in the No. 1 hold were designed for interchangeable transport of hydrochloric acid or caustic solution. These tanks were fitted with special rubber linings supplied by the Goodyear Tyre and Rubber Company. After installing the linings, steam under pressure was introduced to vulcanize the rubber. Over 35,000lb. of rubber were used in lining the tanks and connecting piping.—*Marine Engineering, September 1954; Vol. 59, pp. 38-45.*



Profile and deck plans showing cargo tank arrangement

Latest Italian Liners

The main machinery of the Ansaldo-built Italia Line passenger ships *Andrea Doria* and *Christoforo Colombo* consists of a twin-screw set of Ansaldo-Parsons geared turbines taking steam at 600lb. per sq. in. and 840 deg. F. from four Ansaldo-Foster Wheeler boilers. The turbines are three-casing sets in which the impulse-reaction h.p. elements have double-reduction and the all-reaction i.p. and double flow l.p. components have single-reduction gearing. The normal service output of 35,000 s.h.p. is obtained with three boilers. The maximum contract speed requires 50,000 s.h.p. and is normally obtained with four boilers, but it can be attained by using three boilers at higher rating. The h.p. turbine is of impulse-reaction type, having a two-row Curtis wheel and nineteen reaction stages. Steam is admitted to the first stage through controllable groups of 9, 15, 13 and 9 nozzles. A bypass valve is fitted for overloads. The primary reduction is 2·14 to 1 and the secondary ratio is 11·456 to 1. The i.p. turbines have 24 reaction stages with a bleeder point for the h.p. feed heater, corresponding to a pressure of 35·5lb. per sq. in. after the seventeenth stage. A three-row Curtis h.p. astern element is incorporated in the casing. The l.p. turbine is of double-flow type, with sixteen stages in each flow. A three-row Curtis wheel is fitted for astern running. The main boilers are of the two-furnace controlled-superheat type, with normal and maximum outputs of 50 and 60 tons per hour respectively. Howden's balanced draught system is employed and E. Green and Son's economizers are fitted to the boilers. There are two auxiliary boilers of Ansaldo-Babcock type, producing steam at 150lb. per sq. in. for the ship's domestic services and the evaporating plants. The main boilers are located in pairs fore and aft and the side-fired auxiliary boilers are installed in the wings between the main ones. A central firing space with the heaters and pumps is thus conveniently arranged. The total dry weight of the machinery, including the auxiliaries for ship's service located in the engine spaces, is 3,365 tons. The weight of the propelling machinery only, including pipework is about 3,000 tons, which for 50,000 s.h.p. corresponds to about 132lb. per s.h.p. Power and consumption trials were carried out at a displacement of 26.800 tons. These consisted of six hours at progressive speeds, followed by forty-seven hours at 23 knots with three boilers in service. The forty-seven hour consumption trial provided the following information. Fuel with a net calorific value of 17,500 B.Th.U. per lb. was used: —

Average power (torsion meter)	33,000 s.h.p.
Average fuel consumption (boilers)	17,983lb. per hr.
Average Diesel fuel to generator sets	407·7lb. per hr.
Average fuel for 50 t/day evaporators (pressure auxiliary boilers) ...	353·5lb. per hr.
Average total fuel rate (all purposes)	18,647lb. per hr.
Daily fuel rate (all purposes)	198·2 tons per day
Fuel rate for propulsion only	0·562lb. per s.h.p. per hr.

The full power trials were carried out during the following week at a displacement of 25,340 tons. These trials consisted of three hours at 23 knots with four boilers, six hours at maximum speed with four boilers and three hours at 25·3 knots with three boilers. A last, absolute maximum, run was made with 60,200 s.h.p. at 172·5 r.p.m. when 26·4 knots were attained. *Christoforo Colombo trials.* The initial displacement was 25,685 tons, some 200 tons more than the contract figure for the trials. The consumption trial held over three hours, returned a figures of 0·575lb. per s.h.p. per hr. Whereas the *Andrea Doria* has a D.C. electrical system fed by five 750 kW. Diesel generators and two 1,000 kW. turbo generators, the *Christoforo Colombo* is an all-A.C. ship, the first Italian vessel of substantial size to have such an installation. The power plant generates three-phase alternating current at 440 volts, 60 cycles. The total generating capacity is about 6,500 kW. There are four self-contained Ansaldo 1,000 kW. turbo alternators taking steam at 600lb. per sq. in. and three 750 kW. Diesel-alternators, each driven by an Ansaldo seven-cylinder four-stroke engine. An emergency generating set, arranged on the boat deck, consists of an Ansaldo Diesel engine driving a 150 kW. alternator.—*The Marine Engineer and Naval Architect, Annual Steam Number, 1954; Vol. 77, pp. 405–408.*

The Measurement of Power

The measurement of power in transmission dynamometers is difficult, although it is all-important in research, development, and last, but not least, operation. In absorption-type dynamometers, provided a size of brake is chosen appropriate to the load, accuracies of the order of ± 0·2 per cent are obtainable. The hydraulic dynamometer is at its best in low-speed work, particularly where very high powers are concerned. The dynamatic brake due to the simple form of rotor without excrescences is very suitable for high-speed power absorption, and almost keeps pace with the well-known ability of a turbine to develop large powers at high speeds. Up to the present engineers have not been able to fit a torsionmeter in the main shaft of a marine propulsion installation to give a better accuracy than ± 2 per cent. The ideal torquemeter should have an accuracy of ± 0·25 to 0·5 per cent. It should be easy to fit into the line shafting and it should measure an average torque integrated over a period many times longer than the period of any cyclic variation which may occur, either related to the shaft speed, such as torsional oscillation, or related to time, such as the slow variation in torque which takes place due to the rise and fall of the screw in the water from the motion of the ship. Most torquemeters relay on the measurement of twist of the shaft. This is a small quantity. The turning moment in the shaft, on the other hand, is a large quantity, and it could well be that direct measurement of this larger quantity would give greater accuracy. The instrument which provides the most direct measurement of torque is the hydraulic torsionmeter. At high speeds this instrument may suffer from inaccuracies arising from the difficulty of measuring pressures in a high centrifugal field, but for high torques at low speeds, where the torque-resisting pressures are very high compared with the centrifugal pressure, the desirable accuracy should be possible of achievement. This type of meter also virtually eliminates any errors due to hysteresis in the metal or variation of its physical properties with temperature or time. The rigidity modulus of elasticity of mild steel is reduced by approximately 1 per cent when the temperature varies from 40 to 100 deg. F. This is very little reduced by the effect of temperature on the shaft dimensions, so there is an error of the order of ± 0·5 per cent in the torque. It is therefore necessary if this error should be excluded from the large general torsionmeter error to have a thermometer in the shaft space adjacent to the torsionmeter so that the necessary correction for rigidity modulus can be made. Similarly hysteresis effect may change the zero setting of torsionmeters in marine propeller shafting. There are many opinions on the method by which hysteresis loss varies with number of cycles, but there is enough evidence to show that hysteresis loops plotted in static tests carried out after only a small number of cycles of stress are representative of the condition of the marine shafting on trial in which there may have been one or two astern runs before proceeding on trial. It should also be remembered that the stress in the shafting may be higher going astern due to the higher torque of astern turbines at zero speed. The error introduced by hysteresis in normal ahead operation may be of the order of ± 0·1 per cent, but going full astern may increase this figure to 1½ per cent. It is therefore all-important to check torsionmeter zeros if full astern running has occurred before accurate readings are required. Possible errors of these sorts with mild steel do suggest that a special shaft in which hysteresis is less would be of advantage where reliable shaft horsepower figures are desired. Shafts carrying torsionmeters should as far as possible be free from bending stresses. This can largely be taken care of by suitable positioning of the shaft bearings. Using higher quality material the stress in the torsionmeter shaft could be much higher than at present with a proportional reduction in error as the value of G does not vary much for mild and alloy steels. A torque stress of 10,000lb. per sq. in. for example would almost halve the error in the torsionmeter and ± 1 per cent accuracy with present commercial instruments be attainable, particularly if the ordinary structure of the torsionmeters be properly engineered. A special short torsionmeter shaft calibrated for a range of torques would be a feasible solution. This shaft could then be replaced by a normal mild steel shaft after completion of trials. The calibration of the integrating power meter could be carried out when the special shaft and torsionmeter was in place.—*Paper by T. W. F. Brown, presented at the Autumn Meeting of The Institution of Naval Architects, 22nd September 1954.*

Swedish Tanker with Turbo-compressor Engine

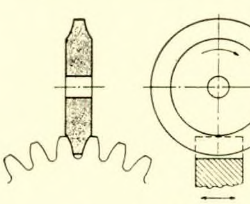
Helsingborgs Varfs A.B. have delivered the tank steamship *Soya-Lovisa* to A.B. Walltank (Olof Wallenius), of Stockholm. The ship has been constructed for carrying heavy oils and the cargo spaces are divided into six centre tanks and eight wing tanks, with double-bottom ballast tanks. There is a stepped trunk with external stiffeners in way of the centre tanks. This feature, with the double-bottom tank top and corrugated transverse and longitudinal bulkheads, facilitates the cleaning of these spaces. All the cargo tanks except No. 1 wing tanks are provided with heating coils. The principal dimensions are as follows: —

Length overall	358ft. 11in.
Length between perpendiculars	330ft. 0in.
Moulded breadth	48ft. 0in.
Moulded depth to main deck ...	24ft. 4in.
Deadweight capacity	4,585 tons
Draught	21ft. 5¼in.

The ship is rigged with one mast and two derrick posts on the foredeck. The windlass and steering gear are of the builders' own design and make. Accommodation for the deck officers is provided in the deckhouse amidship and for the engineer officers in the deckhouse aft. The crew are berthed aft under the poop deck, mostly in single-berth cabins. The ship is equipped with an echo-sounder, SAL log, radar, gyro-compass and gyro-pilot. The propelling machinery consists of a Helsingborg-Götaverken quadruple expansion steam engine equipped with a turbo-compressor and developing about 2,500 i.h.p. at 95 r.p.m. Steam is generated in two Prud'hon-Capus oil-fired boilers fitted for Ray automatic combustion.—*The Marine Engineer and Naval Architect (Annual Steam Number), 1954; Vol. 77, pp. 421-423.*

Manufacture of Gears

Manufacture of cut gears for high-power capacity is done on conventional gear-hobbing machines. Available accuracy is determined by the worm and gear which turn the large gear blank, by the rigidity of the cutter mounting, and the uniformity of the hardness of the blank. The large forces occurring during cutting, and the resulting deflections of blank and tool, with their mountings, will affect surface finish of the cut teeth and, in case of non-uniform hardness, the spacing and tooth form. Subsequent shaving will improve surface finish and tooth form but will not correct spacing errors. Grinding of gear teeth will permit correction of virtually any discrepancy produced in cutting and hardening, if the grinding process is done in a manner that will not reflect grinding-wheel wear into the gear teeth. However, it is important to use a type of grinding wheel that is subjected to continuous and rather rapid wear. If the wheel is of the hard-bonded type which wears only slowly, then the same grit will come in contact repeatedly with the work, and the originally sharp edges of the grit will be dulled. Then the grinding wheel will rub on the surface of the teeth, producing friction and high surface temperatures. The sharper the grit of the grinding wheel, and the smaller the contact area between grinding wheel and gear teeth, the less will be the heat generation.



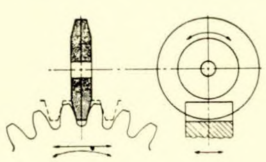


FIG. 8—*Formed-wheel gear-tooth grinding*

FIG. 9—*Gear-tooth generating grinding with straight-contour wheel*

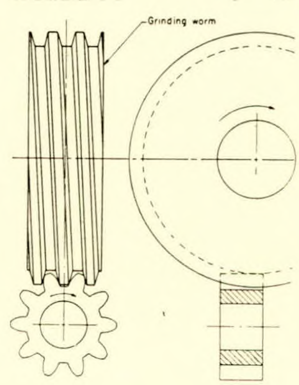


FIG. 10—*Gear grinding with worm-type wheel*

Processes like form-grinding, Fig. 8, or generating with a straight-sided wheel, Figs. 9 and 10, will give good results only on small gears, and with application of ample liquid coolant. On gears having large numbers of teeth, grinding-wheel wear in these processes would result in errors in tooth form and spacing. If wheel wear is reduced by use of hard-bonded wheels, then the danger of overheating becomes acute. Overheating during grinding would lead to softening of the carburized surface and to tensile-stress areas on the tooth flanks. If grinding extends into the roots, this area would be sensitive to grinding burns. Gears that were ground by one of the foregoing processes should be checked for grinding burns by applying a mixture of alcohol and nitric acid (nital test) which produces dark discoloration on areas overheated during grinding. A process that permits continuous grinding-wheel wear together with smallest contact areas is shown in Fig. 11. In this process the grinding wheel touches the gear teeth only in two points, at the thin outer edge of the wheel. The latter is of the soft-bonded type which breaks down fast and brings new sharp grit into operation. The wear of the wheel is compensated automatically. A flat diamond touches the edge of the wheel every few seconds, and if the edge has worn more than a preset amount (0·00004 inch) the wheel is readjusted automatically to the original position. Thus any number of teeth on gears of any size may be ground

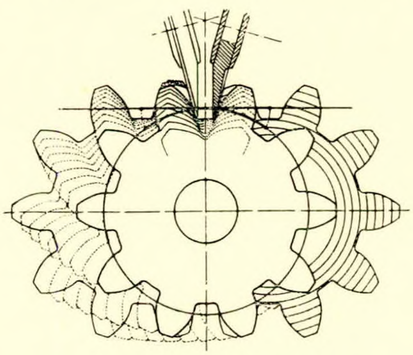


FIG. 11—*Gear-tooth point-generating grinding process*

uniformly within the capacity of the machine.—*D. W. Botstiber, Mechanical Engineering, September 1954; Vol. 76, pp. 735-738.*

Ship Structural Members

This paper presents the results of further investigations carried out at Glengarnock on behalf of the British Shipbuilding Research Association, on the strength and stiffness of ships' structural members, and describes a series of tests of a rather more fundamental nature than those already reported in earlier papers. An attempt is made to determine to what degree the simple beam theory applies to stiffened panels of plating, and to confirm the Glengarnock experimental technique. The sections consisted of flat bars welded to strips of plating, the majority of the specimens being freely supported at the ends. One was of uniform section, but the others had bracket-like appendages at the ends which were welded to the stiffeners, although unconnected to the end structure of the machine. By this means, the effect of brackets on the stiffness of beams—as distinct from their function as a means of applying fixing moments—has been investigated. Broadly, the tests indicate that the simple bending theory applies within close limits provided that the stiffener and plating are of uniform cross-section. Where the cross-section is rapidly changing, however, as at brackets, and at uniform sections in the vicinity of brackets, the simple theory does not hold.—*Paper by J. McCallum, read at a Meeting of the North-East Coast Institution of Engineers and Shipbuilders, 29th October 1954.*

Mechanism of Cavitation

An experimental investigation of incipient cavitation is described and the relationship between these experiments and current theories is discussed. Experiments have indicated that the cavitation number for inception varies with free-stream velocity and body size for both streamlined and bluff bodies. Preliminary studies of the mechanism of cavitation were conducted in order to understand the reasons for the variations in cavitation performance. Photographic investigations have disclosed the effect of the boundary layer on cavitation inception for both streamlined and bluff bodies. In addition, this study has shown that tensions exist in the flow of ordinary water at incipient cavitation.—*R. W. Kermeen, J. T. McGraw and B. R. Parkin, 1954 A.S.M.E. International Meeting; Paper No. 54-Mex-1.*

Admiralty Experiment Works, Haslar

The paper is presented as a tribute to William Froude on the occasion of the unveiling of the memorial to his genius on the site of the original experiment tank at Torquay. His main work was carried out for the Admiralty, and continued at Torquay after his death for a few years until transferred to Haslar nearly seventy years ago. An account of the investigations at Torquay has been given previously and the present paper deals with the work at Haslar. The scope has widened to meet the exacting requirements of the modern Navy, but the methods rest on the sure foundations securely laid by William Froude. Formerly the investigations were largely concerned with propulsion. This item is now more comprehensive, and seaworthiness, manœuvring, vibration and other qualities are also fully considered. The approach is through model experiments, ship trials, associated theory and research. The original ship tank at Haslar is now supplemented by one of greater size and carriage speed. There are two cavitation tunnels and other laboratories. Some account is given of the facilities and the requirements which led to their provision, together with examples of some of the lessons learnt, including the value of regular tests of standard models. The programme is closely co-ordinated by the Director of Naval Construction to suit the requirements for designs of H.M. ships. Valuable guidance and help is afforded by Sir Geoffrey Taylor, Sir Thomas Havelock and other consultants. Not the least of our debts to William Froude was the training of his gifted son and successor. A brief appreciation is included of R. E. Froude's achievement at Haslar.—*Paper by R. W. L. Gawn, O.B.E., presented at the Autumn Meeting of the Institution of Naval Architects, 21st September 1954.*

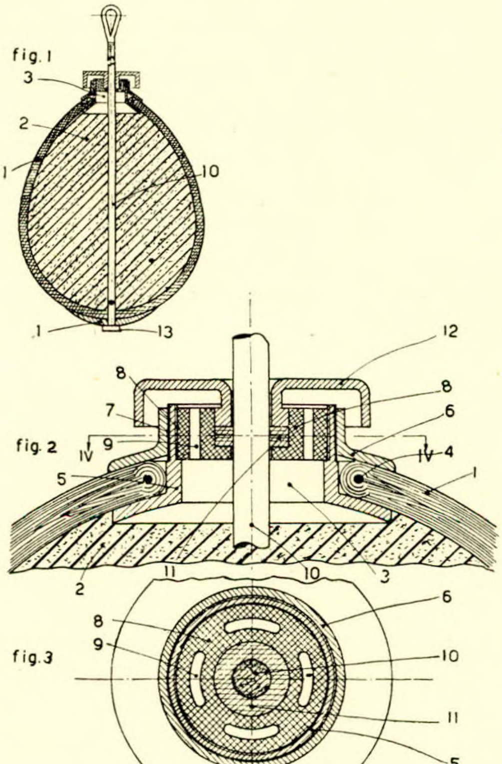
Progress in Stainless Steels

A new alloy has been introduced which is said to be suitable for replacing stainless steel in many applications. It consists of 84 per cent iron and 16 per cent aluminium. If an addition of vanadium or molybdenum to the extent of about 5 per cent is made, the alloy combines a remarkable range of properties in a single material. It is 25 per cent lighter than stainless steel and it is claimed that it can be heated to 1,300 deg. C. with low heat losses; also it has a high resistance to corrosion and oxidation. Another alloy which is being recommended as a substitute for nickel stainless steel wire in applications requiring good workability, resistance to corrosion, low electrical conductivity and absence of magnetism is a nickel-free, high strength steel wire. The alloy contains, in addition to steel, about 14·5 to 18·5 per cent manganese, 9·0 to 11·0 per cent chromium and small amounts of silicon, copper, carbon and nitrogen. In many instances, bolts, screws, pins, bushes, etc., made from stainless steel, which are to be used in contact with light alloy components, are now being cadmium-plated to reduce the risk of contact corrosion with the light metal. Because of the passive film on such steels, special plating techniques are essential, and the recommended procedure is contained in a Ministry of Supply Memorandum R.D. Mat. (A) No. 3. Among the interesting new applications of stainless steel is a float trap designed to drain corrosive liquids from air, gas or steam confined in piping systems. It is claimed that it does not allow air or gas to escape while liquid is being drained, and the float is designed for operating at 300lb. pressure and 800 deg. F. maximum temperature. Galling of stainless steel nuts and bolts is a recurring problem which is being overcome in a number of ways. One is to ensure that the one stainless steel is much harder than the other, for example, an 18-8 type may work against one of similar composition, but chromium-plated, or against a hardened straight chromium or a 2 per cent nickel steel, hardened by heat-treatment. Alternatively, the threads may be coated with an oil containing carborundum powder of about 400 mesh in suspension. Another remedy is to apply litharge and glycerine or a compound of powdered mica and any good vegetable oil. A colloidal silver powder painted on with a brush has been found useful for austenitic steel bolts subjected to a temperature of about 650 deg. C. Finally, there is a film combining pure graphite and molybdenum sulphide, which amalgamates with the material to be covered. It is applied by vapour blast and later heated to 204 deg. C. for 40 minutes to produce the amalgam. The non-galling films should be of a thickness within the range 0·0002 to 0·0003in. A special 17-7 chromium-nickel stainless steel is being successfully used for check valves. In the soft condition, this alloy may be readily formed and blanked into check valve discs. The correct hardness is attained without appreciable distortion, by a double low-temperature heat treatment, which produces a 0·2 per cent offset yield strength of 74 tons per sq. in. in tension or compression, as a minimum. Check valves of this alloy show no evidence of deterioration after having been in operation for over twelve months, and it is estimated that they will last indefinitely in the same service. A hardenable corrosion-resistant casting alloy of ferrous type, which has hitherto been available in wrought form only, has now been put on the market. It contains carbon 0·05 per cent, chromium 16·5 per cent, nickel 4 per cent, copper 4 per cent, and is claimed to show resistance to corrosion superior to that of the 12-14 per cent chromium-iron alloys, and to be suitable for use in abrasive conditions. In the solution-treated condition, it has a tensile strength of 60·5 to 73·5 tons per sq. in., and in the hardened condition, of 80·5 to 94 tons per sq. in.—*J. Lomas, Engineering and Boiler House Review, November 1954; Vol. 69, pp. 343-344.*

Patent Specifications

Fender

The fender according to the invention comprises a strong and comparatively thin flexible covering, a deformable yielding core inside it and a suspension cable penetrating into the covering through an air passage and anchored to the covering at the point opposite to the air passage. According to Figs. 1-3 the fender consists of a strong flexible covering (1)



which is completely filled up by a core (2) of sponge rubber. The flexible covering (1) of parabolid shape has a large circular opening (3) reinforced at its edge by a steel wire (4). The edge of the opening is enclosed between two parts (5 and 6) screwed together (7) forming a nozzle within which is located a disc (8) provided with peripheral ports (9) for the passage of air and with a central opening for the cable (10). Tight closure of the cable passage is ensured by the stuffing box (11) in disc (8). The cap (12) protects the ports (9) from water spray.—*British Patent No. 719,575, issued to Pirelli Societa per Azioni. Application in Italy made on 30th April 1951. Complete specification published 1st December 1954.*

Forced Through-flow Boiler

This invention relates to forced through-flow boilers. The first forced through-flow boiler which could be operated practically was developed as a high pressure boiler. The circulation of the operating medium steam/water through the tube system was effected by means of a pump. From the knowledge that the difficulties encountered with high-pressure boilers are mainly due to the sharp demarcation between steam and water in the vaporization zone, a proposal was made to raise the pressure and temperature of the working medium in the boiler at least up to a critical value in order to generate the steam to a certain extent by a limit transition at the critical point, that is to say, in order that operating medium is homogeneous and to prevent the operating medium being present simultaneously in two different states of aggregation. As compared with the state of the art at that time, this principle was shown to be correct and valuable. It was, however, shown that this principle was not so easy to carry into practice as was thought, since the salt content of the feed-water settled in the tubes in and after the zone of complete vaporization, with the result that undesirable burning of the tubes occurred. A way was found of removing this salt-endangered zone from the radiation section to an area of lower gas temperature. It is true that the salt deposit as such could not be avoided, since this was due to physical reasons. However, the salt deposit no longer assumed the form of a hard insoluble crust, but was deposited in a form in which it could be removed comparatively easily by flushing. Boilers constructed on this basis proved to be very satisfactory and represented the first serviceable high pressure boilers. There always remained one point which was undesirable, namely, the necessary high working capacity of the feed pump due to the high pressure (227 atm. and more). Attempts have been made to operate with a pressure below the critical pressure by using the forced circulation principle and the cooler vaporization zone. The experiments produced surprisingly favourable results and thus led to the modern forced through-flow boilers which operate below the critical pressure. In the meantime, on the basis of further experience, the procedure was adopted of preparing the feed-water in such manner that it is practically free from salt. By this means, it was in itself no longer necessary to displace the last part of the vaporization zone to a lower temperature field, since it was now no longer necessary to expect disturbing salt deposit. On the basis of new knowledge, it is now proposed, in accordance with the invention, to increase the boiler pressure again up to, and in certain cases beyond, the critical pressure, and to design the superheater surface in such manner that a steam temperature of about 1,110 deg. F. is reached or exceeded, and also to construct the vaporization heating surface as a radiation heating surface, while feeding the boiler with water which is practically free from salt. The invention has advantages if the power plant is operating with reheating, that is to say, when a reheater is built into the boiler, and particularly when the reheating temperature is approximately equal to the live steam temperature. However, desirable results are also obtained with boilers for power plants without reheating when the turbine works as an exhaust turbine. In this way, the old idea of the so-called "Benson boiler" is taken up again, but fundamentally modified in two respects, viz. first as regards the temperature level and secondly as regards the feed-water. It is essential for the purpose of the invention that only practically salt-free water is used for the forced through-flow boiler operating with at least critical pressure. Practically free from salt means that the residue does not amount to more than 1-2 mg/l. It is only possible to expect adequate reliability in operation if this condition is fulfilled. Finally, the following is to be said concerning the construction of the vaporization heating surface as a radiation heating surface. It has been possible, when operating the forced through-flow boiler with salt-free water, to dispense with the subdivision of the vaporization heating surface. The maintenance of the subdivision is carried out from the point of view of avoiding corrosion effects in that zone at which the vaporization is completed. At this point, the proportion of steam greatly exceeds the proportion of water, but it is probably just this quantity ratio between steam and water which favours the corrosion by O_2, CO_2 and the like. However, the use of the critical pressure and the critical temperature prevents the operating medium being present simultaneously in two different states of aggregation, it being always homogeneous, a strict distinction between steam and water no longer being possible upon reaching the critical temperature. Thus the essential condition for corrosion is also removed. As the cause of the trouble is eliminated, there is no longer any reason to retain the feature of a vaporization surface having a higher and lower temperature zone. The complete vaporization process can now be shifted to the radiation heating surface and thus it is possible to arrive at a substantially lower constructional cost for the forced through-flow boiler. A forced through-flow boiler which operates at least with critical pressure, operates with feed-water which is practically free from salt, the vaporization surface of which is constructed as a radiation heating surface and the superheating temperature of which reaches or exceeds approximately 1,110 deg. F., represents an essential technical advance over the forced through-flow boiler according to the existing state of the art.—*British Patent No. 717,637, issued to Siemens-Schuckertwerke A.G. Complete specification published 27th October 1954.*

Oil-operated Reverse-reduction Gear Drive with Multiple Engines

An arrangement of oil-operated reverse-reduction drive for an installation of multiple engines is shown in Fig. 1. The input shaft (1) carries the ahead and astern pinions (2, 3) which are free to rotate on the shaft and are provided with clutches

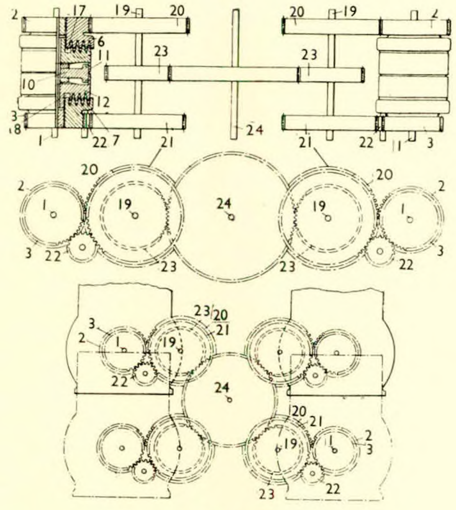


FIG. 1

(6, 7). Between the clutches is a disc (10). The clutch members (11, 12) are splined on the shaft (1). Oil for disengaging the clutches is supplied through cavities (17, 18). The idler wheels (22) give the desired reverse rotation. The intermediate shafts (19) carry fixed pinions (20, 21). The wheels (23) transmit the drive to the output shaft (24), on each side of which are, for example, two or four engines, any of which can be coupled while the others are in motion. The reverse slipping clutches are in engagement before the shaft comes to rest.—*(British Patent No. 709,530. T. Hindmarch, Chesham.) The Motor Ship, October 1954; Vol. 35, p. 309.*

Double-acting Portable Hand Pump and Deck Socket

This invention has for its object the provision of an improved double-acting continuous flow portable hand pump of the plunger type, to provide a large pumping capacity in a pump of short length and of easily manageable proportions and designed for quick attachment to one or more mating sockets or deck fittings. To enable this to be achieved a pump working on a double-acting principle is employed, having one suction valve in the base and two delivery valves and one suction valve in a detachable head, delivery and suction to these valves being effected by the use of transfer tubes. The barrel (A) Fig. 1, together with two transfer tubes (B and C) are secured between head and base castings (D and E). The base casting is divided into two separate chambers (a and b) communicating one with the other through a foot valve (G). The chamber (a) is on the suction side of the valve (G) through which fluid is drawn on the up stroke of the plunger (N). The other chamber (b) communicates via the transfer tube (C) to the pump head casting (D) (also divided into two chambers (d and c)) through the suction valve (K), thus enabling fluid to be drawn into the pump on the down stroke. Delivery takes place in a similar manner through two delivery valves (L and M) in the head casting, one (M) communicating directly with the upper side of the pump plunger (N) and the other (L) with the lower side via the second transfer tube (B). The pump base is provided with a short coarse screw thread and taper seating for quick attachment to a suction pipe socket or boat's deck fitting, provided with a

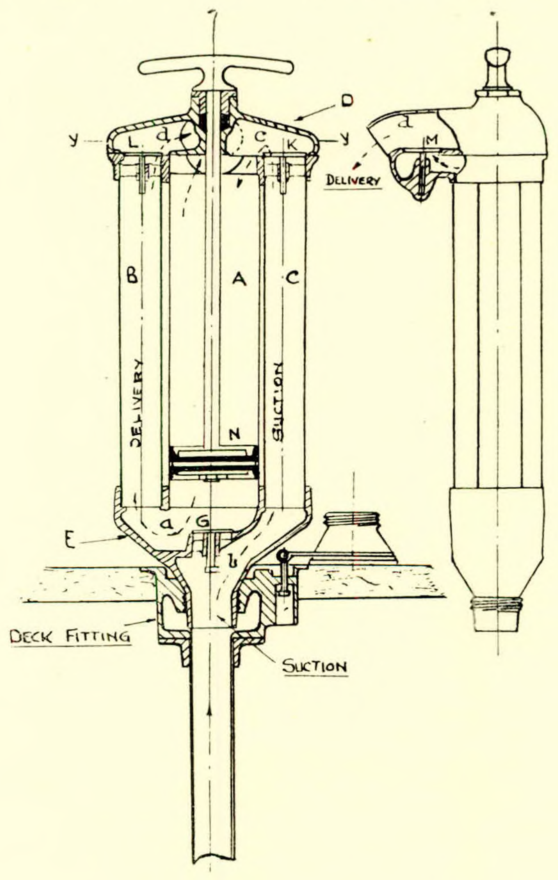


FIG. 1 FIG. 2

rubber conical seating and hinged plug for sealing the socket when the pump is not in use. The pump is provided with a detachable head for easy access to valves and plunger.—*British Patent No. 718,655, issued to E. Clare. Complete Specification published 17th November 1954.*

Heating and Circulating Heavy Fuel for Sulzer Engines

A method of heating and circulating fuel round the supply system, including the pumps and injectors of a Diesel engine using heavy oil, is shown in Fig. 4. Fuel is drawn from a tank

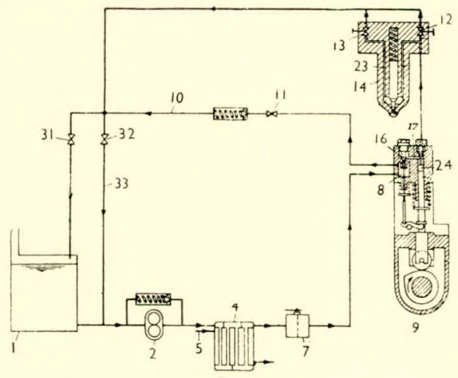


FIG. 4

(1) by a pump (2) and delivered to the heater (4), which may be supplied with steam through a pipe (5). The fuel then passes through a filter (7) into the suction space (8) of the injection pump (9). In the first instance, all the fuel is delivered to the pump and returned to the fuel tank through a pipe (10). When the valve (13) is opened, the heated fuel flows into the passage (23) and raises the temperature of the injection valve (14). Before the engine is started, the valves (12, 13) are closed and the valve (11) is opened. The relatively small amount of fuel for injection is taken through the suction valve (16) into the pump cylinder (24). The major portion of the fuel returns from the suction space (8) through the pipe (10) to the tank (1). If the fuel is too hot for returning to the tank on account of its liability to ignite, it is circulated through a bypass pipe (33). Only the amount of fuel actually required for injection then flows out of the tank into the supply pipe. The valves (31, 32) are adjusted to suit the conditions.—*(British Patent No. 712,143. Sulzer Frères S.A., Winterthur.) The Motor Ship, October 1954; Vol. 35, p. 309.*

Air Supply for Supercharged Opposed-piston Engines

An engine fitted with an exhaust gas turbo-blower for supercharging is illustrated in Fig. 3. Either a steam turbine or an electric motor is employed to drive the blower when the exhaust gas supply is insufficient at reduced loads. The blower

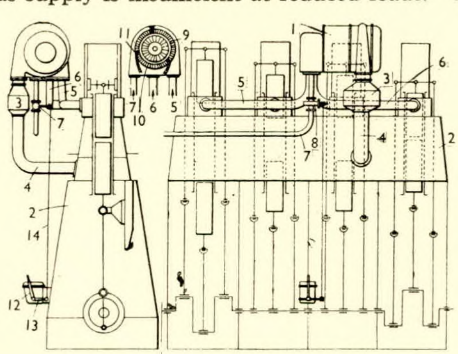


FIG. 3

(1) is shown mounted on one side of the engine (2), the air being delivered through an intercooler (3) and a pipe (4). During normal running, exhaust gas is supplied through pipes (5, 6). For starting and at slow speed, the turbine is supplied with steam through a pipe (7) and a control valve (8). The nozzle ring is divided into three sections (9, 10, 11) of which two (9, 10) take exhaust gas and one (11) takes steam. The valve (8) is controlled by the lever (12) so that with an increase in speed, the lever (13) and the rod (14) progressively close the valve (8). The specification describes a clutch which may be fitted between the steam turbine, air motor or electric motor and the turbo-blower. The clutch may be operated pneumatically or a switch may be used to control the supply of current to an electric motor. The auxiliary drive is stated to be capable of running the rotor and blower at about half the normal full-load speed and the power required is about an eighth of that needed for the full engine output.—*(British Patent No. 711,904. Wm. Doxford and Sons, Ltd., W. H. Purdie and P. Jackson, Sunderland.) The Motor Ship, October 1954, Vol. 35, p. 309.*

Evaporating Plants

This invention relates to evaporating plant of the type in which the vapour from the liquid being evaporated is led to and compressed in a compressor, from here it passes to a heat exchanger in the evaporator where the heat contained in the compressed vapour passes to the liquid being evaporated. The invention consists in an evaporating plant comprising an evaporator vessel, a heat exchanger associated with this vessel, and a compressor arranged to withdraw vapour from the vessel and to deliver compressed vapour to the heat exchanger. An open pipe for the passage of air into and out of the heat exchanger is connected to the vapour space of the condensate drainage compartment of the heat exchanger, and the area of the heating surface of the heat exchanger is considerably greater than that necessary to condense the compressed vapour in the heat exchanger when the heating surface is unscaled. Referring to Fig. 6, the main evaporator vessel (1) is provided with a

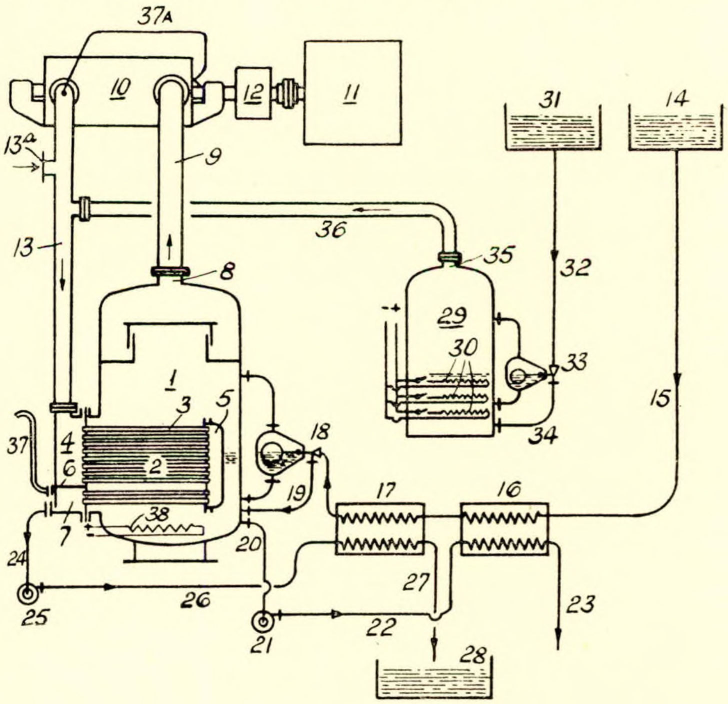


FIG. 6

heat exchanger (2), comprising tubes (3) secured at one end to a steam inlet and to a floating header (5) at the opposite end. The header (4) is divided into two compartments by a rib (6) which divides the tubes into two banks. The tubes in the upper bank are so arranged that the heating steam enters these tubes at the inlet header (4), passing along the tubes to the header (5), from which the condensate and any residual steam flow back through the lower bank of tubes into a drainage compartment (7) of the header (4). The header (4) forms a door on the evaporator vessel (1) whereby the complete heat exchanger (2) may be removed from the evaporator vessel (1) for servicing. The evaporator vessel (1) is provided with a vapour outlet (8), which is connected by a pipe (9) to the suction of a vapour compressor (10), which is driven by an electric motor (11) through gearing (12). The compressor (10) discharges compressed vapour through the discharge pipe (13) to the header (4) from which it passes into the tubes (3) of the heat exchanger (2). The temperature and pressure of the compressed vapour within the tubes (3) are greater than the pressure and temperature of the vapour in the evaporator (1). Raw water to be evaporated is contained in a supply tank (14) which may be arranged at a level high enough for water to be fed by gravity into the evaporator. Blow-down from the evaporator may be withdrawn through the pipe (20) by the brine pump (21) and discharged through the pipeline (22) and heat exchanger (16) and pipe (23) to waste. The condensate formed as the result of condensing the steam within the tubes of the heat exchanger (2) is withdrawn from the drainage compartment (7) through the pipe (24) by the drain pump (25), and discharged by way of the pipeline (26) through the heat exchanger (17) and pipeline (27) to a distilled water tank (28); (29) denotes an auxiliary evaporator containing electric heating elements (30) which are arranged below the water level in the auxiliary evaporator (29). A water tank (31) is provided to supply distilled water or other fresh water with a minimum solid content by way of the pipe (32), feed regulator (33) and pipe (34) into the auxiliary evaporator (29). A pump may be provided to discharge the distilled water from the tank (31) into the evaporator (29). The use of distilled water is necessary in order to reduce to a minimum the formation of scale on the surface of the electric heating elements (30). Arrangements may be made whereby the number of electric heating elements in service can be varied to suit operating conditions. The distilled water in the auxiliary evaporator (29) is evaporated by the heat supplied by the electric heating elements (30), and the vapour formed is led through the evaporator outlet (35) and pipe (36) to the heating surface of the main evaporator by way of the pipe (13) and the main inlet header (4). It will be understood that the vapour pressure in the auxiliary evaporator (29) is slightly higher than that in the heat exchanger (2) of the main evaporator (1). If so desired, provision may be made for a supplementary supply of steam to the heat exchanger (2) from an external source such as a steam generator, said supplementary steam supply being fed, for example to the pipe (13) by way of the branch (13a).—*British Patent 712,834, issued to G. and J. Weir, Ltd. and H. Hillier. Complete specification published 4th August 1954. Engineering and Boiler House Review, October 1954; Vol. 69, pp. 324-325.*

Steam Traps

This invention relates to steam traps with float-controlled shut-off elements. The steam trap, according to Fig. 7, comprises a housing *a*, a cover *b*, and an outlet pipe *c* integral with the cover *b* and supporting a slide plate *d* and a guide *e* having guiding grooves *f*, f_1. A float lever *g* is connected to a float *h* and pivots around a bolt *i*. The lever *g* is provided with a trunnion or journal *k* which slides in an elongated hole *l* of a control arm *m* rigidly connected to a box-like housing *n*. The housing *n* has two pivots *o* which engage in the guiding grooves *f*, f_1 of the guide way *e*. The housing *n* surrounds, with some free space and on 3, 4 or 5 sides, a shut-off element *p* of rectangular prismatic or cube-like shape. The shut-off element *p* can seat and seal without any hindrance on the sliding plate *d* and so close the outlet aperture *r*. The float lever *g* is provided with a nose *q*. By means of the nose *q* of the float lever *g* the housing *n* and thus the sealing member *p* can be loaded additionally in the sealing position. The control arm *m* has a lug *s* at its free end and the lever *g* engages this lug to move the control arm as shown in Fig. 8. The steam trap described above operates as follows: The float lever *g* is turned upwards as the float *h* rises, thereby pushing the control arm *m* upwards by means of the trunnion *k* moving in the elongated hole *l*: the housing *n* tilts the shut-off element *p* around the pivots *o*, thereby relieving the sealing member *p* of the pressure, the float *h* being in position II of Fig. 7. Consequent to this relief of pressure the float *h* rises more quickly, the housing *n* sliding in the guiding grooves *f*, f_1, together with the sealing member *p* to one side of the outlet *r*, the float *h* now being in position III in Fig. 7. Due to the escape of the condensate the float *h* sinks, the housing *n* moving along the groove *f*, f_1 together with the shut-off element *p* back to their initial positions, as shown in position I of Fig. 7, at which position the shut-off element closes the outlet *r*. By means of the lug *s* the travel of the housing *n* is extended and the opening process is accelerated. Illustrated in Fig. 9 is a trap in which is used a double-armed or bell-crank lever *g*, g_1, which is pivoted around the bolt *i*. The shorter arm g_1 of the lever has a trunnion which engages in the elongated hole *l* of the control arm *m*. When the float lever arm *g* moves upwards in the direction indicated by the arrow the control arm *m* is pressed downwards so that the housing *n* and shut-off element *p* are tilted and the latter is relieved of the steam pressure. When the float lever *g* moves on in the direction indicated by the arrow, the lug *s* provided at the end of the lever arm g_1 pushes against the housing *n* and, therefore, pushes the shut-off element *p* to one side of the outlet *r*.—*British Patent No. 712,949, issued to P. Graeff. Application in Germany made on 15th June 1951. Complete specification published 4th August 1954. Engineering and Boiler House Review, October 1954; Vol. 69, p. 325.*

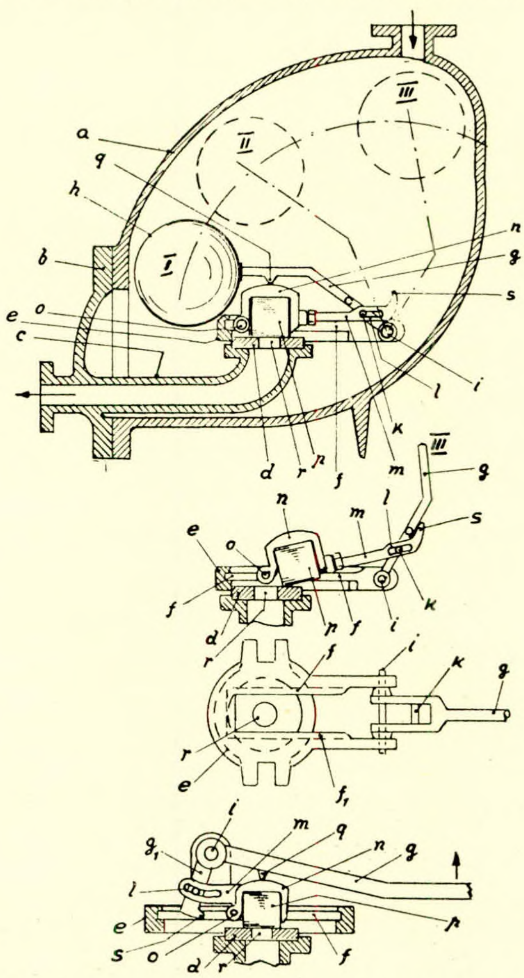


FIG. 7 (*upper*); FIG. 8 (*middle*); FIG. 9 (*lower*)

Marine Engineering and Shipbuilding Abstracts

Volume XVIII, No. 3, March 1955

Plastic-coated Heat-exchanger Coils

It is well known that coils for heating corrosive solutions are frequently subjected to more drastic changes in temperature, thermal stress, and agitation than tanks, pipe lines, and other equipment. Moreover, many of the materials available for heat exchangers designed for corrosive conditions, such as glass, are fragile, while others are extremely expensive, and the remaining materials are restricted in their applications. It is difficult, therefore, to find an economical material for heating coils which can be used in severely corrosive solutions. One solution which is said to have given satisfaction in numerous cases is the application of a coating of fluor-carbon plastic material to common metals. Two of these plastics are reported to be resistant to many corrosives at temperatures up to 400 deg. F. The monochlorotrifluoroethylene polymer, for example, has been used to apply coatings, up to 0·2in. in thickness, which proved free of pinholes and flaws. As the interiors of the tubes are necessarily unprotected, only steam and non-corrosive heat-carrier fluids may be passed through these tubes. Initially coatings of only 0·005in. in thickness were used, presumably in order to keep the loss in heat transfer owing to the coating to a minimum. It was, however, found advantageous to increase the thickness of the coating to 0·2in., so as to lengthen the service life of the tubing under corrosive conditions. Coatings of both thicknesses have been tested for flaws and pinholes by means of a 30,000 volt spark source. Where the unavoidable, but not excessive, loss in heat-transfer efficiency and the limiting temperature of 400 deg. F. are acceptable, plastic insulated tubing may provide a solution, but the brief service life, although longer than that of competitive equipment will, of course, have to be accepted.—*The Engineers' Digest, October 1954; Vol. 15, p. 415.*

Air-blast Atomizer for Viscous Fuels

To meet the difficulties inherent in the atomization of liquid fuels such as Bunker C, which has a viscosity of 6,000 seconds Redwood No. 1 (about 1,500 centistokes) at 100 deg. F., an air-blast atomizer has been developed at the National Gas Turbine Establishment. An essential feature of the atomizer is that fuel is introduced radially inwards into a swirling air stream just before this debouches from the atomizer into the combustion chamber. This paper describes the performance by discussing measurements on two such atomizers with mixing sections such that one was of twice the linear dimensions of the other.—*H. Clare and A. Radcliffe, Journal of the Institute of Fuel, October 1954; Vol. 27, pp. 510-515.*

Structural Design of Tankers

Whereas the structure of the dry cargo carrier has become more or less standardized, that of the bulk oil carrier shows great variety, reflecting the diversity in views of the many people who have given detailed study to the problems posed in the design of a structure suitable to satisfy the various requirements for a ship of this type. This interest is largely due to the desire to eliminate certain basic defects of previous designs or to detailed consideration of certain factors which assume greater importance in the tanker than in the dry cargo carrier. For instance, there is the integrity of bulkheads, which has been very difficult to attain in the past; the problems of adequate drainage occasioned by the use of large capacity cargo pumps; the problem of cleaning tanks; and above all the big factor of corrosion. Then again, some designers have been motivated by the aim to reduce weight as much as possible consistent with strength, though it should be noted that the initiative in this direction rarely comes from the tanker owner. To some extent some of the above factors are incompatible, and as their relative importance is initially hard to assess it is hardly surprising that great variety in structural design is a peculiarity of the tanker. The successful designer will keep all these factors in mind, however, giving to each its due, always remembering that reliable and continuous operation unhampered by delays to rectify minor structural defects is the factor the owner values most; deadweight records are usually of more satisfaction to the shipbuilder than to the shipowner. In what follows an attempt has been made to cover the more important aspects of structural design of oil tankers; to mention those features which experience has shown to be troublesome; to outline some of the fundamental problems which must be met in the production of a satisfactory design; and to express an opinion on the merits and demerits of various constructional features. Generally, unless otherwise noted, welded construction is assumed, riveted construction in tankers being now a matter of history. The basic systems of construction are grouped for convenience under the following three headings:—(*a*) Transverse framing; (*b*) Complete longitudinal framing; (*c*) Combination system of framing where the framing on the bottom and deck is run longitudinally and that on the sides vertically. Transverse framing is nowadays adopted only for very small vessels, the fundamental objections to it being outlined later. For larger vessels, therefore, the choice lies between the completely longitudinally-framed vessel and one constructed on the combination system. These two systems, both of which show considerable variety in detail design, might be said to be representative of the American and Anglo-European schools of design, respectively, as it is within these geographical boundaries that they respectively predominate. One of the disadvantages of transverse framing in an oil tanker is that the bottom and deck lack sufficient rigidity to resist buckling in compression. This effect is not so marked in small tankers (up to 250 feet), largely due to the more conservative stresses adopted, but the effect is noted even on these ships when the effects of corrosion reduce the thickness of the deck and bottom plating somewhat. The effect can be controlled by increasing thicknesses and reducing spacing of frames, but even these measures do not prove entirely satisfactory in welded ships where a certain amount of distortion or preset, of the type of buckling to be expected in compression, is initially present due to contraction of the transverse welds. A small amount of initial distortion can greatly reduce the load carrying capabilities of the envelope in compression. Severe buckling of bottom shell has been experienced on a number of transversely framed dry cargo carriers of all-welded construction. A disconcerting feature is that, once present, the buckling is progressive and eventually reaches serious proportions, when expensive corrective measures require to be taken. The use of longitudinal framing, with the resulting large increase in the radius of gyration of the local section of material concerned, is structurally very much more desirable for deck and bottom stiffening which is required to resist compressive loads of a large order in the course of normal service. The wave form of buckling due to distortion is then in a direction where it will be largely unaffected by longitudinal strains. Transverse framing, except at the ends of the vessel, where bending moments are greatly reduced, should therefore be confined to small vessels in sheltered trades, and longitudinal framing of deck and bottom accepted as standard in the construction of larger vessels built for ocean trade. This is usually referred to as the Isherwood system, in which all plate stiffeners for the envelope and the longitudinal bulkheads run in a longitudinal direction, the stiffeners being broken up into suitable spans by the transverse bulkheads and transverse frames. Originally introduced by Curchin, the combination has become easily the most popular with British shipowners. With longitudinal framing on the bottom and deck, and vertical framing on the sides, longitudinal bulkheads and transverse bulkheads, the distribution of stiffening material is most logical. A few years ago a new combination system of interest was introduced by Sir Joseph Isherwood and Co., Ltd., in which, in place of horizontal stringers in the wing tanks, diagonal bracing on every frame was introduced. The designers claim a reduction in weight of about 7½ per cent over their complete longitudinal system. So far, service experience with a structure of this type is not known. The system would appear to be inferior to the Curchin type in regard to ease of tank cleaning.—*E. E. Bustard, The Shipping World, 25th August 1954; Vol. 131, pp. 216-217.*

Inspection, Explosion and Breakdown of Boilers

The most frequent cause of boiler failure is damage due to loss of water consequent on neglect or misreading of the water gauges. The results are usually serious as, apart from personal injuries, the affected plant may be out of commission for a protracted period for major repairs and often complete replacement is necessitated. External-flue-gas explosions which usually result in grave structural damage to the flue and furnace settings are more common than is usually realized and protection measures against these are discussed in the paper. Many violent explosions have occurred of economizers of the cast-iron tubular type, which are liable to graphitic disintegration and other forms of chemical attack. The greatest danger for such plant is that of steam formation, usually due to neglect in not bypassing the flue gases under conditions of static feed water, with the possibility of resultant explosion from water hammer. Steam and feed pipes are subject to failure for similar reasons as those for pressure-vessel breakdown, but are particularly liable to explosion from water hammer, the causes of which are often not properly understood. Major disasters, for instance, have resulted from the opening of drain taps on steam pipes containing condensate when under pressure. An interesting case of failure due to corrosion fatigue of the cover-retaining bolts of a high-pressure feed-water heater is quoted and an analysis is given of the formation of typical corrosion-fatigue cracks throughout the body of the bolts. The initial cause of this breakdown was variation in temperature between the bolts and the heater cover occasioned by sudden changes in the feed flow which permitted slight leakage to occur through the cover joint. Details of explosions of air receivers are quoted. Such vessels may be the subject of concentrated internal chemical attack, and particular care has to be taken in the examination and testing of such plant. The majority of explosions of air receivers is, however, due to the detonation of internal inflammable vapours caused by high operating temperatures. Such explosions may normally be prevented by the fitting of a correctly designed fusible plug.—*Paper by J. Eyers, read at a General Meeting of The Institution of Mechanical Engineers, 12th November 1954.*

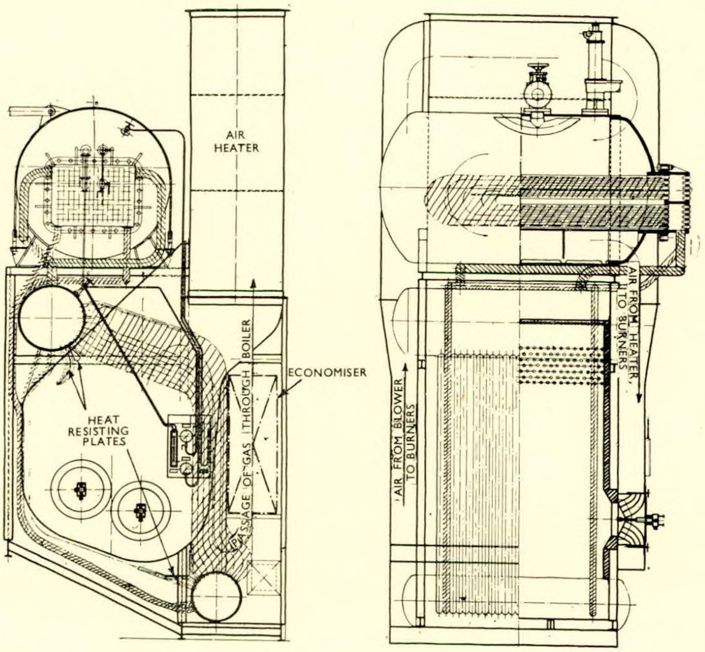
Side Launching of Ships

Over the past few years an investigation into the side launching of ships has been in progress at the Ship Division of the National Physical Laboratory. Ship launches have been recorded by means of gyroscopic equipment placed on board and wired to recording units on shore. Continuous records of the roll, pitch and yaw of many vessels were taken during launching, together with the motion down the ways. A series of model experiments has been conducted with a trawler model, in which the main factors affecting side launching were systematically varied. From this work a method has been developed, using design diagrams, to enable launching characteristics to be predicted in advance, with practical accuracy. Two ciné films showing aspects of the work on model and ships have been prepared. Further experiments are envisaged for other ship types, with a view to extending the range of the design diagrams.—*Paper by D. J. Doust, presented at the Autumn Meetings of the Institution of Naval Architects, 24th September 1954.*

The Boiler Problem in Supertankers

The chief disadvantage of the motor tanker in relation to the steam-driven ship is that fairly large boiler plant is required in the case of the former, mainly utilized, whilst the ship is in port, for the supply of steam to the engines driving the cargo pumps. It therefore represents high cost capital equipment with comparatively little earning capacity except when the tanker is in port. The principle adopted in this connexion is, naturally, to endeavour to reduce the size of this plant as far as possible, and with it the capital cost involved; also to supply all the steam necessary whilst the ship is at sea from an exhaust-gas boiler, so that the large oil-fired boilers are not required, as they would operate with reduced efficiency. A rough value for the hourly cargo pumping capacity is 10 per cent of the ship's deadweight, so that the pumping output required ranges from 2,000 tons to 3,000 tons hourly. In the case of one motor ship, the

Bergeboss—of 32,250 tons d.w. with 12,000 b.h.p. machinery—there are four pumps each of 750 tons hourly capacity and two are driven by motors installed in the engine-room, the driving shafts passing through the bulkhead. The other two are steam-driven so that the demand for steam, and with it the size of the boilers, is limited to half the total cargo pumping requirements. The two 300-kW. Diesel generators supply the current for driving the motor-driven pumps when in port, also for the auxiliaries when at sea and there is a 75-kW. set receiving steam from the exhaust-gas boiler. In the 29,500-ton M.T. *Biblos* and her sister ships, however (with



New design of Burmeister and Wain boiler with special application for tankers

10,000 b.h.p. machinery), there are two cargo pumps, of 1,000 tons hourly capacity, and these are driven by steam turbines also installed in the engine-room and with shafts passing through the engine-room bulkhead. The exhaust-gas boiler supplies steam to a 200-kW. turbo alternator and the propeller shaft drives a 50-kW. alternator. Normally these two alternators meet the power requirements when the ship is at sea and the main boilers are not needed. It will be seen that the exhaust-gas boiler plays an important part in the engine-room arrangement of large Diesel-engined tankers. With the modern waste heat boiler it is possible to produce, from the exhaust gas of a two-stroke turbocharged engine, 0·9lb. of steam per b.h.p. per hr., at 180lb. per sq. in., or about 0·95lb. per b.h.p. per hr. at 100lb. per sq. in. With a 10,000 b.h.p. engine, therefore, there should be a maximum of 9,000lb. to 9,500lb. of steam per hr., and, even allowing a margin, and providing over 6,000lb. of steam for a 220-kW. turbo-alternator there should be sufficient for heating the accommodation and the fuel oil tanks, and for supplying the evaporators. With the increasing requirements of steam in large tankers, Scotch boilers are being replaced by watertube boilers which are smaller, cheaper and lighter, but are not so easy to maintain and are more sensitive. Burmeister and Wain have developed a special boiler for this purpose with a closed primary system in which pure feed water circulates, together with a secondary system heated by steam from the primary system. The primary oil-fired section, as shown in the illustration, comprises two boiler drums and tubes.—*The Motor Ship, December 1954; Vol. 35, pp. 392-393.*

Scale Effect Experiments on Victory Ships and Models. Part I

This paper deals with the resistance- and thrust-measurements on a series of six geometrically similar Victory ship models, ranging in scale from 60 to 18, and on a Victory model boat of scale 6. The first part of the paper gives a description of the ship, the propeller, the models, the test apparatus, the test performance, the instrument calibration and the necessary corrections. The measured resistance and thrust values of the model boat are correlated with the results of the tests with the models using three methods, i.e. the Schoenherr, the Hughes and the Lap-Troost extrapolators. The results of experiments for the determination of the wall-interference are given. These experiments are based on the principle of the "image method". An attempt is made to analyse the thrust deduction factors as a function of the Reynolds number. Conclusions and a programme for the continuation of the tests are presented at the end of the paper. Test measurements of the models and the model boat are tabulated in the Appendix.—*Paper by W. P. A. van Lammeren, J. D. van Manen and A. J. W. Lap, presented at the Autumn Meeting of the Institution of Naval Architects, 22nd September 1954.*

N.E.M. Reheater and Bauer-Wach Combination

The *Baron Ardrossan* is an open shelter decker with raised forecastle. She has been built under Lloyd's Register of Shipping special survey and has the following principal particulars:—

Length overall	422ft. 8in.
Length between perpendiculars	410ft. 0in.
Moulded breadth	57ft. 0in.
Moulded depth to main deck...	28ft. 9in.
Moulded depth to shelter deck	37ft. 9in.
Deadweight capacity on 25ft. 5¼in. draught	9,072 tons
Cargo capacity (bale)	476,500 cu. ft.
Gross tonnage	5,254 tons
Net tonnage	2,649 tons
Service speed	12 knots
Bunker capacity	947 tons
Ballast capacity...	1,670 tons

The hull form has resulted from resistance and propulsion tests carried out at the Teddington tank of the National Physical Laboratory. The *Baron Ardrossan* is the first ship to go to sea with a North Eastern Marine triple-expansion reheater engine working in conjunction with a Bauer-Wach exhaust steam turbine. The North Eastern Marine Engineering Co. (1938), Ltd. and its associate, George Clark (1938), Ltd., have delivered over 150 reheater engines and a number of triple-expansion engines with Bauer-Wach turbines, but until now, the two refinements have not been combined in one installation. The main engine is of the usual N.E.M. design having cylinder diameters of 24 inches, 39 inches and 68 inches with a stroke of 48 inches. Cam-operated poppet valves for steam admission and exhaust are fitted to the h.p. and i.p. cylinders, which are disposed at the forward and after ends of the engine. By arranging the comparatively large, heavy and cooler l.p. cylinder in the centre, improved running balance, symmetrical thermal expansion and accessibility to the h.p. and i.p. poppet valves is obtained. The i.p. cylinder is fitted with an Andrews and Cameron double-opening slide valve. All the valve gear is operated by Stephenson link motion with all-round reversing mechanism. The designed service output of the combination is 2,650 equivalent i.h.p. at 75 r.p.m., of which 1,925 i.h.p. are delivered by the reciprocating element, and 725 equivalent i.h.p. are contributed by the exhaust turbine which runs at about 3,860 r.p.m. An overload rating of 3,050 equivalent i.h.p. is available, when the two components develop 2,165 and 885 i.h.p. respectively. The Bauer-Wach exhaust turbine is of the usual pattern and was supplied by Barclay Curle and Co., Ltd. The l.p. exhaust steam, after leaving the Andrews and Cameron balanced slide valve, passes through an oil and water separator to the after end of the turbine casing where there is an oil-operated servo-motor controlling a balanced flap valve. This is fitted so that when the main engines are reversed, steam is bypassed round the turbine and flows back through the square-section fabricated eduction bend to the condenser. Steam is raised in two N.E.M. Scotch boilers having a diameter of 17 feet, a length of 11 feet 6 inches and a heating surface of 3,325 sq. ft. Each boiler has three 4 feet 3 inches diameter furnaces fitted for oil firing. A Nemenco superheater, designed to raise the steam temperature at the outlets to 750 deg. F. is fitted in each wing furnace. A Howden tubular preheater is fitted in the uptakes to raise the air temperature to 220 deg. F. The auxiliary machinery is arranged to take superheated steam at a temperature of 600 deg. F. A new type of de-superheater has been developed by North Eastern Marine Engineering Co. which is designed to prevent any carry over of excess water. In this model the inlet and outlet branches are arranged in the side of the casing, the steam flowing downwards and past a series of baffles before rising upwards in an annular space to the outlet. The four spray nozzles are arranged vertically and the drains are led through a gauge column and drain-trap to the hotwell. As with all N.E.M. reheater installations, two-stage feed heating is employed. The l.p. heater is used as a condenser for the auxiliary machinery exhaust and the h.p. heater takes bled steam from a tapping after the h.p. cylinder. In view of her technical importance, the *Baron Ardrossan* has been the subject of special loaded trials and arrangements somewhat after the lines of the classic *Sussex Trader* trials were made before she left the Wear. The ship sailed in ballast for San Domingo in Cuba, where she loaded a full cargo of sugar for London. On her return voyage to this country, an N.E.M. trials party joined at Falmouth, and a further series of tests was taken during the voyage up Channel. The trial was carried out under steady steaming conditions and an i.h.p. check with "turbine out" was made before and after the trial which gave 2,100 i.h.p. at 72·54 r.p.m., from which the total equivalent i.h.p. with "turbine in" has been calculated. During the trial the engine was set with inlet valves operating at "line in line" (h.p. cut off 60 per cent). All main steam valves were fully open except the engine stop valve which was adjusted to give the required service power, the amount of throttling at this point being reduced by keeping the boiler pressure at a steady 200lb. per sq. in. pressure (designed pressure 220lb. per sq. in.).

Baron Ardrossan loaded trial figures

Draught forward	25ft. 1in.
Draught aft	25ft. 3in.
Mean draught	25ft. 2in.
Displacement	12,546 tons
Barometer	30·07in. Hg.
Sea	Smooth
Wind	Light breeze
Ship's speed by log	12·8 knots
Propeller revolutions	77·96 r.p.m.
H.p. cylinder	624 i.h.p.
M.p. cylinder	611 i.h.p.
L.p. cylinder	581 i.h.p.
Equivalent turbine output ...	787 i.h.p.
Equivalent total output ...	2,603 i.h.p.
Shaft horse power (by torsiometer)...	2,378 s.h.p.
Mechanical efficiency	91·4 per cent
Equivalent m.i.p. referred to l.p. piston	38·0lb. per sq. in.
Boiler steam pressure	199·5lb. per sq. in.
Engine stop valve pressure ...	187·5lb. per sq. in.
H.p. chest pressure	164·5lb. per sq. in.
M.p. chest pressure	49·8lb. per sq. in.
L.p. chest pressure	9·75lb. per sq. in.
Turbine inlet vacuum	16·68in. Hg.
Condenser vacuum	28·57in. Hg.
Augmentor steam pressure ...	55·3lb. per sq. in.
Superheater outlet temperature:	
(*a*) Calibrated thermometers	733 deg. F.
(*b*) Pyrometers	743 deg. F.
Auxiliary steam temperature ...	593 deg. F.
Engine stop valve	605 deg. F.
H.p. exhaust	461 deg. F.
M.p. chest	594 deg. F.
L.p. chest	414 deg. F.
Turbine inlet	232 deg. F.
Turbine outlet	91 deg. F.
Feed at filter outlet	120 deg. F.
Feed after first-stage heater ...	195 deg. F.
Feed after second-stage heater...	287 deg. F.
Air at furnaces	222 deg. F.
Gases after air heater	409 deg. F.
Fuel oil:	
Specific gravity at 60 deg. F.	0·952
Viscosity, Redwood No. 1 at 100 deg. F.	731 sec.
Gross calorific value... ...	18,405 B.Th.U.
Net calorific value	17,259 B.Th.U.
Oil fuel, all purposes	1,771lb. per hr.
Oil fuel, all purposes	18·98 tons per day
Water consumption, all purposes	22,050lb. per hr.
Water consumption, main engine and bleeding	19,997lb. per hr.
Oil fuel rate, all purposes ...	0·680lb. per i.h.p. per hr.
Water rate, all purposes ...	8·47lb. per i.h.p. per hr.
Water rate, main engine and bleeding	7·68lb. per i.h.p. per hr.
Oil fuel consumption, all purposes	0·745lb. per s.h.p. per hr.
Water consumption for auxiliaries...	2,053lb. per hr.
Bled steam (h.p. exhaust) ...	1,915lb. per hr.
Boiler efficiency...	82·1 per cent

—*The Marine Engineer and Naval Architect (Annual Steam Number), 1954; Vol. 77, pp. 413-420.*

Marine Plywoods

Plywood is extensively employed in many types of ships and some data given in the Journal of the American Bureau of Ships should therefore be of interest. Most of the plywood used by the U.S. Navy is produced from Douglas fir and manufactured in two types according to whether it is used for external or internal purposes, panels of both types being graded according to the appearance of their face veneers. The exterior type is manufactured with waterproof adhesives so that it can be safely exposed to open weather conditions or used in atmospheres having a high moisture content. The interior type, although manufactured with moisture-resistant glue, is of such quality that it should only be used in enclosed spaces. No matter in what conditions that are employed, the edges of plywood are susceptible to deterioration due to infection by various decay fungi and in order to prevent rapid deterioration, all plywood for naval use is treated with suitable chemical preservatives. There are several chemical compounds which, it is claimed, provide effective protection, among which are acid cupric chromate, chromated zinc chloride, ammoniacal copper arsenate and Wolman salts. Attention is drawn to the fact that since all these preservatives are water-soluble chemical salts, the dryness of the plywood before the treatment is applied is of considerable importance.—*Shipbuilding and Shipping Record, 21st October 1954; Vol. 84, pp. 527-528.*

Replacement of Liberty Ships

A large number of cargo shipowners in all countries must consider their future building policy, not merely on the basis of the replacement of tonnage in the normal way, but of the replacement of Liberty ships, of which there are still over 600 in service in countries other than the United States. It is understood that some of the owners placing orders for motor cargo ships during the past month or two (particularly Norwegian) have had this problem in mind. It is felt that the inefficiency of the Liberty vessels, both from their loading characteristics, their low speed and very high fuel consumption, may render it necessary to take them out of service before the end of their natural period of life, especially on account of the higher speed which the modern tonnage will supply and which will attract the charterer at the expense of Liberty ships. The question concerns Norway, which has about 50 such vessels, but it is more important in Great Britain with over 100 Libertys, also in France with between 70 and 80 and in Italy with about 90 Liberty cargo ships. The daily fuel consumption of the Liberty ship carrying a little over 10,000 tons and maintaining a speed of between 10 and 11 knots at sea is in the neighbourhood of 25 tons. For the same amount of fuel, a motor ship with equal carrying capacity will have a service speed of $14\frac{1}{2}$ knots in average weather or 4 knots greater than the Liberty ship. Alternatively, if a speed of 12 knots is decided upon, the ship will consume about 12 tons daily, or rather less than half of that of the Liberty ship with an increased speed of at least one knot, and probably a better cubic carrying capacity.—*The Motor Ship, December 1954; Vol. 35, p. 358.*

Nuclear Ship Propulsion

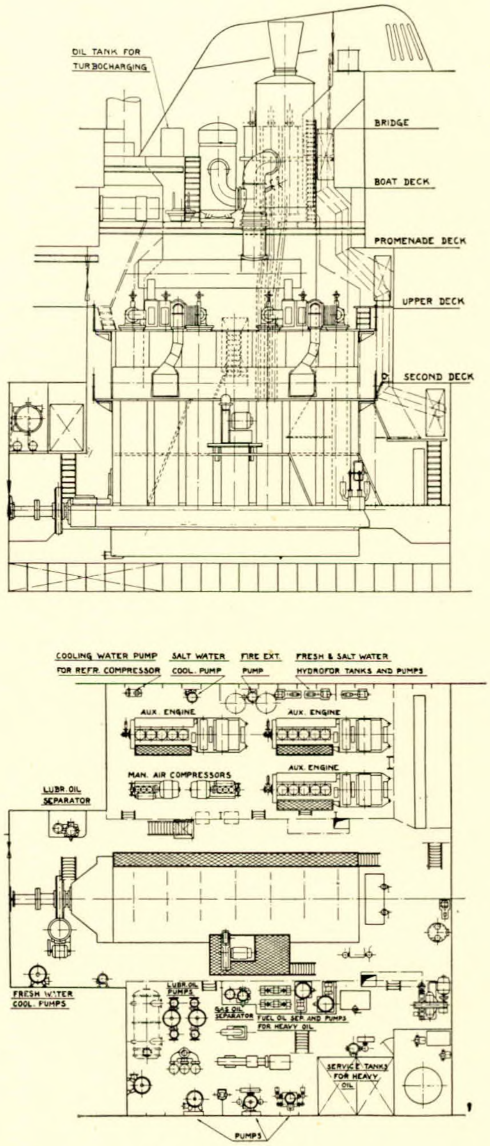
A proposal, by the shipbuilding division of the Bethlehem Steel Company, of New York, to study the application of nuclear power to commercial ship propulsion has been approved by the Atomic Energy Commission in Washington, D.C. This study is the second in the field of ship propulsion under the Commission's industrial participation programme, and it brings to seventeen the number of independent studies on the development of industrial nuclear power and by-products. The study will be undertaken by the Bethlehem Steel Company's central technical department at Quincy, Massachusetts, who were associated with the development of machinery for the submarines *Nautilus* and *Sea Wolf*. Preliminary studies will be made of reactor types, steam cycles, control methods, machinery arrangements and weight. These studies will be followed by detailed designs of reactions and other major propulsion equipment. It is also proposed to explore the development of a package reactor plant for small land-based electric-power stations, or for distilling plants to provide fresh-water in remote regions. The Bethlehem Steel Company will bear the cost of the study, other than that incurred by the Commission and their contractors, in making available the information and consultation services necessary to the study. The contract will run for one year, after which the company will report their findings and recommendations for further study or development work.—*The Shipbuilder and Marine Engine-Builder, December 1954; Vol. 61, p. 715.*

Danish 17-knot Cargo Vessel

The Danish shipowning company Ove Skou, Ltd. are building up a fleet of fast, moderate-sized cargo liners, and their recently-completed vessel *Susanne Skou* is notable for the installation of a turbo-charged two-stroke engine which gave her a speed of 17 knots on loaded trials. Two further ships of about the same size are under construction, one at Burmeister and Wain's yard, where the *Susanne Skou* was constructed, and the other by the Elsinore S.B. and E. Co., Ltd. Particular interest in the *Susanne Skou* centres around the relatively small engine room, rendered possible by the adoption of turbo-charging, whilst allowing the installation of sufficient power to give the high speed mentioned. The engine is designed to develop 7.500 b.h.p. and the length of the engine room as shown in the illustration is about 41 feet, apart from the thrust recess; moreover, at the forward end a portion of the engine room reaches only to the second deck. The propelling engine is designed to operate on heavy oil carried in double-bottom tanks, the daily service tank being installed at the starboard side of the machinery space. The following are the main particulars of the new vessel: —

Length b.p.	121·92 m. (400ft. 0in.)
Breadth moulded ...	17·22 m. (56ft. 6in.)
Depth to upper deck ...	10·82 m. (35ft. 6in.)
Depth to second deck...	8·23 m. (27ft. 0in.)
Draught to summer load line	7·52 m. (24ft. $7\frac{7}{8}$in.)
Corresponding dead-weight capacity ...	7,050 tons of 1,016 kg.
Total cargo capacity, including deep tanks	434,000 cu. ft. (bale)
Speed on loaded trial trip	17 knots

Three holds are arranged forward and two aft of the engine room and in the after part of hold No. 3 are two deep tanks for the carriage of vegetable oil. Fourteen 5-ton derricks serve



Engine room plans

the five hatches and in addition there are two 25-ton derricks at hatches No. 2 and No. 4 respectively. The 5-ton electric winches, also the two 3-ton capstans, the electrically-operated windlass and the steering gear are all of Thomas B. Thrige's manufacture. There are two 29-ft. lifeboats, one being equipped with a Diesel engine, in addition to which there is an 18-ft. motor boat and a 16-ft. 3-in. dinghy carried aft. The propelling engine is a standard B. and W. single-acting turbo-charged two-stroke six-cylinder unit designed to develop 7,500 b.h.p. or 8,300 i.h.p. at 115 r.p.m. The cylinders are 740 mm. in diameter with a piston stroke of 1,600 mm. Current is supplied throughout the ship from three 5-cylinder B. and W. four-stroke engines coupled to 200-kW dynamos and running at a speed of 500 r.p.m. The cylinders have a bore and stroke of 245 mm. and 400 mm. respectively. Most of the engine room pumps are electrically driven and of the vertical type with flexible couplings between the motor and the pump. For the circulation of cooling and lubricating oil there are two 200-ton pumps, whilst three 240-ton centrifugal pumps provide the circulation of fresh and salt water for the main engine. For the auxiliary engines there are two 30-ton pumps. The two electrically-driven compressors have a capacity of 200 cu. m. of free air per min. compressed to a pressure of 25 atmospheres. In addition to the oil-fired boiler with a heating surface of 23 sq. m. there is an exhaust-gas boiler, the heating surface of which is 100 sq. m.—*The Motor Ship, December 1954; Vol. 35, p. 377.*

Twin Screw Passenger Car Ferry

Built by Ailsa Shipbuilding Co., Ltd., of Troon, for the Bland Line, Ltd., Gibraltar (Watts, Watts and Co., Ltd., London agents), the twin-screw motorship *Mons Calpé* has been designed to provide a passenger, mail, car-ferry and cargo service between Gibraltar and Tangier, a route which is exceedingly popular with travellers visiting the North African International Seaport. The principal dimensions and other leading characteristics of the *Mons Calpé* are given in the accompanying table.

Length overall	283ft. 0in.
Length on load water-line ...	268ft. 0in.
Breadth moulded at car deck ...	48ft. 0in.
Breadth moulded at load water-line	46ft. 0in.
Depth moulded to shelter deck	26ft. 0in.
Depth moulded to car deck ...	17ft. 0in.
Load draught	10ft. 6in.
Gross tonnage	2,000
Number of passengers	600
Number of motorcars	70
B.H.P.	2,620
Speed, knots	16

The *Mons Calpé* is propelled by twin screws, driven by two British Polar engines (type M47M) capable of developing a total of 2,620 b.h.p. Air for starting is stored in two cylindrical air vessels, which are located at the port side of the engine room. For charging these pressure vessels, there is a motor-driven compressor (type C5A6), with a capacity of 50 cu. ft. of free air per minute at a pressure of 350lb. per sq. in., manufactured by Reavell and Co., Ltd., Ipswich, who have also provided the emergency Diesel-driven compressor. This unit, which has a capacity of 45 cu. ft. of free air per minute, at a pressure of 350lb. per sq. in., is driven through a friction clutch by a hand-starting Diesel engine.—*The Shipbuilder and Marine Engine-Builder, December 1954; Vol. 61, pp. 707-710.*

Economics of the Trailer Ship

Research in the field of ocean transportation has been concerned for many years with the problem of reducing cargo handling time and expense and, as a consequence, increasing productive time at sea for ocean cargo carriers. Much of the coastwise and intercoastal shipping operations in the United States have wasted away, in face of rising costs and increasing competition servicewise from land carriers, for lack of an adequate answer to this problem. The trailer ship idea is not a new one. Its development into a finished operating plan by the McLean Trucking Company came as the result of its growing awareness of a public need for the re-establishment of a low-cost medium of transportation for shippers who can accept a somewhat deferred service but for whom ordinary break-bulk water service is not satisfactory. The McLean Sea-Land trailer ship service, then, is a plan devised for the more economic and efficient movement of an established volume of traffic, rather than one devised purely to mend a sick industry. One of its results, however, is the development of at least a partial answer to the insistent cargo-handling problem of the water carrier. This is accomplished through adaptations of two familiar cargo-handling devices—cargo containers and a conveyor loading and unloading system. The containers are ordinary truck-trailers and are, therefore, already in general use. They may be loaded at the point of production or first handling, and need not be unloaded until arrival at the point of use or final distribution. The conveyor system is simply an inclined plane between the ship and shore. The cargo will roll up and down this gangplank on wheels which are already integral parts of the containers. Transportation of trailers by water is already in operation on the Hudson River, on the Gulf Intracoastal Canal and between Seattle and Alaska ports, in addition to short-run ferry services. The McLean Sea-Land trailer ship plan is an innovation chiefly because it proposes the use of large ocean-going ships over lengthy runs. Each of the ships will have the following dimensions and characteristics: length 638ft.; beam overall 87ft.; cruising speed 19 knots; gross tonnage approximately 10,000; displacement tonnage loaded approximately 17,000; indicated horsepower of 14,000 from geared steam turbines, driving twin screws; maximum draught loaded 20ft.; and a crew of forty-two or forty-three men. The ship will have four cargo decks with a total capacity of 286 trailers of 35 feet length, or a proportionately larger number of shorter trailers. The two principal trailer decks will have a capacity of 208 trailers. An additional 78 trailers will be accommodated in the hold and on the boat deck. Ramps or elevators will be used to move trailers from the main deck to the hold and from the first deck to the boat deck. Stern ports at the level of both the main deck and the first deck will provide for loading and unloading directly to and from each such deck. The trailers will range in length from 16 to 35 feet and in height (for vans) from 12·5 to 14 feet. Their width is 8 feet. Provision will be made for handling special trailers, including flat-body trailers, open-top trailers, drop-frame (household goods and furniture type) trailers, vehicle carriers and refrigerated trailers. The average empty weight of a trailer is 5 tons; at full capacity, 1,430 tons of trailer weight will be carried. Based on known density factors, the average net weight of cargo expected to be carried has been computed at 15·38 tons per trailer. On this basis, total payload will equal 4,399 tons at maximum utilization.—*D. G. Macdonald, The Shipping World, 17th November 1954; Vol. 131, pp. 543-544.*

Tanker of Unusual Design

A motor tanker of unusual design has recently been completed by A/B Lindholmens Varv, Gothenburg, for the Rederi A/B Nordstjernan (the Johnson Line). This vessel, the *Oceanus,* is of 24,500 tons deadweight. She is notable partly as being the largest ship yet built by Lindholmen, but more particularly on account of her having the bridge house combined with the after superstructure at the stern of the ship. The advantages of this design were discussed in *The Shipping World* of 9th June at the time of the vessel's launch. It is, of course, the layout commonly—though not universally—adopted for ore carriers, and it would appear to have definite advantages for oil tankers also. The removal of the weight of the bridge structure from the centre of the ship has the effect of reducing the bending moments at the place where they are in any case at a maximum, while the transfer of their weight to the stern of the ship should make it easier to achieve satisfactory trim in the loaded condition. From the shipbuilder's point of view the construction is likely to be somewhat simpler, but a more important gain results directly from the grouping of all the accommodation in a single structure. This will give a saving in the weight of steel required, and should also give notable savings in the lengths of cabling and piping runs. The obvious disadvantage, of course, is the effect on the navigation of the ship of removing the bridge to a point very much further aft. In fact, however, this will not have the effect of altering the angle of sight over the bow, as the greater distance between bow and bridge is compensated by the greater height of the bridge when it is on top of the poop structure. Another possible disadvantage concerns the centre of wind resistance of the ship, which may be inconveniently far aft. But the fact that a growing number of ore carriers are in service with the bridge right aft shows that these difficulties are not of too great importance. Apart from the points mentioned, the design of the *Oceanus* is fairly typical of modern all-welded practice. The principal particulars of the ship are as follows:—

Length b.p.	575ft. 0in.
Breadth moulded...	76ft. 0in.
Depth moulded	43ft. 3in.
Mean draught	32ft. 8in.
Cargo capacity	1,180,000 cu. ft.
Horsepower	8,150 b.h.p.
Service speed	14·5 knots

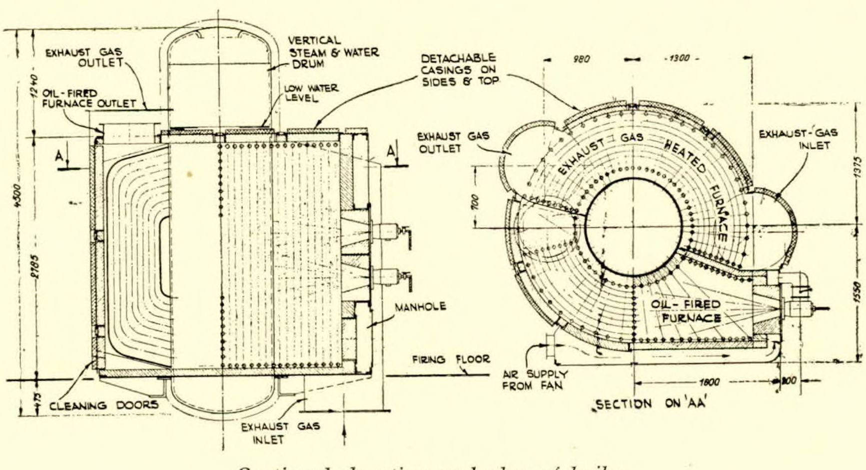
The propelling machinery consists of a single Lindholmen-GV Diesel engine of single-acting two-stroke type, with cylinder bore of 760 mm. and stroke of 1,500 mm. It develops 8,150 b.h.p. at 112 r.p.m. to give the service speed of 14·5 knots. On loaded trials a speed of 15·1 knots was attained.—*The Shipping World, 17th November 1954; Vol. 131, pp. 549-550.*

New Class of Lifeboat

Replacing the 41ft. T. S. Watson type non-cabin lifeboat, which first saw service twenty-one years ago, is a new 42ft. cabin class, the prototype of which, built by Wm. Osborne, Ltd., Littlehampton, has recently gone into service at Coverack, Cornwall. In several directions this new boat shows radical departures from normal R.N.L.I. practice and reflects the new policy of using standard, rather than specially manufactured equipment, wherever possible. Requiring a larger reserve of power than has previously been regarded as necessary, the Institution decided to install standard marine engines, a practice unknown since the first powered boats, after which the R.N.L.I. designed and manufactured their own machinery. The engines chosen were Gardner LW4 type, lightweight, alloy construction Diesel units each developing 48 b.h.p. at 1,200 r.p.m. and giving the lifeboat a top speed of 8·38 knots. The engines are rendered "splashproof" in standard Gardner manner, the only non-commercial features are the carrying up of the air intakes to the top of the engine room by way of flexible pipes and the fitting of a second oil pump to ensure safe operation of the engine at fore and aft inclination angles of 12½ degrees. The use of hydraulically operated gears in conjunction with Bloctube throttle controls is another innovation as is the use of standard instrument panels in place of the sealed units previously employed. The engines are fresh water cooled in common with all R.N.L.I. boats—it is understood that Serck heat-exchangers will be fitted in future rather than specially built units—and the exhausts discharge through a common mast pipe. The standard R.N.L.I. capstan is driven from the fore end of the starboard main engine and operation is by way of a foot-operated hydraulic control pedal located on the deck alongside the capstan. The hull is of normal lifeboat form and construction; an innovation is the use of a double bottom in the engine room on which the vessel could float should the shell in this space be pierced. This inner bottom, which extends under the engines, is of light alloy construction and, in conjunction with watertight fore and aft wing bulkheads, virtually makes the engine room a watertight box within the hull. A range of 238 miles at a top speed of 8·38 knots is a great improvement over the 125 miles at 7·78 knots endurance of the replaced petrol-engined class, the fuel capacity in either case being 112 gallons.—*Shipbuilding and Shipping Record, 21st October 1954; Vol. 84, pp. 533-534.*

Novel German Exhaust Boiler

A new type of composite boiler for exhaust and/or oil firing has been installed in several recently-built motorships in German yards. A single vertical drum is surrounded by a cylindrical casing. D-shaped tubes radiating from the drum form the heating surface. In the case of the exhaust gas furnace, the tubes extend throughout the space, whereas for the oil-fired furnace there is an open space immediately in front of the burner openings and another beneath the uptake, as can be seen from the drawing. The two furnaces are quite separate, and ready access to the tubes for cleaning is obtained through detachable casings. The unit installed in the *Lichtenfels* took the exhaust from the two 2,800 b.h.p. Diesel engines and had two oil-fired burners. Its performance was as follows: When supplied with 6,160lb. of exhaust gas at 662 deg. F. per hr., the output was 2,200lb. of steam at 85lb. per sq. in. per hr. This is achieved with a heating surface of 990 sq. ft. The 538 sq. ft. of oil-fired heating surface can produce 3,300lb. steam per hr. The boiler weighs 14½ tons empty, and contains 3½ tons of water at working level.—*The Marine Engineer and Naval Architect, Annual Steam Number, 1954; Vol. 77, p. 428.*



Sectional elevation and plan of boiler

Interim Report on U.S.S. *Timmerman*

During the past twenty years, the Navy's Bureau of Ships has from time to time undertaken to procure and evaluate ships having experimental machinery installations. These projects have been a part of a continuing research and development programme directed toward improved fuel economy and reduced machinery weights. Machinery plants installed in experimental ships utilize improved or advanced design components, improved systems or cycles, or a combination thereof. The most recent and far reaching of these undertakings was the design and development of the U.S.S. *Timmerman* (EDD828). In this instance, several hull innovations were incorporated in a ship having a highly experimental propulsion plant. The contract set forth the following general characteristics of the machinery plant: Full power, 100,000 s.h.p.; Number of shafts, 2; Ship's speed, not less than 40 knots; Shaft revolutions per minute, about 350; Steam drum pressure, not less than 750lb. per sq. in. gauge; Steam temperature at outlets of superheaters, not less than 1,025 deg. F.; Steam pressure at superheater outlets, not less than 700lb. per sq. in. gauge; Steam pressure at main turbine control valve, not less than 650lb. per sq. in. gauge; Steam temperature at main turbine control valve, not less than 1,000 deg. F.; Shaft horsepower astern, 20,000 (later modified to 8,000); Ship's speed for best economy operating split plant, 20 knots. The starboard machinery installation utilizes two single-furnace, 875lb. per sq. in. 1,050 deg. F. boilers equipped with fully automatic combustion controls. These boilers supply steam to a main propulsion unit consisting of cruising, high pressure and low pressure turbines with a full power rating of 50,000 s.h.p. These turbines are connected through a single reduction gear to the high-speed line shafting through a planetary second reduction gear and low-speed line shafting to the propeller shaft. This array is one of the most interesting features in the ship. The high-speed shafting between the first and second reduction gears is 7¼ inches in diameter, turns at 1,800 r.p.m. at full power, and is made up of four sections totalling approximately 72 feet. It is supported by seven trunnion roller bearings. The planetary gear is the first of its size to be used in Naval service. The outboard shafting is supported by sealed anti-friction bearings, lubricated through the struts. This machinery combination consists of two single-furnace, 2,000lb. per sq. in., 1,050 deg. F. controlled circulation boilers equipped with fully automatic combustion controls. These boilers furnish steam to a main propulsion unit consisting of cruising, high pressure and low pressure turbines rated at 50,000 s.h.p. at full power. The reduction gear for this plant is a double reduction unit. The shafting arrangement for the port plant is a conventional design. The ship is equipped with a new type steering gear which utilizes a ball bearing nut and screw with hydraulic drive, in lieu of the conventional hydraulic ram system. This unit weighs approximately one-fourth as much as a standard destroyer steering gear. There is also provided an automatic steering system. The twin rudders are mounted in roller bearings and fitted with seals. Provision was made for the installation of conventional sleeve bearings in the event of the roller bearings failing. One of the rudders has medium steel plating, while the other has plating of special treatment steel, thus enabling a comparative evaluation of the effects of water flow, erosion, and corrosion.—*D. G. Phillips, S.N.A.M.E. Bulletin, October 1954; Vol. 9, pp. 19-26.*

New Process of Stud Welding

The fixing of studs, carried out in the past mainly by drilling and tapping, has for some years been accomplished in many countries by using the much quicker method of drawing a welding arc between the stud and the work piece on to which the stud is to be fixed. When the high temperature of the arc has made the surface of the stud and the work piece sufficiently fluid, the stud is brought into contact with the plate (e.g. by means of spring pressure) and the arc is interrupted at the same time. The solidification of the metal creates a strong weld. The normal method of carrying out this operation is to hold the stud in a special gun or holder, press it on to the plate, and switch on the welding current. Immediately, the stud is lifted a short distance from the plate, and an arc is thus created. A separate regulating device connected to the gun switches off the current after a predetermined time and allows the stud to contact the plate again. Unless special measures are taken, the metallurgical characteristics of the weld are those of a weld made with an uncoated electrode. These measures consist in selecting special types of steel for the stud material, together with a de-oxidizing medium applied to the tip of the stud if necessary. In general, therefore, specially manufactured studs are used. Shortly before the end of the second World War, the Philips Laboratories developed the "Contact" arc-welding method in which thickly coated welding electrodes are used. These are made semi-conducting by incorporating part of the metal in the coating; this makes ignition and re-ignition easy and, moreover, makes touch-welding possible. The possibility of applying the same principle to other aspects of welding was then studied, and it was found that a simple stud-welding method could be realized. In the new method a special ignition body is placed between the work piece and the stud. During welding that part of the ignition body on which the stud rests is fused away and the released stud is driven into the weld pool. The ignition body comprises a cartridge which is a semi-conductor; it starts the arc between stud and plate. The composition of the cartridge resembles that of a normal electrode coating. A simple stud-welding gun containing only a spring pressing the stud-holder on to the work piece is used to carry out the process.—*W. P. Van Den Blink, E. H. Ettema, and P. C. Van Der Willigen, British Welding Journal, October 1954; Vol. 1, pp. 447-454.*

Unusual Accident

A first assistant was down in a cargo tank setting up on a 12-in. cargo pipe line expansion joint. The ship was in the yard and gas-free. The man was balancing on the pipe line, handing on with one hand and working with the other. He was holding a flashlight in his mouth. The wrench slipped and the man fell, jamming the flashlight back into his throat. He broke a tooth and sustained injuries to the roof of his mouth. His esophagus was cut. The ship reports: "The basic cause of this accident was an improper method of doing work. The man should have established himself more securely on the pipe line and made provision for better lighting. At the very least, he should have had someone to hold the flashlight".—*Marine Engineering, September 1954; Vol. 59, p. 47.*

Oil Drain Practices for Diesel Engines

The determination and establishment of appropriate oil-change periods for Diesel engines is of serious concern to operators, owners, and engine builders. Oil kept too long in service results in dirty engines, often accompanied by excessive wear or damage to parts due to impaired lubrication. Unnecessarily frequent oil changes, on the other hand, are wasteful and costly. Methods and techniques for the examination of used-oil samples are described and the significance of various oil-inspection tests is discussed. Consideration is given to the many factors influencing engine circulation-oil condition. Examples of used-oil analysis and interpretations are presented. These illustrate typical situations covering engine sizes, types, and service applications from automotive Diesels to large slow-turning marine and electric-power-plant units of several thousand horsepower. In normal operation of an engine the following conditions usually prevail: Less than perfect removal of abrasive material from the air entering the engine, less than perfect combustion leading to fuel-decomposition products that contaminate the oil, oil filtration not complete, operation of the engine under varying conditions of load and climates. These conditions result principally from: Fuel injectors out of balance or with poor spray pattern; poor cooling control, such as inoperative thermostats, dirty oil coolers, rust and scale in water jackets; poor fuels characterized by high end point, high sulphur, high-carbon residue or dirt; lack of filter and air-cleaner maintenance; infrequent oil-filter replacement; insufficient crankcase ventilation; and coolant leakage into the crankcase. These, in turn, produce certain changes in the lubricating oil, the degree of which guides the establishment of the drain period. Symptoms of these changes are the presence of organic insolubles from fuel and air, fuel dilution, and moisture. Nothing can substitute for careful analysis of oil and engine condition at regular intervals as the most reliable guide to establishment of proper oil-change interval. The best oil cannot overcome the detrimental effects of poor maintenance, poor operation, and poor fuel. Careful control of these factors reduces the load on the oil and gives the operator the opportunity of longer drain periods consistent with longer engine life. *—L. F. Moody and J. C. Gibb, 1954 A.S.M.E. International Meeting; Paper No. 54-Mex-3.*

Thermal Insulation Flame Test

Fire tests carried out with 85 per cent Magnesia thermal insulation demonstrate that the material will withstand a 2,000 deg. F. flame for an extended period without appreciable damage. The insulation, applied to section of steel pipe by standard methods, showed no damage beyond slight surface cracks after two hours of the intensely hot flame. Temperature on the surface of the insulated test pipe was below 600 deg. F. The results indicate that 85 per cent Magnesia would remain intact and would protect pipe lines, flues, ducts and other equipment from fire damage. During a plant fire it would also prevent sudden temperature rise of the contents of the pipes and vessels. The tests showed further that this insulation resists fire damage equally well when covered with canvas, galvanized steel, aluminium, or various asphaltic jackets. However, the type of jacket did not have any marked effect on the amount of heat transmitted through the insulation to the pipe surface, although somewhat lower temperatures were recorded for the canvas-covered samples. The tests were carried out with empty 6-ft. lengths of 3-in. standard steel pipe covered with 1 inch of insulation. Chromel-alumel thermocouples cemented at the top and bottom of the pipe under the insulation showed the amount of temperature increase at the pipe surface. The pipe was supported horizontally in a test stand about 10 inches above the orifice of a burner firing a mixture of compressed air and manufactured gas. An 8-gauge chromel-alumel thermocouple gave temperature readings for the flame jet at a point 1 inch below the insulation. The 2,000 deg. F. temperature was maintained within a range of ± 20 deg. F.—*Marine Engineering, September 1954; Vol. 59, pp. 59-60.*

German Fuel Blending System

In employing boiler oil in their Diesel engines for ships, some shipowners specify that the fuel should not exceed a given viscosity, very often 1,500 seconds Redwood No. 1 being stipulated. At bunker stations such fuel is not always available and the desired viscosity may be obtained by blending two fuels of different viscosity, the mixing being carried out merely by delivering the two fuels direct to the bunkering tanks. The degree of mixing is obviously not perfect and for this reason owners will not always use blended fuels in their ships. A system has recently been developed by which it is claimed that homogeneous and permanent blending is effected and that there is no stratification. Moreover, variation of the ratio of mixing within a wide range is possible and this ratio, once decided upon, is exact and can be steadily maintained throughout the bunkering period. The delivery quantity can be adjusted to accommodate itself to the demand from the ship. It is reported also that a portable design is being developed and that this equipment, when available, may be installed in the ship, so that at any bunkering station, whether it has mixing facilities or not, the fuel can be blended in any required degree. The blending equipment, which has been developed in collaboration with Siemens and Halske, is automatic and, it is claimed, represents a saving in manpower at the bunkering station. If necessary the variation in mixing may be as wide as 90 per cent Diesel oil and 10 per cent boiler oil, or 10 per cent Diesel oil and 90 per cent boiler oil. It is emphasized that the mixing is carried out completely in the apparatus so that the fuel can be delivered direct to the bunker tanks in the ship, no storage tank being necessary. The rate of delivery can be varied as required. There are two general systems. The first is an equipment by which the measurement and control apparatus is arranged in the suction piping leading to a single mixing pump, which also acts as delivery pump. This is illustrated in Fig. 1. The second arrangement is that in which the quantity measurement and control apparatus is fitted in the pressure piping. This requires two pumps and one mixing pipe or one mixing pump. The system may also be varied by the adoption either (1) of quantity blending, with pneumatically operated control valves, or (2) of volumetric control, also under valve regulation but with a so-called differential regulator. Whilst

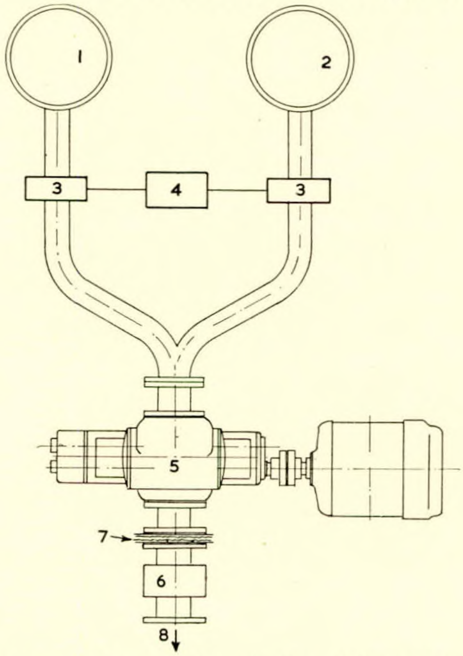


FIG. 1—*Blending installation with a single mixing pump*

1.—Diesel oil tank. 2.—Bunker oil tank. 3.—Pneumatic quantity controls. 4.—Relay. 5.—Special mixing pump. 6.—Quantity measuring gauge. 7.—Mixing pipe. 8.—Discharge.

the quantity mixing arrangement is only suitable for fuels with a maximum viscosity of 50 deg. Engler (1,500 seconds Redwood No. 1) the alternative system is independent of the viscosity, but on the other hand can only be loaded at a pressure of about 60lb. per sq. in. With both systems it is possible to operate efficiently, down to outputs of 20 per cent of the normal discharge. It is further to be noticed that the second, or ring piston system—that is the volumetric control—is easier to operate, owing to its independence of viscosity and it can be run down to outputs of only 10 per cent of the rated production. The question whether the mixing pipe or mixing pump system should be adopted is entirely dependent on cost, the latter being more expensive than the former. It provides, however, a more intensive and homogeneous mixing, particularly when the quantity must be variable. The reason is that the intensity of mixing in the mixing pipe is dependent on the velocity of the oil, but when the pump is employed at lower discharges, say 20 per cent of the normal output, the remaining oil continues to circulate and is even more thoroughly mixed.—*The Motor Ship, December 1954; Vol. 35, pp. 386-387.*

Experiments with a Low Drag Hydrofoil

If laminar flow can be persuaded to persist over parts of a ship's wetted surface and appendages without an adverse separation of boundary layer occurring, a reduction in resistance will result. Similarly, propeller performance can be improved if a part of the boundary layer over the blades can be made laminar. That the persistence of an extensive laminar layer is difficult to achieve in practice, may be seen from this paper, even though the case treated is a simple one. In this connexion the hydrofoil used could be regarded as a full-scale A bracket strut tested at full-scale Reynolds' numbers. The author found that it is possible to achieve an extensive, controlled, laminar boundary layer and the low drag associated with it, in the water of a towing tank. In these experiments any residual turbulence produced by the passage of the model through the water has no apparent effect on subsequent runs. Deliberate attempts were made to trip the flow by running down the tank in the reverse direction at a high speed and then immediately running forward at high speed and observing the streamflow. No difference in the streamflow could be observed from that obtained with a high-speed run taken in the morning after the tank water had become still overnight. Care must be taken, however, that wave-making does not influence either the drag of the section considered or the readings of the Pitot comb measuring the drag. Any estimate of the influence of the orbital motion in a wave on the wake velocity measured by a Pitot comb is very difficult. All the drag readings taken by the Pitot comb when the hydrofoil was vertical show the influence of wave-making, even though it is small at the deepest comb position. Moreover, the influence appears to be most irregular, depending, as it does, not only on the comb depth but also on its distance from the wave-making body in relation to the speed.—*Paper by R. L. Townsin, presented at the Autumn Meeting of the Institution of Naval Architects, 24th September 1954.*

Preventing Fires in Boiler Air Heaters

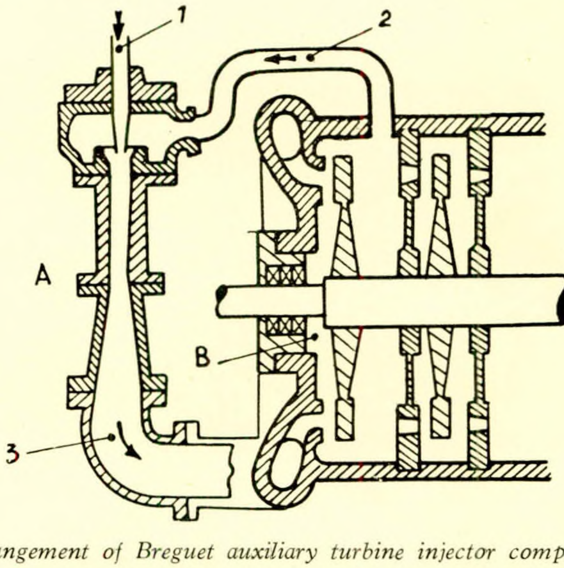
Air-heater fires occur when soot or other combustible deposits accumulated in the gas passages of the air heater become ignited through some cause. This article summarizes the precautionary measures to be taken to avoid such fires, and indicates the action to be taken when they occur. The precautions include the correct positioning of the bypass damper, regularly checking the uptake gas temperature, frequent use of soot blowers, and periodic examination of all external heating surfaces and gas passages to see if deposits tend to accumulate at any particular place. In case of fire, the following action should be taken immediately:—Shut off the oil supply to all oil burners and remove the atomizers; close the air registers on all oil burners; secure the forced- or induced-draught fans of the affected unit and close the inlet-air dampers; raise the water in the steam drum so that the level is just in sight at the top of the gauge glass; locate the fire by inspecting the upper rows of air-heater tubes and then, by means of the access door in the boiler uptake, flood the uptake area over the air heater with CO_2 or other suitable fire extinguisher; do not use the soot blowers during the fire. Some advice is also given on the carrying out of repairs after an air-heater fire.—*Bureau of Ships Journal, No. 4, 1954; Vol. 3, p. 34. Journal, The British Shipbuilding Research Association, October 1954; Vol. 9, Abstract No. 9,535.*

B.S.R.A. Resistance Experiments on the *Lucy Ashton*

This is the third paper dealing with the resistance experiments carried out on the *Lucy Ashton* by the British Shipbuilding Research Association. It is concerned with the ship-model correlation for two conditions involving different types of twin-screw shaft appendages, namely, bossings and shaft brackets. The resistance increments due to fitting these appendages have been determined and the full-scale values have been correlated with results of similar tests carried out in an experimental tank on the same series of six geometrically similar models used in the correlation of the naked-hull results. The lengths of these models ranged from 9 to 30 feet. Ship-model correlation has also been made in terms of the total measured resistance. The findings have been reviewed in the light of previous work and in relation to current practices in the assessment of scale effect on appendage resistance. This appears to be the first occasion on which the parasitic resistance due to shaft appendages has been measured on a full-size ship. A detailed summary together with the conclusions is given at the end of the paper. Full details of both ship and model results are given in appendices.—*Paper by H. Lackenby, read at the Autumn Meetings of the Institution of Naval Architects, 21st September 1954.*

Ejector-compressor for Auxiliary Turbines

The recently-completed French tanker *Aramis* has Breguet generators which supply current for all auxiliary services, including pumping. Each set consists of a 670 h.p., 8,174 r.p.m. multi-stage turbine taking steam at 580lb. per sq. in. and 790 deg. F., coupled to a 375 kW, 230-volt, 1,200 r.p.m. D.C. generator. The generators have over-compounded fields and are capable of an output of 450 kW for one hour. A special requirement was that the sets should be able to develop the normal power with steam conditions of 400lb. per sq. in. and 790 deg. F. and the overload power at 465lb. per sq. in. and the same temperature. The steam consumption at normal load of 375 kW was specified as 4,525lb. per hr. or 12·07lb. per kW hr. The operation of the turbine is particularly interesting for the employment of high pressure, high temperature steam permits the use of a large heat drop, but in the case of small powered condensing turbines, such as these, the internal efficiency is seriously affected by the small volume of steam. In general, it is necessary to throttle the inlet steam upstream of the first stage in order to increase the specific volume—the pressure being considerably reduced. The gain in internal



Arrangement of Breguet auxiliary turbine injector compressor

A.—Upstream ejector compressor. B.—First stage of turbine. 1.—H.P. steam inlet, controlled by admission valve. 2.—Aspiration of steam downstream of first disc. 3.—Steam returns upstream of first nozzles.

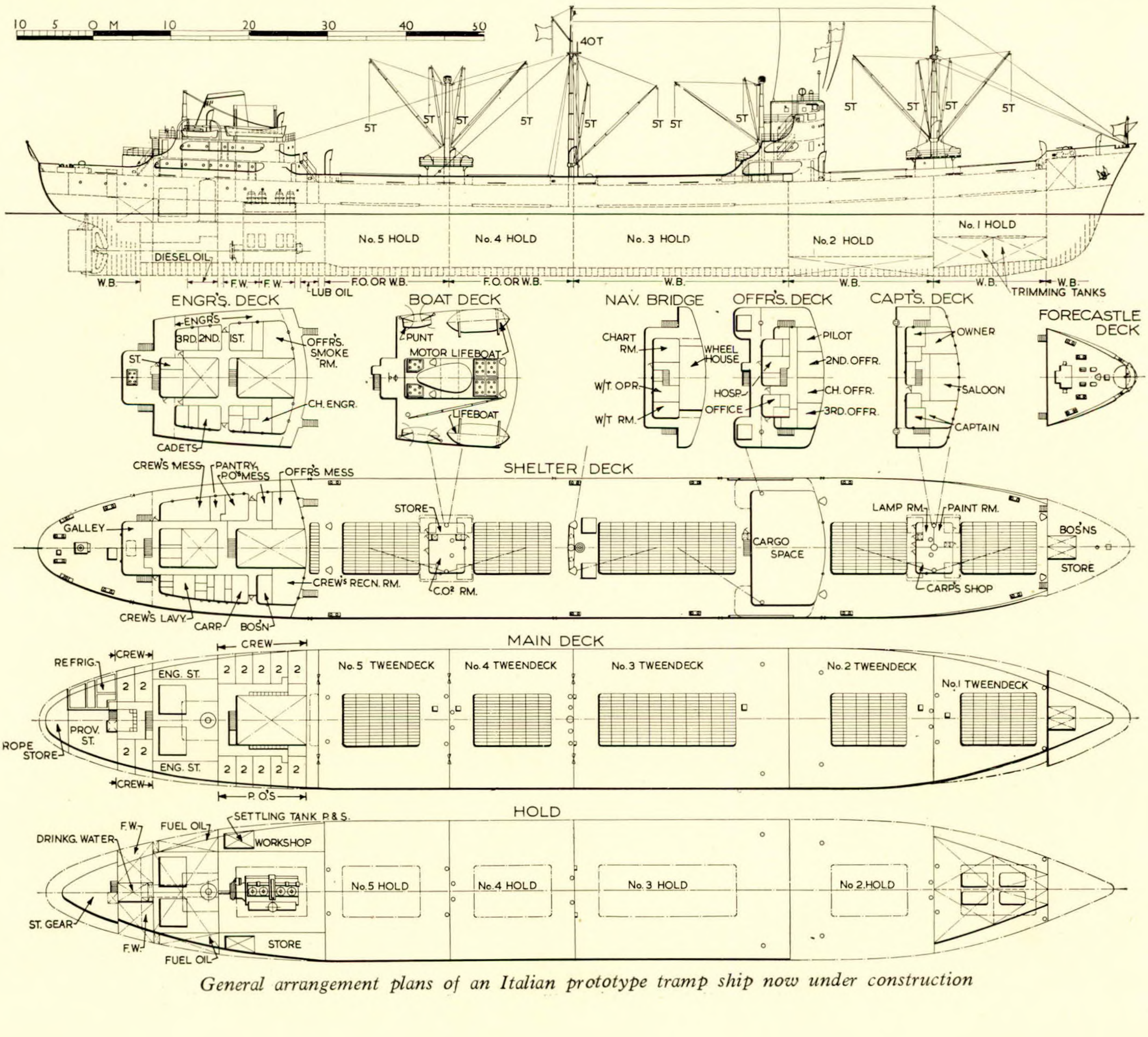
efficiency obtained under these conditions compensates for the loss in heat drop. The replacement of throttling by an upstream ejector compressor—a type of apparatus of which Breguet have long experience—makes it possible to improve the efficiency, for the energy resulting from the first expansion of the steam is used to draw the expanded steam through the first stage of the turbine and to inject it again into the same stage mixed with the live steam expanded. With this arrangement the first stage of the machine is fed with a much greater volumetric quantity of steam, thus improving the efficiency by a coefficient increased by injection. The ejector-compressor is essentially a static device requiring practically no attention. The figure shows the arrangement diagramatically.—*The Marine Engineer and Naval Architect, Annual Steam Number, 1954; Vol. 77, p. 423.*

Accident Due to Unsafe Practice

A fireman-water tender was painfully burned one day aboard a tank vessel as a result of an unsafe practice in which the chief engineer was employing an open flame blowtorch on a fuel oil line to heat the oil in the line. The boiler, which had been in idle status, was being lit off, with the chief engineer assisting the fireman. The chief had partially closed the bypass valve on the fuel oil manifold header until the oil in the header had warmed up to approximately 100 deg. F. Apparently not wanting to wait until the fuel oil to this boiler could be heated to a higher temperature by recirculating through the fuel oil heater, and believing the oil in the line to be too cool to safely light off, the chief lighted a blowtorch and played it back and forth along the fuel oil service line to the burner, testing with his hand to see how hot the line was getting. When the line reached a temperature estimated by the chief to be about 160 deg. F., he shut off the blowtorch and instructed the fireman to light off the burner in the boiler with the lighting off torch. Just as the fireman crouched over to insert this torch, the fuel service line ruptured and hot oil sprayed on to the fireman's face and right hand and arm. The chief immediately shut off the fuel oil header valve to stop the flow and administered first aid to the fireman. The burns received, while not incapacitating, were painful and required hospital treatment. The rupture in the fuel service line was only a slit about $\frac{3}{16}$th inch by $\frac{1}{32}$nd inch and occurred in a point in the steel tubing which was later found to be thin. While it could not be definitely proved that the rupture was caused directly by the impingement of the flames from the blowtorch, the only logical conclusion is that the actual failure of the line at that time was caused or precipitated by the use of the blowtorch. The chief engineer received an official admonishment for poor judgment in using an open flame on a high pressure fuel line—an unsafe practice, which was bound sooner or later to lead to misfortune.—*Proceedings of the Merchant Marine Council, United States Coast Guard, November 1954; Vol. 2, p. 185.*

New Italian Tramp Design

With the object of producing a tramp ship having the lowest capital cost and highest operating efficiency, the Cantieri Riuniti dell'Adriatico, Trieste, has developed a design which is now published, and the prototype ship is under construction. The size decided on was a vessel of 9,550 metric tons deadweight with an engine of 3,600 b.h.p., operating on boiler oil



General arrangement plans of an Italian prototype tramp ship now under construction and installed aft, so as to give five unobstructed holds, practically all rectangular in section. The first ship has been laid down to the order of the Società di Navigazione Lussino of Trieste and contracts have been made for two further vessels of the same design. The following are the main characteristics: —

Length b.p.	129·60 m.
Breadth, moulded...	18·10 m.
Depth, moulded to main deck ...	8·95 m.
Depth, moulded to shelter deck	11·60 m.
Draught, loaded	7·87 m.
Deadweight (metric)	9,550 tons
Gross register, about	6,550 tons
Cargo capacity	16,700 cu. m. (bales)
Machinery	3,600 b.h.p. at 115 r.p.m.
Speed on trials, fully loaded ...	13·2 knots

The propelling engine is a C.R.D.A.-Sulzer two-stroke single-acting unit of the new RSD type with a cylinder diameter of 760 mm. and a piston stroke of 1,550 mm. At 115 r.p.m. the output is 3,600 b.h.p. when operating on boiler oil. The speed on trials, as stated, will be 12·2 knots when fully laden. In normal service the speed will be 12 knots and the fuel consumption should be about 12 tons per 24 hr. for all purposes. The exhaust gas from the engine is passed through an exhaust-gas boiler, raising steam for driving the 40-kW steam-driven dynamos and for other purposes, so that it will be unnecessary to utilize oil fuel when the vessel is under way. In port, the Scotch boilers will be oil-fired and supply steam to the steam winches on deck. The ship will have a complement of eleven officers, twelve petty officers and eighteen men, totalling forty-one, in addition to which there is accommodation for the owner and a pilot. Water ballast is carried in the fore peak and in the trimming tanks under hold No. 1 or, alternatively, general cargo can be loaded in these tanks. Water ballast is also carried in the double-bottom tanks under Nos. 2 and 3 holds, and in the after peak. In Nos. 4 and 5 holds either fuel or water ballast may be accommodated, whilst under the engine room are fresh water and lubricating oil tanks: there is a Diesel oil tank under the boiler room. Fuel oil or water ballast may be carried in wing tanks in the boiler room. In the after peak there is water ballast. Two settling tanks are provided in the wings of the engine room.—*The Motor Ship, January 1955; Vol. 35, pp. 434-435.*

Norwegian Bulk Carrier

A recent delivery of A/S Fredriksstad Mek. Verksted to Jacob Kjode A/S, of Bergen, is the bulk carrier *Ingertre.* This ship, which is of 7,000 tons deadweight, is intended principally for the carriage of coal from Spitzbergen, though she will no doubt be employed in other trades in the winter months. She is a single-deck vessel with machinery arranged aft and bridge structure amidships. The four holds, two forward of and two abaft the bridge, are self-trimming, and topside ballast tanks are arranged for use on the outward passage. Each hold is served by a pair of light derricks of sufficient length to give a good outreach. The holds have a capacity of 325,850 cu. ft. excluding the deep tanks. The principal dimensions of the ship are as follows: —

Length o.a.	388ft. 10in.
Length b.p.	360ft. 0in.
Breadth moulded	54ft. 0in.
Depth	30ft. 2in.

Passenger accommodation is provided for six passengers in the midships deckhouse, which also contains cabins for the master and deck officers, and the wheelhouse, chartroom, etc. The rest of the crew is accommodated aft. The propelling machinery consists of a Fredriksstad steam motor, taking steam from two Scotch boilers arranged on deck in the shipbuilders' usual fashion. This arrangement is most commonly seen in ships where the machinery is amidships, though mention may be made of the steam tankers *Golden West* and *Amica* and the heavy lift vessel *Christen Smith,* completed by the Fredriksstad yard in 1945, 1946 and 1947 respectively, in all of which machinery of this sort is arranged aft. The engine develops 1,850 i.h.p., giving a loaded service speed of 11½ knots. The boilers burn oil under forced draught, and are fitted with superheaters and air preheaters. The auxiliary machinery includes two steam-driven generators of 25 kW capacity each, and a Diesel generator for use in port of 20 kW capacity.—*The Shipping World, 22nd December 1954; Vol. 131, p. 663.*

Residual Stresses in Hollow Cylinders Due to Creep

This paper describes investigations into the residual stresses produced in cast-iron cylinders by the creep-relaxation of thermal stresses, with the object of determining the magnitude of the residual tensions in relation to the initial compressions. The connexion of the work with failures of combustion-chamber parts in large internal-combustion engines is discussed. A series of thick hollow cylinders were subjected to a radial flow of heat by heating the bore and water-cooling the outer diameter for a chosen period of time, during which the thermal stresses were relaxed by creep. The residual stresses due to relaxation were found by slitting the cylinder and boring out successive layers of the material from the inside. The changes in diameter caused by this operation were measured by electrical-resistance strain gauges fixed to the outer diameter of the cylinder. The results indicate that the most rapid release of the total residual-stress component occurs within the first half-inch of boring, indicating that creep strains of any appreciable magnitude are confined to this region. The highest residual stress at the bore was four-and-a-half times that at the outside diameter. Creep in compression at the bore is the predominating factor in causing residual stresses on cooling. —*Y. G. Attia, D. Fitz-George, and J. A. Pope, J.Mech.Phys.Solids, 1954; Vol. 2, p. 238. Journal, The British Shipbuilding Research Association, October 1954; Vol. 9, Abstract No. 9,480.*

New Diesel Fuel-injection System

A new simplified Diesel fuel system, making Diesel fuel injection as simple if not simpler than gasoline-engine carburation and ignition systems, has been developed by Cummins Engine Company. The system uses one of the simplest types of fluid-metering devices known, namely, a fixed orifice with a variable fuel oil pressure behind it. The metering orifice is now a part of the fuel injector located in the cylinder head, thus adding the metering function to the injector. Although the injector now serves the dual purpose of metering and injecting the fuel, the construction and servicing of the injectors have been greatly simplified. With the metering function now in the injector, the once-complicated fuel pump has become simply a source of fuel pressure with a pressure regulator, throttle control, governor control, and "start-stop" control serving to restrict the flow of fuel and so varying the fuel pressure at the metering orifice. Injector plungers are operated from lobes on the main engine camshaft and control the timing of the engine, as in the past. However, since the timing is controlled by the engine camshaft, the new system need not be timed to the engine. Briefly, the fuel system operates as follows: A gear pump picks up the fuel from the fuel tank, creating the basic fuel pressure for the system. This pressure is then controlled by a pressure regulator which is designed to control the maximum pressure in the system and to give the proper torque and horsepower characteristics. The fuel pressure from this regulator is controlled by a simple throttle-control valve, a simple governor-control valve, and an "on-and-off" shutdown valve. From this point the fuel flows from the fuel pump to a common manifold leading to each of the injectors located in the engine cylinder head. The pressure from the fuel pump is applied to the metering orifice in the injectors and, depending upon the pressure in the manifold, more or less fuel is metered through the orifice into the injector cup in the top of the engine combustion chamber. At the proper

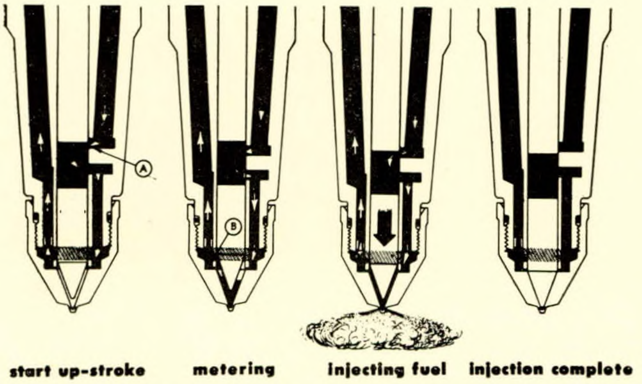


FIG. 8—*Fuel metering injection cycle of PT injector*

Start Up-sroke. As plunger moves up, fuel supply hole is uncovered, allowing fuel to circulate through the injector and out of drain at left. About four-fifths of the fuel delivered to the injector is returned to the fuel tank. *Metering.* Now plunger has uncovered metering orifice, *at left,* allowing fuel to enter injector cup. The length of time this opening is uncovered and the pressure supplied by the fuel pump determines the quantity of fuel injected. *Injecting Fuel.* As the plunger moves down it "pushes" fuel through holes in the tip of the injector cup and into the cylinder as an exceptionally fine spray, insuring thorough mixing with the air and complete burning of entire fuel charge. *Injection Complete.* Following injection the plunger remains seated until it is time to repeat cycle. Action of the injector plunger is controlled directly by engine camshaft assuring injection of the fuel charge at the most effective time.

time for firing, the injector plunger is forced down by the action of a cam on the main engine camshaft, forcing the metered fuel through the spray holes in the injector cup. Since all injectors are connected to the common manifold, all receive the same pressure and equal charges of fuel are injected into each of the engine's cylinders, assuring that each cylinder develops equal power without the necessity for individual adjustment of each injector. — *Mechanical Engineering, November 1954; Vol. 76, pp. 914-915.*

Condenser Tube Jet Impingement Tests

Co-operative research on materials for condenser tubes has for some time past been carried on in the United Kingdom and the U.S.A. Particular attention has been given during recent years to behaviour under the jet-impingement test which, in the early stages of condenser tube investigation, was found to simulate the type of attack encountered in sea water service. This paper describes the operation of jet-impingement test apparatus in the laboratories of the British Non-Ferrous Metals Research Association in London, and at The International Nickel Company's Marine Corrosion Test Station at Harbor Island, N.C. Differences in results obtained on identical materials in the two laboratories are discussed in relation to variations in test conditions, particular emphasis being laid on the differences observed between results of tests made with water many times recirculated (as in the B.N.F.M.R.A. laboratories) and those obtained with water which is passed through the apparatus only once (as has been regular practice at Harbor Island). The materials used in various series of tests (the results of which are individually reported) included arsenical copper, 70-30 brass, Admiralty brass, aluminium brass, and 70-30, 65-30, 90-10 and 95-5 cupro-nickels containing varying percentages of iron. The following general conclusions are drawn:—"Severity of jet-impingement tests with once-through sea water at Harbor Island was much greater than with recirculated sea water, under otherwise similar testing conditions, in the laboratories of the British Non-Ferrous Metals Research Association. Materials that were badly attacked in recirculated water were, however, no more severely attacked in once-through water. Tests with once-through water in England gave less severe attack than those with once-through water at Harbor Island, but results otherwise were very similar. Under severe impingement-test conditions it was not necessary to have air bubbles in the water for attack to occur. Air bubbles had more effect in recirculated water in the B.N.F.M.R.A. laboratory than in recirculated water at Harbor Island. Relative behaviour of common condenser tube alloys when tested at the level of severity provided by tests with recirculated water at 15ft. per sec. in the B.N.F.M.R.A. laboratory was in line with their performance under many conditions of practical use. Tests at Harbor Island can be made to duplicate results from the B.N.F.M.R.A. laboratory either by reducing jet velocity from 15ft. per sec. to about 4ft. per sec., or by recirculating the water at 15ft. per sec. The presence of suspended plankton in the water at Harbor Island is suspected as being the principal factor in increasing the severity of tests with once-through water. The greater severity of the test with once-through water at Harbor Island should provide a means for qualifying materials to resist exceptionally severe conditions of impingement attack in practical service".—*P. T. Gilbert and F. L. Laque, Journal Electrochemical Society, September 1954; Vol. 101; pp. 448-455. Abstract in The Nickel Bulletin, No. 11, 1954; Vol. 27, pp. 218-219.*

Inspection, Explosion and Breakdown of Boilers and Pressure Vessels

An interesting service failure recently occurred in the cover bolts of a high pressure feed water heater, details of which are given in Fig. 9. The bolts were $2\frac{1}{16}$in. in diameter by 2ft. $1\frac{1}{4}$in. long with a longitudinal hole $\frac{5}{8}$in. in diameter drilled through the centres and made of nickel-chrome steel having an ultimate tensile strength of 60 tons per sq. in. Several were found to be badly cracked after six years' service. Owing to the length of bolts and the massive thickness of the flanges of the cover and water box, there was an appreciable time lag between the temperature equalization of the bolts and the surrounding metal, and trouble had been experienced in keeping the cover joints tight under those conditions. Metallurgical examination of one of the bolts revealed two sets of fractures in the root of the collar fillet, and at the line of the cover joint respectively. A third family of cracks tending to run longitudinally in the bolt was present at the bottom of a small drilled hole in the circumference of the collar into which a locating pin had been driven. Selected portions of the fractures were broken open and the surfaces examined, but no evidence of the presence of alkali or other boiler water salts was found. There was present, however, a considerable quantity of rust, confirming the fact

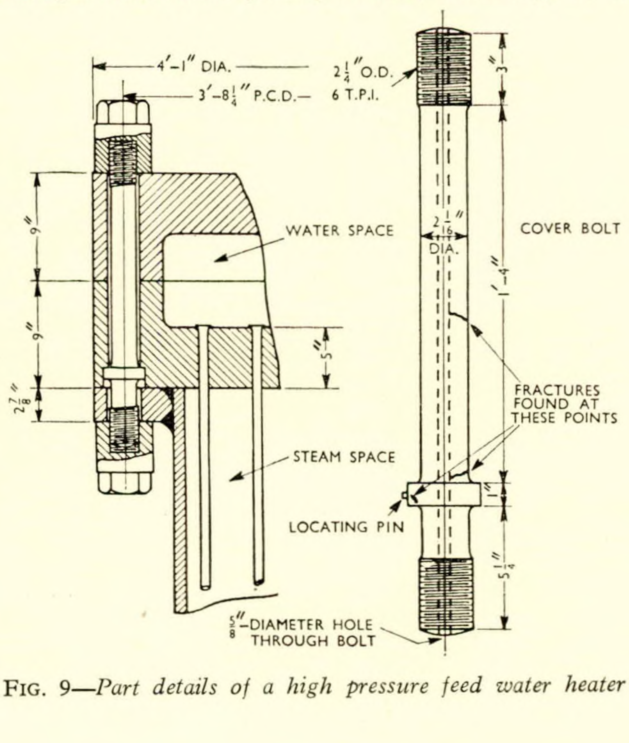


FIG. 9—*Part details of a high pressure feed water heater*

that the bolts had been in the presence of water and, in fact, various small corrosion pits had been produced on the bolt surfaces. Micro-examination revealed the presence of numerous non-metallic inclusions in the steel and it was noted that the cracks had a pronounced tendency to associate with these inclusions, branching from one to another. The examination indicated that the cracks in the bolts were of a fatigue type and that several factors were associated in their development. One of the principal causes of breakdown was the presence of water on the stressed material which not only would have caused a serious reduction in the fatigue limit but also, by the production of pitting, would form highly stressed centres favourable to the origination and development of fractures. The dirty nature of the steel would also help in the rapid propagation of the cracks once these had been initiated. A major engineering development during the past two decades has been the general adoption of welding in the construction and repair of pressure vessels. It is of primary importance that the stress welds of such plant should be of the highest quality, and fabricated by certified welders. The history of numerous breakdowns due to defective welding gives impressive evidence of the necessity for this precaution. In this connexion the recent explosion of a welded-steel hot water cylinder projected the vessel upwards with great violence. This cylinder was directly connected to the jacket of a chemical vessel through which cooling water was circulated by means of a pump at a pressure of 60lb. per sq. in. Investigation made after the occurrence proved that the vessel should never have been used for pressure purposes. The welded seams viewed externally appeared to be of excellent quality but, in point of fact, embodied serious faults. No edge preparation of the plates had been carried out prior to welding, which had been undertaken from one side of the plates only, and little attempt appeared to have been made to secure proper penetration. In addition, the junction of the endplates to the shell was effected by corner welding without any knuckle radius, with the result that a high stress concentration was imposed favourable to the formation of fractures and, in fact, it would appear that cracking had been initiated at this point during manufacture. Failure occurred through the bottom endplate being blown out of the vessel owing to the rupture of the corner welding. The cylinders had been subjected to hydraulic test by the manufacturers at a pressure of 120lb. per sq. in., but no evidence of weakness was then observed. The facts of this failure emphasize that a hydraulic test can only be regarded as a measure towards proving that a vessel is safe for the required working pressure which must primarily be assessed by direct calculation based on full knowledge of the construction. Many accidents have resulted from the dangerous practice of conducting a pressure test by the medium of air or steam. It may be remarked that the same danger exists if, during the application of a hydraulic test, substantial pockets of air are entrapped. Some time ago, during the initial hydraulic test of a large watertube boiler, the ends of two tubes were suddenly blown out of position and the subsequent whip of one of them killed an engineer who had been witnessing the test. On another occasion a surveyor was hammer-testing the welding of a large new cylindrical steam drum under hydraulic test when a longitudinal seam failed for a length of 6 feet. Air had been trapped within the drum with the consequence that the inspecting engineer was struck by a jet of water propelled with such force through the ruptured seam that he died a few minutes later. It was afterwards discovered that the welding of this vessel was seriously defective. Unfortunate as were the results of the test, it is evident that this prevented a major disaster through failure of the vessel when in commission.—*J. Eyers, Engineering and Boiler House Review, January 1955; Vol. 70, pp. 11-17.*

Hydrogen Engine

A report by Canadian scientists in the journal "Nature", states that the successful operation of an internal combustion engine using a hydrogen-air fuel has been carried out. The experiments were carried out in the department of mechanical engineering at the University of Toronto for the Defence Research Board of Canada. Previous attempts to use hydrogen-air mixtures as fuels had failed because of the high temperatures of the ceramic core of the sparking plugs and of the exhaust valves, causing premature ignition. When this was overcome and the progress of the lubricating oil (which had been involved in the premature ignition cycle) into the combustion chamber was slowed down, the engine ran successfully at 1,800 r.p.m. Compression ratios of up to 12 to 1 and fuel mixtures ranging from 80 per cent weak in fuel up to 70 per cent rich in fuel were, it is stated, used with success.—*Gas and Oil Power, December 1954; Vol. 49, p. 311.*

Lubricating Oil with Additive

A study of this problem of high rates of wear associated with boiler fuel which appears to apply particularly to two-stroke engines indicated that, although the choice of cylinder liner material was an important factor, the main cause of excessive liner wear was corrosion due to acidic combustion products. An appropriate line of attack, therefore, would be the use of a lubricant which would effectively prevent this type of corrosion. It was already known that some types of "H.D." Diesel engine oil are very effective in reducing cylinder wear in engines in which the pistons and cylinder walls are copiously lubricated with the crankcase oil, but it was not known whether such oils would be sufficiently powerful to cope effectively with the problem in larger marine Diesel engines where oil is fed in relatively small quantities through mechanical lubricators. Test bed and sea trials in engines burning boiler fuel showed that, in fact, wear rates of the order of 0·4 mm. per thousand hours can be reduced substantially by the use of some H.D. type oils of high additive content. However, at higher levels of wear the reduction obtained was relatively small and in some cases the wear rate was not significantly lower than with straight mineral oils. The trials were pushed to the extent of using oils with very high concentrations of additive without really substantial reductions in wear rate being obtained and the higher additive levels required to produce even significant reductions in wear did not afford an economic solution to the problem. Consequently, a new approach was made, the outcome of which was the development of the new cylinder lubricant which is, at present, branded Shell Oil S.4264. The preliminary trials on this lubricant were carried out in marine-type Diesel engine in the laboratory and gave such encouraging results with boiler fuel that they were soon extended to trials with marine Diesel engines in service. The first sea trials with Shell Oil S.4264 began in December 1953, on ships using boiler fuel which were experiencing abnormally high cylinder liner wear. For comparison purposes the oil normally used was retained in some of the cylinders of the engines concerned. Since then the number of ships using Shell Oil S.4264 has increased considerably and, whilst it is still too early to obtain reliable wear data from the majority of them, results are now available from the earlier trials which show conclusively that the new lubricant has reduced considerably the wear rates obtained with both straight mineral and H.D. type oil. The Shell Oil S.4264 is a stable water-in-oil emulsion having a viscosity lying in the S.A.E.50 class and containing special additives to prevent corrosion wear.—*The Motor Ship, January 1955; Vol. 35, p. 425.*

Patent Specifications

Surface Condenser

This invention relates to surface condensers, the object of the invention being to provide a condenser, wherein the heat transfer from steam to the cooling liquid is at an extremely high rate, with the result that the dimensions of the condenser may be small and, consequently, the manufacturing cost may be low, and wherein the resistance to flow of the steam passing through the condenser is so slight as to reduce the counter-pressure of the machine connected to the condenser. The condenser shown in the line drawing Fig. 1, is intended for the condensing of steam, the cooling liquid used being water. It consists of a cylindrical sheet-metal container (1) provided with a steam inlet (2) at the top and an outlet (3) for condensate at the bottom. Tube plates (5 and 6) are provided at opposite ends respectively of the container (1) to cover circular openings eccentrically located in the lower portion of end walls (4). Tubes (21) are fitted in and between the tube-plates symmetrically about a vertical central free space (22). The tubes are subdivided into two horizontal groups one comprising nests of tubes (25, 26), and the other comprising nests of tubes (27, 28). The groups are arranged in superimposed relation, and these nests are mutually separated by horizontal sheet metal screens (24). A fifth nest (15) is located centrally of the lowermost nest (25) and is shielded off from the latter by plates (14) and from the central free space (22) at the upper region of this nest by vertical plates (18). The screens (24) sloping slightly inwards towards the centre are intended to collect and drain off towards the central space (22) condensate formed from the condensing steam, and also to lead air separated from the steam to space (22). The screens (14 and 18) together form a kind of hood into which open a number of air exhaust pipes (16). The pipes (16) through openings (20) in the walls, interconnect the hoods with longitudinal headers (17) attached to conventional means (not shown in the drawing), such as a suction pump of the condenser, for sucking-off air separated from the condensing steam. All the tubes (21) open into chambers (10, 11, 12) outside the tube plates (5 and 6) and formed by walls (7 and 8). The wall (7) and tube plate (5) form firstly, a lower admission chamber (10) with an inlet (9) for cooling-water, and also an upper outlet chamber (12) with an outlet (13) through which the used cooling-water is discharged. At the opposite end of the container, the chamber (11) is common to all the tubes. The tubes of the nests (15, 25 and 26) open into the admission chamber (10) and comprise parallel paths leading the entering cold cooling-water to the common chamber (11) whence the tubes of the upper nests (27 and 28) lead the cooling-water to the outlet chamber (12) from which it is discharged through outlet (13). Since the cooling-water is first admitted to the lower tube group formed by nests (25 and 26), the cooling is more intense in this group than in the upper group formed by nests (27 and 28). In each of the nests (25 to 28) the tubes are arranged in straight rows, disposed side by side and diverging upwards from the central space (22), the divergency of the rows of the nests (25 and 26) being greater than that of the rows of the nests (27 and 28); for example, about $1\frac{1}{2}$ to 2 times as great. Thus the tenth row as shown in Fig. 1 from the centre may form an angle with the innermost row of about 35 degrees to 45 degrees in nest (26) and 15 degrees to 20 degrees in nest (27). The rows of tubes in each nest (25 to 28) of the condenser illustrated are located in lines radiating from a common centre.—*British Patent No. 715,162, issued to Aktiebolaget De Lavals Angturbin. Application in Sweden made 9th June 1951. Complete specification issued 8th September 1954. Engineering and Boiler House Review, December 1954; Vol. 69, p. 390.*

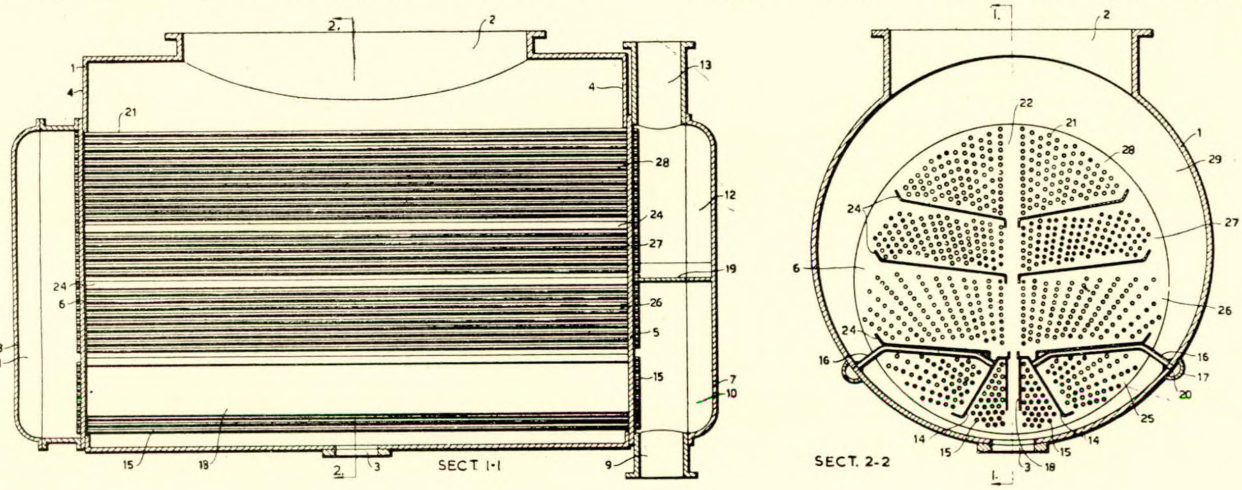
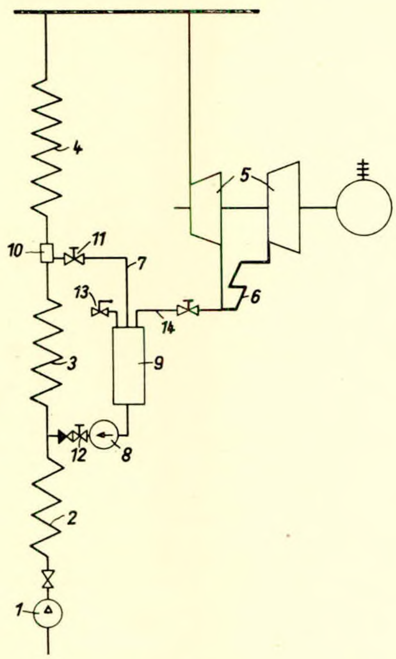


FIG. 1

Improvement in Forced Through-flow Boilers

With every type of boiler, regardless of whether it is of the natural circulation type, forced circulation or forced through-flow type, there is a lower load limit below which considerable difficulties arise, both from the firing and the water feed points of view. The invention provides a method and a boiler whereby it is possible, with comparatively simple means, to start with a load range down to about 10 per cent of the normal load and in certain cases even less. The drawing shows a construction of a forced through-flow boiler with the feed pump (1), the preheater (2), the vaporization section (3) and the superheater (4). The turbine is (5) and the steam reheater is designated by (6). An arrangement for operating at minimum load comprises the by-pass pipe (7) for the endangered vaporizer section (3) and the circulation pump (8). As a result of this shunt to the part or section (3), it is always ensured that a sufficiently large quantity of water is circulated even at minimum load operation. It is, however, expedient to connect a drum or collecting container (9) in this pipe, in order to have a certain buffering possibility in the event of any fluctuations. The pipe (7) is not connected directly to the tube system of the forced through-flow boiler, but to a water and steam separator (10) in which steam and water are separated. A shut-off valve is indicated by (11) on the return flow pipe (12). The safety valve (13) has already been mentioned. For the purpose of producing a sufficiently large pressure drop between the separator (10) and the container (9), there is provided the pipe (14) which connects the container (9) with the intermediate steam reheater (6) and thus with a stage of correspondingly lower pressure.—*British Patent No. 719,753, issued to Siemens-Schuckertwerke A.G. Complete specification published 8th December 1954.*



Ballast for Sea-going Vessels

The prior known arrangements for ballasting sea-going vessels have many disadvantages. The introduction of sand or other solid ballast is expensive and time-consuming, particularly because of the loading and unloading operations of the ballast and the loss of time caused thereby. Furthermore, additional expense is involved by the buying and transporting of the ballast. As water-ballast provided in the double bottom and similar available spaces in the vessel is not usually enough to submerge the propeller sufficiently deeply and give the vessel the necessary stability in rough seas, there are many vessels in existence today which need to take on further solid ballast on long journeys. Liberty ships always have stone ballast rigidly built into them. This permanent ballast is, of course, detrimental to the carrying capacity and is, therefore, extremely uneconomical. The disposition of supplementary water tanks above the double bottoms has not been found suitable since a considerable part of the hold space is lost. According to the invention, a ballasting arrangement for sea-going vessels consists in the provision within the hold space of the vessel of a number of receptacles which are arranged to be assembled and filled with water to ballast the vessel and to be dismantled and stowed away when ballasting is not required. In Figs. 2 and 3 is shown in sectional plan view and side view respectively how a hold space is divided into separate cells, tanks or chambers (32) by means of additional intermediate walls (28, 29) stretched between the vessel's walls, bulkheads, propeller shaft tunnel and so on, for example, textile materials, rubber strips or other webs. Fig. 4 shows an embodiment of the water-receiving receptacles formed of hollow bodies, which in this embodiment consist of watertight sacks (33), which are conveniently made of sail cloth, nylon fabric or rubber webs. In order to give the water-sacks (33) or, as mentioned above, the walls (38-31) greater stability, they can be stiffened and strengthened with boards (40) carried in brackets (39), which can conveniently take the form of flexible loops.—*British Patent No. 721,405, issued to H. Schluter and Co. Application in Germany made 26th January 1952. Complete specification published 5th January 1955.*

Fig. 2.

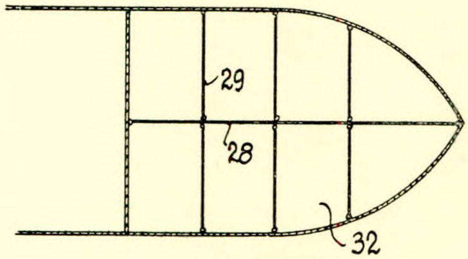


Fig 3.

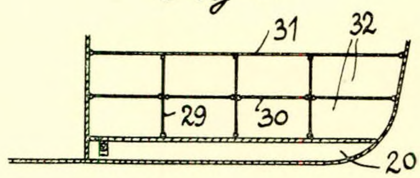
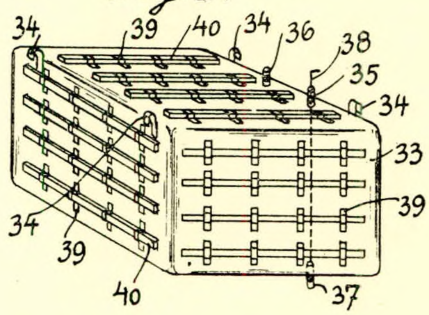


Fig. 4.



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Marine Engineering and Shipbuilding Abstracts

Volume XVIII, No. 4, April 1955

German Built Vessels for Indian Owners

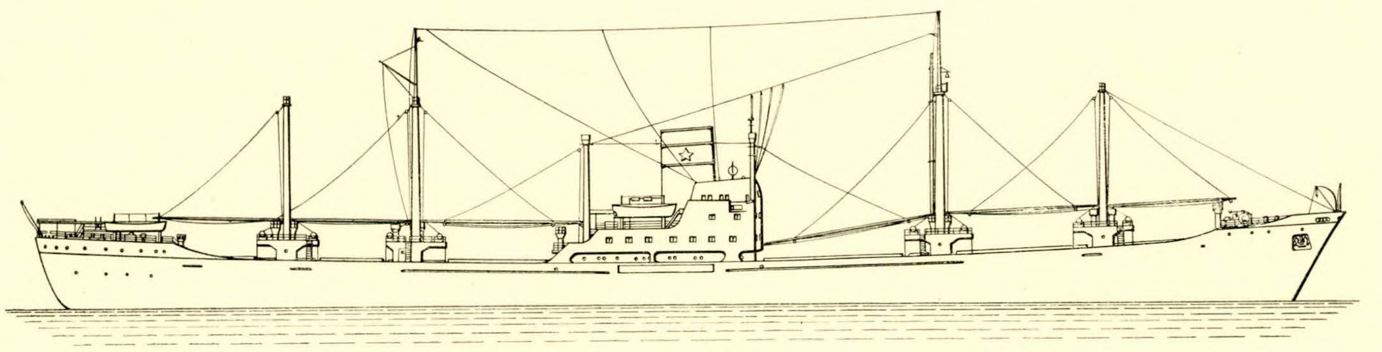
The *Indian Reliance* and *Indian Renown*, recently launched for the India Steamship Company by Howaldtswerke A.G. Hamburg, are the largest and fastest vessels built to date for the Indian mercantile marine. Both vessels were launched on the same day, the *Indian Renown* entering the water first, only fifteen minutes before her sister ship. The principal particulars are:—

Length overall ...	531ft. 3in.
Breadth moulded...	63ft. 0in.
Depth to first deck ...	39ft. 4½in.
Depth to second deck ...	30ft. 2¼in.
Deadweight tonnage ...	10,000 tons
Service speed ...	17½ knots

Intended for the owners' India/United Kingdom/North Continental service, the ships have six cargo holds, fitted with MacGregor steel hatch covers and have an approximate grain capacity of 680,000 cu. ft., including vegetable oil tank space of 24,000 cu. ft. Two special compartments are also provided in the forecastle for dangerous cargoes. Cargo handling appliances include a total of twenty derricks, all served by electric winches. The derricks can lift varying loads from 3-10 tons and in addition a heavy derrick with a safe working load of 50 tons is provided. A full range of radio and navigation equipment will be installed, including Marconi Radiolocator Mark IV radar, Sperry gyro and automatic steering equipment, visual and recording echo sounders, etc. Accommodation for officers and crew will be of a high standard. The propelling machinery will consist of steam turbines developing a maximum of 9,900 s.h.p. at 115 r.p.m., steam being supplied by two oil-fired watertube boilers with a steam output of 40,000lb. per hr. Working pressure will be 605lb. per sq. in. with a final steam temperature of 830 deg. F. The necessary electric power will be provided by a 400 kW turbo generator and three 200 kW Diesel generators. The first of the vessels was expected to be delivered by the end of 1954 and the second early this year.—*Shipbuilding and Shipping Record, 16th December 1954; Vol. 84, p. 817.*

Large Turbine Tanker

The Gulf Oil Corporation of New York have an important new construction programme in hand at European yards, the vessels, when completed, being operated by their subsidiary, the Afran Transport Co. of Liberia. The Furness Shipbuilding Co. Ltd., of Haverton Hill-on-Tees have delivered six of these vessels, mainly in the 25,000-ton d.w. class, but the subject of this article is the largest so far built on the river, and represents a new type of Furness-built vessel. The new ship has been constructed under special survey to the highest requirements of Lloyd's Register of Shipping and the American



Outboard profile of the Indian Reliance *and* Indian Renown

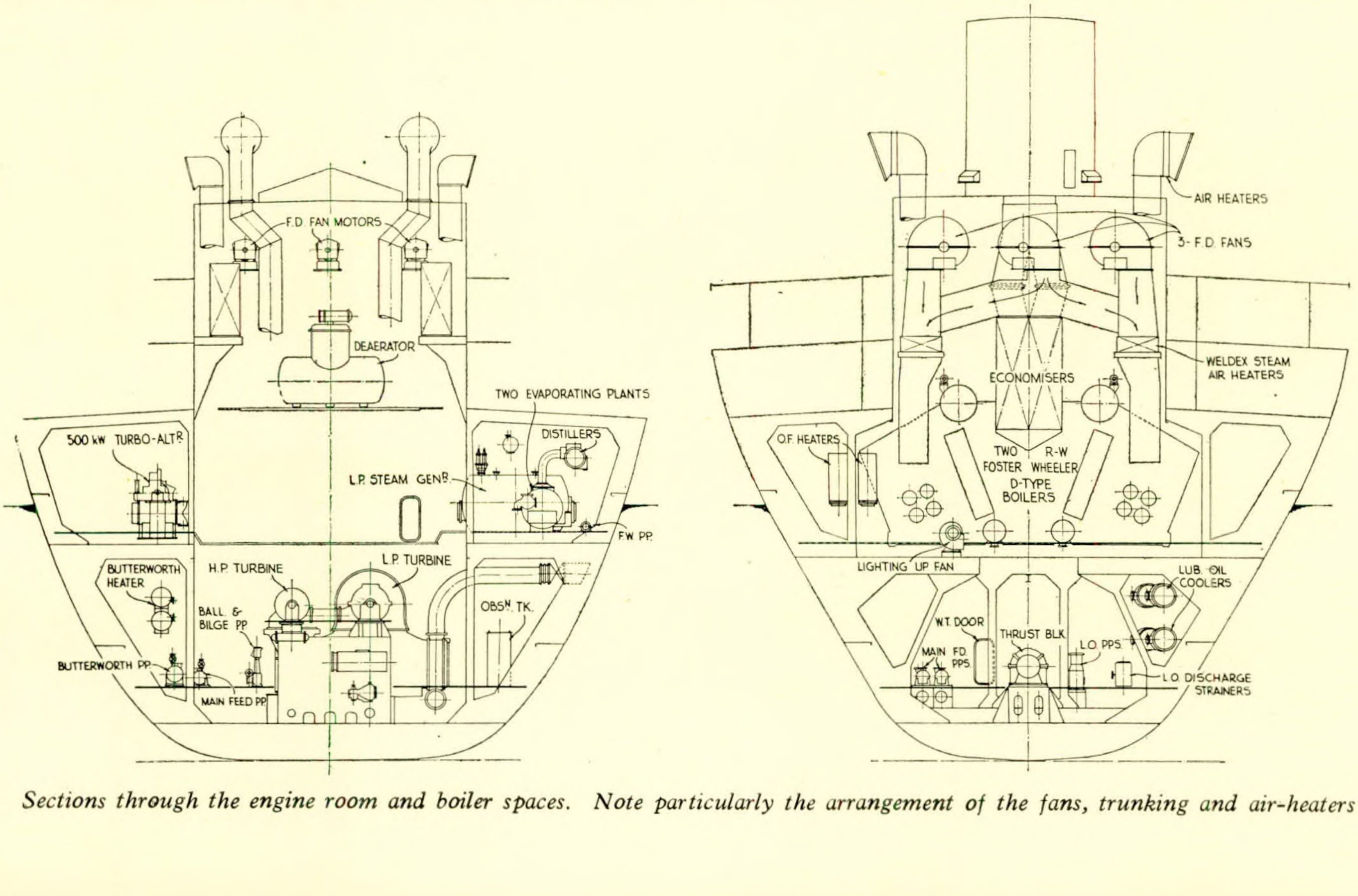
Bureau of Shipping. She has the following principal dimensions:—

Length overall	661ft. 7in.
Length b.p.	630ft. 0in.
Moulded breadth	87ft. 0in.
Moulded depth	45ft. 6in.
Summer draught	34ft. 2in.
Corresponding deadweight ...	32,020 tons
Bunkers	2,838 tons

The longitudinal system of framing has been adopted. Electric welding has been used extensively, the shell and decks being almost completely welded and the longitudinal and transverse bulkheads being welded by the Union-Melt system as especially developed by the builders. The propelling machinery of the *Melika* consists of a single-screw set of Richardsons Westgarth-Brown Boveri double-reduction geared turbines designed to develop 12,500 s.h.p. at 105 r.p.m. in normal service and a maximum continuous output of 13,750 s.h.p. The h.p. ahead turbine is of impulse-reaction design and the l.p. turbine is of all-reaction design. The h.p. turbine casings are of cast steel and the l.p. turbine cylinder is of fabricated construction with cast steel shells built into a mild steel fabricated casing. At normal power the h.p. turbine runs at 4,350 r.p.m. and the l.p. turbine at 3,000 r.p.m. The astern turbines, comprising an h.p. turbine of impulse design and an l.p. reaction element, are incorporated in the h.p. and l.p. casings respectively and are capable of developing 65 per cent of the normal ahead power. Suitable bled steam tappings are provided for feed heating and air preheating purposes. The main condenser is of Weir regenerative type, manufactured by the engine builders, and with a cooling surface of 14,000 sq. ft. it is designed to maintain a vacuum of 28¾ inches at normal full power with a sea water temperature of 75 deg. F. The condenser shell is of mild steel plate, with aluminium brass tubes expanded in the rolled brass tube-plates at one end and fitted with Crane's packing at the other. Two Richardsons Westgarth-Foster Wheeler D-type boilers are carried on a flat aft of the gearing and above the thrust block. Each boiler has a total generating surface of 15,065 sq. ft., a superheater surface of 1,680 sq. ft. and a maximum evaporation of 80,000lb. of steam per hour at stop valve conditions of 600lb. per sq. in. and 850 deg. F. The steam and water drums are fusion welded and the convection type superheaters consist of U-type elements expanded into mild steel headers, the outlet header being of alloy steel. An economizer of all-welded construction and consisting of solid drawn 2in. o.d. steel tubes with shrunk-on cast iron gilled rings has been supplied for each boiler by E. Green and Sons, Ltd., of Wakefield. An internal coil desuperheater capable of reducing the temperature of 60,000lb. of steam per hour from 850 to about 600 deg. F. is fitted in each steam drum. Desuperheated steam is used by the turbo feed pumps, the cargo pumps, and for other purposes. The Bailey system of automatically proportioning the fuel/air ratio is employed. A Weldex bled steam air heater with a surface of 3,160 sq. ft. is arranged in each downcast trunking. These heaters, of the horizontal stack type, are exceptionally neat and take steam bled from the main turbines between the h.p. and l.p. stages. Each heater is designed to raise 5,500lb. of air from 80 to 216 deg. F. per hr., when supplied with steam at 35lb. per sq. in. Two Weir distilling plants are carried on the same flat as the l.p. steam generator. Each is of horizontal, single-effect, submerged-tube type having an internal vapour purifier, a two-stage air ejector and a combined distiller condenser and feed preheater. These sets, which are complete with observation and test tanks, brine and extraction pumps, are capable of providing 47 tons of made water per day when supplied with saturated steam at 5lb. per sq. in. In normal service the evaporators will be supplied with steam bled from the turbines at a pressure of 7½lb. per sq. in. abs.—*The Marine Engineer and Naval Architect, November 1954; Vol. 77, pp. 431-437.*

Six-bladed Propeller

A six-bladed propeller, reported to be the first of its type on a Wear-built vessel, has been fitted to the 10,000-ton d.w. motorship *Christina Pezas,* at Sunderland South Docks by the North-Eastern Marine Engineering Co. (1938), Ltd. It weighs 7 tons 18 cwt. and has a diameter of 15ft. 6in. and is made of



Sections through the engine room and boiler spaces. Note particularly the arrangement of the fans, trunking and air-heaters manganese bronze. The *Christina Pezas,* which Short Bros. are building for Mr. Apostolos Pezas, of Chios, is the first Greek cargo motorship to be launched since the war.—*Shipbuilding and Shipping Record, 2nd December 1954; Vol. 84, p. 746.*

Coal and Iron Ore Carrier for Australia

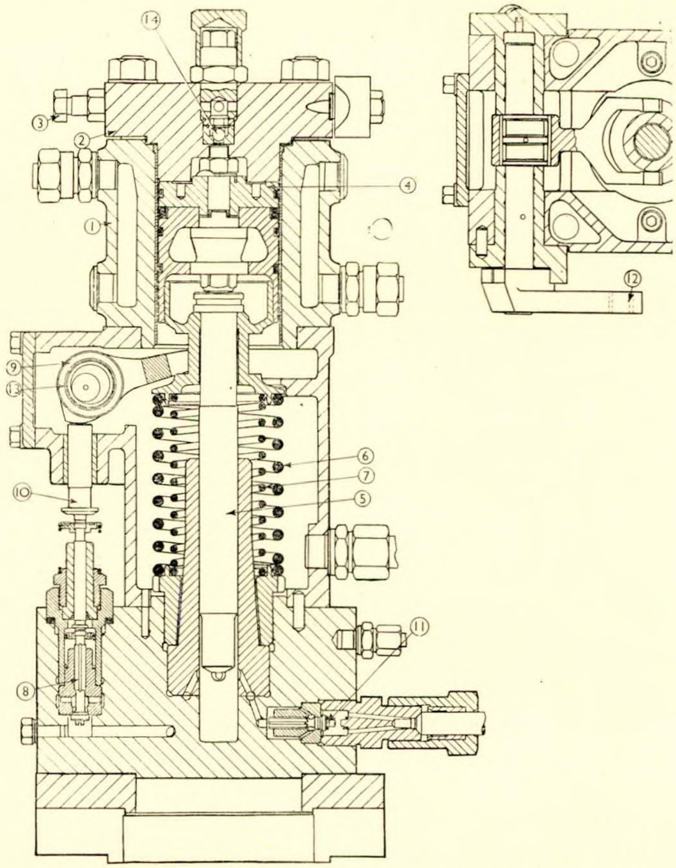
After the completion of successful trials, the single-screw steamship *Timbarra,* built by the Blyth Dry-docks and Shipbuilding Co., Ltd., Blyth, for the Australian Shipping Board, has entered her owners' service. The vessel has been specially designed for the carriage of coal and iron ore, and has been constructed under the special survey of officers of Lloyd's Register of Shipping for the highest classification of that society. The construction and outfit of the vessel are also in accordance with the requirements of the Ministry of Transport and the Australian Navigation Act. The principal dimensions and other leading characteristics of the *Timbarra* are as follows: —

Length b.p.	435ft. 0in.
Breadth moulded...	58ft. 6in.
Depth moulded	34ft. 0in.
Load draught	26ft. 0in.
Corresponding deadweight, tons	10,000
I.H.P. on trial	3,500
Corresponding r.p.m.	90
I.H.P. in service	3,100
Corresponding r.p.m.	86

Electric welding has been used extensively in the construction of the vessel, riveting being confined to the strength-deck stringer angle, the main girder connexions and the shell bilge strake. The remainder of the shell, decks, frames, double bottom and bulkheads are all welded. The bulkheads, it may be mentioned, are of the corrugated type. The rudder is actuated by means of an electro-hydraulic steering gear, manufactured by Donkin and Co., Ltd., of Newcastle on Tyne. The gear is housed on a special flat, and is of the two-ram type, with duplicated variable-stroke pumps and motors. The arrangements are such that either pump can be used for normal steering, and both pumps can be brought into operation for rapid manœuvring. Control is from the bridge by means of a Donkin automatic by-pass telemotor. The vessel is propelled by a single four-blade, manganese-bronze propeller of Unislip design, manufactured by the Manganese Bronze and Brass Co., Ltd., of Birkenhead, who have also provided the spare propeller. Power for driving the propeller is supplied by a superheated, triple-expansion, reciprocating steam engine constructed by the North-Eastern Marine Engineering Co. (1938), Ltd., at Sunderland; the North Eastern Reheat system is incorporated. Cam-operated poppet valves are fitted to the h.p. and m.p. cylinders, and a Martin and Andrews balanced-type slide valve is fitted to the l.p. cylinder. The balance weights are forged solid with the h.p. and m.p. crankwebs, and the sequence of the cylinders from fore to aft is h.p. (26in. diameter), l.p. (73in. diameter) and m.p. (41in. diameter). With a stroke of 48in., the engine is capable of developing 3,500 i.h.p. at 90 r.p.m. on trial and 3,100 i.h.p. at 86 r.p.m. in service.—*The Shipbuilder and Marine Engine-Builder, December 1954; Vol. 61, pp. 694-702.*

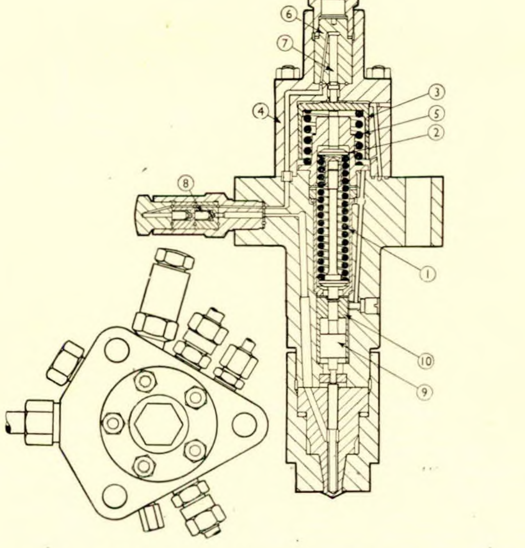
New Fuel Injection System

Shortly after the war, Wilson and Kyle, Ltd., undertook the manufacture of solid injection conversion equipment for marine propulsion and auxiliary engines originally fitted with air injection. This system consisted of compression gas-operated combined fuel pumps and fuel valves, and the engines of about sixty-five vessels have been converted by this equipment. The system was later developed to suit new engines and has been installed on Harland-B. and W. two-stroke single-acting, opposed-piston engines, having cylinders 750 mm. in diameter, the piston stroke being 1,500 + 500 mm. These engines were built by Harland and Wolff, Ltd., and J. G. Kincaid and Co.,



Section through compression-pressure-operated fuel pump

Ltd., for six Blue Funnel Line vessels, namely, the *Adrastus, Alcinous, Elpenor, Eumæus, Laomedon* and *Lycaon.* Supercharging is not employed and the high pressure gas pumps deliver fuel to two injection valves per cylinder. Metering of the fuel is carried out by a separate pump driven from the crankshaft. A new type of gas-operated fuel pump, incorporating a metering arrangement, has been developed in order to dispense with the need for a separate pump and accumulator. In the new design the capacity has been increased to meet the



Details of fuel injection valve

demand for a fuel pump capable of delivering the much larger volume of fuel per stroke required for supercharged engines. An important feature of this system is that no camshaft is required. Messrs. Alfred Holt and Company have adopted the new fuel pumps and fuel valves (for which applications have been made for patents) for the supercharged engines now under construction for their fleet. These engines will operate on boiler oil. One of the accompanying illustrations shows a section of the gas-operated fuel pump. Compressed gas from the engine cylinder is fed to the pump cylinder (1) through a pipe and a passage in the cylinder cover (2), the rate of flow being regulated by a screw (3). The gas piston (4) and the plunger (5) have a differential ratio of 10:1, so that if the compression pressure is 500lb. per sq. in., the pressure exerted on the fuel in the lower pump chamber will be 5,000lb. per sq. in. The piston and plunger have a constant stroke and after each delivery they are returned to the top position by double springs (6, 7). Fuel is delivered from the usual low pressure surcharge pump through a suction valve (8) to the chamber below the plunger. The fork lever (9) follows the movement of the plunger and the cam portion of the lever acting on a tappet (10) opens and closes the suction valve. As the piston and plunger move down, fuel is spilled at a low pressure through the suction valve, back to the supply line until the valve closes. Delivery then commences under high pressure through the valve (11) to the injector and ceases when the plunger covers the discharge port to the delivery valve. The oil trapped below this port acts as a cushion for the plunger. Metering of the fuel is carried out by moving the lever (12) on the eccentric shaft (13), thus varying the point at which the suction valve closes and delivery commences. The point at which this valve closes regulates only the quantity of fuel to be delivered and the actual timing of injection is carried out by the fuel valve. On the upward stroke of the piston the gas in the cylinder is expelled through a ball valve (14) back into the pipe, thus bypassing the regulating screw so that the delivery chamber can be recharged quickly. The gas cylinder (1) has a cooling jacket and oil or water can be used as a cooling medium. The gas piston is lubricated through two non-return valves in the gas cylinder. The fuel valve has been developed to work in conjunction with a gas-operated fuel pump on engines where the compression pressures vary considerably. Injection commences at the correct moment in relation to the position of the engine piston over the complete range of revolutions, from starting to full power. The initial opening pressure of the nozzle is controlled by a comparatively light spring (1). The top portion of the upper spring keep (2) bears on the spring casing (3), which is held against the housing (4) by a spring (5). The load of this spring does not act on the nozzle needle. A sleeve (6) is mounted in the top housing carrying a plunger (7) which is in contact with the spring casing (3). Fuel at the delivery pressure is fed to an annulus in the upper portion of the sleeve, through a double check valve (8). Under starting and slow-speed conditions the oil pressure acting on the plunger (7) is not sufficient to compress the upper spring (5). As the engine revolutions increase, the pump delivery pressures rise in direct ratio to the engine cylinder compression-pressure and the plunger (7) forces the spring casing (3) downwards, thus compressing the spring (1) and raising the injection pressure. Conversely, this pressure will be lowered as the engine revolutions and fuel delivery pressures are reduced. A small amount of oil is allowed to leak past the plunger (7), so that there will be no delay in reducing the injection pressure as the engine revolutions are lowered.—*The Motor Ship, December 1954; Vol. 35, pp. 388-389.*

New Glass-like Fibre Product

A considerable improvement in glass-fibre hull board is reported. Known as "Microlite" lightweight hull board, it has been well received by shipyards and the U.S. Navy as an alternative to the JAN-G-742 Specification. The marine field has long recognized the excellent insulating properties of glass-fibre insulation. For the past fifteen years the United States Navy has specified a glass board-like product faced with glass cloth as the interior finish for living quarters and operational areas aboard ship where insulation is required for personnel comfort. The product used for the past fifteen years has a nominal fibre diameter of 0·0006 inch. The new product has a nominal fibre diameter of 0·000125 inch. The new fibre product, therefore, has a cross-sectional area 1/23rd the cross-sectional area of the old, which means that the improved product offers twenty-three times the fibre length from a pound of glass as is offered by the old product. Converted to fibre length per cu. ft. of volume, considering the variation in density of the two products, this amounts to 8·2 times the fibre length offered in the new product per unit of volume, as compared with that offered in the old product. This greater fibre length, through smaller fibre diameter, permits each single fibre to be bonded to neighbouring fibres at many more points along its length. Also, smaller diameter fibre means more fibres per pound of weight and, therefore, again more bonds between fibres. These refinements in fibre diameter and fibre length offer advantages of greater tensile strength, resiliency, puncture resistance, and tenacity.—*Marine Engineering, October 1954; Vol. 59, pp. 65-66.*

Sand Blasting and Painting Costs Reduced

Tests have been carried out at the Philadelphia Naval Shipyard on a mobile staging unit for use when sandblasting and painting ships' hulls in dry dock. The unit consists of a petrol-driven tractor fitted with an air compressor and a jointed boom, at the other end of which are two buckets each capable of holding a man and his equipment. The boom can be rotated through more than a full circle, and the workmen may be positioned at any point above the dock floor up to a height of 37 feet. Controlled tests were carried out on the hulls of three vessels, conventional staging methods being used in preparing and painting the starboard sides of the three hulls, whilst on the port sides, two of the new units were employed. The labour cost of painting and sandblasting the starboard side was approximately 25,000 dollars and the comparable cost for the work on the port side was approximately 17,000 dollars; the corresponding times taken were approximately 775 and 475 man-days. In addition to this direct saving, there is the saving of time and expense in erecting and dismantling staging. It is stated that with the mobile unit the boom operator easily learns to anticipate the workers' needs and move the boom accordingly. In addition, the unit provides suitable mobile support for other dry-dock operations, such as caulking, riveting, closing sea chest, and installing scuppers. The air compressor of the tractor is rated at 105 cu. ft. per min. at 100lb. per sq. in.; this capacity is sufficient to operate two paint-spray guns. The rated lifting capacity of the fully extended boom is 500lb.—*Bureau of Ships Journal, 19th October 1954; Vol. 3, p 41. The British Shipbuilding Research Association, December 1954; Vol. 9, Abstract No. 9,715.*

Ceramic Coatings

Radically new and flexible ceramic coatings can now be applied to a wide variety of materials as the result of a process developed at Armour Research Foundation of Illinois Institute of Technology, Chicago, Ill. The new coatings differ from conventional coatings such as porcelain enamel, in that they are not brittle and can be applied to almost any clean solid surface at a few hundred degrees Fahrenheit. Called "solution ceramics" because they are applied from chemical solutions, the new coatings do not contain any adhesive or binding agent. Unusual features of the coatings are at follows: 1. Application is easy, requiring neither expensive equipment nor protected heating. 2. The temperature required by the coating process is far lower than that required for porcelain enamel. 3. The thickness of coatings can be controlled accurately. While films as thick as 0·01 inch can be deposited, some coatings only a few millionths of an inch thick have been made. 4. Solution ceramic coatings are highly resistant to heat, and the more refractory ones can be used to protect a metal, for instance, against molten metal or slag. 5. The coatings are less hard than ordinary ceramic materials and can be scratched with a knife blade. This allows sheet metals coated with solution ceramics to be stamped after coating. 6. Most solution ceramics are so adherent to the surface of the underlying material that other coatings can be anchored to them. 7. Like most ceramic materials, solution ceramic coatings are themselves resistant to chemical attack, even at high temperatures. This is especially evident when the ceramic itself is refractory and stable.—*Mechanical Engineering, November 1954; Vol. 76, pp. 918-919.*

Components of Propulsive Efficiency

In a complete sense of the term the efficiency of screw ships should include consideration of such qualities of the hull form as would ensure the least resistance to self-propulsion. In the modern practice of ship design the information concerning good hull form is abundant, and it is only occasionally that a proposed hull form can be improved, after model tank testing, by more than some 3 per cent. Too frequently, however, when the attempt is made in shipbuilding research to correlate the effective power assessed by an experiment tank to the actual power measured on trial, discrepancies of much more than 3 per cent are found. This paper assumes that effective horsepower for a good hull form has been correctly assessed in an experiment tank, and confines its attention to the ratio, effective power: trial-measured power. By examination of the component factors of this coefficient it attempts to eliminate or at the least to segregate clearly these larger discrepancies. The relationship between trial-measured and effective power is wholly empirical, being founded on observation—during trials or tank experiments—rather than on abstruse theory. The relationship is a matter of great commercial importance to ship and marine engine designers and builders. A large part of the cost of any ship is proportional to the power necessary to propel it. A contract is usually made definite by the builder's promise to propel the vessel at a definite speed; and the commercial success of a firm or of a national group of firms competing for contracts at home and abroad owes much to the skill and enterprise with which designers can estimate precisely the power necessary for self-propulsion. This important commercial consideration makes it proper to examine the practical importance of each component factor. In the last few years the hope of an adequate numerical assessment has been encouraged by the comprehensive and accurate work done by the British Shipbuilding Research Association on the correlation of tank and trial data. The general aim of the Association is to provide ship and machinery designers with data and information that will enable them to produce better designs. Since the Association does not do design and development work it is evident that the successful use of the data will depend upon the designers themselves. In order to facilitate this task the author has undertaken to define precisely the correlation of tank and trial data.—*Paper by G. Johnson, read at a meeting of the Institution of Engineers and Shipbuilders in Scotland, 30th November 1954.*

Net Winches for Trawlers and Luggers

In recent years the electric drive for net winches of up to 300 h.p. has gained increased importance in modern fishing vessel design. The great degree of adaptability of the electric winch drive has enabled the net hoisting gear and the screw drive to be combined. Thus in new ships built in recent years the entire mechanical plant, i.e. propulsion unit and electric winch gear, has been merged into a single unit. According to choice, the winch Diesel engine may power either the winch generator or the screw drive, a combination which can be carried out either electrically or mechanically. The most complete merging of the screw drive with the electric net winch drive is achieved with Diesel-electric propulsion. The Diesel-generator units can operate both the screw drive and the net winch (Fig. 1). Altogether four fishing vessels have so far

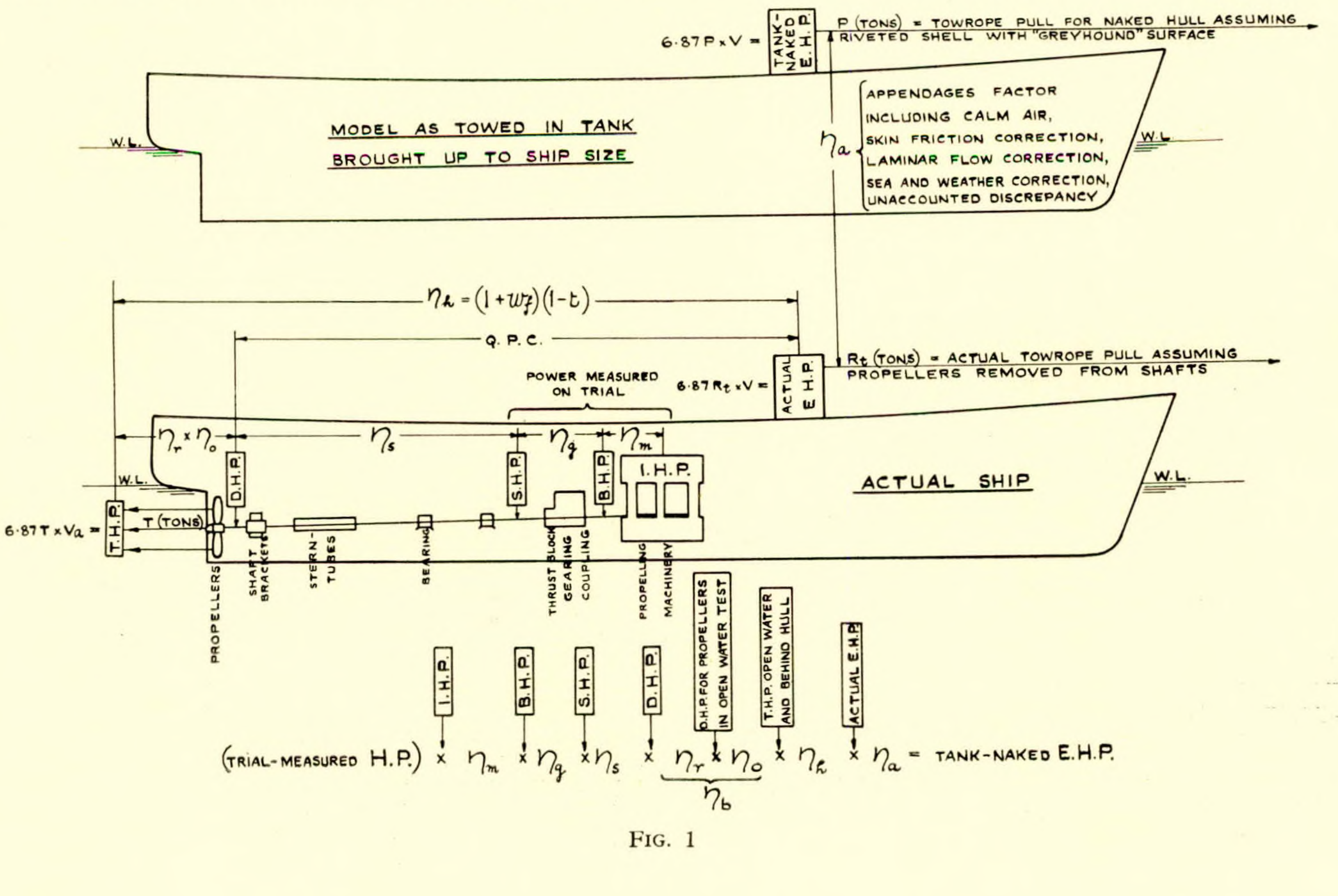


FIG. 1

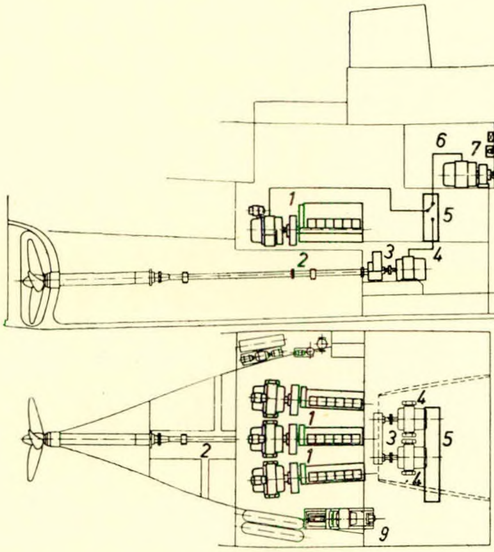


FIG. 1—*Basic circuit diagram of electrically powered trawler*
(1) Main Diesel unit; (2) Screw shaft; (3) Screw transmission gearing; (4) Screw drive motors; (5) Main switchboard; (6) Net winch motor; (7) Net winch controls; (8) Command post bridge; (9) Auxiliary Diesel; (10) Net winch.

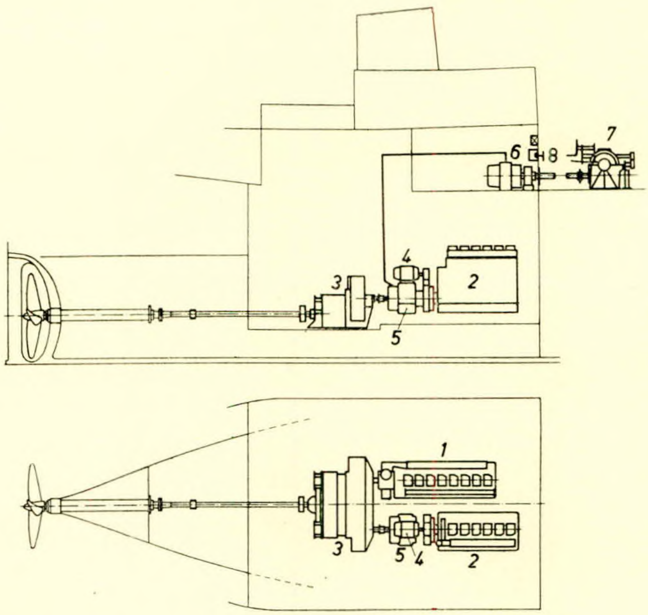


FIG. 2—*Basic circuit diagram of mechanical "Father and Son" installation*
(1) "Father" Diesel engine; (2) "Son" Diesel engine; (3) Reduction gear; (4) Board supply generator; (5) Net winch generator; (6) Net winch motor; (7) Net winch; (8) Net winch controls.

been equipped with Diesel-electric drives of this kind. In fishing circles the term "Father and Son" equipment has been coined to describe the mechanical variation of combining the net winch Diesel with the screw drive. Here the main Diesel engine is referred to as "Father", the winch Diesel as "Son". Both engines drive the ship's screw through a common drive (Fig. 2 shows an arrangement of this type). The winch generator has become part of the screw shaft transmission, a fact which called for special constructional consideration in fitting the generator. The working conditions of the dragnet call for special features on the part of the winch drive. In a swell the trawling lines must always remain taut to prevent tilting of the yawboards. The winch motor speed must, therefore, rise rapidly when the strain on the line decreases, in order to pick up the slack. When fishing in rough seas or if the dragnet fouls the sea-bed, the lines are subject to heavy stresses, and a sharp drop in engine speed is then required in order to protect the winch tackle. These operating conditions had hitherto been met by the inherent characteristics of the steam winch. In designing the electric winch motor, therefore, at least equal advantages had to be provided. By applying the Ward-Leonard system, the desired characteristics may be obtained.—*W. Hollmann, A.E.G. Progress, No. 3, 1954; pp. 217-218.*

Little Supertanker

The *P.C. Spencer* belonging to the Sinclair Refining Company, is a specially designed 604-ft. "little supertanker" of 25,214 deadweight tons. Termed the T-2 tanker of the next decade, the *Spencer* and her sister ship, *M. L. Gosney* represent a distinct and highly developed type of ship construction for the transportation of crude and refined petroleum products. In preparing for the design of the *P.C. Spencer* an intensive study was undertaken of the economic requirements which this new class of tanker must meet. Working in collaboration with the Central Technical Division of the Bethlehem Steel Company, Shipbuilding Division, at the Quincey yard, the Sinclair staff developed a design based on modification of Bethlehem's 28,000-ton supertanker which has been built in considerable numbers. These tankers are 20ft. shorter and have 3,146 tons less deadweight than the Bethlehem supertankers. The *Spencer* has a liquid-cargo capacity of 212,405 bbl (100 per cent full) as compared with the Bethlehem-built 28,000-ton supertanker's 240,000-bbl capacity (100 per cent full). Planned primarily for American coastwise service, the ship's basic design was developed within definite limitations of length, breadth and depth prescribed by her domestic ports of call. However, the company also wanted the vessel to have additional sea qualities and cruising range to ensure economical operation in the event the vessel was required for foreign trade. In design the *Spencer* is a three-island, single-screw, geared-turbine-driven tanker having a curved raked stem and cruiser stern with machinery aft. The amply proportioned outer stack, located well forward on the house aft, and the absence of a forward catwalk and mainmast enhance the outward appearance of the vessel. The hull is constructed on the longitudinal type of framing with plate and angle-type bulkheads to American Bureau and Lloyd's Register of Shipping classification requirements. There are two continuous longitudinal bulkheads 20ft. at either side of the centreline extending from the pump-room bulkhead to the forward cofferdam. These two oiltight longitudinals together with eight transverse bulkheads divide the cargo-oil space into twenty-seven tanks which have a total capacity of 212,425·70 bbl (100 per cent full). The *P.C. Spencer* is powered by Bethlehem impulse-reaction, high-pressure steam turbines delivering 12,500 normal and 13,750 maximum h.p. at propeller speeds of 100 and 103 r.p.m., respectively. This is the same type of propulsion machinery that has been installed in Bethlehem's 28,000-ton supertankers. Two Foster-Wheeler two-drum water-tube boilers are fitted with superheaters, desuperheaters, economizers, steam-type air heater, steam soot blowers; combustion controls, feedwater regulators and water-walled furnaces are provided. Sinclair's marine staff departed from the conventional machinery arrangement plan and placed the vessel's boilers forward of the main engines. With the boilers in this location, the numerous boiler auxiliary units are more accessible, thereby facilitating better inspection and easier repair by the engine room personnel. A substantial reduction in the amount of line shafting required, also was realized by this arrangement. An electrically operated, General Regulator automatic-combustion control system is provided. The control instruments are grouped on a single panel which is located in the boiler room and is visible from the manœuvring platform. The main condenser is of the horizontal, two-pass, reheating, surface type and has 13,000 sq. ft. of cooling surface. Two auxiliary condensers, of the same type as the main unit, are provided. one for each turbo-generator. Two turbo-generator sets are provided; and each unit consists of a De Laval 6-stage, impulse-type steam turbine driving a Westinghouse 400-kW., 450-volt A.C. generator through a single, double-helical reduction gear. The vessel also is provided with a six-cylinder, 4-cycle Cummins supercharged Diesel engine which drives a 75-kW. A.C. emergency generator. This unit starts automatically on loss of power from the vessel's turbo-generator sets.—*Marine Engineering, December 1954; Vol. 59, pp. 64-65, p. 93.*

Repairing Modern Tankers

The increase in the size of tankers has brought the ship-repairing industry face to face with important problems relating to large drydocks and, in addition, the necessity for complementary services at the drydocks and wharves and at wet berths, as well as in the workshops, where heavier items of machinery from vessels under repair are handled. The usual services provided in repair yards, such as cranes, ballast water pumps, compressed air and electric current, etc., must equally be matched up with the magnitude of the drydocks to ensure that the time factor in turnround can be maintained at an absolute minimum. The drydocks must be capable of accommodating and sustaining the weight of the ships together with very considerable amounts of ballast water utilized in testing cargo tanks. It must be possible to deliver large quantities of water rapidly to the ship for testing purposes and subsequently, when this is drained into the drydock, to dispose of it equally quickly. The cranes must be capable of lifting heavy parts of the hull of the ship as well as heavy parts of machinery and it should be noted that the propellers of these larger tankers weigh up to 33 tons and rudders 43 tons, while main gearwheels weigh as much as 61 tons. With their high-pressure boilers and propelling machinery, basically in the same form as a modern shore power station, there are considerable auxiliary services and equipment in the engine room which must be kept running as long as there is steam being generated in the boilers. In order to carry out repairs and maintenance to the machinery, it is essential that the whole of this plant be shut down and the maintenance of services to the ship under these conditions calls for considerable quantities of both electric current and shore steam in the drydock and at the repair berth. Because of the high pressure and high superheat developed, a large amount of auxiliary equipment is installed in the boiler and machinery spaces and the number of auxiliaries is much greater than hitherto. All these factors call for specially skilled labour in repair yards, together with the additional plant which can deal with all the modern equipment to ensure efficient and quick repairs. From the electrical aspect the trend again is to have an increasing amount of electrical equipment for the operation of machinery and for aids to navigation. Additionally, a great many of these newer vessels have talking-picture equipment, laundries and washing machines and other crew amenities installed. The recent International Convention on Oil Pollution has called for reception facilities for dirty ballast water to be available at certain ports and in order that the minimum of time shall be lost in obtaining gas-free certificates to enable repairs to proceed quickly, oily ballast water separating plant, together with hot water washing facilities and tank cleaning machines, are becoming yet another adjunct to the many facilities to which the modern ship repair yards must give attention. The increasing application of welding as applied to the structure of the hull is one of the changes which has come about in shipbuilding technique today. In the comparatively early days of the modern tanker the problem of wastage and deterioration brought about by the constant carrying of crude oil, which applies not only to shell plating but to heating coils and other internals in cargo tanks, may not be apparent. However, as classification surveys become due this problem must also be met by the shiprepairer in the provision of extended welding facilities and an adequate highly skilled labour force. It is true to say that because the tanker is modern the question of what may be required in the way of repairs and maintenance has hardly yet been resolved, but from what has been revealed as these vessels come into the picture it is apparent that only those ship-repairers who are prepared to enlarge and develop their organizations to deal with the new situation can hope to be favoured with the repair and maintenance of the modern tanker.—*H. A. J. Silley, The Shipping World, 12th January 1955; Vol. 132, p. 31.*

The "White" Oily-water Separator

The "White" oily-water separator, developed by White's Marine Engineering Co., Ltd., is designed to provide a method of dealing with oily bilge and ballast water, and can either be provided with automatic control or arranged for manual operation. The principle of separation is as follows. The oil and water mixture to be separated is pumped direct through the mixture inlet valve and enters the primary separation chamber through vertical apertures in the distributor. 90 to 95 per cent of the oil in the mixture is separated in the primary chamber, the oil rising and collecting in the upper section. The remaining mixture passes down through the primary chamber and because of the convergent passage, the oil is compacted into larger droplets and progressive separation occurs. The mixture then passes into the secondary chamber via compacting nozzles, where a similar separation action takes place, the mixture rising to pass through further compacting nozzles before entering the tertiary chamber containing the filter bed. As the mixture changes direction to enter the filter bed, the small amount of compacted oil remaining is released and rises through passages into the top section of the primary chamber. The filter bed mentioned above performs an important function by removing the small percentage of oil, or grit having a coating of oil, still remaining in suspension in a very finely divided condition. The filter consists of specially graded material down through which the discharge water passes and where all traces of oil contamination are removed. The makers claim that tests on an oily-water mixture ranging from 25 per cent oil and 75 per cent water to 75 per cent oil and 25 per cent water gave results with a residual of oil in the water amounting to only 1·7 parts per million. With the automatic control system the control of the oil and water discharge is achieved electronically, with electric motor operation of the valves. The electronic method of control is completely independent of the specific gravities of the liquids involved, and therefore no adjustments are necessary on change of specific gravity. For this type of control the electrostatic properties of the liquids are utilized. Because of the low voltages and powers applied to the probes, there is no danger of spark or fire hazard to the oil in the separator.—*The Steam Engineer, December 1954; Vol. 24, pp. 117-118.*

Norwegian Cruising Vessel

The Aalborg Vaerft shipyard, Denmark, have very recently delivered the passenger and mail motorship *Meteor* to her Norwegian owners, Det Bergenske Dampskibsselskab (The Bergen Line), Bergen, who will use her for cruises and as a relief ship on their other services. She has been built to Det Norske Veritas classification, and is generally electrically welded. The deckhouses on the boat deck and the bridge have been constructed in aluminium. Principal particulars of the *Meteor* are as follows:—

Length overall	296ft. 8in.
Length b.p.	291ft. 7in.
Breadth	45ft. 0in.
Depth to upper deck	24ft. 6in.
Load draught	16ft. 0in.
Gross tonnage	2,850 tons
Passengers, cruising	157

The insulated cargo hold has a total capacity of 18,000 cu. ft. and is capable of being cooled down to −20 deg. C. The hatch is served by two 1·75-ton and one 3-ton electric cranes, all supplied by Thrige, of Odense. Both the bipod type foremast and the main mast have been fabricated from aluminium. There are six aluminium lifeboats, one with motor, and with a boat winch for each boat. On cruises, a motor-driven barge for passengers' use replaces one lifeboat. The propelling machinery consists of a nine-cylinder B. and W. single-acting two-stroke, trunk Diesel engine, with exhaust gas turbo charging. The engine develops 5,000 b.h.p. at 200 r.p.m., giving the *Meteor* a speed of a little over 15½ knots. She is fitted with a reversible propeller, controllable from the wheelhouse. The auxiliary machinery comprises three six-cylinder Diesel engines, each developing 300 b.h.p. at 425 r.p.m., and supplying 200 kW.—*Shipbuilding and Shipping Record, 27th January 1955; Vol. 85, pp. 111-114.*

Fast Twin-screw Motorship

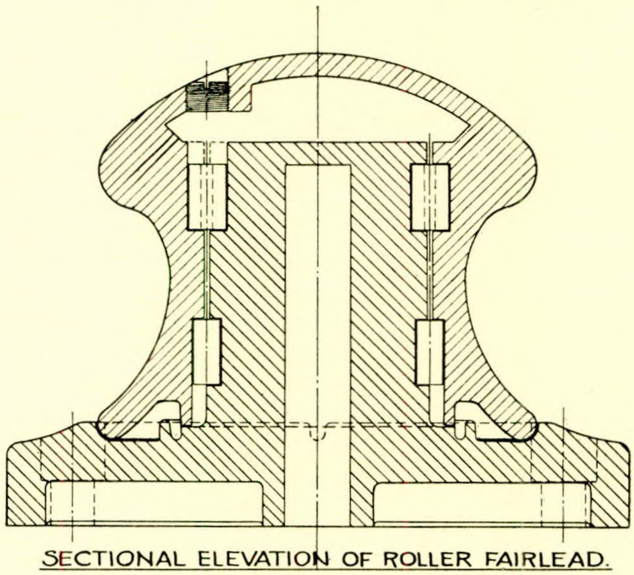
The twin-screw motor ship *Canopic* was recently delivered from the Walker yard of Vickers-Armstrongs, Ltd., to the Shaw Savill and Albion Co., Ltd., London. This is a high-class refrigerated cargo vessel of similar construction to the *Cymric,* built in 1953, and the *Cedric,* built in 1952. The *Cretic,* also a sister ship to the *Canopic,* was launched on 25th January 1955, from the Wallsend shipyard of Swan, Hunter and Wigham Richardson, Ltd. All these vessels have been designed for a speed of 17 knots in service. The *Canopic* is powered by two Harland and Wolff type engines of 14,300 s.h.p. and attained a speed of 18½ knots on her trials. The principal particulars of the *Canopic* are as follows:—

Length o.a. (about)	512ft. 0in.
Length b.p.	481ft. 0in.
Breadth moulded	69ft. 0in.
Depth moulded to upper deck ...	33ft. 4½in.
Draught loaded	30ft. 4½in.
Deadweight	11,390 tons
Gross tonnage	11,166 tons
Net tonnage	6,350 tons
Horsepower	14,300 s.h.p.
Speed in service	17 knots
Cargo capacity:	
Insulated cargo	348,280 cu. ft.
General cargo (bale)	308,635 cu. ft.

The construction of the *Canopic* has been carried out under Lloyd's Register of Shipping special survey for classification 100A, and also complies with the latest Factory Acts, Ministry of Transport Regulations and the International Convention for the Safety of Life at Sea. The design incorporates a long bridge and topgallant forecastle and a well raked stem. The stern is cruiser type with a semi-balanced streamlined double plate rudder. The shelter and upper decks extend continuously the full length of the ship, the main deck is worked at different levels, and a lower deck is arranged in Nos. 2 and 3 holds and aft. The cargo space is divided into six holds, four of which, together with their lower and main tweendecks, are insulated. The main propelling machinery consists of two Harland and Wolff two-stroke opposed-piston oil engines having a bore of 750 mm. and a combined stroke of 2,000 mm., designed to develop 14,300 b.h.p. (max.) at 118 r.p.m. They are designed to burn boiler fuel of about 1,500 sec. Redwood viscosity and there are two heavy fuel oil purifiers and two clarifiers, each having a capacity of 1,500 gall. per hr., which have been supplied by the Alfa Laval Co., Ltd. Two Alfa Laval Diesel fuel purifiers, each of 250 gall. per hr. capacity are also installed and, in addition, two similar size and make of purifiers are installed for the lubricating oil services. The working propellers are of the four-bladed type of solid manganese bronze supplied by the Manganese Bronze and Brass Co., Ltd. The refrigerating plant, which is arranged in the upper deck to port and starboard of the engine and boiler casings, comprises four 125-b.h.p. 8-cylinder Freon machines supplied by J. and E. Hall, Ltd. This equipment has been designed to deal with the four holds and eight tweendeck spaces, including twelve chilled meat lockers, having a total capacity of 348,280 cu. ft. for the carriage of meat and/or dairy produce and/or fruit. It also deals with the ship's provision chambers in the shelter tweendeck, and two low temperature chambers at the after end of No. 3 main tweendeck. These latter chambers are so arranged that when low temperature produce at 0 to 5 deg. F. is not being carried they are suitable for the carriage of goods at 15 deg. F. and above. The method of cooling is by means of brine cooling batteries and fans.—*The Shipping World, 9th February 1955; Vol. 132, pp. 196-199.*

Improved Warping Roller

One of the problems of warping rollers and roller fairleads is that of keeping them free to rotate. They are nearly always installed in a position on deck where they are exposed to salt air and water, which often causes them to become seized up. An improved type of roller has been designed by Mr. G. W. Rooney, of Goole, Yorkshire, and is known as the G.W.R. warping roller. This is a watertight roller which cannot be damaged unless by accident or misuse, thus saving the expense of having the rollers freed when the vessel comes into dock—a not infrequent occurrence. The object of the design is that the roller should be so constructed that little, if any, attention is necessary, allowing it to be left for long periods without requiring maintenance. It is also claimed that it is competitive in price with the conventional type of roller. The design is simple. The roller consists of a large rotating head revolving on rollers mounted on the pillar. The centre post, or pillar, which is mounted on the sole plate, is grooved to take an upper and a lower set of small mild steel or bronze rollers. The larger roller is grooved to register with the grooves on the pillar; the simple method of assembly eliminates bolts, nuts and washers, and once it has been lubricated the unit will remain free without further attention. Several of these rollers have been fitted to vessels now at sea with highly satisfactory results.—*The Shipping World, 29th December 1954; Vol. 131, p. 685.*



The G.W.R. warping roller

Atomic Ship Model

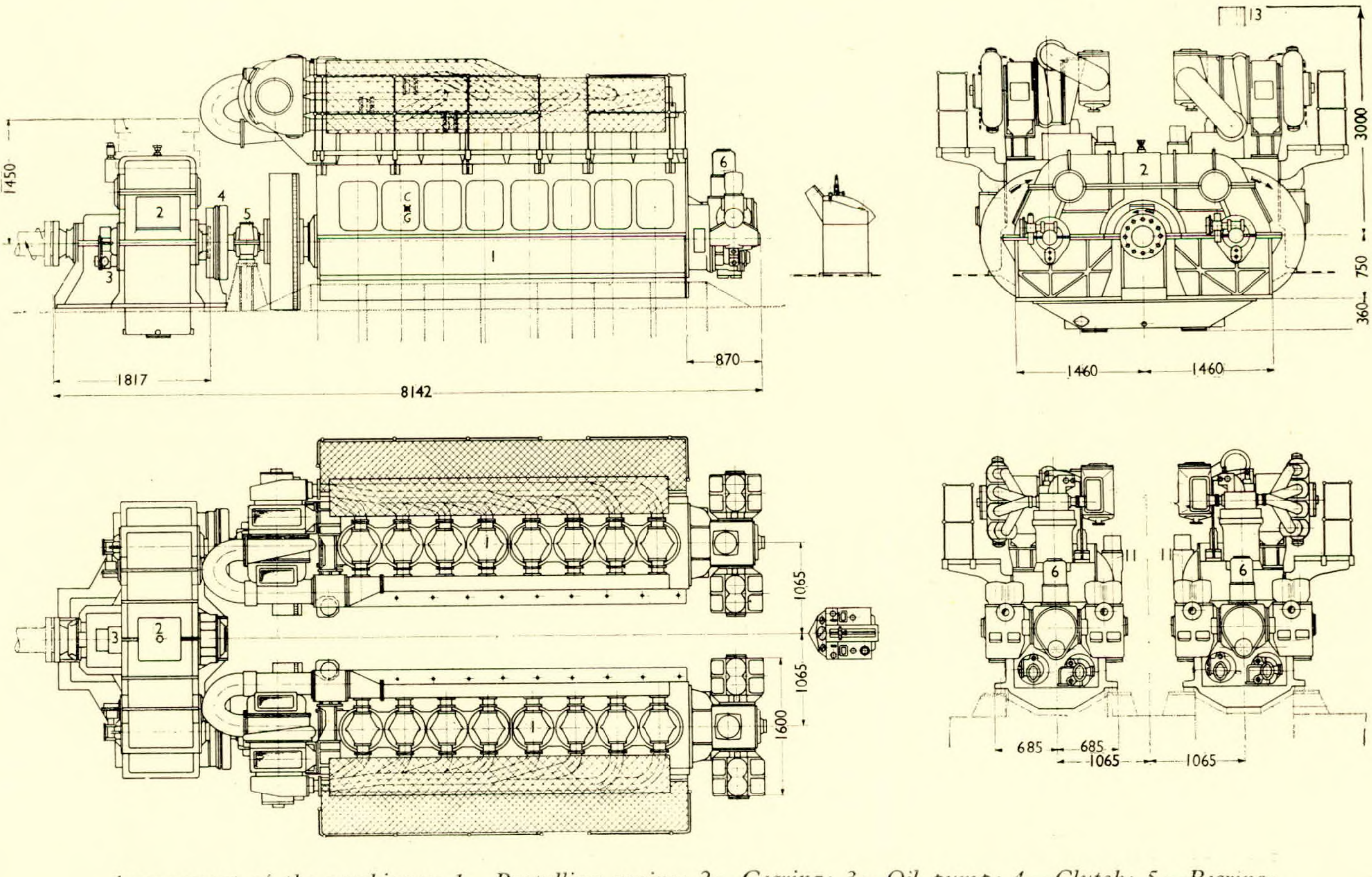
A scale model showing the application of nuclear power for propulsion of a cargo vessel, has been built by the Newport News Shipbuilding and Dry Dock Company, at the request of the United States Information Agency. The model is a replica of the latest and fastest type of cargo vessel, the 560-ft., 20-knot Mariner Class. With the use of atomic power, such a ship could stay at sea for years without refuelling. Instead of a firebox and boiler, the ship's atomic power plant would have a reactor which would generate and change water into steam. Machinery such as steam turbines, pumps, condensers, etc., would be the equivalent of those on a normal ship. The engine and boiler rooms on the atomic-propelled ship would require about the same space as those of a normal ship. However, some additional cargo space would be created by elimination of fuel-oil tank within the ballasting requirements of the ship. The outstanding difference in the appearance of an atomic-powered vessel would be the absence of a smokestack. Nuclear power would create no smoke or soot.—*Mechanical Engineering, December 1954; Vol. 76, p. 1,011.*

Free-piston Gas-turbine Prime Mover

The free-piston turbine combination consolidates the principal assets of the opposed-piston combustion cylinder with the good points of the gas turbine, cancelling out a large number of the wearing parts, such as the crankshaft and the connecting rods, in converting linear motion to rotating motion. Crankshaft engines have attained 40 per cent thermal efficiency. The all-rotating gas turbine with its complexities of intercooling, reheat and regeneration, and multiple components in high and low pressure compressors with high and low pressure turbines, have not equalled the crankshaft-engine efficiency. The simple and inexpensive free-piston unit attains the lowest number of reciprocating and wearing parts and has the efficiency of the piston crankshaft-type engine together with the turbine as the driving medium. It operates much as a steam plant with a turbine driving a load by pumping hot exhaust gas under pressure from one or more gas generators as boilers deliver their steam, but without the necessity for using an intermediate medium like steam between the source and the turbine. The free-piston turbine runs on the actual exhaust combustion gases without the heat having to pass into steam, which has to be superheated, condensed, and returned. The exhaust gases serve their purpose directly in the free-piston turbine where the network energy is extracted. There is heat remaining in the turbine exhaust which may be regained for auxiliary uses. The greatest outstanding attribute of the free-piston engine is its essential simplicity. The elements of its design, which are attached to the free-piston principle, are absolutely trouble-free. For instance, the mechanism that keeps the pistons synchronized has never been known to give any trouble to any free-piston unit in service or on test. The pneumatic controls to regulate the stroke are taken from railroad freight-car braking systems and are simple and trouble-free.—*Paper by R. P. Ramsey, read at the 1954 A.S.M.E. Petroleum Mechanical Engineering Conference, Paper No. 54-PET-23.*

French Cargo Vessel with Geared Diesel

The *Ville de Montréal,* which was completed by the Amsterdamsche Droogdok Mij. N.V., Amsterdam, in December, is for the Great Lakes service of the Cie. Générale d'Armements Maritimes, a subsidiary of the Cie. Générale Transatlantique. She is of 2,800 tons deadweight, with a speed of 13 knots and is equipped with two S.L.M. eight-cylinder four-stroke supercharged Diesel engines, coupled to a common S.L.M. oil-operated reverse and reduction gearbox driving on to a single propeller shaft. The general arrangement and the layout of the machinery is shown in the accompanying plans. The normal speed of each engine is 475 r.p.m., the rated output at this speed being 1,250 b.h.p., but the engines have an overload capacity up to 1,375 b.h.p. at 490 r.p.m. for two hours. The b.m.e.p. at full load is 8·5 kg. per sq. cm., or 121lb. per sq. in., whilst at the overload the corresponding b.m.e.p. is 9 kg. per sq. cm., or 128lb. per sq. in. The engines are of the non-reversible type, and fitted with a Brown Boveri exhaust turbo-charger with



Arrangement of the machinery: 1.—Propelling engine; 2.—Gearing; 3.—Oil pump; 4.—Clutch; 5.—Bearing; 6.—Air compressor an after cooler. Each engine drives a starting air compressor, sea-water circulating pump, bilge pump and lubricating oil and fuel pumps. Between the reverse-reduction gear and the engines are semi-flexible couplings. The gear is fitted with a built-in Michel thrust block bearing taking care of the propeller thrust, also built-on oil pressure pumps, oil filters, oil coolers and pneumatically operated control cocks. The forward speed reduction ratio is 1:3·812, giving a propeller speed of 125 r.p.m. at full load. The astern reduction gear is practically the same. The manœuvring controls are common for both engines, including a control drive for the reverse-reduction gear and a locking arrangement for operation on one engine only. During the course of the trials at the manufacturers' works the following fuel consumption figures were attained:—

Load	B.H.P.	Consumption per b.h.p. per hr.
Half	622	166 gr.
Three-quarter	937·5	162 gr.
Full	1,250	165 gr.
Overload	1,375	166 gr.

At full load the temperature of the exhaust gas before the turbine was 475 deg. C. and the temperature at the manifold 392 deg. C. The consumption figures given above refer to a speed of 374 r.p.m., 432 r.p.m., 475 r.p.m. and 490 r.p.m. respectively. The speed of the turbo blower at full load is 10,500 r.p.m. The blower pressure is 1·337 kg. per sq. cm.—*The Motor Ship, November 1954; Vol. 35, p. 332.*

First Japanese Gas Turbine Ship

The first Japanese gas turbine for marine use was installed in the training ship *Hokuto Maru* in September 1954, and has since been operating in a satisfactory manner. With the help of a subsidy from the Japanese Ministry of Transport, the construction of a 500 h.p. gas turbine was started by Mitsubishi Zosen in July 1952, for the study of applied technology. In December 1953, it was reported that the gas turbine had been running continuously on full load, and on 11th February 1954, official shop trials took place in the presence of persons concerned in the various fields of science and industry. Some modifications were carried out in the April of the same year in order to improve the output of the turbine, after which it was tested thoroughly on the test bed so that its endurance could be checked. The vessel in which this gas turbine is now installed, the *Hokuto Maru,* is a training ship of 1,617 tons displacement, built at the Fujinagata shipyard in 1952 for the Government of Japan (Navigation Training Establishment). The vessel is equipped with a steam turbine and has run at a service speed of 12 knots. The purpose of installing the 500 h.p. gas turbine was to operate it separately from the steam turbine so that the ship would run at a speed of about 8 knots on the gas turbine alone. The change over from the steam turbine to the gas turbine is effected by means of a clutch. The gas turbine operates on the two-shaft open cycle and is fitted with a conduction-type heat exchanger. The principal details of this engine are as follows:—

Output at turbine coupling	450 h.p. at 20 deg. C.
Maximum output	500 h.p. at 20 deg. C.
Pressure ratio...	3·5 : 1
Temperature at turbine inlet	65 deg. C.
Fuel consumption	0·858lb. h.p. per hr.

During the sea trials of the *Hokuto Maru* at the beginning of October 1954, the following data was recorded:

Speed	9·605 knots
Shaft revolutions...	105 r.p.m.
Output at turbine coupling ...	403 s.h.p
Pressure ratio	3·08 : 1
Gas temperature	654 deg. C.
Output (theoretical)	403 h.p.
Atmospheric temperature ...	24·6 deg. C.
Fuel consumption	1·05lb. h.p. per hr.

The above output of 403 h.p. has been determined taking into consideration a presumed loss of 80 h.p. due to the reduction gear and the idling of the steam turbine, the calculation of the loss having been assumed at the time of analysing the data recorded during sea trials.—*The Shipping World, 19th January 1955; Vol. 132, p. 124.*

The Prevention of Pitching

The stabilization of ships against rolling having been successfully accomplished, attention is now being turned towards achieving similar success in preventing, or at least reducing, pitching. For over a year now experiments have been conducted by the Holland America Line in this field, the idea being to reduce the pitching of a ship by means of fins installed right forward in the ship. The fins are presumably similar in character to those used in the Denny-Brown equipment for the reduction of rolling, but arranged so that the two fins tilt together instead of in opposite directions, following in this respect the hydroplanes of a submarine. After experimental work involving Dutch shipbuilders and the Dutch shipbuilding research institute at Wageningen, trial equipment was installed in the liner *Ryndam.* The ship sailed from Rotterdam, and it appears that on passage to Southampton the gear operated very successfully. On leaving Southampton, however, the *Ryndam* ran into heavy head seas, and the vibration was such that she had to put back to Southampton, where the gear was removed. The forces needed to correct pitching are, of course, very much larger than those associated with rolling, and it is not surprising that problems of a practical nature have been encountered. Since pitching can induce sea sickness as effectively as rolling, further results of this work will be awaited with interest.—*The Shipping World, 2nd February 1955; Vol. 132, pp. 159-160.*

German Built Research Vessel for Argentine

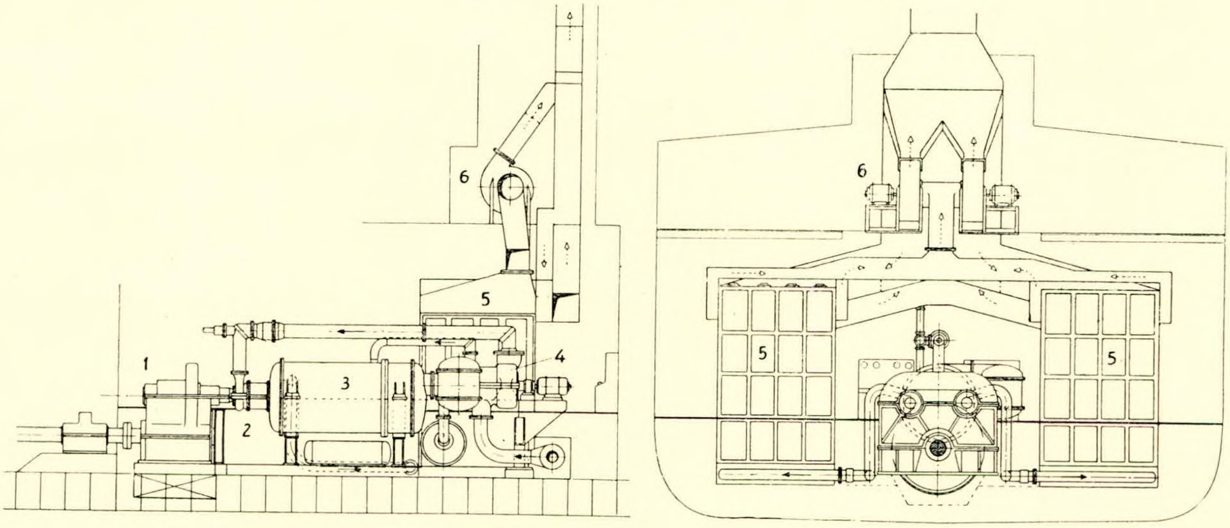
Built for service as a research and supply ship in the Antarctic, the *General San Martin,* constructed by the A.G. Weser, Werk Seebeck, for the Argentine Government, is equipped with Diesel-electric machinery of 6,500 b.h.p. The vessel is 84·70 m. in length overall with a beam of 19 m., and has a gross tonnage of 3,640·37. There are two engine-rooms and in each is a nine-cylinder four-stroke M.A.N. engine with cylinders 400 mm. in diameter and a piston stroke of 600 mm., coupled to a dynamo, the current being supplied to two A.E.G. electric motors driving the propelling shafts at 138 r.p.m. The survey work will involve service in waters in which there is much heavy ice, and for this reason arrangements have been made to control the vessel from the crow's nest, all the engine controls being taken to that point. The ship can also be steered and manœuvred from the wheelhouse in the ordinary manner.—*The Motor Ship, January 1955; Vol. 35, p. 432.*

Hungarian River Floating Dock

A new floating repair ship has recently been moored in the Danube, at Budapest, for the Hungarian River Transport Authorities. When the river is not ice-bound, the ship can be towed, at short notice, to vessels in need of repair. The craft is of massive barge construction, and is equipped with a series of workshops to deal with welding, metal-cutting, machinery repairs, electrical repairs and wood-working. Power for machinery, lighting and heating is supplied by the repair ship's own generators. All the workshops have insulated walls, to reduce the transmission of noise. Accommodation is provided for a permanent crew, and includes offices, stores, sleeping and dining quarters, a kitchen, a bathroom and showers, while a system of central heating is fitted. To assist in communicating with ships in need of help, there is radio apparatus, and an internal telephone installation is also provided.—*The Shipbuilder and Marine Engine-Builder, February 1955; Vol. 62, p. 120.*

Swiss Closed Cycle Air Turbine

A paper by Keller and Spillmann read at the annual meeting of the Schiffbautechnische Gesellschaft at Hamburg gives information on progress made in the application of the



Hot air turbine plant for 8,000 s.h.p. single screw drive vessel

1. Reduction gear.
2. Power turbine.
3. Heat exchanger.
4. Turbo compressor.
5. Air heater.
6. Combustion air blower.

Escher Wyss closed cycle air turbine system. The first pulverized fuel fired plant of this type with an output of 3,000 h.p. will be placed in commission during the summer of 1955 in an industrial power plant in Southern Germany. Plants for Scotland and Japan are in process of manufacture. For marine propulsion plant with powers up to 12,000 s.h.p. closed cycle air turbine plants without reheating are proposed. These plants are to be used in conjunction with variable pitch propellers. The fuel oil consumption is estimated at 210 gal. per s.h.p. hour. Unit plant weight is given as 16 kg. per s.h.p. It is expected that with the use of radial turbines, unit plant weight can be reduced to 12 kg. per s.h.p. Expected service life is 50,000 to 100,000 hours with turbine inlet temperatures of 680 deg. C. A typical plant for an 8,000 s.h.p. single screw drive equipped with two air heaters operating in parallel is shown in the accompanying illustration.—*Schiff und Hafen, December 1954; Vol. 6, pp. 772-773.*

Large Norwegian Cargo Ship

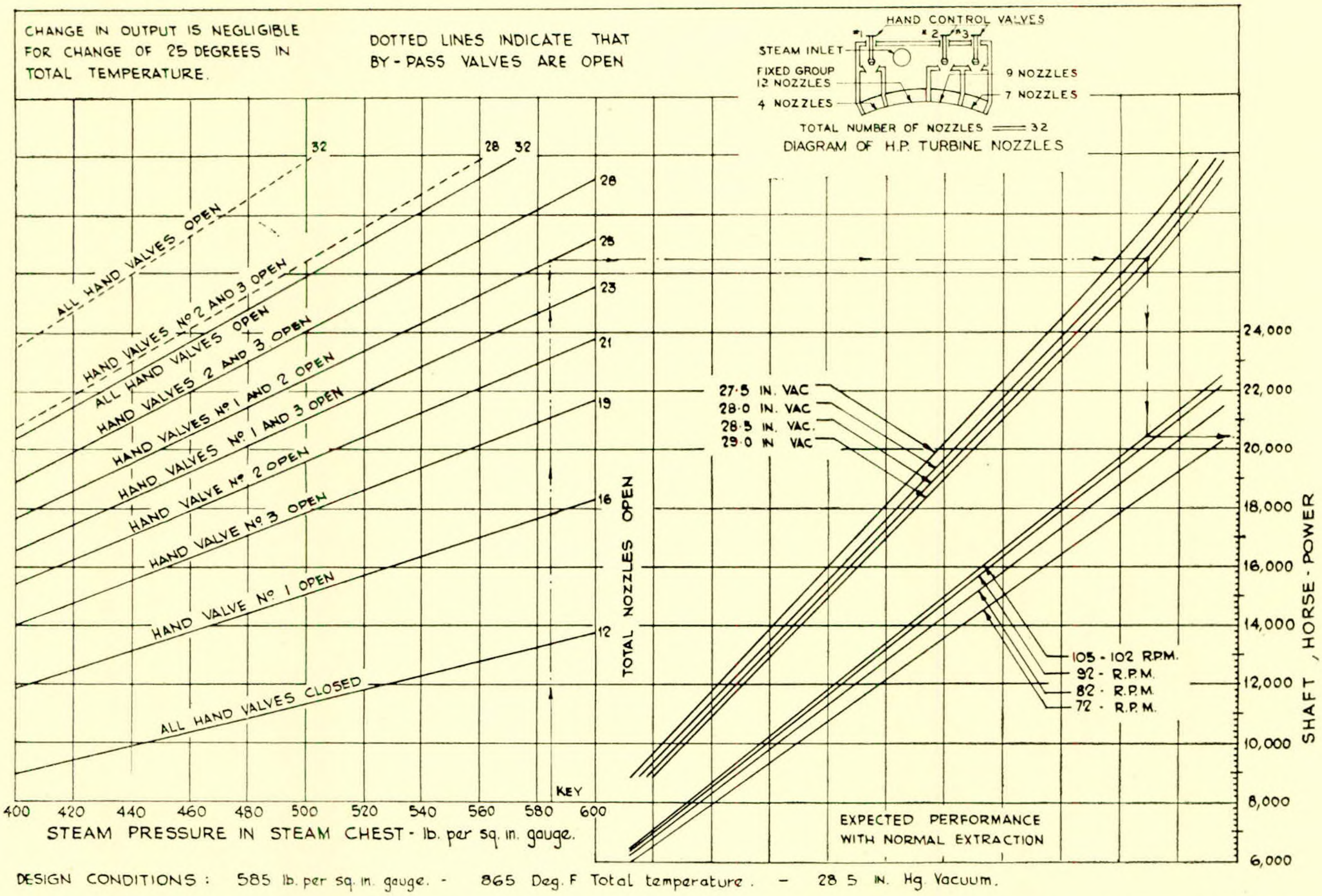
The *Jakara* is being built for Anders Jahre and Co. A/S (A/S Kosmos), Sandefjord, Norway, by Oresundsvarvet Aktiebolag, Landskrona, and the following are the main characteristics:—

Deadweight capacity	12,300 tons
Cargo capacity, grain, about ...	628,000 cu. ft.
Length overall	483ft. 6in. (147·37 m.)
Registered length...	458ft. 5in. (139·73 m.)
Breadth, moulded	59ft. 0in.
Depth, moulded to upper deck...	35ft. 6in. (10·82 m.)
Depth, moulded to second deck...	27ft. 8in. (8·42 m.)
Draught on summer freeboard, about	27ft. 6in. (8·38 m.)
Machinery...	6,200 i.h.p.
Speed, fully loaded	13½ knots

Welding has been employed exclusively throughout the construction and the *Jakara* is being built to the highest class of Det Norske Veritas, being designed for the occasional carriage of ore cargoes. She has one continuous deck, a forecastle deck and a long poop deck. There are five cargo holds (all with 'tween deck spaces), two being aft and three forward of the 'midship house. For hold No. 5 there are two hatches, but each of the other holds is served with one hatch, those on the weather deck forward being fitted with MacGregor steel covers. The double bottom extends the whole length and is utilized for water ballast or fresh water, fuel oil being carried in tanks under the engine-room, where also fresh water, cooling water and lubricating oil tanks are arranged. The remainder of the fuel is accommodated in wing tanks. The total amount of ballast in the double bottom tanks is 2,092·6 tons, and the maximum water ballast in all tanks, 3,418 tons. The bunker capacity totals 715·4 tons. A deep tank is arranged to port and starboard, between Nos. 3 and 4 holds, whilst there are, in addition, two tanks at the sides. The total volume of all tanks is 3,780 cu. m. or about 134,000 cu. ft. The Götaverken two-stroke single-acting propelling engine is a standard unit with seven cylinders, having a diameter of 680 mm. and a piston stroke of 1,500 mm. The speed is 112 r.p.m. and the normal output at these revolutions is 6,200 i.h.p. In this type of engine the under side of the piston is employed for the supply of scavenging air, augmented by separate plunger pumps driven from each of the crossheads. The fuel consumption is 0·35lb. per b.h.p.-hr., and the daily fuel consumption of the ship for all purposes will probably be under 20 tons. The engine is designed to operate on boiler oil and three Titan NS66 self-discharging centrifugal purifiers are installed.—*The Motor Ship, January 1955; Vol. 35, pp. 442-444.*

22,000 s.h.p. Single-screw Tanker

The *W. Alton Jones* is the first American-built tanker designed for the transport of crude oils to be fitted with Cargocaire dehumidification equipment for the prevention of tank corrosion. A smaller tanker thus fitted but for lubricating oil cargoes is owned by the Cities Service Company. The equipment is of the Water Winner type, using a liquid absorbent, triethylene glycol, which has an intense affinity for water and here again the ship is the first tanker to have this plant. She is also the largest vessel to be equipped with Cargocaire. The main machinery consists of a 22,000 s.h.p. single-screw set of cross-compound double-reduction geared turbines built by the De Laval Steam Turbine Company of Trenton, New Jersey. The h.p. turbine is of all-impulse pattern, having eleven stages. The l.p. turbine is also of all-impulse type, having eight stages; the last two rows of moving blades have anti-erosion shields welded to their leading edges. A two-row Curtis wheel followed by one single stage is arranged on the forward end of



Performance chart of the 22,000 s.h.p. De Laval machinery, showing powers developed with various nozzle combinations and steam conditions. The arrowed line indicates how the chart should be used

the l.p. rotor for astern running. Deflection baffles are fitted behind the last moving rows of ahead and astern blading. The l.p. rotor weighs 6½ tons and is of the semi-built-up type with only the astern and last two forward discs solid with the shaft. The gearing is of the double helical articulated type and the teeth have been treated by a shaving process after hobbing. Dehydrators are fitted to extract any water vapour in the gear cases. There are three extraction points for feed heating and the maximum permissible extraction flows are given below. The ahead turbines are provided with hand-controlled nozzle valves and bypass valves which allow the unit to obtain a relatively flat water rate characteristic over the range 12,000 to 22,000 s.h.p. The propelling machinery is designed for the following operating conditions:—

	Normal	Maximum Control
Power at propeller, s.h.p.	20,000	22,000
Propeller speed, r.p.m.	102	105
Initial steam conditions :		
Pressure, lb. per sq. in.	585	581
Temperature, deg. F.	865	865
Condenser vacuum, in. Hg.	28·5	28·0
H.P. turbine speed, r.p.m.	5,448	5,608
L.P. turbine speed, r.p.m.	3,279	3,375

The astern turbine is designed to develop at least 80 per cent of the normal ahead torque when operating astern at 50 per cent of the normal ahead power when exhausting to a vacuum of 27·5in. Hg. Both these conditions are based on supplying steam to the turbine throttle at 585lb. per sq. in. and 865 deg. F. but the turbines have been designed to operate continuously with a receiver steam pressure of 620lb. per sq. in. and a temperature of 875 deg. F.—*The Marine Engineer and Naval Architect (Annual Steam Number), 1954; Vol. 77, pp. 400–403.*

Dutch Built French Cargo Vessels

Built by Amsterdamsche Droogdok-Maatschappij N.V., Amsterdam, the single-screw motorship *Ville de Montréal* has recently carried out satisfactory sea trials, and has been handed over to her owners, Compagnie Générale d'Armements Maritimes, Paris. The *Ville de Montréal* has been designed by the builders in close co-operation with the owners, and is the first of two similar vessels suitable for operating between Europe and the North American Great Lakes during the summer season, and for short-sea or Mediterranean trading during the winter. The second vessel—the *Ville de Quebec*—is now under construction by N.V. Scheepswerf en Machinefabriek "De

Bleeder point	H.P 8th stage inlet		H.P. exhaust		L.P 5th stage inlet	
s.h.p.	Flow lb. per hr.	Pressure lb. per sq. in. (abs.)	Flow lb. per hr.	Pressure lb. per sq. in. (abs.)	Flow lb. per hr.	Pressure lb. per sq. in. (abs.)
12,000	460	92·0	0	40·5	8,100	9·0
17,500	650	133·5	9,000	54·2	11,250	11·9
20,000	750	149·0	10,300	60·0	13,600	13·1
22,000	825	170·0	11,300	69·5	15,750	15·1
Maximum permissible extraction flow	7,000	—	12,000	—	17,000	—

Biesbosch", Dordrecht, and is expected to enter service in March 1957. The principal dimensions and other leading characteristics of the *Ville de Montréal* are given below.

Length overall	258ft. 0in.
Length b.p.	249ft. 0in.
Breadth moulded...	42ft. 4in.
Depth moulded to upper deck ...	26ft. 3in.
Depth moulded to second deck...	18ft. 3in.
Gross tonnage	2,388
Net tonnage	1,236
Load draught	19ft. 3in.
Corresponding displacement, tons	4,240
Corresponding deadweight, tons	2,835
Tons per inch immersion at load draught...	21·4
Light displacement, tons... ...	1,405
Light draughts—	
Forward	3ft. 2in.
Aft	11ft. 3in.
Tons per inch immersion at light draught...	18·2
Centre of gravity of light ship above keel	19ft. 4in.
Metacentric height of light ship...	2ft. 10in.
Cargo capacities—	
Bale, cu. ft.	147,010
Grain, cu. ft.	155,916
Maximum continuous b.h.p. ...	2,500
Corresponding engine r.p.m. ...	475
Corresponding propeller r.p.m. ...	125
Propeller—	
Diameter	13ft. 1½in.
Face pitch	11ft. 7¾in.
Disc-area ratio...	0·475
Number of blades	4
Average service speed, knots ...	13

The vessel is of the closed shelter-deck type, and has been built in accordance with the requirements of the Bureau Veritas. The hull is strengthened for navigating in ice. The construction has followed the builders' usual practice, electric welding having been used to a large extent. The hull structure was erected, in large prefabricated sections, in a floating dock. This building method reduced the assembly period to eight weeks. The principal riveted connections are those of the frames to the shell, the longitudinal seams of the shell plating above the bilge strake and the stringer angle. The stern-frame is of all-welded construction. The propelling machinery consists of two non-reversible, four-stroke cycle, eight-cylinder, supercharged Diesel engines, coupled to a common oil-operated reverse and reduction gearbox. The engines and gearbox have been constructed by Société Suisse pour la Construction de Locomotives et de Machines (S.L.M.), Winterthur. With a cylinder diameter of 310 mm. and a piston stroke of 460 mm., each engine has a normal rated output of 1,250 b.h.p. at 475 r.p.m., and an overload capacity of 1,375 b.h.p. at 490 r.p.m. for two hours. The supercharger for each engine consists of a Brown-Boveri exhaust turbo-charger (Büchi system), with a speed, at full load, of 10,500 r.p.m. Fresh-water cooling, in a closed circuit, has been adopted for the main engines; but in an emergency, sea-water cooling can be introduced.—*The Shipbuilder and Marine Engine-Builder, February 1955; Vol. 62, pp. 113-115.*

Gas Turbine Operation at 1,900 deg. F.

Continuous 100-hr. operation of a gas turbine at a temperature of 1,850 to 1,900 deg. F. and speed of 30,000 r.p.m. was revealed recently by Kennametal Inc., Latrobe, Pennsylvania. Using its own specially designed and operated test turbine, Kennametal subjected the turbine parts to the same conditions of temperature and stress which they would encounter in a jet engine operating at full power. This unit also permitted the testing of new component designs required for the optimum use of their super high-temperature carbide, Kentanium. Engineers found little change in the critical parts of the turbine, particularly in the impeller, which is its one moving part. Ordinarily, gas turbines can withstand operating temperatures in the range of only 1,400 to 1,600 deg. F., which greatly limits their power and efficiency. Key to the successful test, according to company metallurgists, was the fact that the critical parts of the turbine were made of Kentanium, a sintered titanium carbide, specially developed to withstand high temperatures and speeds. Their successful results were seen having great implications for the future of gas turbine power plants. Kennametal engineers envisage the day when further development of these Kentanium materials made by modern powder-metallurgy techniques will enable gas turbines to operate at temperatures well in excess of 2,000 deg. F.—*Mechanical Engineering, December 1954; Vol. 76, p. 1,008.*

Heat Transfer in Rotating Air Preheater

Several analytical and experimental studies of the moving surface or rotating-element regenerative heat exchangers have been made. These studies have been based, in several cases, upon steady-state conditions for the heat-transfer surfaces although it has been recognized that the elements were heated and cooled alternately. This paper presents experimental results to show the effect of the plate thickness, for a plate-type matrix for the rotor, and the time of contact of the "hot" and "cold" air on the total heat-transfer rate from the hot air to the plate or from the plate to the cold air. It also compares the experimental results with an analytical procedure for transient flow which is based on the assumption that the plate is alternately immersed in constant-temperature hot air and constant-temperature cold air. In this analytical procedure it is assumed that the stream at exit from the rotor is mixed thoroughly so as to give a uniform temperature of each fluid at exit. It also is assumed that the air temperature for both hot and cold streams varies linearly with length of passage through the rotor. Some conclusions reached are as follows:—Heat-transfer coefficients in the rotating-element heat exchanger do not appear to differ appreciably from those in the stationary-surface type. Speed of rotation lessens the time of contact of the plates with the fluid and affects the plate-temperature change but has no effect on the total heat transferred within the speed range of these experiments. Plate thickness has no effect on the total heat transferred for the same terminal temperatures and fluid velocities. The effect of the plate thickness is in its effect on the weight of the rotor for the same heat-transfer surface, a thin plate being usable for this type exchanger where it would not be structurally acceptable for an exchanger with the fluids separated by the plate material unless the pressures were equal.—*Paper by C. M. Simmang, F. T. Saadeh, and B. E. Short; 1954 A.S.M.E. Autumn Meeting, Paper No. 54-F-22.*

Shipbuilding Behind the Iron Curtain

At the 1954 Leipzig Fair the model of a twin-screw passenger liner with 8,000 s.h.p. Diesel propulsion plant to be built by the VEB Mathias Thesen Yard, Wismar, was shown. The model of the 10,000-ton cargo motor ship of the VEB Warnow Yard, Warnemünde was also shown. The VEB Yard Neptun, Rostock, exhibited a model of their Type II trawler of 1,000 tons displacement, which is completely welded. The Diesel engine of 970 s.h.p. is equipped with reduction gearing and has a flexible coupling. The VEB Karl Liebknecht Works, Magdeburg, exhibited Buckau-Wolf marine Diesels such as a directly reversible Diesel of 400-540 s.h.p. and 275/375 r.p.m. A 150-200 s.h.p. Diesel of 375/500 r.p.m. with reversing-reduction gear was also shown. A typical cabin furnished with plastic materials was exhibited. The Chinese exhibits included an echo-sounder and a water cooled six-cylinder marine Diesel of the four-stroke cycle type developing 150 h.p. at 550 r.p.m.; the cylinder diameter is 187 mm. and the stroke is 270 mm.; the weight is 5 tons. The model of a Chinese-built dredger of 100 to 150 cu. m. per hr. capacity for depths up to 12 metres was also on exhibition.—*Schiffbautechnik, 1954; Vol. 4, No. 11, p. 362.*

Patent Specifications

Improvements in Ships' Davits

The object of the invention is to produce a relatively simple and reliable construction of davit with adequate outboard reach when in the lowered outboard position whilst avoiding undue height when in the raised inboard position. Referring to Fig. 1,

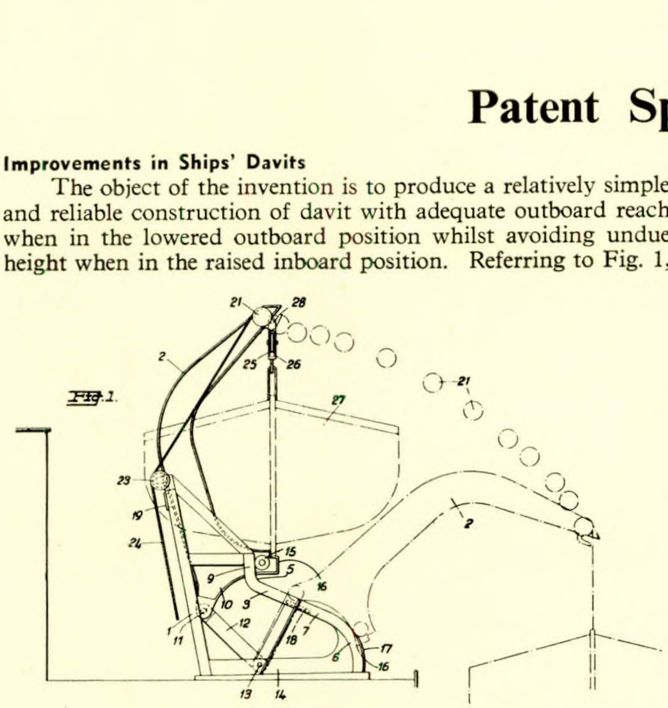


FIG. 1

a structure (1) for the support and manipulation of a davit arm (2) includes a ramp (3) consisting of a pair of ramp elements (4) which are generally inclined in a downward and outboard direction. The ramp (3) includes an upwardly or outboardly concave portion (6) with a substantially straight downwardly inclined portion (7).—*British Patent No. 720,397, issued to Campbell and Isherwood, Ltd. and S. Southall. Complete specification published 22nd December 1954.*

Improved Fuel Oil

This invention relates to a fuel oil of relatively high sulphur content, modified so as to reduce significantly the corrosive effect of the sulphur oxides in the flue gases from the burnt oil. The theory has been advanced and is supported by practical tests that the presence of excessive smoke in the flue gas has the effect of lowering the dew point with the concomitant effect of reducing corrosion. This phenomenon has been demonstrated in furnaces burning solid fuels, both with carbon smoke produced by incomplete combustion, and with zinc oxide smokes produced by the introduction of zinc blende into the furnace. Both with coal- and oil-fired furnaces, a corrosion reducing smoke may be produced by the injection of a suitable smoke-forming medium e.g., zinc blende, titanium tetrachloride or a finely-divided refractory material, such as silica into the furnace, or by incomplete combustion. The first of these methods is inconvenient and the second uneconomical. The present invention is concerned with the use of slurry oil obtained as the bottoms product from the fractionator of a fluidized catalytic hydrocarbon cracking process using catalysts comprising silica and/or alumina. The quantity of the solid components of the slurry oil required will be a function of the sulphur content of the fuel and of the particle size (i.e. a ratio of surface area to total weight) of the solid particles. It appears also that the requirement is dependent on the sorptive capacity of the smoke particles, hygroscopic smokes being more effective than non-hygroscopic smoke in equivalent quantity. As a general rule of thumb solid particle contents of the order of double the sulphur content may be used. In general, therefore, from 0·5 per cent to 8 per cent, preferably from 1 to 2 per cent by weight of solid particles based on the total composition will give satisfactory results, and this represents the total amount of solid additive necessary, excepting any peptising agent.—*British Patent No. 721,271, issued to Standard Oil Development Co. Complete specification published 5th January 1955. (Inventor: D. A. C. Dewdney.)*

Counteracting Corrosion in Diesel Engines

In ship Diesel engines, especially slow-running engines, corrosion and wear often will develop incidentally and without apparent reason on cylinder covers, pistons and cylinders. It happens for instance that Diesel engines of the same type and working under apparently the same conditions will show essential differences of wear, and the damages and losses herethrough may rise to great costs being a real worry for all ship owners. Scientific researches to find the source of all the troubles mentioned above have resulted in the discovery that during abnormal combustion detonations in the Diesel engines, electrical phenomena are created and electrolytes are generated within the cylinders creating a passage for ions, and that these phemonena are the real cause of the corrosions and wear. The present invention essentially consists in that said electrical phemonena within a Diesel or other combustion engine are counteracted by the connexion of an electrical source, one pole of the electrical source being in conducting connexion with the piston of the engine, and the other pole in conducting connexion with the cylinder of the engine. It is disclosed by researches that electrical voltages are generated when abnormal combustion detonations occur close to metal surfaces, that a freeing of electrons is initiated by ultra violet light, that combustion detonations generate ultra violet rays, and that normal combustions, incidentally, may take the shape of abnormal combustion detonations. Further, the researches have shown that electrolytes are momentarily generated within the cylinder and serve as a current conductor for the electrons which are being freed by the abnormal combustion detonations, and that measureable electrical voltages thereby are generated, by which, in fact, the abnormal combustion detonations are able to be verified. Electrical currents generated in such a way give the pistons of the engine a positive voltage with reference to the cylinders. The voltage difference is small, usually lower than 0·5 volt, the same being, however, sufficient to cause the ion to move in the electrolyte, and thereby to cause the corrosion within the cylinders of the engine. In application of the opposing voltage, by the employment of conventional means, the piston and the cylinder are in conducting connexion with different poles in the circuit of the electrical source. A potentiometer may be inserted in the circuit of the electrical source in order to regulate the counter voltage to such a degree that the corrosion and wear are reduced to a minimum. It is to be understood that by abnormal combustion detonations" is meant those where the temperature and the pressure are much higher than the normal combustion, even reaching a temperature of 5,000 deg. C.; it having been found that these detonations, which also have a much higher combustion speed than those of normal combustion, depend on the high cetane number in the Diesel fuel, and as this cetane number has been increased during the last ten years, it appears that these abnormal detonations occur, practically speaking, with each combustion. As the harmful current is active only for a short moment during the stroke cycle of the engine the opposing voltage may, if desired, be brought intermittently into action in synchronism with these combustion detonations.—*British Patent No. 721,401, issued to E. T. Christansson and K. H. T. Bergenheim. Complete specification published 5th January 1955.*

Ship's Derrick with Ball or Roller Bearing

In a ship's loading gear the derrick or boom swings about an axis which is parallel to the mast, the lower end of the derrick being rotatable. The upper end of the derrick is supported by a rope or cable, the so-called pendant, which serves the purpose of controlling the inclination and the reach of the derrick boom. At the upper end of the mast the rope or cable runs over a sheave or pulley which is pivotally mounted on a journal pin arranged on a masthead member. Fig. 1

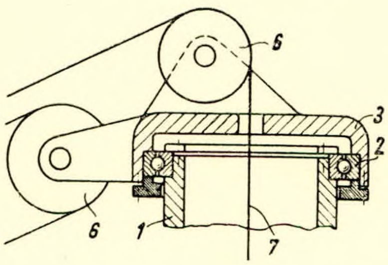
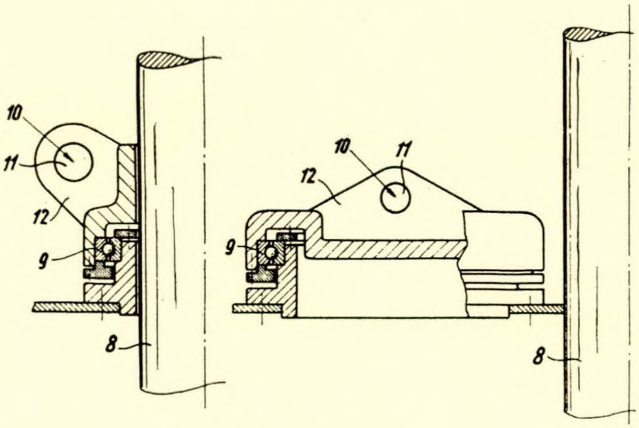


FIG. 1

represents a rope pulley bearing according to the invention. An annular ball bearing (2) is mounted on the upper end portion (1) of the derrick mast. The ball bearing is enclosed in the rotatable housing (3) to which the pendant is attached. The large diameter of the housing (3) makes it possible to provide on the housing a number of eyes or lugs between which there may be arranged all the sheaves (6) necessary for the tackle required for the operation of a derrick of high load-lifting capacity. This arrangement makes it possible to dispense with intermediate members of high weight and to facilitate the topping of the boom when the derrick is out of operation. Figs. 3 and 4 represent two preferred embodiments in

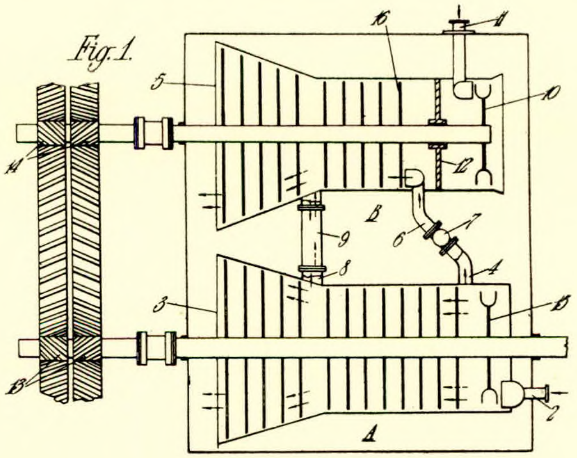


FIGS. 3 and 4

which the invention is applied to the standard or mast bearing of the derrick boom. In Fig. 3 an antifriction bearing (9) surrounds the lower portion of the mast (8). With this arrangement it is usually possible to dispense with the boom standard proper and, as shown in Fig. 3, to transmit the pressure of the boom, indicated by arrow (10), to the housing of the bearing (9) by means of a pivot comprising a horizontal axle (11) and a vertical lug (12). The pivot is carried in antifriction bearings.—*British Patent No. 722,657, issued to Kugelfischer Georg Schaefer and Co. Application in Germany made on 14th January 1952. Complete specification published 26th January 1955.*

Improvements in Steam Turbines

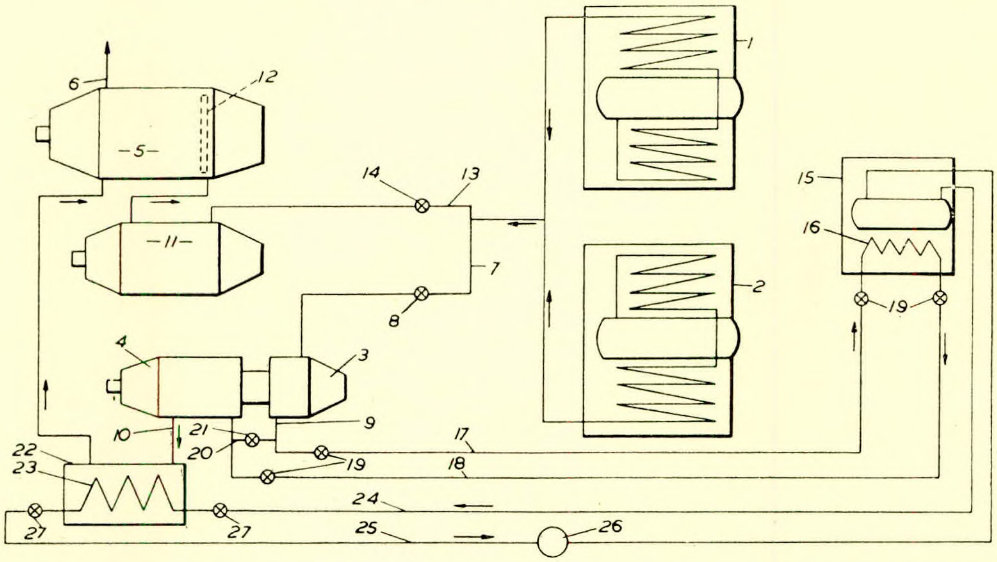
In steam turbines designed to maintain a high efficiency over a very wide range of power, such as, for example, ship propulsion turbines, it is well known to design and use the greater part of the turbine blading for low power operation and to obtain the full power by bypassing some of the stages. Two ways of doing this are known, referred to as live steam bypass and internal bypass respectively. With these arrangements it is necessary to ensure that the pressure drop across the bypassed stages does not fall to too small a value, since otherwise insufficient cooling steam would flow through these stages to carry away the heat developed by friction between the rotating parts and the elastic fluid. With too small a pressure drop the bypassed stages cease to be capable of efficient working and instead of deriving mechanical work from the action of the elastic fluid with a corresponding fall in temperature, they do work on the steam accompanied by a rise in temperature. The bypass valve and pipe must, therefore, restrict the flow of steam to some extent, and this naturally results in a loss of available power. The object of the present invention is to overcome this drawback. In carrying the invention into effect according to one arrangement illustrated by way of example in Fig. 1, two turbines are provided each driving a separate pinion



of a reduction gearbox, the blading of the main turbine (A) being designed to expand steam efficiently at a desired fraction of maximum power from the live pressure admitted through a main steam inlet (2) to the exhaust pressure at (3), the second turbine (B) being designed to deal efficiently with steam expanding from the second stage pressure (i.e. the pressure at (4) between the first and second stages) of the turbine (A), down to the exhaust pressure at (5). A cross-connexion is provided for this purpose comprising a pipe (6) embodying a control valve (7). The two turbines are also cross-connected at an interstage point where a conveniently low pressure obtains, e.g., about two-thirds of the way down the turbines, the connecting pipes (9) being permanently open and not under the control of a valve. The second turbine (B) also includes an entirely separate group of stages indicated diagrammatically at (10) for astern working by steam admitted at (11); this group is isolated from the ahead stages by a suitable division plate (12). At low powers with the bypass valve (7) shut, those stages in both turbines (A and B) beyond the permanently open cross-connexion pipe (9) share the steam flow in proportion to the blading areas, and are conveniently designed to have identical blading, in which case each takes half of the total flow. Earlier stages of the turbine (A) carry the whole flow at low powers and the early stages of the second turbine carry no steam flow except for gland leakage elastic fluid.—*British Patent No. 721,287, issued to The Parsons and Marine Engineering Turbine Research and Development Association. Complete specification published 5th January 1955. (Inventor: H. G. Yates.)*

Marine Steam Turbine with Interstage Reheating

This invention relates to marine steam turbine installations of the type in which a steam boiler system supplies steam to turbines that drive the propellers, and in which the steam passes through the turbines in stages of different steam pressure conditions. The object of the invention is to improve the thermal efficiency of such installations by inter-stage reheating of the steam in an improved manner. Referring to the drawing, the installation shown includes main watertube steam boilers (1 and 2). These boilers generate steam at a pressure of 950lb. per sq. in., superheated to a temperature of 850 deg. F. The installation includes one or more two-stage ahead turbines, one being shown; and this turbine, or each such turbine, consists of a high pressure stage (3) and an intermediate pressure stage (4). The installation also includes one or more low pressure turbines, one being indicated at (5), exhausting in conventional manner as indicated by (6) to one or more condensers (not shown). Steam is led from the boilers (1 and 2) by way of piping (7) through a manœuvring valve (8) direct to the high pressure stage (3) of the one or more ahead turbines from which the steam passes through the first inter-stage piping (9) to the intermediate stage (4), after which the steam is passed through the second inter-stage piping (10) to the one or more low pressure turbines (5), passing finally at (6) to the one or more condensers. The astern turbine steam system is any conventional arrangement. In the example, steam is led by way of piping (13) through a manœuvring valve (14) direct from the boilers (1 and 2) to the one or more high pressure astern turbines (11), from which the steam is passed to the astern wheel (12) of each low pressure turbine (5), finally passing by way of the exhaust (6) to the one or more condensers. The turbines drive the ship's propeller shafting through reduction gearing, as usual. There is also an independent auxiliary oil-fuel watertube boiler (15), which generates steam at the extra high pressure of 2,000lb. per sq. in. This auxiliary boiler is equipped with a tubular heater (16) in the passage for its combustion gases. The installation also includes a tubular steam heat exchanger (22). The tube system (23) of this exchanger is connected by extra high pressure piping in the steam circuit of the auxiliary boiler (15). In operation of the installation for ahead propulsion, steam from the first inter-stage pipe (9) is passed through the gas-heated heater (16) in the auxiliary boiler (15) and reheated there to a temperature of about 950 deg. F., and steam from the second inter-stage pipe (10) is passed through the steam heat exchanger (22) and raised to a temperature there slightly below the saturation temperature of the extra high pressure steam. For manœuvring or astern propulsion, the oil-fuel supply to the furnace of the auxiliary boiler (15) is cut off and the appropriate steam valves (19 and 21) are manipulated to shut down and isolate the reheating system.—*British Patent No. 721,276, issued to The Fairfield Shipbuilding and Engineering Co., Ltd. Complete specification published 5th January 1955. (Inventor: A. W. Davis.)*



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Marine Engineering and Shipbuilding Abstracts

Volume XVIII, May. No. 5 1955

X-ray versus Ultrasonic Inspection

There are many points of similarity between radiography and ultrasonic inspection. Both are pentrating methods, seeing right through the metal. Both find inherent defects. Both are most economically applied to parts before any processing is performed, in order to weed out any rejectable material. Both require skilled interpretation. Both may be used on ferrous or non-ferrous metals, any shape. Both are useful in detecting cracks or porosity in weldments. Both may be portable, for use in the field. However, there are just as many differences between radiography and ultrasonic inspection. Radiography is most sensitive to voids with the major axis parallel to the beam of radiation; a lamination parallel to the surface and normal to the beam of radiation cannot be detected unless its thickness approaches 2 per cent of the thickness of the part. Ultrasonic is most sensitive to voids with the major axis normal to the ultrasonic beam; a thin seam or crack running normal to the surface may be undetected because the beam will pass all around it and still reach the back surface and be reflected, as though no crack were present. Radiography indicates the size and thickness of a void in metal, but does not locate it in all three dimensions. A radiograph does not tell whether the porosity in a casting is just below one surface, in the centre, or just under the other surface of the part. Ultrasonic locates the surface level of the defects, but not its thickness. To find out whether an indication means a minute lamination or an open cavity, one must inspect ultrasonically from both sides and note the depth indicated from each surface. For radiography, it is usually necessary to have both surfaces accessible. Ultrasonic inspection can often be accomplished from one side. Surface smoothness is of no importance in radiography, but rough surfaces may cause undue crystal wear, or may give rise to extraneous signals, in ultrasonic inspection. All flaws which affect density, regardless of location on or near the surface, are recorded by X-ray. Much ultrasonic inspection, however, is "blind" for some $\frac{1}{4}$-in. under the surface because of the large signal caused by the initial pulse entering the metal. (There are some new "skip" techniques which overcome this disadvantage of ultrasonic inspection.) Radiography is limited to parts a few inches thick, while ultrasonic waves travel through 20 or 30 feet of metal in almost no time. For the reasons cited above, radiography is most useful on castings and weldments.—*R. H. Sparling, Non-destructive Testing, September-October 1954; Vol. 12, pp. 19-28.*

Use of Digital Computer for Analysis of Structures

The basic principles of the electronic digital computer at Manchester University are described briefly. It is emphasized that although such machines are popularly referred to as "electronic brains", they merely carry out numerical computations in accordance with instructions fed to them on punched teleprinter tape. Thus, while these machines are incapable of thought, they are capable of dealing with computations of enormous complexity. The instructions for a given kind of computation are known as the "programme"; the numerical data of a particular problem are fed to the machine independently of the programme but in a similar manner. One of the first engineering problems to be programmed was that of structural analysis and the choice of method is discussed. The kind of programme adopted is based upon a rigorous method of analysis whereby the equations of equilibrium and compatibility of strains are solved simultaneously and use is made of matrices. The operation of the programme is described and details of the analysis of a power station building frame using the computer are given. The problem of economical frame design by the methods of the elastic and plastic theories is discussed in relation to the use of computers. Other applications of computers, including vibration analysis, are considered briefly. The general problem of programming is discussed with particular regard to the human factor. Finally, the future use of computers is considered; it is suggested that computers will be used to best advantage only when engineers in industry are prepared to understand the technique of programming and use it for their own problems. A survey of the various available methods of structural analysis is contained in the Appendix.—*Paper by R. K. Livesley and T. M. Charlton, read at a meeting of the North East Coast Institution of Engineers and Shipbuilders on 26th November 1954.*

Australian Cargo Vessels

The new construction programme of the Australian Shipbuilding Board includes several types of vessels ranging from 12,500-ton d.w. 14-knot turbine-driven ore-carriers to 200-ton dumb hopper barges. An important group, both numerically and as effective cargo-carrying units is the B-class, the design for which was based on a survey of the requirements of the various Australian coastal shipping companies. In these vessels it was found possible to achieve standardization of the hull and yet to provide alternative types of machinery and modifications in the arrangement of holds, accommodation and other details to meet special requirements. The original plans provided for two oil-fired steamships, two coal burners and six vessels powered by Commonwealth-Doxford Diesel engines. The two oil-fired vessels are the *Bilkurra* and *Binburra,* whose names are taken from the Aborigine words for flowers. They were constructed by Evans Deakin and Co., Ltd., Brisbane, and launched complete with all machinery and boilers in place. They were commissioned some time ago and their performances on the Australian east-coast run from the aspects of economy of running, cargo handling, etc., have been excellent. The vessels are of the open shelter deck type, built to the highest class of Lloyd's Register of Shipping, with principal particulars as follows:—

Length between perpendiculars	380ft. 0in.
Moulded breadth	53ft. 0in.
Moulded depth to shelter deck	33ft. 0in.
Moulded depth to second deck	24ft. 3in.
Deadweight capacity on 22ft. 6in. draught	6,474 tons
Bale capacity	350,445 cu. ft.
Grain capacity...	382,023 cu. ft.
Gross tonnage...	3,952 tons
Net tonnage	2,088 tons
Speed	12½ knots
Power	3,000 i.h.p.

The hulls are generally of riveted construction, but welding has been introduced to a considerable extent, including the ends of the shell plating, the inner bottom, keel, centre and side girders. The main transverse bulkheads, decks, casings and masts are completely welded. Oil fuel and fresh water are carried in the double bottoms. The main bunker and settling tanks are at the sides of the boiler room. There are four holds with 'tween-decks, three forward and one aft of the machinery space which is consequently somewhat aft of amidships. Rapid discharging capacity is an important consideration on the Austrialian seaboard where stevedoring charges are heavy and the ships are accordingly equipped with sixteen 5-ton and one 25-ton derricks. The deck machinery includes one direct-acting steam steering engine of Wilson-Pirrie type, sixteen 8in. by 12in. steam winches and a 10in. by 14in. steam windlass. The holds are clear of obstructions and are without pillars. Small side doors are fitted in the shell above the main deck. As the illustration shows, a thoroughly workmanlike appearance has resulted. The main machinery consists of a three-crank triple-expansion reciprocating steam engine with a Bauer-Wach exhaust turbine. The cylinder diameters are 23½in., 38¾in., and 64in. with a stroke of 45in. The engine develops 2,120 i.h.p. at 85 r.p.m. and the exhaust turbine provides an additional 880 equivalent i.h.p. Steam is generated at 220lb. per sq. in. and 620 deg. F. in two Babcock and Wilcox oil-fired header-type watertube boilers. Each is capable of providing 17,500lb. of steam per hour, under normal service conditions. Inter-deck superheaters are provided through which all the steam passes and spray-type de-superheaters are installed for the main and auxiliary steam systems.—*The Marine Engineer and Naval Architect, November 1954; Vol. 77, pp. 450-452.*

Barge Propulsion from Shore Current?

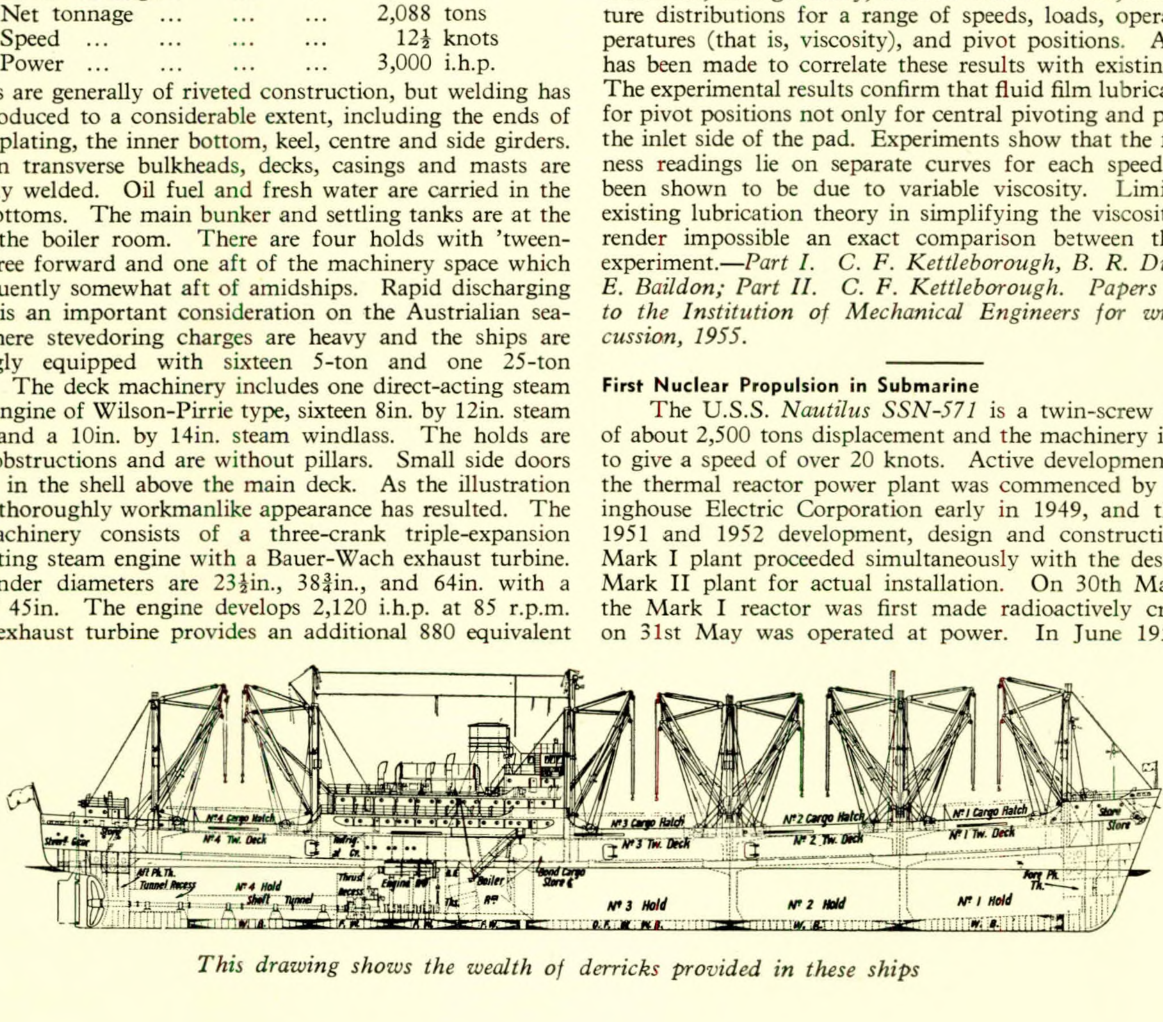
First experiments with electrically-driven vessels are taking place on inland waterways near Moscow, according to a recent issue of *Hansa.* Diesel-electric passenger ships plying on the Moskua Canal switch from Diesel power to electricity taken from overhead cables installed over a trial length. An automatic cable winder enables ships to change course within the canal fairway. Another experiment with power supply is being carried out on barge-trains which take power on the induction principle from two high tension cables running along both banks of the canal. The overhead cable on land serves as a primary winding of a high tension transformer while the intake grid is mounted on special masts on the barges—not the tug, which is "too short" for this purpose. This innovation is credited to the Russian Dr.Ing. G. J. Babat, who is considered the most important constructor of contactless electric transport in the Soviet Union today.—*Shipbuilding and Shipping Record, 16th December 1954; Vol. 84, p. 801.*

Michell Bearing Lubrication

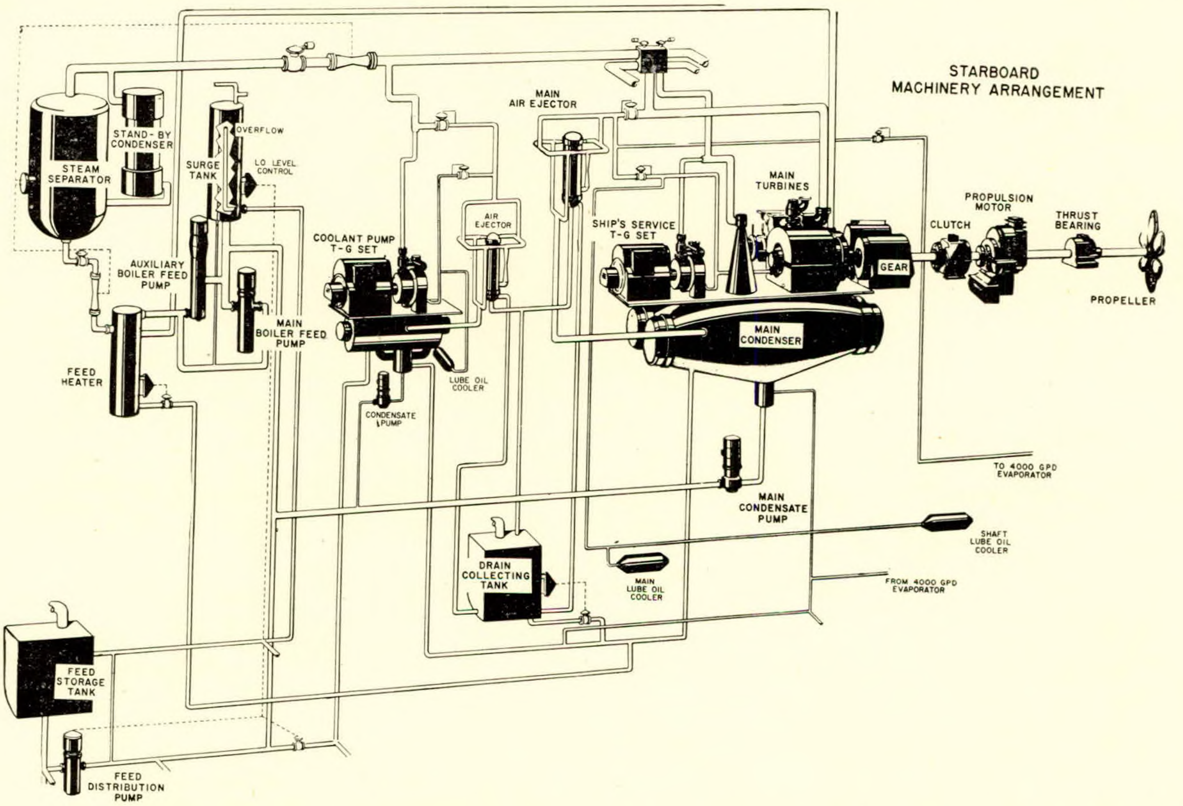
An experimental investigation into the operation of tilting-pad thrust bearings has been carried out. The following experimental determinations have been made: film pressure distributions, film geometry, friction characteristics, and temperature distributions for a range of speeds, loads, operating temperatures (that is, viscosity), and pivot positions. An attempt has been made to correlate these results with existing theories. The experimental results confirm that fluid film lubrication exists for pivot positions not only for central pivoting and pivoting on the inlet side of the pad. Experiments show that the film thickness readings lie on separate curves for each speed; this has been shown to be due to variable viscosity. Limitations to existing lubrication theory in simplifying the viscosity changes render impossible an exact comparison between theory and experiment.—*Part I. C. F. Kettleborough, B. R. Dudley, and E. Baildon; Part II. C. F. Kettleborough. Papers submitted to the Institution of Mechanical Engineers for written discussion, 1955.*

First Nuclear Propulsion in Submarine

The U.S.S. *Nautilus SSN-571* is a twin-screw submarine of about 2,500 tons displacement and the machinery is designed to give a speed of over 20 knots. Active development work on the thermal reactor power plant was commenced by the Westinghouse Electric Corporation early in 1949, and throughout 1951 and 1952 development, design and construction of the Mark I plant proceeded simultaneously with the design of the Mark II plant for actual installation. On 30th March 1953, the Mark I reactor was first made radioactively critical and on 31st May was operated at power. In June 1953, a sus-



This drawing shows the wealth of derricks provided in these ships tained full power run simulating an Atlantic crossing submerged was concluded successfully. When the *Nautilus* was launched on 1st January 1954, the major components were already installed. During the "run" across the Atlantic the "submarine" proceeded at top speed. It was never necessary to shut down the reactor or even to "surface". Only three times during the entire "crossing" did the unit slow down. Once it was throttled back to two-thirds power for seven minutes and twice to half power for a total of less than 1½ hours. In each case the necessary adjustments were made to permit continued operation. The equipment to be mounted in the hull of a nuclear-powered submarine is so functionally interrelated that it must be considered as a single power plant. However, it requires hull space of such size that it was necessary to divide it into two physically separated compartments to provide adequate watertight integrity in the event of battle damage to the submarine hull. The reactor compartment contains the nuclear reactor, all steam-generating equipment and the auxiliary systems. The engine room contains the propulsion generators. The reactor vessel is located vertically in the compartment. In the lower part of the compartment are the primary coolant pumps. Outboard of the pumps is the steam-generating equipment. In the upper part of the compartment above the steam generating equipment, is the steam separator, from which emerge the main steam headers that lead aft to the engine room. Aft of the main pumps is located the pressurizer unit. These components outline the main areas in the reactor compartment. The areas between them are filled with secondary equipment and related piping and cables, electrical and pneumatic control panels, valves, motor generator sets, and instruments. The engine room is located adjacent to and aft of the reactor compartment. This space is divided into upper and lower levels. Each level is provided with a walkway running fore and aft along the horizontal centreline. Components also have to be located on intermediate levels because of their varying size. These intermediate levels are accessible from either the lower or upper level. To convert the shaft horsepower from the low and high pressure turbines into useful thrust,



Diagrammatic arrangement of the engine room in the submarine Nautilus *with nuclear power propulsion plant*

equipment, all steam-driven items, associated control panels and switchgear, as well as the main control point for the equipment in both compartments. The primary coolant system, located in the reactor compartment, consists of a reactor pressure vessel, which contains a nuclear reactor and coolant loop. The water coolant, called primary water, which is also the moderator, is circulated by canned motor-type pumps through the reactor vessel to be heated, and then through the steam generators for transfer of heat to the water on the secondary side. The wet steam rises to the steam separator where the water is removed, providing dry and saturated steam. The coolant loop is provided with stop valves to permit isolation of parts for maintenance purposes. From the separator the steam is carried through pipes which penetrate the bulkhead and diverge in the engine room to supply steam to the main propulsion turbine and to the ship's service turbine generator sets. Condensate is pumped back to the steam it passes through the reduction gears, the clutch and finally through the propulsion motor shaft. The power is then applied directly to the propeller. The propulsion system clutch enables the propulsion motor and propeller section to be disengaged from the turbine and main reduction gear section. If a clutch were not used the propulsion motor, when in operation, would have to overcome the inertia and friction losses of the turbine rotors and reduction gear in addition to overcoming the normal propeller load. The clutch also permits some angular and parallel mis-alignment of the drive shaft. It incorporates a thrust link which serves to locate the reduction gears axially and transmit any axial forces due to these gears to the main thrust bearing.—*The Motor Ship, December 1954; Vol. 35, p. 400*

Stresses in Cylindrical Shells and in Plain and Pierced Drumheads

This paper presents experimentally verified methods and curves for determining the stress ranges which occur in non-circular shell sections and plain drumheads. It points out, however, that such stress ranges, which are commonly measured by a strain-gauge test on a pressure vessel, are often higher than, and seldom directly comparable with, the maximum absolute stress in the vessel. It is shown that, in the case of drumheads where some yield usually takes place on the first application of the hydraulic test pressure, it is not feasible to base design procedure on the maximum absolute working stress. The most satisfactory procedure is to base on stress range, and a means of evaluating the maximum safe stress range from the yield properties of the material at temperature is tentatively suggested. Work on this subject is continuing, but there is an important qualification which has to be remembered. Experimental work may show that comparatively high stress ranges are permissible from the point of view of the stress-carrying properties of the material, but it may be either the maximum permissible surface elastic strain range or the total elastic deflexion of the head, which is found to be the true limiting factor in the design. Particular conclusions of the paper may be enumerated as follows:—(1) Haigh's method of calculating bending stress ranges due to out-of-circularity in cylindrical shells agrees closely with experimental results. (2) From this it may be deduced that, in vessels which are thick for their radius, greater local deviations from the circular could be permitted than present codes allow. (3) The formation of residual compressive stress on release of the first application of pressure to a drumhead has been studied experimentally and confirmed. (4) Well's method of calculating elastic stress ranges in torispherical heads, using a continuous beam-on-elastic-foundation formula, agrees closely with experimental results from a wide range of drumheads. (5) From results of this form of calculation it may be shown that the method of plain drumhead design used in the British code is more than safe, based on "no reverse yield" subsequent to any occurring during the initial hydraulic test. Some thickness reductions could be obtained, particularly among the very thick drumheads. (6) The maximum absolute stress in a drumhead at working pressure is not a suitable criterion for design purposes; it would lead to excessively thin drumheads having too high a strain and deflexion. (7) Drumheads having flanged elliptical manholes have performed well in the low and medium pressure ranges. When designed for high pressure, according to present rules, they tend to be rather flexible and the manhole could with advantage be compensated more locally. (8) A solution to this problem is to design all heads initially as plain heads, and to compensate manholes and other openings individually by the best local means known.—*Paper by W. B. Carlson, and J. D. McKean, presented at a General Meeting of the Institution of Mechanical Engineers, on 10th December 1954.*

Lloyd's Register Rules for Modern Cargo Ships

As a result of the Society's investigations into cases of structural damage and failure which have occurred during recent winters to some modern types of dry cargo ship, the Committee of Lloyd's Register appointed a special panel in January 1954, to consider what amendments to the Construction Rules were necessary to ensure a satisfactory standard of longitudinal strength in this type of ship. After examining the results of these investigations and considering various proposals, the panel recommended important amendments to the Rules. Following their consideration by the Technical Committee, these amendments were adopted by the General Committee on 9th December 1954, and will be published and applied forthwith to new ships for which plans are submitted to the Committee for approval. Broadly the amendments can be divided into two main parts. First, formulæ have been devised and introduced into the Rules whereby the minimum longitudinal scantlings, which are based on dimensions, will require to be increased when necessary to take account of modern design characteristics in respect of cargo distribution and speed. The object of this amendment is to maintain the same standard of longitudinal strength for all types of dry cargo ship, and will apply irrespective of the mode of construction of the hull. The second part deals with welded construction and the amendments require a considerably extended use of specially approved notch-tough steels for the main structure of welded ships. A limited use of such steels was first introduced into the Rules in 1950. More specific requirements relating to detailed structural design have also been prescribed. The underlying considerations governing the extended provision of special quality steels in welded dry cargo ships apply generally to tankers, and the Rules for this type of ship have also been amended. The Technical Committee have also appointed a special panel to review the Rules for the quality and testing of notch-tough steels suitable for welded construction.—*The Motor Ship, January 1955; Vol. 35, p. 466.*

Cavitation Tunnel Tests Compared with Open-water Tests

Hand in hand with the development of model research on ships' propellers in the cavitation tunnel, intensive investigations are being carried out on problems arising from cavitation erosion. Fig. 1 shows the general plan of the cavitation tunnel

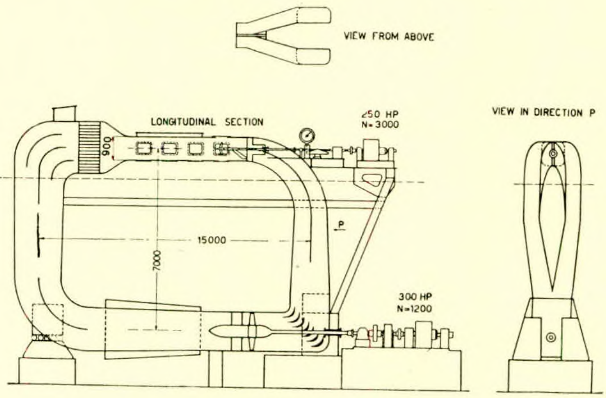


FIG. 1—*Cavitation tunnel of the N.S.M.B.*

of the Netherlands Ship Model Basin. The tunnel, built in the form of an enclosed tubular channel, is placed in a vertical plane in order to eliminate the danger of cavitation of the impeller which generates a velocity of about 12 m. per sec. of the water in the vicinity of the model screw under observation. The power required is 300 h.p. The model screw is driven by a d.c. motor with an output of 250 h.p. at 1,000-3,200 revs. per min. It is customary to carry out the tests at a constant velocity in the measuring channel of about 5-8½ m. per sec. The bronze model screws are made with a diameter of 40-46 cm. (maximum 50 cm.). The number of revolutions varies between 700 and 2,000 per minute. The construction of the tunnel ensures an even flow at the measuring point. By means of a vacuum pump it is possible to reduce the pressure above the free surface of the fluid—necessary for obtaining the required static pressure—and to maintain this during the test. While tests are in progress, the water velocity and the static pressure in the vicinity of the model screw are constant, whilst the number of revolutions is varied, in order to cover a range of loads. Measurements are taken of the torque, thrust and the number of revolutions, thereby establishing the characteristics of the screw (Fig. 2). At the same time, cavitation phenomena, if any, are observed and, if necessary, photographed through the observation window. For this purpose a Philips stroboscope is used, which is equipped with two lamps illuminating the model screw from above and below. By means of a contact on the propeller shaft, the stroboscope lamps can give flashes of 10^{-5} sec. duration in synchronism with the number of revolutions. This enables excellent observations to be made of the screw. which appears

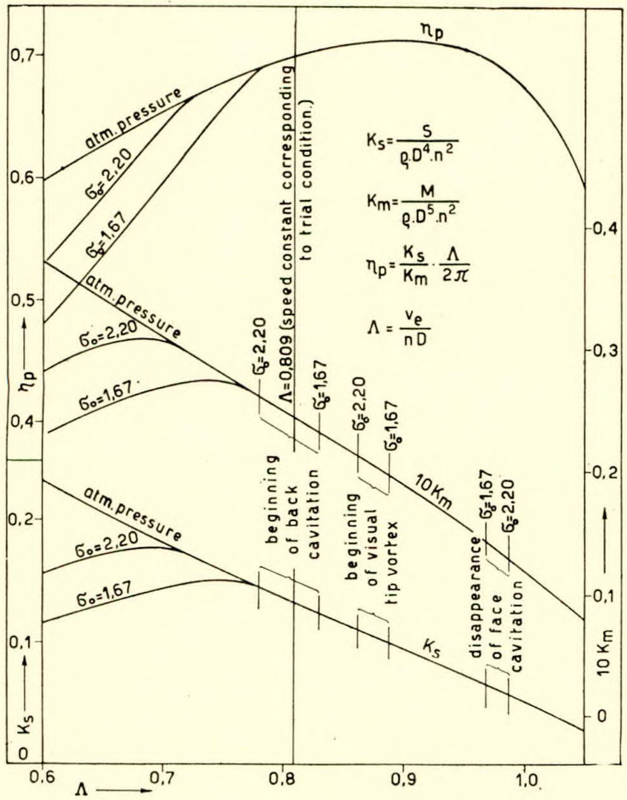


FIG. 2—*Results of a cavitation test*

to be at rest. By turning the contact around the propeller shaft, the observer can examine the different blades of the model screw in any position. An attempt has been made to explain the discrepancies which occur between the results of open-water tests and those carried out in the cavitation tunnel. As a result of these investigations the following conclusions may be drawn to aid future research to be carried out in cavitation tunnels. 1. In order to arrive at accurate testing of model screws in the cavitation tunnel, it is necessary to be able to test these same screws with the same measuring apparatus in the towing tank at the same number of revolutions. For this purpose a dynamometer with a sufficiently large measuring field is required. The K_s-identity method can then be applied, either for each model screw separately or for three model screw series with varying screw diameters, so that a calibration diagram can be made for the velocity in the cavitation tunnel. 2. The object of this K_s-identity method is not only to measure the power on the model screws in the cavitation tunnel as accurately as possible, but also to obtain as exact a prediction as possible of the cavitation phenomena. The question as to how far the radial distribution of the screw's load on the model screw is influenced by wall effect still needs to be investigated. In order to accomplish this measurements can be taken behind the model screws with the help of a blade-wheel apparatus, both in the open-water test and in the cavitation tunnel. The number of revolutions of these axial blade wheels is a measure of the tangential disturbance velocities and, therefore, of the circulation of the blade sections. By means of this blade-wheel apparatus, it is possible at the same time to study more closely the phenomenon of thrust influence.—*J. D. Van Manen, International Shipbuilding Progress, No. 3, 1954; Vol. 1, pp. 149-155.*

Automatic Steering Device

A new automatic steering device introduced by the Marconi International Marine Communication Company is a compact, simple "electronic helmsman" designed for small craft such as yachts, drifters and trawlers. Named the Marconi Martinet, it is a joint product of the Marconi Marine Company and Hartley Electromotives. It does not depend on a gyro-compass installation for control, but merely on a small magnetic and high frequency electronic assembly powered by a 12 V or 24 V battery or from 110 V d.c. mains. Once the ship is set on her course, the Martinet may be switched on and will then hold her "as she goes", firmly correcting any tendency to yaw. A useful feature is the provision of a hand-held remote control unit which permits accurate conning and fine adjustments of steering when the helmsman is away from the actual steering position. The Martinet is controlled primarily by terrestrial magnetism. When set on a course any deviation caused by yawing due to wind or sea brings into operation electronic circuits whose action is governed by the relation between ship's head and magnetic north. A small electric motor connected to the steering wheel then puts helm on in the appropriate direction to bring the vessel back on course.—*Shipbuilding and Shipping Record, 16th December 1954; Vol. 84, p. 806.*

Actual and Model Tests of Japanese Cargo Ship

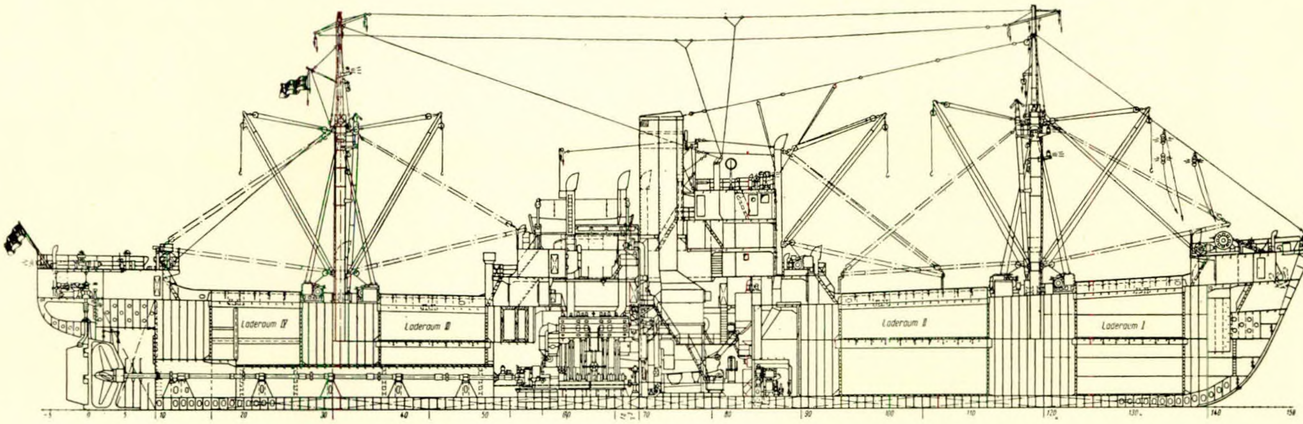
This report describes extensive full-scale and model experiments carried out on the cargo ship *Nissei Maru*. The full-scale results, dealt with in Part I of the report, were obtained during speed trials and during the ship's maiden voyage from Yokohama to Vancouver, Honolulu, Singapore, Bombay, Mormugao, Singapore, and back to Yokohama, a distance of 21,700 nautical miles. Simultaneous readings were taken at regular intervals of the ship's speed; propeller revolutions; torque; angle of helm; angles of yawing, pitching, and rolling; heaving motions; hull stresses in several positions; and sea conditions. The main particulars of the ship are as follows:—

Length, b.p.	419·9 feet
Breadth, moulded	54·7 feet
Depth, moulded	34·1 feet
Draught at full load	27·1 feet
Displacement at full load ...	13,870 tons
Deadweight	9,914 tons
Block coefficient at full load ...	0·728
Midship coefficient at full load...	0·987
Position of LCB...	1·38 ft. forward of amidships
Designed service speed	13·4 knots

The main engine is a cross-compound impulse turbine with double reduction gearing, developing a maximum continuous power of 4,000 s.h.p. at 105 r.p.m., and a normal service output of 3,400 s.h.p. at 99 r.p.m. The four-bladed propeller has a diameter of 17·2 feet. The butts of the shell plating are all-welded; the seams in the bottom are riveted, and in the sides alternately welded and riveted. All plating is riveted to the frames. Particulars are given of the instruments used in the tests and full results are presented in graphs and tables and discussed. It is thought that much valuable information was gained; it was particularly noticed that the effects of wind and waves on the propulsive performance were far larger than expected. This fact led to a series of model tests, described in Part II, whose aim was to clarify the nature of the loss of speed of a ship in a seaway and the effect of wind and waves on the seaworthiness of vessels under any weather and sea conditions. Models of the *Nissei Maru* were tested in an experiment tank and in a wind tunnel. The wind tunnel experiments were carried out on an "image" model, which consisted of two models of the above-water portion joined together symmetrically at the water plane. The tank experiments included rolling, pitching, heaving, and surging tests; resistance and self-propulsion tests; and propeller open-water tests. These were carried out in smooth water and in waves. The results are given and discussed. The results of the self-propulsion tests agree well with the full-scale data in the case of near-ideal conditions, i.e. no wind, no waves, clean bottom, no effect of shallow water, etc. Therefore, the usual method of self-propulsion tests, in which the friction correction is calculated by using Froude's friction constants, is considered to be satisfactory for ships similar to the *Nissei Maru.—Report No. 1 of Japanese Shipbuilding Research Association, 1954; Journal, The British Shipbuilding Research Association, November 1954; Vol. 9, Abstract No. 9,545.*

East German Cargo Boats

The VEB Deutsche Seereederei, Rostock, took delivery of two 3,000-ton cargo boats with the VEB shipyard Neptun, Rostock. These two vessels, *Rostock* and *Wismar*, are the first cargo vessels to be built by a state-owned East German shipyard. Their design is based on Russian practice. The vessels are equipped with the latest equipment, including automatic boiler control. The leading dimensions are 102·4 m. overall, 95·84 m. between perpendiculars, 14·4 m. moulded breadth, 6·03 m. draught loaded, 6,739 tons displacement. The hull is completely welded. Service speed loaded is 13·3 knots. The main engine is a double compound engine built by VEB Karl Liebknecht, developing 2,450 i.h.p. at 90 r.p.m. in conjunction with an exhaust turbine with seven reaction stages. Maximum power at 94 r.p.m. is 2,750 i.h.p. Steam is produced at 16·2 atmospheres and 325 deg. C. and admitted to the engine at 14·0 atmospheres and 300 deg. C. The watertube boiler of 186 sq. m. b.h.s. is equipped with a mechanical grate for coal firing having 8·86 sq. m. grate surface. Maximum firing rate, based on 6,000 kcal. per kg. bituminous coal is 112 kg. per hr. per sq. m. grate area.—*Schiffbautechnik, No. 11, 1954; Vol. 4, pp. 335-352.*



East German cargo boats Rostock *and* Wismar

The Supertanker *World Glory*

The Quincy Yard of the Bethlehem Steel Company, Shipbuilding Division, recently completed one of the world's largest bulk-oil carriers. The *World Glory,* with its length overall of 736ft. 3¾in. and its moulded breadth of 102ft., has the distinction of being, by a substantial margin, the largest cargo ship yet built on the American continent. In the linear dimensions of length and breadth, she also exceeds all but one of the merchant vessels of all types which American shipyards have produced so far, the single exception being the passenger liner *United States.* The *World Glory* is 13ft. longer and nearly 10ft. wider than the liner *America,* next in order of size among American-built merchant ships, and exceeds the *Independence* and *Constitution* by 54ft. in length and 13ft. in breadth. The *World Glory* was designed by the Central Technical Department of the Bethlehem Steel Company, Shipbuilding Division. Working plans were developed in the drafting rooms of the Quincy Yard. Before construction of this ship began, three models of the same dimensions but different hull form, all from lines developed in Bethlehem's Central Technical Department, were subjected to extensive tests at the David Taylor Model Basin. One of these three models was selected for the construction of the actual ship; tests showed for this model an advantageous combination of low tow-rope power, high propulsive efficiency, and good distribution of wake over the propeller disc. The value of these tests and the good qualities of the hull form were demonstrated by the performance of the completed ship on trials. When run on the measured mile at Rockland, Me., ballasted with water to the designed service draught of 34ft. 6in., the *World Glory* averaged 17·1 knots with 15,000 s.h.p., and 16·1 knots with 12,000 s.h.p. Allowing the customary 25 per cent margin for wind and weather conditions at sea, the vessel can thus be credited with a "sea speed" of better than 16 knots. Throughout her trials, the ship showed freedom from vibration to a remarkable degree in all conditions of operation. The single-screw, geared turbine propelling machinery of the *World Glory* follows present Bethlehem practice for merchant ship installations of the same type and power, and is almost identical to that of several other large and recently built tankers. The turbines were designed by Bethlehem and built in the shops at the Quincy Yard. There is a high pressure turbine of the impulse-reaction type and a low pressure turbine of the single-flow reaction type. The impulse-type astern turbine is incorporated in the exhaust end of the low pressure ahead turbine. End thrust in both turbines is balanced by dummy pistons and equalizing pipes, and each turbine rotor is fitted with a Kingsbury thrust bearing. Each turbine rotor is machined from a single forging of open-hearth steel. Turbine casings are made in upper and lower halves of cast steel. The double-reduction, double-helical, nested-type gears are of Falk manufacture. The pivoted, segmental-type main thrust bearing is incorporated at the forward end of the reduction gear casing. Propeller shafting is solid forged steel; the diameter of the tailshaft, under the bearing sleeve, is 24 inches. The propeller is five-bladed, solid, cast manganese bronze unit having a diameter of 22ft. 6in. and a pitch of 17ft. 4in., and weighs 66,930lb. It was designed by Bethlehem and made in the propeller shop at Bethlehem's Staten Island Yard. Steam is supplied by two Combustion Engineering, Inc., boilers of the two-drum bent tube, vertical-superheater type. Normal power evaporation rate is 52,500lb. of steam per hour per boiler with steam conditions of 600lb. per sq. in. gauge and 850 deg. F. at the superheater outlet. Each boiler has a maximum total evaporation rate of 72,000lb. per hr. Furnaces have water walls at the sides, rear and roof. Boilers also are fitted with auxiliary desuperheaters, steam air heaters and aluminium extended surface economizers. A low pressure steam generator takes steam normally from the auxiliary steam line or from a turbine bleeder and returns the condensate to the deaerating feedwater heater. Heat given up by this steam is used to generate up to 45,000lb. per hr. of saturated steam at 120lb. per sq. in. This steam is supplied to certain services where low pressure steam is needed, or where steam and exhaust lines are subject to a certain degree of contamination so that the condensate cannot be mixed safely with the boiler feedwater.—*Marine Engineering, October 1954; Vol. 59, pp. 42-51.*

Cargo Vessel with All-welded Aluminium Superstructure

The largest ocean-going cargo vessel ever built in Canada is the 12,700-tons deadweight turbine steamship *Sunrip,* which completed trials successfully in November from the Davie Shipbuilding yard at Lauzon, Quebec, where she was built for Sun Steamships, Ltd. She has been chartered on a long-term basis to Saguenay Terminals, Ltd., the steamship subsidiary of the Aluminium Company of Canada, Ltd. (Alcan). From a ship operator's point of view one of the most interesting features is the extensive use of aluminium in the vessel. Of outstanding interest is the superstructure, which is the largest all-welded aluminium superstructure ever built in the world. The *Sunrip* is a two-deck vessel of the closed shelterdeck type with raised forcastle, raked stem and cruiser stern. She is designed to maintain a service speed of 13·5 knots on about 33 tons of bunker "C" fuel oil per day under normal weather conditions. Her general particulars are as follows:—

Length, overall...	475ft. 0in.
Length, b.p.	450ft. 0in.
Breadth moulded	62ft. 6in.
Depth moulded to upper deck	40ft. 9in.
Depth moulded to second deck	31ft. 3in.
Deadweight (approximately) ...	12,700 tons
Summer draught	29ft. 0in.
Cubic capacity (grain) ...	615.000 cu. ft.
Block coefficient	0·725

Designed primarily as a bulk carrier, the *Sunrip's* main function will be to transport alumina (aluminium oxide) from Port Esquivel, Jamaica, to the new Alcan aluminium smelter at Kitimat, B.C. She is the first ship specially constructed for this trade. Her design includes a number of interesting technical developments which will facilitate operations considerably. To overcome any possible instability caused by the shifting of cargo the ship has been equipped with a system of centreline bulkheads, feeders and trimming hatches similar to those on grain vessels. A unique feature of the *Sunrip* is that key sections of the feeders are capable of folding away from the square of each hatch, thereby enabling her to convert to general cargo in the matter of a few hours. The *Sunrip* is powered by a set of geared turbines of Pametrada design driving a single screw. This machinery, which has been manufactured by John Inglis and Co., Ltd., Toronto, is designed to develop 5,000 s.h.p. in service at a propeller speed of 115 r.p.m. Steam is supplied by two Babcock and Wilcox marine type watertube boilers of the single-pass, sectional header type at a working pressure of 470lb. per sq. in. and a temperature of 760 deg. F. The boilers are equipped with a complete system of automatic combustion control by Bailey Meters and Controls, Ltd.—*The Shipping World, 5th January 1955; Vol. 132, pp. 13-15.*

Ships' Structural Joints

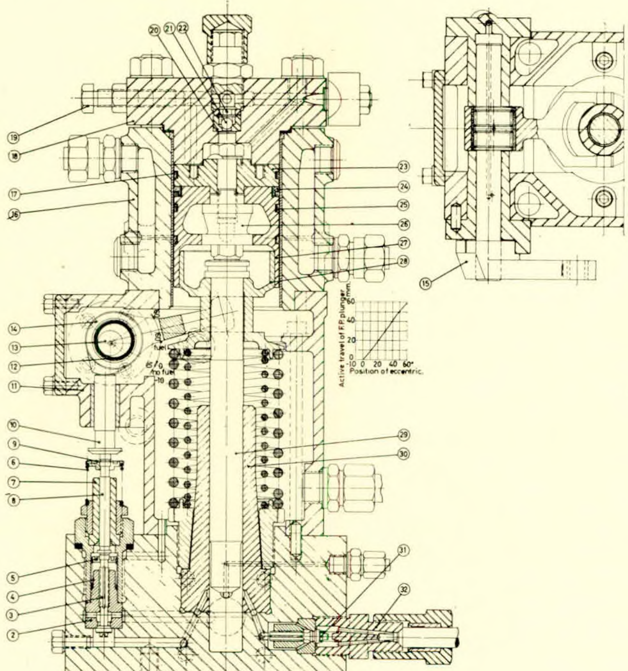
The change-over from riveting to welding in ship construction has raised doubts about the need for brackets at the joints of structural members. In general, the brackets have been reduced in size, but the desire to omit the brackets altogether remains in order to conserve materials and to facilitate the installation of ship's fittings, pipe and trunk systems, and furniture. Before investigating the effects of brackets and the possibility of replacing them by convenient joints of sufficient strength and stiffness, the author studied the literature on the subject and in this report he summarizes forty-nine papers, mentioning the most important conclusions and contents of each. The reports are summarized under five headings, namely plastic design, analysis of redundant frameworks, analysis of brackets and joints, the Glengarnock tests, and civil engineering joints. Some of the conclusions reached by the author after study of the papers are as follows:—1. Further consideration should be given to the application of plastic theory to the design of structural joints. 2. A number of methods are available for a reasonably thorough analysis of the transverse strengths of ships under assumed loading conditions. The actual loading of a ship frame is, however, not known with any degree of accuracy; and, in particular, further investigation into the proportion of the racking and docking loads taken by transverse bulkheads and by framing is required. 3. The Haigh joint with the diagonal stiffener is a mechanically satisfactory type of joint if properly made, and is recommended in several papers. Two of the practical disadvantages associated with this joint are that, if the diagonal stiffener is laminated, premature failure may occur, and that careful fitting and fabrication of the joint is required. These disadvantages may be overcome by a second type of joint described by Haigh. 4. Since the stress distribution determined by Hendry and others indicates a low stress value at the outer sharp corner when a joint is on the point of failure, no advantage will be gained by rounding it off unless a considerable saving in weight, offset against the increased time of fabrication, results. 5. Where a curved knee is used, the most uniform stress distribution is obtained when the radius of the inner flange is about twice the girder depth, and the stress concentration cannot be materially reduced by increasing the radius above this value. 6. Since the ship's side is relatively flexible, permitting rotation of the beam end, there is a point beyond which added stiffness of the knee becomes uneconomical. This optimum stiffness is of the order of stiffness of an equivalent length of beam. 7. The stress concentration at the inner corner of a square knee may be reduced to any value by local reinforcement.—*K. J. Rawson, Naval Construction Research Establishment, Dunfermline; Report No. N.C.R.E./N49. Journal, The British Shipbuilding Research Association, December 1954; Vol. 9, Abstract No. 9681.*

Turbocharged Machinery

With the advent of turbocharged two-stroke Diesel machinery, the perhaps natural interest is in what maximum power can be developed by a single engine. It is fair to say that the exhaust-gas turbocharging of such engines was largely accomplished to meet the higher powers required in tankers of increased carrying capacity. However, with substantially fewer tankers being ordered, and with owners of dry cargo ships seeking to take advantage of the new tonnage regulations, supercharged two-stroke engines are being specified for medium- and quite low-powered ships. Thus, there have been ordered or completed two-stroke engines ranging from 12,500 b.h.p. down to 1,970 b.h.p. in pressure-charged form. The B. and W. engine was, of course, the first to enter service as an exhaust-gas turbocharged unit and its adoption has hitherto been the most extensive. Well, nearly one hundred and fifty such engines are now on order or in service, the continuous rating at present being limited to a value corresponding to an m.e.p. of about 7·1 kg. per sq. cm., or 101lb. per sq. in., compared with about 5·2 kg. per sq. cm., or 74lb. per sq. in., for the non-turbocharged unit. This represents an increase in the specific output of about 35 per cent and, on a horsepower basis, a reduction in weight and space requirements of about 25 per cent and 20 per cent respectively. Owing to the higher mechanical efficiency—89·5 per cent—the fuel consumption is about 3 per cent lower.—*The Motor Ship, January 1955; Vol. 35, p. 411.*

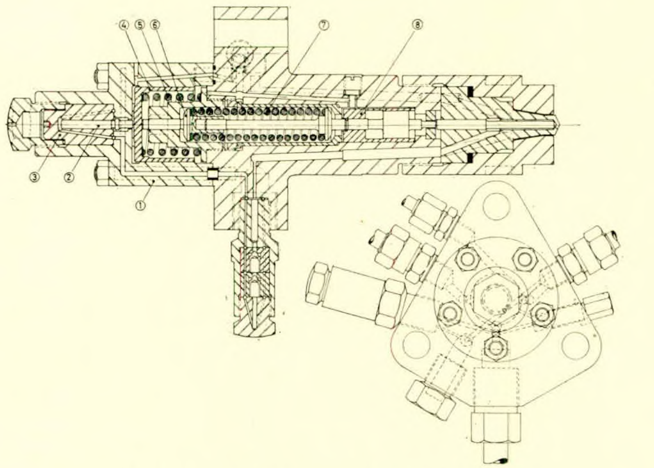
Gas Operated Fuel Injection System

A new self-metering fuel injection system has been developed by Wilson and Kyle, Ltd., for use in conjunction with supercharged Diesel engines running on boiler fuel. This system is being used for the first time on the six-cylinder, 8,000-b.h.p. Harland and Wolff pressure charged single-acting, opposed-piston Diesel engine recently built at the Belfast works. The new type of gas-operated fuel pump, which incorporates a metering arrangement, has been developed in order to dispense with the additional pumps and accumulators. The capacity has also been increased to meet with the much larger volume of fuel required per stroke by supercharged engines. It is an important feature of this system that a camshaft is no longer required. Details of the gas-operated fuel injection pump are shown in an accompanying sectional drawing. Compression gas from the engine cylinder is fed to the pump cylinder (16) through a pipe and through a passage in the cylinder cover (15), the rate of flow being regulated by a screw (19). The gas piston (27) and the plunger (29) have a differential ratio of 10:1, so that if the compression pressure is 500lb. per sq. in. the pressure exerted on the fuel in the lower pump chamber will be 5,000lb. per sq. in. Piston and plunger have a constant stroke and after each delivery are returned to the top position by springs (2). Fuel is delivered from a low pressure surcharge pump through the suction valve (3) to the chamber below the plunger. The forked lever follows the movement of the plunger and the cam portion of the lever acting on a tappet (10) opens and closes the suction valve. As piston and plunger move downwards, fuel is spilled at low pressure through the suction valve back to the supply line until the suction valve closes. Delivery then commences under high pressure through valve (31) to the injector and ceases when the plunger covers the discharge port. Oil trapped below the discharge port acts as a cushion. The fuel is metered by moving the lever (15) on the eccentric shaft (13), thus varying the point at which the suction valve closes and delivery begins. The point at which this valve closes regulates only the quantity of fuel to be delivered; the actual timing of injection being carried out by the fuel valve. On the upward stroke of the piston, the gas in the cylinder is expelled through a ball valve (21) back into the pipe, thus bypassing the regulating screw so that the delivery chamber can be recharged quickly. The gas cylinder has a cooling jacket and oil or water can be used as a cooling medium. Carnea 15 flushing oil has been found most suitable for this purpose. The Wilson and Kyle fuel valve has been developed



Section through Wilson and Kyle gas-operated fuel pump

(1) Base plate; (2) Suction valve seat; (3) Suction valve; (4) Spring; (5) Spring keep; (6) Spring; (7) Sleeve; (8) Rod; (9) Spring keep; (10) Tappet; (11) Housing; (12) Bush; (13) Regulating shaft; (14) Fork; (15) Regulating lever; (16) Cylinder; (17) "Chromard" liner; (18) Cylinder cover; (19) Adjusting screw; (20) Valve seat; (21) Ball; (22) Plate; (23) Piston crown; (24) Piston ring; (25) Piston ring; (26) Thrust piece; (27) Piston; (28) Spring keep; (29) Plunger; (30) Sleeve; (31) Valve; (32) Delivery union.



Section through Wilson and Kyle fuel valve

to work in conjunction with a gas-operated fuel pump on engines where the compression pressures vary widely. Injection begins at the correct moment in relation to the position of the engine piston over the entire engine speed, from starting to full power. The initial opening pressure of the nozzle is controlled by a comparatively light spring (7). The top portion of the upper spring keep (6) presses against a spring casing (4) which is held against the housing (1) by a spring (5), the load of which does not act on the nozzle needle. A sleeve (3) is mounted on the top housing carrying a plunger (2) which is in contact with the spring casing. Fuel oil at delivery pressure is fed through a double check valve to an annulus in the upper portion of this sleeve. The initial opening pressure of the fuel valve is 3,200lb. per sq. in. Under starting and slow-speed conditions, the oil pressure acting on the plunger (2) is insufficient to compress the upper spring (5). As the engine speed increases, the pump delivery pressures will rise in direct ratio to the engine cylinder compression pressures and the plunger will force the spring casing (4) downwards, thus compressing the spring and raising the injection pressure. Conversely, the injection pressure will be powered as the engine speed and fuel delivery pressures are reduced. A small amount of oil is allowed to leak past the plunger so that there will be no delay in reducing the injection pressure as the engine speed is lowered. The pressure spindle which acts on the nozzle needle is, in effect, a piston working in a cylinder (8) forming a dashpot. The cylinder is filled with fuel oil at atmospheric pressure and has a very small spill orifice at the top. When the nozzle needle is lifted by the fuel pressure, the oil in the dashpot is forced through this orifice. This restriction slows down the lifting of the needle and controls the rate of injection, so that initially a small quantity of finely atomized fuel is delivered to the engine cylinder to start combustion. The rate of injection is progressively increased up to the end of the delivery period. At the end of injection the piston and needle are forced sharply downwards by the spring and oil is once more drawn into the dashpot; this oil is taken from the normal nozzle back. By this arrangement the rate of combustion is controlled so that a more even rise in cylinder pressure is obtained; thus not only eliminating Diesel knock and reducing engines noises, but consequently reducing engines stresses.—*The Shipping World, 2nd February 1954; Vol. 132, pp. 170; 172.*

New Naval Engine

The highly supercharged Mercedes Benz marine Diesel engine has a rated output of 1,350 b.h.p. at 1,500 r.p.m., with intercooling, giving an m.e.p. of about 200lb. per sq. in. With normal supercharging the output is 1,000 b.h.p. and as an unsupercharged engine 700 b.h.p. The bore and stroke are 175 mm. and 205 mm. respectively, and there are twelve cylinders in Vee formation. The propeller drive is taken through a reverse reduction gear with a ratio of 1·5 to 1 or 2 to 1. The weight of the engine alone is 6,336lb. A 16-cylinder engine of the same design has been produced and this has cylinders with a diameter of 185 mm., the piston stroke being 230 mm. Its normal rating is 1,870 b.h.p. at a speed of 1,400 r.p.m. and 2,000 b.h.p. at 1,500 r.p.m. The corresponding outputs at 1,200 r.p.m. and 1,000 r.p.m. are 1,600 b.h.p. and 1,290 b.h.p. The consumption of fuel is 170 gr. or 0·37lb. per b.h.p. per hr. at full load. The overall length, width and height are respectively 2,890 mm., 1,400 mm. and 2,030 mm., and the weight without water and lubricating oil is 4,400 kg. or 9,750lb. The engine has two exhaust supercharging blowers of the Brown Boveri type, the supercharging being on the Büchi system. The blowers are mounted on the engine, and there are two intercoolers. The speed governor is of the hydraulically operated type while the fuel injection pumps, of which there are two, are of the Bosch design. The engine may be started electrically or by means of compressed air, and in the former case there are two 24-volt 15-h.p. starting motors, also two 1,800-watt generators. A 20-cylinder supercharged engine is being developed, with a cylinder diameter of 185 mm. and a piston stroke of 250 mm., the total cylinder capacity being 134·4 litres. The compression ratio is 14 to 1 and the weight of this unit, including reduction gear and exhaust pipes, is 4,675 kg. The output is 2,500 b.h.p. at a speed of 1,630 r.p.m. For continuous output at 1,510 r.p.m., at 2,000 b.h.p., the fuel consumption per b.h.p. per hr. is 180 gr. and at the maximum output (half-an-hour) at 1,630 r.p.m., 2,500 b.h.p., the consumption is 200 gr. per b.h.p. per hr.—*The Motor Ship, January 1955; Vol. 35, p. 435.*

Preservation of Bearings

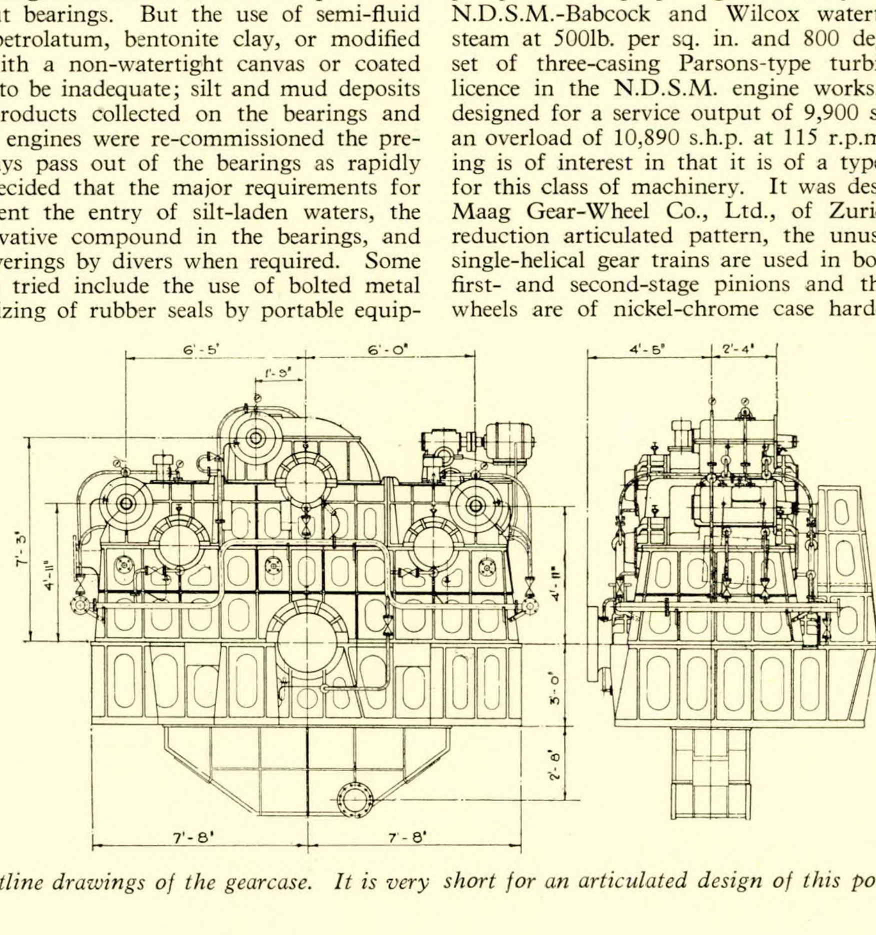
When ships were first being laid up in the reserve fleet after the war, difficulties were experienced in maintaining the water-lubricated bearings in good condition. It was at first believed that prevention of marine growth would be sufficient protection for stern-tube and strut bearings. But the use of semi-fluid preservatives such as petrolatum, bentonite clay, or modified castor oil base soap with a non-watertight canvas or coated glass-cloth seal proved to be inadequate; silt and mud deposits as well as corrosion products collected on the bearings and journals, and when the engines were re-commissioned the preservatives did not always pass out of the bearings as rapidly as expected. It was decided that the major requirements for the seals were to prevent the entry of silt-laden waters, the retention of the preservative compound in the bearings, and easy removal of the coverings by divers when required. Some of the sealing methods tried include the use of bolted metal casings and the vulcanizing of rubber seals by portable equipment. In the method that has been adopted since 1952, sheet-rubber coverings used in conjunction with rubber cement are wound and held around the bearing by metal cables, which are secured by specially designed clips. The seal is easily removed by cutting through the sheet rubber.—*E. A. Bukzin, and J. C. Reid, Bureau of Ships Journal, October 1954; Vol. 3, p. 8. Journal, The British Shipbuilding Research Association, December 1954; Vol. 9, Abstract No. 9,771.*

Dutch-built Turbine Tanker

The Netherlands Dock and Shipbuilding Company, Amsterdam, have recently delivered to Wilhelmsen Lines of Tonsberg, their first turbine-driven tanker. These owners, who before the war had only two old tankers in their large fleet, have expanded the oil-carrying side of their business considerably in recent years, and have taken delivery of several medium-sized units from Clyde and Swedish shipyards. The *Tarim,* as the new ship is named, is considerably larger and has the following principal dimensions:—

Length overall	630ft. 4in.
Length between perpendiculars	600ft. 0in.
Moulded breadth	76ft. 6in.
Depth	43ft. 0in.
Deadweight capacity	25,200 tons
Corresponding draught ...	33ft. 1¼in.
Trial speed	15·9 knots

There are nine main cargo tanks divided by two continuous longitudinal bulkheads into twenty-seven compartments having a total capacity of 1,204,000 cu. ft. As is customary in vessels built by the N.D.S.M. yard, welded construction has been applied, with the exception of a riveted stringer-angle in way of the tanks and two riveted seams on each side of the shell. The longitudinal bulkheads are horizontally fluted, and the transverse bulkheads are corrugated vertically. There are two pump rooms each with three horizontal duplex steam-driven pumps. The propelling machinery consists of a pair of N.D.S.M.-Babcock and Wilcox watertube boilers supplying steam at 500lb. per sq. in. and 800 deg. F. to a single-screw set of three-casing Parsons-type turbines, also built under licence in the N.D.S.M. engine works. The machinery was designed for a service output of 9,900 s.h.p. at 108 r.p.m. and an overload of 10,890 s.h.p. at 115 r.p.m. The reduction gearing is of interest in that it is of a type not hitherto supplied for this class of machinery. It was designed and built by the Maag Gear-Wheel Co., Ltd., of Zurich, and is of double-reduction articulated pattern, the unusual feature being that single-helical gear trains are used in both reductions. All the first- and second-stage pinions and the first-reduction gearwheels are of nickel-chrome case hardened steel. The main



Outline drawings of the gearcase. It is very short for an articulated design of this power gearwheel is of nickel-chromium-molybdenum heat-treated steel. All the teeth have precision-ground profiles. The gear casings are of cast-iron. Castings are used for reasons of rigidity and silence, and also because there is no shortage of skilled iron-founders in Switzerland. The weight of the entire gear assembly is 55 tons. It is to be noted that there is no flexible element such as a quill shaft between the primary and secondary trains. The secondary shafts are located by thrust collars at their after ends. A separate main thrust block is fitted.—*The Marine Engineer and Naval Architect, January 1955; Vol. 78, pp. 24-25.*

American-built Feeder Ship for African Coast

The latest addition to the Farrell Lines' African feeder fleet is the motor vessel *Kpo*. The vessel is used primarily for shuttle service along the African Coast, working out of Monrovia. The draught has been held to a minimum to make it possible for the vessel to enter the small harbours and river ports in that area and link these terminals with Farrell's ocean-going vessels. The design is a development of the type previously worked out by the designers and now in use by the Farrell Lines for handling rubber products from Firestone plantation in Liberia, down the Farmington River, across a 4½-ft. bar and into the harbour of Monrovia. The leading data are:—143ft. length o.a., 135ft. length b.p., 31ft. breadth, 6ft. draught, 8 knots sea speed. Main propulsion is supplied by two Detroit Diesel Engine Division engines, Model No. 6-71A, developing 120 s.h.p. at 1,600 r.p.m. Throttle and clutch control are located in both the engine room and pilot house. These engines drive the twin screws through a reverse and reduction gear with a reduction ratio of 4·45 to 1. A spare engine complete with reverse and reduction gear, a spare propeller shaft, and two spare propellers are carried on board. The port auxiliary plant is a Detroit Diesel engine developing 72 h.p. at 1,800 r.p.m. This engine is provided primarily to operate the hydraulic pumps for cargo handling. However, it can be clutched into one of the two ships-service 2-kW generators. Duplicate bilge and fire pumps also can be driven either from this engine or the port main engine.—*Marine Engineering, December 1954; Vol. 59, pp. 91-92.*

American Liberty Ship Conversions

Award of two contracts by the Maritime Administration totalling $4,149,000 have been made in the Liberty Ship Experimental Conversion Programme, calling for the installation of a steam turbine, a geared Diesel drive, a cargo handling system of revolving cranes, and the lengthening of a Liberty ship by 25 feet. The contract awarded to Ira S. Bushey and Sons, Inc., calls for the conversion and reactivation of the *Benjamin Chew* to a Design EC2-S-8a (steam turbine) and improvement of the cargo handling gear. The contract awarded to the Bethlehem Steel Company, Baltimore, Maryland, calls for the conversion and reactivation of the *Thomas Nelson* to a Design EC2-M8b (geared Diesel), the installation of a new system of cargo handling cranes, and the lengthening of the ship by 25 feet. Main improvements to be embodied in the steam turbine conversion will be the replacement of the 2,500 indicated shaft horsepower steam reciprocating engine, which originally powered the Liberty ships, with a 6,000 shaft horsepower steam turbine reduction gear drive. In this conversion the present cargo handling gear will be improved and modernized by increasing the capacity of the present 5- and 50-ton booms to 10- and 60-ton booms and adding topping winches. In the vessel to be converted to geared Diesel drive by the Bethlehem Steel Company, the main improvements will be embodied in the replacement of the present steam reciprocating engine with modern geared Diesel engines totalling 6,000 shaft horsepower. Revolving gantry cranes will replace the traditional booms of the merchant ship in the Diesel conversion. Five of these cranes, of a new and improved type, will be placed on the ship in a special arrangement, three forward of the amidship house and two aft. All of the cranes will be mounted on tracks. The forward cranes will be capable of being positioned athwartship. The two cranes aft of the amidship house will be able to move under their own power along the length of the vessel on tracks located on each side of the cargo hatches. All the cranes will be of five-ton capacity and capable of full rotation and mobility under load. They will be able to "spot" cargo in the vessel hold and thus place or "break out" specific items from the holds with a minimum of labour within the holds. This advanced cargo handling gear with its new principles of application to ship loading and unloading is expected to offer the possibility of epochal improvement in the ancient problem of cargo handling and the speeding of ship turnaround. The cranes to be used will employ electric and hydraulic drives and will present the possibility for exhaustive studies on the efficiencies of each. In the vessel to be equipped with the Diesel drive, the bow of the ship will be lengthened 25 feet, giving the converted Liberty an overall length of 466 feet. The additional length is expected to improve the sea-keeping qualities of the vessel and with the extra horsepower the sustained speed of the vessel will be increased from about 10½ knots to better than 15 knots. In the steam turbine installation no changes will be made to the bow of the ship, but the shaft horsepower will be increased from 2,500 to 6,000.—*Marine News, December 1954; Vol. 41, pp. 55-56.*

Atomic Propulsion Study in U.S. Shipyards

The development of nuclear propulsion of ships has been spurred by recent authorization of studies and contracts issued by the U.S. Government. The Atomic Energy Commission and the Defence Department, working jointly, have awarded design-study contracts to Newport News Shipbuilding and Dry Dock Company and Bethlehem Steel Company's Quincy plant. These separate design studies are for large nuclear-powered vessels. The research and development work on a reactor suitable for large-ship propulsion has been assigned to Westinghouse Electric Corporation.—*Marine Engineering, December 1954; Vol. 59, p. 95.*

Rectifiers for Floating Docks

Owing to the considerably increased use in recent years of welding for ship repairs and due to the tendency to build ships in docks, multi-point welding plant has lately been increasingly used. By this method, the welders draw the current they require at each individual welding location from a constant-voltage d.c. busbar system. Equipment of this kind not only dispenses with the numerous individual welding sets otherwise required, but is also very economical. Multiple-point welding plant is fed by a d.c. generator or rectifier at a voltage of 65 V and several thousand amperes. Since ships' engine plant cannot be kept running while in dock, ships must rely on a power supply from external sources. The floating dock power plant must, therefore, be capable of giving an adequate supply at all voltages and types of current normally found on ships, current ratings of up to 1,000 amperes being not infrequent. Thus, the field of application of rectifiers in floating docks is twofold, namely the operation of multiple-operator welding equipment and the supply of power to docked vessels.—*G. Lay, A.E.G. Progress, 1954; No. 3, pp. 218-219.*

Electroplating Method for Finding Stress Concentration in Shafts

When a specimen plated with copper which contains some impurities is subjected to reversals of a suitable load, the surface of the specimen changes in colour owing to the flecks produced by fatigue. The variation in the appearance of flecks, which is due to the change of load, is very sensitive at an appropriate load called by the authors the "proper" load; and thus an accurate determination of the surface stress can be made from a study of the flecks produced by that load. The value of the cyclic load below which these flecks do not appear is characteristic of the plating metal and the impurities contained in the plating solution, if the specimens are plated under the same conditions. In this paper, the torsion of shafts with transverse holes is investigated experimentally by this electroplating method, which was recently developed by one of the authors. Maximum stresses were measured and stress-concentration factors found for various diameters of hole. The authors conclude from their research that the usual methods for stress measurements, such as the freezing method and the use of strain gauges, are not suitable for an accurate determination of the stress concentration in shafts with small transverse holes.—*Paper by H. Okuba and S. Sato, read at the A.S.M.E. 1954 Annual Meeting, Paper No. 54-A-88.*

Speed Trials with Decca Navigator

Recently two ships of the Holland-Amerika Line carried out speed trials, using the Decca system. These trials show that, in daylight, the accuracy of the Decca method of speed determination is comparable with that of the measured mile. Each run should be of about ten minutes' duration with photographic registration of decometers and stop-watch taken about every thirty seconds, so as to avoid lane-identification kicks. The final accuracy is not likely to be increased and may even be decreased by longer runs because of the increased likelihood of a change in the value of the set over the period of two consecutive runs. For the same reason it is an advantage to complete each series of runs without undue delay, and experience indicates that less time is wasted in the turn-round using the Decca method than on the measured mile. Six runs —three in each direction—are considered sufficient for an accurate speed determination. The current information will indicate whether one or perhaps two of the runs should be discarded because of great or irregular changes. An accuracy comparable with that of the measured mile can only be obtained when carefully adjusted equipment and the special Decca charts are used.—*J. Th. Verstelle, The Journal of the Institute of Navigation, January 1955; Vol. 8, pp. 41-44.*

Aluminium Anode for Cathodic Protection

Alcoa Anode Plate is a new product developed by Aluminium Company of America. It consists of a special aluminium alloy, and is designed primarily for the cathodic protection of steel ship hulls in sea water. Anodes cut from Alcoa Anode Plate serve as sacrificial electrode, which relieve galvanic attack on the hull. Other potential applications of this new product include the protection of heat exchangers, storage tanks, various types of condensers and small metal-hulled boats. The first installation of this new product was made in January 1953. Thirty-eight anodes cut from Alcoa Anode Plate, were bolted to the hull and rudder post of the *Alcoa Clipper*. The ship was drydocked in August of the same year, at which time it was observed that the steel on the stern frame, rudder post and rudder had retained its coating of paint in its entirety. There was no indication of severe corrosion-erosion attack. This initial inspection revealed that the aluminium anodes were half consumed and were, therefore, providing the desired protection. In January 1954, after one year of continuous service, the anodes had been consumed to the point that they were ready for replacement. This later inspection showed that the same excellent condition still existed around the stern of the ship as revealed in the earlier inspection.—*Marine Engineering, December 1954; Vol. 59, pp. 143-144.*

Turbocharged Diesels for U.S. Coast Guard Cutter

Evaluation tests recently completed on the U.S. Coast Guard cutter *Sundew* reveal many noteworthy advantages gained by increasing marine-engine power through turbocharging. These advantages are of particular interest to operators of Diesel-power craft. Early in 1953 the United States Coast Guard converted two Diesel engines on its ice-breaker *Sundew* from atmospheric to turbocharged units. The output on the two Cooper-Bessemer GN-8 Diesel engines involved was boosted from 700 to 1,060 h.p. This was an increase of 51 per cent in power developed. Performance tests were made with the turbocharged engines at the Straits of Mackinaw in icefloes ranging from 16in. to 38in. thickness. During evaluation tests, the vessel ran into one windrow found to be 8ft. thick. Upon order for full power astern, the *Sundew* backed out immediately from the heavy floe. Prior to turbocharging of its Diesel engines, the cutter had difficulty manœuvring through 15in. of ice. In contrast, the cutter now successfully rams floes 38in. thick. Operating under normal ice-free conditions, the vessel now manœuvres on one engine, with the resultant saving of 30 per cent in fuel consumption. Before the vessel's Diesels were converted to turbocharging, normal operation of the cutter required two engines pulling at partial load. This resulted in less efficient performance than is now possible by operating on one engine pulling near full load.—*Marine Engineering, December 1954; Vol. 59, p. 146.*

Diagrams for Determining Resistance of Single-screw Ship

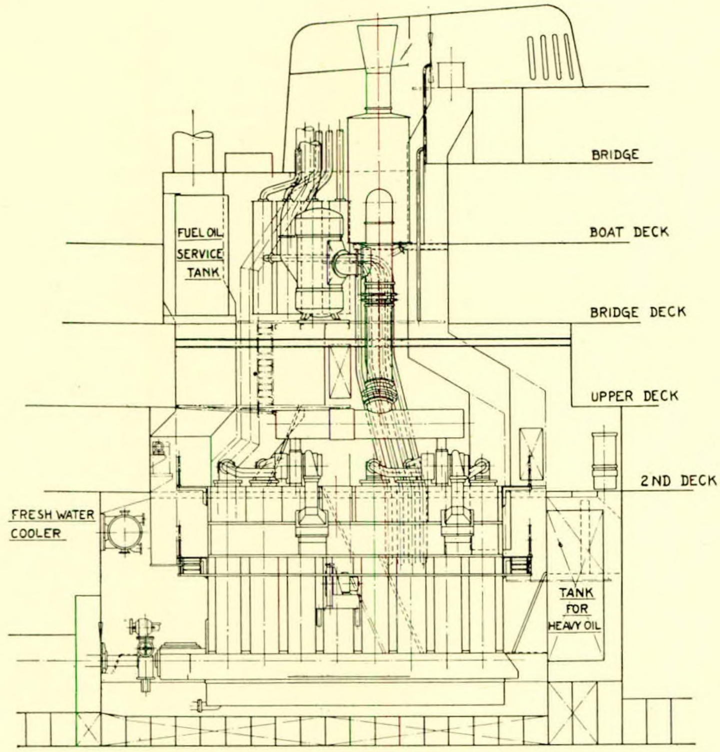
With the aid of the results of many model experiments which have been carried out by the Netherlands Ship Model Basin for upwards of twenty years, attempts have been made at devising a method to determine ship resistance. To this end, investigations have been made with a view to finding out what parameters are of importance for characterizing the ship form and next to ascertaining what influence each of these parameters has on ship resistance. Diagrams are provided for determining the residual resistance as a function of speed. The construction of these diagrams is described. Moreover, some conclusions are mentioned which could be drawn during the investigations. Finally, the total efficiency of propulsion is given as a function of the number of revolutions of the propeller and the ship's length. By these means it is possible to derive the ship's power under service conditions from the resistance of the smooth ship, after various allowances have been taken into account.—*International Shipbuilding Progress, 1955; Vol. 1, No. 4, pp. 179-193.*

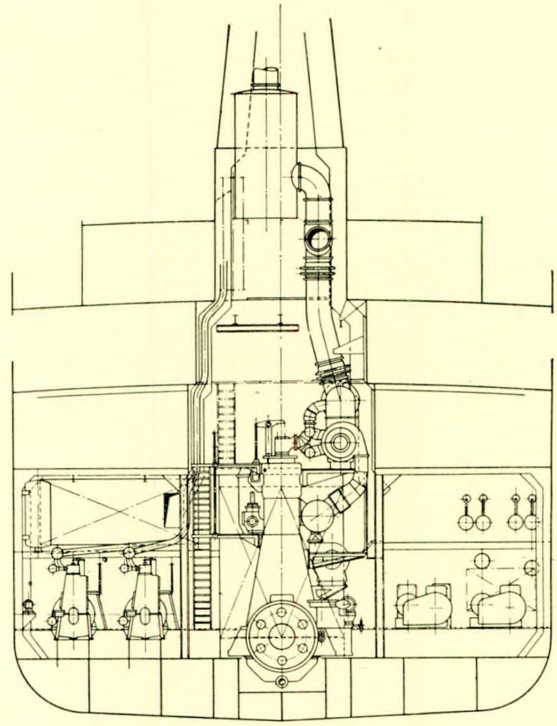
Stresses from Radial Loads in Cylindrical Pressure Vessels

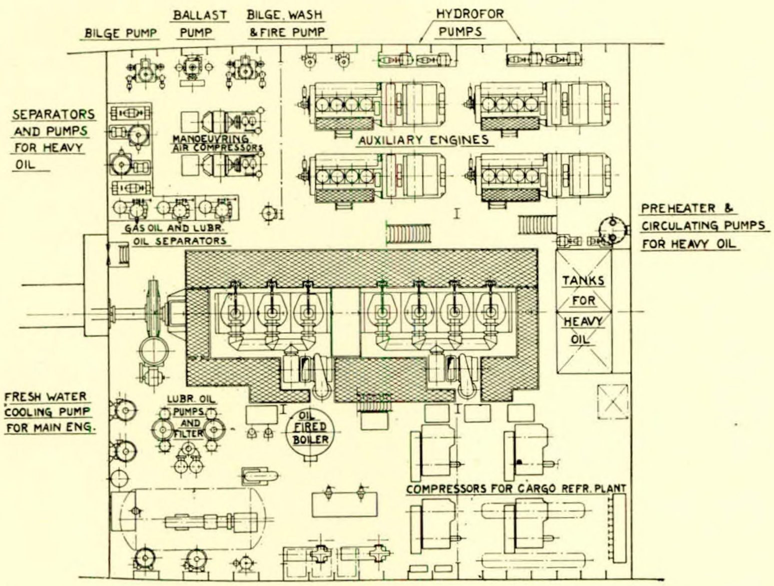
Local forces transmitted to cylindrical pressure vessels through various attachments (nozzles, lugs, clips, etc.) often result in local stresses which cannot be neglected in design. This paper contains design information on deflexions, bending moments and membrane forces caused in cylindrical vessels by radial loads uniformly distributed over a rectangular area. The cylindrical shell wall is assumed to be simply supported at the ends, which means that there the radial deflexions, the bending moments and the tangential displacements are zero. The data published allow the direct determination of maximum stresses and deflexions for design purposes. Comparison with the scattered, available test results reveals good agreement with the computed values. The content of this paper represents part of the results of a research project on Effects of External Loads on Pressure Vessels sponsored at Cornell University by the Pressure Vessel Research Committee of the Welding Research Council. Similar data for loadings by local longitudinal and circumferential moments will be published later.—*P. P. Bijlaard, The Welding Journal, December 1954; Vol. 33, pp. 615-s-623-s.*

Danish Built 17½ knot Fruit Carrying Motor Ship

The motor ship *Cooranga,* which was completed by Burmeister and Wain recently for the Rederi A/B Transatlantic, Gothenburg, is one of the first typical fruit-carrying motor vessels in which advantage has been taken of the employment of a turbocharged two-stroke engine, specially suited by its reduced weight and smaller size for the high-speed, comparatively small ships of this class. These vessels, first introduced by Norwegian owners some twenty years ago, have been very popular with passengers on account of their speed and a feature is made of the accommodation. There are eight single-berth and two double-berth cabins on the boat deck, where are also a smoke-room and verandah, the captain's accommoda-







Engine room plans of the Cooranga

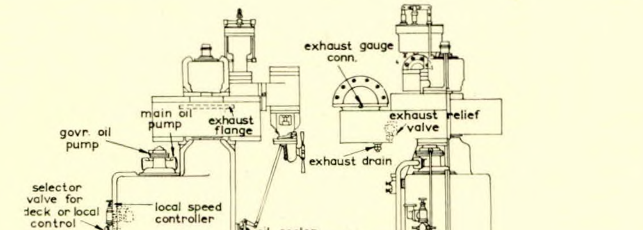
tion being on this deck. The chief characteristics of the *Cooranga* are given below:—

Length b.p.	111·25 m.
Breadth moulded	16·459 m.
Draught	7·112 m.
Corresponding deadweight ...	4,270 tons
Refrigerated hold capacity (grain)	6,260 cu. m.
Machinery	5,750 b.h.p.
Trial trip speed, loaded ...	17·5 knots

Each of the four holds, two forward and two aft of the engine room is divided into ten separate compartments. The cargo is handled at the four hatches by six three-ton and two five-ton derricks, and two two-ton electric cranes, all of the Thrige type. The STAL refrigerating machinery in which Freon 12 is used as a cooling medium is of sufficient capacity to maintain a temperature of − 13½ deg. C. in the tropics. The engine room plans give an indication of the arrangement of the machinery in a modern cargo ship utilizing turbocharged engines operating on boiler oil and it may be noted that the whole of the plant is accommodated in an engine room with a length of slightly more than 50 feet. The propelling unit is a seven-cylinder B. and W. standard turbocharged engine with a cylinder diameter of 620 mm., and a piston stroke of 1,150 mm., the normal output being 6,400 i.h.p. or 5,750 b.h.p. at 150 r.p.m. The exhaust gases are discharged through a boiler with a heating surface of 80 sq. m. and in addition there is an oil-fired boiler with a heating surface of 23 sq. ft. The generating plant consists of three B. and W. 240 b.h.p.-engined dynamos running at 500 r.p.m., the voltage being 220 and the output 160 kW.—*The Motor Ship, March 1955; Vol. 35, pp. 538-539.*

High Capacity Cargo Pump

A new size of high-capacity vertical centrifugal cargo pumps has recently been designed and is now being produced by Hayward-Tyler and Co., Ltd., Luton, Bedfordshire. These pumps, which have a capacity of 1,600 tons of water per hour, are being supplied to Shell Tankers, Ltd., for four of their new 31,000-tons d.w. tankers. With oil tankers increasing in size it was only reasonable to foresee that cargo pumps having a higher rate of discharge than hitherto would be required, and these new pumps are probably the largest yet to be installed in a tanker. In addition it is also desirable to have as quick a turnround as possible. These Hayward-Tyler pumps, four of which will be supplied to each tanker, are driven by Peter Brotherhood 22-in., 800-b.h.p. vertical steam turbines running at 6,000 r.p.m., the running speed of the pump being 1,800 r.p.m. The turbines are mounted vertically above the pump room and are connected to the pumps by means of Hardy-Spicer couplings. These couplings are provided with a splined sleeve which allows a vertical movement of plus or minus 1½ inches. A Drysdale patent gas seal is interposed between the turbine and the pump on the engine room deck. The pump of the single stage, vertically-split volute casing design with suction and discharge branches incorporated in the pump body.



Arrangement of Hayward-Tyler centrifugal oil cargo pump driven by 800 b.h.p. vertical steam turbine

This allows the pump casing cover to be removed and the pump completely dismantled without breaking any pipe joints. The casing is provided with two alternative suction branches, either one of which can be used to suit the layout of the tanker. A venting connexion to the stripping pump is also provided. The impeller is overhung on the shaft and has a double eye entry. It delivers 1,600 tons of water per hour against a discharge pressure of 140lb. per sq. in. The shaft is carried in two bearings. The top main bearing is a deep groove ball bearing which takes thrust and radial loads, and the lower main bearing is of the roller type. Both these bearings are oil lubricated. The pump shaft has also been extended downwards through the impeller to form a journal for an emergency sleeve-type steady-bearing incorporated in the lower part of the pump casing. This bearing is intended only as a precautionary measure and, as it has a large running clearance of 0·010in., can operate only as a bearing in the event of shaft deflexion. During tests no sign of rubbing was detected on this bearing surface. It will be appreciated that, should failure of the ball and roller bearings occur, the provision of this steady-bearing will protect the impeller and casing wearing rings from damage and the pump shaft from distortion. The stuffing box is fitted with a mechanical seal, and a drain hole in the casing allows any liquid leaking through this seal to run to the bilges.—*The Shipping World, 23rd February 1955; Vol. 132, p. 248.*

Gas Turbine for U.S. Minesweeper

The 57-ft. minesweeping boat (MSB) introduced to the United States Operating Forces a completely different type of shaft horesepower prime mover—the gas turbine engine. Designated the Boeing model 502-6, the 160 h.p. engine is the first equipment of its kind to be used in the U.S. Navy. Two of these engines drive a generator for minesweeping purposes through a combining reduction gear. In 1948, the Bureau of Ships became interested in this engine and provided funds to aid in its development. Before the development phase was completed, however, a pressing demand was placed on the Bureau of Ships for a lightweight engine to be installed in the MSB-5 class vessels. The Boeing model 502-6 fulfilled the space and weight requirements, and production was authorized without a prototype. However, taking the engine away from the contractor before the prototype was made meant that certain components still needed development and testing. Because of the results obtained from Boeing tests after the first deliveries of the new engine and early reports submitted by the Operating Forces and Boeing service engineers, the Bureau authorized Boeing to conduct additional development and testing of components with a view toward increasing life and reliability. Developments and tests presently under way include: 1. A second stage governor to withstand higher ambient temperatures. 2. An oil pressure switch to prevent tripping out for reasons other than low pressures. 3. A locking device to prevent rotation of the nozzle box under repeated heating cycle. 4. An exhaust collector heat shield to prevent actuator "O" rings from cracking due to excessive operating temperatures. 5. A new design overrunning clutch to prevent binding during single engine operation. 6. A new design ignitor plug with increased life and reliability. 7. Method to eliminate chronic failure of shear pins in the fuel pump and governor. 8. Two new design starter motors to overcome difficulties encountered in original design. 9. A new design burner liner and crossfire tube to control more adequately the cooling air flow into the nozzle box, and eliminate both liner buckling and crossfire tube bellows failures. 10. An automatic starting system that will bring the engine up to idle speed by depressing a single starter button. The two new design starters are undergoing evaluation testing on the MSB-19 attached to Cominpac. The overrunning clutch phase has been given priority due to recent failures.—*J. A. Culver, Bureau of Ships Journal, January 1955; Vol. 3, p. 17.*

Patent Specifications

Doxford Fuel Pump

An arrangement of fuel pump control which allows reversing to be effected without the use of special cams and other mechanism is shown in Fig. 1. When the eccentric (3) moves

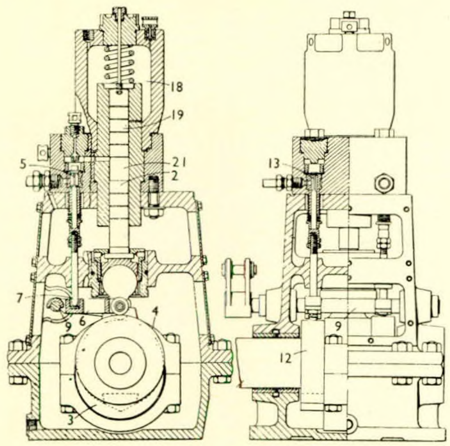


FIG. 1

the fuel-pump plunger (2) upwards during the compression stroke of the engine, fuel is at first returned past the control valve (5) to the supply line. Rotation of the eccentric (4), together with the movement of the lever (6) and the tappet (7), causes the fuel for injection to be trapped in the space (21) between the plungers (19, 2). Continued upward movement of the plunger (2) raises the accumulator plunger (19) against the pressure in the chamber (18). At the correct moment during the compression stroke of the engine, the cam (12) lifts the discharge valve (13) and injection takes place due to the downward movement of the accumulator plunger. As the eccentric (3) continues to rotate, the plunger (2) returns on the down-stroke and fuel is drawn into the pump chamber through the valve (5). The eccentric (3) is timed so that the end of the upward stroke of the plunger (2) coincides with that of the compression stroke of the engine. The cam (12) is in phase with the eccentric and the cam (4) is 180 deg. out of phase, both being symmetrical with respect to their high and low points. In consequence, the pump delivers the same quantity of fuel for any given position of the shaft (9), and gives the same timing for the beginning of injection without regard to the direction of rotation.—*(Patent No. 710,533. W. Doxford and Sons, Ltd., W. H. Purdie, P. Jackson, and J. G. Gunn, Sunderland.) The Motor Ship, November 1954; Vol. 35, p. 356.*

M.A.N. Fuel Injection Pump Control

With the fuel pump illustrated in Fig. 4 the commencement and termination of injection are effected by two levers carried on a common eccentric. When the roller (7) is raised by the cam (3) the pump plunger in the cylinder (4) is lifted by the tappet (5). Fuel is not delivered while the relief valve (15) remains open. With the movement of the lever (6) the pushrod (10) is raised and the lever (12) is turned on its shaft (14). The pressure stop (13) moves down and the valve (15) follows until the spring (17) keeps it on its seat. Injection is then maintained until the pushrod (9) has rotated the lever (11) to bear on the end of the pushrod (19). The forked arm of the lever then raises the pushrod (19) and opens the relief valve (15), so that injection terminates. The amount of fuel injected is determined by the time between the closing of the relief valve and its re-opening by lifting the pushrod (19) and the spindle (18). This period is regulated from zero to the

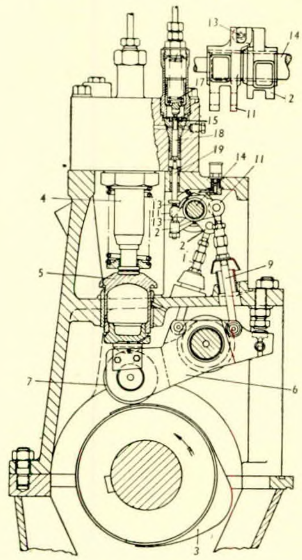


FIG. 4

predetermined maximum by the rotation of the shaft (14). The delivery stroke being advanced at the same time, it follows that an advance in the commencement of injection is obtained in addition to an increase in the quantity of fuel injected.—*(Patent No. 722,626. Maschinenfabrik Augsburg-Nürnberg, A.G., Augsburg.) The Motor Ship, March 1955; Vol. 35, p. 550.*

Removable Grain Anti-shifting Partitions of Metal

Partitions for dividing a hold by means of "Z"-form metal plates are illustrated in Fig. 2. Inside the hold is a

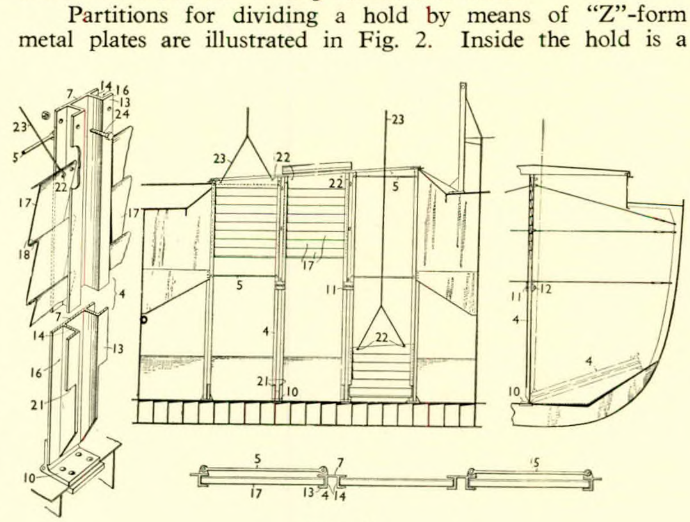


FIG. 2

number of channelled posts (4) and if required, spreader stays (5) are fitted. The lower ends of the posts (4) are provided with hinges (10) and in certain instances the posts are made in two parts with hinges (11) and pins (12). The posts are designed with a back (7), front flanges (13) and webs (14). Sliding in the channels (16) are the plates (17) for filling the spaces between the posts to a depth of 89ft. from the deck downwards. One of the diagrams shows the elongated "Z" section with a flange (18) at each end. For inserting the plates into the channels, the flanges of the posts are cut away (21). Holes (22) receive hooks suspended from lifting tackle (23), so that the ship's winches may haul the plates up one step at a time to fill the spaces between the posts from top to bottom, or as high as necessary. The top plate (17) is secured to the posts by bolts (24).—*(Patent No. 722,546. R., Jos. and Jas. MacGregor, Whitley Bay.) The Motor Ship, March 1955; Vol. 35, p. 550.*

Napier Compound Power Plant

With the arrangement shown in Fig. 2, the engine (A) drives the propeller shaft (B) through gearing. The shaft (D) carries a spider on which are mounted the planet wheels (K) of a differential gear. The ring (L) has internal and external teeth, the latter engaging a gearwheel (E) on the shaft of the friction-type, infinitely variable transmission gear (F). For every setting of the lever (M) a predetermined transmission ratio

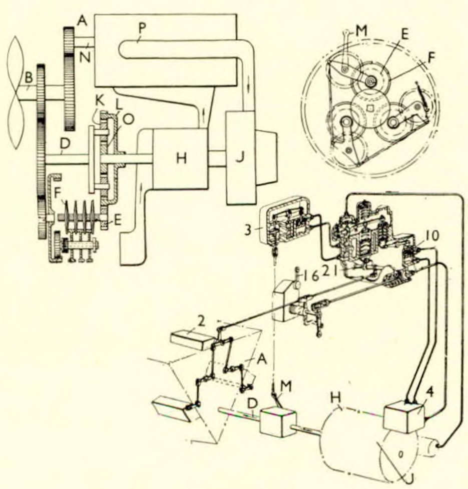
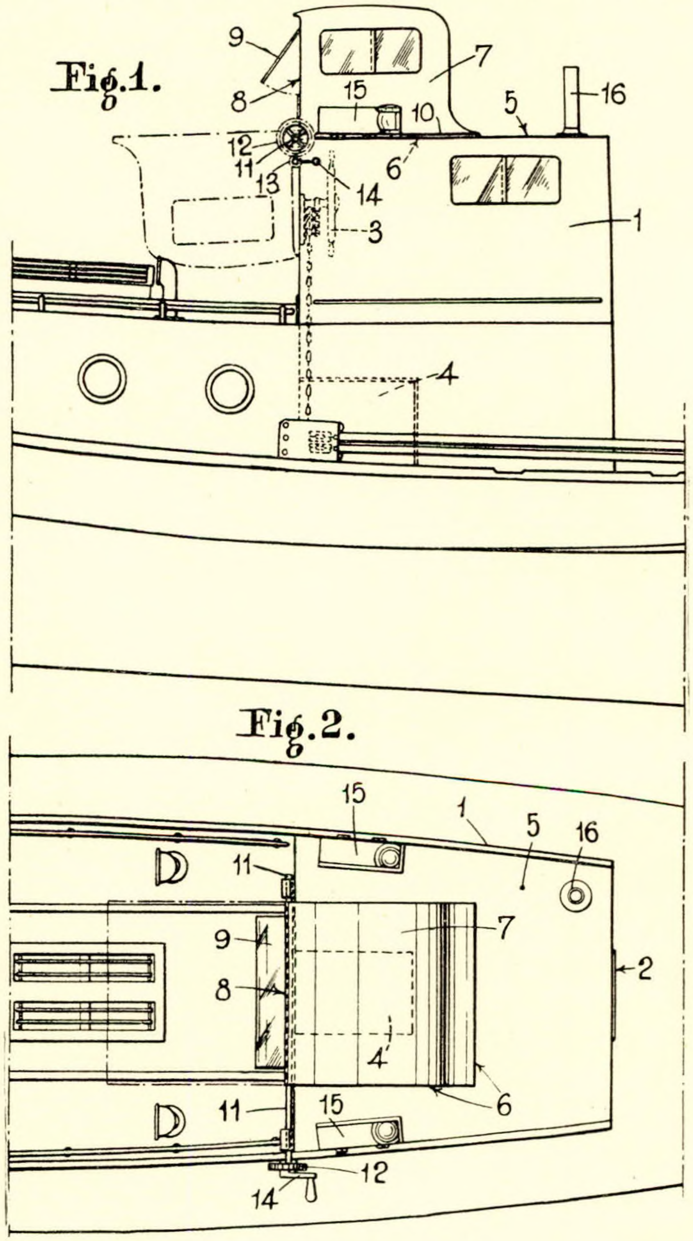


FIG. 2

is established between the crankshaft (N) and the gearwheel (O), which is connected to the shaft of a compresser (H). The compressor forms part of a turbo blower, the turbine (J) being driven by exhaust gas discharged through a pipe (P). The lever (16), shown in the diagram of the control gear, alters the setting of the fuel pumps (2). The boost control cam (21) regulates the servo motor (3), connected to the lever (M), which operates the variable-transmission gear (F). The apparatus includes a speed governor (4) for the turbo blower (H, J). This governor controls the supply of oil under pressure to a cylinder (10). The mechanism alters, simultaneously, the speed of the engine, the quantity of fuel delivered per cycle and the transmission ratio of the drive between the engine and the turbine rotor.—*(Patent No. 718,100. E. E. Chatterton., D. Napier and Son, Ltd., London.) The Motor Ship, February 1955; Vol. 35, p. 512.*

Wheel House of Waterborne Vessel for the Propulsion of Barges

This invention has reference to improvements in waterborne vessels designed for the propulsion of barges and the like and has for its primary object to enable covered deck accommodation to be provided on craft used on inland waterways, such as canals and canalised channels. The invention is particularly applicable to vessels such as tugs constructed mainly for pushing the barges or the like, but is equally applicable to tugs in general. In Figs. 1 and 2 the wheelhouse is made with a permanent roof (5) at a height sufficiently low to be within the necessarily limited head room, and a rectangular opening (6) is made at the forward end immediately above the wheel and the steersman's platform. On top of the permanent roof and covering the opening is a square conning tower, and this conning tower has side and aft walls with necessary port-holes or windows, its own roof, and a forward flat front and observation wall (8) with windscreen opening (9). The height



of the conning tower roof is such as to give ample head room for a steersman of normal height standing on his platform (4) with head and shoulders through the permanent roof (5), so as to permit an unobstructed view forward over a push-propelled barge or the like. Any suitable means may be incorporated to enable the tower to be swung into its position of use or out of action, and in this respect a rotatable hinge-pin (11) fast with the forward edge of the tower may be provided with a toothed wheel (12) geared to a smaller pinion (13) mounted on the wheelhouse and having an operating handle

(14). Thus when a bridge is about to be encountered, the tower is swung down, much the same as smokestacks on larger craft, and the steersman then remains erect being fully sighted up to the last moment, when he stoops as is the practice with exposed steersmen.—*(British Patent No. 723,378, issued to E. G. Jones. Complete specification published 9th February 1955.)*

Anchor

This invention relates to anchors of the hinged twin fluke type and has for its main object to provide an improved anchor of this type which is of relatively lightweight construction particularly suited to manufacture by fabrication methods from sheet steel or light alloy material with a consequent reduction in production costs, and which, at the same time, is capable

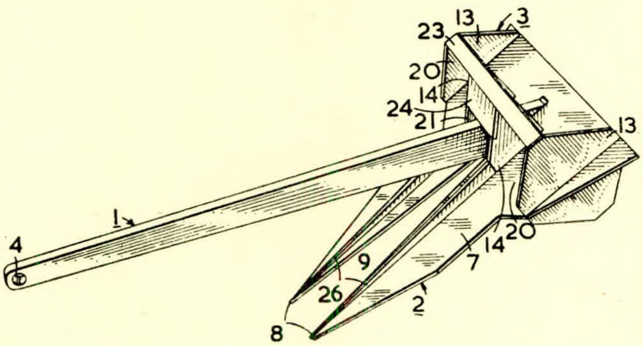
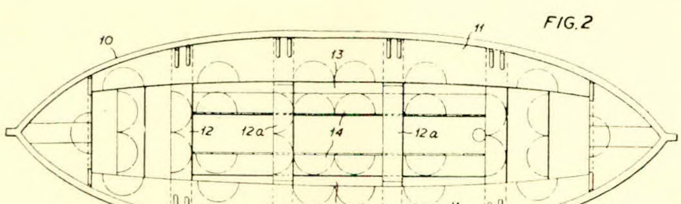


FIG. 5

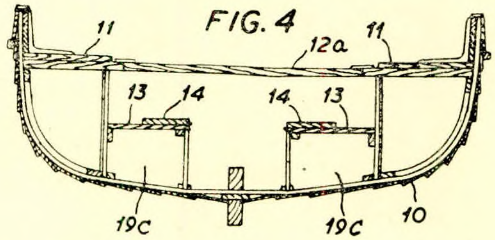
of withstanding loads normally requiring a much heavier and larger design of anchor. Another object of the invention is to provide an improved anchor which is of compact design such as to be particularly suited for use on flying boats and high-speed marine craft where restricted weight and stowage considerations are of the highest importance. Fig. 5 is a view of the anchor in perspective.—*(British Patent No. 724,005, issued to National Research Development Corporation (Inventor: S. T. Cope). Complete specification published 16th February 1955.)*

Improvements in Ship's Lifeboats

There is an urgent requirement at the present time for a lifeboat in which there is provided an open space in which injured passengers or stretcher cases can lie down. An object of the present invention is to meet this requirement without adversely affecting, at least to a material extent, the passenger accommodation of a lifeboat. Another object of the invention



is to facilitate the speedy and orderly embarkation of passengers. According to the present invention there is provided an unobstructed space for the accommodation of prostrate passengers, said space extending in the direction of length of the boat. Referring to Figs. 2 and 4, in the amidships section of the boat there are provided, on opposite sides of the boat, two longitudinal extending or fore-and-aft lower seats or side benches (13). The thwarts at this part of the boat have a portable or removable section (12a). Hinged to the inboard edge of each of the lower seats or inboard edge of each of the lower seats or benches (13) is a deckboard (14). In an emergency or at any other time when it is desired that some of the passengers



should lie prostrate then the portable thwart sections (12a), or one of them, is removed so as to leave a clear space amidships and the deckboards (14) are folded inboard, on to supports, so that they meet at the centre of the boat.—*(British Patent No. 723,645, issued to I. R. Fleming and F. E. Fleming. Complete specification published 9th February 1955.)*

M.A.N. Landing Gangway

A safety arrangement which separates a gangway from the ship automatically if the distance should exceed a certain length is illustrated in Fig. 1. The ship (3) is connected with the quay

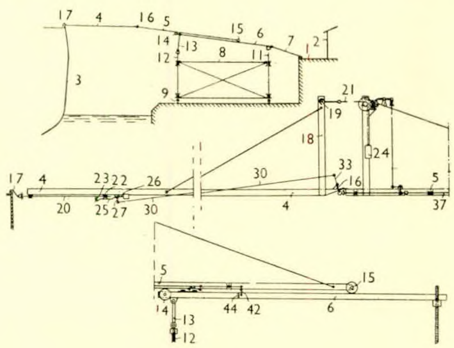


FIG. 1

(1) or the Customs house (2) by the portions (4, 5, 6, 7), one of which (6) is supported on a frame (8) transported by rollers (9) on rails. The portion (4) has a support (18) at the top of which is a hook (19) on a cable (21) to which is attached a counterweight (24). Adjustable lifting spindles (11, 12) are provided, together with a link (13). One part of the gangway (5) slides on the portion (6) by means of rollers (14, 15). An articulated member (16) is provided and the portion (4) engages the side of the ship by a hook (17). If the ship moves away from the quay, the lever (42) strikes the stop (44) so that the rod (37) is displaced and rocks the lever (33). The rod (30) moves the three-armed lever (25, 26, 27) out of engagement with the projection (23). The rod (20) is drawn out of its guide (22) and moves the hook (17) upwards so that the connexion between the ship and the gangway is broken.—*(Patent No. 719,178. Maschinenfabrik Augsburg-Nürnberg A.G., Nürnberg.) The Motor Ship, April 1955; Vol. 36, p. 37.*

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Marine Engineering and Shipbuilding Abstracts

Volume XVIII, No. 6, June 1955

New Welding Process

Higher welding speeds, better quality welds, and lower operating costs are the main advantages of a new consumable-electrode, inert-gas arc-welding process. It is claimed that field tests on the new process disclose at 15 to 20 per cent faster welding speed at a 25 to 50 per cent reduction in costs over other manual and semi-automatic inert-gas methods. Key to the new process, which uses ordinary welding grade argon to shield the arc, is a new coated wire. This wire produces best results when used with a newly designed welding gun, wire control and a constant potential 500-amp., d.c. arc welder. The new process produces welds that can be painted without cleaning. There is virtually no spraying or spatter. Nor is there any slag-covering on the weld. The new process may be used on mild steels, with or without normal mill scale, on thicknesses from $\frac{1}{16}$-in. up. It can be used as either a semi-automatic process—where an operator holds the gun; or in automatic processes—where the gun is clamped into position and the work moved, or where the wire is fed through an automatic head. Primarily intended for horizontal and flat position welding, the new process is not yet developed for vertical or overhead welding. The new process, in contrast to most other processes of this type, uses straight polarity.—*The Welding Journal, December 1954; Vol. 33, pp. 1187-1188.*

American-built Tanker

The *Delaware Sun* is the first of a class of four sister ships for Sun Oil Company operation. The general arrangement of the *Delaware Sun* is in keeping with best modern efficient tanker practice. She is of the three-island type—poop, set-in bridge, and forecastle; with raked stem incorporating a bulbous bow, cruiser stern and single continuous steel deck. Propulsion is by a single screw with the conventional location aft for propelling machinery, consisting of geared turbines and two watertube boilers. The main cargo pump room is located at the aft end of the cargo oil space. The cargo tank space is divided transversely into two wing tanks and a centre tank by two longitudinal oil-tight bulkheads. Longitudinally there are ten subdivisions by transverse oil-tight bulkheads, making a total of thirty cargo-oil compartments. Generally, the tanks are 39ft. 4in. long and the overall length of the oil space is 380ft. The principal characteristics of this vessel are: 641ft. o.a. length; 615ft. b.p. length; 84ft. breadth moulded; maximum draught 34ft.; 30,200 tons d.w.; 16·5 knots speed loaded. The vessel is welded throughout, with the following exceptions riveted: Two seams of bottom shell, upper and lower seams of bilge strake, lower edge of sheer strake, seam of stringer strake, and two seams of deck plating. In addition, the outboard edge of the deck stringer is welded to a $9\frac{1}{2}$-in. by $1\frac{1}{4}$-in. gunwale plate which is double riveted to the sheer strake. The vessels are intended for the transportation of a single grade of petroleum cargo; however, the cargo-oil system is so designed that several grades may be handled properly. The pumping time required to discharge the vessel is between ten and fourteen hours, depending upon the required discharge pressure. The vessel has two boilers of Babcock and Wilcox make located on a flat at the after end of the main engine room space and above the line shafting; the firing aisle is located at the forward end of the boilers. Each boiler is of the two-drum, vertical bank battery-set type fitted with superheater, economizer and internal boiler-drum desuperheaters. Each boiler has a unit heating surface of 7,550 sq. ft. including water-wall surface, and each economizer has 2,450 sq. ft. of surface; the furnace space is water cooled on the sides, roof, rear wall and floor, and the front wall is refractory lined. The boilers are designed for continuous operation at 600lb. per sq. in. pressure and 835 deg. F. at the superheater outlet. Normal rating of each boiler is 53,750lb. of steam per hr. Maximum rated capacity of each boiler is 78,750lb. of steam per hr. at 600lb. per sq. in. pressure and 835 deg. F. at the superheater outlet. The propelling machinery for the *Delaware Sun* comprises a Westinghouse cross-compound turbine unit consisting of high pressure and low pressure turbine elements. The normal rating is 13,500 s.h.p. at a propeller speed of 100 r.p.m. The total shaft horsepower is equally divided between the high pressure and low pressure turbines, which run at 6,310 and 4,230 r.p.m.

respectively, at normal power. The Westinghouse reduction gear that transmits the power of the turbine to the propulsion shaft is of the two-pinion, articulated, double-reduction type.—*Marine Engineering, December 1954; Vol. 59, pp. 74-75; p. 94.*

Light Tunnel

The light tunnel at the Material Laboratory, New York Naval Shipyard, is one of few such facilities in the U.S.A. A light-tight chamber 200ft. long, 12ft. wide, and 10ft. high, it is divided into four rooms of unequal size by three transverse walls containing apertures for the passage of radiation. Running down the centre of the chamber is a track for moving photometric apparatus or radiation sources along the length of the tunnel. One of the inner rooms contains the optical attenuator, which is used to diminish to any desired value the apparent intensity of a visible, infra-red or ultraviolet source without appreciably diffusing the radiation or changing its spectral character. This device can make a light source which is just a few feet away appear as it would if viewed through a vacuum at distances up to thousands of miles. Because of its ability to simulate various operational situations, the attenuator is used when there is no complete body of theory to permit prediction of how new equipment will function. Precise control is necessary because communications systems ultimately pass information to a human observer and there is no complete theory covering audio-visual psychophysiology. One assignment given the light tunnel was to determine the range of communications system composed of an infra-red searchlight for sending Morse code and an infra-red viewer through which the receiving signalman viewed the transmitter. By using the attenuator to simulate various distances, it was found that the range of the apparatus depended considerably on the skill of the receiving signalman. The results obtained with the attenuator could not have been predicted on the charted values for reception of visual signals. Also, its use explained why apparently inconsistent results had been obtained in shipboard trials. Aside from its work with the optical attenuator, the light tunnel is valuable as a photometric chamber for measurements of large and small searchlights, navigation lights, and other radiation sources. It is also suitable for conducting experiments in which the observer's eyes must be adapted to various ambient light conditions, and for providing backgrounds of varying brightness and colour. In most of this work, it is required that the observer's eyes be dark-adapted and the signals be of such low intensity that the slightest light leak into the tunnel would distort the results. However, it also has been necessary on occasion to provide backgrounds which match in colour and brightness such conditions as a clear, blue sky or a dense, daytime fog.—*Bureau of Ships Journal, January 1955; Vol. 3, pp. 19-20.*

New Principles for Calculation of Welded Joints

For the calculation of the strength of welded joints, use has been made of a critical surface of the ultimately allowable values for the average stresses "*p*" either in the smallest longitudinal section (throat section) or cross-section of the weld. In statically loaded structures the shape of the critical limit curve for loads perpendicular to the longitudinal axis of the weld can be deduced from the experiments of the Netherlands Foundation for Applied Scientific Research (T.N.O.). For the shape of the surface of limits, a surface of revolution has been assumed, of which the meridian section forms the above-mentioned limit curve. Starting from the average stress in the throat section of the weld no theory of rupture or plastic flow has, up to now, given a satisfactory explanation of all phenomena of rupture observed in end fillet welds at the moment of fracture. This is to a great extent caused by the uneven distribution of stresses. In comparison with Kist's method of calculation in use at present in the Netherlands, the principles here presented allow not only some simplification but above all economy in the case of weld sections predominantly loaded in shearing or compression. In dynamically loaded structures a symmetrical surface of revolution has been adopted as the shape of the critical surface of allowable effective stresses. This effective stress is to be obtained by multiplying the numerically greatest value of the oscillating stress by a coefficient γ, which is consistent with the fatigue diagram of the material and the type of weld. The values of γ are given both for butt and fillet welds for two frequency ranges of load oscillations. A full explanation based on both theory and experimental work is given.—*C. G. J. Vreedenburgh, International Shipbuilding Progress, 1955; Vol. 1, No. 4, pp. 200-223.*

Effects of Blade Section Shape on Model Screw Performance

The main purpose of the work described in this paper was to study some effects of variations in blade-section shape on the performance of model screws by analysis and experiments on a group of screws in which all the other design features remained unchanged. To some extent this might be described as an attempt to make comparative tests on propellers in a water tunnel analogous to experiments made on aerofoils and wings in a wind tunnel. Such comparative tests are not necessarily a good basis for decisions on ship-screw design problems, and any conclusions which may be drawn from them can only be applied to ship-screw design with reservations. Three points of doubt, among many others, may be quoted; first, these model tests were made in uniform flow conditions while ship screws operate in irregular flows; second, equivalent ship screws incorporating different blade section shapes will often have different blade-area ratios; third, merchant ship screws often operate in conditions in which cavitation occurs but does not affect the screw performance, and in such cases the extent and nature of cavitation damage, which depend strongly on the form of cavitation, can be significantly affected by blade-section shape even when performance is unaffected. With these reservations in mind, some tentative conclusions are stated as follows: (*a*) The design method given by Hill, and his form of curvature and tip loss corrections, are reliable. (*b*) The lift-camber relation quoted by Hill for a parabolic mean line requires further investigation. (*c*) Under non-cavitating conditions, provided that the blade sections used have good lift and drag characteristics, variations in section shape have little effect on efficiency. However, it is possible that some gains can be achieved by using sections near the root with their maximum thicknesses forward of mid-chord, and also by avoiding high cambers at these sections. (*d*) It is doubtful whether adhering to the theoretical optimum radial thrust distribution is of importance. (*e*) Under cavitating conditions changes in blade-section shape can cause significant differences. In particular, constraints imposed by geometrical conditions, such as flat face sections or fixed camber ratio, are harmful. (*f*) Comparison of thrust breakdown conditions, and thrust and efficiency reduction factors, suggest that "shockless entry" designs are inferior to those with a small amount of positive incidence.—*Paper by A. Silverleaf and T. P. O'Brien, read at a meeting of the North East Coast Institution of Engineers and Shipbuilders, 28th January 1955.*

Electrolytic Descaling

Electrolytic descaling is a rapid and possibly less expensive method than manual and mechanical methods for removal of heavy scale in large tanks. It has the additional advantage that no metal is removed from the tank surfaces, and the descaled metal is left in a passive condition less subject to rapid rust formation. The process is applicable to large floodable voids, large structures which can be immersed in sea water, and small parts which can be immersed in tanks containing the electrolyte. It has particular application in removing rust from cargo tanks, being cleaned for converting to light fuels such as aviation gasoline where contamination must be avoided. Removal of oil and sludge is a prerequisite prior to descaling operations. Essentially demountable grid anodes arranged in curtains roughly conforming to the geometry of the tank are energized by low voltage high current sources such as d.c.

welding generators. The tank acts as the negative ground for the electrical circuit. The sea water electrolyte which fills the tank completes the circuit. The electrolysis reactions which take place during descaling electrochemically reduces the iron oxides at the metal interface and causes the scale to slough in sheets. When operating at current densities of half to three-quarters amp. per sq. ft., descaling can be carried out in about thirty to forty hours. Draining and flushing the tanks complete the operation.—*F. E. Cook, H. S. Preiser, and J. F. Mills, Journal of the American Society of Naval Engineers, 1954; Vol. 66, No. 4, pp. 1005-1049.*

Norwegian Motor Ship

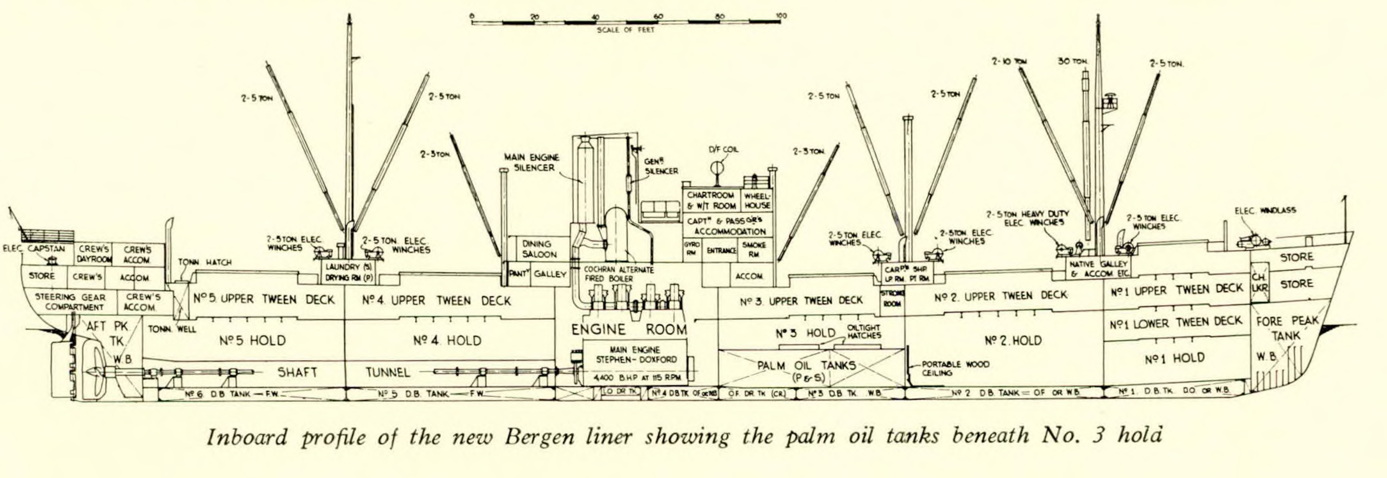
The single-screw motorship *Irma* has been completed by Alexander Stephen and Sons, Ltd., for Det Bergenske Dampskibsselskab (the Bergen Line) of Norway. She is the first of three similar vessels, one of which will be transferred to Fearnley and Eger of Oslo, and will be employed on the service between Scandinavia and West Africa which is maintained jointly by the Bergen Line, Fearnley and Eger and Rederi A/B Transatlantic. The new ship has been built to the highest class of Det Norske Veritas and fulfils all the requirements of the International Convention for the Safety of Life at Sea. She has the following principal particulars:—

Length between perpendiculars ...	400 feet
Moulded breadth...	57 feet
Moulded depth to shelter deck...	35 feet
Loaded draught	23 feet
Deadweight capacity	6,530 tons
Gross tonnage	4,441 tons
Power	4,400 b.h.p.
Speed	14½ knots

The *Irma* is an open shelterdecker, but she has been built with scantlings as for a closed shelterdeck vessel which would allow her to carry an increased deadweight on a deeper draught. There are seven watertight bulkheads dividing the ship into five separate cargo spaces, each with upper 'tween decks. The raised fo'c'sle has a cargo space beneath it and there is an additional lower 'tween deck in No. 1 hold. The lower part of No. 3 hold takes the form of four palm oil tanks which are closed by large oiltight hatches and which can be used, if necessary, for general cargo. There is a pipe tunnel running fore and aft on the centre line between the tanks. The *Irma* is propelled by a four-cylinder Stephen-Doxford opposed piston two-stroke engine having a cylinder bore of 670 mm., a combined stroke of 2,320 mm. and an output of 4,400 b.h.p. at 115 r.p.m. The engine is of the normal welded plate construction, having the controls at the forward end and two lever-driven scavenging air pumps driven by the centre crossheads of the after pair of cylinders. A Michell thrust bearing is incorporated at the after end of the crankshaft and a Doxford Bibby Detuner is arranged at the forward end. The engine is cooled by distilled water and the essential auxiliaries consist of two Drysdale Rotary Centrex 200-ton jacket and piston cooling water pumps, a 270-ton Drysdale Upright salt water pump and two Drysdale Vertoil 45-ton lubricating oil pumps, all these, of course, being motor driven. Starting air for the main engine is provided by two Reavell vertical three-stage compressors. The engine is arranged for running on heavy grades of fuel and three De Laval VIB 1929 purifiers and clarifiers have been provided. These are installed in a group together with two similar units for lubricating oil and Diesel oil treatment. Auxiliary power is provided by three 180-kW, 500 r.p.m. Ruston and Hornsby-engined generators for which a separate 40-ton salt water circulating pump is provided. Steam for heavy fuel, cargo tank, galley and accommodation heating is provided by a Cochran vertical auxiliary boiler fitted with Wallsend oil-burning equipment. This boiler is contained in the port after corner of the engine room. There is also an alternately-fired waste heat/oil Cochran boiler which is carried at passenger deck level in the casing. Equipment in connexion with the boilers includes two Weir direct-acting feed pumps, a 10-ton Caird and Rayner evaporator and distiller and a Carruthers feed filter.—*Marine Engineer and Naval Architect, February 1955; Vol. 78, pp. 41-43.*

Experiences in the Stabilization of Ships

Following upon the satisfactory reports from the owners of the *Chusan*, the Cunard Company decided to install a Denny-Brown stabilizer on a ship in service, and in 1950 a contract was placed for equipment for the *Media*. The fin design followed generally the lines of the fins installed in the *Chusan* which had been in service twenty-one months prior to trials of the *Media* installation. All possible trials were carried out satisfactorily on the *Media* prior to handing over, and the report that she had lost one of her stabilizer fins in mid-ocean after three days in operation on her first voyage came as a great shock to all concerned; design and construction of similar installations in hand was at once halted. Examination in drydock of the broken finshaft revealed the design error, and an error in welding technique. The shaft had fractured at the place where the shell plating of the fin had been welded to the inner part of the radius of the half coupling, and an examination of the fracture revealed an obvious fatigue failure initiated by a crack developing between the fin cover plate and the shaft coupling. The design error was the reliance on welding a thin section to a heavy section, without the issue of a rigid specification of the welding technique. The finshafts on the *Chusan*, of similar design, possibly with more careful welding, operated satisfactorily for three-and-a-half years, and were replaced as a precautionary measure by the improved design of outboard shaft and fin. A stabilizer of approximately the same size as those fitted in cross-Channel vessels, and generally similar in design to the large number running trouble-free was installed in the French liner *Maroc*, and it was naturally a shock when it was learned that this



Inboard profile of the new Bergen liner showing the palm oil tanks beneath No. 3 hold ship had lost one of her fins during a storm. Examination of the broken shaft showed that the fin had broken off half way along the tapered part of the outboard part of the shaft—there was no obvious explanation as to the cause of this fracture—and to ensure as far as humanly possible no risk of a subsequent failure in a foreign-owned ship, both fins were replaced by a special design—fin and shaft being forged as one unit. The next mishap came with the loss of one fin off the Bergen steamship *Leda*. Here defective material was suspected, but unfortunately, by an error, the recovered part of the shaft found its way into the melting shop before tests had been carried out; that the design, etc., was sound is indicated by the fact that, until the installation was restored to its original condition, the *Leda* was very satisfactorily stabilized with only one fin. The stabilizer for the Orient liner *Oronsay* was partially constructed prior to the mishap on the *Media*. The outboard finshafts were therefore scrapped and shafts similar to the redesigned *Media* fin were constructed and installed. All trials as usual were satisfactorily carried out, and on service this ship provided the best record so far available of a stabilizer bringing the roll down to a figure lower than was thought theoretically possible. This stabilizer was in service for eighteen months, during which time the gear operated without a hitch for 1,239 hours, but on 16th June 1954, one fin was lost, and seven days later the other fin was lost, due to failure in fatigue of the eight bolts in the finshaft couplings; moreover, from the same cause, the *Viet-Nam* lost one fin after sixteen months in service. Complete examination of the bolts received from the *Oronsay* appeared to demonstrate that they had never been really hardened up when fitted under uncomfortable conditions of dry-dock. All ships at sea are now having the coupling bolts replaced with Nitralloy steel nitrided bolts, as undoubtedly while the original bolts were theoretically of ample strength, their fatigue life in salt water was underestimated. All bolts now fitted are tightened to a predetermined stressed length.—*Paper by Sir William Wallace, read at a meeting of the Institution of Engineers and Shipbuilders in Scotland, 11th January 1955.*

Installation of Machinery Aft

A feature of post-war cargo ship design has been that several proposals have been made for the installation of machinery aft in vessels of sizes up to 12,000 tons deadweight and some ships have been built in which this arrangement has been adopted. The advantages to be gained from installing machinery aft may be summarized. First, there is a considerable simplification of structure and machinery, as no need exists for a shaft tunnel and the length of tail-shafting is greatly reduced. Both these features have a beneficial effect upon the deadweight displacement ratio of the ship and upon the first cost of the vessel, not to mention a saving in maintenance due to the avoidance of a long shaft and the tunnel steelwork. Secondly, there is what is perhaps the most attractive feature to the shipowner, namely that the midships portion of the ship is free for the carriage of cargo. With the current trend towards fairly high speed tramps and cargo liners, the ends of the ship have become increasingly fine and this leads to holds which are of an awkward shape and of much lower earning ability per cubic foot than the more or less rectangular space amidships. The disadvantages are not many, but do require some careful consideration in the initial stages. The main disadvantage and, in fact, the only real one, is that, due to the light machinery room being aft, the trim loaded with a homogeneous cargo may present difficulties in the design stages, as the ship may exhibit a tendency to trim by the head. A careful disposition of weights and of buoyancy can take care of this, but in the author's opinion, it does produce a limitation for fast ships insofar as the longitudinal centre of buoyancy of such ships is in a relatively aft position compared with slow ships, and thus the trim problem is accentuated. It should be emphasized that for the normal cargo vessel with a speed up to approximately 15 knots, and 470ft. in length between perpendiculars, this problem can be solved quite simply in the design stages, but for ships operating at relatively high speed/length ratios, this may indeed be a limiting criterion forcing the installation of machinery amidships. A further feature of the "engines aft" ship is that of the altered longitudinal bending moment characteristics in various conditions. Considering, first, the loaded condition, it is clear that as the engine room is lighter per cubic foot of volume than are the holds, there must be an increase in sagging moment and a decrease in hogging moment in the machinery aft ship *vis-à-vis* the machinery amidships vessel. In general, sagging moments are more important than hogging moments and, therefore, consideration must be given to this effect on the longitudinal bending moment. This affects mainly the buckling stresses in the deck plating amidships and, the compressive deck stress being greater, the resistance to buckling must be higher in the engines aft ship. This leads to a design feature, consistent with the longitudinal coamings, namely, of introducing longitudinal stiffen-

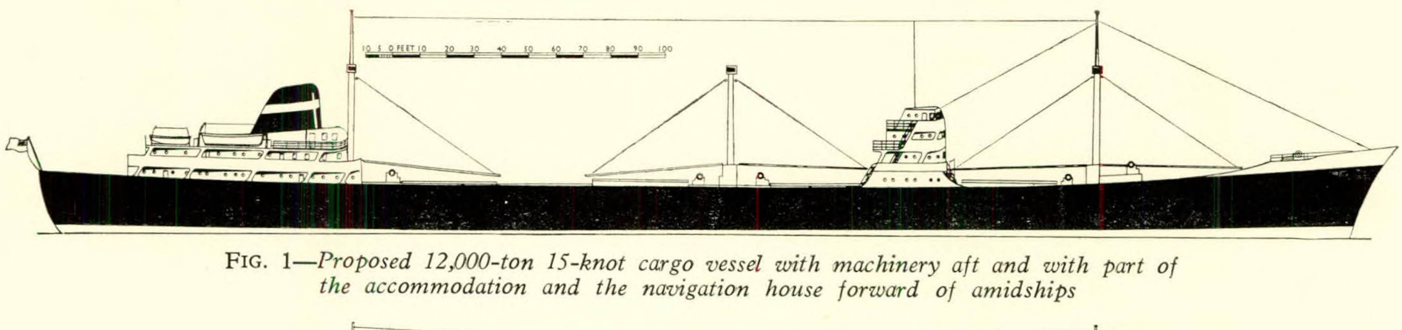


FIG. 1—*Proposed 12,000-ton 15-knot cargo vessel with machinery aft and with part of the accommodation and the navigation house forward of amidships*

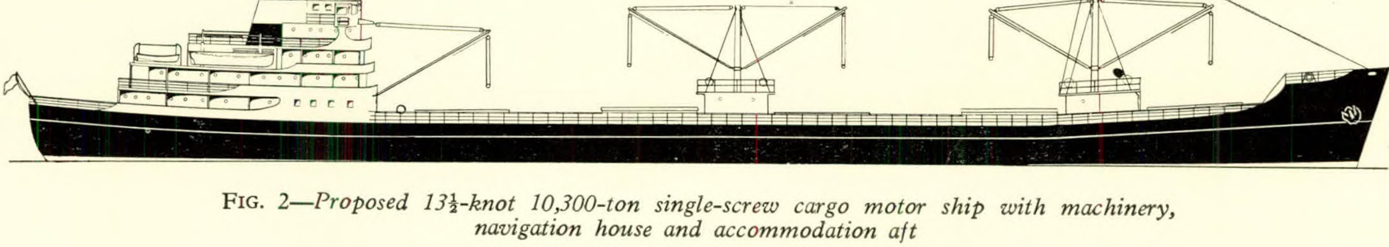


FIG. 2—*Proposed 13½-knot 10,300-ton single-screw cargo motor ship with machinery, navigation house and accommodation aft*

ing in the upper deck. This has been done in a number of cargo vessels and is undoubtedly most successful from the points of view of lightness and resistance to compressive stresses. —*E. C. B. Corlett, The Motor Ship, February 1955; Vol. 35, pp. 483-485.*

New Hatch Tent

A new type hatch tent has already proved itself exceptionally economical in use, according to purchasers of these tents who have carefully tested them. Chief advantage of the tent, according to users, is its lightness and ease of handling. The tent, which can be carried by two men and easily hoisted by hand, can be lowered into a carrying bag and stowed wet, if necessary. The tent is so designed that it is "spread" across the hatch rather than hung, with supporting webbing of high tensile strength angling from offshore bottom corner to inshore end of slot. This permits tent to be positioned without offshore boom being directly over tent headboard. Between the legs of the headboard, which divide the weight of the tent into two parts, and the webbing running diagonally across the wide walls, the tent is very light and flexible. This materially reduces strain and abrasion along the slot. In some instances, particularly when tents are used over a number of different sized hatches, or where winch driver positions are sometimes on deck and sometimes on platforms, an optional feature allows for the tent to be made in two sections, overlapped in back with sufficient material added to allow expansion of approximately twenty to forty feet. This also allows the overlap to be staggered to permit strong winds to pass out of the enclosure of the tent without "lifting".—*Marine News, December 1954; Vol. 41, p. 79.*

Heavy Cargo-handling Gear

Much of the varied cargo carried on the Persian Gulf run is of a heavy nature, and the cargo liners operating on this route are fitted with heavy cargo-handling equipment on a large scale. On the *Bärenfels,* which is one of the three ships of 7,200 tons d.w. ordered by the D.D.G. Hansa from the A.G. Weser, Werft Seebeck at Bremerhaven, the gear has been adapted to these varied needs. In the layout of the heavy lifting tackle the increased weight of the average locomotive has been kept in view; thus the gear has been designed to take a maximum load of 205 tons. Comparison of the derrick capacity of the cargo liner *Bärenfels* with that of the heavy-lift carrier *Belkarin,* built by N.V. Scheepswerf De Hoop, Lobith, for Christen Smith and Co. (Belships Co., Ltd.) shows that the capacity of the heavy derrick on this vessel, which is specially designed for the carriage of heavy lifts with 140-tons capacity, does not come up to the capacity of the gear of the "normal" cargo liner *Bärenfels.* On the *Belkarin,* the heavy-cargo gear, like its counterpart on the *Bärenfels,* is attached to the ship's main mast. The *Belkarin's* total derrick capacity is 203 tons. Another comparison is offered by the two American C-4 ships recently converted by the Bethlehem Steel Company for the U.S. Military Sea Transportation Service. The first vessel of this type to be completed was the U.S.N.S. *Pvt. Leonard C. Brostrom,* which can handle heavy lifts up to 150 tons. This vessel, with two 150-ton, two 30-ton, eight 10-ton, and four 5-ton derricks, has a total derrick capacity of 460 tons. The requirement for a long, straight hold for the carriage of locomotives and other heavy lifts of great length, either on the deck or in the hold, has led the owners of the *Bärenfels* and *Belkarin* to adopt a design with the machinery space arranged aft. The winches serving the heavy derricks on the *Bärenfels* are of the electric type, two heavy winches, each of a lifting capacity of 24 tons, being powered with 110-h.p. motors. On the *Belkarin* there are two 30-ton cargo winches driven by a 70-h.p. motor. The *Pvt. Leonard C. Brostrom* has four 200-h.p. winches of 17·5 tons to serve the two 80-ft. heavy derricks, one on either side of the mast. These derricks are said to be the heaviest ever made; they are stowed one below the other in the ship's second and third decks.—*Holland Shipbuilding, 1954; Vol. 3, No. 7, p. 18. Journal, The British Shipbuilding Research Association, January 1955; Vol. 10, Abstract No. 9,838.*

Swedish Marine Steam Turbines

For the past two years there has been some talk of a steam renaissance in Scandinavia. It is only a question of how long this new steam epoch will last and how large the turbine field will be in the future. The main reason for the steam renaissance in Scandinavia is the development of very large tankers during recent years. Ten years ago the maximum size was more or less stabilized to 16,000-17,500 t.d.w. The fact is, however, that if harbour conditions permit larger ships and if sufficient quantities of cargo are on hand, the costs of freight become smaller the larger the ship. The economical speed increases also approximately with Froude's number, i.e. the power of the engines has increased rapidly. Nowadays 12,500 s.h.p. is most commonly asked for, but tankers with up to 20,000 s.h.p. are built in Scandinavia as well as in the U.S. Only the turbine engineers have been able to offer such high power units. Besides there are some important advantages of steam turbines on tankers. It is interesting to note that more turbine ships than Diesel ships have been built during the last year—despite the fact that the production in the U.S. has been rather modest—while about two or three years ago Diesel ships formed the largest share of new constructions. Three Swedish built and three Italian built tankers with De Laval turbines have been in service for some time, and not less than a dozen Scandinavian installations and some Italian

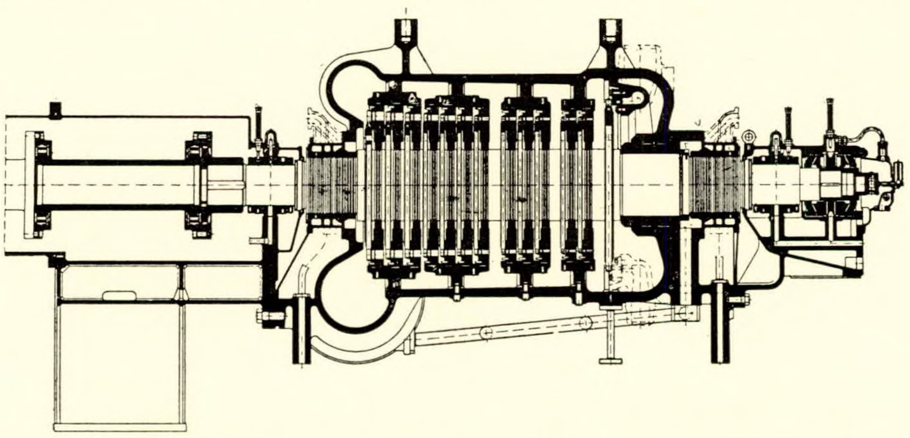


FIG. 3—*Section through a 10,000 s.h.p. solid h.p. rotor. Steam conditions 43 atmospheres (600lb. per sq. in. gauge), 450 deg. C. (850 deg. F.), speed 4,070 r.p.m.*

are to follow. Thev are to be completed within the next two years. The high pressure turbine is usually designed for a speed of 4,000-6,000 r.p.m., making it small and efficient. Nowadays it is commonly designed with a solid rotor, Fig. 3. The groups of nozzles are welded in position on the De Laval turbines. Such an arrangement excludes any possibility of direct contact between the high pressure steam and the casing. The latter can, therefore, be fabricated in a material of lower quality than that chosen for the nozzles. A.B. De Lavals Angturbin makes the low pressure rotors with the wheels shrunk on but in the astern wheels; where high temperatures occur, the design includes special radial pins as locking devices. The shrink fit is not reliable at temperatures of 150 deg. C. (300 deg. F.) and above; sudden astern operation in combination with the heat is apt to loosen the wheels from the shaft. The radial pins centre the wheels even if the shrink fit should loosen because of the temperature shock. The low pressure turbine casing is fabricated in welded cast steel and plating and is directly connected to and resting on the condenser which is supported on springs.—*I. Jung, European Shipbuilding, 1954; Vol. 3, No. 6, pp. 150-154.*

Torque of Semi-balanced Rudders on Multiple-screw Ships

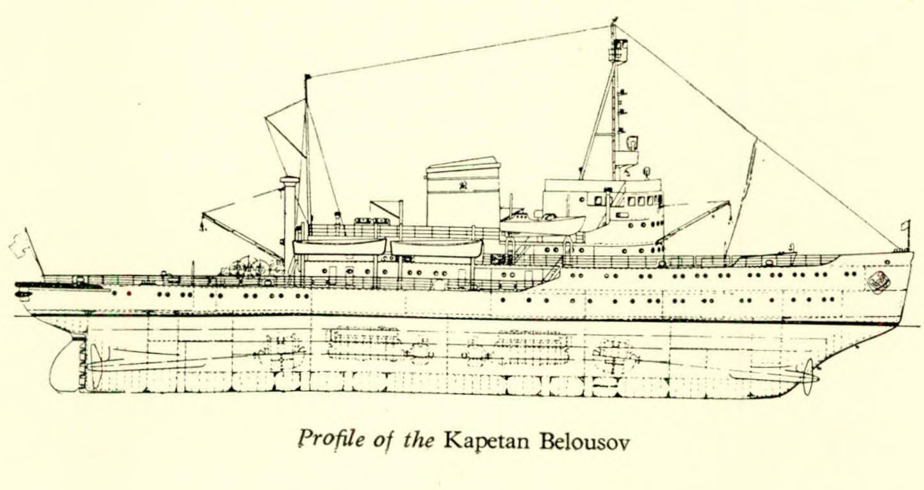
An empirical method is presented for predicting the torque required for semi-balanced rudders not directly in the propeller slipstream. An example of the method is given and theory, full-scale trials, and model test results are compared. The method discussed in this report deals with the rudder torque to be overcome in initiating a free turn from a straight course. Although larger torques are sometimes found in zigzag manœuvres, in restrained model tests, and in manœuvres going astern, the values of initial torque are the prime requisites in rudder design calculations. The basis of the method is the effect of rudder shape on the location of the centre of pressure and on the normal force coefficient. It was found in evaluating considerable model and full-scale rudder torque data that the most significant parameter was the height to chord ratio of the lower portion of the rudder. Curves for normal force coefficients and centre of pressure for the whole rudder plotted against h_1/c_1 (height to chord ratio of lower portion of rudder) were derived from tests of a systematic series of rudders. The torque calculated from these curves was in reasonable agreement with measured values, indicating that other parameters such as section, aspect ratio of the upper portion, thickness, and percentage balance or hull effects such as wake, drift angle, and reduction in speed were either not critical or were implicitly taken into account in the analysis. It was believed advisable, however, for estimating the effect of rudder modifications to take into consideration the shape of the upper portion of the rudder also. Therefore, it was assumed that the rudder may be considered as two relatively independent rudders consisting of the upper and lower portions whose principal parameters are height to chord ratios. The torque of each portion may be calculated from the coefficients for normal force and centre of pressure given by empirical curves. The curves were adjusted from those previously mentioned so that they could be applied to the individual portions of the rudder. The proposed method of predicting torques for semi-balanced centreline rudders on multiple-screw ships appears to give fairly reliable design data. Further tests are needed on single-rudder destroyers to determine what correction factors are necessary to make this method more accurate for this type of vessel. The rudder torque values predicted by this method agree much closer with actual measured torques than do values obtained with Joessel's formula.—*S. C. Gover and C. R. Olson, David W. Taylor Model Basin, Report No. 195, 1954.*

Transverse Strength of Tankers

The question of the transverse of tankers covers a large number of strength problems. To arrive at an estimate of many of the questions it is necessary to know the bending moment distribution in the transverse framework formed by the transverse members. Usually the bending moment distribution in the transverse frameworks is calculated by means of the "Hardy Cross" method. In applications of this method the transverse members have been analysed without considering the longitudinal continuity of the ship. The longitudinal continuity means that the behaviour of the transverse members is affected by the interaction of longitudinals and transverses. Longitudinals and transverses form grids, also called girder networks. Vedeler has given details of methods for grid calculations. It is thus possible to perform accurate analyses of the transverses, provided the end fixities are known and the transverses have constant moment of inertia. The Hardy Cross method as well as Efsen's primary moment method, applied to frameworks having movable joints, mostly will be very intricate unless some approximations are introduced in the calculations. A framework may have joint displacements at the ends of the struts in the wing tanks. When considering these joint displacements the bending rigidity of the struts is mostly neglected. The error, hereby introduced, is defined by the bending rigidity of the struts compared with the bending rigidity of adjacent transverses. However, great bending moments arise at the ends of the struts. It is known that in some tankers the struts have worked loose and fallen to the bottom of the tanks.—*E. Steneroth, European Shipbuilding, 1954; Vol. 3, No. 6, pp. 136-143.*

Finnish-built Icebreakers for Russia

The icebreaker *Kapetan Belousov,* which was completed towards the end of last year by the Wartsila Koncernen, Helsingfors, for the Soviet, is the first of three of the same

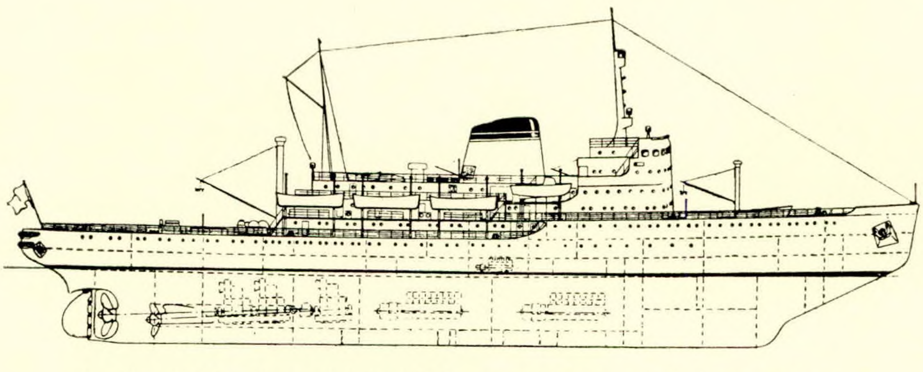


Profile of the Kapetan Belousov

size and is stated to be the largest operating in the Baltic. The main particulars are:—

Length overall	83·16 m. (273ft.)
Length on the waterline	80·76 m. (265ft.)
Maximum beam	19·4 m. (63ft. 9in.)
Beam on the waterline ...	19·21 m. (63ft.)
Depth to upper deck ...	9·5 m. (31ft. 1in.)
Draught	7·0 m. (23ft.)
Displacement	5,360 tons
Machinery	10,500 h.p.
Speed	16½ knots

There is a propeller at the forward, as well as the after end and each is driven by an electric motor supplied from six Polar Diesel-engined generators arranged in two machinery compartments. The engines have eight cylinders 340 mm. in diameter with a stroke of 570 mm. The speed is 400 r.p.m.



Profile of icebreakers with 22,000-h.p. machinery to be built in Finland for the Soviet Government

The second similar ship, the *Kapetan Voronin,* has been launched and the keel of the third vessel laid. In addition, the Soviet Government has now ordered from the same yard two icebreakers which, when completed, will be the largest in the world, each having a displacement of 12,840 tons; the output of the machinery is 22,000 h.p. The other main particulars are:—

Length overall	118·9 m. (390ft.)
Length on the waterline	110·8 m. (363ft. 5in.)
Maximum beam	24·5 m. (80ft. 4in.)
Draught	9·5 m. (31ft.)

Diesel-electric propulsion is to be employed in these ships, as in the previous vessels. There are to be eight generating units running at 330 r.p.m. supplying current to three aft propellers. There is no forward propeller. The generating plant is installed in two engine rooms, four units being fitted in each. The centre propeller is driven by an electric motor of 11,000 h.p. and each of the side propellers by motors of 5,500 h.p. The first ship is due for delivery in 1958 and the second in 1960. Both will have an operating radius of 20,000 miles.—*The Motor Ship, February 1955; Vol. 35, p. 503.*

Shipbottom Paint

Future research and development in organic protective coatings for ships' bottom and other underwater structure will take more cognizance of cathodic protection as a valuable method of preventing corrosion. It now seems obvious that a good underwater paint system in conjunction with a suitable cathodic protection system (either with sacrificial anodes or impressed current) will be a great step forward in the solution of the problem of preventing corrosion and extending the service life of metal in sea water. Paints will be required to withstand the effects of electric currents flowing from the anodes to the ship's hull cathode. The protective paint film will therefore require electrical resistance, water impermeability, adhesion and alkali resistance to a high degree. Considerable research efforts of paint technologists are being applied towards the improvements of shipbottom paints to decrease their susceptibility to film failure due to cracking, peeling, embrittlement and loss of adhesion during the drydocking of a ship. Such coatings should be capable of repeated drydockings and the consequential exposure to the atmosphere without requiring renewal or extensive touching-up of the coating. Relatively hard, tough, smooth chemical and water resistant films of the vinyl copolymer type should offer excellent possibilities for the development of coatings with the desired aforementioned features. The boot-topping belt or waterline area is the most abused outside surface of a ship. Alternate exposure to salt air and immersion in sea water, together with water erosion and the abrasion of fenders and debris, destroys any but the toughest and most waterproof paint films. A boot-topping paint should also have good antifouling properties to prevent fouling at the waterline by marine growth, especially by algæ (seaweeds). A better black antifouling boot-topping paint has been developed and put into use, but further development is under way to produce a superior coating based on tougher synthetic resins of the vinyl type.—*W. J. Francis, Journal of the American Society of Naval Engineers, 1954; Vol. 66, No. 4, pp. 857-866.*

New Canadian Survey Ship

A notable hydrographic survey vessel for the Canadian Department of Mines and Technical Surveys, Montreal, the first to be built in forty years, is shortly to be laid down by Canadian Vickers, Ltd., Montreal. She is for service on the St. Lawrence River, in the Gulf of St. Lawrence and adjacent Atlantic waters, also in the Arctic Ocean, being designed for operation in heavy ice. She will have a round nose, raked stem, ice-breaking bow contour and cruiser stern, with continuous upper and main decks. A hangar is provided for two Bell helicopters and there will be a flight-deck aft. The main particulars are:—

Length overall	285ft.
Length b.p.	262ft.
Breadth moulded	49ft. 6in.
Depth moulded to upper deck	29ft. 6in.
Displacement	3,700 tons
Maximum speed	15½ knots
Cruising speed...	12 knots
Range	14,000 nautical miles
Machinery	7,060 b.h.p.

The propelling machinery comprises four Canadian-built Fairbanks, Morse opposed-piston engines running at 750 r.p.m., each pair driving a propeller at 200 r.p.m. The drive is taken through British-built Modern Wheel Drive oil-engined reverse reduction gears.—*The Motor Ship, March 1955; Vol. 35, p. 531.*

Marine Auto-Alarm Equipment

The auto-alarm, switched on when the radio officer ends his watch and goes off duty, maintains an unceasing reception watch on the 500 kc/s distress frequency until he returns to the radio room for his next spell. If, in his absence, another ship within range transmits the auto-alarm signal, the equipment responds just before the end of the third four-second dash in the series of twelve, by actuating an alarm bell circuit. Bells ring on the bridge, in the radio officer's own cabin, and in the wireless room itself, calling attention to the fact that a ship in the vicinity is in distress. The radio officer, alerted by the bells, then hurries to his post—the remaining nine dashes giving him time to do this before the distress message itself begins—and is then able to listen to details of the distress signal on headphones plugged in to the receiver unit of the auto-alarm. Auto-alarms marketed by British manufacturers are necessarily similar in basic principles and performance, being designed in accordance with the specification laid down. There are, of course, differences in circuits and in the methods used to obtain the necessary acceptance and rejection characteristics of the selector. Generally speaking, the auto-alarm consists of two main units—the receiver and selector—with the necessary power arrangements and the bell circuit, which is powered by a separate 24 volt battery supply. This is in accordance with regulations so that, should the set be rendered inoperative by failure or disconnexion of the ship's mains, the alarm can be given and the matter put right. The receiver unit is of an optimum sensitivity carefully calculated to ensure reception of alarm signals from ships within reasonable assistance range. Transmission from vessels that are too far away to make steaming to their positions impracticable are also rejected—it being assumed, of course, that such distant transmissions will be picked up by ships much better placed, geographically, to render assistance. The selector, for its part, must be capable of the most rigid "censoring" of all incoming combinations of dots, dashes and static passed to it from the receiver. Moreover, it must be capable of excluding any which may closely resemble the auto-alarm signal without being so narrow in its acceptance stringency that there is any risk that a call for help might be rejected. The characteristics of the auto-alarm receiver must be such that it will readily receive weak signals slightly off tune without taking in unnecessary signals or static which might conceivably overwhelm a genuine auto-alarm signal. It is accordingly pre-tuned on a frequency band of eight or ten kc/s either side of 500 kc/s. This is to ensure adequate coverage on the actual distress frequency while still allowing for the reception of alarm signals from a transmitter which may be slightly off tune caused by emergency operation due to the use of a "jury" aerial or other causes.—*S. T. Andrew, Communications and Electronics, February 1955; Vol. 2, pp. 62-65.*

Pulsating Flow Measurement

The problem of measuring the mean rate of mass flow of a fluctuating steam is one which has attracted the notice of many investigators but, although a limited degree of success has been achieved by more than one method, it may be said that a simple, inexpensive, and compact device which is capable of dealing with flows subject to severe pulsations without loss of accuracy and without ancillary apparatus has yet to be described. Meters of the orifice type, used in conjunction with large smoothing capacities, have been widely employed, in particular for the measurement of the air consumption of reciprocating internal combustion engines, but the smoothing capacity often requires to be inconveniently great and too small a volume will lead to surprisingly large errors, whilst the range of flows covered by a single orifice is rather restricted since the rate of flow is proportional to the square root of the head. Electrical methods, depending on the transfer of heat from an electrically heated wire to the steam, have been tried, but are also not free from errors when velocity fluctuations are present. Other methods adapted to particular circumstances have had some success but do not represent a satisfactory general solution of the problem. So-called "viscous meters" in which the pulsating steam is directed through a number of narrow passages across which the pressure difference is measured and in which the Reynolds number is kept so low that the flow is laminar may be shown theoretically to have certain advantages, notably that of a straight-line law between head and mass flow even when the instantaneous velocity is subject to fluctuations. Within limits these advantages have been confirmed in practice but under conditions of severe pulsation such meters have shown large and puzzling errors. The investigation described was undertaken with the primary aims of studying the performance of viscous meters under the most severe conditions of pulsating flow, with special reference to the measurement of the air consumption of reciprocating internal combustion engines, and of developing an accurate meter suitable for use under such conditions. It is believed that these aims have been at least partly achieved and, whilst many of the experiments relate to engine air-flow measurements, it is considered that the results are capable of wider application and that they will be of value in a variety of problems concerned with fluctuating flows.—*Paper by L. J. Kastner and T. J. Williams, read at a meeting of the Institution of Mechanical Engineers on 18th February 1955.*

Corrosion Due to Vanadium Pentoxide in Fuel Oil Ash

Air-corrosion of heat resistant alloys at high temperatures is known to be extremely rapid under conditions where surface contamination by certain low melting constitutents occurs. The corrosion rate actually accelerates with time and reaches catastrophic proportions; corrosion rates actually surpassing a hundred inches per year (approximately three yards per year) are encountered! This poses a critical problem in gas turbines, boiler tubes and other applications where normally heat resistant alloys are exposed to hot gases resulting from the combustion of certain grades of fuel oil which contain trace amounts of vanadium. Although the vanadium content of the fuel oil may be less than a hundred parts per million, it can nevertheless become concentrated to more than 50 per cent in the resulting fuel oil ash. These ash paticles generally are carried in the gas stream and are deposited on impinged metal surfaces. Instances of oxide dripping from ship boilers and their supports have been observed. Attempts to find a heat resistant alloy capable of resisting catastrophic corrosion have not met with success. However, nickel-rich alloys appear to be more resistant than nickel-free alloys, but their resistance to attack is still far below par. Some success in counteracting the effects of vanadium pentoxide with the use of additives such as CaO, MaO or carbon has developed, but here, too, success has been only partially achieved. The most promising approach to the problem appears to be that of developing an additive to the oil or a porcelain-type coating to be applied periodically to exposed metal surfaces.—*Corrosion, January 1955, Vol. 11, pp. 17-18.*

Symposium on Turbine Operating Experience

This symposium is in five parts and represents a wide variety of operating and maintenance experiences by some of the principal turbine builders of America. The papers presented were:—

Part 1. Turbine Operating Experience, by A. Dudley Haff.

The author discusses the operating experience obtained from service and inspection of main propulsion turbines over a period of the last fifteen years, during which time the author's company have built upwards of two hundred units. Blading failures are described, which include fatigue failure from vibration, damage caused by foreign material, damage caused by improper steam conditions, and lack of clearance. Experience with gland and dummy labyrinth packings and rotor vibration are included. The usual maintenance problems encountered with bearings, oil baffles, and valves are described.

Part 2. Some Operating Experience with Westinghouse Marine

Turbines, by Harold Anderson.

This is a case history of unsatisfactory operating experiences encountered with the new designs of marine turbines of the author's company, either on trial trips or during early service. In the past four years, several vessels have entered service with propulsion turbines designed to operate with a steam temperature of 1,000 deg. F. at the turbine inlet. This new venture in marine practice has proved entirely successful, and after initial adjustments only minor troubles have occurred in these installations.

Part 3. Corrosion-Erosion Problems in Steam-Turbine Operations, by F. H. Pennell.

Corrosion, erosion, and a combination of these two factors as encountered in steam-turbine operation are discussed by the author, and a considerable part of the paper is devoted to consideration of erosion by impact of water-droplets. This frequency occurs in the later stages of low pressure turbines when the linear speed of the blading exceeds 900ft. per second, and the steam contains 10 per cent or more moisture. Methods of counteracting this destructive condition are by design and metallurgy. Illustrations show the older types of design, and the newest one for extraction of moisture. A number of methods are discussed for hardening the area eroded by water-droplet impact, together with some mechanical tests. The author concludes that at the present time the separately cast or forged strip, silver brazed to the blade, still seems to be the most favoured method despite its deficiencies.

Part 4. Turbine-Operating Experience, by J. R. Kane.

The author discusses the family of merchant-type turbines now being designed and built by the Newport News shipyard, which were first developed and put into operation in 1949. No serious incidents or difficulties have arisen with these machines in service, and maintenance has been moderate.

Part 5. Turbine-Operating Experience, by A. D. Somes.

This discussion of marine steam turbines is confined to those designed and built since 1946. Some of the difficulties encountered with the new design and the measures taken to overcome them are described. The design of buckets is treated in detail, and reference is made to their failure due to torsional vibration. Other operational difficulties with thrust bearings, rough operation of high pressure and low pressure turbines, and oil leakage from bearings are discussed.—*S.N.A.M.E. papers read at Annual Meeting, 10th-13th November 1954. Journal, The British Shipbuilding Research Association, January 1955; Vol. 10, Abstract No. 9,821.*

Device for Testing Deck Sockets

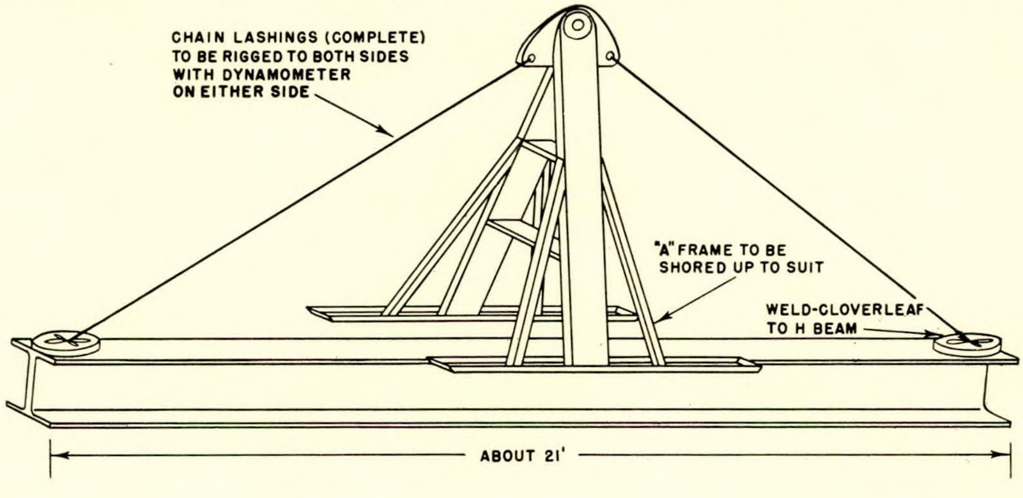
The use of an "A" frame for testing vehicle-lashing deck sockets and side sockets has resulted in considerable savings. Eliminating most of the need for a shipyard crane, the portable "A" frame permits testing of all lashings at any convenient place prior to ship installation. When testing deck pads and bulkhead fittings, any two previously tested lashings can be used to test all fittings. Use of the "A" frame eliminates need for welding brackets on the hull and chipping them off upon satisfactory completion of the test. Its use in testing deck sockets and side sockets permits the testing of two fittings simultaneously at the proper angle.—*Bureau of Ships Journal, February 1955; Vol. 3, p. 48.*

Wall Corrections for Flow About Two- and Three-Dimensional Symmetrical Bodies

When a body is placed along the centre line of a rectangular channel which is infinitely long, an infinite two-dimensional array of images is formed by the multiple reflections of the body in the four walls of the channel. The flow in a channel is then equivalent to the flow past such an array of bodies. Velocity and pressure measurements made at any point in the channel are affected not only by the original body but by the whole field of images. If the channel is finite in length, the flow is further restricted by the end walls which, by the multiple reflections of the two-dimensional array of images, form an infinite three-dimensional array. The present study was undertaken to determine the magnitude of the wall interference for models of different sizes and shapes in a rectangular electrolytic tank which is equivalent to a rectangular channel of finite length. Reflections in the tank bottom and free water surface are identical with mirror reflections in solid walls. Sources and doublets are reflected with a change of sign in the end walls. An optimum model size for an electrolytic tank or wind tunnel would be one for which the wall correction falls within the experimental error of the measurement. As it often happens that this optimum size is too small to produce the required experimental accuracy in the measurements, methods are presented for finding the wall correction at any point in the channel for both two- and three-dimensional symmetric bodies. Many papers have been written on wall interference, particularly for wind tunnels. Lock has calculated the wind tunnel interference at the position of the body for both two- and three-dimensional symmetrical bodies which are small compared with the width of the tunnel. His results will be extended to include larger bodies and to find the pressure at any point in a channel of infinite or finite length. By the present method, expressions may be found for the wall interference of bodies which may be represented by a source-sink or doublet distribution along the axis of the symmetry.—*A. Borden, Navy Department (U.S.A.), The David W. Taylor Model Basin; Report No. 864, December 1954.*

Flow of Air Through Radial Labyrinth Glands

Radial labyrinth glands are employed in radial flow steam turbines, in certain steam and gas turbines, and in certain turbo-compressors. The subject of leakage through radial



glands has been neglected in technical literature. A theory for the flow of air through a radial gland has been worked out for both outward and inward flow. Expressions are derived for the pressure distribution in each kind of flow, and it is shown that the critical pressure ratio can be reached only in the final constriction. Experiments were made on a gland having a single ring and also on a 20-ring gland of the staggered type. The discharge coefficients in the latter gland were lower than those in the single-ring gland, possibly due to the different approach conditions. The observed pressure distribution in the multi-ring gland agreed well with the theoretical value. Finally, some experiments were made with unstaggered radial glands under various conditions. Discharge coefficients greater than unity were measured.—*Paper by W. J. Kearton, submitted to the Institution of Mechanical Engineers for written discussion, 1955.*

American Controllable-pitch Propeller

A new line of controllable-pitch propellers for installation on tugs, ferries, ore carriers, and similar vessels has been announced by Allis-Chalmers Manufacturing Company. The propeller eliminates the need for reversing gears and provides maximum engine operating efficiency at all conditions of loading. Work on the new propeller began in 1945. In 1952, the first commercial propeller was tested in a Diesel-electric tug in New York harbour. Tests covered every phase of tug operation over a period of several months. Final designs, based on results of these tests, now cover propeller ratings from below 300 h.p. to 10,000 h.p. and above. The unit consists of a hydraulic servo system inserted at a convenient place in the drive shaft. This operates a link and lever in the propeller hub to adjust the blade pitch. Thus, only a single lever is needed to control the forward and reverse motion of the ship. With completely automatic control, the operator quickly can throw the ship from full speed forward to full speed reverse. The control system follows through this order, utilizing the full output of the engine at every point during the change but without overloading. The same control can be set to hold motor horsepower constant, regardless of changing load conditions. The Allis-Chalmers propeller mechanism is under a slight oil pressure at all times, eliminating any possible inleakage. The propeller is compact, can be coupled to almost any size shaft, and is installed easily. It can be adapted to Diesel engines or to gas or steam turbines maintaining high efficiency at all loads.—*Marine Engineering, February 1955; Vol. 60, p. 68.*

Propulsion of Ships

The huge peak outputs, from one hundred to two hundred thousand h.p. required by large naval craft to attain the highest possible speed in an emergency (and for a short period only), have long ago introduced the problem of running the main engines at much lower powers than the trial overload and at the prolonged tactical speed, with a good efficiency. The planned ship was designed with a main engine consisting of two sets of turbines with reduction gears of 48,400 s.h.p. each (fitted with Troost-type high efficiency propellers) for full power and strategical cruising speeds, and an auxiliary propelling engine consisting of fast running Diesel motors with a.c. electric transmission of the maximum output of 5,000 s.h.p. on each shaftline for low cruising speed (20 knots) and manœuvring. The arrangement of "Vulcan"-type hydraulic couplings between each prime mover and its adjoining pinion (namely, four to each set; see Figs. 1 and 2) permits alternating

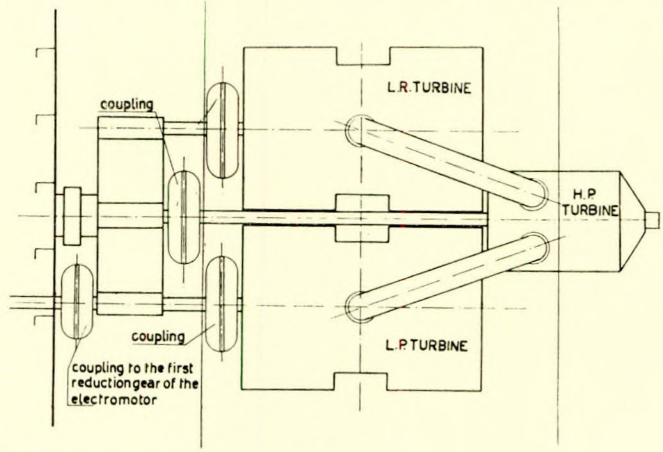


FIG. 1

For each group: h.p. turbine, r.p.m. 3,500; l.p. turbines, r.p.m. 3,000; screw, r.p.m. 320; maximum power: steam, h.p. 48,400, each electric motor, h.p. 5,000

the couplings in the best possible manner, according to the speeds. For instance, at *maximum power:* the h.p. turbine exhausts into the two l.p. turbines; electric motor uncoupled. *Strategical cruising:* the h.p. turbine, sectioned off, exhausts only into the l.p. turbine containing the backing turbine; the other l.p. turbine and electric motor being uncoupled. *Low cruising and manœuvring:* electric motor connected; all turbines uncoupled. A serious inconvenience is the power loss in the coupling emptied of oil but not of air. This power loss can be eliminated by means of a small air pump which maintains a vacuum in the airtight casing of each coupling and in the upper and lower vessels connected to them. This arrangement of main engines would remain substantially unchanged if, at *low cruising speeds,* one preferred adopting *steam turbines with thermo-nuclear boilers* instead of the Diesel-electric installation, apart from the normal boilers employed for high speeds.—*G. Rabbeno, European Shipbuilding, 1955; Vol. 4, No. 1, pp. 2-4.*

Stress Distribution in Corrugated Bulkheads

This report deals with the stress distribution in a transverse corrugated bulkhead with vertical troughs. Its aim is to arrive at a method of calculation suited for use in practice.

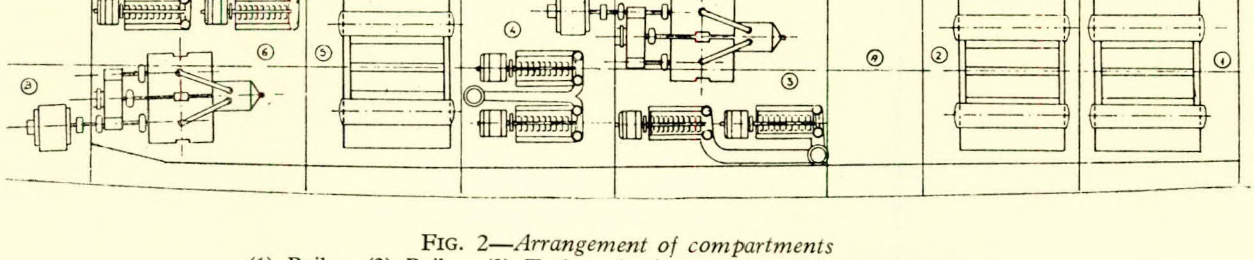


FIG. 2—*Arrangement of compartments*

(1) Boiler; (2) Boiler; (3) Turbo-reduction gear and Diesel alternator set; (4) Electric motor and Diesel alternators; (5) Boiler; (6) Turbo-reduction gear and Diesel alternator set; (A) Diesel-electric auxiliaries; (B) Electric motor

A method, based on elementary principles, and the necessary assumptions are discussed and an example is given. The problem of local instability is discussed and the stresses in the welds of the plating are considered. The effect of the adoption of corrugated bulkheads on weight is discussed briefly. In Chapter II some tests on a corrugated bulkhead are described and a comparison is made between the calculated and measured stresses. It is shown that there is a reasonable agreement which, in the opinion of the authors, justifies the method of calculation discussed in Chapter I. The desirability of more extensive stress measurements on a bulkhead of this type is mentioned.—*H. E. Jaeger, B. Burghgraef, and I. Van Der Ham; International Shipbuilding Progress, 1955; Vol. 2, No. 5, pp. 3-29.*

Controlling Smoke by Sludge Removal

Among the causes of smoking power plant and furnace installations, the factor of sludge deposits in fuel oil seems most frequently neglected. Yet with today's heavy oils and the ability of modern burners to handle oils which would not function in the burners of yesterday, this factor of sludge is of prime importance. Storage tanks tend to accumulate sludge contents which seriously interfere with efficient functioning of the burners, with the result that smoking, often blamed on other causes, is the rule rather than the exception.—*Marine News, December 1954; Vol. 41, p. 79.*

Hydrodynamic Forces Acting on a Slender Body of Revolution

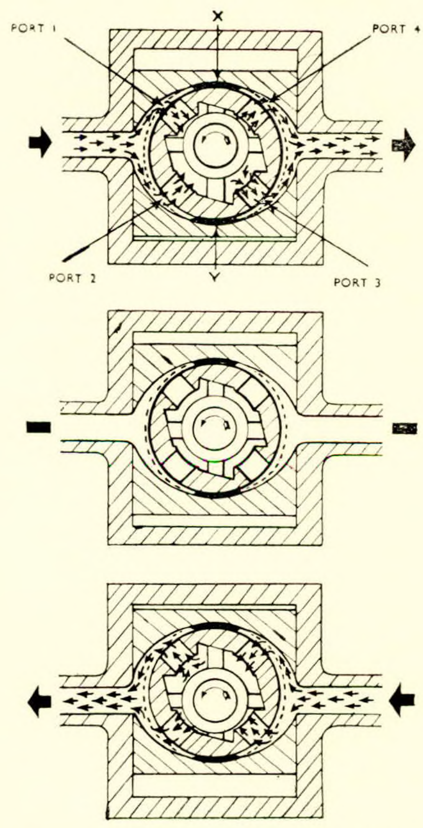
The theoretical determination of the forces and moments on a body due to waves on a free surface is one of the outstanding problems of hydromechanics and is of considerable practical as well as theoretical interest. The case in which the waves are generated by the body has been extensively studied. The case in which the surface is independently disturbed by a train of waves has received much less attention, but it is probably of greater importance in naval architectural applications since this case is closely tied up with seaworthiness. The usual method for estimating the effects due to a wave train is based on the Froude-Krylov assumption, which considers the pressure variation due to the structure of the wave but neglects the dynamics of the flow around the body. For submerged bodies, this assumption is known to be quite inaccurate, and a solution which takes account of the boundary condition at the surface of the body is highly desirable. In the present report, such an analysis is carried out for a slender body of revolution. The body is represented approximately by a system of singularities distributed along its axis, and the forces and moments are determined by means of an extension of Lagally's theorem developed by the author in a previous report. A detailed computational procedure is outlined for computing the forces and moments in terms of the sectional area curve of the body. Although the derivation is for a body of revolution, the method may be used for other slender bodies in which the cross-sections depart somewhat from circular, with the expectation that the answer will at least be correct in order of magnitude. The principal limitations on the analysis are that the fineness ratio of the body be large and that the ratio of the wavelength to body diameter also be large. This latter restriction is not serious, since the forces and moments are usually negligible for wavelength less than half the body length. The limitation on fineness ratio will be relaxed somewhat in a report to appear shortly, in which a more precise analysis is carried out for the particular case of the spheroid.—*W. E. Cummins, Navy Department (U.S.A.), The David W. Taylor Model Basin, Report 910, December 1954.*

Italian Built Diesel Engine of Non-magnetic Materials

A twelve-cylinder Mercedes Benz Diesel engine, type MB 820 Eb, built entirely of non-magnetic materials, has passed a 200-hour acceptance test supervised by the Italian Ministry of Marine. Mercedes Benz engines are built by the Fiat works at Turin under licence from the German firm.—*Schiff und Hafen, December 1954; Vol. 6, p. 827.*

Large Bunkering Pump

Plenty and Son, Ltd., have completed, at their Newbury works, one of the largest rotary positive-displacement pumps yet built in this country. The particular unit is of the maker's own design, first used in 1935, and has an output of 350 tons per hour. It has now been despatched to Aden where it will be installed at the Anglo-Iranian Oil Company's new refinery for bunkering visiting vessels. A particular feature of the design is that although it is uni-rotational and runs at constant speed, the output can be varied from full flow through zero to full flow in the reverse direction. This feature is of special value for the oil industry as it enables pipe lines to be emptied in readiness to receive the next consignment of oil, which may be of a different grade, simply by altering the control mechanism. Of modest dimensions, the pump consists of a main body having suction and delivery branches on the horizontal plane. Carried within the main body is an inner sliding block which can be raised or lowered relatively to the shaft centre line by operating a handwheel on the top casing. A cylindrical rotor, carried within this block, revolves in a specially-ported liner, having four rectangular ports spaced at 90 degrees. This



Diagrams showing the action of the pump. With the slide block in the central position no flow occurs

rotor acts solely as a distributor and is caused to revolve by the blades which, in turn, are driven by the shaft. The driving shaft is a solid member supported by ball and roller bearings concentrically with the main body. It has two slots cut through it at right angles into which the blades fit. As the shaft rotates the blades slide radially through the slots and traverse the angled faces in the rotor bore, so that they form four sealed chambers, as can be seen from the accompanying sketches. If the slide block (Y) is held so that the liner (X) is concentric with the shaft, there will be no displacement or change of volume in the four chambers as the rotor revolves. As soon as the slide block is raised above or lowered from the central position, displacement does occur in the chambers as the rotor revolves and the liquid flows in one direction or the other.—*The Marine Engineer and Naval Architect, February 1955; Vol. 78, pp. 49-50.*

Noise in Reduction Gears

Future trends in power plant design point towards the increased use of reduction gears. High rotational speed-type machines for prime movers such as high-speed steam turbines and gas turbines must have a suitable speed reducer interposed between the output shafts and the propeller shafts. With the advent of nuclear power, the steam turbine will be given new emphasis. The marine propeller is still considered the best and simplest underwater power transmission mechanism. Effective noise reduction of all machinery components in naval vessels is a vital necessity in the ability to detect the enemy and escape detection by him. Gears are inherent noise producers by virtue of their construction and function. Heavily stressed rotating structural elements, transmitting large loads at high speeds, must produce vibration and its accompanying noise. It has been repeatedly shown that tooth spacing errors and hobbing machine inaccuracies are certain causes of noise generation. Modifications of design and geometry and the use of new materials may contribute to noise reduction; but it is not possible to ascertain their contribution without extensive comparative tests. The gear manufacturer must make use of the most modern engineering methods to reduce errors and thus control the generation of noise at the source. The Bureau of Ships gear noise reduction programme has been set up with the ultimate goal of presenting the gear manufacturer with a qualitative specification which will guarantee the production of gears with a predetermined noise output. These specifications will definitely set down criteria for minimum machining tolerances, tooth characteristics, special design features, balance, materials, proper lubrication, casing construction, and isolation of components and the completed unit. An analytical review of this programme has been made in light of the numerous suggestions which have been published regarding the causes and correction of gear noise. This review indicates that, if the ultimate goal can be reached, the investigations which have been initiated will eventually lead to a solution. The programme is well balanced between basic research, empirical investigations, and practical applications.—*R. Taggart, Journal of the American Society of Naval Engineers, 1955; Vol. 66, No. 4, pp. 829-847.*

Experiences with Mariner-type Ship

The Mariner is a high-speed freighter of the C4 type, designed to carry somewhat greater cargo than the C3 class ship, and move it faster and more efficiently. In the design it was attempted to meet both the requirements of an all-purpose commercial carrier and a naval auxiliary without destroying the usefulness of either. In this regard particular attention was paid to ensure the handling of military-type material. The vessel was designed to maintain 20 knots sustained sea speed. With the exception of one item involving the main turbine, there have been few failures of any major importance on the Mariners to date. With regard to the main propulsion turbines, though most of these units have given excellent service, several failures of one make have necessitated design changes. In each case the trouble first was indicated by a noticeable vibration of the turbine. The first breakdown occurred in the high pressure unit, and came soon after the ships went into operation. Tenons in the first stage wheel were found to be cracked as a result of resonant vibration. Calculations indicated that the natural frequency of the blade-shroud combination closely coincided with the frequency of the first stage nozzle impulses at maximum speed, and the high stresses thereby set up on these tenons caused failure. The manufacturer corrected the situation by separating these frequencies and reducing the load on the tenons. Impulse frequencies were increased by redesigning the nozzle block, still retaining the same nozzle area. The system frequency was decreased by machining the shroud band. The net result was removal of the blade frequency from the critical range and a stress reduction in the tenons. The second turbine trouble occurred in the low-pressure unit, but did not show up until after more than 100,000 miles of service. A number of buckets at the fourth stage failed from fatigue, breaking off at the bottom of the blade just above the top of the skirt. Some indications of high stresses were noted in third stage buckets. This trouble appeared to be another case of resonant vibration, for which the correction was felt to be a redesign of the blades. Though the calculated stresses due to variable forces are very low compared with the endurance limit of the material in wet steam, the condition resonance amplified these stresses until the limit of the material was reached after about 92 billion cycles. In the redesign, blade frequency was removed from the operating range and the stresses reduced to from one-sixth to one-quarter of their original values by modifications to the bucket design and stage arrangement.—*W. G. Allen and E. K. Sullivan, Marine Engineering, January 1955; Vol. 60, pp. 44-50.*

Choice Between Single- and Multi-combustion Systems for Gas Turbines

The diversity of applications for which gas turbines are being considered is matched by the wide variety of external configurations of these prime movers, even when limited to the combustion system selected on the overall design and layout of the engine. The paper also considers the influence of the combustion system selected on the design of the individual combustion chambers making up the combustion system. The main conclusion drawn is that in spite of its greater bulk the single chamber combustion system is to be preferred on the grounds of reliability and ease of maintenance.—*Paper by I. G. Bowen and W. Tipler, read at a meeting of the North East Coast Institution of Engineers and Shipbuilders, 14th January 1955.*

American Radar Set for Small Boats

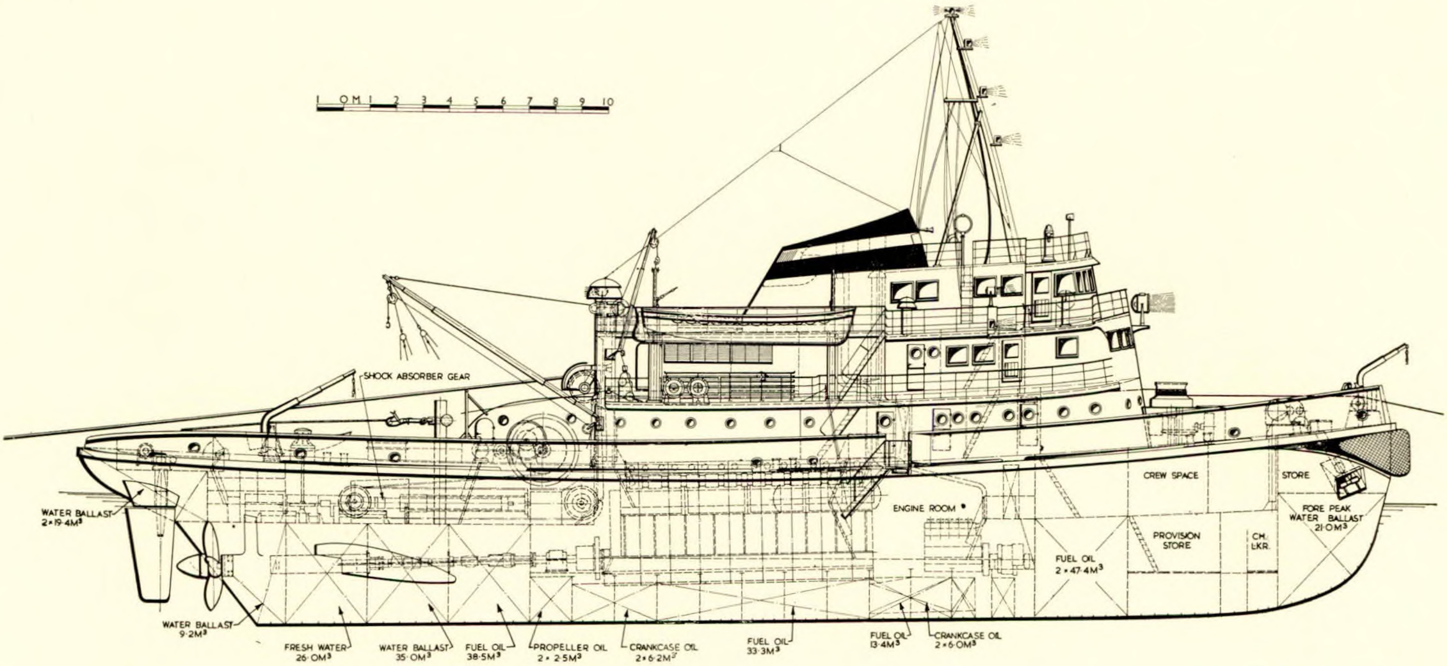
A new small boat radar system, developed and created by Lavoie Laboratories originally for U.S. Army and Navy use, is now ready for universal production. The manufacturer states that "The Bat" is an entirely new departure in electronics. Designed especially for small craft, it is usable also for larger ships as auxiliary radar. It is said to have a mimimum range of 40 yards and a maximum of sixteen miles. Its total weight, including antenna, receiver-transmitter, and indicator, is approximately 130 pounds. It has been tested through 1,400 hours of all weather conditions, including four hurricanes, and withstood winds of over one hundred miles an hour. The manufacturer claims that simplicity of pattern enables the units to be placed anywhere aboard without affecting the balance or the effectiveness of operation of the equipment.—*Marine News, January 1955; Vol. 41, p. 65.*

Powerful Dutch-built Tug

The *Edgar Bonnet*, one of the highest powered tugs yet constructed, was recently completed at L. Smit and Zoon's yard, Kinderdijk, Holland, the tug having been ordered with the I.H.C. Holland last year. She is for the Cie Universelle du Canal Maritime De Suez, Paris. The following are the main characteristics:—

Length overall, metres	50·125
Length b.p., metres	46
Beam, metres	12
Mean draught, metres	4·5
Machinery, b.h.p.	5,500

Specially designed towing equipment has been provided with a hydraulic-pneumatic winch capable of absorbing the strain on the cable when operating in very heavy seas and on a short tow.



Profile of the tug Edgar Bonnet

The equipment includes four electrically driven salvage pumps, two with a capacity of 100 tons hourly and two of 60 tons. There is also a 150-ton fire pump. The two engines are of the Smit-M.A.N.-type, each having eight cylinders, 480 mm. in diameter with a piston stroke of 840 mm. Each drives a Kamewa variable-pitch propeller. The normal output in a tropical climate is 2,250 b.h.p. and on trials in Holland, was 2,750 b.h.p. The speed on trials, running free, was 15 knots. Current is supplied for the various auxiliary machinery and for other purposes throughout the vessel from three 115-kW 220-volt M.A.N.-engined dynamos and a Pelapone-engined harbour generator. The steering gear is of the electro-hydraulic type. The bunker capacity of the tug is 120 tons of Diesel oil, and 50 tons of fresh water are carried. The fuel is sufficient for five days of twenty-four hours when the tug is proceeding at full speed.—*The Motor Ship, April 1955; Vol. 36, p. 23.*

Rubber Bearings

Rubber as a bearing material is adaptable to any underwater location or where water can be piped to the bearing for lubrication purposes. The hardness of the rubber used is comparable to the hardness of an automobile tyre. With a well-lubricated rubber bearing, the coefficient of friction is about the same as for a well-lubricated metal bearing; it is especially low at higher speeds. The most usual applications of rubber as a bearing material include marine propeller installations, rudder pintle heads, dredge cutter heads, deep well pumps, coal washers, underwater conveyors, and hydraulic turbines. Provided the type of rubber in the bearing is suited to the service conditions, it affords a definite advantage as a protective medium for metal shafting by reason of its ability to accommodate itself to abrasive particles. In other words, the softness of the rubber enables any foreign abrasive solid matter, carried in by the water, to be depressed into the bearing surface, rolled into the longitudinal grooves, and flushed out rather than being embedded in the surface of the bearing, as might occur with some metal bearings, to cut or score the shaft. The shaft should be bronze, "monel" metal, or stainless steel because steels of conventional type present a rusting problem especially when the machine is idle. Rubber is possessed of another natural advantage by reason of its flexibility which enables it to conform to some shaft misalignment without undue vibration. While it has been noted that the coefficient of friction of a well-lubricated rubber bearing is low, this can be high unless the bearing is copiously lubricated by water from the very start. Even a few turns of the shaft on a dry bearing surface could cause serious abrasion, tearing, and overheating. It is well, also, to bring the unit up to speed before the load is applied because even though rubber, as a bearing material, can accommodate itself to loads around 50lb. per sq. in., temperature control and complete lubrication are necessary to protect the surface. The normal maximum shaft speed limit for such bearings is around 800 to 1,000ft. per minute. The normal accepted maximum permissible temperature is approximately 150 deg. F. Higher temperatures will cause marked decrease in life. Most complete protection of the bearing by water lubrication is assured when the bearing is designed to ensure complete flooding by the flow of water. Chamfering of bearing edges and angular grooving enable the turning shaft to exert a pumping action, just as it does in a well designed oil-lubricated metal bearing. Flood lubrication also permits the water to function not only as a lubricant but also as a means for conducting heat and foreign materials away from the bearing.—*A. F. Brewer, Lubrication Engineering, 1954; Vol. 10, No. 6, p. 316.*

Forty Years of Change at Portsmouth Dockyard

This paper has been prepared to provide a general appreciation of the principal changes which have effected the production activities at Portsmouth Dockyard during the past forty years. It also includes a brief outline of the general organization of a dockyard, together with comments on some present-day problems, most of which are common to the other home dockyards, and some to commercial shipyards, particularly those engaged on warship work. The arbitrary period chosen embraces both world wars, and coincides with the author's career in the Admiralty Service, which commenced as a shipwright apprentice on the eve of World War I.—*Paper by I. E. King, read at a meeting of the Institution of Naval Architects on 5th April 1955.*

Patent Specifications

Marine Fuel Oil Sprayer

This invention relates to an oil burner for marine purposes of the kind comprising a sprayer body which is fitted at its discharge end with atmomizer plates or an atomizer cap through which the oil passes and is atomized before entering the furnace. Referring to Figs. 1 and 2, the sprayer comprises the usual body (1) which is fitted at its discharge end with an atomizer (2) which is detachable for cleaning purposes. The means provided by this invention for preventing discharge of oil when the atomizer cap or plates are not in position, comprise a spring loaded valve (3) which is fitted within the discharge end of the sprayer body (1) and which when the atomizer (2) or plates (10) are not in position is urged against a seating (4) formed in the inner wall (5) of the body (1) so that no oil can pass. The spring may conveniently be interposed between a shoulder (6) at the discharge end of the spray tube (7) and the rear end of the valve (3) which is thus pressed

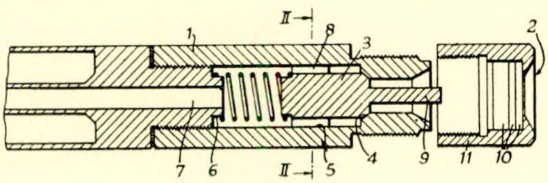


FIG. 1

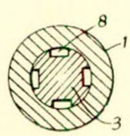


FIG. 2

at its forward end against the seating. The circumferential wall of the valve (3) is formed with grooves (8) through which the oil can pass from the rear end of the valve to the front end. The front end of the valve is formed with a rod or web (9) which, when the atomizer (2) or atomizer plates (10) in the cap (11) are fitted to the discharge end of the body (1), is engaged and moved axially rearwardly off its seating against the influence of its spring.—*British Patent No. 722,951, issued to Associated British Combustion Ltd. Complete specification published 2nd February 1955. (Inventor: R. B. Cooper.)*

Gas Turbine Plant

This invention relates to open-cycle gas turbine plant capable of being run on a variety of fuels having different calorific values, without serious loss in efficiency. In conventional open-cycle gas turbine plant, the mass flow through the turbine is the sum of the mass flow of the air from the compressor and of the fuel admitted to the combustion chamber with corrections for leakages. When running on fuel having relatively low calorific value, the mass flow through the turbine will be higher than when running on fuel having relatively high calorific value because, in the former instance, the low calorific value will necessitate the burning of a greater mass of fuel in order to achieve the nozzle temperature for which the turbine is designed. Thus, if it is desired to run a given plant on any one of a variety of fuels having widely different calorific values, the problem arises of adjusting the turbine "swallowing capacity", i.e. nozzle area, to accept correspondingly different quantities of gas. For, if the design performance of the turbine blading is not sufficiently flexible to accept such variations in gas flow, the efficiency of the plant will suffer in consequence of any deviation from the design basis. It is an object of the invention to provide plant whereby a change over from fuel having high calorific value to fuel having low calorific value may be readily made without the need for adjusting the nozzle

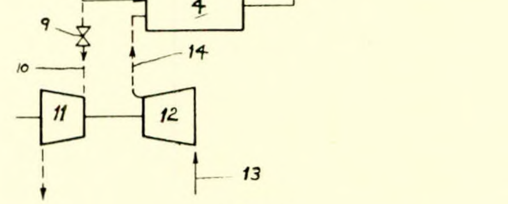


FIG. 4

area and without undue loss of efficiency. In Fig. 4 a turbine (1) designed for fuel of relatively high calorific value is mechanically coupled to a compressor (2). Air from the compressor is conveyed through a duct (3) to a combustion chamber (4), into which liquid fuel is injected by a fuel pump (5) driven by an electric motor (6). The gaseous combustion products are then conducted through a duct (7) to the turbine. The turbine is coupled to an electric generator (8). When it is desired to run the plant on fuel having a lower calorific value than that for which the turbine is designed, for example, blast furnace gas, a greater mass flow of fuel has to be provided in order to attain the desired nozzle temperature in the turbine. In order that the total mass flow through the turbine shall not be greater than that for which the turbine is designed, it is necessary to reduce the mass flow of air supplied to the combustion chamber (4) by an amount equivalent to the additional mass flow of fuel. For this purpose a controllable bypass valve (9) is provided in a duct (10) connected with the duct (3). By opening the bypass valve (9) some of the pressurized air can be diverted to operate suitable means for converting its energy into useful work, for example, an air turbine (11) which can be used to drive a power take-off shaft. In the case of plant designed to utilize gaseous fuel of low calorific value, it is convenient to couple the air turbine (11) to a compressor (12) to which the gaseous fuel is supplied through a duct (13). The fuel after leaving the compressor (12) passes through a duct (14) to the combustion chamber (4).—*British Patent No. 722,230, issued to Metropolitan-Vickers Electrical Co., Ltd. Complete specification published 19th January 1955. Engineering and Boiler House Review, March 1955; Vol. 70, pp. 104-105.*

Boiler Feed-water Deaerator

This invention relates to an apparatus for removing gas from boiler feedwater. In the plant outlined in Fig. 11, at one end of the top of the vessel (1) a spraying device (2) is fitted to which the boiler feedwater to be degasified is supplied (3) in order to be sprayed into the vessel in the form of a

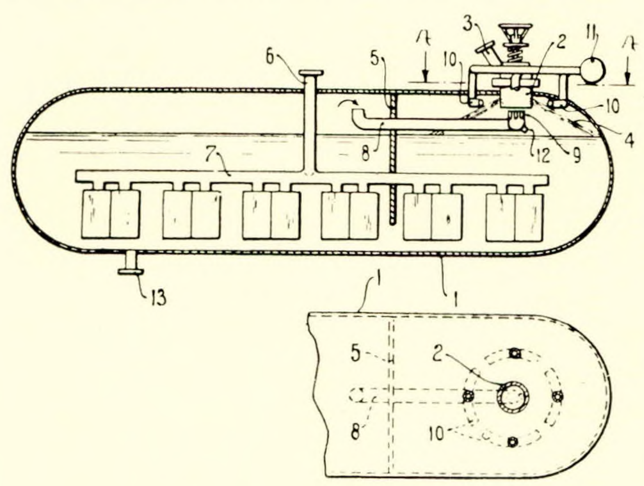


FIG. 11—*Section A-A below*

hollow cone (4). The compartment in which the water cone (4) is formed, is separated from the remaining part of the vessel by an upright partition (5) extending below the water level. Steam is supplied to the vessel by a tube (6) which has a distributing conduit branch (7). Above the water level in the vessel there is arranged a tube (8) which is completely open at one end and at its other end is provided with vertical outlet slots (9), so that the steam and gases rising out of the water at the left-hand side of the partition (5) are discharged through the slots (9) below the water cone (4) and pass through the water cone, while for the withdrawal of the gases segmental pipes (10), or a perforated annular tube, are located above the cone and connected to a common discharge conduit (11). The tube (8) has an outlet (12) for the condensate, and the vessel (1) is provided with an outlet (13) for the degasified water.—*British Patent No. 724,840, issued to Koninlijke Machinfabriek Gebr. Stork and Company. Application in Holland made on 9th November 1952. Complete specification published 23rd February 1955. Engineering and Boiler House Review, April 1955; Vol. 70, p. 137.*

Spheroidal Graphite Cast Iron

In view of the growing importance attached to the manufacture of cast iron with spheroidal graphite particles, as distinct from the normal form of graphite inclusions comprising laminations, it is of interest to note that Sulzer Bros., Ltd., have taken out a British patent dealing with this development. The claim is made by Sulzer Bros., Ltd., that they have developed a readily reproducible method for producing the separation of spheroidal graphite particles. The whole of the iron melt is desulphurized by adding cerium and is then divided into two portions. The first of these is treated with 0·05-0·15 per cent magnesium and the slag is removed. The two portions are recombined after the second has been heated to such a degree that the melt is then at the required casting temperature. The formation of the spheroidal graphite particles is effected by the pouring of the hot second portion into the cooler first portion. In order to increase the desired spheroidal graphite particles when recombining the two portions of the melt, the silicon content of the untreated iron should be 0·5-1·0 per cent. During the magnesium treatment of the first portion, the silicon content of the second, while in the furnace, is adjusted to produce 1·3-2·5 per cent silicon in the finished melt.—*British Patent No. 722,368. Sulzer Bros., Ltd., Winterthur. The Motor Ship, April 1955; Vol. 36, p. 37.*

H. and W. Piston with Scrapers for Breaking up Deposits

A mechanically operated device for breaking up deposits within pistons is illustrated in Fig. 1. The upper piston (1) of an opposed-piston engine is secured to a yoke (2). The

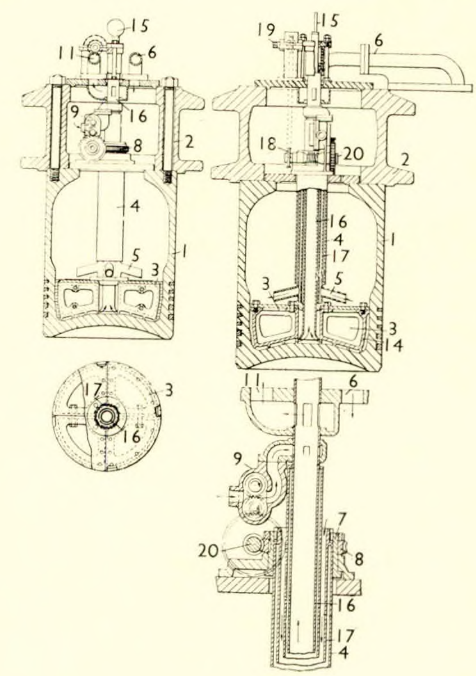
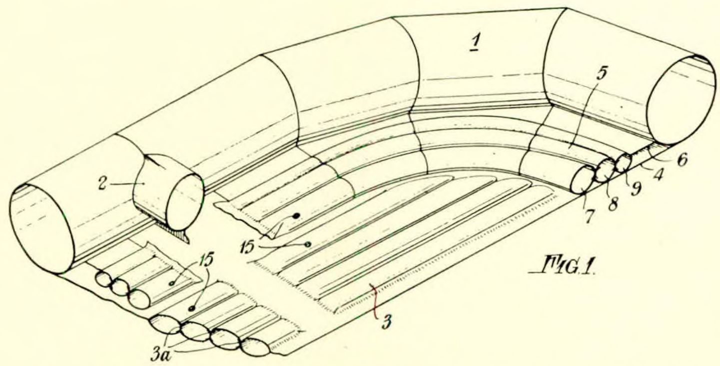


FIG. 1

casing (4) surrounding the inlet and outlet pipes of the cooling fluid is insulated from the tube (17), which has branches (5) set to give a swirling motion to the supply. Teeth on the casing (4) engage a bush (7) carrying a wormwheel (8). A motor (9) drives the wheel (8) and is rotated by the cooling fluid which enters the yoke through a pipe (6). After passing through a circulator (3) the fluid leaves by a pipe (11). The circulator consists of a number of scrapers (14) and there is an indicating disc (15) on the top of the pipe (16). For turning the scrapers by hand, a chain wheel (18) is rotated by a crank fitted on the spindle (19). The chain wheel is secured to a worm pinion shaft (20) which drives the wormwheel (8).—*British Patent No. 725,227. W. A. Harper, C. C. Pounder, and Sir F. E. Rebbeck. Harland and Wolff, Ltd., Belfast. The Motor Ship, May 1955; Vol. 36, p. 81.*

Improved Inflatable Life Raft

This invention relates to inflatable life rafts of the kind comprising an inflatable main buoyancy tube adapted to constitute the sides of the raft, there being an inflatable floor for the raft secured to the main buoyancy tube. With life rafts of the kind indicated, it is desirable to provide seating accommodation, and preferably the occupants of the raft are seated with their backs against the main buoyancy tube and with their feet extending towards the centre of the inflatable raft floor. Inflatable seats may therefore be secured over the inflatable floor adjacent to the inner side of the main buoyancy tube. Fig. 1 illustrates a life raft having a main peripheral buoyancy tube (1) of generally oval shape. A main central inflatable thwart (2) divides the interior of the raft into two halves. A floor (3) of flexible material is attached to the main tube (1) and to the thwart (2). The floor (3) is made inflatable within each half of the raft, except for a strip (4) adjacent the main tube (1), and is divided throughout its longitudinal extent



by reedings shown at (3a). Above the strip (4), lies the seating means which comprises an inflatable envelope (5) shaped to conform with the inner periphery of the main buoyancy tube (1), the envelope (5) being joined to the main tube (1) by a sheet of flexible material (6). The arrangement of the sheet (6) is such that when the inflatable envelope (5) is inflated and the raft is in use, the sheet (6) is substantially parallel to the floor (3) and extends over about half the total seat area afforded by the envelope (5) and the sheet (6).—*British Patent No. 727,187, issued to Elliott Equipment, Ltd. and V. G. Shepheard, J. H. Clarke, and J. F. Coates. Complete specification published 30th March 1955.*

Ricardo Oil-cooled Piston

With the trunk piston shown in Fig. 4, oil cooling is effected by discharges from the connecting rod through a passage (4) and a nozzle (5). Oil is retained in the space (6) and a circular plate (1) has an oval aperture (2) through which the jets of oil are delivered into a recess (3), where they collect

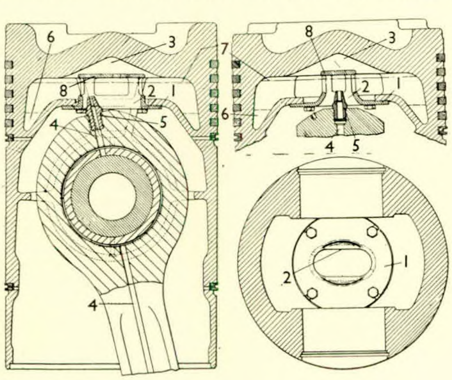


FIG. 4

by the inertia forces. During operation, the oil which remains between the space (6) and the crown of the piston provides at the mid-point of the down-stroke a depth of oil retained at the level indicated (7) lying immediately below the surface of the baffle plate (8). By altering the distance between the piston crown and the baffle plate (8) or by modifying the size of the latter it is possible to vary within wide limits the quantity of oil retained between the crown and the space (6).—*British Patent No. 724,354. G. A. Holt. Ricardo and Co., Engineers (1927), Ltd., London. The Motor Ship, April 1955; Vol. 36, p. 37.*

Means for Hoisting and Lowering Small Craft from Ships

This invention relates to means for facilitating the hoisting and lowering of small craft such as boats, lighters and the like to and from seagoing ships by derricks or similar equipment in such a manner as to prevent collision between craft and ship while permitting relative movement due to swell. By means of the present invention it is possible to carry out loading, transferring and unloading cargo on board ships in circumstances wherein the port facilities are practically non-existent or such that sea-going ships are unable to approach the shore and to have their hold unloaded. Means for avoid-

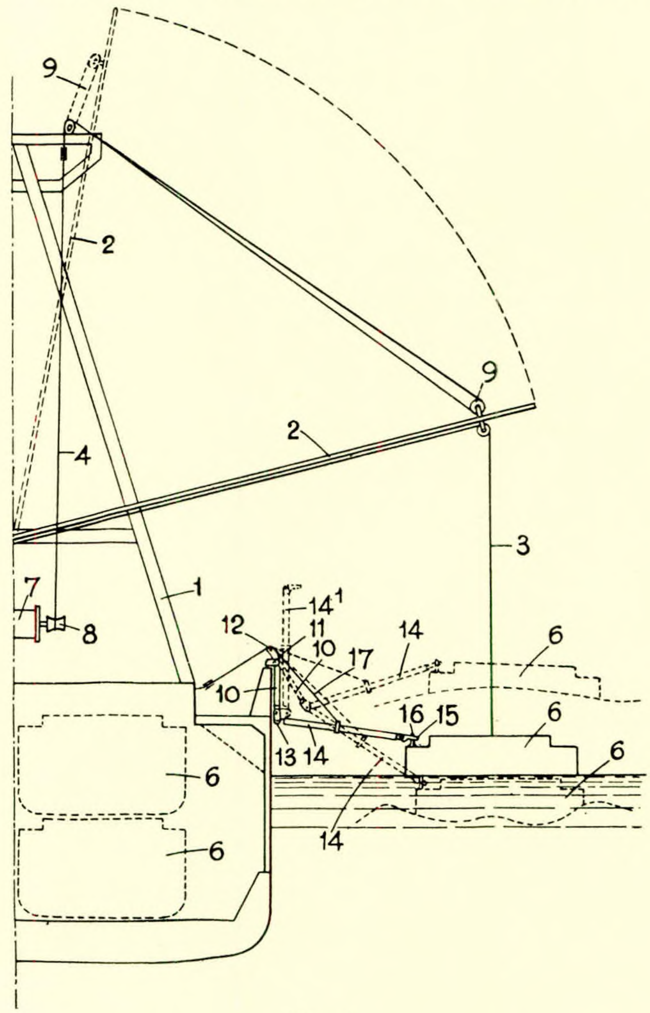


FIG. 3

ing the risk of collision between the ship and the launched or approaching boats, consists of the device shown in Fig. 3. This device consists of an articulated boom having a first section or arm (10) pivotally mounted (11) on board the ship and carrying at its free upper extension or end past the pivotal point (11) a sheave (12), whilst the other end of the arm is hingedly connected, for example through a link (13), to an outboard beam or second section (14) preferably of tubular section and made of light metal, if desired. The free end of this beam is provided with a collar or like element (15) which may be hingedly mounted (15^1), and constitutes a readily detachable pivotal connexion with a pintle (16) for example, fixed on the boat (6). It is apparent that the relative motion between the auxiliary craft and the ship is taken up by the pivotal action of the boom about its pivot (11) and by a flexing of the boom about the articulation (13). The dotted lines in Fig. 3 show various possible relative positions of the boat and ship.—*British Patent No. 725,451, issued to R. MacGregor. Complete specification published 2nd March 1955.*

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Marine Engineering and Shipbuilding Abstracts

Volume XVIII, July, No. 7 1955

Improved Aluminium-Tin Bearing Alloys

Although tin-base babbitt is a satisfactory bearing alloy for loads up to 2,500lb. per sq. in., its tendency to fail by fatigue at higher loads has led to a demand for a bearing alloy possessing the softness and good anti-friction properties of babbitt with a fatigue resistance at least as high as that of copper-lead. Attempts to meet this by aluminium-base alloys containing tin were at first unsatisfactory, as an increased amount of tin content in these alloys led to a deterioration in their mechanical properties at the working temperature. Means have now been found of overcoming this disadvantage and of raising the tin content to 30 per cent or more while still maintaining acceptable mechanical properties at temperatures up to at least 300 deg. F. Bearing tests and preliminary engine trials have shown that these new aluminium-tin alloys are satisfactory bearing materials, with adequate fatigue resistance to meet heavy duty demands, as encountered in Diesel engines, for instance; compared with copper-lead, they are less prone to cause journal wear, and no more expensive in cost of metal. The use of bearings of this new alloy is likely to be greatly increased if a satisfactory method of bonding them to steel can be devised. So far, the greatest success has been achieved with a rolling-on rather than a casting-on process, although one leading bearing maker has successfully bonded aluminium-tin alloys to duralumin by a casting-on process. It is recommended that before being put into service, all aluminium-tin bearing alloys should be tinned by immersing them for a few seconds in a hot sodium stannate solution.—*J. W. Cuthbertson and E. C. Ellwood, Metal Industry, No. 5, 1954; Vol. 85, p. 3. Journal, The British Shipbuilding Research Association, January 1955; Vol. 10, Abstract No. 9,814.*

Modern Cargo Vessels with Three-phase A.C. Equipment

A new cargo ship is under construction at the Nordseewerke Emden for Norwegian owners, in which extensive use of a.c. will be made for auxiliary drives. The cargo winch motors will be of the Siemens multiple-speed squirrel-cage type; they will have three stator windings with 32, 8, and 4 poles, corresponding to hoisting speeds of 0·49, 2·13, and 4·3ft. per second respectively when driving a 3-ton winch. The performance of these winches is said to compare favourably with that of d.c. winches, and their maintenance is simpler. The anchor windlass of the new ship will be driven by two a.c. motors, the larger of them to be used for weighing anchor and warping the ship, and the smaller one for hauling the free hawser at twice the speed when under load. To haul in the free hawser, the larger motor, which is of the wound-rotor slipring type, is first accelerated in stages up to its full speed, after which the load is switched over to the smaller and faster motor. Should the latter become overloaded, the drive is switched back to the larger unit. The various engine room auxiliaries and the steering gear incorporate squirrel-cage motors arranged for direct-on-line starting. The electrical power plant of the ship consists of three main 170-kVA Diesel alternators, and one 36-kVA Diesel alternator for port duty. The three main alternators work in parallel, and are equipped with specially developed voltage regulating devices which will almost instantly compensate for the voltage drops caused by the starting up of the squirrel-cage motors. The exciting current for the alternators is taken from the bus-bars through reactor coils and rectifiers; it rises as the bus-bar voltage drops, thereby bringing the generated voltage back to its normal value. No moving contacts are needed in this type of regulation. The ship will make extensive use of fluorescent lighting.—*Norwegian Shipping News, 1954; Vol. 10, p. 1261. Journal, The British Shipbuilding Research Association, January 1955; Vol. 10, Abstract No. 9,835.*

Lauritzen Arctic Ship

Lauritzen Lines, the owners of the famous Arctic vessel *Kista Dan,* have ordered a similar but larger and more powerful vessel from the Aalborg Vaerft. The new ship will have a d.w. capacity of 1,800 tons in three holds and a 2,200 i.h.p. Burmeister and Wain Diesel engine will give a speed of 13 knots. Following the successful features of the *Kista Dan,* she will be fitted with an Aalborg variable-pitch propeller with ice fins forward of it and an ice knife to protect the rudder when running astern. Naturally, she will be built to verv heavy scantlings, and the owners claim that she will be the most heavily-built Arctic vessel in existence (ice-breakers excepted). Another interesting item will be the enclosed navigation room at the head of the foremast where there will be a control position, with wheel, engine speed and propeller pitch controls, etc. Access will be through the hollow mast. The superstructures will be extended in size and thirty-six passengers can be carried. An unusual feature is that the holds are not only arranged for cooling down to − 20 deg. C. under tropical conditions, but they can also be heated when required for Arctic service.—*The Marine Engineer and Naval Architect, February 1955; Vol. 78, p. 68.*

Submarine Television Checks Fish Nets

Underwater television was used recently to evaluate new equipment for the fishing industry. At the request of the Department of Interior's Fish and Wildlife Service, the U.S. Navy Bureau of Ships used television apparatus to test a midwater trawl used for herring, sardines and other fish. The net is a Swedish development and its original model has expensive component parts and requires several men to handle it. Experiments were conducted to ascertain whether an adapted form, constructed at lower cost and requiring fewer men to operate, would be as effective. The net, about 150 feet long, has a 40-foot square mouth. To monitor the net while it caught fish, a television camera was attached to a towed buoy which permitted a control of the depth while in tow. An image orthicon designed for underwater use was used. The camera is 5 inches in diameter and 42 inches long without the pressure housing. It is capable of picking up scenes having a considerably lower light level than most conventional equipment. Change of focus, aperture, and lens is accomplished by remote control from the surface operating position. As a result of the tests it was determined that the modified version of net performed as well or better than the more complicated original net.—*Electronics, March 1955; Vol. 28, p. 26.*

Danish Two-stroke Diesel Without Crankshaft

A new type of opposed-piston two-stroke Diesel engine without crankshaft, designed primarily for trawlers, tugs, and launches, has been developed by Svanemölle Company, of Copenhagen. It is a small and compact engine, capable of rapid acceleration, while retaining the features of a slow-running, heavy-duty, direct-coupled marine propulsion unit. For economy and reliability, uniflow scavenging was adopted. The piston forces are transmitted by rocker arms, in the same way as in previous designs of Junkers and Sulzer, so that the side thrust of the piston is extremely low. Previous double-piston designs had normal crankshafts with two throws per cylinder, but these, in multi-cylinder engines, gave rise to serious torsional vibration problems. These problems have been avoided in the new design. The method of power transmission from the piston via a rocker arm to the cam on the main rotor shaft eliminates vibration and also permits a reduction in the piston speed. The engine is a three-cylinder, double-piston design, with a bore of 150 mm. and 2 × 150 mm. stroke. The rotor shaft, provided with cams, is supported in roller bearings and runs at 350 r.p.m., corresponding to twice as many working cycles per minute for each cylinder. The engine develops 150 b.h.p. at a mean effective pressure of 87lb. per sq. in. The cams, made of ball bearing steel and hardened to 63 Rockwell-C, are mounted, together with cylindrical intermediate sections, on a plain rotor shaft which carries

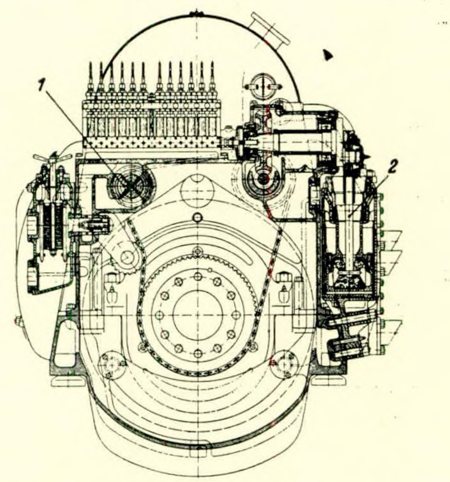


FIG. 2—*Section through auxiliary drives*

(1) Oil pump; (2) Double-acting water pump

a flywheel at its flanged end. Each line of cylinders is cast integrally in one block and bolted to cast steel cross webs; these transmit the pull of the rocker arm bearings in such a way that the grey cast iron parts are not subjected to tensile loads. The cylinder block, the cast steel cross webs, and a set of cams and rocker arms thus form a sub-assembly, and make possible the construction of a range of engines with two,

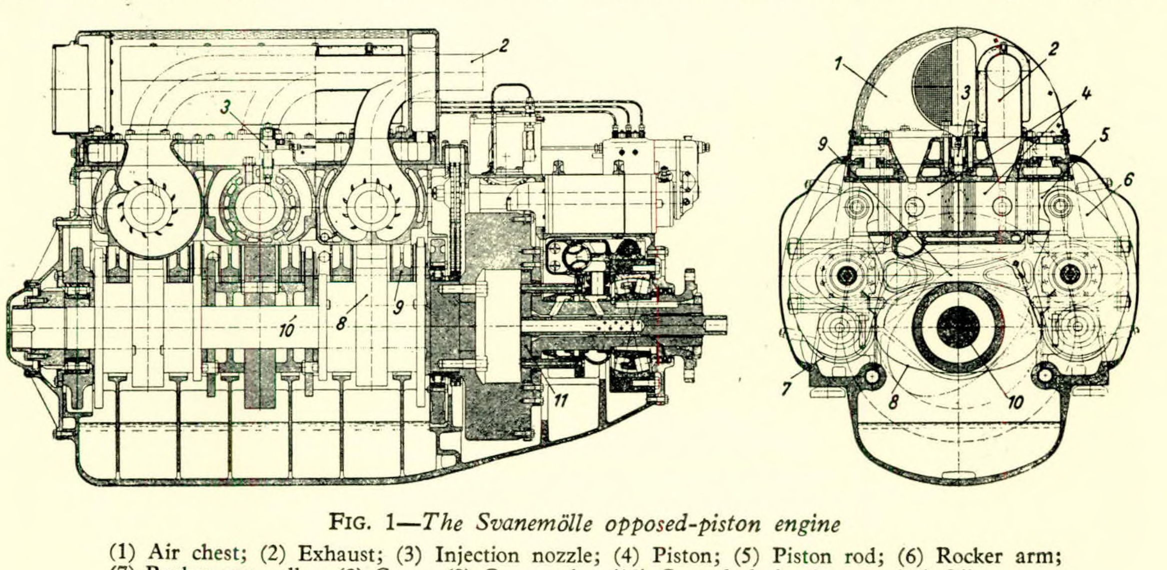


FIG. 1—*The Svanemölle opposed-piston engine*

(1) Air chest; (2) Exhaust; (3) Injection nozzle; (4) Piston; (5) Piston rod; (6) Rocker arm; (7) Rocker arm roller; (8) Cams; (9) Cross webs; (10) Central shaft of rotor; (11) Oil pressure piston for adjustment of variable pitch propeller.

three, four, five and six cylinders, all of which can have the same auxiliaries, controls, etc. The lubricating oil vane-type pump is also designed as a servo-motor for the control of variable pitch propellers. The piston-type water pump, driven through a worm gear (on the right), serves as a bilge pump on the lower side and as a cooling water pump on the upper side of the double-acting piston. The same pumps are employed on all engines; their delivery can be adapted to the number of cylinders by using a worm drive with a number of starts equal to the number of cylinders.—*The Engineer's Digest, March 1955; Vol. 16, pp. 106-107.*

Contamination in Lubricating Oil Systems

During the last several years, contamination in lubricating oil has caused an excessively large number of casualties to sliding surface journal bearings in United States Navy machinery. These and many other casualties would not have occurred if dirt and contamination had been kept out of the lubricating oil systems. A contaminant in the oil is any substance that the oil refinery did not put there for a specific purpose. The term includes water, sand, weld splatter, glass wool, and just plain dirt. The worst offender of all is the agglomerate into larger particles which may obstruct oil flow. The thickness of the minimum clearance space may be calculated, or it may be directly measured with a newly developed instrument. Fig. 2 is a graphical record of actual measurements of the minimum oil film thickness observed at various shaft speeds for a vertical mounted main feed pump thrust bearing. Fig. 2 shows that at full speed the thrust bearing developed a film of oil slightly less than one one-thousandth of an inch in thickness at the point of closest approach of the rotating thrust collar to the bearing. Thus, abrasive particles must be smaller than one one-thousandth of an inch if they are not to damage the bearing face during normal operating conditions for this pump. The graph shows that the oil film thickness rapidly approaches zero as speed drops below 2,000 r.p.m. Since it is impossible to stop the pump without its passing through the lowest speed range, it would appear that abrasive particles, no matter how small, will damage the bearing. Were the machine to run without ever having to stop, and if abrasive particle sizes contaminating the oil were less than one one-thousandth of an inch in diameter, the bearings would never wear out as a result of any abrasive contaminant in the lubricant. Measurements taken on journal bearings of

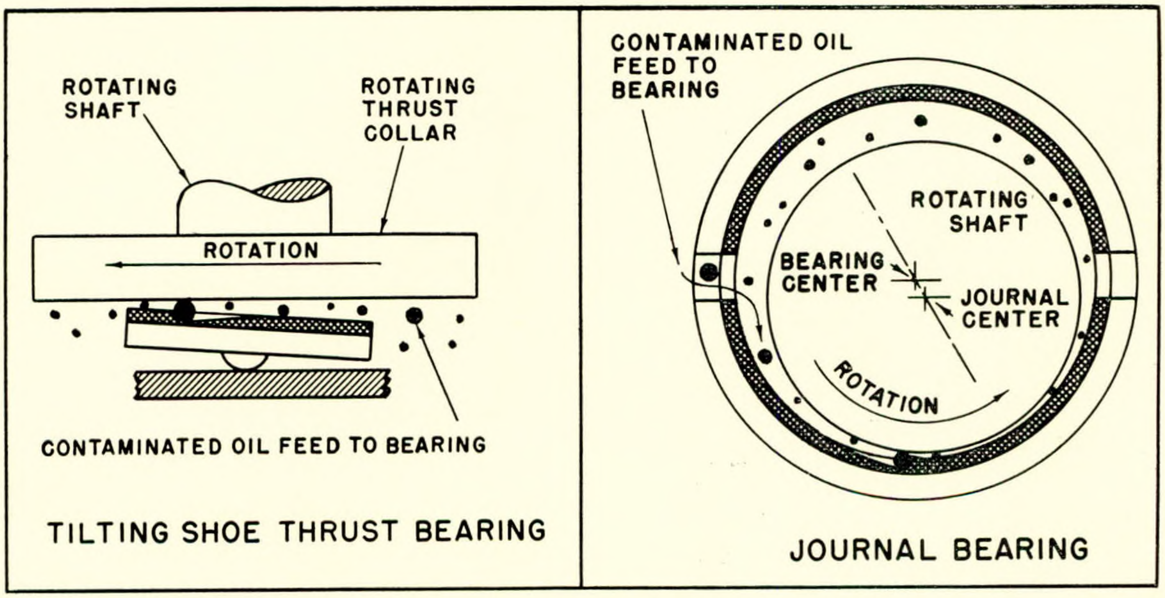


FIG. 1—*Contamination particles in the bearing oil. Note how the larger particles, which are thicker than the thin oil film, cut grooves in the bearings and become embedded*

relatively small grit or abrasive particle, which does not form plugs to shut off oil supply, but flows along with the oil, cutting and grinding away bearings and journals. Sand particles, metal particles, rust, glass, nuts, bolts, cotter keys, and tachometer tips are some of the solid contaminants which have been found in supposedly clean lubricating oil systems. Nuts, bolts, and tachometer tips generally settle out in the sump or are retained in the strainer. Unless they pass through a gear mesh, these large particles do not cause nearly so much trouble as the small abrasive particles that pass through the coarse mesh of the strainer and enter the bearing. The oil film clearance, measured at the point where the rotating shaft approaches closest to the bearing, is the bearing bottleneck. The abrasive particle must pass through this minimum clearance space in the manner shown in Fig. 1. If the hard particle is larger than the minimum oil film thickness, it must cut, displace, or grind a channel in the bearing and the journal. If the hard particle is smaller than the minimum clearance space, it may pass through without appreciably damaging either the bearing or the journal. Some particles may be crushed, breaking into smaller pieces which safely pass through, but the damage is done at the point of crushing. Other particles may feed pumps, propulsion turbine, turbo-generator sets, and the like, give essentially the same results. Abrasive particles smaller than one one-thousandth of an inch can be tolerated as long as the machine is running continuously and provided the finish of the bearing and journal surfaces is smooth. However, when the equipment slows down and stops, the oil film thickness decreases almost to zero. Similarly, during a brief period when the equipment is starting, the supporting oil film is practically non-existent and the bearing is lubricated by an extremely thin layer. Any solid contaminant will damage the bearing surface during this critical period. Aggravating the situation is the present-day tendency to design rotating shafts in a vertical position that requires the entire rotor weight to be carried by a thrust bearing. In such a case, the full weight load must be carried by the thrust bearing during starts and stops, and during periods of idleness. When vertical units are secured, the pivoting action built into the supporting thrust bearing shoes allows the shoes to align themselves parallel to the thrust collar. After some hours, the oil film separating the parallel shoes from the steel thrust collar squeezes out. The two almost perfectly flat surfaces trap abrasive particles between them. At first glance, the situation seems hopeless.

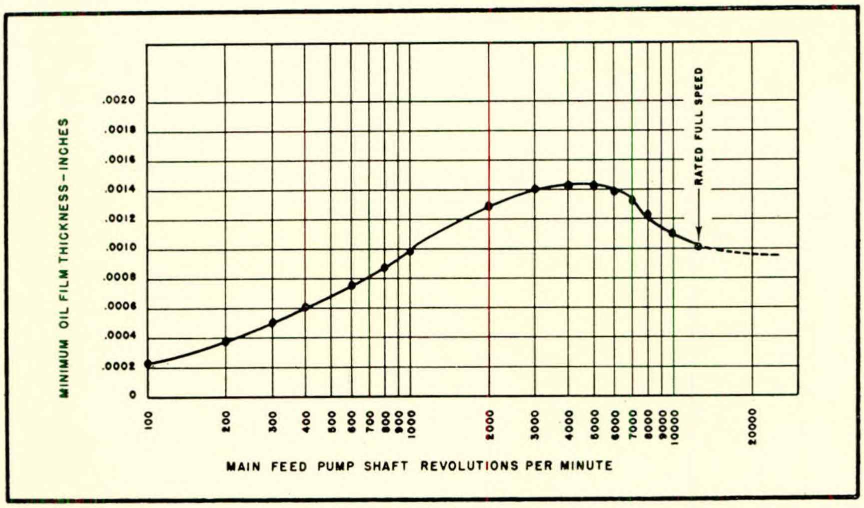


FIG. 2—*Bearings are particularly liable to damage when the shaft is turning at low speeds just after starting and just before stopping. At these instants the protective oil film is at its thinnest*

for any solid contaminant, however minute, cannot be tolerated. However, the situation is far from hopeless. Much is to be gained by reducing the amount of contamination of lubricating systems. Bearing life can be prolonged. Assuming no interruption of oil flow, wear will occur only during the starting and stopping periods of machinery operation. Therefore, the smaller the dirt particle, the shorter the time for abrasive grinding of babbitt and journal during start-stop intervals. The fewer the abrasive particles in the lubrication system, the fewer the particles that will be carried through or trapped in the thin oil film. Consequently, bearing life will be prolonged by reducing the size of abrasive particles in the lubrication system. This key fact must be kept constantly in mind.—*G. C. Ambler, Bureau of Ships Journal, February 1955; Vol. 3, pp. 13-16.*

Running-in of Engines

In the paper are described factors which affect the mating of piston rings and cylinder bores in newly assembled Diesel and petrol engines. It is shown that, if the correct cylinder bore surface finish is used, rapid running-in, or break-in as it is sometimes termed, may be achieved when either straight mineral or additive-type engine oils are used. When a reciprocating internal-combustion engine is first assembled, perfect congruence between the piston rings and the cylinder bores does not exist. During the process of running-in the high spots on piston rings and cylinder bore are worn away, and break-in may be considered complete when mating of these surfaces has occurred. Since diametral wear of the piston ring usually occurs at a greater rate than that of the bore, it is normally the rate at which the bore wears that determines when the engine is run-in. Although wear of the bore is relatively rapid at the ends of the piston stroke, in the middle of the stroke wear occurs much more slowly. However, the engine is not completely run-in until satisfactory mating of rings and bore is achieved throughout the stroke. Until this process is complete the engine exhibits a relatively high rate of oil consumption and there is a danger that "scuffing" or "galling" of the rubbing surfaces may occur under high load conditions. Since the time taken for the engine to break-in completely is determined by the rate at which the "high spots" are worn away from the new components, the duration of the running-in period is generally extended by factors which reduce the rate of wear, for example, improved materials of construction or the use of heavy duty lubricants having anti-wear properties. While in the latter case a straight mineral oil may be used for running-in, this solution is not always practicable as many modern engines are so highly rated that an additive-type engine oil must be used at all times to prevent the formation of excessive deposits. In such circumstances the use of a moderately rough cylinder bore surface finish has been found to give a satisfactorily short running-in period.—*Paper by K. R. Williams and S. G. Daniel, read at a general meeting of the Institution of Mechanical Engineers, on 8th February 1955.*

Terylene Ropes

Trials of barge ropes—in sisal, nylon and Terylene—recently carried out on the Thames, showed that, of the types under test, Terylene was the most efficient and economical. All the ropes tested were of equal diameter and were used for identical towing duties. It was found that a sisal 6-in. circumference rope—the standard rope used on the Thames—lasted six days, while a Terylene 6-in. circumference rope lasted 137 days. Although synthetic ropes are more expensive, their extremely long life make them economical. The trials were conducted by one of the largest lighterage companies operating on the Thames, viz. William Cory and Son, Ltd., with the object of reducing the frequency with which their barge tow ropes have to be replaced. Because the fibre absorbs little moisture, Terylene ropes remain flexible when wet, and the lightermen say that they are much easier to handle. They are stated to be twice as strong as equivalent ropes made from natural fibres, are rot proof, and have good resistance to sunlight. An additional advantage, especially important when the ropes are to be used in confined waterways, is that they stretch much less than some other types of synthetic rope. Friction between strands, therefore, is low—a factor which makes for long life. Other marine outlets for Terylene are in hauling lines, whaling fore-loopers, lifeboat covers, awnings, tarpaulins, log-lines, fish netting and trawl cod-ends.—*The Shipbuilder and Marine Engine-Builder, March 1954; Vol. 62, p. 169.*

New German Dry Dock

An important addition to the ship repairing facilities on the North Sea coast of Germany is the new dry dock recently completed at the shipyard of Nordseewerke Emden, G.m.b.H., Emden. This dock, which is designed to be used both for ship repairing and also for shipbuilding purposes, is capable of accommodating vessels of up to 38,000 tons deadweight, and as such can provide docking facilities for all but the largest tankers now being built. The dock is 714ft. 8in. in length, with a breadth at the entrance of 98ft. 4in., and ships can enter at draughts of up to 22ft. 10in. The dock gate is of the caisson type, and the pumping arrangements allow the dock to be pumped dry in rather over two-and-a-half hours when empty.

With a vessel of 38,000 tons deadweight inside, this time is reduced to two hours. The flooding of the dock takes about one hour. The dock is situated parallel and adjacent to two building berths. It is served by two travelling cranes of 25 tons capacity each, and one of 16 tons capacity, two of these cranes being also available for the nearest berth. At the head of the dock is a clear space for welding prefabricated assemblies: the crane tracks extend far enough back to allow the dock's cranes to cover this space. The dock is built on sand, and the dock bottom has a thickness of over 6ft. of concrete at the centre.—*The Shipping World, 16th February 1955; Vol. 132, p. 218.*

Failure of Diesel Engine Pistons and Cylinder Covers

This paper reports on research which was carried out on behalf of the British Shipbuilding Research Association. The stresses that occur in combustion chamber parts are discussed with regard to their nature, magnitude, and likely contribution to failure. The physical and mechanical constants of the material which affect the thermal stresses are given, and the modifying influence of plastic deformation is considered. It is deduced that cracks which commence at the hot side of the walls are due to residual stresses on cooling down, following plastic flow while running. This form of crack is shown to be most likely to occur in cast iron (where ductility is low) and the authors have not so far encountered it in steel components. The danger spots are in the hot face of the cover (particularly where high thermal compressive stresses occur initially as a result of asymmetry) and in the hot face of the piston crown. The second principal cause of failure is the fatiguing action of the repeated combustion pressures, combined with the tensile stresses occurring as a result of temperature gradients in the walls. The cracks occur at the horizontal junction of the crown and wall of pistons, and where webs and other scantlings meet the cover plate in cylinder covers. Failures of this type commence at the cooled side of the walls and occur in steel as well as cast iron components. The effects of growth, additional stresses accompanying load changes, and the fluctuating surface stresses caused by the varying gas temperature are small, but the authors prefer to regard them as contributing factors causing somewhat earlier failures. Typical failures are discussed. Radial cracks in cylinder covers and circumferential cracks in pistons are selected for detailed study. Direct estimates of the initial thermal compressive stresses in an unsymmetrical cover range from 16 to 28 tons per sq. in. Being higher than previous estimates, these values support the residual stress theory of failure. A theoretical investigation of the stresses in an idealized piston shows that cracking at the junction of the crown and skirt is caused by fatigue. Examination of four actual failed pistons confirms this finding. Calculated stress curves are drawn showing the variation of piston stresses with changing wall and crown thickness. The stresses due to combustion pressure are independent of the scale of the piston, but dependent on its proportions. This applies to the thermal stresses also, provided that the temperature distribution is the same. Since the piston temperatures increase with increasing size, the thermal stresses in large pistons are greater than in small ones. The stresses produced by the combustion load are shown to be smallest when the crown and wall are thick. For low thermal stresses, a thin crown is necessary; for low thermal stresses in the wall, the wall should be thick, but a thick wall causes high thermal stresses in the crown. An example has been worked out to show graphically the variation in the overall stress conditions at the inner edge of the wall and at the inside face of the crown centre as the crown and wall thicknesses are varied. For the conditions assumed, the wall and crown should be approximately of equal thickness and one-tenth of the piston diameter. These figures include consideration of the compressive stresses in the hot face of the crown. The conclusion is drawn that where circumferential cracking has occurred in the wall, in the absence of cracks elsewhere, design changes should be directed at reducing the stresses at the position of failure, even though the stresses at other vital positions may be increased as a result. This requires the wall to be thickened, which would raise the temperature of the upper ring, but the tendency would be counteracted by the cutting of suitable grooves in the piston wall. Alternative designs are suggested which might be useful in overcoming cracking troubles in the pistons of large or highly rated engines. —*Paper read by D. Fitzgeorge and J. A. Pope, at a meeting of the North East Coast Institution of Engineers and Shipbuilders, 11th February 1955.*

Measurement of Sand Transported by Suction Dredger

In Holland, where the use of suction dredgers for land reclamation is extensively practised, means have recently been developed for measuring the total quantity of solid matter moved by the dredger. The method of operation of the installation is as follows: (*a*) A Tobiflux flowmeter measures the rate of flow of the sand/water, or earth/water mixtures, a function which cannot be performed satisfactorily by any other type of flowmeter hitherto available. This flow measurement is independent of temperature, sand concentration, weight, electrical conductivity, salt-content, etc.—speed only is measured. (*b*) A Tobicon concentration meter measures the concentration of sand in the water as a percentage, also independently of temperature, sand concentration, weight, electrical conductivity, salt-content, etc. (*c*) A third instrument gives the product of flow and concentration, i.e. quantity being transported, by means of which optimum operating conditions can be maintained at all times. In addition if required a totalisator can also be fitted, showing total quantity of solid transported over an hour, day, week, etc. The construction of the equipment is such that no obstacles or restrictions exist in the pipe, measurement being via electrodes.

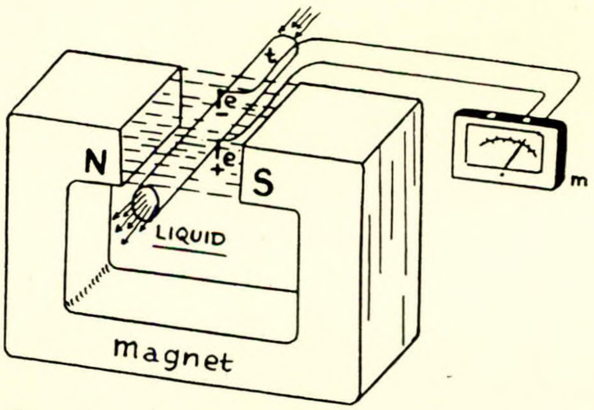


FIG. 1—*Principle of operation of the Tobiflux Flowmeter*

The method of operation of the Tobiflux flowmeter is as follows: When an electric conductor is moved in a magnetic field, an electromotive force (EMF) is built up in the conductor, proportional to the number of magnetic force lines cut by the conductor during unit time. If, as a moving electric conductor, a conducting liquid is taken flowing through a tube of non-conducting material in a magnetic field an EMF will be built up in the liquid. By applying two electrodes opposite each other and perpendicular to the magnetic field, through the tube wall into the liquid, a potential can be measured between the electrodes. This is dependent on:—

1. The distance between the electrodes.
2. The strength of the magnetic field.
3. The velocity of the liquid flowing through the tube.

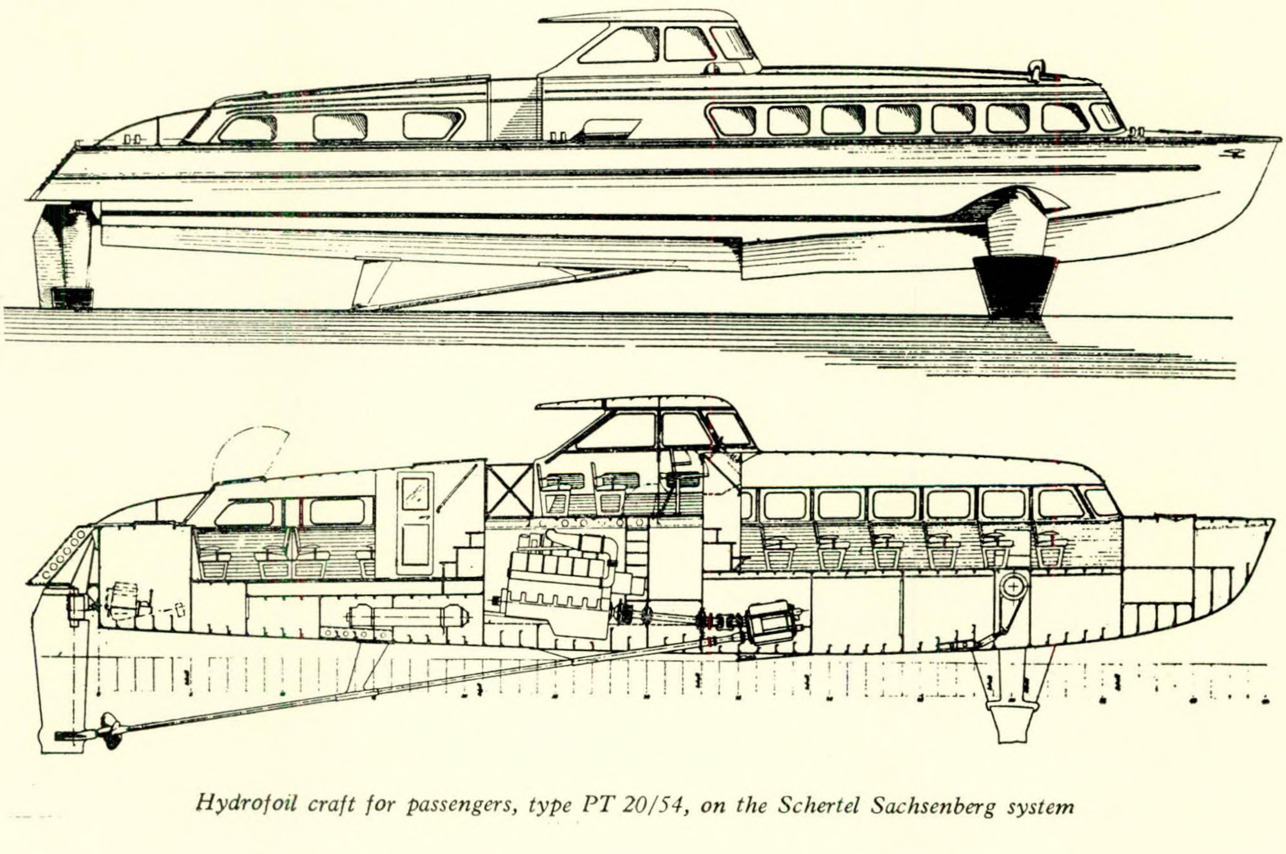
So if the geometric dimensions and factors 1 and 2 are kept constant the velocity or rate of flow of the liquid can be measured by measuring the potential between the electrodes. Fig. 1 illustrates this principle. Electrodes "e" are let into the insulating wall of tube "t". As soon as liquid flows through the tube, the magnetic field will induce an electromotive force between the electrodes. This EMF can be read on millivoltmeter "m", directly calibrated in units of rate flow. Although it has been explain the preceding section, that a direct linear measurement of liquid flow should be possible, a number of factors tend to interfere. Special precautions in the measuring apparatus are necessary to eliminate or compensate these factors. The Tobicon concentration meter measures the concentration of solid matter carried in a transporting fluid, the percentage concentration being read off a linear scaled indicator. The instrument is based on the principle that the conductivity of a mixture of fluid and solid matter is a function of the concentration of solid matter in the carrying fluid where the electrical conductivity of the fluid and the solid matter are different. The design of the instrument includes two continuous flow type measuring cells, the electrodes of one being immersed in the fluid at an appropriate position in the pipe line where solid matter is being transported hydraulically. The electrodes of the second cell (reference cell) are immersed in fluid at a position where solid matter is not being carried.—*Process Control, No. 1, Vol. 2, pp. 18-20.*

New and Larger Hydrofoil Boats

Based on experience gained with various hydrofoil boats which have been constructed in recent years, the building of some new and all-round interesting types has now been started. These new units are of the larger class and are equipped with several innovations which shall now be tried for the first time on passenger boats. Two of the new types, of approximately equal size, deserve special attention: one is built by Lürssenwerft, Bremen, the other one by Cantiere Navale Leopoldo Rodriquez, Messina. In Messina, three boats were simultaneously put in hand; they represent a standard type without any complicated details. The Messina boats are intended for passenger traffic along the coast, while the Bremen boat is built for inland service. Both types have been designed by Supramar, Ltd., Zug (Switzerland), according to the Schertel-Sachsenberg system. Some details of the Bremen boat have, however, been designed by the firm who had ordered her because this firm (Nautik GmbH., Bremen) intends taking up technical activity in the future. Following are the data of the "Supramar" boats above mentioned and henceforth called "Bremen"-type and "Messina"-type:—

	Bremen	Messina
Year of design...	1953	1954
Displacement, tons	22	26
Overall length, metres	20·0	20·7
Beam, metres	4·8	4·8
Span of hydrofoils, metres	7·8	7·8
Draught of hull, metres	0·95	1·0
Draught, hydrofoils included, floating on spot, metres	2·6	2·6
Draught, cruising on hydrofoils, metres	1·1	1·15
Draught, hydrofoils retracted, floating on spot, metres	1·6	—
Engine power, h.p.	2 × 650	1,350
Maximum speed, approximately, knots	38	40
Cruising speed, approximately, knots	34	35
Fuel consumption for cruising speed, approximately, kg. per hr.	185	140
Range, km.	300	450
Deadweight capacity, tons	6·0	7·0
Number of passengers	60	72

The light alloy metal out of which the hulls are made is proof against corrosion from sea water; exterior shape and size of both types are practically identical. Both types have a double bottom, subdivided into a large number of watertight compartments, thus providing the best safety in case of accident. The superstructures of the Messina boats are of a stronger build because they must be able to resist rough sea, also when floating, even if they are used only for traffic along the coasts and in favourable weather conditions, according to their comparatively small displacement. Yet both hulls should be considered as light constructions, similar to the build of flying boats. The light construction is decisive as regards the commercial (and eventually military) utilization of hydrofoil boats, because the deadweight capacity can only be used to a satisfactory extent if the weight of the hydrofoils (built of steel)



Hydrofoil craft for passengers, type PT 20/54, on the Schertel Sachsenberg system is somehow compensated by a lesser weight of engines and hull. The hulls are riveted, because welding of light metal did not seem to provide sufficient safety. The stresses on the hull of hydrofoil boats greatly differs from the stresses on a normal high-speed craft and require that the strength of materials be fully and entirely utilized. The available power of the engines is practically the same in both types, but the engines and gears are quite different. The Bremen boat has two Hall Scott petrol engines of 650 h.p. each, driving two propeller shafts. The Messina boats have one Daimler-Benz Diesel engine of 1,350 h.p. This means that the power available on the Bremen boat is 59 h.p. per ton and on the Messina boats 55 h.p. per ton. Yet the maximum speeds will be practically identical because of the less favourable resistance conditions of the twin propeller shafts on the Bremen boat.—*K. Büller, European Shipbuilding, No. 1, 1955; Vol. 4, pp. 5-8.*

Oil Sump Tank for Main Reduction Gears

Development work on the design of lubricating-oil sump tanks for the USS *Saratoga* (CVA-60) at the New York Naval Shipyard has resulted in some significant changes in the structural, lubricating, and production aspects of this unit. Design details were worked out by using a scaled mock-up of the sump tank made of transparent plastic. Thus details which would be difficult to visualize on blueprints could easily be seen in the working model. In addition, the dynamic actions involved could be readily demonstrated. The joints of the model in the sump tank region were sealed to permit filling with liquid. The motion of the oil due to the ship's movement was duplicated by placing the model on a rocker platform, and a protractor with weighted pointer indicated the angular position for each trial inclination (see Fig. 1). Since the model is very accurate, the effect of oil-volume change could be checked directly by changing the amount of liquid in proportion to the scale of the model. The major factors to be considered in the design of the sump tank were its structural features, oil level and flow path, air venting, and deaeration of the oil. In order to make the sump tank structurally sound, it had to be formed within the confines of the main reduction gear foundation without penetrating the inner-bottom plating. This plating is an important part of the ship's girder, and such penetrations are undesirable especially in the vicinity of amidships where they would necessarily be located. Lubrication requirements dictated that an adequate suction head on the foot valves be maintained for all positions of heel. The top baffle plates control this submergence and also act as structural ties between the transverse framing under the bull-gear gearing supports. Cross-venting the upper portions of the sump's port and starboard wings under the baffle plates is accomplished by using the chamber located in the foundation structure immediately forward of the bull-gear chamber as an air passage. With suitable web cutouts and high corner vent openings, the air is permitted to circulate freely from port to starboard without pocketing. To permit maximum deaeration of the oil, the flow path from the gravity-dump point directly beneath the bull gear is arranged in two symmmetrical paths. The first moves athwartships from the gear centreline to port, then aft, then returns athwartships to starboard to the gear centreline. The second path was to starboard, aft, and return. These paths join at the foot-valve suction face of the circulating pump line. Drain lines from the bearings discharge into the same path as the oil from the gears. The oil moving through the sump tank has a low velocity, and the diverted flow path provides ample surface area in which to permit the rise of any entrained air bubbles. A low point is created in the sump by inclining the bottom plating in the general direction of oil flow. This permits stripping off water and contaminated oil and also provides for complete drainage of the tank. All pockets formed in the structure above are drained by limber holes. A clearance

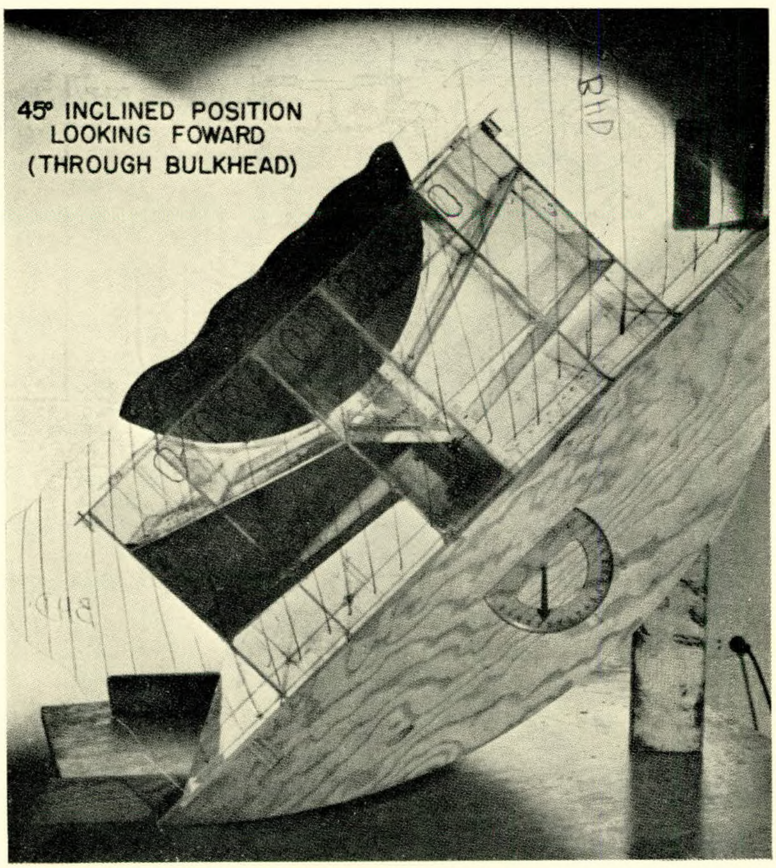
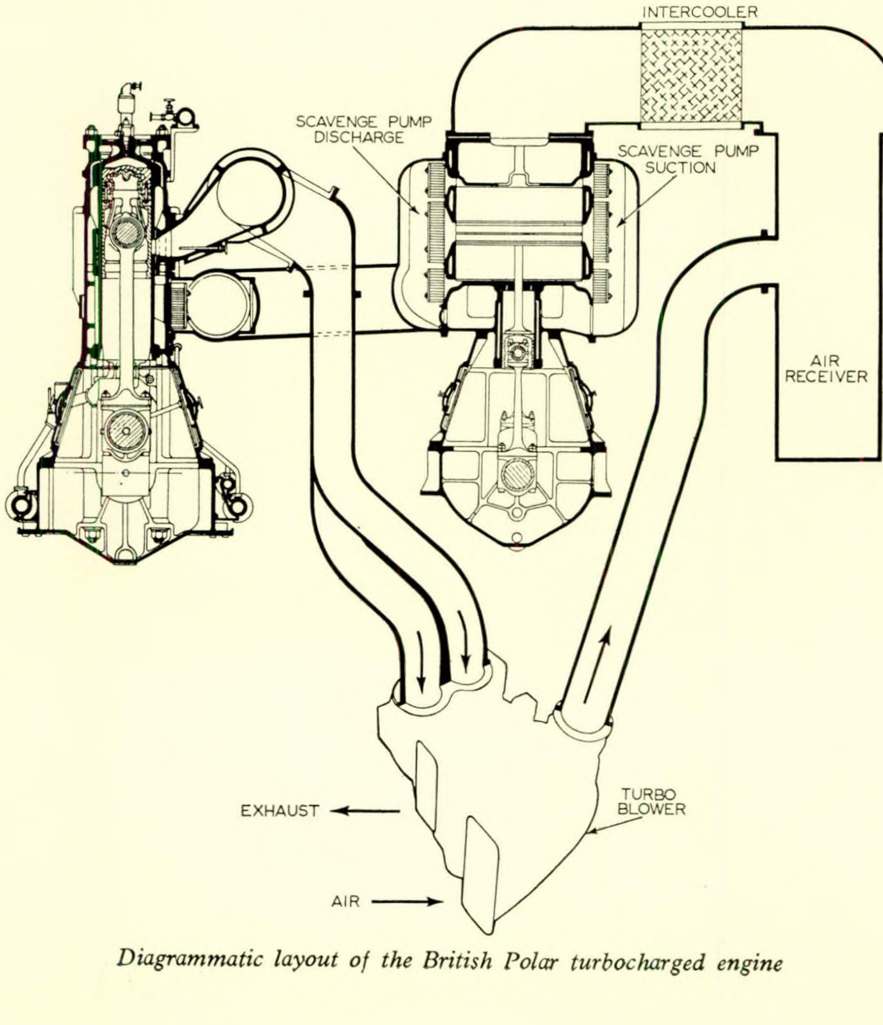


FIG. 1—*45 degrees inclined position*

of six inches is maintained between the foot-valve face and the sump bottom to permit the contaminated oil to collect. Access to the voids under the sump is provided by manholes. To prevent leakage of oil and oil vapour into the machinery room, a seal is provided between the bottom of the reduction-gear case and the gravity-dump opening in the sump. This prevents leakage adjacent to the liners at the bolting faces between the gear case and the top of the foundation. Production requirements that the reduction gears be shop-assembled necessitated the development of an attached seal. By having the manufacturer provide relatively thin (¼-inch) gear case attachment strips, the seal can be welded in place on the ship without distorting the heavy, machined plates of the gear case. The connexion to the gravity-dump opening is made by welding strips between the seal and the tank. Welding in the vicinity of the exposed bull-gear teeth is thus kept to a minimum. The seal serves a second function in protecting the gear during shipment from the assembly shop to the ship.—*Bureau of Ships Journal, Vol. 3, February, 1955; pp. 44-45.*

Turbocharged Engine

A British Polar two-stroke engine giving one-third greater output than the normally-aspirated unit has been running experimentally for some time, on marine Diesel fuel, on one of the test beds at the works of British Polar Engines, Ltd. The engine employed is a seven-cylinder example of the M.M. class with a 340-mm. bore (13.39in.) and a 576-mm. stroke (22.44in.). It is a reversible open-chamber loop-scavenge marine propulsion unit with oil-cooled pistons and is normal in all respects, except for the addition of a Napier turbocharger, air receiver and Serck gas-to-water heat exchanger. Ability thus to adapt standard existing or new engines having from four to nine cylinders offers useful commercial advantages. The turbo-blower has its own independent cooling and lubricating systems. In its unsupercharged version the rating is 1,310 b.h.p. at 300 r.p.m.; when the turbocharger, receiver and cooler are added, the output is raised to 1,750 b.h.p. at the same crankshaft speed, the respective mean effective pressures being 78.3lb. per sq. in. and 104.8lb. per sq. in. at full load; the piston speed remains the same at 1,120 ft. per min. There is considerable saving in length. The normally aspirated nine-cylinder engine measures 33ft. and develops 1,700 b.h.p. The pressure-charged engine with seven cylinders is 27ft. 5in. in length and has an output 50 b.h.p. greater. The general layout of the aspiration system of the British Polar engine is shown in the accompanying diagram. Air is induced by a Napier TS/400 machine, having a two-entry turbine casing. All seven cylinders discharge into a single large-diameter manifold; the down pipe bifurcates on its way to the turbine, which has a maximum working temperature of 1,200 deg. F. On the same rotor shaft as the turbine is a centrifugal compressor running up to a normal speed of 10,500 r.p.m. (11,500 r.p.m. is the maximum speed). With this system of exhaust gas supply to the turbine, constant working is obtained, as compared with the impulse principle more usual with turbocharging. At full load the air discharged from the Napier compressor is at about 4.5lb. per sq. in. After this first stage of compression, air passes into the receiver, then through the Serck heat exchanger, being cooled before entry to the engine's normal opposed-piston reciprocating scavenge pump of some 87 per cent displacement efficiency. The pump has automatic valves on both inlet and discharge sides. In the standard engine this pump delivers air to the cylinders at about 2.25lb. per sq. in., but when functioning as the second stage of compression, its output is under a pressure of 7-8lb. per sq. in. at full load. The admission of air to each



Diagrammatic layout of the British Polar turbocharged engine cylinder is through automatic valves. The compression ratio has been reduced somewhat by omitting the shims generally used between the big-end bearing and the connecting-rod palm; the rod is unaltered in other respects. Despite the increase of 33 per cent in power, the maximum cylinder pressure is raised by only 10-12 per cent under pressure-charged conditions. The rotor speed of the Napier set is found to follow crankshaft speed changes very rapidly. It is reported that, raising the charge pressure has quietened the running and smoothed combustion. British Polar Engines, Ltd., will shortly commence the construction of the largest size of Polar engine, namely, the "T" series, which has an output of about 400 b.h.p. per cylinder, and this class of engine may also be developed as a turbocharged unit, with a corresponding increase in output.—*The Motor Ship, May, 1955, Vol. 36, pp. 50-51.*

Doxford Diaphragm Engine

The modified design of Doxford opposed-piston engine has been developed to obviate the possibility of corrosion of the crankshaft and other working parts within the crank-chamber when operating on boiler fuel. This design now incorporates a diaphragm plate and gland which completely isolates each cylinder from the crankcase, yet allows ease of access to the piston rod gland. Two large doors give ample access to the gland, which can be inspected without entering the crank chamber.—*The Motor Ship, October 1954; Vol. 35, pp. 293-297.*

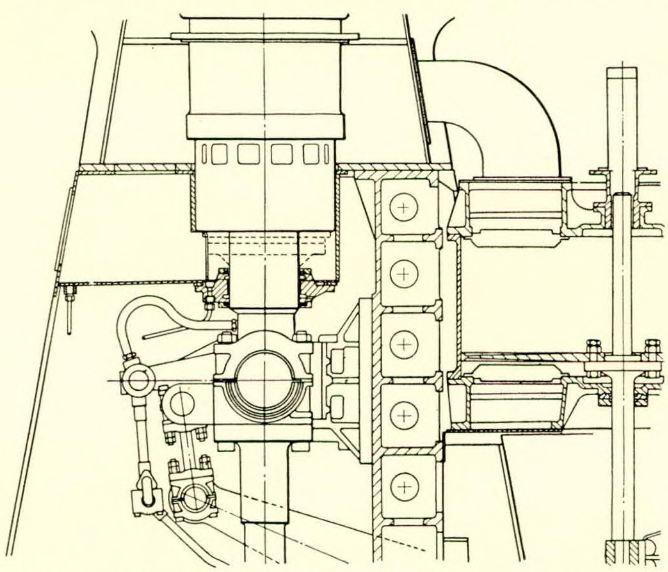


FIG. 1—*Sectional arrangement of diaphragm engine with a bore of 670 mm. and a connecting rod/crank ratio of 3·55:1. The lower piston is oil-cooled through telescopic piping*

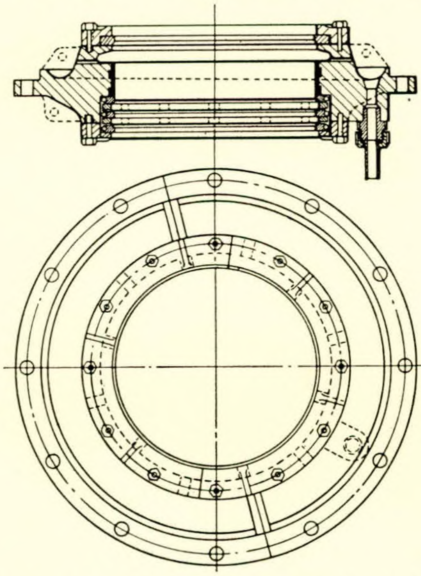


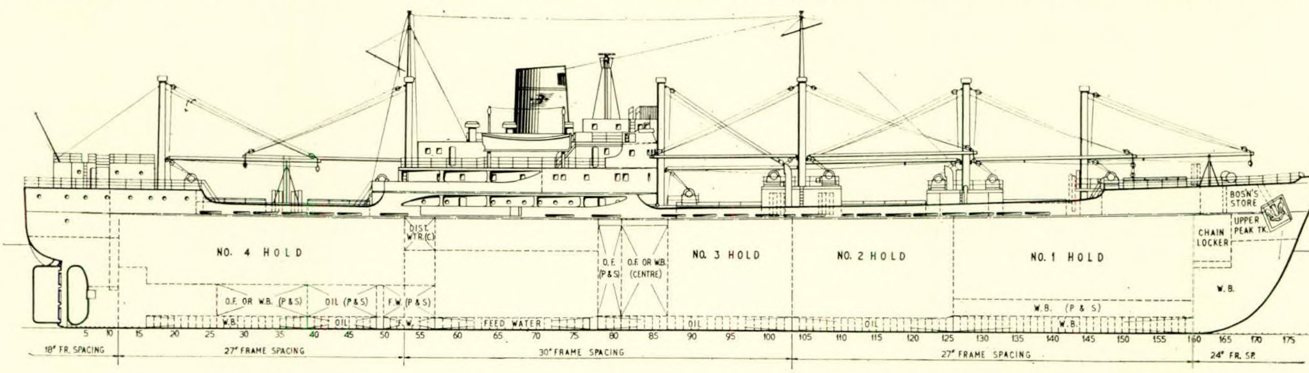
FIG. 2—*Details of gland as now fitted to the single-cylinder engine. The bore is to suit a 370 mm. diameter rod*

Swiss Gas Turbine Progress

Escher Wyss, Ltd., of Zurich, Switzerland, have established close contact in recent years with their licensees in Germany, Great Britain, the U.S.A. and Japan, and to date ten units to their design are under construction or in operation for a wide variety of applications. They use fuel ranging from oil and natural gas to pulverized coal or peat and the most prominent among them is the industrial closed-cycle set of 2,300 kW useful output which will soon be set in operation in a German factory. This unit will be fired with pulverized coal. A new closed-cycle gas turbine of 2,000 kW useful output has now completed extensive trials with excellent results on the test-bed at the Zurich works. The outstanding feature of the new prime mover is its radical simplification compared with earlier plants. In the new standard design only three radial flow compressor stages are required to compress the working medium from an intake pressure of 114lb. per sq. in. to a delivery pressure of 455lb. per sq. in. Owing to the fact that the useful output of a gas turbine unit is the surplus of the turbine output over the compressor input, the efficiency of the compressor as well as that of the turbine is of prime importance. As the efficiency of axial flow compressors was much higher than that of centrifugal compressors at the time, the first pilot plant was equipped in 1945 with three axial flow compressors having a total of 27 stages. Research conducted since then has resulted in such an improvement in centrifugal compressors that present efficiencies approach those of the axial design. The adoption of the centrifugal compressors in the new Escher Wyss closed-cycle gas turbine units has brought about a drastic reduction of compressor stages. There exists today two principal types of gas turbines, both having achieved their respective field of application. In the open combustion turbine ambient air under normal atmospheric pressure is compressed by the compressor, then blown into the combustion chamber for combustion. The flue gases after having driven the turbine are then exhausted into the open air. In the Escher Wyss closed-cycle gas turbine the working medium, in this case exclusively pure air, enters the compressor already at an elevated pressure level. It is then compressed by the compressor and consequently indirectly heated in an air heater by heat transfer through tube walls, as in a steam boiler, before it passes through the turbine. The exhaust air from the turbine is then passed through a heat exchanger and a precooler before entering the compressor again. One of the fundamental advantages of the closed-cycle gas turbine in comparison with the open combustion turbine used in its simplest and most consistent form as a jet engine is that the working medium flowing through the entire plant is and stays clean. In the case of the units described it is pure air. Another great advantage of the closed-cycle is that any fuel can be combusted that can be fired in a steam boiler. This feature leads to a much wider field of application and a much larger economic basis for the gas turbine in general. Furthermore, turbines and compressors as well as heat exchangers can be designed for highest efficiency and best heat transfer conditions without the risk of fouling or corrosion. The circuit pressure can be chosen to suit the load and therefore an extremely flat efficiency curve over a wide range of load is maintained. This is particularly attractive for peak load power plants as well as for ship propulsion, because in both cases the load is extremely variable. A flat efficiency curve as far as the lowest partial loads means great saving for peak load power plants or power units working on small grids or in remote places. This applies especially in the case of naval vessels where the turbines operate most of the time at only one-eighth or one-tenth of their output for cruising and are practically never at full load except during combat. The closed-cycle gas turbine with its flat efficiency curve means a greater range for warships for the same amount of fuel or less fuel and more tonnage for heavier armour or guns.—*Gas and Oil Power, February 1955; Vol. 50 pp. 43-44; 50.*

Newsprint Carrier

The first of two single-screw turbine-driven vessels specially designed and equipped for service in the newsprint trade has been completed by William Denny and Brothers, Ltd., for the Bowater Steamship Co., Ltd. The *Margaret Bowater* has been built to Lloyd's Register of Shipping highest class and survey and her hull structure, forward and aft, is specially strengthened for navigation in ice. The shell plating all fore and aft, between the light waterline and the load waterline, has also been increased in thickness in excess of Lloyd's requirements. The importance of stiffening the structure for ice is fully realized when it is considered that in a normal "ice weather" trip she may have to proceed through something like twenty miles of ice about 18in. thick. The vessel has been built with a raised forecastle and poop, full cruiser stern and a raked straight stem very well rounded at the forefoot. Oil fuel for the boilers is carried in deep tanks and several of the double bottom tanks, the total oil fuel capacity being sufficient for steaming thirty days at sea.



Profile of the Margaret Bowater, *built by William Denny and Bros., Ltd., for the Bowater Steamship Co., Ltd.*

The principal particulars of the *Margaret Bowater* are as follows:

Length o.a.	419ft.
Length b.p.	395ft.
Breadth moulded	58ft. 6in.
Depth moulded	35ft. 6in.
Draught (carrying newsprint) ...	22ft. 6in. (A)
Draught (carrying bulk cargo) max.	26ft. 6in. (B)
Deadweight (including hold and deck cargo), tons on A ...	6,400
Deadweight (maximum), tons on B	8,400
Speed, knots	15
Power, s.h.p.	4,700

Sparring in the cargo holds amidships consists of portable flat bars faced with ½-in. thick bonded rubber to prevent contact of the paper rolls with the steel plating. In the forward and after holds, at the ship's side, the shelving arrangements have received special attention, as the rolls of newsprint, which are carried stowed on end, vary considerably in diameter and length. The permanent shelving fitted is totally enclosed in portable light sheet metal to prevent any accumulation of paper and dunnage scraps behind the shelving constituting a fire hazard. These arrangements will also reduce to a minimum the ship preparation work necessary before proceeding to load paper.

The stowage capacity of the holds is supplemented by arrangements which permit the carriage of 500 to 700 tons of newsprint on deck, special provision being made for the safe stowage and efficient securing arrangements which are necessary for these deck cargoes. The *Margaret Bowater* is propelled by a set of single reduction geared turbines of Parsons' type, consisting of one h.p., one i.p., and one l.p. turbine, working in series, the h.p. ahead turbine having a two stage impulse wheel and the remainder of the blading being reaction type. The blading of the i.p. and l.p. turbines is of the all-reaction type. For astern working, an h.p. astern impulse wheel is incorporated in the i.p. turbine, separated from the ahead portion by a diaphragm, and working in series with an l.p. astern turbine of impulse reaction type fitted in the exhaust casing of the l.p. ahead turbine. The turbines are capable of developing 4,700 s.h.p. continuously at 90 r.p.m. The astern turbines develop 60 per cent of the normal ahead power. Steam for the main and auxiliary machinery is supplied by two oil-fired Foster Wheeler design boilers, fitted with economizers and superheaters, external type desuperheaters and mechanical sootblowers. The boilers are designed for 250 lb. per sq. in. pressure and 650 deg. F. at the superheater outlet for operating turbines only. The boiler furnaces are arranged for oil firing with a system of forced draught, the fans being supplied by James Howden and Co., Ltd. Oil fuel equipment, supplied by the Wallsend Slipway and Engineering Co., Ltd., consists of duplicate vertical steam-driven pumps, heaters, and filters.—*The Shipping World, 27th April 1955; Vol. 132, pp. 463-464; p. 467.*

Danish Oil Tank Motor Ship

Built by Messrs. Burmeister and Wain, for A/S. Dampskibsselskabet "Vendila," both of Copenhagen, the single-screw oil-tank ship *Tove Vendila* recently completed successful trials. The vessel has been built in accordance with the requirements of Lloyd's Register of Shipping for the classification ✠ 100 A.1., while the construction and equipment are in accordance with the requirements of Danish Law. The principal dimensions and other leading particulars are given in the accompanying table.

Length b.p.	526 feet
Breadth moulded	72 feet
Depth moulded to main deck	38ft. 3in.
Load draught	29ft. 6¾in.
Corresponding deadweight, tons	18,000
Capacity of cargo tanks, cu. ft.	860,000
B.H.P.	7,380
Corresponding r.p.m. ...	115
Speed on loaded trial, knots ...	15

Transverse corrugated bulkheads divide the cargo space into ten tank sections, with two main pump rooms; there is also a smaller pump room forward. Two longitudinal bulkheads further subdivide the tank range into ten centre and ten wing compartments (five port and five starboard). Longitudinal framing has been used for the centre and the wing tanks. Wherever possible, electric welding has been used in the construction of the hull. The shell plating, with the exception of the sheer and bilge strakes, which are riveted, is completely welded, as are the decks, bulkheads, frames and beams. The cargo-piping system comprises one 12-inch main pipe with 10-inch suction ports. In the cargo-pump rooms there are four vertical, duplex, compound pumps, each with a capacity of 500 tons per hour, and two duplex bilge pumps, each of which has a capacity of 40 tons. The forehold bilge pump and the fuel-oil transfer pump are located in the smaller pump room. Fuel-oil is stored in deep tanks under the forehold, in deep tanks at the fore end of the engine-room, and in the double bottom. Tanks suitable for the carriage of Diesel oil, lubricating oil and feed water are also located in the double bottom. Further capacity for feed water is provided in the upper after-peak tanks. For the carriage of fresh water, there are tanks in the poop, as well as two fresh water reservoirs amidships. The *Tove Vendila* is propelled by a single-screw, which is driven by a B. and W. single-acting, two-stroke cycle, eight-cylinder, crosshead, direct-reversing engine, with airless injection. With a cylinder diameter of 740 mm. and a stroke of 1,600 mm., the engine is capable of developing 7,380 b.h.p. at 115 r.p.m. Arrangements are made for burning Diesel oil as well as heavy oil. The electrical requirements are met by three 160-kW dynamos, each of which is driven by a B. and W., type 425-MTH-40, single-acting, four-stroke cycle, four-cylinder, trunk-piston Diesel engine. In addition, there is one steam-driven 110-kW dynamo. Steam for a variety of purposes is generated in two oil-fired watertube boilers, with indirect evaporation, and one exhaust-gas boiler. Each of the oil-fired units has an output of 22,000lb. per hour, while the exhaust-gas boiler has a heating surface of about 2,150 sq. ft. The boilers are all designed for a working pressure of 180lb. per sq. in.—*The Shipbuilder and Marine Engine-Builder, January 1955; Vol. 62, pp. 42-45.*

Tests on Turbocharged Sulzer Engine

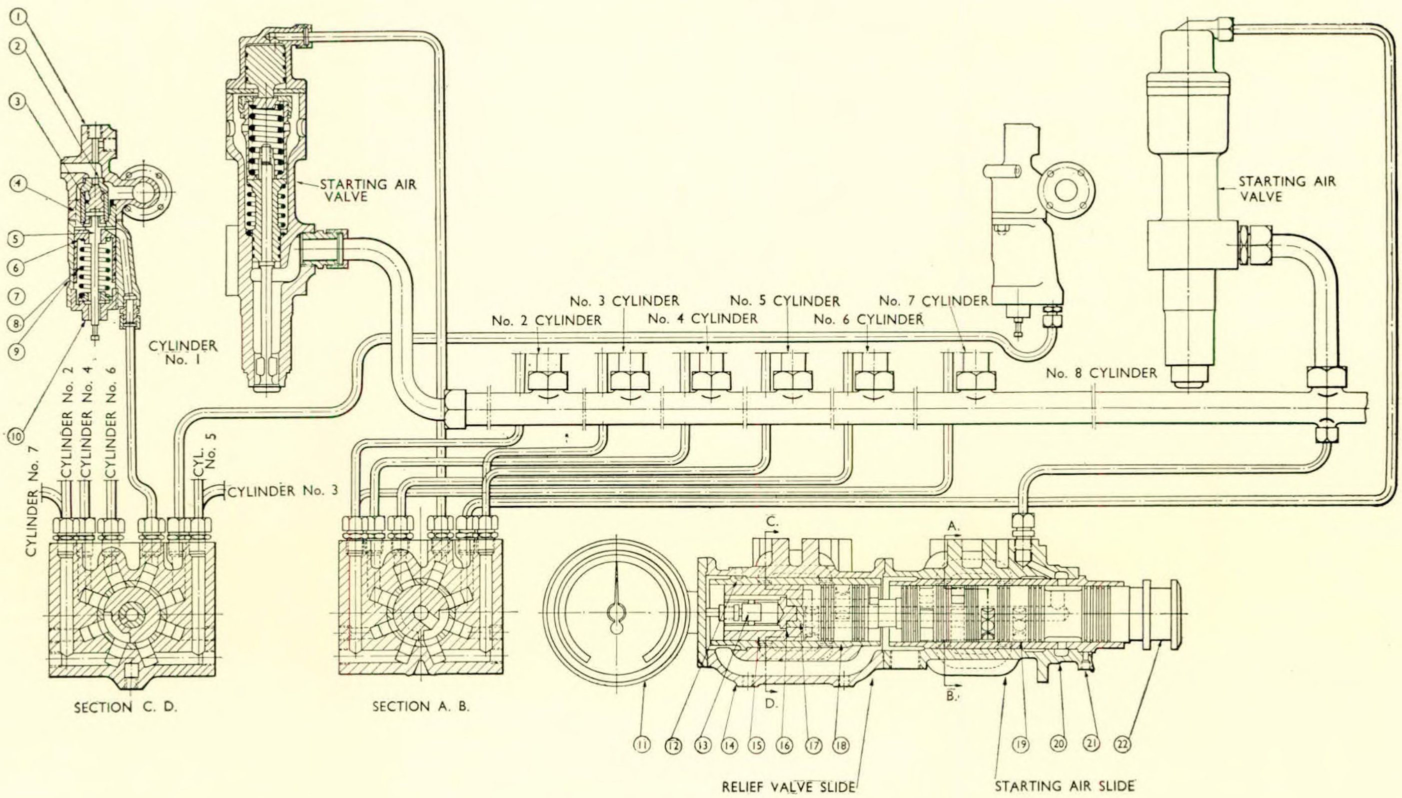
Prolonged trials have now been carried out by Sulzer Bros. at Winterthur on turbocharged two-stroke single-acting engines and, in particular, on the new R.S. model, of which type a 10-cylinder 9,000 b.h.p. unsupercharged unit has now run 3,000 hours without trouble. This design was developed specially with a view to operation on boiler oil, and with this in view the lower part of the cylinder is separated from the crankcase by a diaphragm in which is a stuffing-box, through which the piston-rod passes. A relatively short piston is employed, there is a single row of scavenging ports instead of two, as in previous Sulzer two-stroke engines, the exhaust ports are higher in the cylinder, and in the exhaust manifold is an oscillating valve which closes the exhaust ports when the piston is at top dead centre, and at the end of the scavenging period. A seven-cylinder turbocharged Sulzer engine of the new type of 8,400 b.h.p. has been completed in Japan, and trials will be carried out this month. A commencement had been made at Winterthur, on the building of a 12-cylinder 15,000 b.h.p. set, but orders have since been received for two nine-cylinder engines, and the first of these, using some of the material intended for the 12-cylinder plant, will, instead, be completed and ready for trials at the end of this year. Further, a six-cylinder engine has been ordered for installation in a ship for a Norwegian owner, also one with 12 cylinders, which will be the highest-powered Sulzer engine yet constructed. This R.S. model has a cylinder diameter of 760 mm. with a piston stroke of 1,550 mm., and runs at a speed of 115 r.p.m., with a brake mean effective pressure of 6·5 kg. per cm.^{2} (92·4lb. per sq. in.) for continuous service at sea, this being equivalent to 1,200 b.h.p. per cylinder. The normal rated power on trial is 1,300 b.h.p. (7·0 kg. per cm.^{2} b.m.e.p.), and tests on the single-cylinder experimental engine of 580 mm. bore with a b.m.e.p. of 7·5 kg. per cm.^{2} (106lb. per sq. in.) show that the engine has an overload capacity up to nearly 1,400 b.h.p. per cylinder. The trials on the single-cylinder engine indicated that the fuel consumption is well below that of the corresponding unsupercharged engine and it is anticipated from these results that the specific fuel rate of the multi-cylinder unit will be about 150 gr. per b.h.p. per hr., or some 10 gr. per b.h.p. per hr. better than the normally aspirated design. The exhaust temperature was 680 deg. F. at full load and the maximum firing pressure 850lb. per sq. in., which is slightly higher than when no supercharge is employed. It may be added that the single-cylinder experimental engine (580 mm. diameter) had a continuous run of three weeks with a b.m.e.p. of 7 kg. per cm.^{2}, or 100lb. per sq. in.—*The Motor Ship, March 1955; Vol. 35, p. 525.*

Heavy Lift Ships

In addition to the new vessels which have been equipped with heavy-lift gear since the war there have been a number of conversions carried out. An interesting example of this work is the replacement of the existing 50-tons derrick in the *Nigaristan*, 9,850 tons d.w., owned by the Strick Line, Ltd., with one of 100 tons lifting capacity. This conversion was done by the Wilton-Fijenoord N.V. last year. The original mast was removed, the decks strengthened and new bulkheads were fitted to form a winch compartment on the tweendecks. The heavy gear fitted consists of a mast about 85 feet 6 inches in height and 3 feet 6 inches diameter, extending to the double bottom tank top. The derrick is 67 feet 4 inches long, and 2 feet 2¼ inches diameter with a single sheave at the top end. The topping lift has two four-sheave blocks and is led to the 20-tons starboard winch. The cargo fall also runs through two four-sheave blocks, through the sheave at the top of the derrick and to the winch on the port side of the ship. The steel wire used is 5 inches diameter, and the winches, which are steam driven, have been placed at an angle of 5 degrees so as to ensure a good lead. To prevent the wire rope from jumping the grooves in the winch barrel, a rope guiding device has been fitted. A 25-ft. lifting beam is included in the lifting equipment and this together with the weight lifted amounts to about 110 tons. With the losses due to friction in the blocks, the minimum pull on the winch barrel will be about 20 tons. The speed of take-in is 69 feet per minute, giving a hoisting speed with 100 tons load of 6½ feet per minute. The winches were supplied by Figee, Harlem, Holland, and the masts and derricks by Wilton-Fijenoord, Schiedam. Other conversions include the fitting of 80-ton derricks in the American Naval vessels, *Short Splice* and *Col. Wm. J. O'Brien*. The bulk of the work on each ship was in the section bounded by the forward end of No. 2 hatch and the after end of No. 3 hatch. Both these hatches were lengthened, the main and tweendeck girders were strengthened, watertight bulkheads moved, the inner bottom structure in the after half of the conversion area modified and 1,200 tons of ballast were placed in Nos. 1, 2 and 3 holds and in the lower winch machinery room. The heavy-lift gear consists mainly of a single centreline unstayed mast mounted between Nos. 2 and 3 hatches, having one 80-ton derrick at the forward and one 80-ton derrick facing aft. The mast, which is 76 feet high, extends from the tank top to about 50 feet above the main deck. One section of this mast is 100 inches diameter and is 39 feet long, and extends upwards from the tank top. It is of welded steel plate of varying thickness from 1½ inches at the heel to 1⅝ inches where it passes through the main deck.—*The Shipping World, 9th March 1955; Vol. 132, pp. 295-298.*

Quick-manœuvring Engine Gear

The new deep-sea salvage tug *Marinia* is propelled by a six-cylinder British Polar direct-reversing engine developing 960 b.h.p.at 250 r.p.m. in service, the b.m.e.p. at full load being about 78 lb. per sq. in. A feature of the engine is that special quick-manœuvring gear is fitted; although not entirely a new development, it is nevertheless of interest, and has proved to be reliable in service on a number of well-known tugs. The equipment, which is shown in diagrammatic form, consists of two air distribution slide valves, one for normal reversing and the other used only when exceptionally quick manœuvring is required The starting-air distributor piston has two sets of radial ports, disposed on each side of similar ports in the slide-valve liner. When the piston is moved axially into the ahead position, the ahead ports line up with the ports in the liner, and as the distributor piston turns at crankshaft speed, the starting air escapes from the hollow centre through the liner ports to the starting-air valves in the cylinder covers. This air serves to open these valves, allowing a large volume of starting air into the cylinders as the pistons pass the top dead centres, thus providing the necessary starting impulse. It will be seen that the ports in the manœuvring piston for ahead and astern operation are arranged so that the starting valves are opened exactly at the correct time. If, when the engine is running ahead, the



Schematic arrangement of the British Polar engine quick-manœuvring gear

(1) Relief valve body; (2) Valve seat; (3) Valve; (4) Valve seat holder; (5) Valve spindle; (6) Piston; (7) Piston rings; (8) Spring; (9) Cylinder; (10) Spring support; (11) Tachometer; (12) Stop screw; (13) Catch; (14) Slide housing (aft); (15) Air slide (aft); (16) Tachometer sleeve; (17) Lost motion shaft; (18) Liner (aft); (19) Liner (forward); (20) Slide housing (forward); (21) Stop screw; (22) Air slide (forward).

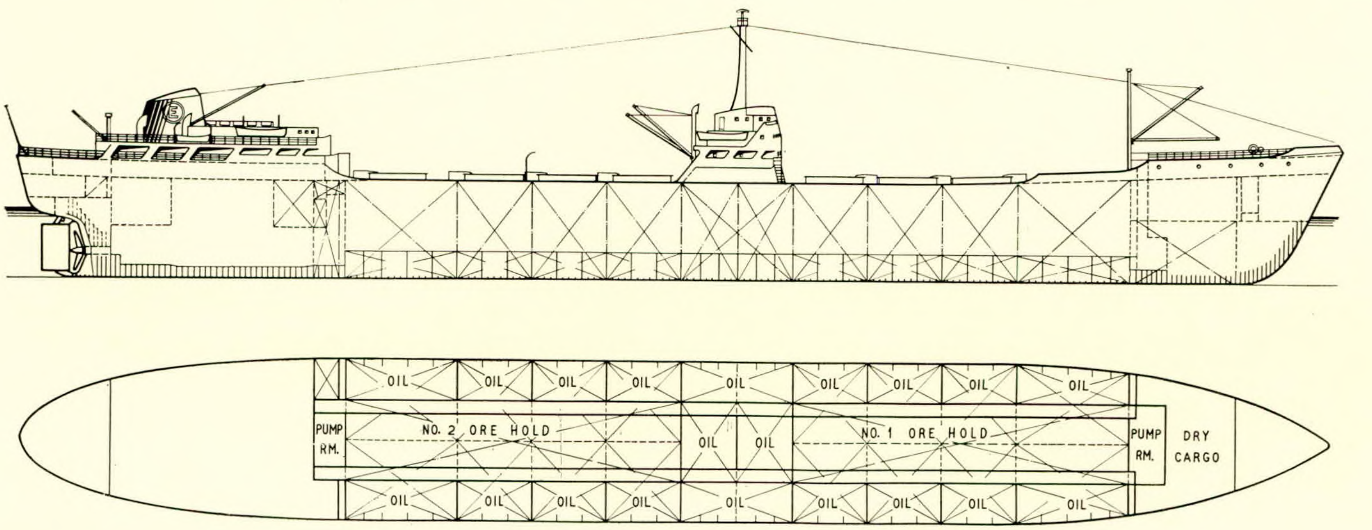
manœuvring lever is brought into the astern position, starting air is admitted to the cylinders and is compressed by the ascending pistons, thus tending to stop the engine. If this air is allowed to expand, its usefulness as a brake is largely cancelled. The purpose of the quickmanœuvring gear is to avoid this by means of relief valves on the cylinder covers, which are opened in sequence as the pistons pass through their top dead centres. The relief valves are operated by air from the relief-valve slide, which works in tandem with the main-slide valve distribution. As may be seen in the diagram, it has radial ahead and astern ports. For each cylinder there is one ahead port and one astern port drilled in an angular position, so that with the distributor piston turning at crankshaft speed, one or other of the ports is in the correct alignment when the corresponding main piston reaches top dead centre. Air thereby flows through the slide-valve liner to the relief valve on the cylinder cover, opens the valve and relieves the pressure in that particular cylinder. In this way a smooth effective brake is applied, and when manœuvring, reversal from full ahead to full astern or *vice versa* can be executed in four to five seconds. Even with full "way" on the ship, the time required is normally not more than twelve seconds. Movement of the manœuvring lever into the astern position when the engine is running ahead, or into the ahead position when the engine is running astern, admits air to the main cylinders, as explained, to provide a brake which brings the engine to rest quickly. However, this movement of the manœuvring lever also disengages the clutch through which the camshaft is driven, and thereby cuts off the supply of fuel to the engine. The arrangement is such that injection of fuel cannot take place until the engine is turning in the correct direction of rotation.—*The Motor Ship, May 1955, Vol. 36, p. 71.*

German Ore and Oil Carrier

The first German example of a type of ship that is becoming increasingly popular—the combined oil and ore carrier—has been completed by Kieler Howaldtswerke for Thomas Entz Tanker G.m.b.H., of Rendsburg. The shipowners, as their name indicates, are tanker owners. The ship is the *Bertha Entz*, and is a motorship of some 21,600 tons deadweight with a speed of 15½ knots. The *Bertha Entz* closely resembles a tanker in appearance, having her propelling machinery aft and a bridge deckhouse amidships. A difference from the normal tanker, however, is the absence of any catwalk, covered access fore and aft being provided in normal ore carrier fashion by means of alleyways in the space behind the hoppering at the upper corners of the ore holds. The oil tanks extend beneath the ore holds, and are separated by a centreline division in the double bottom. There is an exception to this in way of the bridge deckhouse, where oil is carried in all cargo spaces, there being two centre and two wing tanks separated by full depth vertical bulkheads. The principal particulars of the ship are as follows:—

Length o.a.	603 feet
Length b.p.	568 feet
Breadth moulded	73ft. 9in.
Depth	44ft. 9in.
Draught (summer)	32ft. 3in.
Capacity of ore holds ...	355,000 cu. ft.
Capacity of oil tanks ...	795,000 cu. ft.

The *Bertha Entz* has been classed with the American Bureau of Shipping and Germanischer Lloyd. Her gross tonnage is 15,004, and net tonnage 2,528. Each of the two ore holds is served by four hatches with MacGregor hatch covers, while there are eighteen wing oil tanks and two centre tanks. Two



Arrangement of the ore holds and oil tanks in the Bertha Entz

pump rooms are fitted, one at either end of the cargo spaces, access to the forward pump room being gained through the fore and aft alleyway. The main engine is an eight-cylinder double-acting two-stroke M.A.N. Diesel engine with normal aspiration, developing 8,000 b.h.p. at 118 r.p.m. The speed of the ship in the laden condition is 15·7 knots. There are two Diesel generators of 150 kW each, and one Diesel and one steam generator of 100 kW each. Steam is provided by two Scotch boilers and an exhaust gas boiler.—*The Shipping World, 20th April 1955; Vol. 132, p. 440.*

Bulk Cement Carrier for New Zealand

A twin-screw bulk cargo motorship embodying some unusual features has recently been completed by Henry Robb, Ltd., Leith. The *Golden Bay* has been specially designed for the carriage of cement in bulk and is equipped with mechanical conveying and pumping equipment for loading and discharging. Before the final design was reached, a considerable amount of research was carried out in a dummy hold made of concrete at the A.P.C.M. works at Swanscombe, Kent. The *Golden Bay* is powered by twin British Polar Diesel engines, each driving a propeller through electro-magnetic coupling/alternator units. When in harbour these alternators supply power for the cargo handling machinery. This is believed to be the first vessel to be fitted with an installation of this type in which the main engines provide the power for cargo handling as well as for ship propulsion. In addition to carrying cement, the *Golden Bay* has been designed for carrying coal; the cargo being transported to the port side of the ship by means of a screw conveyor and discharged on to a portable conveyor belt which is slung into position between the ship and the jetty through a large watertight door in the side of the ship between the upper and lower forecastle decks. The *Golden Bay* sailed for New Zealand at the beginning of February 1955, after having completed extensive loading and discharging trials. Her principal particulars are as follows:—

Length o.a.	245ft. 9¾in.
Length b.p.	230 feet
Breadth moulded	42 feet
Depth moulded to upper deck	18 feet
Draught	14 feet
Deadweight	1,400 tons
Service speed	10 knots
Grain Capacity	
Excluding hatches	54,160 cu. ft.
Including hatches	60,360 cu. ft.
Hold Capacity	
Bulk cement	1,200 tons
Slack coal	1,000 tons

The ship is powered by two British Polar type M44M Diesel engines of 630 b.h.p. each, transmitting power at 250 r.p.m. through two electro-magnetic coupling-alternator units designed and manufactured by the British Thomson-Houston Co., Ltd., Rugby. As previously stated, when the ship is at sea, these coupling/alternators operate as normal electro-magnetic couplings and transmit the power of the engines direct to the propellers. They are also used as alternators for supplying electric power for the extensive cargo handling machinery when the ship is in harbour: when used for this purpose the outer members are locked stationary by means of a steel wedge which engages in a slot cut in the rim of the outer member. Electrical connexions are then made to the three-phase windings of each outer member and the units then become conventional alternators each driven by one of the main engines. Under such conditions the engines run at 300 r.p.m. and each alternator is capable of developing 500 kW each at 440 volts, three-phase 50 cycles, this frequency being chosen to permit the use of previously developed 50-cycle motors.—*The Shipping World, 2nd March 1955; Vol. 132, pp. 270-272.*

Ocean Going Motor Tug

A new motor tug for coastal and deep sea operation has been delivered to the Overseas Towage and Salvage Co., Ltd., London, by her builders, Cook, Welton and Gemmell, Ltd., Beverley, Yorks. This vessel, the *Marinia,* is powered by a British Polar Diesel engine of 960 b.h.p. driving a single screw. The *Marinia* brings the number of tugs operated by the owners up to four. As her first tow, the *Marinia* is bringing back the *Baltic Clipper,* 1,627 tons d.w., from Casablanca with a damaged reduction gearbox. The principal particulars of the *Marinia* are as follows:—

Length	120ft. 9in.
Breadth moulded	28ft. 6in.
Depth moulded	13ft. 9in.
Gross tonnage	392 tons
Engine output	960 b.h.p.

The *Marinia* has been constructed with a single deck with enclosed forecastle built into the main part of the superstructure which extends slightly abaft amidships. She has a semi-cruiser stern and a well raked soft nosed stem, and the hull is protected by a steel rubbing strake. An electric winch of Clarke Chapman make is installed for towing purposes in a house at the after end of the forecastle with direct access from the engine room. The electric windlass is also of Clarke Chapman manufacture, the motor being on the main deck and driving a vertical shaft to the windlass. The same arrangement applies to the electric capstan fitted aft. A 20-tons Monarch type tow hook is mounted on the top of the tow line winch-house, at the aft end of the forecastle. This is in addition to the towing winch. The two towing beams are of tubular section and special attention has been given to the rounded stern rail. The propelling machinery consists of a six-cylinder British Polar type M46M Diesel engine designed for a rating of 960 b.h.p. at 250 r.p.m. in regular service. The cylinder diameter is 340 mm. and stroke 570 mm.; the brake mean effective pressure is about 78lb. per sq. in. at full load. The engine drives its own sea and fresh water circulating pumps, starting air compressor, as well as two gearwheel type lubricating oil circulating pumps. A tug must be able to manœuvre quickly and accurately, and for this reason special quick manœuvring gear has been fitted to this British Polar engine. This consists of a rotating air distributor, which allows air to be admitted to the cylinders as the pistons approach top dead centre, and opens relief valves to dissipate energy thus stored in the compressed air. In this way an effective brake is applied to stop quickly the engine and propeller, permitting reversal from full ahead to full astern and *vice versa* within five seconds. The propeller, to which the engine is direct coupled, is of Manganese Bronze type, while the spare propeller is of cast iron.—*The Shipping World, 9th March 1955; Vol. 132, pp. 301-302.*

Estuary Tankers

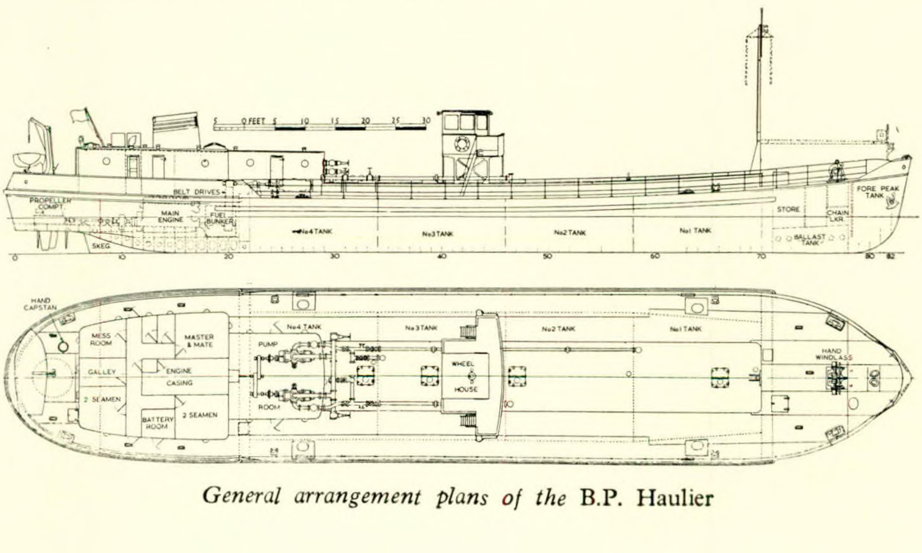
Nine small motor tankers are among those under construction for Shell-Mex and B.P., Ltd., and one of these vessels, named *B.P. Haulier*, was launched on 22nd April from the Faversham shipyard of James Pollock, Sons and Co., Ltd. She is intended for estuary service, principally in the Thames and Medway. It is understood that the vessel is the first tanker built in this country fitted with a Voith Schneider cycloidal propeller, and she is also the largest self-propelled tanker, for the transport of petroleum spirit, built in Kent. The following are the main details of the vessel:—

Length overall	148 feet
Length b.p.	140 feet
Depth	8ft. 6in.
Deadweight capacity, tons ...	300
Machinery, b.h.p.	242
Speed, r.p.m.	600

A six-cylinder Lister-Blackstone engine is installed, operating the propeller through an M.W.D. oil-operated isolating clutch. The engine can be used for driving two Stothert and Pitt 8-in. screw displacement pumps through a belt drive, the pumps being in a compartment at deck level. Two 18 h.p. Lister-engined auxiliaries are installed, one driving a generator and the other a Hamworthy compressor and a 30-ton pump. The engine controls are of the Bloctube type, led to the bridge amidships and to the pump-room aft. There are four tanks for the carriage of petroleum spirit. A 20-ton ballast tank is arranged forward, in addition to a 19-ton fore-peak tank. A 14-ton double-bottom tank is also provided for water ballast, below the propelling engine.—*The Motor Ship, May 1955; Vol. 36, p. 80.*

U.S. Navy Heavy-lift Ships

The Todd Brooklyn Shipyard has recently completed the conversion of the C-1-MAV-1 vessels, U.S.N.S. *Short Splice* and the U.S.N.S. *Col. Wm. J. O'Brien* to heavy-lift ships. The major part of the conversion work was carried out between the forward end of No. 2 hatch and the after end of No. 3 hatch. Both these hatches were lengthened; the main and second deck girders were strengthened; a new watertight bulkhead was installed and another was re-positioned and 1,200 tons of concrete and pig iron were installed in Nos. 1, 2 and 3 holds and in the lower winch machinery room. The 80-ton cargo-handling gear consists of a single unstayed mast located on the centre line between Nos. 2 and 3 hatches, with 80-ton derricks located both forward and aft of it. The 76 feet mast extends from the tank top to approximately 50 feet above the main deck. The lower section of this mast is 39 feet long, 8 feet 4 inches in diameter and is of welded steel plate varying from 1½-inch to 1⅝-inch thick. There is a fore-and-aft cantilever structure at the masthead and an athwartship's cantilever crosstree at approximately 42 feet above the main deck. In addition to the 80-ton derricks, four 10-ton booms are installed for general cargo handling. These are rigged in accordance with the Farrell improved burtoning system. There are 100 h.p. electric winches to serve the 80-ton topping and hoisting lifts. The smaller electric winches for topping, hoisting and burtoning the 10-ton booms are also used when required as vang winches for the 80-ton booms. In accordance with modern practice, the vangs are led to the lower block of the hoist and not to the derrick head, a method which can impose tremendous compressive strains on the boom, if unskilfully tended. The boom will swing through an arc of 70 deg. each side of the centreline, and when topped to 45 deg., the maximum outreach over the ship's side is 18 feet. This condition produces a maximum list of 8½ degrees which is within the specified maximum of 10 degrees.—*The Marine Engineer and Naval Architect, February 1955; Vol. 78, p. 65.*



General arrangement plans of the B.P. Haulier

Patent Specifications

B. and W. Compression-operated Fuel Pump

In Fig. 2 is illustrated a fuel pump, operated by compression pressure. Following the firing stroke, the plunger (1) together with the piston (3) moves to the upper position, and the pressure chamber of the pump is filled with fuel from the opening (12). The compression pressure moves the piston (3) downwards and as the pump outlets close, the pressure of the enclosed fuel increases in proportion to the gas pressure, until it reaches that required for opening one of the fuel valves (20). The fuel pressure continues to rise, and the pump plunger moves downwards. When the opening pressure for

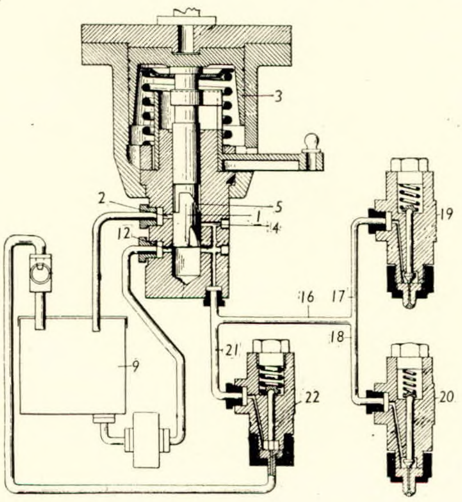


FIG. 2

the second fuel valve (19) is reached, this valve injects fuel into the same engine cylinder. During normal running, injection continues through both valves until the oblique edge (5) on the plunger opens the channel (4). When this occurs, the fuel pressure is relieved and the oil passes back to the tank (9) through the channel (2). Both fuel valves close immediately, and the motion of the fuel pump piston stops. The total quantity of fuel injected depends on the length of the fuel pump stroke, from the commencement of injection until the pressure drops. The overflow valve (22) interrupts the injection if, for any reason, the pressure in the engine cylinder becomes too high so that the pressure in the pipes (21, 16, 17, 18) is relieved, whereupon the fuel valves (19, 20) close at once. In the following table are examples showing the pressure in the cylinder at which the fuel and overflow valves may advantageously be arranged for opening and closing.

	Opening pressure kg. per sq. cm.	Closing pressure kg. per sq. cm.
Fuel valve 20 ...	... 25	20
Remaining fuel valve	... 32	25
Overflow valve ...	... 52	18

British Patent No. 725,585. Burmeister and Wain, Copenhagen. The Motor Ship, May 1955; Vol. 36, p. 81.

Marine Boiler

This invention relates to watertube marine boilers. In marine turbine power plants, efficient operation at normal cruising speed is necessary or desirable, and high efficiency calls for highly superheated steam. In some instances, however, a relatively high maximum speed is required and efficiency during the attainment of such speed is of secondary importance, since the speed is normally required only for short periods of time. A lower final steam temperature may therefore be tolerated. Such a lower temperature is, however, higher than the temperature required for astern operation. In some ships, limits of space and weight are of great importance and difficulties arise in providing compact, light-weight boilers capable of giving the required output and steam temperature characteristics. The invention includes a marine boiler comprising a combustion chamber, three parallel-connected gas passages, leading from the combustion chamber, and dampers for controlling the distribution of flue gases to the gas passages. The main superheater is confined wholly or substantially wholly to one of the gas passages, and the auxiliary superheater is confined wholly or substantially wholly to a second of the passages. During normal speed ahead a major part, if not all, of the gases from the combustion chamber is passed over the main superheater, and the remainder of the gases, if any, is passed over the auxiliary superheater, in order to obtain a relatively high degree of superheat. During full speed ahead the gas flow resistance past the boiler and superheater surfaces is reduced by opening a bypass to the superheaters and passing the gases partly over the main superheater, partly over the auxiliary superheater and partly through the bypass so as to obtain a high output at reduced steam temperature. During astern operation some of the gases are passed over the auxiliary superheater in order to obtain a relatively low steam temperature, and the remainder of the gases is passed through the bypass.—*British Patent No. 728,500 issued to Babcock and Wilcox, Ltd. Complete specification published 20th April 1955.*

Starting Gear for Doxford Engines

Reversing mechanism for a three-cylinder Doxford engine is illustrated in Fig. 3. The necessary operations are carried out by compressed air instead of by mechanical linkage from the control station. Air is supplied to the starting valve (10) from a pipe (13). On the admission of compressed air to the cylinder (15) through the inlet (16) the valve opens, to permit the starting air to enter through the aperture (11). The engine-driven shaft (20) is carried in a casing (21) within which is a piston (25). The three outlet ports (26, 27, 28) are connected respectively to each cylinder (15) of the air inlet valves (10). To start the engine, the handle (41) is moved, say, to the ahead position admitting air to the pipe (43), which is connected with the ahead end of the casing (21). The air acts on the piston (25), and passes through the groove (30) to the outlet port (26, 27 or 28). The starting valve remains open until the piston (25) is rotated to move the inlet groove (30) out of register with the outlet port, thereby shutting off air from the cylinder (15) and opening the outlet port through a relief groove to the port (29). This permits the starting air valve to close, and simultaneously the groove (30) admits air

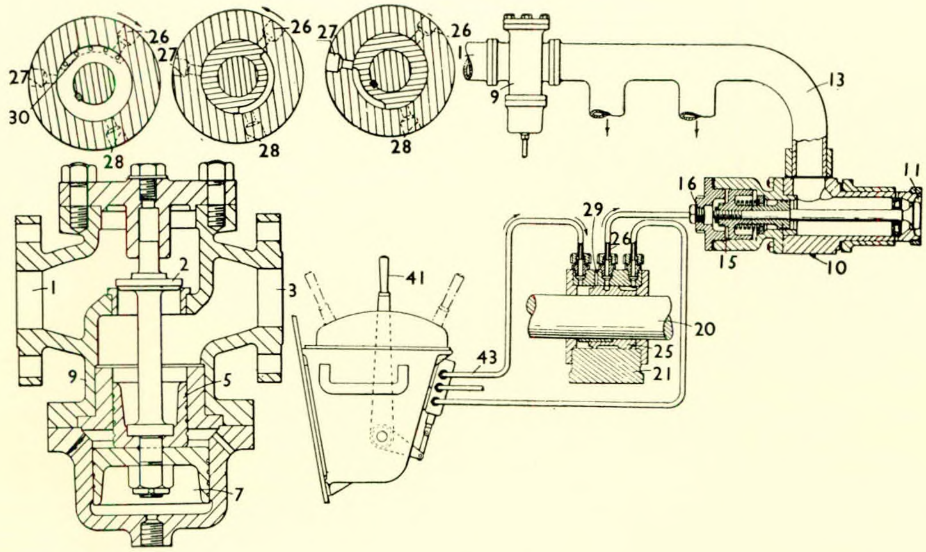
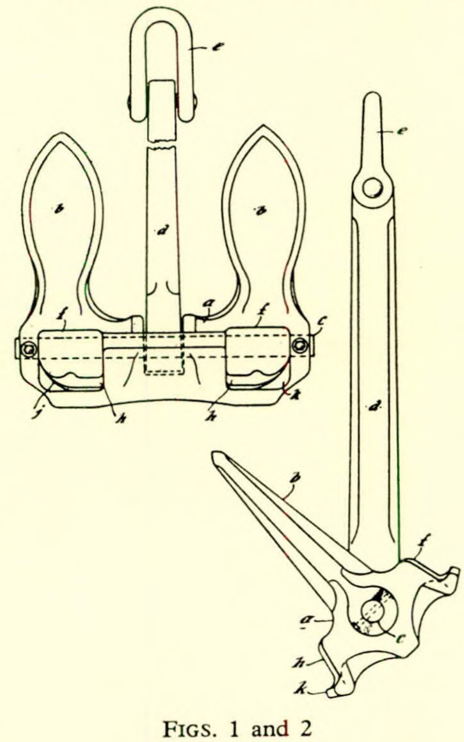


FIG. 3

to the starting valve of the following cylinder, the sequence being repeated for the three cylinders of the engine. The purpose of the valve (9) is to prevent the flow back of gases from the engine cylinders, should one of the starting valves stick open. Air from the starting container is admitted to the cylinder (7) when the handle (41) is moved either to the ahead or astern position. If when the engine is running, the pressure in the outlet (3) should exceed that in the inlet (1), the load on the piston (5) is greater than that on the valve (2), so that the latter remains closed.—*British Patent No. 728,453. Wm. Doxford and Sons, Ltd., W. H. Purdie and E. Taylor, Sunderland. The Motor Ship, June 1955; Vol. 36, p. 118.*

Stockless Anchor

This invention relates to stockless anchors. An anchor in accordance with the invention is characterized in that the outer rear corners of its transversely-spaced wings are rounded and in that the outer corners of the tripping ridges of the wings conform with the rounded portions and present contours which slide over and so avoid fouling any obstructions in the way of the anchor. Referring to the drawing, the crosshead (a) of the anchor is formed in one with a pair of flukes (b) and is attached by a main bolt (c) to the eye of a shank (d) provided with a pivoted shackle (e) as usual. On each side of the crosshead is a laterally-projecting portion (f) which is symmetrical about a plane passing through the axis of the main bolt and parallel with the central plane of the flukes. The portions (f) form a pair of transversely spaced approximately flat wings. The operative faces of the wings are outwardly inclined relatively to the central plane of the flukes (b), and each wing extends transversely from a line about level with the outer edge of the adjacent fluke over a width slightly less than that of the fluke. The edges of the wings remote from the flukes are shaped to form projecting tripping ridges (h), and the outer corner of each wing (f) is rounded, shown at (j), and the corresponding corner of its tripping ridge (h) is merged in it, shown at (k). The radius of the curve (j) may be approximately equal to the distance between the front and rear edges of the wing. The wings (f) and tripping ridges (h) thus present rounded outer corners (j, k) to obstructions in the way of the anchor during lowering and raising, and any



FIGS. 1 and 2

risk of the tripping ridges fouling such obstructions is eliminated.—*British Patent No. 731,563 issued to W. L. Byers and Co., Ltd., and F. J. Charlton. Complete specification published 8th June 1955.*

These extracts from British Patent Specifications are reproduced by permission of the Controller of H.M. Stationery Office. (Complete British Specifications can be obtained from the Patent Office, 25, Southampton Buildings, London, W.C.2. Price 3s. 0d. each, both inland and abroad.)

Marine Engineering and Shipbuilding Abstracts

Volume XVIII, No. 8, August 1955

Atomic Power for Ships

A number of American industrial groups were given contracts to study the possible application of atomic power for ship propulsion, with special reference to ships of the "Mariner" class cargo vessel. The prime mover was still to remain as a steam turbine, so that the reactor plant would replace the oil-fired steam boilers. A summary of six proposals is given in a paper read before the American Institute of Electrical Engineers by Messrs. H. L. Witzke and S. A. Haerstick, entitled "Nuclear-power Plant for Ship Propulsion—Application." Most reactors have a moderating medium, which serves to slow down the neutrons given out in the fission process to speeds most effective for producing further fission, and to which the heat made available by atomic fission is first transferred. The generated heat is transferred, by means of a coolant circuit, from the moderating medium (or from the core, if no moderator is used) to the steam-raising plant. By using the same fluid for coolant and moderator, the plant can be simplified to a certain extent. In reactor No. 1 of those quoted in the paper, moderation and cooling is by means of heavy water, natural uranium being the fuel. Reactor No. 2 uses a fuel of slightly enriched uranium, and has light water for moderation and cooling. The third reactor is moderated by heavy water, cooled by light water and uses natural uranium fuel. Graphite is used to moderate reactors Nos. 4 and 5, and liquid sodium for a coolant permits the use of reasonable temperatures without the need for pressurizing the coolant circuit; slightly enriched uranium fuel is proposed. As it is fuelled with highly enriched uranium fuel, reactor No. 6 requires no moderation, and sodium is proposed as coolant. A summary is given of the estimated cost of each proposal, and the conclusion is reached that nuclear-power plants of this type cannot at present compete on an economic basis with conventional power plants. A further serious disadvantage is the considerable weight of the shielding necessary to surround the plant, for the protection of the operating personnel. Information concerning the two nuclear-power plants being actively developed for submarine propulsion is given in a paper read before the American Institute of Electrical Engineers by Messrs. F. E. Crevet and T. Trocki, and further details concerning one of the plants (S.T.R.) is included by Messrs. L. H. Roddis and J. W. Simpson in their paper presented to the American Society of Naval Architects and Marine Engineers, in November 1954. The submarine thermal reactor (S.T.R.) is already in operation, and the submarine intermediate reactor (S.I.R.) is under construction. Both include the same basic components, and both are to be used as steam generators, the principal source of difference being in the coolant systems. The S.T.R. is cooled with pressurized water, and the S.I.R. is cooled with liquid sodium. The nuclear, thermodynamic and mechanical characteristics of the two systems are discussed, and the points of difference summarized. Also reported is a preliminary study of a unit using helium as the reactor coolant, the heat being transferred to a closed-cycle gas-turbine plant. Helium does not become radio-active, and thus the heat-transfer system requires no shielding, with consequent reduction of plant weight and improved flexibility of machinery layout. Certain assumptions have been made in this preliminary study which require experimental verification, but it has considerable interest, and much useful information is given in the comparison of the main mechanical characteristics of nuclear plant cooled by water, sodium and helium.—*The Shipbuilder and Marine Engine-Builder, January 1955; Vol. 62, p. 32.*

Effect of Torsional Overstrain upon Strength of Spring Steel

This paper gives the results of an investigation into the effect of low-temperature heat-treatment upon the static properties of a hardened and tempered spring steel which had been previously subjected to torsional overstrain. The results show that the textural stresses may be relieved by low-temperature heat treatment without reducing the beneficial body stresses produced by the torsional overstrain. At large overstrains the textural stresses are so great that plastic deformation in the reverse direction takes place on unloading. Such deformations reduce the residual body stresses, and therefore the elastic limit strength, after low-temperature heat-treatment. To prevent the plastic deformation on unloading, the specimens should be given the low-temperature heat-treatment before removal of the load. Tests on six springs showed that the results obtained from torsional specimens are directly applicable to springs. The improvement in the elastic-limit strength after low-temperature heat-treatment is considerable, about 33 per cent, while the increase in 0·1 per cent proof stress, although marked, is somewhat less and of the order of 8·5 per cent. After the low-temperature heat-treatment the elastic-limit strength range (that is, the sum of the elastic-limit strengths in the positive and reverse directions) and the 0·1 per cent proof stress range are unaffected by the degree of overstrain. That is, for the steel tested and within the range of strain examined, torsional overstrain does not work-harden the material, any increase in strength in one direction being offset by an equal reduction in strength in the reverse direction. The change of strengths in the two directions is approximately equal to the corresponding body stresses induced by the overstrain.—*Paper by J. A. Pope and J. E. Andrew, submitted for written discussion to The Institution of Mechanical Engineers, 1955.*

Experiments on Oscillating Bearings

The top-end bearings of marine steam and Diesel engines constitute a special case from the lubrication aspect owing to the reversal of relative rotation which takes place once per stroke. Little information is available on the effect of intermittent motion on lubrication conditions and an investigation has been carried out for the British Shipbuilding Research Association both to provide a basis for rational design and to assess the effect of various materials, methods of lubrication, etc., on the reliability, endurance, and efficiency of bearings of this type. Consideration of reported experience with top-end bearings on different types of engine indicated that conditions were more severe in engines such as single-acting two-stroke Diesel engines wherein no reversal of loading occurred during the cycle. It was accordingly decided that uni-directional loading would be appropriate to the present experiments. The loading applied to the top-end bearings of an actual engine will vary throughout the stroke in accordance with variations in steam or gas loads combined with inertia forces. This variation will depend on the design and condition of working of different engines and, for the purpose of a general investigation, there is no justification for assuming that a bearing would be relieved of load during the most critical part of the stroke, that is, at the instant of reversal. Accordingly it was decided that the load applied during the tests should have a constant value throughout the stroke. Although frictional losses in a top-end bearing may be of little importance in practice, in experimental work they provide the best means of indicating the efficiency of lubrication and the incidence of any critical condition. Arrangements were made therefore to obtain continuous records of friction force throughout the tests. The following conclusions may be drawn from the results of this investigation: As is so for bearings generally, the operation of oscillating bearings may be divided into two phases, representing the running-in and "no-wear" conditions. Performance during the running-in period is shown to be markedly dependent on the nature of bearing lining and lubricant. With certain combinations it was impossible to attain the no-wear condition under the severe test procedure employed in the present investigation. Contrary to the case of bearings in which the shaft revolves continuously, the presence of oil grooves in the loaded region may improve lubrication. Form and disposition of oil grooves have a marked effect on performance. Under the conditions of load and speed applied during tests, bearings made from leaded bronze and lubricated with a compounded oil wore at a lower rate than white metal. Under favourable conditions, for example, after an initial wearing in period, the rate of wear was negligible (less than 0·0001 inch in 500 hr.). The maximum safe continuous intensity of loading, using a leaded-bronze bearing lubricated with compounded oil under force-feed conditions, was 1,200lb. per sq. in. of projected area. Under conditions where lubricant was fed by gravity the intensity of loading at failure was 800lb. per sq. in. of projected area. The use of leaded-bronze or lead-base white metal appears to offer scope for improvement in top-end bearings of marine steam engines lubricated with compounded oil. There is no reason to believe that there would be any improvement in Diesel-engine practice by the use of these materials in place of tin-base white metal. That it is possible to obtain conditions at least analogous to hydrodynamic lubrication in bearings whose motion is intermittent. —*Paper by F. T. Barwell and A. A. Milne, read at a meeting of the Institution of Engineers and Shipbuilders in Scotland on 25th January 1955.*

Marconi Seagraph II Echometer for Trawling

After close consultation with trawling interests, the Marconi International Marine Communication Co., Ltd., of London, have developed a completely new technique of echometer transmission and reception. This technique is embodied in the Seagraph II. the latest Marconi marine echometer. This equipment employs a recording unit similar to that of the Seagraph, but includes phasing—an expanded-scale technique, which enables a detailed examination to be made of the sea-bed at greater depths. An entirely new transceiver has been designed, which, with specially wound projectors, is much more powerful, and has, in practice, proved capable of indicating sparse bottom fish, such as hake, at depths as great as 300 fathoms. The power of the new equipment is such that a firm sounding line, with the ground painted-in solid, is obtained, even under adverse weather conditions, when considerable aeration is present in the vicinity of the projectors. A number of Seagraph II. installations are already in service in fishing vessels, and reports indicate the high performance of this equipment. Dan-buoy anchors can be watched all the way down to the bottom. One vessel reported that, on several occasions, it has shown the point where she has entered a cold stream; while another, finding fish abundant near the surface off Spitzbergen, waited four hours, during which the Seagraph II. was used to keep watch on them until they went to the bottom in 150 fathoms. The new transceiver uses a system of pulsed continuous-wave transmission, as distinct from the more general method of discharging, into the projector winding, the energy stored in a capacitor. Thus, instead of shock excitation by a single high-current pulse of short duration, the new system applies the output of a high-power tuned oscillator to maintain the excitation current for a controlled period, during which the projector reaches maximum amplitude of oscillation. With this type of transmission, the receiver band width necessary can be substantially reduced to make a highly-efficient transmission and reception circuit, which results in a greatly improved signal-to-noise characteristic. This contributes largely to the ability of the Seagraph II. to record echoes from fish at great depth. Two pierced-hull projectors are used for transmission and reception, respectively. Outer dimensions of the new projectors are similar to those of the normal Seagraph units, so that, in cases where a two-projector Seagraph is to be replaced by a Seagraph II. installation, existing positions can be used.—*The Shipbuilder and Marine Engine-Builder, April 1955; Vol. 62, pp. 220-221.*

Non-magnetic and Non-sparking Tools

Various non-magnetic and non-sparking tools have been developed, tested, and approved by the Bureau of Ships for use in the presence of explosive gases or when handling explosive material, and for use when the build-up of magnetic field is undesirable. Tools made of beryllium copper, aluminium bronze, and aluminium with beryllium copper inserts are both non-sparking and non-magnetic. However, non-magnetic tools are to be used only to prevent the build-up of a magnetic field and the alloys used are not necessarily non-sparking. Therefore, care must be taken so that non-magnetic tools will not be used where the creation of a spark might constitute a hazard. More and more non-sparking and non-magnetic tools are being furnished aboard ship to ensure the highest degree of safety. In addition to various non-sparking hand tools, such as adjustable end wrenches, pipe wrenches, wrench sockets, and pliers, cutting tools are also available for use in hazardous areas. While these tools cost more than the best steel tools, they do not have their strength or hardness and should not be used except where the creation of a spark or the presence of a magnetic field is hazardous. Cutting tools of beryllium copper alloy, for instance, can be heat treated to a Rockwell C hardness of only 38 to 45 in comparison with steel tools which can be heat treated to a Rockwell C hardness of 60 to 65. Aluminium bronze alloy cannot be hardened to more than 20 to 30 on the C scale. The relative bending moment and torsional strength of the alloys used in safety tools is approximately 60 per cent of forged steel tools.—*Bureau of Ships Journal, February 1955; Vol. 3, p. 29.*

Progress in Boiler Water Testing

There never has been a time when United States fleet water tests were completely adequate. Complexity of equipment and techniques, instability of reagents from the standpoint of shipboard supply and obscurity of end point signals are common shortcomings. These deficiencies are most pronounced in the present tests for hardness and dissolved oxygen. The soap hardness test requires a five-minute waiting period which is very trying for the patience of an operator who consistently has recorded zero many times before. The test is liable to false end points in the presence of magnesium salts and reads traces of heavy metals as hardness. Search for a better hardness test was started in 1949. During the past three years very promising progress has been made in adapting the newly developed Schwarzenbach complexometric titration to Navy use. Contrary to its title, this is a very simple test. A liquid indicator turns the sample red if hardness is present and titration to a blue colour with a second chemical measures the hardness. The test is sensitive enough to detect 1 ppm of calcium or magnesium hardness. The colour change is sharp and distinct. Recognized interferences such as manganese, copper, and zinc generally are not pertinent in Navy boiler water. The only real shortcoming is instability of reagents. Effort has been concentrated on this problem and, to date, stable forms have been obtained for two of the three chemicals required. Informal trials of the method have been made on several ships and formal trials recently were recommended to the Bureau. The existing shipboard dissolved oxygen test has many disadvantages. Two of the four reagents are extremely corrosive and one causes continuous trouble with frozen stopcocks. It will not detect oxygen below 0·02 ppm, though a modern deaerator is guaranteed to supply water of 0·014 ppm or less. A long search for better oxygen methods finally was rewarded in 1952 with discovery of a promising indigo carmine colourimetric procedure. Basically this consists of adding a mixed reagent to the water sample, shaking it and comparing the colour with a series of standards. The water sampling and mixing procedure is simple but unique and employs the sampling bottle sketched in Fig. 24 (not reproduced). The mixed reagent of indigo carmine, glucose, caustic soda, glycerine, and water is pipetted into the small vial inside the sample tube. The vial is sealed with a glass bead and the sample water is flowed into the bottom of the tube and overflowed past the vial of reagent until the tube is thoroughly flushed. The stopper is then wetted and inserted, and the tube is inverted several times to mix the reagents and develop the colour. Numerous laboratory and service trials of this method have shown it to be as sensitive and almost as reproducible as the Engineering Experiment Station's referee method for dissolved oxygen. Instability of the mixed reagent again is the major problem. Attempts are being made to provide the necessary chemicals as two or three stable components with simple directions for mixing prior to use. Perfection of this method would make it easy to test for dissolved oxygen as to test for alkalinity. A Hartmann-Braun oxygen meter (German) is also being tested. Needs for other special tests have grown out of the special treatments discussed earlier. For example, the cracking of stainless steel boiler parts resulted in the need for a microchloride test. (0·5 ppm and lower). The Station has adapted the standard mercuric nitrate titration to this need as an interim measure and now is working on more sensitive mercurimetric spot tests and colourimetric tests. Adoption of hydrazine as a water testing chemical also will involve some puzzling test problems. The small concentrations involved (approximately 0·05 ppm) are difficult to determine by any means. In 1953 a simple shell and coil heat exchanger finally was adopted as standard equipment for cooling boiler water aboard ship. The cooler can also be used for collecting dissolved oxygen test samples if the sea water temperature is below 70 deg. F.—*F. E. Clarke, Journal of the American Society of Naval Engineers, February 1955; Vol. 67, pp. 11-43.*

German Cargo Liner with A.C. Winches

The German motor cargo liner *Cap Blanco*, which has been delivered by Lubecker Flenderwerke to the Hamburg-South America Line (Hamburg-Sudamerikanische Dampfschiffahrts Gesellschaft), is the first of a group of eight similar vessels intended for the South American service of the company. Four will be owned by the firm itself, and the other four by Rudolf A. Oetker. With a loaded speed of 17½ knots, they will be the fastest German ships on this run. The principal particulars of the *Cap Blanco* and her sisters are given below:

Length	493ft. 6in.
Breadth	60ft. 2in.
Loaded draught	24ft. 9in.
Gross tonnage	5,899
Deadweight tonnage	8,300

There are five cargo holds, giving in all 570,000 cu. ft. of space for general cargo, and refrigerated space amounting to 100,000 cu. ft. A particularly interesting feature of the design is the electrical installation, in which alternating current is used throughout, even the cargo winches having A.C. motors. The installation has been carried out by Siemens-Schuckert. The generators are of the firm's self-excited constant-voltage type, and are arranged to give almost instantaneous compensation for the line voltage drops resulting from the starting of squirrel-cage motors, which are used for all auxiliary machinery in the engine room. The principal deterrent to the use of alternating current in cargo ships is the difficulty of providing an economic drive system for the cargo winches affording accurate speed control. Where A.C. installations are employed in such ships, the practice normally followed is to fit rotary convertors near the winches so that the actual drive can be through a D.C. motor in the ordinary way. In the *Cap Blanco*, however, three-speed squirrel-cage motors are employed, in which three sets of stator windings allow four, eight or thirty-two poles to be used. Thus, in addition to the normal speed, speeds of one-quarter and one-eighth of this speed only are available. The main engine in the *Cap Blanco* is an M.A.N. single-acting Diesel engine developing 7,200 b.h.p. to give the loaded speed of 17.5 knots. A feature of the engine room is the use of fluorescent lighting in it.—*The Shipping World, 23rd March 1955; Vol. 132, pp. 350-351.*

Shipboard Radio Sextant

The development of a shipboard radio sextant which automatically tracks the sun through rain, snow, and clouds was disclosed recently. Built for the Bureau of Ships, this new instrument has three main advantages over the hand-held optical sextant used at present. It can furnish information continuously from sunrise to sunset. It can operate under all weather conditions, including blinding snowstorms, heavy rains, and dense overcast. Its calculations can be recorded automatically for future reference. The radio sextant is equipped with a 30-inch diameter parabolic reflector, which has all high-frequency circuitry housed in the rear. Total weight of the sextant is about 400lb.—the topside installation weighing about 275lb. and the stable table on which it rests about 125lb. A desk-size box weighing 75lb. or less can serve as the control console for the equipment. Using a sensitive radio receiver, the radio sextant detects microwaves emitted from the sun.

The receiver operates a "servo" system which moves the parabolic reflector to track the sun and measure its position. From this information, a navigator can readily pinpoint his ship's precise latitude and longitude. Cited by a well-known navigator as one of the greatest advances in navigation in the last fifty years, the radio sextant is made possible by the fact that the sun radiates in the microwave spectrum. This thermal radiation, in the form of noise, is of very low level and was not detectable until the impact of radar led to the development of receiver and antenna techniques for the low centimeter wavelength region. Using these techniques, engineers developed an antenna and receiver at a wavelength of 1·25 cm. which was capable of easily detecting radiation from the sun in a 15 megacycle band-width. As a result, the feasibility of an automatic solar radio sextant was established, and the decision to attempt such a development was made. A wavelength of 1.9 cm. was chosen for the initial development model of the sextant because at this wavelength atmospheric absorption during all weather conditions appeared to be sufficiently low. In addition, an experimental sextant operating at a wavelength of 0·87 cm. is being tested daily to provide accuracy information as well as atmospheric absorption and refraction magnitudes. These tests indicate that an 0·87 cm. radio sextant can be made to present the altitude and bearing angles of the sun with a probable error of one minute of arc except during severe rainstorms. Use of this wavelength permits the design of a radio sextant with an antenna only one foot in diameter, making it attractive for many applications.—*Bureau of Ships Journal, February 1955; Vol. 3, pp. 31-32.*

Engines-aft Cargo Liners

Two of the large cargo liners which the East Asiatic Co., Ltd., has on order will have their machinery arranged aft. The first of these 10,000 tons d.w. vessels will be delivered by Burmeister and Wain in July 1956, and the second will come from the Nakskov Skibsvaerft in April 1957. The ships will have four holds closed by five hatches with upper and lower 'tween decks in each. There will be cargo deep tanks beneath the bridge, which will be situated between No's. 2 and 3 hatches. Hallen bipod masts will be fitted forward and aft of the bridge, the latter one carrying a 60-ton derrick. In each case an eight-cylinder turbocharged Burmeister and Wain 74-VTBF-160 engine developing 10,000 b.h.p. at 115 r.p.m. for a service speed of 17½ knots is fitted. Auxiliary power will be provided by four 120 kW generators, driven by three-cylinder 325-MTH-40 engines, these being arranged on flats in the wings of the engine room. Accommodation will be provided in the two deck houses, the bridge structure carrying two two-berth passenger cabins, with dining room, smoking room and the navigating officers' cabins.—*The Marine and Engineer and Naval Architect, March 1955; Vol. 78, p. 91.*

Turkish-built Diesel Electric Ferries

The Istinye Shipyard at Istanbul are constructing a pair of twin-screw Diesel-electric ferries for local service. The two craft are being built under the survey of the American Bureau of Shipping. They have principal particulars as follows:

Length overall	106ft. 3in.
Length b.p.	98ft. 8in.
Moulded breadth	21ft. 7¼in.
Moulded depth	8ft. 2⅝in.
Draught	5ft. 3in.
Gross tonnage	131
Passenger seats	250
Speed, knots	11

The owners are the Denizcilik Bankasi T.A.O. Sehir Hatlari Isletmesi who operate the principal short and long distance ferry services around the Turkish city. Twin-screw Diesel-electric machinery developing a total of 320 s.h.p. is installed. There are two propulsion generator sets, each consisting of an MWM type-418A eight-cylinder four-stroke Diesel engine running at 1,350 r.p.m. The bore is 140 mm., the stroke is 180 mm., and the output is 220 b.h.p. at 1,350 r.p.m. The engines are self-contained and have integral starting air compressors, fresh and salt water circulating pumps, a lubricating pump and a bilge pump. Each generator is mounted on a common bedplate with its Siemens-Schuckert 131 kW propulsion generator. The screws are each driven by a 160 b.h.p. 1,250/300 r.p.m. geared Siemens motor. Control is carried out on the Ward-Leonard system and the 10 kW propulsion exciter is driven by Vee-belts from either of the main generators, as can be seen from the arrangement drawing. The machinery can be controlled from either the engine room or the bridge, and change-over switches are provided to enable the alternative positions to take over. Arrangements have also been made for the vessel to run with power supplied from only one generating set. When the vessel is lying alongside a pier, the Diesel-generating sets can be set to idle at 800 r.p.m. As soon as movement order is received, the speed is automatically increased to 1,350 r.p.m.—*The Marine and Naval Architect, March 1955; Vol. 78, pp. 90-91.*

Ship's Stabilizers

In this article the author undertakes a critical review of ship's stabilizer types in use and proposed. Reference is made to the possible use of the Voith-Schneider propeller as stabilizer. This is considered to be superior to the Denny-Brown stabilizer insofar as its fin size is smaller, while at the same time its use dispenses with the need of a propeller. With the Voith-Schneider type the efficiency of the "activated" fins is considerably increased, since the cosine-component of the thrust exerted by the Voith-Schneider propeller is utilized as ship's propulsion means, and the sine-component acts as stabilizer. A typical proposed arrangement is shown in Fig. 4, which is based on German Patent No. 690,383.—*C. Von den Steinen, Schiff und Hafen, March 1955; Vol. 7, pp. 132-137.*

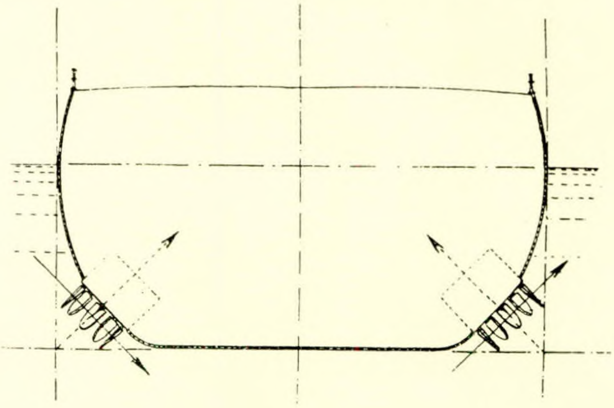


FIG. 4

First Ferry Ship of its Type for Preston-N. Ireland Service

The Atlantic Steam Navigation Co., Ltd., London, have placed an order on behalf of their Continental Line, with Wm. Denny and Bros., Ltd., Dumbarton, for the construction of a new vehicle and passenger-carrying ferry. This vessel, of about 3,000 gross tons, will be placed in the Preston-Northern Ireland service in the autumn of 1956 to supplement the daily transport ferry service maintained by the company. The first of its type to be built in this country, it will carry more than eighty laden road haulage vehicles, trailers and containers which will be driven or hauled on and off the ship. Special accommodation will be provided for lorry drivers and good-class facilities will be provided for first- and tourist-class passengers. The length when loaded will be 325ft., the breadth 55ft. and the load draught 12ft. 6in., the depth to the vehicle deck being 16ft. and to the upper deck 32ft. A twin-screw vessel, it will be propelled by two Sulzer engines each of about 3,000 b.h.p. to give a service speed of 14 knots. It is likely that stabilizers will be fitted.—*The Motor Ship, April 1955; Vol. 36, p. 31.*

American Built Largest Deadweight Carriers

The last two years have seen some striking advances in the size of bulk cargo-carrying vessels. Four tankers of over 45,000 d.w. tons have now been completed and six even larger ones are on order at Barrow and in France. These ships are very large by any standards, all having a length between perpendicular of over 750 feet and are numbered among the largest fifteen merchant ships of any type afloat. If carrying capacity is to be the yardstick, then the group of three vessels operated by National Bulk Carriers for a specific service across the Caribbean Sea are easily the greatest. These vessels can only carry their deadweight of 59,200 tons when relatively heavy cargoes such as iron ore at 16½ cu. ft. per ton are concerned. The three ships are the *Ore Chief*, *Ore Transport* and *Ore Titan*, the first two of which are already in service, transporting Venezuelan iron ore to steel plants on the Gulf of Mexico and United States eastern seaboard. These ships have been built to the American Bureau of Shipping Rules and have the following principal particulars:—

Length b.p.	756 feet
Moulded breadth	116 feet
Moulded depth	56 feet
Deadweight capacity, (ore), tons	59,200
Corresponding mean draught	38ft. 6in.
Gross tonnage	20,910
Loaded displacement, tons ...	79,700
Block coefficient	0.826
Midship coefficient	0.989
Prismatic coefficient	0 835
Length/depth	13.5
Breadth/draught	3.01
Breadth/depth	2.07
Trial power, s.h.p.	17,000
Trial speed, loaded, knots ...	14
Fuel consumption for all purposes (lb. per s.h.p.-hr.) ...	0·55

In order to keep the draught at a moderate figure, the beam is very great indeed, so much so that these vessels have to make their delivery voyage from Japan to the Gulf round the Cape of Good Hope, as they are unable to pass through the locks of the Panama Canal. The ballasting arrangements of such large vessels are of the greatest importance. There are six large double-bottom tanks, a large forward deep tank and a cross ballast tank beneath the bridge structure. These are always kept void, even on ballast voyages. The wing tanks, of which there are fourteen pairs, are divided in two by a watertight flat at about 47 feet above the keel line. On ballast voyages all the upper wing tanks are filled and also Nos. 4, 8, 11 and 13 lower ballast tanks, making a total of 18,220 tons of ballast. Under these conditions, the mean draught is 20 feet 4¾ inches, with a trim aft of 15 feet 2½ inches.—*The Marine Engineer and Naval Architect, March 1955; Vol. 78, pp. 98-100.*

Vibration in Marine Geared-shaft Systems

The paper presents a calculation method based on an electrical analogy in which attention is focused on the relation between applied harmonic forces and the velocity of harmonic vibration produced. The mathematical properties of sine and cosine functions, and their relation to simple harmonic motions, enable damping terms to be handled as well as those relating to mass and flexibility. The steady-state behaviour of a system under any assumed harmonic excitation may then be calculated for any desired frequency, both in phase and in magnitude. This approach has led to a very simple graphical method for determining the natural frequencies in the type of system usually encountered in marine practice. Two types of excitation are considered, of which only the first has previously received much attention in published work. This first type is force excitation, as, for example, the periodic forces arising from propeller rotation which remain constant irrespective of the amount of vibration which they may produce. A second and very important type is displacement excitation; this may arise, for example, from eccentricity of a pinion, which brings about harmonic variation of velocity ratio in the gear at the frequency of pinion rotation. The rotating masses provide whatever torque is necessary to maintain the difference between pinion and wheel velocities prescribed by the error. The method is equally applicable to both types of excitation, and the paper includes worked examples of each type with sufficient detail to enable similar calculations to be made in a design office, and gives damping factors based on measurement at sea. Propeller damping is reasonably amenable to theoretical treatment, but damping in other parts of the system is less clearly understood. Both bearing friction and the motion of turbine blades under steam flow are found to be quite inadequate to account for the observed behaviour, and the author suggests that the principal seat of such damping is the oil film in gear bearings. When gears are subjected to alternating torque the journals must oscillate laterally in the bearings, and this gives rise to axial and peripheral flow of the oil with consequent dissipation of energy. Damping is introduced into the calculations in the form of dynamic magnification factor, Q, and it has been found that Q-values of 10 for a propeller and 20 for all other masses in a geared-turbine system fit measured results very well, even over a wide range of vibration frequencies (about 4-100 cycles per sec.).—*Paper by H. G. Yates, read at a General Meeting of the Institution of Mechanical Engineers on 1st April 1955.*

Multi-purpose Dredger for Italy

A recent delivery from I.H.C. Holland, The Hague, a group of shipbuilding, engineering and dredging companies, was the self-propelled cutter, suction hopper dredger *Silm*, built to the order of the Societa Italiana per Lavori Maritime. The hull was constructed by Verschure and Company, Amsterdam. The principal particulars are:—

Length b.p.	213ft. 1½in.
Breadth	39ft. 4in.
Depth	17ft. 8½in.
Hopper capacity	35,710 cu. ft.
Gross tonnage	1,223

A typical example of a multi-purpose dredger, the *Silm* is equipped with a flexible suction pipe for stationary dredging, a cutter installation for working in hard soil and a draghead for trailing. The hopper contents can be dumped through bottom doors, or alternatively, can be discharged to a shore delivery pipe by the dredge pump. Propulsion is effected by two sets of triple-expansion steam engines. The engines can be disconnected from the propeller shafting when the vessel is at rest, and the starboard engine used to drive the dredge pump, while the port engine can be used to drive a Ward Leonard generator. This generator is driven by a V-belt from the port engine shaft and supplies current for the cutter motor mounted on the suction ladder. In order to step up the generator speed from the low revolutions of the propelling engine, the generator is filled with a pulley of suitable size. A 25 kW compound generator is directly coupled to the Ward Leonard generator and supplies current for the excitation of the main generator and cutter motor field and also for ventilation and lighting purposes. A steam driven auxiliary generator serves as standby for the above-mentioned 25 kW generator and for feeding the ship's main circuit when the cutter installation is not in use. There is also a Diesel-driven generator for harbour use. All engine room auxiliaries are steam driven with the exception of a lighting-up unit for the oil-fired boilers and the pumps for the fresh and salt water pressurized systems. As previously mentioned, the dredge pump is directly driven by the starboard main engine. Suction of the pump is either from the suction tube in the ladder or from the special self-discharging ducts in the hopper. The pump discharges into the hopper through an open chute with diffusers over the hopper. This system of loading chutes has been adopted by I.H.C. Holland for dredging fine sand after extensive research in the Mineral Technological Institute, because of the excellent performance in providing a well-filled hopper with low overflow losses. The suction ladder and suction tube are of unconventional design in that it is quite easy to switch from dredging with the cutter installation to stationary dredging with the flexible suction tube. It is only necessary to remove part of the suction tube in the ladder and to fix the bottom part of the flexible pipe. Then the whole suction pipe can be lowered while the ladder rests in its highest position, supported by brackets. Thus equipped, the vessel can dredge in a considerable swell, while the suction mouth is kept in contact with the bottom independent of the movement of the ship in the waves, for the hoisting wires of the pipe are led over sheaves, mounted on pneumatically loaded hydraulic cylinders, which pay out or take in the slack in the wires automatically. The cutter can be replaced by a drag head for trailing, in which case the port engine is available for propulsion. Deck machinery consists of two forward and two after winches, the bow and ladder hoist winch and a windlass. The twin rudders are operated by hydraulic rams situated in compartments on either side of the ladder well.—*Shipbuilding and Shipping Record, 10th February 1955; Vol. 85, pp. 173-174.*

Journal Bearing Performance Under Alternating Loads

Journal bearings operating under non-steady conditions form a most important class of engine bearings; as yet, however, their performance has received little theoretical or experimental attention. On the theoretical side, the main drawback appears to lie in the mathematical complexity of the equations governing the bearing performance; and on the experimental side, the development of a satisfactory apparatus for testing this class of bearing presents certain practical difficulties. Design procedure for such bearings, based on their performance characteristics under steady conditions and corrected by the application of empirically determined factors, is obviously far from desirable. In approaching the general case of a variable load applied to a journal bearing, alternating and fluctuating loads have first been considered. In this respect, the most significant result obtained is the deterioration, in general, in the load-carrying capacity of the bearing as the ratio of the rate of load application to that of journal rotation approached the critical value of 0·5. In the present paper, the authors give a brief account of the results of some experimental investigations on the simple case of a complete journal bearing operating under vertical sinusoidally alternating and fluctuating loads. In the paper is described a testing machine which has been designed and developed to investigate the more elaborate loading conditions. Observations include the recording of the journal centre path and the load wave applied, also the measurement of the frictional torque on the journal, speeds of journal and loading shafts and the oil temperature, all under the same impressed conditions. Contrary to theoretical expectations, no film breakdown was experienced within the critical range, for sinusoidally alternating loads. This may be partly attributed to some micro-roughness effect in the bearing and to the damping imposed on the journal motion. The critical increase of eccentricity in the journal was found to be accompanied by some related increase in friction values. For sinusoidally alternating and fluctuating loads, bearing design based on a steady load equal to the maximum value in the cycle appears to be of adequate safety, provided that the ratio of the rate of load application to that of journal rotation is not below about 0.7. In fact, the improvement in the load carrying capacity of the bearing with higher values of this ratio should be taken into consideration to secure higher efficiency in design.—*Paper by G. S. A. Shawki and P. Freeman, read at a General Meeting of the Institution of Mechanical Engineers, 1st April 1955.*

Testing Hydraulic Fluids for Safety

For over fifty years, petroleum oil has been used as the power transmission medium in shipboard hydraulic power systems as well as in hydraulically driven industrial machinery. This fluid has been used because it is practically incompressible, it has good resistance to freezing, and, since all components of a shipboard hydraulic power transmission system are lubricated by the fluid as it passes through them, because of its excellent lubricating properties. Several events in the last year and a half have speeded up the search for a fluid which has all the required favourable properties which petroleum oil has, but which is also fire and explosion resistant. The main problem has been to find a satisfactory fire and explosion resistant fluid that also will lubricate the pumps, motors, valves, and rams which are components of the power transmission system. Research for fire and explosion proof fluids for hydraulic systems was begun in July 1943 at the U.S. Naval Research Laboratory in connexion with aeronautical requirements. It centred around water-based fluids. Since then, several non-aqueous synthetic fluids have been developed by industry. During the last year and a half, all types of commercially available fluids have been undergoing a series of tests at the Engineering Experiment Station and at the Material Laboratory, New York Naval Shipyard, to determine their suitability for Bureau of Ships applications. Supporting work is being done at the Naval Research Laboratory, the Industrial Test Laboratory, and in the naval shipyards at Philadelphia, Portsmouth, and Mare Island. The satisfactory shipboard hydraulic fluid, when it is developed, should be: fire and explosion resistant; suitably lubricative; non-corrosive; non-toxic; non-foaming; thermally and chemically stable; compatible with other materials used in the system; reasonably inexpensive. A fluid submitted for test is first checked at the Engineering Experiment Station for explosion resistance, corrosion inhibiting properties, and lubricity. The tests are being conducted as follows—1. A standard CFR Method 5 rating engine is modified in such a way that (*a*) the compression ratio is increased by attaching a conical section to the end of the precombustion chamber plug; (*b*) an intermittent fuel injection device is installed so as to give an injection on three successive cycles followed by twenty-seven cycles with no injection; the evaluations may be conducted at temperatures other than the 212 deg. F. water jacket temperature; (*c*) pressures may be determined in a pressure-sensitive pick-up and oscilloscope. If the compression ratio is increased and the fluid is injected into the combustion chamber, a minimum compression ratio required for igniting the developmental fluids can be determined. This method appears to give a satisfactory evaluation of the compression ignition tendencies of the fluids. 2. Correlative work with a compression-ignition apparatus is being utilized to determine the fire resistance tendencies of hydraulic fluids when subjected to high-pressure air released at 4,000 lb. per sq. in. The test chamber is wetted with a test fluid, and, by means of a fast-opening valve, air at 4,000 lb. per sq. in. is released into the chamber. An explosion is recorded if it occurs. Many experimental fluids have failed this test. 3. A standard turbine oil rust test, ASTM Method D665-52T modified to use distilled water, is employed for determining the properties inhibiting against water corrosion. 4. Bearing and pump lubrication properties of the developmental fluids are being investigated in both bench bearing testers and in full-scale pump tests. After a fluid has been evaluated at the Engineering Experiment Station, it gets extensive pump tests and further bearing tests at the Material Laboratory of the New York Naval Shipyard.—*N. E. Wenger and D. P. Schnacke, Bureau of Ships Journal, February 1955; Vol. 3, pp. 17-19.*

Rubber in Engineering

Rubber, of course, has weaknesses which limit its usefulness. Foremost among these is its low tensile strength; because of its low strength rubber is rarely used in applications where it must continually bear a load in tension or in torsion. Rubber is usually stressed in compression. In some applications, such as machinery mountings, it is stressed in shear, but the load is usually limited to values of the order of 25 to 50 lb. per sq. in. Since rubber is in its chemical nature an organic material, it is subject to deterioration by heat, oxygen, ozone, and sunlight. Such deterioration usually results in hardening, decrease in elasticity and surface cracking. Surface cracking is particularly detrimental because it can lead to sudden rupture when the rubber item is stretched. Another consequence of the chemical nature of rubber is its swelling by certain organic fluids. This swelling causes a general loss of tensile properties as well as a change in shape of the rubber item. The mechanical properties of rubber are affected much more by relatively moderate changes in temperature than are the corresponding properties of metal. In general, the softness of rubber increases and its hysteresis decreases as its temperature rises above 0 degrees F. The rubber becomes hard and brittle at some temperature below 0 degrees F.; the specific temperature at which brittleness occurs depends on the type of rubber. In contrast to the stability of a tempered steel spring subjected to continuous stress, continuous stress on rubber, such as occurs in a rubber mounting supporting machinery, results in a gradual creep or drift of the loaded rubber item. A mounting which has drifted is less efficient for insulation of vibration and protection against shock. If the rubber item is held for a period of time at constant deflexion instead of constant stress, as in the case of most rubber gaskets, the stress exerted by the rubber against the restraining surfaces gradually decreases. When the stress exerted by a gasket falls below the pressure of the fluid being sealed, leakage is liable to occur. After suffering either drift or stress relaxation while deflected, the rubber item does not fully recover its original shape when released, but exhibits a permanent set. Both drift under constant stress and stress relaxation at constant deflection are accelerated by increase in temperature. Rubber is not fire-proof, but burns or is decomposed in a flame.—*Paper by R. E. Morris, abstracted in Bulletin, The Society of Naval Architects and Marine Engineers, February 1955; Vol. 10, p. 47.*

Ring Spring Clamping

The use of ring springs, consisting of a number of ring elements with conical surfaces placed in contact with each other to form a column (see Fig. 1) is fairly widely known.

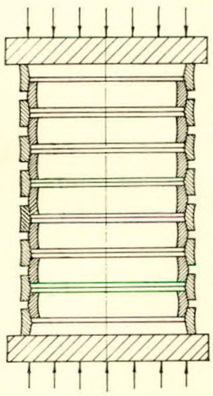


FIG. 1—*Normal ring spring compression unit*

These compression springs are particularly suitable for installations where a high degree of damping is required, e.g. railway buffer springs, because of the large energy dissipation through friction between the spring elements. A more recent development, carried out by Ringfeder, (Uerdingen), G.m.b.H., takes advantage of the fact that when a number of rings are placed together to form a column and are compressed, the outer rings expand and the inner ones contract radially. Normally this

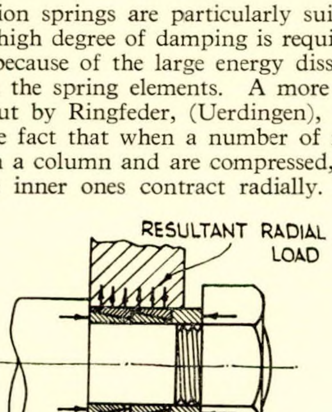


FIG. 2—*Principle of ring spring clamping*

expansion and contraction is allowed to take place quite freely. If, however, the inner rings are placed over a shaft and the outer ones enclosed inside the hub, the expansion and contraction are constrained and high radial pressures are set up across the ring elements, which lock the hub and shaft together. Fig. 2 shows such an installation, in which the rings are compressed axially between a collar and a clamping nut carried by a shaft. The radial pressure and hence the maximum torque, which may be transmitted across the spring element, depends upon the axial clamping force. This method of clamping offers a number of advantages over more conventional methods. The most obvious one which comes to mind is that splines, slots, flats, keyways or complicated machinery are unnecessary, thus eliminating the dangerous stress concentrations, which are attendant upon these discontinuities. The only preparation required is accurate machining of the shaft. Chatter and impact action, particularly under conditions of reversed torque or severe torsional vibration, are minimized. Provided the shaft and hub have been accurately machined, the double-taper action of the rings ensures accurate concentricity between them. The method is therefore highly suitable for high-speed machinery. The rings are uniformly stressed over their whole cross-section and hence the maximum use can be made of the material. Further, because of the uniform concentric clamping over the whole circumference of the rings, very compact weight-saving assemblies are possible. Relative adjustment between the shaft and hub can be carried out over a wide range in both the axial and rotational directions before the assembly is finally clamped. Subsequent adjustment may also be carried out quite simply by slackening off the clamping nut. If highly elastic steel or beryllium bronze is used for the ring elements, quite large clearances may be used to facilitate assembly. Another advantage claimed is that the clamped assembly is liquid tight up to high pressures.—*The Marine Engineer and Naval Architect, March 1955; Vol. 78, pp. 95-97.*

Gear Drives for Diesel Powered Vessels

The marine gear drives used with the Diesel engines have been developed from industrial speed reducer practice with the modifications dictated by shipboard operation. In general, the propeller thrust bearing is built into the marine gear drive—this has required strengthening the gear case. The torque pulsations from the propeller have required that the engine builder give special consideration to the torsional stiffness of the shafts and the inertia of rotating parts. While the operating speeds of these gear drives do not approach those of steam turbine gearing, it is customary to balance all major rotating members to ensure smooth operation. Reliability being of prime importance in marine equipment, all design work and selection of auxiliary equipment is on a conservative basis. This is the reason that ship specifications often call for gear tooth loading or K factors of 60 to 75 where K factors of 90 to 110 would be used in industrial practice. These higher K factors will still permit a service factor of $1\frac{1}{3}$ to $1\frac{1}{2}$ based on the recommended practice of the American Gear Manufacturers' Association. These service factors permit occasional peak loads of twice normal and appear to be adequate for these applications. Navy vessels, due to their special operating conditions, often permit higher gear loading. A K factor not exceeding 175 is sometimes specified for this class of gearing in Navy service. Loading of gear teeth is usually limited by wear or the possibility of surface failure such as pitting due to high contact stresses rather than by breakage. Contact stress is determined by the tooth loading, the modulus of elasticity of the materials and the radii of curvature of the mating surfaces. For a given size of gearing with a given load, the contact stress on helical or herringbone gearing is appreciably less than it would be on spur gears due to the greater length of tooth contact. This class of gearing will usually have a contact stress of 40,000 to 60,000 lb. per sq. in. at full load. Since K factor is related to contact stress, it is often used as a quick check of gear loading. The marine gear drive is an accurately made piece of machinery and as such deserves careful installation and reasonable care in operation. The output torque of the gearing and the propeller thrust load carried by its bearings require a good foundation structure to carry these forces as they usually exceed the weight of the gear drive. The gear drive is manufactured with all shafts parallel within 0.002-inch. The gearcase cannot be expected to maintain shaft alignment within these limits unless the foundation structure is adequate. This means a true surface with sufficient chocks properly fitted between the gearcase and the substructure, which must have adequate stiffness to withstand the applied forces without appreciable distortion under all normal operating conditions. Most of these gear drives have double helical or herringbone gearing as these have smoother action than spur gears and avoid the end thrust produced by single helical gearing. Hobbing is generally considered to produce more accurate gearing than the planing methods. Gear tooth profile and spacing errors are held to 0.0003-inch or less with a surface finish of 32 micro inch or better. Lead errors are held to less than 0.0001-inch per inch of face width.—*Paper by R. A. Frankenberg, abstracted in Bulletin, The Society of Naval Architects and Marine Engineers, February 1955; Vol. 10, p. 43.*

Comparison of Weight and Bulk of In-line Naval Gas Turbines

The effect is investigated of progressively increasing the complexity of gas turbine engines for naval application. The basis of comparison are:—the weight and bulk of the plant plus the fuel for a given operating schedule, plant layout in a typical ship, and size of deck openings. The weights and sizes of each of the four basic power plants considered are estimated from detailed design studies. Each complete power plant consists of a cruising and a boosting engine, and where possible, the engine components form in-line arrangements. An operating schedule of the ship is taken typical of medium sized naval vessels. It is concluded that, of the plants considered, the double-compound intercooled plant with heat exchange, used on the cruising engine, yields the least value of (plant plus fuel) weight and size of deck openings. The single-compressor-engined plant gives the shortest machinery length, though this plant is among the worst as regards overall bulk and size of deck openings. Of the double-compound-engined plants considered, that without intercooling and heat exchange is the worst as regards total weight, bulk and size of deck openings.—*Paper by D. F. Collins and D. W. Thomas read at a meeting of the North-East Coast Institution of Engineers and Shipbuilders on 11th March 1955.*

Carbon Dioxide as Protective Atmosphere in Welding

The potential widespread use of carbon dioxide gas as a protective atmosphere for gas-shielded consumable electrode welding of mild steel is of tremendous significance to the whole welding industry. Applications of this welding process for mild steel using an inert gas shield have gone forward in spite of the relatively high gas costs. The rate of argon gas usage is approximately fifty cubic feet per hour. It has long ago been recognized that carbon dioxide alone would be a more economical shielding gas if ways could be found to utilize it effectively. The most important factor with respect to the use of carbon dioxide is the low cost of shielding gas. Under the influence of arc heat it is generally believed that carbon dioxide breaks down into carbon monoxide and oxygen. In the area of the arc itself the gases break down further into atomic form. In the area surrounding the arc where the temperature drops to 634 deg. C. the carbon monoxide reacts with oxygen to reform carbon dioxide. An arc in carbon dioxide atmosphere is actually shielded by carbon monoxide and oxygen. Neither carbon dioxide or carbon monoxide are soluble in molten metal to any great extent, although the oxygen present due to decomposition is quite soluble in molten metal. The main drawback to the immediate widespread application of carbon dioxide has been the erratic actions of the arc. It has been found that erratic arc condition and extensive spatter can be materially reduced to the point where it is practical to use carbon dioxide as the sole shielding medium at tremendous savings. The way to control spatter using 100 per cent carbon dioxide as a shielding gas is through the use of high current density and short arc length. Many times the arc voltage is specified as the important variable. Actually, arc voltage is a convenient method of measuring arc length but only for a specific current. Holding constant arc voltage does not maintain constant arc length under different current conditions. Constant arc length is the important factor because arc length is a factor that controls bead contour, penetration, porosity, pool agitation, and undercut. The arc voltage, needed to maintain certain bead contour and penetration factors, changes as the current changes, but usually the same arc length will produce the same physical arc characteristics even though the current is changed. Therefore, the variable to control is arc length not arc voltage. The other variable that should be controlled to produce the best welds with carbon dioxide is the current density. It is important to use a very high current density for optimum results, about 100,000 amp. per sq. in. and approaching 200,000 amp. per sq. in. Since very high currents, deeply penetrating arcs and fast travel speeds are needed with carbon dioxide welding, an automatic travel mechanism is desirable. Work with carbon dioxide can be done manually but fast travel speeds are required. In the operation of hand welding, however, a stable welding generator is necessary to permit the arc length to be held to that value necessary for good welds.—*R. W. Tuthill, The Welding Journal, February 1955; Vol. 34, pp.137-141.*

Creep Data on Molybdenum, Chromium-molybdenum, and Molybdenum-vanadium Steels

Creep tests have been carried out to show how the creep resistance of a steel changes owing to the effect of prolonged exposure to stress and temperature. The effect of prior heat-treatment on this change is also considered and the limitations of short-time creep tests are discussed. Long-time creep and rupture tests have been carried out on test pieces from about ten open-hearth casts of each of three types of steel: 0·5 per cent Mo; 0.8 per cent Cr; 0.5 per cent Mo; and 0.5 per cent Mo, 0.25 per cent V. From these data an estimate has been made of the creep and rupture strength for a life of 100,000 hours. The results show that the Mo-V steel has creep and rupture properties much superior to the other two steels. The elongation at fracture of the rupture tests was measured, and it was noted that at any given temperature the elongation at fracture first decreased with increasing time to fracture and then increased with still longer times. It was also noted that as the temperature of testing was reduced a longer time was required to reach the minimum elongation and that the value of elongation at the minimum decreased with decreasing temperature. The elongation of the 0.5 per cent Mo steel became very low with long times of testing, but the elongation of the Cr-Mo and Mo-V steels remained high even in periods approaching 100,000 hours.—*J. Glen, Journal of the Iron and Steel Institute, April 1955; Vol. 179, pp. 320-336.*

Bunkering

The theories of bunkering apply almost exclusively to the fuelling of cargo ships, and are particularly significant in planning the bunkering of ships which are not confined to regular repetitious voyages over the same route. Such vessels, operating in other than liner services, most often carry heavy, dense cargoes, which pay transportation charges subject to frequent and extreme variation depending upon factors beyond the control of the ship operator. The success or failure of the voyage often depends upon the wisdom with which is determined the balance between cargo and fuel. Combination passenger-cargo ships, whose cargoes are made up principally of bulk commodities, freighted by measurement rather than by weight, and the large passenger vessels which rely on passengers as their main source of revenue, have ample reserve weight-carrying capacity for fuel. As a result, problems arising from the need to balance weight of pay-load against weight of fuel to be carried are absent or minimized. Basically, bunkering depends upon the deadweight carrying capacity of a ship, and the load-line requirements. By established usage, deadweight tonnage is considered to include water, stores, crew, cargo, and fuel. Of these five components, only the last two are variable. The quantity of water taken aboard at a port usually is the capacity of the tanks, and the stores and crew generally are lumped into a single gross figure of between 100 and 125 tons. The total tonnage devoted to water, stores, and crew for a standard 10,000-ton cargo ship comes to only a small percentage of her total tonnage, or a maximum of perhaps 300 to 350 tons. Cargo and fuel remain, then, the important variables. Since the reason for the ship's existence is cargo, the more cargo she carries, the greater will be her revenue. For example, the fuel tanks on a C-2 type cargo ship have a capacity of 1,675 tons, and the vessel's deadweight is about 8,750 tons. Obviously, the smaller the proportion of fuel-tank capacity actually used on a voyage, the greater can be the weights of cargo before the maximum of 8,750 tons lifting capacity of the ship is reached. Suppose that a C-2 type cargo ship were assigned to a voyage which, with fuelling stops, required twenty-five days to complete. If she took 350 tons of fuel and 150 tons of water and stores, she could lift 8,250 tons of cargo. By eliminating the fueling stops, the voyage could be shortened to twenty-two days, but the cargo capacity would be reduced by 500 tons, since the fuel requirements would be 850 tons. The shorter voyage, however, would permit 16.5 trips to be made in one full year, whereas the longer voyage would give time only for 14.6 trips. The total carrying capacity for the non-stop voyage would be 7,750 tons of cargo per trip, or 127,870 tons per year, whereas for the longer trip the carrying capacity would be 8,250 tons per trip, or 120,450 tons per year. —*Marine Engineering, March 1955; Vol. 60, pp. 54-57.*

Design of Wake-adapted Screws

No important ship is nowadays built without previous elaborate model scale experiments on hull and propeller, and owners and builders have become quite familiar with this mode of scientific research into problems of naval architecture. During the development of scientific research, the purely theoretical treatment of marine hydrodynamics fell into the background. Now, however, as in aerodynamics, a start has been made to investigate the field of unstable flow phenomena, of which very little is known, and the necessity of carrying out more fundamental research in this field is apparent. If it is remembered that the outcome of experimental research is almost directly dependent on the fundamental theoretical basis on which the experiment is founded, this leads to the conclusion that the recent theoretical work performed by various experimenters in the field of marine propeller design, must be regarded as an important advance. With regard to the theory of propellers adapted to the unequal velocity field behind the ship, the situation is less favourable. For propellers rotating in a radially unequal velocity field, the Goldstein reduction factors and the induction factors are at present being calculated in Holland and the United States respectively. The Goldstein reduction factors are "average" factors, which determine the decrease in circulation due to the finite number of blades. They are calculated for optimum distribution of circulation corresponding to the condition of minimum energy loss. The induction factors are non-dimensional quantities of the induced velocity components. They are calculated for an arbitrary distribution of circulation. When these factors are known, it will also be possible to introduce vortex sheets instead of vortex lines in order to calculate the curvature or deflexion of flow in way of the blade sections of these propellers. In the calculation of the influence of the circumferential inequality of the velocity field in way of the propeller in the behind condition, considerable difficulties are still encountered. It will probably be necessary to deal with this essentially unstable phenomenon in a way analogous to that in which the phenomena of aerofoil flutter are treated. In addition, the effects of the ship's movements due to a seaway on the intake velocities of the water into the screw disc will probably demand a similar treatment. Next, it should be kept in mind that the flow around the propeller is influenced by the presence of the rudder, which must, therefore, be taken into account in the calculation of the propeller. Owing to the risk of erosion, close attention must be paid to the choice of profile because it is, in practice, impossible to design a cavitation-free propeller operating in the circumferentially unequal velocity field behind a ship. Therefore a profile will have to be found that will show little, if any, erosion with the inevitable cavitation. The magnitude of each section's average angle of attack during one revolution is also of importance to the solution of this problem. Finally, an improvement in the present simplified strength calculation for the marine propeller should be aimed at with the aid of the above-mentioned refinements of the theory. It has been learned from experience that the design of wake-adapted screws according to the modern circulation theory leads to propellers with too small a pitch. Corrections are needed to give the screw the correct pitch and render it cavitation-free in a velocity field that is radially, but not circumferentially unequal. The authors therefore first consider the imperfections of the theory for wake-adapted screws more closely, and next discuss the compromise between theory and empirical data that has been arrived at by the Netherlands Ship Model Basin in order to design a wake-adapted screw which come closest to both the correct pitch and freedom from cavitation. By means of this compromise a series of 4-bladed wake-adapted screws is calculated, followed by a numerical example.—*Paper by J. D. Van Manen and W. P. A. Van Lammeren, read at a meeting of The Institution of Engineers and Shipbuilders in Scotland on 8th March 1955.*

Explosion Tests of Ship Plate Welds

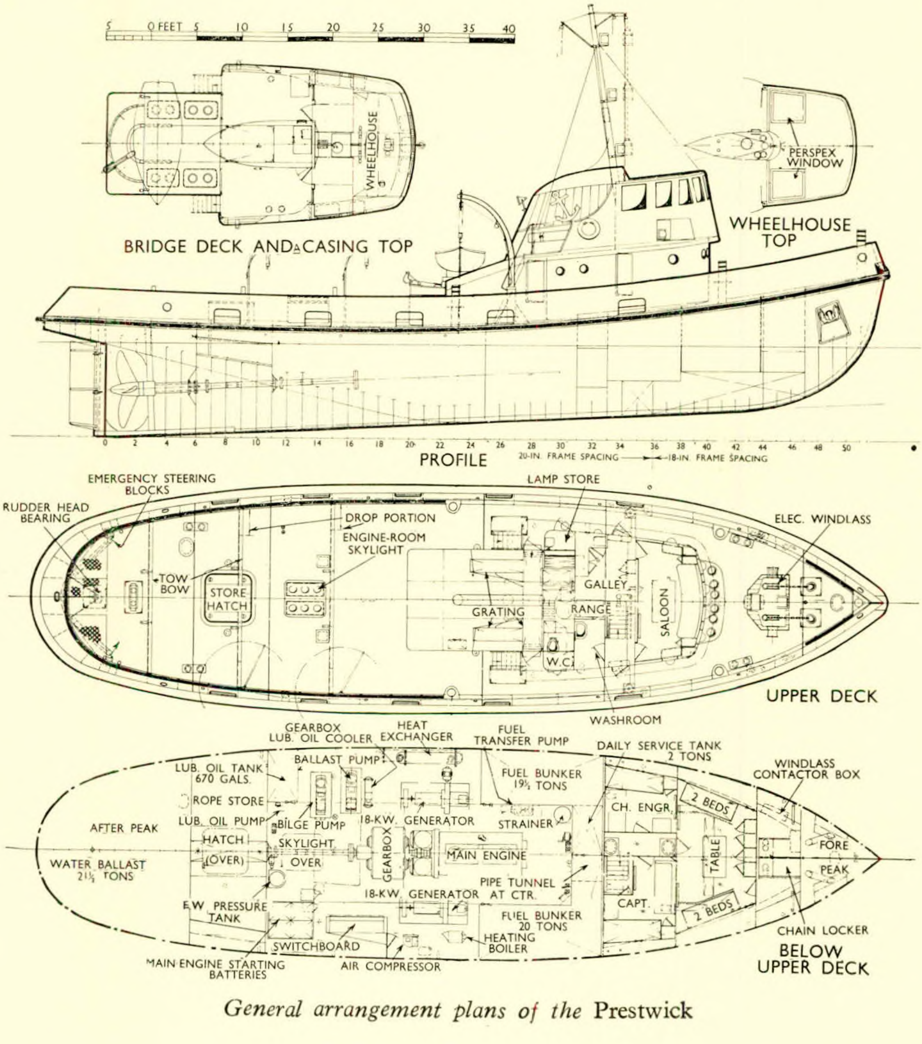
A series of investigations sponsored by the Bureau of Ships, Department of the Navy, revealed considerable difference in performance of welded alloy steel plate when subjected to the system of triaxial stresses believed to exist in the Direct Explosion Test. The Ship Structure Committee became interested in determining whether corresponding differences in performance would not be found in low-carbon steel plate used for the construction of merchant marine vessels. The Committee was particularly interested in determining whether differences in welding procedure would not result in a marked difference in performance. Accordingly, the Ship Structure Committee sponsored an investigation wherein two heats, one of fully-killed ABS Grade C and the other semi-killed ABS Grade B, both 1-inch thick, were welded with different welding procedures and subjected to Direct Explosion Tests. The principal result of this investigation indicated a marked improvement in performance of fully-killed steel when welded with low-hydrogen electrodes over the performance of the same steel when welded with cellulose type electrodes. The difference in performance of semi-killed steel when welded with the two respective grades of electrodes was less pronounced, the net effect being to approximate the performance of killed steel when welded with cellulose type electrodes. Although considerable scatter was observed to exist in the performance of semi-killed steel plates welded with low-hydrogen electrodes, even the specimens exhibiting the poorest performance still appeared to maintain a substantial, although not spectacular, superiority over the plates welded with cellulose type electrodes. Accordingly, it was decided that an additional investigation should be conducted to determine the degree of magnitude of improvement of structural performance which the use of low-hydrogen electrodes produced in semi-killed steel and, if possible, to establish whether it was of real significance. In addition, a brief investigation of the performance of rimmed steels under the Direct Explosion Test was also undertaken. However, part way through the investigation it became apparent that performance of a single heat of semi-killed steel was not uniform but varied very considerably depending on the particular slab and even plate used. As a result, a secondary objective of the investigation developed, and an attempt was made to establish the degree of variation of performance which can exist within one heat of steel and to correlate, if possible, this variation with conventional notched sensitivity tests, such as Charpy impact and Navy tear tests. An additional original objective of this investigation was a comparison of performance evaluation by Direct Explosion Test with performance evaluation by the Stand-off (Explosion Bulge) Test as developed by the U.S. Navy Research Laboratory. However, because of the difference in performance existing between different plates of the heat of steel purchased, the results of this comparison appear to be somewhat inconclusive. The Direct Explosion Test consists of subjecting a number of identical specimens to a blow produced by an explosion of a cylindrical charge of an explosive powder packed to a desired density. The magnitude of each charge is progressively increased until an energy value is reached which just fractures a specimen. The extent of deformation of the specimen subjected to the explosion of a charge just below the minimum charge to fracture is noted and provides an indication of the maximum deformation the plate tested can sustain under the test condition. Specimen failure is said to have occurred when the total length of all fractures exceeds 18 inches. (Actually, in the present investigation the overwhelming majority of plates which fractured at all, fractured into several pieces).—*G. S. Mikhalapov and W. A. Snelling, The Welding Journal, February 1955; Vol. 34, pp. 97-s-104-s.*

Devon-built Motor Tug

The first of five new motor tugs for France, Fenwick Tyne and Wear Company, ran trials last month and came to the Thames for inspection before sailing for the North-East Coast. Intended for harbour service on the Tyne and Wear, this tug, the *Prestwick*, has been built at the Appledore, N. Devon, shipyard of P. K. Harris and Sons, Ltd. The four subsequent vessels will be of identical design to the *Prestwick*, which has a hydroconic hull form designed by Burness, Kendall and Partners, Ltd. This hull design is contrived to give high hydro-dynamic efficiency, using conical development of the hull surfaces, and with it substantial economies in building, time and costs can be made. The overall length is 92 ft., the length b.p. 84 ft., and the moulded breadth 23 ft. 3 in., the depth being 11 ft. and the draught aft in service 10 ft. 6in. A 12-cylinder vee-type General Motors engine is installed developing 1,080 b.h.p. at 744 r.p.m., and coupled to a Lohmann in-line hydraulically operated reverse-reduction gearbox with a reduction ratio of 4 : 1, giving a propeller speed of 186 r.p.m. The cylinder bore is 216 mm. ($8\frac{1}{2}$ in.), the stroke 254 mm. (10 in.) and the m.e.p. at 750 r.p.m., 5.93 kg. per cm.^{2} (83.7 lb. per sq. in.). Scavenging, on the uniflow principal, is effected by a helical lobe Roots-type blower.—*The Motor Ship, April 1955; Vol. 36, p. 36.*

Effects of Underwater Explosion

In the course of the detonation process, an explosive material is transformed into gaseous products which ultimately reach a temperature of the order of 3,000 deg. C. and a pressure of the order of 50,000 kg per cm^2. The state of equilibrium of the detonation products represents an internal energy which is lower than that of the initial explosive material. The difference is the amount of energy released, which manifests itself in an enormous increase in the pressure and the temperature of the detonation products. The pressure disturbance initiated at the instant when an explosive charge detonates is propagated in all directions, and the pressure decreases as the distance from the detonation centre becomes greater. The pressure/time curve at a point situated outside the charge comprises two branches, i.e., a rising pressure branch, in which the pressure



General arrangement plans of the Prestwick increases to its maximum value in about one microsecond, i.e., practically instantaneously, and a falling pressure branch, in which the pressure decreases approximately in accordance with an exponential law. After a brief description of the loads caused by underwater explosions, an expression for the correction of the pressure acting on a surface in water for the motion of this surface is analytically deduced. The sum of the pressure exerted by an underwater explosion on a structure in water and the correction pressure due to the motion of the structure constitutes the load acting on the structure. The deformations of a clamped circular steel plate submitted to this load have been calculated analytically and investigated experimentally, and these results are compared. The principle of similarity is used for evolving a method of calculating the relation between loads and strains for plates having the same diameter/thickness ratio as those plates for which the corresponding relations have been determined by tests.—*The Engineers' Digest, March 1955; Vol. 16, p. 120.*

New Ship Hydrodynamics Laboratory

Work has commenced on the new test tank for the National Physical Laboratory at Feltham, Middlesex. The existing Yarrow Tank is 550 feet long, 30 feet broad and 12 feet deep, with a carriage speed of 20 feet per second and has been in service since 1911. This will be supplemented for ship model research by the new tank, 1,300 feet long, 48 feet broad and 2 feet deep, spanned by a carriage with a speed in excess of 50 feet per second and fine control down to 1 foot per second. Sufficient space is available to enable the tank to be extended to 2,000 feet if considered necessary. A wave-making machine will be installed. In the same building will be a steering and storage pond, 140 feet by 100 feet and 8 feet deep, in which radio controlled models will be used for steering and manœuvring trials. There will also be a workshop for the study of vibration problems on ship hulls. A separate building will house a water tunnel with a working section 44 inches in diameter and a flow of at least 50 feet per second. This will be pressurized and will be able to test propellers of up to 24 inches in diameter. It is planned that the new laboratory shall be fully completed by mid-1958.—*The Motor Ship, April 1955; Vol. 36, p. 31.*

Modern Diesel Tugs

Motor tugs have not only greatly increased in number, but have in many cases also increased in size and power. During this period they have been severely tested and have now developed into a sound workmanlike tool, reliable and efficient for their difficult and exacting work. Tugs have an arduous duty. They frequently have to vary speed and power in a very short space of time. The engine room telegraph registers "go ahead", "go astern" or "stop" in rapid succession, at both light loads and at full loads, hundreds of times a day, and the engine has to respond to these orders with infallible reliability. Failure to go "astern" or "ahead" when required in an emergency, might easily result in very serious damage, not only to the vessel itself, but possibly to other vessels. Therefore, although economy of fuel is important and desirable, reliability in performance and prompt and accurate response to the telegraph is of still greater importance. There is nowadays a distinct trend towards making it possible for the captain to control the movements of the tug from the wheelhouse instead of the orthodox method of transmitting his orders to the engine room by the ship's telegraph. This is deemed by some people to be a considerable advantage in that the captain can instantly get what he orders, leaving the engineer below free to devote the whole of his attention to the running of his engine.—*F. J. Mayor, Diesel Engine Users' Association, February 1955; pp. 1-2.*

Design of Marine Refrigeration Plants

The principal technical developments in ship design have been made during the current century, particularly in the last two decades. Some of the developments have been in the applications of refrigeration, the betterments having come from the demands made upon industry in carrying perishables. There are two classifications of ships carrying perishables, these being the all-refrigerated ship and the combination carrier. The trend in American ships has been toward the combination arrangement. The larger compartments of the all-refrigerated vessel do not provide the most suitable conditions for perishable cargoes as the volumes are such that they are subject to weather influences for the long periods of loading and discharge. The interval of loading is very critical in the keeping life of fresh fruits and vegetables. In the combination ship, the matter of location involves considerations of trim, stability and port operations. Perishables should be the last cargo loaded and the first discharged, and the uppermost location will facilitate that plan. Conditions of arrangement will suggest that the central machinery plant be located close to accessible compartments and adjacent to the main engine room where the operation can be under the scrutiny of watch engineers. The choice of insulator for the refrigerators is important and is an extremely controversial subject, not without the influence of salesmanship. The greatest error in judging insulators is that they are generally thought of only as a heat barrier when, for practical reasons, their value as moisture-vapour barriers is of almost equal importance. There are countless kinds of insulators, all having quite comparable insulating values in their dry condition. Almost without exception the elements of insulators are non-hygroscopic and are claimed as moisture resistant, but if the interstices are permeable and the construction does not provide the moisture-vapour barrier, atmospheric moisture will gain ingress there to be condensed, frosted, and frozen. There are few who seem to have full appreciation of the force of moisture-vapour, and it is of such magnitude that it should have greater consideration in the design of refrigerators and in the insulation of cold pipes and apparatus. Refrigerator linings in freight ships may be of water-proofed plyboard, while the most satisfactory lining for passenger ships is the cement-and-asbestos hard-board as instituted by the author in 1936. Floor wearing surfaces may be made of emulsified asphalt, sand, and cement. It must not be thermoplastic. Anhydrous ammonia is the best refrigerant, but the toxic hazards on vessels which may see war-time service have caused the general adoption of Freon-12. Direct expansion plants have their control at each compartment resulting in a decentralized system, and the piping is extended into inaccessible cargo spaces. Long lines cause suction pressure losses and a tendency to impound vaporized lubricating oil. A case is intentionally made in favour of indirect brine systems, which have simplified central plants, close connected and with small refrigerant charges. With such a plant the refrigeration load may be shared in any degree by the compressors, a condition not feasible for direct expansion cargo systems. A two-temperature brine system is recommended with cascade arrangement for the low temperature compressors. With brine, absolutely constant conditions can be maintained in the refrigerators, which places the carrier in a very desirable position when deliveries are less than satisfactory. In small rooms the wall coil is not as obsolete as some would have it. The cold air system, common to banana ships is highly efficient, but is seldom used on other carriers. The packaged cold air unit is popular but is not without faults. These faults may be minimized by studied cargo stowage methods. Manufacturers favour finned tubes, but the 1¼-inch prime surface coil is superior due to reduced frosting problems. Recording thermometers are essential to cargo refrigerator operation. With cold air units they should show the supply air temperature which is critical to fresh fruits and vegetables.—*Paper by L. L. Westling, abstracted in Bulletin, The Society of Naval Architects and Marine Engineers,1955; Vol. 10, p. 42.*

Canadian Twin Screw Ferry

During the course of this year the vehicle and passenger ferry *Bluenose*, being built by the Davie Shipbuilding Co., Lauzon, Quebec, for operation between Yarmouth and Nova Scotia, and Bar Harbour, Maine, U.S.A.—approximately a one-day journey across the wide entrance to Fundy Bay—will go into service. She will carry six hundred passengers in addition to cars, coaches and lorries, the length overall being 346 feet, the beam 45 feet 6 inches, the draught loaded 16 feet 6 inches and the speed 18½ knots. The machinery installation comprises six 2,000 b.h.p. Canadian Fairbanks-Morse opposed piston engines running at 700 r.p.m. and driving twin screws. Each set of three drives a propeller through two Hindmarch M.W.D. oil-operated reverse-reduction transmissions. Three engines are thus coupled to each gearbox and when running at 750 r.p.m. the propeller speed is 200 r.p.m. Each gearbox transmits 6,000 b.h.p. and the engine arrangement is in the form of a pair of the letters H with the inner after-leg omitted in each case. There are two engines forward and one engine aft of each of the two gearboxes each gearbox weighing 35 tons. The astern power is equal to that ahead. Each box is about 15 feet square and 9 feet high. A large lever gives "Forward," "Stop" and "Astern" movements on the whole transmission, whilst the three smaller levers isolate or engage the three engines individually. Any engine can be disconnected at will. The cylinder diameter is 8⅛ inches and the combined piston stroke 10 inches, the engines being of a type which was built largely for installation in American submarines during the war. *The Motor Ship, May 1955; Vol. 36, p. 80.*

French Turbine Tanker

The first of a group of four tankers of 31,500 tons d.w., ordered from the Chantiers and Ateliers de Saint-Nazaire Penhoet, by Societe Maritime Shell, has now been delivered. This tanker, the *Isanda*, launched on 20th July 1954, is one of twelve vessels of similar tonnage on order in British, Dutch, French and German shipyards for the Royal Dutch/Shell group of companies. At present the *Isanda* is the largest vessel in service for Shell, but two tankers of 38,000 tons d.w., the *Zenatia* and the *Zaphon*, are on order from British shipyards for delivery in 1956 and 1957 respectively.

Length o.a.	660 feet
Length b.p.	635 feet
Breadth moulded	84ft. 3in.
Depth moulded to upper deck ...	46ft. 3in.
Draught (summer)...	34ft. 8⅝in.
Deadweight, tons	31,500
Gross tonnage	20,713
Net tonnage	10,420
Service speed, knots	16½
Horsepower (normal), s.h.p. ...	13,750
Cargo capacity:	
Centre compartments, tons (oil)	25,000
Wing compartments, tons (water)	14,750

In addition to the *Isanda* there are three other vessels of 31,500 tons d.w. under construction at the Penhoet shipyard, the *Isidora*, *Isocardia* and *Isomeria*. The *Isodora* and *Isocardia* have already been launched and are likely to be delivered this year. The main propelling machinery in the *Isanda* consists of a set of h.p. and l.p. C.E.M.-Parsons type turbines driving a single screw through reduction gearing. Both turbines and gearing have been built at the St. Nazaire works. In this type of turbine two astern units are fitted, one being incorporated in the l.p. turbine casing and one being a separate turbine. These turbines are capable of developing a service power of 13,750 s.h.p. at 105 r.p.m., and a maximum designed power of 15,000 s.h.p. at about 108 r.p.m. The steam conditions are 600lb. per sq. in. and 850 deg. F. at the superheater outlet, and 585lb. per sq. in. and 840 deg. F. at the turbine throttle. The designed fuel rate (all purposes using oil at 18,500 B.Th.U. per lb.) is 0.53lb. per s.h.p. per hr. at maximum power and 0.533lb. per s.h.p. per hr. at normal speed. For ahead running, steam is supplied to the h.p. ahead turbine, the exhaust being led to the l.p. ahead turbine. For astern running, the power is obtained from one h.p. turbine which is connected to the h.p. primary pinion and from an l.p. turbine which is incorporated in the l.p. ahead turbine casing. These astern turbines develop together at least 80 per cent of normal ahead torque at 50 per cent of normal ahead speed. The astern power developed is at least 9,000 s.h.p. The main reduction gearing is of the double reduction type arranged in a rigid steel casing, with double helical type pinions and gears. The bearing bushes are arranged for easy replacement without removal of the rotating parts. A main shaft thrust bearing of pivoted segmental type is fitted. The propeller is of solid cast manganese bronze with five blades 21 feet dia., 16¼-inch mean pitch and weighing 24½ tons. The two boilers are of the Foster Wheeler type and have been made at Penhoet. They have been designed for a maximum pressure of 670lb. per sq. in. and for the following continuous operating conditions when burning oil of 18,500 B.Th.U. per lb.

Evaporation, normal, lb. per hr. ...	51,640
Efficiency, normal, per cent	88
Oil consumption, normal, lb. per hr. ...	3,655
Superheater outlet, pressure, lb. per sq. in.	600
Superheater outlet, temperature, deg. F.	850

Each boiler is fitted with four Penhoet fuel burners suitable for burning bunker C fuel oil. Hagan combustion control is fitted as well as manual control.—*The Shipping World, 6th April 1955; Vol. 132, pp. 402-406.*

Coated Valves for Engines Operating on Boiler Oil

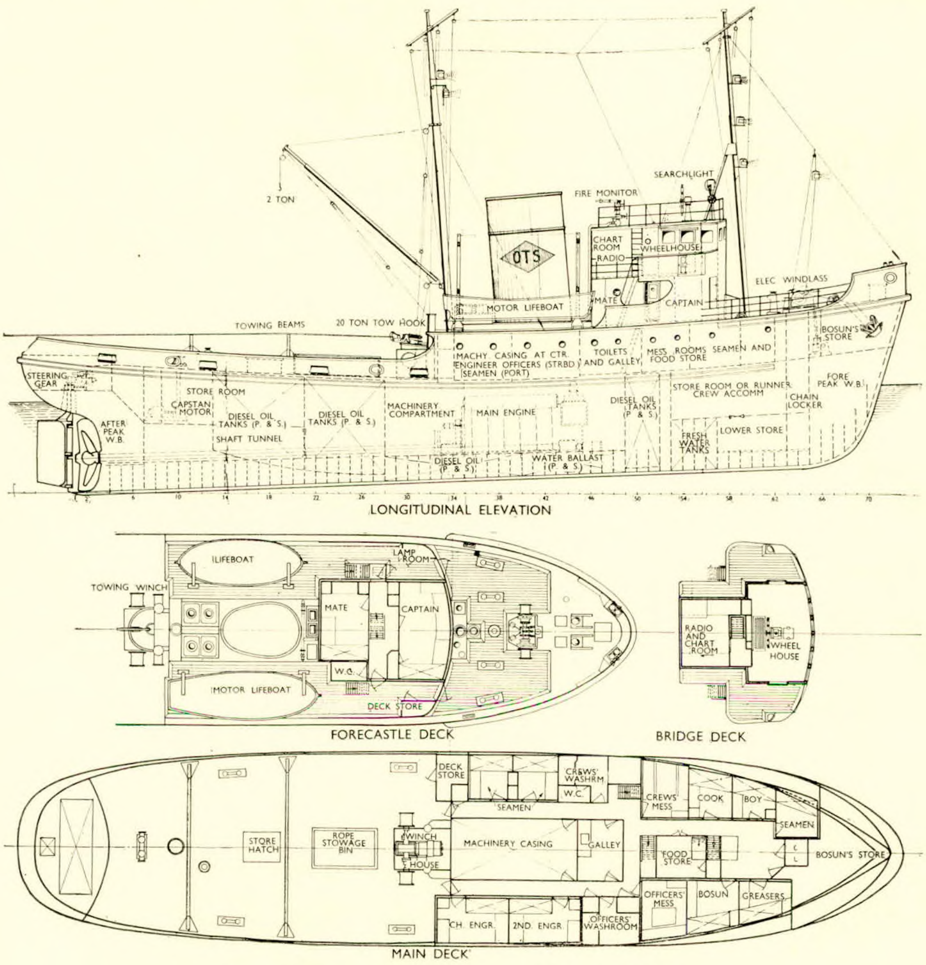
The Mirrlees K-type engine has a bore of 15 inches and a piston stroke of 18 inches. The power range is from 192 b.h.p. (a three-cylinder non-pressure-charged unit with a b.m.e.p. of 80 lb. per sq. in.) to 3,060 b.h.p. at 450 r.p.m. (a 12-cylinder turbocharged engine with a b.m.e.p. of 140lb. per sq. in. and with air cooling). Research was undertaken on a 621-b.h.p. three-cylinder turbocharged engine with boiler oils having viscosities of 220, 950, 3,500 and 6,000 sec. Red.1 at 100 deg. F., the trials proving conclusively that the regular use of such fuels with this engine was practicable and economic. A serious problem met in the earlier tests was the rapid corrosion of the exhaust-valve seating faces. This took the form of "wire drawing" or "guttering", followed by gas blow-by and severe burning of the valve over an arc of about 1½ inches. Standard valves made from a heat-resisting austenitic steel to B.S. Specification En.54 were burnt out in fifty hours or less. The trouble was evidently due to the corrosive action of the fuel-ash deposits, which gradually build up on the exhaust valves when non-distillate fuels are used. Most residual oils contain a small proportion of vanadium in their ash content, and it is considered likely that vanadium pentoxide is formed during the combustion process. Among the various attempts made to extend valve life was the experimental use of a set of En.54 valves coated on the seating face with B.A.C. Brightray nickel-chromium alloy, a product of Henry Wiggin and Co., Ltd., Birmingham. This proved a satisfactory solution, the coating revealing an ability to resist corrosion throughout long periods of continuous service. Furthermore, with the coating proving so effective, the valve material was subsequently changed to a silicon-chromium steel conforming with B.S. Specification En.52. It proved to have the necessary strength at high temperature for this application. As Brightray-coated valves made of silicon-chromium material are competitive in cost with the austenitic steel valves previously used, the Mirrlees company has, consequently, decided to standardize coated valves for all K-class engines. Users will thus be able to convert to heavy oil-burning without modifying any of the internal parts of their engines. Similarly coated valves will also be fitted to Mirrlees high-speed engines of the J type when these are required to operate on non-distillate fuels. The B.A.C. Brightray alloy is supplied in the form of wire and applied to the valve seat by means of an oxy-acetylene welding process, after primary machining operations have been performed on the valves. Following the deposition, the valves are fully heat-treated and finally machined to extremely close tolerances.—*The Motor Ship, April 1955; Vol. 36, p. 19.*

High Frequency Ignition System

The Plessey Company, in conjunction with D. Napier and Son, Ltd., the patentees, have developed a new type of ignition system operating on the capacitor discharge principle. The necessity for this new type of ignition results from the greater energy requirements for the ignition of fuels of low volatility used in gas turbines or oil-fired boilers. This high frequency system has definite advantages for some applications because of its high spark repetition rate, its low initial firing delay, and its use of a conventional type of sparking plug. The insulated gap of the igniter plug employed with the Napier-Plessey system is ionised by means of a high frequency, high voltage initiating surge which is immediately followed by a low voltage discharge of relatively large energy content. The high frequency component is additionally advantageous in ensuring a discharge through fouled plugs, but the combined discharge is, however, non-lethal. The repetition rate of this discharge is approximately 50 per second, and the initial firing delay, that is from switching "on," is less than 10 mS. These factors have proved to be of extreme importance in certain applications where considerable quantities of unburnt fuel would accumulate in the combustion chamber if there were a low repetition rate and a long initial firing delay. Such fuel flooding could easily occur in small turbines where it is vital that a spark should appear during the short period of favourable combustion conditions. Also the occurrence of a spark during this period means that much less energy is necessary for ignition than would be needed at a less favourable time, such as may occur with systems having a low repetition rate. The generally lower volatility of the fuel used for small turbines means that somewhat greater energy is required for ignition than that for example given by a booster coil for petrol ignition. A suitable energy content (actually dissipated at the plug) has been found to be of the order of 0.15 joules per spark with a dissipation time of between 50 and 100 microseconds. Because of the interrelated effects of plug position, chamber shape, method of fuel injection and air flow, the company is collaborating with manufacturers by providing experimental units to determine the best arrangements for effective ignition in various installations.—*The Shipping World, 16th March 1955; Vol. 132, p. 326.*

New Coastal and Deep-sea Tug

The 960-b.p.h. motor tug *Marinia* was delivered to the Overseas Towage and Salvage Co., Ltd., for salvage and deep-sea towage work. Built by Cook, Welton and Gemmell, Ltd., she is of the single-deck type with a registered length of 120ft. 9in., a beam of 28ft. 6in. and a moulded depth of 13ft. 9in. The propelling machinery comprises a 960-b.h.p. British Polar engine. The six cylinders have a bore of 340 mm., and a stroke of 570 mm., the b.m.e.p. at full load being about 78lb. per sq. in. Special quick-manœuvring gear has been fitted, consisting of a rotary air distributor which allows air to be admitted to the cylinders as the pistons approach top dead centre, and subsequently releases the air at the top of the stroke. In this way an effective brake is applied to stop the engine and propeller quickly, permitting reversal from full ahead to full astern within five seconds. Comprehensive fire-fighting and salvage equipment is fitted.—*The Motor Ship, April 1955; Vol. 36, p. 30.*



General arrangement plans of the Marinia

First Italian-built Cable Ship

The first vessel to be built in Italy for the repair and laying of submarine cable is about to enter the service of the Compagnia Italiana Navi Cablografiche. This vessel, the *Salernum*, is a motorship of 2,200 tons d.w. built by the S.A.

Navalmeccanica, Castellammare di Stabia, Naples. The *Salernum* is a twin-screw vessel with alternative direct Diesel or Diesel-electric drive. This type of propulsion has been chosen for the fine degree of manœuvrability which it offers; the vessel can, for instance, run on one engine at a very low speed, but can proceed to where work is to be carried out at 15 knots. In addition to her Diesel-electric propelling machinery, the *Salernum* has electric cable machinery. Although she is not the first cable-layer to be equipped with electric machinery (the *Monarch* has a d.c. system developed by Sir Alexander Gill, and manufactured by the Telegraph Construction and Maintenance Co., Ltd., with motors by the British Thomson-Houston Co., Ltd.), it is unusual, as most vessels of this type have steam cable-laying machinery. There is only one set of picking up and paying out gear, and this is situated forward and serves cable sheaves at both ends of the vessel. In addition to cable laying and repair the *Salernum* is equipped with instruments for hydrographic and oceanographic services. The principal particulars of this vessel are as follows:—

Length o.a., ft. ...	334.64
Length b.p. (B), ft. ...	299.54
Height to main deck, ft. ...	19.36
Breadth, ft. ...	41.45
Draught loaded, ft ...	18.37
Deadweight, tons ...	2,200
Gross tonnage ...	2,830
Speed, knots ...	15
Horsepower ...	3,500
Cable tank capacity, cu. ft. ...	22,354
Length of tank space (A), ft. ...	90.22
Length of engine space (C), ft. ...	72.18
Ratio of A to B, per cent ...	30.12
Ratio of C to B, per cent ...	22.09
Fuel capacity, cu. ft. ...	15.891

The *Salernum* has a clipper stem arranged with three cable sheaves and a cruiser stern with one cable sheave. Below the main deck, proceeding from the bow, are the cable grappling stores and refrigerated chambers, store rooms, galley, three holds, cable tanks, the main electrical power supply, main engine room, oil fuel tanks and steering gear. The P.O.'s and crew's quarters, messrooms and recreation rooms, technicians' quarters, drum room, the jointing workshops, stores and a working space 85.3 feet long by 14.76 feet wide are arranged on the main deck. The *Salernum* not only has equipment enabling her to carry out the maintenance and repair of submarine cables, but also has a large stern sheave which permits her to undertake laying operations aft. For this purpose it is normal practice to install two cable-laying machines, one forward in two sections for all duties, and a second one aft for long laying or the laying of new cable. In this vessel, however, one single powerful unit is installed forward from which cable may be laid from both forward and aft. Two special dynamometers of the plunger type have been fitted on the fore deck. They are arranged along the forward span of the cable and are intended for use during grappling and cable laying forward. For cable laying from the stern, a normal Johnson and Phillips type dynamometer is fitted. In order to obtain a high degree of manœuvrability it is usual to rely on the conventional reciprocating steam engine for propulsion. A cable layer fitted with steam turbines, the *Ampere*, has been built in France, but it was necessary in this case to fit a variable pitch propeller in order to obtain flexibility of control. The *Salernum* is equipped with two Fiat type 457T Diesel engines, each capable of developing 1,750 h.p. at 180 r.p.m. The normal speed of the vessel, 18 knots, is attained with the two engines developing 3,800 h.p. The engines are coupled to the shafts by means of Pomini type couplings. For propulsion during repair or cable-laying operations, two d.c. electric motors are also fitted on the propeller shafts abaft the Pomini couplings. These motors are capable of developing a total power of 500 h.p. at 100 r.p.m., and can be regulated in multiples of two revolutions from 15 r.p.m. upwards. When proceeding to the scene of operations the vessel can proceed on both Diesel engines, but when it is necessary to run at a very slow speed, or to go astern for long periods, the Diesel engines can be uncoupled and the vessel propelled by means of the electric motors.—*The Shipping World, 4th May 1955; Vol. 132, pp. 483-485.*

Electrolytic Descaling

A detailed report of the theory, technique and application of electrolytic derusting to the cargo tanks of United States Navy tankships is given. After a discussion of the various aspects of anodic and cathodic cleaning of rust from steels, the theory of the process, together with an account of patents, army use of the method and initial navy testing and applications is reviewed. Details of the laboratory investigations preceding the application of the descaling method are given, including apparatus and materials. Effects of anode spacing, electrolyte composition, temperature, effect on laying surfaces and on dissimilar metals (aluminium, zinc, lead and brass), additions of sulphuric acid to electrolyte, current reversals, detergents, hydrogen embrittlement are surveyed. Results using anodes of magnesium, aluminium, zinc, copper and platinum are reported. Volume of gases given off by the various anodes is measured. Reports on the electrolytic descaling of nine vessels are given, including an estimate of the final tank conditions, optimum current density, anode area, current distribution, washing down operations, intermittent flushing, hydrogen evolution and cost comparisons with conventional cleaning methods. Specific recommendations and an account of the practical details involved in the prodecure are given.—*F. E. Cook, H. S. Preiser, and J. F. Mills, Corrosion, April 1955; Vol. 11, pp. 31-51.*

Failure of Marine Boiler Tubes

Failures by stress-corrosion cracking of marine boiler tubes are commonly the result of a concentrated caustic solution in contact with stressed boiler metal. Usually the concentration of the caustic results from leakage between seams and around rivets, or capillary action and subsequent evaporation. A rare example of the same type of failure is one where no seams or crevices exist for concentrating the boiler solution by leakage; this is failure resulting from concentration of the boiler solution right on the steam generating surface of the boiler tube. Details of such a failure are discussed. In particular, it is shown that the attainment of the necessary caustic concentration and stress levels is feasible. It is pointed out that stress need not be as high as is commonly believed. The mechanism of caustic cracking as understood today is reviewed; that is, electrochemical action aided by stress. In effect, this serves to include caustic cracking among such phenomena as the season-cracking of brass and the stress-corrosion cracking of stainless steel. It is commonly accepted that this type of failure is rare. That it does occur suggests that some thought might be directed to preventing channelling by hot gases and to alternative chemical treatments which do not rely on caustic.—*R. D. Barer, Corrosion, April 1955; Vol. 11, pp. 18-24.*

Patent Specifications

Reducing Noises and Vibrations Produced by Propellers

In screw-propelled ships the tuning of the propeller almost always produces pronounced vibrations and noise, the sources of which vary somewhat on single and on twin screw ships and also depend upon the position of the propeller with respect to the outside plating of the ship. When the tips of the propeller blades pass closely to the plating, e.g. on twin screw ships or on single screw ships with the propeller turning rather closely underneath an overhanging stern, which is

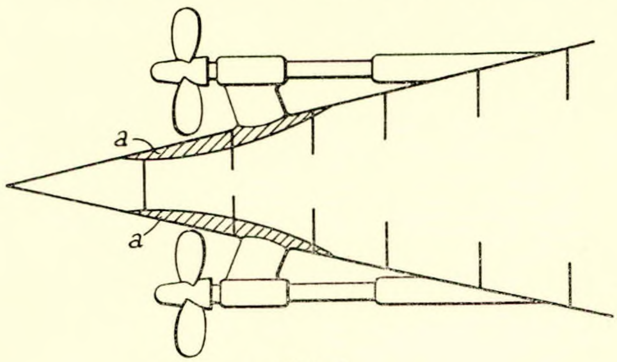


FIG. 1

especially the case on ships with a tunnel stern partly shrouding the propeller disc, or when the propeller disc is entirely shrouded by a double-walled Kort nozzle, strong suction forces of the water forward of the propeller cause the adjacent part of the ship's plates to buckle outward. *Vice versa,* aft of the propeller the strong pressure forces in the race tend to buckle the plates inward. As soon as the blade tips of the

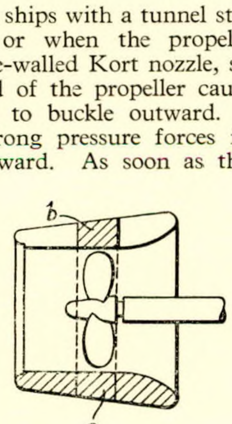


FIG. 2

propeller have passed the plates, the latter swing back due to their inherent elasticity. These movements are repeated when the succeeding blade tip passes the same spot thus producing at that spot continuous back and forward swinging of the plates. This is felt as trilling and heard as noises. In order to counteract these noises, according to the invention, the plating forming the ship's hull and/or appendages to it, e.g. a double walled nozzle ring, is fitted at its inside with anti-vibration or noise-absorbing material in way of or near to the circumference of the propeller disc, the material covering the plates. Fig. 1 is a plan view of the stern end of a twin screw ship, in which in the region of the propeller the plating is covered with a thick layer (a) of anti-vibration or noise-absorbing material. Fig. 2 is a section through a Kort nozzle the double-walled ring of which is either, as indicated (b) in the upper part of the section, partially filled with anti-vibration or noise-absorbing material, or alternatively, as indicated (c) in the lower part of the section, is entirely filled with such material.—*British Patent No. 730,580, issued to L. Kort. Complete specification published 25th May 1955.*

Floating Fish Container

Normally, a fishing trawler is also used for transporting the catch to the landing port. However, if the catching operation can be independent of the transport of the fish, the transport being carried out by a separate vessel, such trawlers may extend their stay at the fishing grounds, thereby saving time and the expense of a number of return voyages to the home port. In addition to reducing the number of voyages to the home port, the necessity is eliminated of providing storage space for the fish and of carrying ice for such storage. The space and the carrying capacity of the holds is thus available for extra fuel capacity, or a smaller type of trawler may be used. The transport vessel may be a factory ship for quick freezing and production of fishmeal and liver oil. The

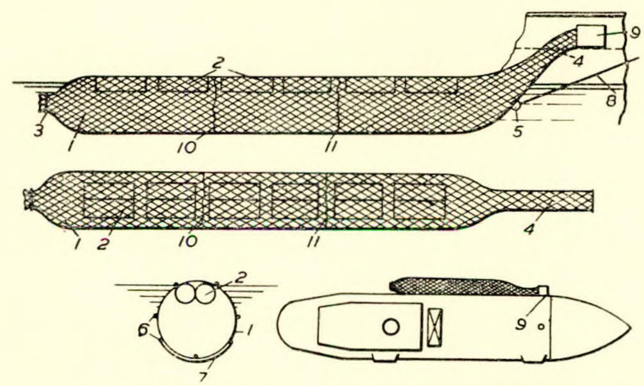


FIG. 1 (upper); FIG. 3 (left); FIG. 4 (right)

device, according to the invention, comprises the fish storage container (1) made of a flexible net-like material, which is supported by floating members such as air-filled rubber floats (2) disposed alongside or inside the container. The rear of the container is closed by a rope sling (3), as indicated in Fig. 1, which is released in order to empty fish from the container, and the front portion of the container is narrowed to form an elongated neck (4) through which the fish may be introduced into the container. The forward end of the neck (4) is provided with a mouth ring which is fixed to a receiving box (9). The receiving box is fastened to the bulwark of the trawler as shown in Figs. 1 and 4. A towing ring (5) is provided to receive a tow line (8), and in addition longitudinal ropes (6), shown in Fig. 3, are connected to the ring (5). The underside of the container is protected by a matting (7) of fibre or leather, so that the container is protected against wear and tear due to friction as the heavy container is pulled aboard the deck of the transport vessel.—*British Patent No. 730,652, issued to E. K. Roscher. Complete specification published 25th May 1955.*

Heating Coil for Tank Ship Cargo Tank

According to the invention, a heating coil or grid for a tank ship cargo tank is built up from pipes made of an alloy in which copper predominates. Preferably the copper content of the alloy will be between 75 and 85 per cent. The copper-base alloy pipes, whilst being resistant to corrosion, will also tend to minimize any possible galvanic action. One alloy suitable for this purpose is that which is generally known as "aluminium brass" containing approximately 76 per cent to 78 per cent copper, 20 to 22 per cent zinc and 1·8 to 2·3 per cent aluminium. Another alloy, described in British Patent No. 577,065, contains approximately 5 per cent nickel, 1 per cent iron and the remainder copper. Fig. 1 is a perspective view looking down into a centre tank of normal construction. The bottom (10) is part of the hull of the vessel, the keel (11), and there are the usual longitudinal reinforcing members (12) and the transverse reinforcing members (13). The latter members have lightening holes (18) in accordance with usual practice, and as the heating coil is built up from a number of lengths of piping (19) between the longitudinal members (12), these holes (18) serve also for the passage of the pipes (19). The lengths of piping (19) may be connected together in any convenient manner. In this particular construction, each longitudinal run of the piping comprises three lengths of the piping (19) connected together end to end by expanding the end of one length to receive the end portion of the next length shown at (20), the ends of the two pipes being secured together by brazing with a metal similar to that of the pipes.—*British Patent No. 732,072, issued to Munro and Miller, Ltd., and W. G. Munro. Complete specification published 15th June 1955.*

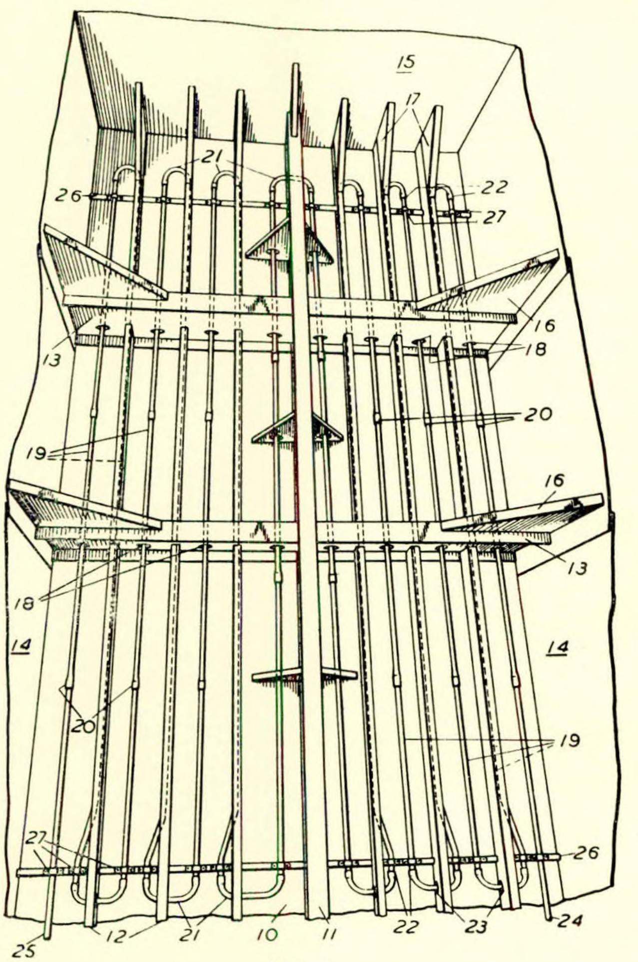


Fig. 1

Production of Ships' Propellers

This invention is concerned with the production of ships' propellers and relates especially to a method and machine for machining the driving surfaces of the rough cast propeller to a form in which they can be hand finished. Nearly all ships' propellers of modern design have blades varying in pitch between the root of the blade and the tip of the blade. Consequently it is not possible to machine the driving surface of the blades by the generating processes which were used formerly when propellers normally had constant pitch between root and tip. According to the invention, a method of machining the driving surfaces of the blades comprises cutting a radial groove

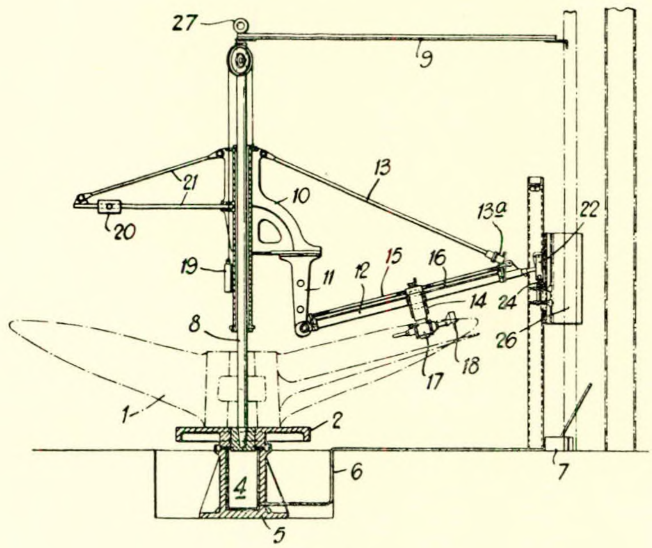


Fig. 1

from root to tip of each blade of the rough casting at the correct rake angle of the blade and then cutting a series of circumferential grooves at close intervals across each blade. The bottom of each circumferential groove is cut at the correct pitch angle corresponding to the part of the blade on which the circumferential groove is cut and at the same depth as the radial groove at its intersection with the circumferential groove. The bottoms of all the radial and circumferential grooves will then lie at such a depth as to provide guidance for finishing operations. In operation, the propeller (1) on table (2) is carefully centred to the post (8) and table (2) is turned with the help of hand pump (7) so that one blade lies towards the centre of the pitch rail (24). By means of the adjustable tie bar (13), the guide (12) is inclined to the correct rake angle of the propeller blade and its length is adjusted, if necessary (Fig. 1). The grinding machine (17) is started and a groove is cut from the tip to the root of the propeller blade on the centre line, feeding the grinding machine (17) along the guide (12). The depth of this radial groove is determined by the amount of metal left on the casting for finishing. This operation is repeated on each blade of the propeller.—*British Patent No. 731, 293, issued to Bull's Metal and Marine, Ltd. Complete specification published 8th June 1955.*

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Marine Engineering and Shipbuilding Abstracts

Volume XVIII, No. 9, September 1955

Weldability of Wrought High-alloy Materials

The increasing use of high alloys as materials of construction has created a demand for more information on their fabricating characteristics. Much development work has been centred on the joining and forming of alloys capable of resisting corrosion, high-temperatures plus oxidation, stresses and mechanical shocks. It is the procedures for joining, and the properties to be expected of weld joints in these alloys, that will be discussed in detail here. Tests were made with nickel-molybdenum alloys, nickel-molybdenum-chromium alloys, cobalt-chromium-tungsten-nickel alloys, and N-155, an iron-base alloy with large percentages of chromium, nickel and cobalt. These were welded by the submerged arc, inert-gas-shielded arc and metallic-arc methods. The properties of the weld joints and the weld metal were satisfactory and fell within acceptable limits for alloys of this type. All of the alloys listed will produce weld joints meeting the minimum tensile requirements for the parent metal. The alloys tested have characteristics of the austenitic type and, generally speaking, can be welded by all of the methods common to the austenitic stainless steels. Certain of the alloys, however, exhibit characteristics that are peculiar to that alloy composition during welding and fabricating operations. Nickel-molybdenum alloys, for example, are subject to stress-cracking in the hot-short range and are the most critical of the alloys in this respect. Nickel-molybdenum-chromium and cobalt-chromium-tungsten-nickel alloys are subject to work-hardening during forming operations. These alloys, however, are not as susceptible to stress-cracking during welding of restrained components as are the nickel-molybdenum alloys. N-155 alloy is subject to microfissuring if proper welding techniques are not used. All of these alloys can be welded satisfactorily when proper procedures are followed.—*R. P. Culbertson, The Welding Journal, March 1955; Vol. 34, pp. 220-230.*

Electrolytic Tank Investigation into Stepped Thrust Bearings

Many years ago the very unsatisfactory bulky and expensive multi-collar thrust-bearing was replaced by the very compact Michell tilting-pad thrust-bearing, which has given very atisfactory service. Of recent years, interest has been focused n simpler types of non-tilting thrust-bearings such as parallel-surface bearings and bearings with a machined taper. There is a general desire for a satisfactory cheap, simple, thrust-bearing. An investigation into stepped thrust-bearings has been carried out with an electrolytic tank, and results indicate that the load capacity is greater than for the tilting-pad bearing. The friction is increased but not in the same proportion as the load capacity. By a suitable choice of the operating conditions it is possible to design a stepped-pad thrust-bearing which operates over a wide range of loads with a greater film thickness and a lower coefficient of friction than a tilting-pad thrust-bearing. Owing to the decrease in friction and the simple and cheap construction, there appears no outstanding reason why this new bearing should not supersede the conventional tilting-pad thrust-bearing.—*Paper by C. F. Kettleborough, read at a General Meeting of The Institution of Mechanical Engineers, on 1st April 1955.*

Temper Brittleness of Pressure Vessel Steels

Temper brittleness is an embrittlement of carbon and low-alloy steels arising from exposure to temperatures above 700 deg. F. but below those at which austenite forms. It is characterized by a rise in the temperature of transition from brittle to ductile failure, with no change in strength or hardness. The severity of temper brittleness in the low-carbon pressure vessel steels commonly used in the as-rolled or normalized condition is generally not appreciable, and no failures from this source have been reported. The effect may be more severe in quenched and tempered low-alloy steels, particularly those containing manganese, chromium and, to a lesser extent, nickel. The mechanism of temper brittleness is not known, but the conditions under which it is produced are fairly well defined. The classic example of temper brittleness is that arising in quenched and tempered alloy steels on slow cooling from the tempering temperature. In general, the obvious method of preventing temper brittleness is to avoid the temperature conditions under which it develops. In quenched and tempered steels, this means, for example, tempering at a high temperature for a short time rather than at a lower temperature for a longer time. Water quenching from the tempering temperature is often practised. Heating above the austenitizing temperature eliminates brittleness resulting from previous treatments. Molybdenum up to about 0.60 per cent is also useful in greatly retarding temper embrittlement. The values obtained from the conventional tensile test give little indication of the susceptibility of a steel to temper embrittlement. The usual method of measuring susceptibility is to use the notched-bar impact test, and to compare the transition temperature of specimens slowly cooled from a tempering temperature of about 1,100 deg. F. with the transition temperature of specimens water quenched from the same temperature.—*L. D. Jaffe, The Welding Journal, March 1955; Vol. 34, pp. 141s-152s.*

Geometrical Size Effect in Notch Brittle Fracture

Fracture size effects are discussed in the light of metallurgical and geometrical variables, and the Docherty size effect in slow notched-bar bend testing is put forward as an example of a purely geometrical effect. The results of repeat tests are given which show that this effect is present when metallurgical differences between specimens are reduced as far as possible. Strain-energy release rates during fracture are determined for the specimens, and checked experimentally by means of the temperature wave method described elsewhere. It is concluded that brittle fracture in the smallest of the geometrically similar specimens is prevented by there being an insufficient strain-energy release rate, and that the critical value is close to values determined for the same steel and temperature by means of a notched tension test. The implications of such a criterion for propagation are then discussed in relation to a particular type of fracture which is common in the case of full-scale welded steel plate structures.—*Paper by A. A. Wells, read at a meeting of the North-East Coast Institution of Engineers and Shipbuilders on 25th March 1955.*

A Hundred Years of Lloyd's Register of Ships

This paper deals with Lloyd's Register Rules for the structure of ships since their rules for iron ships were introduced on 12th April 1855. It is intended to show the development of the Rules as the design of the structure of ships altered and as new materials were introduced. The paper is divided into two sections—one dealing with the Rules proper and the other containing some observations on the underlying developments in ship design which have been reflected in the Rules, and the methods adopted by the Society to ensure that progress has not been retarded. During the one hundred years under review there has been constant development and progress in methods of ship construction and these developments have been progressively reflected in the Ship Rules. Over that period the Society has endeavoured to preserve a just balance. It has not resisted innovations, but at the same time it has fulfilled its function of ensuring that innovations are not allowed to affect adversely the required standards of strength and security of the ship's structure. This work is effected through three agencies: experience, experiment, and calculation, and Lloyd's Rules are founded on these bases. Of these three factors experience is the most important, and Lloyd's Register, through the numerous reports received from its Surveyors all over the world, is able to collate and analyse the information of the behaviour of ships in service, which forms the best means for determining the efficiency or otherwise of any specific arrangement. All experiment and theory must be related to that experience, and the Society can claim that full advantage has been taken of the help given by these means towards the improvement of the structure of a ship. This work has only been made possible through the assistance given by the shipbuilding and shipping industry which, during the period covered by the paper, has co-operated with the Society towards the common object of improving the standards of ship construction. A great amount of pioneer work and research has been done and advantage has been taken of this by the Society in the formulation of the Rules. In this paper only the broadest lines of the Rules and their development are considered. No attempt is made to deal with the subject of classification itself, which necessitates these Rules, or with the survey work which has to be carried out in order to see that the ship is built according to the Rules and maintained in class.—*Paper by J. M. Murray, read at a meeting of the Institution of Naval Architects on 5th April 1955.*

Some Investigations into Singing Propellers

The object of this paper is to review certain cases of propeller singing that have been investigated by the Admiralty in the last few years and to supplement the data that are already available concerning singing propellers in merchant ships and which have been the subject of a number of papers. A propeller is said to be singing when the acoustic spectrum of the noise emitted by the propeller contains one or more distinct frequencies whose level is well above that of the remainder of the spectrum. In other words, a musical note is clearly audible above the normal white spectrum of a ship's propeller. The phenomenon of singing may only be of slight importance in a cargo vessel in peacetime but, in a passenger-carrying ship with accommodation aft, a severe case of singing might cause considerable nuisance. In a warship, where inevitably some accommodation spaces (and often officers' cabins) have to be situated in the stern, the same applies. Moreover, any increase in the noise emitted by a propeller may have serious operational consequences. It is considered that there may be four possible explanations of the mechanism of excitation of propeller blade vibration, namely:—1. Periodic eddy shedding from trailing edges or form tips. 2. Alternating fluctuations in flow conditions or in cavitation at leading edges. 3. A phenomenon similar to aircraft wing flutter, where an unstable or oscillating condition may arise from the coupling of flexural and torsional vibration with a variation of lift due to the consequential change in flow. 4. Self-excited vibrations caused by bearing friction. From the evidence that has been accumulated it can be concluded that any one of the four possible causes of propeller blade vibration may be responsible for the singing of a particular propeller. Although the evidence is not yet complete, it appears that propellers in which the blades are each of slightly different frequencies are more likely to sing than those of precision maunfacture in which all blades being carefully balanced are of identical frequency. In cases where the stern bearing has been giving trouble the possibility of bearing friction being the cause of the singing must not be disregarded. It is also clearly advantageous, if other circumstances permit, to use a propeller material, such as cast-iron, possessing high internal-damping properties. In any case aluminium bronze appears to give this advantage without any penalty such as corrosion or reduced strength. It is also considered that the chances of a propeller's singing are lessened if the lines and appendages of the after end of the hull are designed to give a near uniform flow to the propeller. It is considered that a great deal of research is still needed on this subject before the problem can be completely solved. It is suggested that this research should include the following items:—1. Further experiments on singing vanes, including the effect of different edge shapes and surface finish. 2. Experiments with hot-wire anemometers and pressure pick-ups set into leading and trailing edges of vanes in order to locate and measure exciting forces. 3. Full-scale trials of further modifications to singing propellers including thickening blade edge—leading and trailing separately. 4. An extension of the theory of flutter applied to modes of vibration of propeller blades. 5. Further experiments on singing model propellers. —*Paper by S. G. Lankester and W. D. Wallace, read at a meeting of the North-East Coast Institution of Engineers and Shipbuilders on 4th April 1955.*

Ship Motions in Regular and Irregular Seas

The paper is written as a record of an informal lecture delivered by the authors at the Office of Naval Research in Washington, D.C., during 1954. It consists of two distinct parts: the first describes the application of a theory of rigid body motions to a ship moving in a regular head or following sea; the second part deals with the problem of the representation of an irregular storm sea and of the theory of ship motions in irregular seas. The theory of ship motions in a regular sea, which is presented first, can be considered as the continuation of the work originated by Kriloff in 1896 and re-presented recently in the most developed form in papers by Weinblum and St. Denis. The new development consists in the introduction of coupling between heave and pitch motions, and in the more complete discussion and evaluation of various coefficients in the coupled differential equations of motions, particularly those of the cross-coupling terms. Recent experimental data obtained at E.T.T. on forcing functions due to waves show them to be much smaller than was previously assumed on the basis of the Froude-Kriloff hypothesis. A comparison of computed and experimentally determined ship motions shows quite good correlation. In the second part, some significant features of recent theories for the study of ship motions in irregular seas are discussed. It is shown that progress in this phase of seakeeping research does not need to await the complete solution of the problem of motions in simple seas.—*B. V. Korvin-Kroukovsky and E. V. Lewis, International Shipbuilding Progress, No. 6, 1955; Vol. 2, pp. 81-95.*

Superheater Metal Temperature

One of the critical problems in the design of a modern steam generator is the selection of material for the high-temperature, high-pressure superheaters and for reheaters. Careful calculation of metal temperature is necessary during the design of superheaters, and in this paper the author gives in detail a method of evaluating the several variables involved in the determination of the wall temperature of superheater and reheater tubing. Actual measured temperatures are discussed to show the influence of various elements of boiler design and operation on temperature, and some methods and locations for making metal-temperature measurements are considered. The study of metal temperatures in many superheaters leads to the following general observations:—1) It is recommended that during normal operation temperatures metal be measured on high-temperature superheater installations, and that these temperature measurements should be taken outside of the gas stream. 2) If an unsatisfactory temperature distribution is recorded, the possibility of failures can generally be overcome by burner adjustments, by introducing orifices in individual tubes, by baffling the gas-side of the superheater, or by preventing undesirable gas-side deposits. 3) While extreme care may be exercised in obtaining safe normal working metal temperatures, early failures may occur as a result of incorrect start-up and shut-down procedure.—*Paper by G. Parmakian and N. S. Sellers, read at ASME Annual Meeting, 1954. The British Shipbuilding Research Association, February 1955; Vol. 10, Abstract No. 9908.*

Supercharging the Marine Diesel Engine

The paper outlines the development of the supercharging of 4-cycle and 2-cycle engines by increasing the quantity of air in the cylinder by large scavenging pumps. It then outlines the development of supercharging of 4-cycle engines by exhaust turbo-blowers, and later by high-ratio turbo-chargers. Supercharging of the 2-cycle engine by high-efficiency turbo-blowers is then considered, and its application to the Doxford engine is described. The influence of supercharging on certain engine characteristics is examined. Turbo-blowers are described, and the arrangements for the supercharging of 2-cycle engines of from three to eight cylinders and the influence on the order of firing of the cylinders is considered. Methods of assisting the turbo-blower to give more air under low-power conditions are discussed and finally the possibilities of supercharging and its influence on large propulsion engines.—*Paper by P. Jackson, read at a meeting of the Institution of Engineers and Shipbuilders in Scotland, 22nd March 1955.*

Large French Tanker

The 24,500-ton tanker *Blaise Pascal* has been built by Kockums at Malmo for French owners. The hull, which is longitudinally framed, is entirely welded; the steel employed for the main components is a manganese steel, with a high strength at low temperatures, known as Coltuf 28. A radius is formed at the junction of the deck and the sheer strake. The cargo space is divided into twelve centre tanks and a total of twelve side tanks, corrugated bulkheads being employed. The tanks are divided into three groups by means of two pump-rooms, each of which contains two steam-driven cargo pumps each with an hourly capacity of 400 tons, and a stripping pump with an hourly capacity of 150 tons. The main propulsion machinery comprises a nine-cylinder two-stroke single-acting M.A.N. engine developing 8,100 b.h.p. at 115 r.p.m., which is arranged to run on fuel oil but to use Diesel fuel for manœuvring. Three-phase electric power at 380 V, 50 c.p.s., is generated by two 270-kVA Diesel-alternator sets, and by one 95-kVA steam-driven alternator. Steam is produced by two oil-fired boilers, and by an exhaust-gas boiler which can be operated either independently or in parallel with the main boilers. The principal particulars are:—

Length overall	600 feet
Length b.p.	565 feet
Breadth moulded	71 feet
Depth	42.4 feet
Draught, loaded	31.6 feet
Deadweight, tons	24,530
Speed on trials, knots	15.25

Journal de la Marine Marchande, 14th October 1954; Vol. 36, p. 2389. Journal, The British Shipbuilding Research Association, February 1955; Vol. 10, Abstract No. 9875.

Sea Trials on 9,500-ton Deadweight Motor Cargo Liner

As second part of the programme sea trials of the Centre Belge de Recherches Navales, M.V. *Lubumbashi*, a newly-built cargo liner of the Compagnie Maritime Belge, was equipped with torsionmeter, thrustmeter, pitometer log, anemometer and windvane and again, in varying conditions of draught and fouling weather, numerous records were collected of speed through the water, power and fuel consumption, thrust, revolutions, ship motions, wind and waves. In a similar way as for Victory ship *Tervaete* the service data are analysed and an attempt is made to ascertain the efficiency and economy of the ship and machinery in different service conditions. A predominant part of the programme was the ship-model correlation. Two measured-mile trials were carried out, the first as part of the official trials of the ship in ballast condition, the second in loaded condition at the beginning of her maiden trip.—*Paper by G. Aertssen, read at a joint meeting of the Institution of Naval Architects and the Institute of Marine Engineers on 5th April 1955.*

B.S.R.A. Resistance Experiments on the *Lucy Ashton*

This is the fourth and final paper dealing with the presentation of the results of the experiments carried out on the *Lucy Ashton* by the British Shipbuilding Research Association. It is concerned with certain miscellaneous investigations which included:—(*a*) The effects of fouling on resistance corresponding to various periods of immersion at different times of the year. (*b*) Acceleration and retardation trials from which the virtual mass of the ship was determined. (*c*) Pitot traverses in the ship's frictional belt, giving velocity distributions for various hull-surface conditions. A general appraisal is given at the end of the paper in which the principal findings from the work as a whole are briefly reviewed. Arising from this, some suggestions are made regarding the general lines on which further work might be pursued.—*Paper by S. L. Smith, read at a meeting of The Institution of Naval Architects on 5th April 1955.*

The Cathodic Protection of Ships Against Sea Water Corrosion

When steel (e.g. a ship's hull) is immersed in sea water small galvanic currents flow from anodic parts of the surface of the metal causing corrosion. Cathodic protection is a means of stopping corrosion by suppressing these galvanic cells by a current passed to the hull from external anodes, thus making the whole immersed hull surface cathodic. Current may be obtained by using active anodes of a baser metal, or from an external supply through inert anodes. Various types of anodes for each method have been used, the most common being magnesium alloy and steel anodes respectively. The adequacy of cathodic protection in preventing corrosion is guaranteed if the electrical potential of the hull is maintained, by adjustment of the current, at about 0.8 to 0.9 volt negative to a standard silver electrode. Various applications of cathodic protection to ships and other structures are described, a particular example being that of ships laid up in reserve. The pros and cons of applying the process to active ships are examined. Reference is made to special problems in connexion with cathodic protection including the effect on bottom paint and fouling, the application of cathodic protection in fresh water and to aluminium, and the use of zinc protectors.—*Paper by L. T. Carter and J. T. Crennell, read at a meeting of the Institution of Naval Architects on 6th April 1955.*

Properties of Cast Iron at Elevated Temperatures

The A.S.T.M.—A.S.M.E. Committee on the Effects of Temperature on the Properties of Metals is sponsoring an investigation of the properties of cast iron at high temperatures; from this work it is hoped to determine the load-carrying ability of cast iron in the temperature range 700 deg. to 1,000 deg. F. The work is to be divided into three sections. The first will consist of a literature survey and screening tests on a large number of commercial cast irons to select those with promising high-temperature properties for the main part of the work. In the second section, stress rupture and creep tests will be carried out on the selected material; and finally thermal-shock tests will be carried out on cast irons having good stress-rupture and creep properties. This report covers the literature survey and screening tests on twelve commercial irons to determine their resistance to softening at high temperatures. These tests showed good correlation between alloy content and resistance to softening, but the literature survey showed only partial correlation between creep and stress rupture and other high-temperature properties. A selection of eight irons for stress-rupture and creep tests was made based on the results of the literature survey and screening tests.—*Paper by J. R. Kattus, read at ASME Annual Meeting, 1954. Journal, The British Shipbuilding Research Association, February 1955; Vol. 10, Abstract No. 9904.*

Dynamics of Anti-rolling Tanks

Among the dynamical problems connected with ship motions, considerable attention has been given over the past seventy years to the rolling motion of a ship fitted with stabilizing tanks. However, the current opinion on the subject is still far from being unanimous and a number of erroneous statements can be found in recent publications. This paper is concerned primarily with the theory of U-tube tank stabilizers, and it is the aim of the author to clarify and consolidate the theory as it has been developed so far and to provide certain additions which will make it more complete, more accurate and more practical. Two types of tanks are discussed. In the first type, the fluid in the tank is subjected to no other forces but those which result from the tank being part of the rolling ship; in the second type the fluid, in addition, is acted upon by a pump, compressed air, or similar means. The first type is known as "passive tanks", the second one as "active, or activated, tanks". The first chapter deals with the ship tank system and one of its mechanical analogues, the double pendulum, in a general way, in order to clarify certain aspects of the problem. The second chapter of the paper is devoted to establishing the differential equations of motion for the ship-tank system in a more thorough manner than has been done previously, with one essential simplifying assumption, viz., the axis of rotation is taken as fixed. Two subsequent chapters discuss the effect of the parameters on system behaviour, that is, the motion described by the solutions of the differential equations. The third chapter does this in a general way and the fourth chapter treats the passive tank in particular with special emphasis on obtaining optimum performance. The last chapter extends the discussion to systems with a moving axis of rotation, and presents a review of system parameter values in practical installations. The paper concludes with comments on literature on the subject extant, and a bibliography is also given.—*J. H. Chadwick and K. Klotter, Schiffstechnik, February 1955; Vol. 2, pp. 85-104.*

Nuclear Propulsion for Merchant Ships

In a recent issue of the United States Navy Service Journal an article was published by Holmes F. Crouch entitled "Will Nuclear Fuels Run Merchant Ships?" A careful analysis of the present position and future possibilities was made in this article. Extracts are given below from which it would appear that the outlook for a considerable time to come is by no means optimistic. There has been much speculation and prophecy about the application of "atomic power" to merchant ships. The generalization seems to be that with the advent of nuclear energy, its utilization by large ocean liners is automatic—given time and the proper economic incentives. Evidence of this thinking stems from the many references to the *Nautilus* and *Sea Wolf* as examples of the futureattainment of this goal. But, actually, so very little specific and comparative information has been given out that the realistic question arises: Will nuclear power really benefit merchant ships? Everyone says it will, but will it? Indications are that the two nuclear submarines will prove operationally successful. In fact, publicized long-range plans include the building of two more—making a total of four atomic submarines. Obviously, since nuclear reactors producing propulsion power are feasible for ships of the Navy, why not so for merchant ships? Such interest often leads to thinking in terms of a direct substitution of the submarine reactors for conventional merchant marine boilers. However, this is impracticable . . . for technical and economic reasons which follow. The submarine reactors are of the "once-through" type. That is, each charge of nuclear fuel is used solely for the production of power. Because of nuclear peculiarities, this is not the most efficient reactor design. A better design would be to utilize those neutrons which unavoidably "escape" from the reactor core, for the breeding of new fissionable fuel. While military considerations assign no premium to the conservation or build-up of vital nuclear fuel while it is being burned there would be such a premium on merchant ships. This would stimulate "dual-purpose" type reactor systems wherein new fuel would be bred simultaneously with the development of propulsion power. Though these systems would result in larger reactor plants, the increased space requirements would be of no major consequence on merchant ships. The earlier submarine reactor is designated as STR—"T" for thermal. This means that the neutrons which trigger new fission events have been slowed down—or moderated—to thermal energies. By moderating the neutrons, their probability of interaction with fissionable nuclei of the uranium fuel is substantially increased. Ordinary water (highly purified) is used both as the neutron moderator and the reactor coolant. To maintain the moderating process and simultaneously attain the high coolant velocity necessary for efficient heat transfer from the reactor core, a relatively low water-temperature rise results. The net effect is that the STR produces only *saturated* steam—at 250 lb. per sq. in., 415 deg. F. Steam in this condition would never be acceptable on modern merchant ships, where the boiler and turbine technology is too well advanced, and where systems are standardized in the neighbourhood of 650 lb. per sq. in., 900 deg. F. The SIR ("I" for intermediate) submarine is not so limited in steam conditions. Because intermediate-energy neutrons fission new events, liquid sodium is used as moderator and coolant. The boiling point of sodium is 1,600 deg. F. at atmosphere. Hence, its temperature rise in the reactor core can be allowed to go higher than that of water, subsequently to produce superheated steam—at 500 lb. per sq. in., 750 deg. F. Although this steam is still outmoded by present-day practice, there are other disadvantages of SIR for merchant ships. Sodium burns in air and reacts violently with water, thus requiring extra provisions for handling and leak prevention. Also, radio-active sodium is more hazardous, and its decay is considerably longer than radio-active water. This means that the use of sodium as a moderator-coolant imposes long waiting periods before maintenance and repairs can be made.—*The Motor Ship, May 1955; Vol. 36, p. 53.*

Ship's Speed Measurement with Decca Navigation

Experiments carried out with the Holland-America Line vessels *Arnedyk* and *Diemerdyk* have shown that the Decca method of determining speeds has considerable advantages and that the results are more than accurate enough. Measurements of speed during the night are inadvisable. Certain precautions have to be taken; the accuracy required cannot be obtained with the instruments which are normally available on board ship. It is essential for all methods of measuring speeds that unfavourable conditions (strongly varying current and wind) should be avoided. If this requirement is neglected there will be a risk that by the method of the mean of means the disturbing effects will not be completely eliminated from the final results. It should be left to the staff of Ship Model Basins, of ships' owners and shipbuilders to decide whether Decca can be definitely adopted as a method measuring speed without any further experiments than those mentioned in this article. It may be considered desirable to conduct some additional experiments, results of which might then be analysed by using the method of the least squares as only this method produces reliable data as to accuracy. It is intended in the future to employ only the plotting method for practical use, as it is a much quicker one. Measurements carried out during the trials of newly built ships would prove to be of great value if performed with Decca as well as on the measured mile. During the trials enough time will generally be available for these measurements.—*J. Th. Verstelle and A. Wepster, International Shipbuilding Progress, No. 6, 1955; Vol. 2, pp. 53-60.*

New N.P.L. Tank

The 550 feet long Yarrow Tank at Teddington is the present main research facility available to the Ship Division of the National Physical Laboratory. This is 30 feet broad by 12 feet deep with a carriage speed up to about 20 feet per sec. and has given valuable service both to the industry and to research since it was opened in 1911. It is, however, inadequate for a number of problems which have to be considered to-day in connexion with ship design. The belief widely held in some quarters, that the day of ships is passing can hardly be further from the truth, even from the passenger point of view, while ships are the only answer for carrying heavy freight in bulk quantities. It is clear that shipbuilding and ship-owning will remain industries of national importance both in peace and war, and it is essential that Britain keeps in front in these matters. Merchant ship speeds have increased steadily over the years and with further developments and knowledge in power generation, this trend is likely to continue. Attention is turning more and more to the behaviour of ships in a seaway and to means of making ships more seakindly. A new laboratory has been planned which will enable that research, which had outgrown the existing facilities, to be continued into the foreseeable future. The facilities will include a tank for ship model research, 1,300 feet long on the main waterway, 48 feet broad and 25 feet deep, spanned by a carriage having a top speed in excess of 50 feet per sec., and fine control down to one foot per sec. A feature of the carriage will be the clear platform provided all round and the portable centre beam carrying all the dynamometer gear. This tank will be provided with a wavemaker to simulate a wide range of head and following sea conditions. Ground is available to extend the length of this tank to 2,000 feet if it is ever found necessary. The same building will accommodate a steering and storage basin, 140 feet by 100 feet by 8 feet deep, in which steering and manœuvring experiments will be carried out using 8 feet long radio-controlled models.—*The Marine Engineer and Naval Architect, April 1955; Vol. 78, pp. 139-140.*

Diesel Tug for Middle East

A twin-screw Diesel tug, equipped for the handling of large sea-going oil tankers as well as for buoy tendering, has been built by Richard Dunston, Ltd., Thorne, for service in the Middle East. This vessel, the *Jamoosa*, was built at the Hessle shipyard to the order of Petroleum Development (Qatar), Ltd. Electrically-driven deck machinery is fitted with motors below deck, and comprises a windlass with warping drums and a winch mounted on the main deck under the wheelhouse for use with the two goalpost masts. The latter are fitted with interchangeable 6- or 12-tons S.W.L. steel derricks when buoy tendering. There is a clear space of 20 feet arranged forward of the winch for buoy repair work, and a large salvage hold is arranged below. Two 20-ton tow hooks are attached to the after end of the long bridge. The principal particulars of the *Jamoosa* are as follows:—

Length b.p.	120 feet
Breadth moulded	31 feet
Depth moulded	14ft. 6in.
Total power, b.h.p.	1,520
Static pull, tons	19
Bunkers, tons	90

The *Jamoosa* has been constructed to Lloyd's Register of Shipping and Ministry of Transport requirements, and is subdivided by four main watertight bulkheads. The double bottom in way of the engine room is arranged for fuel oil. Two aluminium lifeboats are carried on the boat deck, the port one being propelled by a 7/9 b.h.p. Diesel engine. The propelling machinery was supplied by the National Gas and Oil Engine Co., Ltd., and comprise two type F4AM8 Diesel engines coupled to 3 : 1 ratio oil operated reverse reduction gearboxes. The engines are fresh water cooled through heat exchangers. The large generating capacity required to handle the heavy electrical load consists of three 45-kW 220-volts d.c. generators each driven by a fresh water cooled National type M4A4 Diesel engine developing 83 b.h.p. The generators are arranged for operation in parallel. Other auxiliaries comprise a 7½-b.h.p. Diesel air compressor; two bilge and general service pumps arranged for either service in the engine circulating system; Diesel oil transfer pump and fresh water pump, all electrically driven.—*The Shipping World, 4th May 1955; Vol. 132, p. 497.*

German Trawler with Variable Pitch Propeller

A motor trawler has recently been completed by the Rickmers Werft, Bremerhaven, for the Gemeinwirtschaftliche Hochseefischerei G.m.b.H. The vessel is named *Gustav Dahrendorff* and has an overall length of 195 feet 10 inches. The breadth is 28 feet 10 inches and the draught 13 feet 7 inches. A Deutz four-stroke pressure-charged engine is installed, the output being 1,000 b.h.p. at 200 r.p.m. and the shaft drives a Liaaen-type variable-pitch propeller. The trawl winch is electrically-operated and is designed for 1,400 fathoms of trawl warp. Current is supplied by a 200-kW Diesel generator of the A.E.G. type, an 80-kW dynamo-compressor set being also fitted. An unusual arrangement is the provision of a second 200-kW machine driven from the propeller shaft, and used as a motor in the event of engine failure. Should this occur, current is taken from the large Diesel generator and the screw can be turned at a reduced speed. Exhaust gas from the main engine is passed through a waste-heat boiler and, in addition, there is an oil-fired boiler. Fresh water cooling is adopted for the main and auxiliary engine systems. A.E.G. electro-hydraulic steering gear is installed, together with an automatic helmsman. Numerous up-to-date devices have been provided for navigation as well as for trawling operations. The equipment includes an Anschutz gyro compass and a bearing repeater, three different types of echo-sounder (one a Fischlupe device used for the detection of shoals of fish), radar, a direction finder, two radio transmitters and receivers and a loud-speaking installation connecting all parts of the ship. Approximately 150 tons of fish can be carried in the hold, and there is a fish-meal plant capable of dealing with 15 tons of raw material daily.—*The Motor Ship, May 1955; Vol. 36, p. 69.*

Cathodic Protection of Salt Water Systems

Maintenance costs, due to the corrosion and scale formation in ships' salt water sanitary systems can become a serious problem to the shipowner, and in the case of the Canadian

Pacific Railway Company have involved the company in an annual expenditure of thousands of dollars. A method of protection, known as the Guldager electrolytic system, was tried out in 1951 in the *PrincessKathleen*, owned by the C.P.R., and after a few months it was found that not only had corrosion been arrested in the sanitary system, but pipes which had been heavily covered in scale were being cleared rapidly. So satisfactory were these results that the owners decided to install a similar system in the *Princess of Vancouver*, now building at Alexander Stephen and Sons, Ltd. This is being carried out by the Cathodic Protection Co., Ltd., London, the firm responsible for the system. The principle on which the Guldager system works is that when electrolytic action occurs, it is the anode which is attacked, while an anti-corrosive film is deposited on the cathode. An aluminium anode is placed in the water supply tank, and is fixed in position by means of specially designed steel bearers, bolts and insulators, the tank forming the cathode. In some installations a special reaction tank is built on to house the anode. The rest of the equipment consists of transformer, rectifier, rheostat, voltmeter and ammeter. When the electricity supply is switched on, the aluminium anode is attacked and a thin film of calcium hydroxide is deposited on the cathode. This film acts as a protective coat which will, providing that the electrical contact between pipes and tank is good, be formed throughout the system. An important feature resulting from the electrolytic process is the generation of aluminium hydroxide which, being a highly gelatinous substance, will clean the water of any colloidal matter or other impurities.—*The Shipping World, 20th April 1955; Vol. 132, p. 445.*

International Internal Combustion Engine Congress in Holland

The 1955 Conference of the International Internal Combustion Engine Association held this year at Scheveningen was opened on 23rd May and ended on 28th May. A summary of some of the papers read is given below.

"Two-stroke Diesel Engines with B. and W. Turbocharging System for Outputs Above 10,000 b.h.p. per Shaft . Experience with the First Plants of this Type", by Soren Hansen, chief design engineer, Burmeister and Wain, Copenhagen.

The author states that at the present stage of development, an output of up to 17,500 b.h.p. on a single shaft can be obtained with the B. and W. turbocharging system applied to the direct coupled, single-acting, turbocharged, two-stroke B. and W. engines, which for many years have been extensively used as ships' propulsion engines. In October 1954, there were already seven installations up to 10,000 b.h.p. of this engine type in service and twenty-two on order. The experience from service with these installations, most of which are run on boiler oil, has proved satisfactory, apart from some slight troubles in connexion with the turbochargers, which have involved little inconvenience to the ships in question. Maintenance costs seem, at a given output, to become lower than for plants without turbocharging, and the cylinder wear does not seem to have increased.

"Development and Service Results of a High Powered Turbocharged Two-Cycle Marine Diesel Engine", by F. G. van Asperen, chief engineer, marine diesel department, Werkspoor N.V., Amsterdam.

The paper contains four chapters, in the first of which is given an account of the technical development during the last ten years of a high powered single-acting two-stroke marine Diesel engine with uniflow scavenging and turbocharging, especially concerning the turbocharging system. Chapter II discloses experience and results obtained in service on Diesel fuel and heavy fuel, followed by conclusions about possible general trends in future. The last two chapters deal with two economic items, closely connected to the main subject. Firstly, the total efficiency of the engine, including the heat recovered from an exhaust-gas boiler placed in series with the turbochargers, is analysed. Finally, maintenance costs concerning cylinder liner replacement are worked out in graphical schemes as a function of liner wear rates and local price conditions, for both cast-iron and chromium-plated liners.

"Supercharging by means of Turbo-blowers Applied to two-stroke Diesel Engines of Large Output, e.g. more than 10,000 b.h.p., for Ships' Propulsion", by G. Wieberdink and A. Hootsen of Koninklijke Machinefabriek Gebr. Stork and Co., N.V., Hengelo.

The application of supercharging by means of turbo-blowers offers the possibility of building a single-acting two-stroke engine of normal size with an output of well over 1,000 b.h.p. per cylinder with the advantage that the turbo-blowers are able to deliver a sufficient quantity of air at any load and speed and also at starting. This makes the normal scavenging pump completely superfluous so that an output of over 10,000 b.h.p. may be developed in eight or more normal single-acting cylinders without extra length to the engine for scavenging. In addition, with this system a better efficiency is obtained.

"High Powered 'UEC' Marine Diesel Engine", by Hideo Fujita, assistant manager of the machinery designing department and chief Diesel engine designer of the Mitsubishi Shipbuilding and Engineering Co., Ltd., Tokyo.

The author's company has recently succeeded in developing a new type of high powered two-cycle single-acting Diesel engine with exhaust gas turbocharger, bearing the trade name "UEC" Diesel engine. The design of this engine is entirely the company's own; it has been completed after many years of exhaustive research. Several 12,000 b.h.p. and 8,000 b.h.p. engines of this type are under construction at the works for installation in high speed cargo vessels. Dealt with in this paper are developments made in the designing of the new engine, including some of experimental engine trial results. The particulars and special features of the engine are also illustrated.

"Development of the Opposed Piston Engine to 10,000 h.p. and Over", by P. Jackson, manager of research and development, William Doxford and Sons, Ltd., Sunderland.

This paper is concerned primarily with the development of the Doxford opposed-piston engine to higher powers. The normal power of the largest engine being built today, 750 mm. bore by 2,500 mm. combined stroke, is 9,000 s.h.p. in six cylinders for continuous service at sea, with an m.i.p. of 90lb. per sq. in. (6·35 atm.) and average piston speed of 900ft. per minute. Research and development work on experimental engines has shown how these ratings can be increased to 105 m.i.p. at an average piston speed of 1,000ft. per minute, which, when applied to the largest engine, give a possible continuous power of over 11,000 s.h.p. A seven-cylinder engine would give nearly 13,000 s.h.p. Engines have been operated at these ratings for months continuously on heavy boiler fuel. Performance curves, temperatures and pressures are given. An experimental three-cylinder engine of 600 mm. bore and 2,000 mm. stroke has been built and supercharged by a Brown Boveri turbo-blower. Development of this engine has been achieved to 130 m.i.p., giving possibilities of engine power on the largest engines of up to 15,000 s.h.p. on a single shaft. Performance data and characteristics of the supercharged engine, are given, together with comparisons of the normal engine with increased m.i.p. and of the supercharged engine. A reduction in engine room space is effected. In conclusion the author states that powers of up to 15,000 s.h.p. in six cylinders will be undertaken on a single shaft, with a big saving in fuel consumption and simpler engine room than at present obtained with steam turbine machinery, and also a much more simple layout and more economical performance than is possible with geared engine arrangements.

"Tentatives, Realizations and Outlook for Marine Engines of more than 10,000 b.h.p. Shaft Output", by M. Zwicky, chief engineer of Sulzer Brothers, Ltd., Wintherthur.

The author relates how during the course of the last fifty years of Diesel engine development, several attempts have been made to build big engine units with large shaft output. The setbacks which occurred and the lack of interest, as well as, perhaps, of confidence, of the shipowners were the main reasons that very few engines with a shaft output of over 10,000 b.h.p.

were built, in spite of the fact that the results of the few units built were very satisfactory. The development of supercharged engines opens up again a new outlook in this field. It is the author's opinion that it appears to be in the interests of shipowners to consider more often the application of big Diesel engines with large shaft output.

"Marine and Stationary Application of the Free-piston Generator with Gas Turbine of more than 10,000 h.p.", by R. Huber, technical director of Societe d'Etudes Mecaniques et Energetiques.

The Societe d'Etudes Mecaniques et Energetiques (S.E.M.E.) has been working for many years on the development of the free-piston generator, started more than twenty years ago by Bureau Technique Pescara. A paper on this new type of internal combustion engine was read at the International Internal Combustion Engine Congress in 1951. Since then, numerous marine and stationary applications have proved that sets composed of free-piston generators and gas turbines give entire satisfaction in industrial service. Almost 100,000 h.p. are in service or under construction. Until now the installations carried out have had relatively small outputs as they consist only of one or two generators per turbine. The present paper treats the application of free-piston generators for installations of more than 10,000 h.p. and shows, both for stationary and marine purposes, the possibilities of installation, ways of regulation and performances to expect. A detailed description is given on the conditions of reversing a marine turbine fed by free-piston generators and fitted with an astern running stage.

"Direct Drive or Geared Diesel Machinery for High Outputs", by Dr. Schmidt of M.A.N., Augsburg.

This paper deals with the conception of high output, as concerns technical development and actual operation. It gives reasons for the increase of power on a single shaft; and an explanation of the different methods employed by the Diesel engine builders. Low-speed, high powered Diesel engines for direct coupling with the propeller shaft and multi-engine installations with gear or electric transmission are discussed, also the different types of engines in use for these installations. A comparison is made of single-engine plants and geared installations with regard to space requirements, weight, first costs, economy, reliability and general performance. Finally, the competition between steam turbine plants and Diesel geared installations as well as Diesel-electric plants is discussed.

"Comparative Analysis of Single-Screw Diesel and Steam Turbine Propelling Plants Developing 10,000 s.h.p. and Over", by MM. Sinobad and Rouet, chief engineer and engineer respectively of Ateliers et Chantiers de Bretagne, Nantes.

With the 10,000 s.h.p. range as a target, Diesel engine builders are now working on designs capable of competing with steam turbines. One of these designs consists of using several Diesel engines to turn a single propeller through a reduction gear and hydraulic coupling. The above arrangement forms the subject of this paper, in which it is compared with a steam turbine machinery of the same horsepower. Dependability, specific fuel consumption, maintenance and initial cost of Diesel and steam plants have been the chief concerns of the authors.

"Twenty-five Years of Actual Operation at Sea of Two Big Passenger Motorships", by Dr. Fontana of Cantieri Riuniti dell' Adriatico, Fabbrica Macchine S. Andrea, Trieste, and S. Filippini of Fiat Stabilimento Grandi Motori, Turin.

The twin passenger liners *Saturnia* and *Vulcania* were the first motorships to be equipped with main engines of more than 10,000 s.h.p. output. Considering that this was a pioneering effort and that these vessels are still—after twenty-five years of continuous running—maintaining a regular service, Fiat and C.R.D.A. thought it would be of interest to publish the main particulars of their construction and operation. The first section of the paper, by C.R.D.A., covers the description of the old propelling machinery of both ships and of the modern machinery that replaced it in 1935-1936. This description is followed by an analysis of the operational data collected during that long period. The second section, by Fiat, covers particularly the propelling machinery of the *Vulcania* and shows its construction features and the results of operation at sea; reference is also made to the experience gained from engines of the same class installed afterwards on board other ships.—*The Shipping World, 25th May 1955; Vol. 132, pp. 566, 568.*

Damping of Surface Waves by Wave Absorbers

The need for preventing the reflection from model basin walls of the surface waves generated incidentally by towed models or specifically by a wavemaker is commonly recognized. In the case of the models, the reflected waves represent conditions not encountered in full-scale operation and are, therefore, unwanted. In the case of wavemaker operation, the reflected waves cause an undesirable and a difficult-to-control non-uniformity in the amplitude of the wave train produced by the wavemaker. The reflection of surface waves is also a matter for concern outside the laboratory, such as in harbour studies or breakwater design. The few reports that are available on the subject of evaluating wave absorbing devices employ widely different techniques. In a report by Thews, one particular wave absorber was chosen as a standard and other wave absorbers were compared with it by making visual observations of the relative sizes of the waves reflected from the test absorbers with respect to the standard. This method permits the detection only of gross differences in absorption properties. The method used by Caldwell was to measure the amplitude of a solitary wave before and after reflection from a wave absorber. One objection to this scheme lies in the lack of assurance that an absorber will exhibit similar damping for a solitary wave and for a train of progressive waves. Another difficulty introduced by the solitary wave method is that the reflected wave is not actually a solitary wave of reduced amplitude; rather the reflected wave energy appears in the form of a train of waves of decaying amplitude. This fact makes extremely difficult the necessary task of comparing the incident and reflected wave energies. The above-mentioned shortcomings can be obviated by making direct measurements with progressive waves. The procedure generally used is to make measurements of an initially uniform train of waves after some of the wave energy has been reflected by the wave damper, but before further wave superposition occurs as a result of subsequent wave reflection by the wavemaker itself. The procedure for separating the amplitude of the reflected wave from that of the incident wave in order to determine the absorptive properties of the wave damper has been detailed by Keulegan. This last method also encounters obstacles in practice. As a wavemaker is put into operation, a number of waves are produced which vary in size and which precede the subsequent uniform waves down the basin. Yet the measurements desired are of the uniform wave train superimposed on the first reflection of this uniform train; i.e. it is required that the first reflection of the non-uniform wave has already passed the point of measurement and that the reflection of these waves from the wavemaker has not yet reached this point. If this state of affairs ever does exist, it is difficult to perceive the period in which it obtains. In this paper, it is proposed to remove the uncertainty attached to the time at which measurements are to be made and to transcend the possibility of "contamination" from the initial waves by making measurements in the channel only after the wavemaker has been in operation a considerable period of time and an equilibrium has been established between the uniform train of waves from the wavemaker and the continued reflections of these waves. Losses of surface wave energy in the basin other than to the wave absorber are considered. It is demonstrated that an equilibrium is expressed in terms of its component parts. A type of wave motion is constructed which yields the desired energy decay, and the manner in which these waves superpose is examined. Finally, the relation needed to determine the absorption coefficient of a given wave absorber from certain wave measurements is given.—*H. R. Reiss, Navy Department, David W. Taylor Model Basin, Report 896, March 1955.*

Cathodic Protection of Aluminium

The use of aluminium alloys for ship construction presents difficulties that must be overcome before its application to large ships becomes a practical proposition. When an aluminium alloy is electrically connected to a more noble metal and immersed in a good electrolyte, such as seawater, the aluminium corrodes at a rate which varies with the electrochemical potential and polarization characteristics of the coupled metals. In large ships it is almost essential to fabricate some of the wetted components, such as shafts and propellers, from non-aluminium materials, and it is difficult to ensure that these components are electrically insulated from the remainder of the hull. In view of this, a series of cathodic-protection trials were made with 65S-T aluminium alloy. From the results of these trials, it appears that aluminium alloy can be protected by electrical means, and that a potential of 1,000 mV., or slightly less with respect to a silver-silver-chloride electrode in seawater, will reduce the corrosion rate to a small value. The possibility of cathodic corrosion by "overprotection" does exist, that the range over which fair protection is obtained is not extremely narrow. The author concludes with the observation that, having regard to the encouraging results obtained, the work on cathodic protection of aluminium alloys should be continued with the object of fixing more definitely the best potential for the purpose.—*G. L. Christie, Naval Research Establishment, Canada, Report No. PHx-85. British Shipbuilding Research Association Journal, March 1955; Vol. 10. Abstract No. 10046.*

Information from Electric Currents in the Sea

Waves, tides and ocean currents might not be expected to produce electric currents, but by moving water, as a conductor, through the earth's magnetic field they do in fact produce measurable voltages and flows of electricity. Young, Gerrard and Jevons, who made the first detailed study of this phenomenon, thought it conceivable that measurements of electric potential gradient might be used to estimate the drift of a ship at sea, and Von Arx showed how indications from electrodes towed behind the research vessel *Atlantis* were used to keep her on a direct course from Bermuda to Woods Hole, across the Gulf Stream. In recent years the method has been used extensively in studies of ocean currents and more ships might use it if the necessary equipment were readily available. In the apparatus designed by Von Arx, recording panels, including all that is useful as well as essential, are housed in a strong, compact steel cabinet measuring approximately 4½ by 2 by 1½ feet, easily shipped and secured. The electrodes are carefully streamlined and can be towed at eight knots.—*G. E. Deacon, The Journal of the Institute of Navigation, April 1955; Vol. 8, pp. 117-120.*

Rudder Preservative Pumping Unit

A rudder pumping unit has been designed which allows rust preservative compound (RPC) to be pumped into and out of ship rudders. The method previously used depended entirely on gravity. To fill the rudder, drums containing the preservative compound were placed on the deck of the ship and poured down into the rudder by means of a length of hose connecting the drums to the rudder. In order to drain the rudder the drums then had to be shifted to the bottom of the dock directly beneath the rudder and the preservative compound drained off into the drums. The new rudder pumping unit has a gear-driven pump. The intake hose is placed in the drum, the exhaust hose is connected to the bottom of the rudder, and the RPC is forced into the rudder until it is full. The pump is then reversed to pump the RPC back into the drums for further use. Since adoption by the Mare Island Naval Shipyard of the rudder pumping unit the time required to fill and empty a ship's rudder has been materially reduced. In addition, there is no longer any waste of the preservative compound through spilling, and the working conditions have become cleaner.—*Bureau of Ships Journal, March 1955; Vol. 3, p. 37.*

Reversible Bending of Turbine Shafts with Temperature

It has been known for some time that some steam turbine shafts can bend when heated. Three types of bending have been recognized. The first is transient and occurs only when the temperature of the shaft is changing, the deflexion being zero when the shaft is at a steady temperature. This can be accounted for by emissivity variations around the surface of the shaft and lack of black body conditions. The second type of bending occurs with new shafts which, on heating up, deflect permanently, this deflexion remaining on cooling to room temperature. This is due to relief of residual stresses which are not axially symmetric, and on subsequent heating no further deflexion occurs. In the third type of bending, the shaft deflexion is proportional to the temperature of the shaft when this is steady. Thus the shaft bends when hot and is straight when cold. The latter type of deflexion is considered in this paper. It has been suggested that this type of deflexion could be explained by chemical segregation which is not axially symmetric and which would result in variations in expansion coefficient in the same pattern as the segregation. Tests have been carried out to verify this. Carbon steel shafts of the dimensions normally used for built-up turbine rotors were deliberately produced so that the position of the chemical axis of the shaft was a significant distance from the geometrical axis. The asymmetry was considered to be greater than would occur in practice. The effect on the thermal stability of the shafts was measured and was either zero or small. Before a shaft is put into service, many turbine manufacturers carry out a "heat indication test" to determine whether the shaft shows any deflexion on heating. The heat indication test equipment used in this investigation consisted of a horizontal gas-fired furnace which heated the body of the shaft, the journals being outside the furnace, supported on bearings and air cooled while the shaft was rotated at low speed. The deflexion of the shaft is measured by a series of dial-gauges and push-rods inserted horizontally through the side of the furnace. Heat indication experiments have shown that some carbon-molybdenum steel shafts can bend when hot and become straight again when cold. This effect in the case of two shafts examined has been shown to be due to a difference in expansion coefficient in the steel on opposite sides of the shafts, and that this difference in expansion coefficient is not caused by chemical macro-segregation but is a result of non-uniformity of heat treatment; in this case, in normalizing and austenitizing prior to normalizing. A change in expansion coefficient of the correct order of magnitude has been produced in two different casts of carbon-molybdenum steel, one having a higher nickel content than the other. The change in expansion coefficient in a given specimen has been shown to be reversible by using the appropriate heat treatment, and a shaft that was originally defective has been corrected by re-treating in such a manner that all parts of the body of the shaft transform in the same manner.—*Paper by A. Barker and F. W. Jones, read at a General Meeting of The Institution of Mechanical Engineers on 6th May 1955.*

De-icing by Gas Turbine Pump

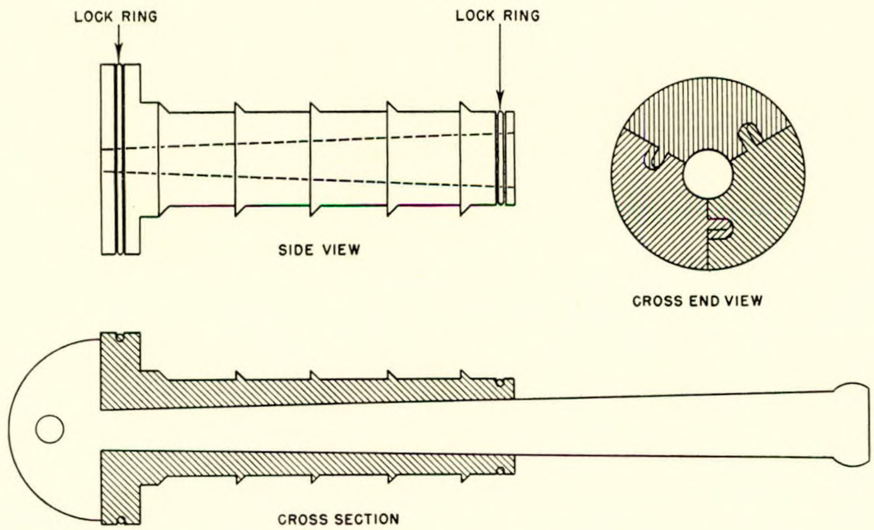
The basic gas turbine is the simplest of all heat engines, but its evaluation as a heater has been postponed during investigation of basic functional problems of this turbine. Now that much progress has been made in the gas turbine field, investigators have begun to cast off convention and are teaching themselves some of the entirely new ways of thinking that are sometimes necessary if satisfactory answers are to be found. With the object of discovering suitable uses for gas turbine exhaust heat, the T-45 gas-turbine-powered fire pump is expected to be tested in arctic regions as a de-icer as well as a fire pump. The reason for evaluating a small gas turbine with this function is that temperatures and transfer coefficients of high-velocity gases at the rim of the turbine are of the same order of magnitude regardless of size. Anticipations based on previous de-icing tests are that turbine exhaust gases can be directed against the metal or iced-structure, loosening the ice. The hose stream can then be played on the area, removing or knocking the ice loose. Previous de-icing tests, in which a salt water hose was used in conjunction with the exhaust gases, proved conclusively that the time required to de-ice an area can be reduced by approximately one-third if high-pressure salt water is used to remove the loosened ice. However, this de-icing test did not utilize the facilities of the T-45 gas-turbine fire pump and did not prove that the ice can be removed faster by this method than it can by men beating the iced surfaces with axe handles. Nevertheless, hot air de-icing will not be as detrimental to the equipment as the axe-handle method sometimes is. By simple redesign of the basic arrangement, the T-45 gas turbine fire pump can be adapted for de-icing purposes so as to utilize its exhaust, a constant 2·4lb. per second flow of hot combustion gases at approximately 1,030 deg. F. Power source of the T-45 is the turbine assembly which operates at a speed of 40,300 r.p.m. on ordinary Diesel fuel, and connects with a reduction gear to reduce this speed to an output of 4,500 r.p.m. A unique feature of the T-45 is its simple and reliable method of starting. It is the first gas turbine ever to be cranked by hand. Only forty-five seconds of cranking by one operator, or, if more convenient, by two, is required to bring the unit up to full pump output.—*D. E. Blackwood, Bureau of Ships Journal, March 1955; Vol. 3, pp. 7-9.*

Cargo Vessel for West African Service

At the Linthouse, Glasgow, shipyard of Alexander Stephen and Sons, Ltd., was recently completed the m.s. *Fearnvalley,* an open-shelterdecker carrying about 6,530 tons d.w. on a draught of 23 feet and having a gross register of 4,550 tons. Owned by Fearnley and Eger, Oslo, this vessel will be employed in the joint West-African service maintained by this company, the Bergen Steamship Co., and Rederi A/B Transatlantic. Although completed as an open-shelterdecker, the scantlings are for a closed-shelterdecker with a corresponding increase in draught and deadweight. The ship has been built to the highest class of Det Norske Veritas and complies with current Safety of Life at Sea Regulations. Welded units of up to 25 tons have been prefabricated before being placed on the building berth. The hull is divided by seven watertight bulkheads into five cargo compartments, the holds having an upper 'tweendeck. In addition, No. 1 hold has a lower 'tweendeck and, in the lower part of No. 3 hold are palm oil tanks which can be used for general cargo as required. Part of the main deck amidships is appropriated for the carriage of 5,000 cu. ft. of refrigerated cargo. Mechanical ventilation of the cargo spaces, with individual vents to holds and 'tweendecks, is fitted. The fire-detection and fire-extinguishing arrangements throughout comply with the requirements of the New York underwriters, the detecting cabinet being fitted on the navigating bridge. The usual automatic alarms are also installed. All deck machinery is electrically driven, this including the sixteen 5-ton winches operating the 5- and 10-ton derricks at the five hatches. There is also a 30-ton derrick at No. 2 hatch. Four aluminium lifeboats are carried, one of them motor driven and, on top of the poop house, there is a small motor tow-boat and a dinghy for harbour use. The Stephen-Doxford main engine has four cylinders with a bore of 670 mm. and a combined piston stroke of 2,320 mm., the service output being 4,400 b.h.p. at 115 r.p.m. To enable residual oil to be used as fuel, the now customary fuel purifying and heating arrangements have been made, this including provision for fuel circulation during standby periods. Diesel oil can, of course, readily be brought into use.—*The Motor Ship, May 1955; Vol. 36, pp. 76-77.*

New Design for Boiler Tube Puller

A boiler tube puller has been designed with which a workman can remove a tube from headers and drums without having to cut the tube from the tube sheet. At present, it is necessary to cut or rip the boiler tube from the tube sheet in order to remove the tube from the boiler. The tube puller now in general use does not work well in most instances because of confined working areas. The procedure recommended for using the puller is as follows:—(*a*) insert the puller into the cut end of the tube; (*b*) attach the tube puller and, using a chain fall, pull the tube free from the tube sheet; and (*c*) remove the loosened tube from the boiler by hand or, if necessary, with the puller. The proposed puller does not slip because the harder the operator pulls, the harder its teeth are forced against the inside surfaces of the tube by the wedging action of the tapered pull pin. The accompanying drawing of the suggested tool does not show dimensions because each size of tube requires a puller that will fit its particular diameter.—*Bureau of Ships Journal, March 1955; Vol. 3, p. 46.*



Water and Steam Velocity in Boiler Tubing

The purpose of this investigation was to establish a sound basis for calculating the water circulation in natural-circulation boilers. The work is based on published literature, and on original experiments with a full-size experimental boiler; in the experiments new measuring methods were used to obtain the mass flow, the distribution of density, and the velocities of the water and steam in both the vertical and the horizontal portions of the tubes. Hitherto, when calculating the water circulation, the velocity of steam had been based on values established for stationary water, and on friction coefficients as known for homogeneous liquids. The new values now obtained for the circulating water-steam mixtures show substantial deviations from values accepted so far; it is thought, therefore, that the latter will have to be revised.—*K. Scharz, VDI Forschungsheft, No. 445, 1954. The British Shipbuilding Research Association Journal, 1955; Vol. 10, No. 3, Abstract No. 10,004.*

Ship Slamming in Head Seas

The most important problem facing naval architects is probably the extension of speed range. It has been suggested that the rough sea corresponds to the sonic barrier of aerodynamics. The maximum speed of a surface vessel is determined not by its power but by its behaviour in a seaway. Increasing the power, making propulsion more efficient, or reducing the still-water resistance will have but small effect on the maximum speed attainable in a seaway. It is a well-established experimental result that fast ships suffer less speed reduction in waves than do slow ships. This type of speed reduction, however, is very often of academic interest only. Voluntary speed reduction (obtained by r.p.m. reduction in order to save the ship) is far more important. In spite of the available power, the speed of advance must be reduced if violent motions are to be avoided. The two types of speed reduction mentioned are shown in Fig. 1 for a Victory ship operating in the North Atlantic in winter. If a larger engine is built into the existing ship, the "academic" speed-reduction curve (speed reduction caused by the waves) may be raised somewhat, but the gap between this curve and the actual speed-reduction curve will be increased. This shows that increasing the power, rather than solving the problem of extending the speed range for the same power, might result in slow ships with large engines. The most violent attack of the sea on a vessel is probably the heavy blow delivered by the waves on the re-entering bow. The ship vibrates for some time after such an impact, and plates under the forefoot may be damaged. Altering course and reducing speed are generally recommended as remedies, which means that the fight against slamming is of a defensive character. If speed and course are maintained in rough seas, a drydocking and repair bill due to slamming may very well result. Ships also slam in hove-to condition, which would indicate that there is really no escape from slamming. The observation might be made that nature seems to try to prevent slamming. Fast ships with less speed loss generally have finer lines than slow cargo ships. Slamming danger is large if the lines are full and the speed high. Since full ships suffer more speed reduction than fine ships, the slamming danger is automatically reduced. Another approach to the solution of avoiding damage is to strengthen the bottom around the bow. The difficulty, however, is that little design information is available with respect to pressure distribution on the bottom of a slamming ship. Over-design is expensive and valuable cargo space is lost. There are many assumptions made in theoretical seaworthiness work which are over-simplified and not totally acceptable when studying slamming. These assumptions are generally made because the actual physical picture either is not clear cut or is too complicated. This report consists of three parts. The first part gives the physical picture, the second analyses the conditions leading to slamming, and the third describes a method for predicting slamming accelerations and pressure distributions. The appendices give mathematical details, computational procedures, and elaboration on some aspects of the discussion presented in the text.—*V. G. Szebehely and M. A. Todd, Navy Department David W. Taylor Model Basin, Report No. 913, February 1955.*

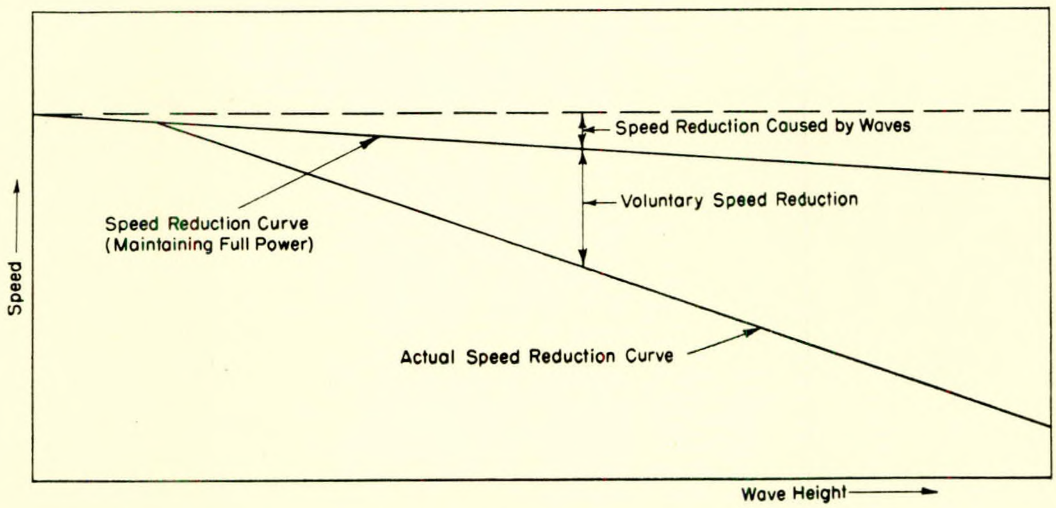


FIG. 1—*Speed reductions for a Victory ship*

Aluminium Hatch Boards

The recently introduced aluminium hatch board, which will last the lifetime of a ship, approved by Lloyd's Register of Shipping and the Ministry of Transport, is one of the innovations in deck equipment made possible by the introduction of all-welded aluminium construction. The *Kentwood,* a France Fenwick collier, was fitted with six aluminium hatch boards in July 1954, since which time she has encountered some heavy weather and has handled twenty-two cargoes. Recent inspection showed the aluminium covers to be in excellent condition. Members of the ship's crew expressed their preference for the aluminium board mainly on the grounds of its unvarying weight of 47½lb.—about one-third the weight of a wet timber board—and the ease of handling. Measuring 6 feet long, 18 inches wide and 3 inches deep, each hatch board is fabricated from two sheets of Noral 54SO alloy, formed to a basically rectangular cross-section, but with additional rigidity provided by a full-depth longitudinal trough formed on the underside. The bottom of the trough is secured by argon-arc plug welds to the underside of the top sheet, and the ends of the board are closed by means of rectangular pieces of 54SO sheet, similarly welded into position. Longitudinal welded seams join the sheets forming the bulk of the shell, and a recessed grip for handling is welded into the top at each end. This form of construction makes the finished hatch board a closed and watertight vessel with two compartments, and ensures buoyancy. Rigorous tests have shown that the board can withstand all the loads and rough usage likely to be encountered in service, despite its lightness, which reduces handling problems to a minimum. In one test, it was called on to show a deflexion of no more than 0·03 inch under an evenly distributed load of 8 cwt. per sq. ft. while simply supported at each end. It was later found that the board was able to withstand a maximum sustained load, after permanent set had ocurred, of 16 cwt. per sq. ft. Dropping the board on to concrete from a height of 10 feet caused no damage. As a final test, to ascertain its resistance to rough treatment, steel weights of various shapes weighing up to 40lb. were dropped on it from heights up to 10 feet. After this severe treatment it was still able to withstand the test load of 8 cwt. per sq. ft.—*The Shipping World, 1st June 1955; Vol. 132, p. 597.*

Free-piston Gas Turbine Installations of More than 10,000 b.h.p.

The following details refer to an installation of 12,000 b.h.p. on board a 31,000-ton tanker. It comprises twelve standard G.S.34 free-piston generators and two gas turbines coupled through gearing to a common shaft. Each turbine has five ahead stages and one astern on the same rotor. The turbines run at 4,500 r.p.m., corresponding to 102 r.p.m. on the shaft, the drive being taken through double-reduction gearing. The following are details of the installation:—

Shaft horse-power	11,400
Efficiency of gearing, per cent ...	96
Losses in gas delivery pipes, per cent	2
Gas pressure, kg. per sq. cm. ...	3
Gas temperature at the entry to turbine, deg. C.	440
Gas temperature at turbine outlet, deg. C.	250
Weight of gas passing through generator, kg. per sec.	3·72
Specific fuel consumption, gr. per b.h.p.-hr.	180

After leaving the gas turbines at a temperature of 250 deg. C., the exhaust is delivered to waste-heat boilers, the exit temperature being 200 deg. C. The heat thus recovered is about 1,500,000 calories per hr. and the steam raised is about 3 tons per hr., which is sufficient to drive a turbine coupled to a 450-kW. generator. The weights of the propelling machinery are:—

	Tons
12 generators at 8 tons ...	96
2 turbines with the head stages	15
Reducing gear	60
Piping and reversing stage ...	14
Total weight	185 =
	15·4 kg. per b.h.p.

Prolonged trials on the test bed have shown that these units in service will operate satisfactorily on Fuel Oil No. 1 or No. 2.—*Paper by R. Huber read at the International Internal Combustion Engine Congress, The Hague, 1955. The Motor Ship, June 1955; Vol. 36, p. 105.*

Peruvian Oil-tank Motorship

Designed and constructed by John I. Thornycroft and Co., Ltd., of Southampton, for the Peruvian Government, the b.a.p. *Sechura* was recently handed over to the Peruvian Navy. The vessel is to operate along the Pacific coastline of Peru. The *Sechura* has been built in accordance with the requirements of Lloyd's Register of Shipping for the classification 100 A.1, for the carriage of petroleum in bulk, and of the International Convention for the Safety of Life at Sea. The construction and equipment of the vessel also comply with the United States Coast Guard Regulations. Before finally deciding on the lines of the vessel, extensive tank tests were carried out with models, and it may be mentioned that the results obtained on trial with the completed ship were better than expected. With a draught corresponding to a displacement of 8,500 tons, a speed of 13·22 knots was attained, to give an Admiralty constant of 404, on a basis of s.h.p. A considerable amount of electric welding has been used in the construction of the hull. Riveting, however, has been employed for the shell seams and for connecting the frames to the shell. The deck is practically all-welded, and is welded to the beams. The principal dimensions and other leading characteristics of the *Sechura* are given below.

Length overall	385ft.
Breadth, moulded	52ft.
Depth, moulded	26ft.
Displacement, tons	8,700
Deadweight, tons	6,000
B.h.p.	2,400
Corresponding r.p.m.	120
Speed on trial, knots	13·25

The main propelling machinery, which has been constructed by Burmeister and Wain, of Copenhagen, consists of a B. and W. single-acting, two-stroke cycle, five-cylinder, crosshead-type Diesel engine. With a cylinder diameter of 620 mm. and a stroke of 1,150 mm., the engine is capable of developing 2,400 b.h.p. at 120 r.p.m. Two four-blade propellers, one working and one spare, and each with a diameter of 14ft. 3in., have been manufactured, to the shipbuilders' design, by J. Stone and Co. (Charlton), Ltd. The propellers, constructed of Stone's bronze, each have a developed area, over the four blades, of 20 sq. ft., and a mean pitch of 11 feet. They each weigh 5¼ tons. Because of the heavy swell in the waters in which the vessel is to operate, special consideration has been given to the mooring arrangements. Provision has been made for six head and six stern lines, of 10-inch manila, to run through roller fairleads on to strong fabricated bollards.—*The Shipbuilder and Marine Engine-Builder, June 1955; Vol. 62, pp. 382-390.*

German Combined Oil and Ore Carrier

The *Bertha Entz,* which recently ran trials, was built at Kieler Howaldtswerke for the Thomas Entz Tanker G.m.b.H., Rendsburg. She was ordered in 1953 as an 18,000-ton motor tanker, but it was afterwards decided to construct the ship as a combined ore- and oil-carrying motor vessel. She is the first motor vessel of this type in the German merchant fleet. The main details are:—

Length overall	607ft. 4in.
Length b.p.	569ft. 1in.
Breadth, moulded	73ft. 10in.
Draught, loaded on summer freeboard	32ft. 2in.
Deadweight capacity, tons ...	21,608
Displacement, metric tons ...	30,410
Speed, loaded, knots	15·7
Machinery, b.h.p.	8,000

The hull has two longitudinal bulkheads. The two ore cargo compartments are amidships, forward and aft of the bridge house respectively, and each comprises four holds; they are provided with MacGregor hatch covers. The oil cargo is carried in eighteen wing tanks and two centre tanks under the bridge house, and these have a capacity of 795,280 cu. ft. All the tanks can be utilized for ballast water and the net tonnage as an ore carrier is only 2,528 tons, the gross register being 15,004 tons. The dry cargo hold under the forecastle has space for 47,780 cu. ft. grain, or 43,542 cu. ft. bales, and is served by two 5-ton derricks. In addition, there are two derricks of similar capacity for handling the cargo hoses when loading or discharging, also a 4-ton derrick and one of 1-ton lifting capacity. There is a pipeline tunnel which serves as a connexion between the fore and after part of the ship for the passage of the crew, on both sides under the deck, leading through the ore cargo holds and the centre oil tanks. With this arrangement the usual bridge and pipelines on deck in a tanker are rendered unnecessary, as these would in any case have been a hindrance to the loading and discharge of the ore through the hatches. The propelling engine is an M.A.N. double-acting two-stroke unit, having eight cylinders 700 mm. bore with a piston stroke of 1,200 mm., the output of 8,000 b.h.p. being developed at 118 r.p.m. The service speed is 15¾ knots with the ship fully loaded.—*The Motor Ship, June 1955; Vol. 36, p. 96.*

Strength of Corrugated Plating for Bulkheads

The behaviour of corrugated plating under lateral load is examined theoretically and experimentally. Tests on panels of steel and aluminium alloy corrugated sheet as beams under simple bending conditions show that, provided instability does not occur, the behaviour is adequately described by the "classical" theories of elastic and inelastic bending, but that in some cases premature failure in bending may occur through local buckling of the compressed plating. A second series of tests was therefore made to investigate the stability of single corrugations in bending, and the results lend support to a theoretical analysis of the problem. It is shown in particular that the shape of the trough has an important effect on the stability of corrugated plating. Finally, tests to destruction on eight model corrugated bulkheads under water pressure are described, which suggest that the non-buckling behaviour of ships' bulkheads can be predicted using a simplified theory, but that their failing strength may be seriously impaired if due precautions are not taken to avoid instability. The tests further demonstrate the greater efficiency of aluminium alloy bulkheads compared with steel. Some conclusions concerning strength criteria for watertight bulkheads and the choice of trough shape are derived from these tests.—*Paper by J. B. Caldwell, read at a meeting of the Institution of Naval Architects on 9th June 1955.*

Partial Superstructures of Aluminium Alloy

This paper gives an account of experiments carried out on behalf of the Aluminium Development Association in the Department of Naval Architecture, King's College, Newcastle on Tyne, on composite model structures consisting of a steel main structure to which were attached aluminium alloy superstructures. Superstructures of three different breadths and of two different thicknesses were tested in a simple bending frame, central loads being applied to the structures. A series of experiments in which the length was systematically reduced was carried out for each superstructure tested, thus enabling the influence of the ratio length of superstructure divided by total length of structure to be investigated. Electrical resistance strain gauges were employed to explore the distribution of strain, and deflexion measurements of the structure were also made at a number of positions in the length. Stress maps were plotted showing how the stress, particularly at the mid-section, was affected by reducing the length. From these results it was possible to assess the degree of effectiveness of a superstructure in contributing to longitudinal strength. Comparison was made with earlier work based on ordinary bending theory. A theory of the behaviour of a partial superstructure is put forward and compared with the experimental results.—*Paper by W. Muckle, read at a meeting of the Institution of Naval Architects on 9th June 1955.*

Inert-gas Metal-arc Cutting

A new process has been devised for cutting non-ferrous metals which normally resist oxidation in the standard oxygen-cutting process. This should prove to be a boon to the metal fabricating field in shaping sections of non-ferrous plates prepartory to welding. Clean, bright metal surfaces are produced at extremely high cutting speeds on aluminium and other non-ferrous metals and alloys. This new process is a metal-arc-cutting operation in which a consumable electrode is used and the arc and active zone are shielded with an inert gas. This process uses inert-gas metal-arc welding apparatus with a few minor changes to take care of the increased power, wire feed rates and different shielding gases demanded by the process for its successful operation. Any standard inert-gas metal-arc welding torch can be used for the type of cutting in question, but the size of the torch that should be used depends upon the thickness of metal to be cut, power capacity and size of the consumable electrode. The consumable electrode is generally a steel wire of suitable size; 3/32-in. diameter steel wire has been used extensively in tests made to date by the author. The wire-feeding apparatus includes electronic controls, wire reel and wire driving unit with suitable gearing to meet wire consumption rates required by the workpiece thickness and wire size. Water cooling of the torch is needed to protect the apparatus at the high currents employed. In some apparatus, water-cooling permits the use of smaller and more flexible power cables between the control unit and torch. A suitable source of D.C. power is required. The capacity which is necessary depends upon the thickness and type of metal being cut. Although inert-gas metal-arc cuts can be made with both manual and mechanized welding torches, the lower capacity of manual torches limits the thickness of metal which can be cut effectively by this equipment. In cutting a given thickness of metal, higher currents give faster cutting speeds than lower currents, but the wire consumption is likewise proportionately higher. In order to more clearly illustrate the cutting process, an enlarged diagrammatic sketch of the cutting reaction is shown in Fig. 2. The torch is advanced over the workpiece along the desired line of cut at a constant rate of speed. As the torch advances and the wire is brought into contact with the workpiece, an arc is established between the wire and the workpiece. The arc energy progressively removes a controlled amount of metal to form a slot or kerf along the desired line of cut. The cutting arc differs from a welding arc in its location. A welding arc extends from the tip of the wire, a cutting arc extends from the side of the wire. The steel wire is fed through the torch at sufficient speed so that the tip of the wire reaches the bottom or underside of the workpiece. For a given power, there is a maximum cutting speed and a most effective wire-feed rate for optimum quality cuts on different thicknesses and types of material. The cutting arc with this process extends over the entire surface of the leading side of the wire from top to bottom of the workpiece. The arc consumes the wire on the leading side so that it becomes tapered in passing through the workpiece. If the tip of the wire is even with the bottom of the workpiece, the kerf or cut width will be uniform and the exhausting slag or dross will be completely discharged. When the tip of the wire is closer to the top of the workpiece, the cut surface is top-bevelled

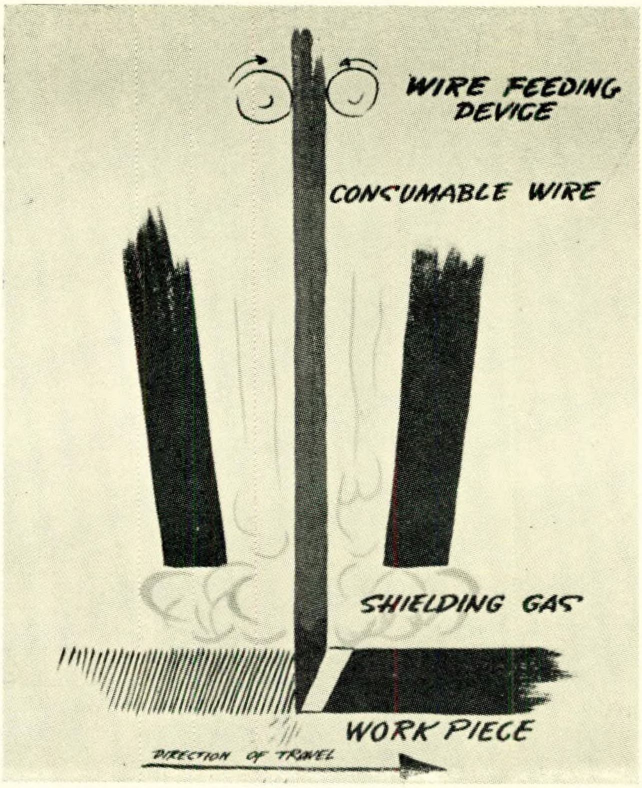


FIG. 2—*Basic inert-gas metal-arc cutting reaction*

and the slag adheres to the bottom of the cut. Excessive wire feed rates produce under-bevelled kerfs, and lead to lost cuts, gouges in the kerf, intermittent piercing and deposit of cold rod on the top of the base metal. It is therefore very important that wire feed and cutting speed be held constant for uniform high quality cuts. Power for inert-gas metal-arc cutting should be obtained from a direct current power supply of adequate capacity. Tests indicate that the range of current that can be used with standard inert-gas metal-arc apparatus is between 450 and 900 amp. This cutting process works best when reverse polarity is employed. The shielding gas best suited for inert-gas metal-arc cutting appears to be argon. Other gases which have been tried include hydrogen, helium, oxygen, nitrogen, air and carbon dioxide. Cut surfaces obtained with these gases were not usable in the as-cut condition due to roughness, discoloration, contaminated metal, adhering slag and/or failure to cut uniformly. Argon flow rates depend on material thickness and are in the order of 20 cu. ft. per hr. for average conditions. Slightly lower flows have been successfully employed in a few instances, but the higher flow seems to give more consistent results. Since the effectiveness of the inert-gas metal-arc cutting process is a result of arc force rather than gas velocity, there is no advantage to be anticipated from the use of higher gas flows, except as required for adequate shielding on thicker material.—*R. S. Babcock, The Welding Journal, April 1955; Vol. 34, pp. 309-315.*

Cavitation Pitting by Chemical Action

The author contends that certain extremely reactive unstable substances are produced locally in water and some other liquids, at the instant of final cavity collapse, by cavitational impacts. These substances complete subsequent chemical reactions in a few millionths of a second. They may react with each other, or with the liquid itself, and disappear by reforming into stable liquid. Those that are very close to a solid wall at the instant of their creation, however, can just as well react chemically with the solid. These reactions occur so quickly that even in a very fast-flowing liquid, the evidence (pitting) may be localized within an inch downstream of the main impact point. The paper represents an idea rather than new experimental data, the conclusions being stated in the form of contentions. The author, however, presents an analysis of known data upon which these contentions are based.—*Paper by I. Taylor, read at the 1954 A.S.M.E. Annual Meeting; Paper No. 54-A-109.*

Slamming of Liberty Ship in Head Seas

Experimental results of motion studies and slamming are presented. A 5·5-ft. model of a Liberty ship was tested in waves of varying length ($0 < \lambda < 9$ft.), of approximately constant wave length/wave height ratio ($\lambda/h \sim 20$), and with different thrusts (0·4 to 1·2lb.) at two draught conditions. Speed reduction, pitching angle, heave, bow acceleration, and slamming acceleration are presented in the form of dimensional and dimensionless graphs. Two experiments are discussed in detail, showing recorded pitching angle, heave, bow acceleration, and keel emergence as well as computed resultant bow displacement, bow velocity, and vertical wave velocity. It is shown that slamming can occur in regular waves under properly selected conditions.—*V. G. Szebehely and S. M. Y. Lum, Navy Department, David W. Taylor Model Basin, February 1955; Report No. 914.*

Screw Propellers and Cavitation

Numerous factors affect the performance of screw propellers. As is the case for ship hulls, there is much empirical data on screw propeller performance which is widely used for design purposes. However, only information regarding thrust, power, advance and rotary speeds for various advance ratios is given by the empirical data and this only for particular types of propeller. As a result of the phenomenon of cavitation and its erosion and noise effects, the empirical approach alone for propellers is becoming increasingly insufficient. Cavitation occurs when the local pressures at the tips or on the blades of propellers are reduced to the vapour pressure of the water, and bubbles, sheets, or clouds of vapour form. Or, in other words, it occurs when the local pressure reductions of the "lift pressures" developed by the blades become equal to the static pressures (at the particular points in the water) minus the vapour pressure of water. The theory of screw propeller action has been developed to a point where it is useful for practical application, and many designs have been considered on this basis in the laboratory. Unfortunately, it is not possible to attain cavitation conditions, similar to those on the full scale, during normal model tests under atmospheric pressure. Experiment work on cavitation requires a circulating water tunnel in which the pressure can be reduced. The static pressure in the experiment tank (atmospheric pressure plus pressure due to depth of propeller blade) is of the same order as on the full scale; the difference in pressure due to the difference in the depth of immersion of the full scale propeller and of the model, is relatively small. However, the pressure reductions on the model under test, when satisfying the Froude number and correct advance ratio, are smaller than on the full scale, approximately in proportion to the linear scale ratio. In order to make the difference—undisturbed static pressure minus pressure reduction—on the model equal to that on the ship, it is necessary to reduce the undisturbed static pressure of the model test approximately in proportion to the linear scale. It is not necessary to satisfy the Froude relationship for open water propeller tests (free from hull). Even so, if the static pressure is not reduced, in order to get the same cavitating conditions as on the full scale, impractical advance and rotary speeds are required.—*S. T. Mathews, International Shipbuilding Progress, 1955; Vol. 2, No. 8, pp. 194-200.*

New Swedish Cargo Liner

A new type of cargo motorship has been built by Eriksbergs Mekaniska Verkstad for the Swedish American Line, Gothenburg. This vessel, the *Vasaholm*, 8,575 tons d.w., represents the first of a new type of ship for the Brostrom group, there having been no ship of this size in the company's fleet before. The object in building this class of vessel is to provide a ship which will meet the special requirements of both the Swedish American Line and those of the Swedish East Asiatic Company, to as large an extent as possible, and to enable the vessels to trade in either company's services. Three vessels of similar size and construction are building at the Eriksbergs yard for the Swedish East Asiatic Company. The *Vasaholm* has been designed for the very exacting conditions of service in the North Atlantic and the Baltic, and is specially strengthened in excess of the classification rules. Ice strengthening is provided to comply with the requirements of the Finnish Board of Trade Ice Class B certificate. The vessel is propelled by a Burmeister and Wain supercharged two-stroke Diesel engine of 9,360 i.h.p. Her principal particulars are as follows:—

Length o.a.	482ft. 5in.
Length b.p.	445ft. 0in.
Breadth moulded	62ft. 0in.
Depth moulded to main deck ...	29ft. 0in.
Depth moulded to shelter deck...	38ft. 6in.
Draught loaded on summer freeboard	25ft. 9¼in.
Horsepower, b.h.p.	8,300
Speed loaded on trial, knots ...	17
Gross tonnage	6,233
Net tonnage	3,094
Deadweight, tons	8,575
Cargo holds non-refrigerated (bale), cu. ft.	504,000
Refrigerated cargo space, cu. ft.	37,000
Capacity of deep tanks for vegetable oil, tons	738

The *Vasaholm* is of the open shelterdeck type with poop and long forecastle, with a well raked stem, cruiser stern, streamlined funnel and bridge structure. The vessel has been constructed under the special survey of Lloyd's Register of Shipping to Class ✠ 100 A1, with scantlings required for an increased draught of 27 feet, and with the strengthenings as previously mentioned, and also to the requirements of Ministry of Transport and international conventions. As a closed shelter-decker the deadweight capacity of the ship can be increased by about 800 tons. The cargo space comprises five main holds, three forward and two aft of the engine room, and in addition 'tweendeck spaces in the poop and forecastle. There are three masts and three pairs of derrick posts with a total of nineteen derricks, eight of which are for 5 tons, four for 7½ tons, six for 10 tons, and one 30-ton heavy lift derrick, the latter being placed at the fore end of No. 3 hatch. These derricks are served by eighteen electric winches of A.S.E.A's latest design with built-in topping winches and automatic speed control. The vessel has five large cargo hatches and in addition a smaller one on the poopdeck. Steel hatch covers are provided for all hatches on the weather deck. The spaces for refrigerated cargo, placed aft of No. 2 hatch in the hold and shelter 'tweendeck, are intended for carrying fruit, frozen meat, fish, etc., and can be cooled down to − 24 deg. C. by a Freon cooling plant. The insulation has been carried out with fibreglass mattresses. The total cargo capacity amounts to 541,000 cu. ft., including the refrigerated cargo spaces of about 37,000 cu. ft. The capacity of the deep tanks, which are arranged at the sides of the tunnel in cargo hold No. 4, is 738 tons. The refrigerating machinery is of the semi-automatic, direct expansion type supplied by L. Sterne and Co., Ltd., Glasgow. There are four compressors, belt driven by electric motors of 40 h.p. each, the total capacity being about 61,800 kg. calories at + 30 deg. C. cooling water and − 10 deg. C. evaporating temperature. Six chilled air circulating fans are capable of circulating the cooled air about sixty times per hour, and arrangements are provided for keeping different temperatures in the individual spaces. Permanent strip lighting is arranged in the cargo holds. —*The Shipping World, 1st June 1955; Vol. 132, pp. 599-601.*

Investigation of Hydraulic Lock

Hydraulic lock is a phenomenon encountered in piston-type control valves governing the supply of high pressure oil to mechanical equipment. It is liable to occur when conditions are such as to produce an axial pressure gradient in the fluid contained in the clearance space surrounding the piston lands and its effect is to cause sticking or locking of the piston in its enclosing cylinder. A preliminary investigation carried out in the Mechanical Engineering Department of Birmingham University indicated that true hydraulic lock is essentially a hydrodynamic effect, resulting from asymmetric pressure distribution in the clearance space between piston and cylinder. Thus, if owing to machining imperfections in the piston lands or cylinder bore, the flow of fluid in the clearance space is either diverging or converging, an asymmetric distribution of pressure will in general result and will be accompanied by a transverse thrust on the piston. If the clearance is divergent in the direction of flow, the piston will be forced against the cylinder wall; if convergent, the action of any thrust arising will be to exert a centralizing effect. The present paper deals with an extension of the investigation referred to above, the object being to examine, both theoretically and experimentally, the effect on hydraulic lock of the employment of tapered piston lands. For this purpose, a mathematical analysis was made of the pressure distribution in the clearance space between piston lands and cylinder bore and of the accompanying side forces on the piston for a range of selected conditions. To establish the validity of the analysis, experiments were carried out on a cylinder of uniform bore in conjunction with dumb-bell-shaped pistons having lands of such form as to give various taper-clearance ratios. In general the procedure was as follows: high pressure oil was supplied through the cylinder wall to the annular space between the two lands of the piston, the oil then flowing outward in both directions through the clearance space. Pressure tappings at a number of points in the cylinder wall allowed the pressure distribution in the clearance space to be determined, and suitable provision was made for the measurement of any locking force acting on the piston.—*Paper by J. Manhajm and D. C. Sweeney, read at a General Meeting of the Institution of Mechanical Engineers, on 13th May 1955.*

Design and Application of the Vertical Axis Propeller

Published literature on the principles of operation and performance of vertical-axis propellers is rather scarce. The author provides a major contribution to this literature by reviewing past history in Germany as well as recent developments in this country. A discussion of the hydrodynamic principles, and of the characteristics and merits of various blade motions is presented. For this a method of plotting comparative efficiencies is employed which relates the propeller efficiencies to the ideal jet efficiency, thus arriving at the term "degree of perfection" which does not become zero at zero speed. Reference is made to vertical-axis propellers constructed in the United States for the Navy and the U.S. Army Transportation Corps. Detailed information is provided on the latest design of two 1,000-h.p. propellers for the U.S. Army Towboat LTI-2194 with particular emphasis on theoretical and experimental stress and deflexion analysis. Remarks and data are included covering trials and experience in commercial service operation.—*Paper by H. F. Mueller, read at the Spring Meeting of the Society of Naval Architects and Marine Engineers on 19th May 1955.*

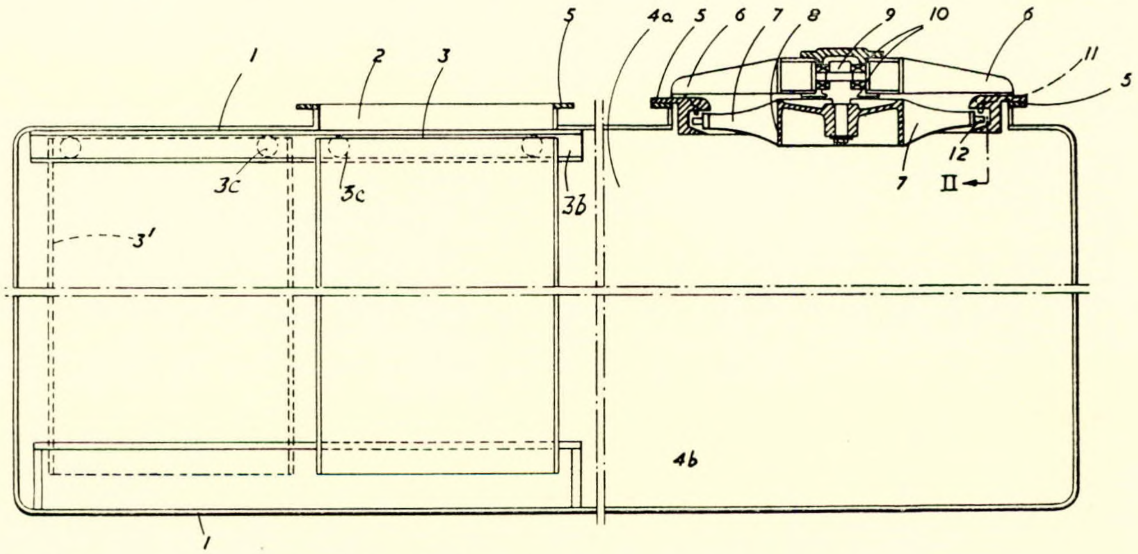
Additives for Preventing Corrosion in Oil-fired Boilers

In recent years, corrosion in the low temperature zones of oil-fired boilers has become increasingly serious, particularly in the cold section of the air heaters. Primarily, two factors have been responsible for the increased difficulties with air-heater corrosion. One is the decrease in air-heater metal temperatures which has resulted from the demand for more efficient boilers. The second is the change in the quality of the residual oil available for central-station consumption. There has been a significant increase in the sulphur and ash content of the oil. In addition, the amount of vanadium and alkalis in the ash has increased. This has been brought about by changes in refinery processes and the utilization of oil from new fields. All boilers which burn sulphur-bearing fuels will produce some sulphur trioxide which combines with the water vapour in the flue gases to produce sulphuric acid. The amount of corrosion in a specific unit depends primarily on the relationship of metal temperatures to the dew-point temperature of the flue gases and the balance between the sulphuric acid condensing and the quantity of basic ash constituents. Residual oil fuel is comparatively high in sulphur and its ash frequently contains a high proportion of constituents which may promote the formation of sulphur trioxide. In addition, oil ash contains limited quantities of materials which will absorb sulphur trioxide or neutralize sulphuric acid. For this reason the combustion gases from most residual oils are considered potentially corrosive. The use of an additive, such as dolomite, will reduce the amount of sulphur trioxide present in the flue gases and will inhibit sulphuric acid corrosion. Furthermore, dolomite will reduce the quantity of ash depositing in the air heater and the amount of work necessary to keep these surfaces clean. The use of dolomite for the prevention of low temperature corrosion is economically feasible and it is recommended where corrosion or pluggage is a problem.—*E. C. Huge and E. C. Piotter, Transactions, A.S.M.E., April 1955; Vol. 77, pp. 267-278.*

Patent Specifications

Evacuating Gases from Tanks

This invention envisages a method for the removal of gases from holds or tanks on board tankers, etc. In the drawing (1) indicates the space to be ventilated and (2) an opening provided with a flange (5). A tube (3) movable on a roller track is provided. This tube can be made of bronze or of brass or of canvas which is held stretched by means of rings. No parts of the device should be made of steel, as steel in striking steel in the equipment could cause sparks which would set fire to the very explosive gas-air mixture. When the tube is out of use, the tube (3) is pushed into the position (3′). In the other opening of the hold there is arranged a ventilation device, a fan. The fan blades are designated (7) and the hub of the fan is designated (8). The shaft of the fan is shown at (9), and (10) is a bearing. The bearing is supported by the arms (6) resting on the flanges (5). The entire fan is supported by a framework so that the complete device is easy to transport. The device can be used in different spaces in one and the same tanker and also on board different ships.—*British Patent No. 731,754, issued to A.B. Svenska Flaktfabriken. Complete specification published 15th June 1955.*



Estimating Longitudinal Bending Moment in Hull

The invention relates to an apparatus by means of which the suitability of a certain distribution of the cargo can be estimated rapidly and reliably in consideration of the strains of the hull as caused by the longitudinal bending moment. The ship is conceived as being divided into two parts, i.e. a forepart and an afterpart, and the cargoes placed astern of a conceived plane in the afterpart give the ship a positive moment, while the cargoes placed ahead of this plane give a negative moment. For the forward part the rule holds good that the cargoes placed astern of a corresponding conceived plane give negative moments, whereas those placed ahead of said plane give positive moments. In both cases the moment is taken to be a function of the weight of the cargo and the distance thereof from the plane conceived of. Within a displacement and trimming range corresponding to the total load of the cargo, the weight of the hull and the buoyancy of the displacement causes a constant moment value, here called residual moment. The algebraic sum of the moments caused by the loads and the residual moment gives a conception of the excess moment for the actual load distribution. The invention comprises an apparatus for the estimation of longitudinal bending moments in the hull as represented by the difference between positive and negative moments and caused by the distribution of the cargo in different holds within the ship. The apparatus comprises an electric balancing means of the bridge circuit type and includes an adjustable resistor for each individual hold. These resistors are adjustable to represent the moment of the cargo in the hold in question with respect to a given reference plane. The resistors corresponding to holds giving positive moments are arranged in series in one branch of the bridge circuit and the resistors corresponding to holds giving negative moments are arranged in series in the other branch of the bridge circuit. The bridge circuit is provided with a further resistor representing the residual moment in the hull as caused by the weight of the hull and the buoyancy of the displacement. A potentiometer is connected between the two bridge branches and is provided with means for zero-setting of an instrument which, in its zero-position, indicates electric balance between the two bridge branches. The setting of the potentiometer then indicates the difference between the positive and negative moments and represents the stresses in the hull exceeding a calculated mean value.—*British Patent No. 731,428, issued to A.B. Götaverken. Complete specification published 8th June 1955.*

Automatic Bottom Valve for Lifeboats

A hole is provided in the bottom of lifeboats so that water which may collect in the boat when it is located on board is allowed to flow out of said hole. The commonly used device for closing the hole is a stopper. In launching lifeboats during accidents, however, panic or hurry often occurs so that it frequently happens that the stopper has been forgotten. The invention describes a valve which, if desired, may be set open, but which automatically closes in launching the lifeboat.

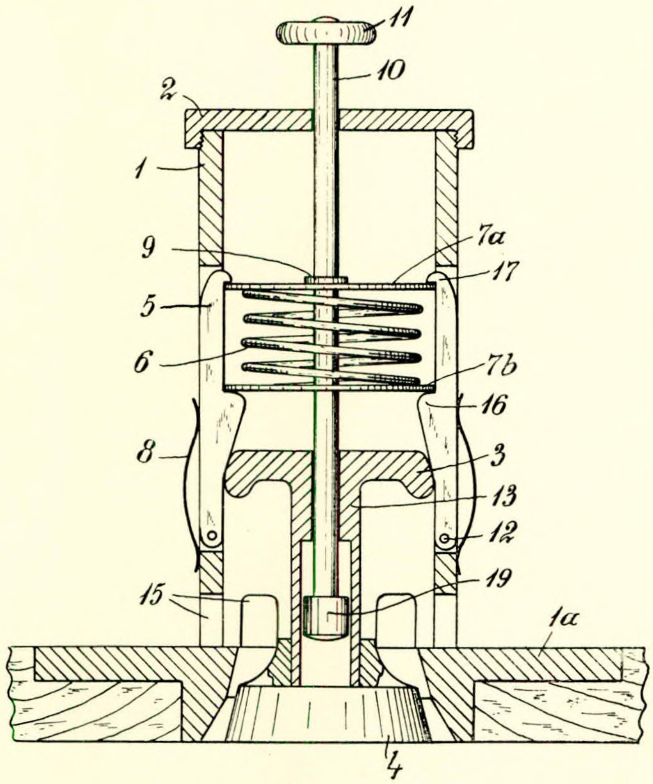


FIG. 1

Referring to Fig. 1, when the valve is to be opened, the rod (10) is pressed down by means of button (11), so that valve disc (4) and the piston (3) are moved downward. When the piston (3) in this way is moved down in the housing, the catch levers (5) will be pressed inwards by the leaf springs (8), and the hooks (17) will grip the edge of the upper disc (7a) and keep this in a position with loaded springs (6), as shown. This is the position in which the valve is set when the boat is placed on the chocks. Water which may collect in the bottom of the lifeboat is then permitted freely to flow out through the apertures (15) and the open valve. Before the boat is to be launched, the valve should be closed by pulling the button (11) upwards. If, however, this should be forgotten, water will, when the boat is launched, flow in through the valve opening and thereby act on the valve (4) and also on the piston (3). The piston and the valve will then be moved in an upward direction. The rounded-off edges of the piston (3) then will move up along the cams or projections (16) of the catch levers and press them outwards. Thereby the hooks (17) go out of engagement with the upper disc (7a), whereby the spring power is allowed to move the rod (10) upwards. By means of the piston (19) in the bore (14) the valve is then pulled up into the closed position.—*British Patent No. 726,875, issued to R. Wilk. Complete specification published 23rd March 1955.*

Operation of Hatch Covers

This invention relates to a device for the operation of hatch covers. Assuming the hatch cover sections to be spread in their closed position (Fig. 1), if the ropes are pulled in the

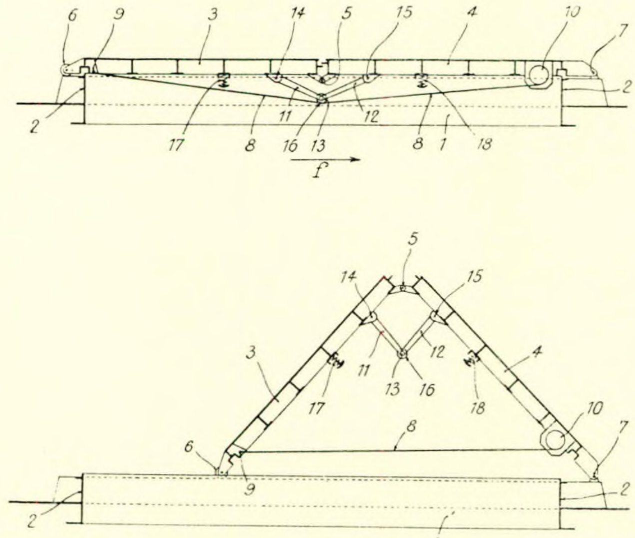


FIG. 1 *(upper)*; FIG. 2 *(lower)*

direction of the arrow (*f*), their sag will decrease progressively and the links (11 and 12) bearing on the ropes will be lifted to a certain extent thereby starting the folding movement of the overlying hinge (5) and, therefore, of the hatch cover sections (3 and 4). By continuing to rotate the winch (10), the sections are successively brought to the position shown in Fig. 2. At this time the springs (17 and 18) engage each other or an appropriate portion of the opposite hatch cover sections. Thus, when the sections are operated in the reverse direction the rope or like traction member (8) is first slackened and the springs (17 and 18) are thus able to repulse the sections and start spreading them. The weight of the hatch cover sections keeps the rope (8) constantly taut.—*British Patent No. 729,379, issued to MacGregor-Comarain. Complete specification published on 4th May 1955.*

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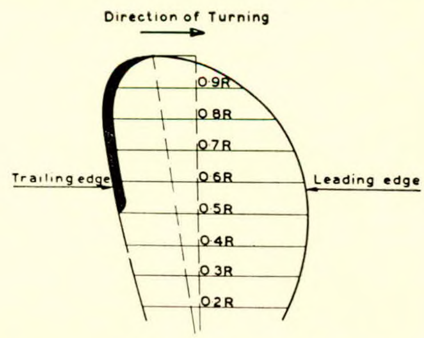
Marine Engineering and Shipbuilding Abstracts

Volume XVIII, No. 10, October 1955

Vibration Problems in Propeller Design

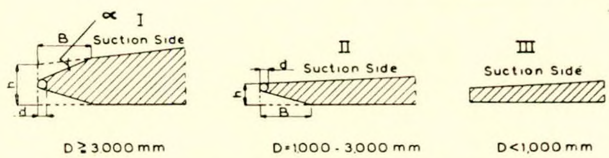
The inconvenient noise frequently noticed with revolving marine propellers and generally termed "singing", is essentially a critical condition of vibration; also, in the case of the subject on hand, with propeller blades. If we want to explain the phenomenon of singing propellers we have to consider the trailing edge of a blade profile. Here the flow leaves the blade profile and forms the leaving vortex. On any propeller blade that is not provided with an anti-singing edge, the releasing point of the leaving vortex shifts within certain limits. This releasing point may be on the pressure side as well as on the suction side of the blade element, while the said limits are, in any case, situated at a very short distance from the trailing edge. If the frequency of the displacement of the releasing point moving up and down agrees with the natural frequency of vibration of the propeller blade, singing will occur. Since the natural frequency of vibration of a propeller blade depends on the shape of the blade, the form of the blade profiles, and the curve of the maximum blade thickness of the profiles, while the frequency of the displacement of the releasing point of the vortex is unknown, it is very difficult to establish beforehand that the phenomenon of singing will occur. It sometimes happens that a damaged propeller which was not singing before being damaged, will sing after having been repaired which, in the authors' opinion, is due to the accurate reconditioning and smooth finishing of the blade profiles at the trailing edges, which enables the releasing point of the vortex to shift. On the other hand, it is not necessary for every propeller to sing when revolving behind the ship. The best method of preventing singing of the propeller blades is to grind anti-singing edges on the trailing edges of the blades in such a way that the releasing point of the leaving vortex is fixed. Fig. 3 shows the different types of anti-singing edges, depending on the screw diameter, for bronze, cast steel and nodular iron propellers manufactured according to the Lips' design. From this figure it is seen that for propeller diameters of upwards of 3,000 mm., the trailing edge between 0·5 *R* and the tip of the blade has to be cut chisel-shaped on either side over a breadth of a few centimeters (I), while the breadth of the anti-singing edge has to be such that the leaving vortex will be forced to leave the blade profile somewhat earlier than would be the case if there was no anti-singing edge. For propeller diameters of upwards of 1,000 and less than 3,000 mm., when the angle *a* becomes too small, it is sufficient to grind the anti-singing edge only on the pressure side (II). For smaller propellers, of less than 1,000 mm. diameter, it is recommended to grind the anti-



D = Diameter of the Propeller

D mm	h mm	B mm	d mm
7,000	9·5	30	3·00
6,000	9·0	29	2·75
5,000	8·5	28	2·75
4,000	7·5	25	2·50
3,000	6·5	22	2·50

D mm	h mm	B mm	d mm
3,000	6·50	22	2·50
2,500	5·75	20	2·25
2,000	5·00	18	2·00
1,500	4·00	16	1·75
1,000	3·00	14	1·50



NOTE The Value of "h" is valid for 0·7R - 0·8R and 0·9R
The Negative Tolerance of "h" may Not Exceed ½ mm

FIG. 3—*Different types of anti-singing edges*

singing edge as given under (III).—*J. A. Van Aken, European Shipbuilding, No. 2, 1955; Vol. 4, pp. 26-33.*

Transverse Strength of Tankers

A ship's hull has to fulfil many requirements. It should furnish the ship with good hydrodynamic and hydrostatic properties, it should provide the cargo space required, it must be of ample strength, it must be divided into watertight compartments, and so on. All the necessary requirements cannot be fulfilled without each taking account of the others, so that compromises are imposed. Generally it may be said that the components of the hull have to be so designed as to suit, so far as is possible, the shape decided upon for the hull and the plan for holds, tanks, etc. Compared with a cargo ship, a tanker facilitates the design of the hull from the purely strength point of view. However, research in this field has not advanced so far as to solve all the problems occurring in connexion with the strength of ships. The hull of a tanker consists mainly of a long parallel middle body, i.e. the shape of a cross-section is the same for a very great part of the ship's length. At the ends the hull is shaped to give the ship suitable stem and stern. In principle, the hull constitutes a closed box of plate (Fig. 1 (a)). The hull is divided into tanks by longitudinal and

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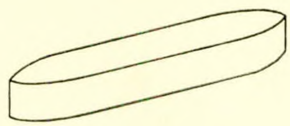

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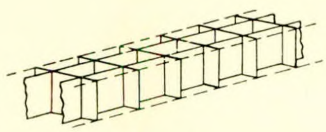


FIG. 1

transverse vertical bulkheads. Fig. 1 (b) shows a very common arrangement. Here similar tank compartments extend over nearly the whole length of the ship. A short pump room is situated between two tank compartments near the middle or at the ends of the ship, but this is of no interest for the present paper. Large plate areas are formed by the bulkheads and between the bulkheads. These plate areas must be reinforced by stiffeners. Because of the high longitudinal stresses occurring in a long tanker, all modern tankers are now being built with longitudinal frames at the deck plating and at bottom plating. Even the side plating and the longitudinal bulkheads are mostly framed longitudinally. In each tank compartment the longitudinal members are supported by deep transverse web frames. In the wing tanks, one or more struts connect the side plating and the longitudinal bulkhead. Fig. 2 shows one type of transverse design. Instead of having longitudinals at the longitudinal bulkheads, these bulkheads are often corrugated, but this does not affect in principle the transverse design. The transverse bulkheads are mostly stiffened by horizontal stiffeners. In this manner one can satisfy the requirement for stronger stiffeners at the bottom than at the top of the tank. These stiffeners, too, are supported by deep vertical members. The transverse bulkheads are often corrugated. In the centre line of the ship, strong girders are applied at the deck and at the bottom. The purpose of these girders is to increase the strength of the ship in the longitudinal as well as in the transverse direction. In the latter case, this will be effected by the supporting action on the transverse frames. When judging the strength of a ship, the longitudinal stresses appear to be the only stresses of fundamental interest. It has been found

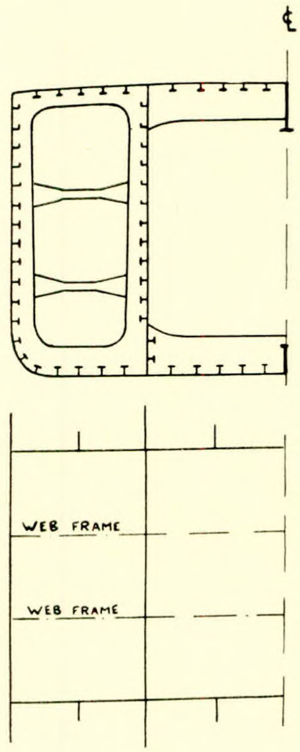


FIG. 2.

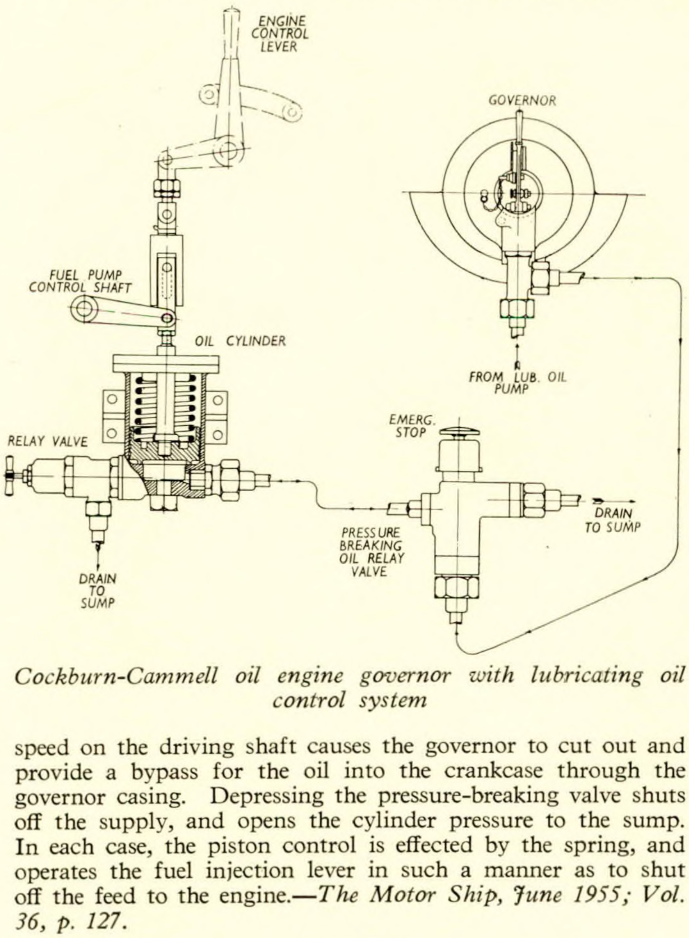
by practical experience, however, that even the transverse strength is of extreme importance. Fractures have frequently occurred in the transverses of tankers, involving expensive repairs and delays. Further, the longitudinal strength of a ship is to some extent dependent on the transverse design, owing to the interaction between longitudinals and transverses. As some of the members of the transverse framework are very deep, the *shear deflexion* of these members is likely to be considerable. The considerable shear deflexion of the deep and short transverses near the bottom of the ship raises the question whether the simple beam theory is valid for these members. It is desirable to get more knowledge about this question. Until more elaborate methods for analyses of such members are known, the method given in this paper seems to constitute a practicable approximation. The analysis of the transverses at the ends, where brackets or similar components are fitted, concerns the deflexion of the transverses and should not be used to calculate the stresses in these regions.—*E. Steneroth, Transactions of the Royal Institute of Technology, Stockholm, No. 90, 1955*

Ship Repair Yard Operation

The subject of production planning for a repair shipyard calls for very extensive treatment and the author has endeavoured to review in general outline those methods now in use or which could be adapted to that type of organization. The author hopes that his study will show that production planning principles can be applied to ship repair yard operation. He indicates some of the advantages which these principles can bring, and some of the more important considerations in the development of such a plan are given. An effort has been made to ensure that planning is not detailed beyond that point where it becomes uneconomical because of the variety and unpredictability of ship repair tasks and the usual limited size of activity area and organization. The planning organization described is especially applicable to ship repair work because of its flexibility and ability to expand. These characteristics are desirable because of the large variety and fluctuations of work load inherent in this industry. The body of the paper is divided into sections which correspond to the different planning activities of the proposed production department. The sections are presented in order of their occurrence in the overall planning operation and although they are interdependent, can be treated separately under the headings of (*a*) Organization; (*b*) Production programme; (*c*) Materials; (*d*) Work operations; (*e*) Machines; (*f*) Labour; (*g*) Inspection; (*h*) Yard maintenance; (*i*) Transportation; (*j*) Yard layout. It is felt that this method of presentation lends itself most readily to use in the development and evaluation of planning procedures to suit requirements of individual ship repair establishments.—*Paper by R. Schor, read at the Spring Meeting of the Society of Naval Architects and Marine Engineers on 19th May 1955.*

Oil Engine Governing Device

A safety device for cutting off the supply of fuel to an engine in the event of overspeed or loss of lubricating oil pressure has been placed on the market by Cockburns, Ltd., Clydesdale Engineering Works. An illustration is given showing the component parts of the apparatus, which is known as the Cockburn-Cammell oil engine governor with lubricating oil control. The system consists of a speed governor, a pressure-breaking relay valve, and an oil cylinder controlling the fuel injection delivery. The oil pressure is taken from the discharge side of the lubricating oil pump, and a connexion is made through the governing device to the pressure-breaking valve, which is set to a predetermined figure. On the oil cylinder, a relief valve is fitted to prevent over pressure in the system, and on the pressure-breaking relay valve there is a hand trip for emergency use. With this arrangement, oil pressure is removed from the oil cylinder in the event of pump failure, while over-



Cockburn-Cammell oil engine governor with lubricating oil control system

speed on the driving shaft causes the governor to cut out and provide a bypass for the oil into the crankcase through the governor casing. Depressing the pressure-breaking valve shuts off the supply, and opens the cylinder pressure to the sump. In each case, the piston control is effected by the spring, and operates the fuel injection lever in such a manner as to shut off the feed to the engine.—*The Motor Ship, June 1955; Vol. 36, p. 127.*

Energizing Propulsion Motor to Prevent Shaft Rotation

It is occasionally necessary to prevent rotation of a propeller shaft when a vessel is being towed or driven by other propellers. This can be done in direct current electric drive vessels by applying power to the propelling motor in such a manner that it produces torque equal and opposite to that resulting from dragging the propeller. Several instances have been noted in recent derangement reports where this method has been used to prevent rotation of one of the propeller shafts in twin-screw Diesel electric direct current propelled vessels. In these cases derangements of the shaft or motor bearings made it necessary to hold the deranged shaft while propelling with the other. Propulsion motors are usually designed with ventilating fans attached directly to the rotors. Therefore, when the armature is held stationary by applying sufficient current to overcome the propeller torque, severe overheating may occur due to the lack of ventilation. In addition, when current is passed through a stationary armature, the commutator bars may be severely burned and the commutators may be warped due to unequal heating. In view of the damage which may result, the practice of holding a shaft stationary by circulating current through the propulsion motor armatures should be avoided wherever possible. If used as an emergency method of control, this method should be limited to the minimum time necessitated by the emergency. Wherever practicable, mechanical locking means should be used to hold the shaft stationary. —*Bureau of Ships Journal, March 1955; Vol. 3, p. 20.*

Suppression of Burner Oscillations

Tests on the suppression of burner oscillations are described. The work produced several positive conclusions which may help in quietening particular combustion installations or in preventing waste of effort in trying ineffective methods. The effectiveness of a quarter-wave tube was found to be critically dependent on length, but relatively insensitive to location, as long as the tube is placed in the region of the pressure antinode. Similarly, the effectiveness of a Helmholtz resonation depended critically on volume for a particular neck, but is quite insensitive to location. In neither case did the suppressor have to be placed near the particular antinode where energy was fed into the oscillation. The degree of suppression obtained was approximately proportional to the cross-sectional area of the tube. The same conclusion appears to be warranted with regard to the cross-sectional area of the throat of a Helmholtz resonator. Almost complete suppression was obtained with holes drilled in the side of the tube and placed within 10 per cent of a wavelength from a pressure antinode. It was found that the diameter of the hole need not be greater than about one-tenth to one-fifteenth of the diameter of the combustion chamber, but should be greater than the wall thickness for best suppression.—*Paper by A. A. Putnam and W. R. Dennis, read at the 1954 A.S.M.E. Annual Meeting; Paper No. 54-A-174.*

Gas as Important Factor in Thermal Conductivity of Insulation

Recent work has shown that the major method of heat transfer through most insulating materials is by gas-transferred heat. This quantity is found in the laboratory by subtracting the heat transfer obtained with gas extracted from that obtained with gas present. In this paper, the author describes an investigation of the reduction in thermal conductivity of some common insulating materials by replacing the ambient air with gases of larger molecular weight, such as carbon dioxide and dichlorodifluoromethane. Experimental data are presented for several materials which show the relative importance of heat transferred by radiation and by conduction through the solid material. The reductions in thermal conductivity vary considerably, and in some cases amount to approximately 50 per cent.—*R. M. Landar, Heating, Piping and Air Conditioning, 26, No. 12, 1954; p. 121. British Shipbuilding Research Association Journal, March 1955; Vol. 10, Abstract No. 10,029.*

Cathodic Protection of H.M.C.S. *New Liskeard*

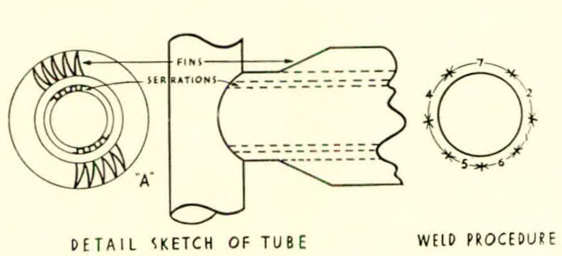
H.M.C.S. *New Liskeard* was the first active ship having an underwater area greater than 1,000 sq. ft., on which a definite attempt was made to protect completely the exposed underwater hull against corrosion by supplementing paint with a cathodic protection system whose current output could be controlled in order to attempt to maintain the hull potential at an optimum value. The original experiment with the *New Liskeard* was begun in June 1949, when a number of magnesium-alloy anodes was fitted to the bilge keel of the ship. The vessel was drydocked in May 1950, and the successful results obtained with this initial experiment were described. The present report deals with further trials to determine if the period between drydockings could be successfully extended from one year to two years, insofar as the effects of corrosion and fouling were concerned. Service conditions prevented a full two-year trial from being carried out, but from the data obtained for the twenty months that the trial was run, and from supplementary experiments, it is considered that, from the corrosion point of view, it is possible to extend the period between drydockings of active ships in seawater to two years and that in East Coast Canadian waters, if painting is done during the spring period of the year, the fouling growth does not become too severe to preclude a two-year period out of dock. The use of a patch of vinyl paint near the anode arrays helped to decrease the current required for the protection of this ship, from an average of 25 amps. on the earlier trials to one of 20 amps. The effect of a complete vinyl system applied subsequently was a further reduction of the average current requirement from 20 to 12 amps. In conclusion, it is pointed out that before a perfectly clean ship can be expected at the end of a two-year period, it will be necessary either to improve the common anti-fouling paints, or the adherence of vinyl paints when cheaper pretreatment methods are used than thorough cleaning and sandblasting. A requirement also exists for segregation studies of the AZ63 magnesium alloys employed for the anodes; also, investigations should be made into the possible use of other magnesium alloys that may show less inclination to segregate inversely when chill cast.—*Naval Research Establishment, Canada, Report No. PHx-83. British Shipbuilding Research Association, March 1955; Vol. 10, Abstract No. 10,043.*

Hot Ductility of High Temperature Alloys

This report summarizes the details of construction of a device for determining the effects of testing temperature and prior thermal history on the hot ductility of structural alloys. The device permits subjecting suitable samples to the actual thermal cycles experienced in the heat-affected zone of an arc weld made in the alloy under study, and evaluating the influence of these thermal cycles on the hot ductility of the material. Two grades of stainless steel, a cast and heat-treated type modified with columbium and a wrought type stainless steel have been subjected to such tests. The results indicate a significant difference in the hot ductility of these materials. Furthermore, tests show that brief exposure to temperatures in the vicinity of 2,400 deg. F., such as would be found in the region immediately adjacent to the fusion zone of an arc weld, severely reduce the hot ductility in both grades. It is believed that the reduction in hot ductility and the incidence of cracking in the heat-affected zone are intimately related.—*E. R. Nippes,* et alii, *The Welding Journal, April 1955; Vol. 34, pp. 183-s-196-s.*

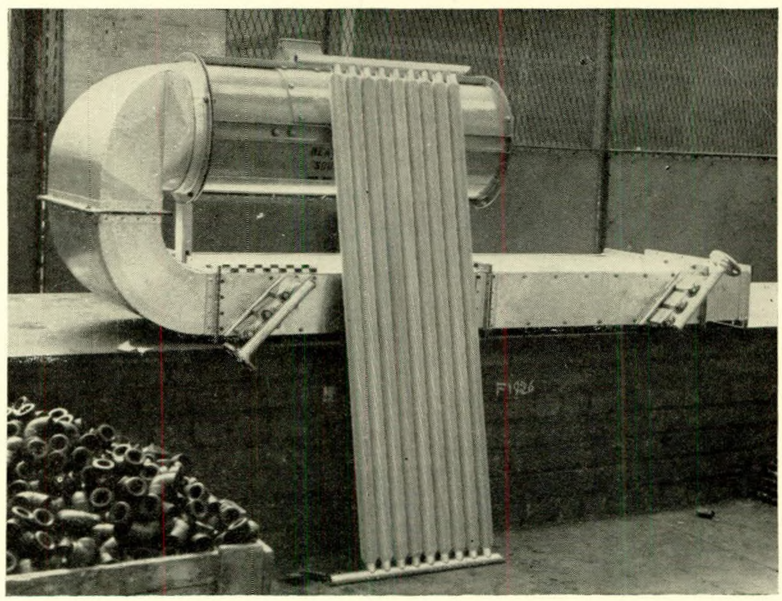
Welded Air Heaters for Whale Factories

Chr. Salvesen and Company, Leith, are proposing to install air heaters in their Antarctic whaling vessels *Southern Harvester* and *Southern Venturer*. James Mitchell and Son (Greenock), Ltd., have been responsible for their fabrication. Each air heater requires four units and each unit has three tube assemblies. These tube assemblies are made from twelve tubes fitting at each end into headers, both tubes and headers being of 99·9 per cent pure aluminium. The tubes are unique in design and are the patent of Chr. Salvesen and Company. Initially they are extruded to give fins on the outside and serrations on the inside circumferences. The tubes are then processed to give a uniform twist lengthwise and the outer fins take on an additional curve. This can be seen more clearly in the detail sketch of the tube. In the assembly of the tubes into the headers the fins are practically in contact so that access for welding proved very troublesome, there being only about ½in. clearance between weld points. These difficulties were overcome by adopting the welding procedure shown. Welds Nos. 1 to 4 were completed by a right-hand operator with the torch in the right hand whilst weld No. 5 was made with the torch in the left hand after the job had been completely turned over. Weld No. 7 was made after turning the job back to its original position. Welds were made on all the ten pipes in the sequence of 1-2, then 3-4, then 5-6, the seventh weld being completed on all ten pipes at both ends. On completion, all the welds successfully passed pressure tests and the job was completed to the owner's entire satisfaction. In the welding of six assemblies (one hundred and twenty complete welds), 330 cu. ft. of Argon and 4lb. of ⅛-in. 5 per cent silicon aluminium welding rod were used. The Argon flow was 12 cu. ft. per hr. and the current



Above: Showing the weld procedure for manufacture of the air heater elements

Right: A completed element and the heater assembled in the ducting



was 110/140 amps. These details were given in *The Torch*, technical journal of the British Oxygen Co., Ltd.—*The Marine Engineer and Naval Architect, May 1955; Vol. 78, p. 204.*

Direct-driven Diesel versus Geared Steam Turbine Installation

It is very difficult to get reliable service data regarding repairs and maintenance of ship machinery. In the literature available very contradictory opinions are expressed. Nearly all the authors are complaining of lack of exact data. The author gives some general remarks on the opinions held in this respect by European and American owners and engineers who have had experience with both types of machinery. The question is easily solved with regard to the Diesel engine, because it is a very complicated structure. It has a lot of highly stressed precision-machined parts, which are subject to wear, fatigue, and rupture. Its maintenance cost is high, as is also the demand for skilled maintenance personnel. The same question, however, is not so easy to solve with regard to steam turbine installations, because one is apt to think only of the steam turbine itself when considering the maintenance problems of a steam turbine plant. The steam turbine propelling plant includes not only steam turbines, but also condensers, condensate pumps, feed booster pumps, de-aerating feed tanks, main feed pumps, boilers, superheaters, economizers, and a vast amount of piping, myriads of valves and hundreds of control accessories. When we add up the number of parts involved in this family of equipment, we shall find almost as many sources of trouble, dislocation and maladjustment as the Diesel installation provides. According to available information steam turbine installations have very low maintenance costs during the first year of service, but these costs will usually increase very rapidly after some years. Several owners are of the opinion that the maintenance cost during the ship's life may be the same for both types of machinery. Norwegian owners have not got much experience with modern turbine installations; they are, however, apt to think that the maintenance cost of a Diesel installation is from 100,000 kr.-200,000 kr. higher per year than that of a turbine installation with outputs of from 5,000-13,000 s.h.p. There is, however, one thing in this connexion which ought to be mentioned. Data for maintenance costs must necessarily be based upon engines constructed and installed some years ago. The comparisons drawn by engine builders, however, are based on their latest constructions and even on designs not yet carried out. In the past few years the Diesel engine has become complicated and the steam turbine installation of simple construction. Now, however, a different tendency is evident. The turbine installations, with their ever-growing auxiliary equipment and with the increasingly expensive materials required, tend to become more elaborate and costly as regards maintenance costs, whereas with Diesel machinery the trend has been towards simpler designs. From now onwards, the attractiveness of steam machinery, on that count may, therefore, be expected to decline. With the use of boiler oil in motor ships, we shall have to reckon with an increase in the cylinder wear up to 100 per cent. The extra cost will be about 11-12 per cent of the gain on the fuel bill. It would be of great interest if in the years to come the owners could draw the maintenance costs from the total amount, so that material could be collected and statistically treated.—*H. Johansen, International Shipbuilding Progress, 1955; Vol. 2, No. 8, pp. 179-193.*

Quayside Wind Break

The problem of berthing large ships in a cross wind has become so serious in the port of Marseilles that it has been necessary for Estrine et Cie., agents for the P. and O. and the Orient Line, to take steps to avoid damage to the vessels using the Leon Gourret Mole. Accordingly, the Institut de Mécanique des Fluids, of the Aix-Marseilles University, was asked to make a study of the stresses sustained by large vessels tying up in the port, and to suggest some method by which berthing conditions affording a maximum of security could be obtained. On the findings of Professor Valensi, a director of the Institute, it was obviously impossible, with the normal method of berthing, to hold large vessels of 28,000 to 30,000 tons at this quay when the mistral reached about 75 miles per hour; it is known that during certain gusts a speed of about 95 m.p.h. is reached. Some special form of protection was therefore necessary, and a perforated wall, which had already been tried out by Professor Valensi at Le Havre, was suggested. Under the direction of M. Flinois, managing director of the concessions of the Chamber of Commerce, a study was made of the practical conditions under which a wind break of this type could be erected at the Leon Gourret Mole. The height of the bridge above sea level of the large P. and O. and Orient Line ships is 78 feet 9 inches and the windscreen wall which was erected is 73 feet 9½ inches high. The original length of wall decided on was 820 feet 3 inches, but it was later discovered that a wall 393 feet 9 inches would not only reduce the effect of the wind sufficiently, but would also be very much cheaper to build. In addition, it was found that a considerable improvement could be obtained by extending the length of No. 19 shed by about 82 feet, and this would mean that the wind break would be lengthened by a corresponding amount. The benefit derived from this type of shelter when the vessel is in the final stages of berthing and when she is tied up alongside is considerable. Once the vessel is alongside the quay the wind speed, under normal mistral conditions, is reduced from 74 m.p.h. to about 17 m.p.h. This not only allows the cranes greater freedom of operation but also safeguards the passengers while embarking and disembarking, and facilitates the ship's sailing.—*The Shipping World, 15th June 1955; Vol. 132, p. 645.*

U.S. Ocean Radar Station Ships

Four former Liberty ships were converted recently into the U.S. Navy's first ocean radar station ships. Designated YAGR, the newly commissioned ships operate out of Newport, Rhode Island, as offshore units of the continental air defence system. Although other Navy ships have been employed in radar picket duty off the Atlantic coast, the YAGR's are the first which have been detailed for this specific purpose. Conversion work on the radar station ships consisted of the installation of extensive communication and electronic equipment and additional berthing and messing facilities for the crew. The electronic equipment includes air and surface search radar. Because cargo vessels without cargo in their holds generally roll uncomfortably, ballast is used in the YAGR's to make them less susceptible to the effects of rough seas. The combination of ballast and additional structural bulkheads also greatly increases the safety of crew members.—*Bureau of Ships Journal, May 1955; Vol. 4, p. 19.*

New Metal Powder Deposition Process

The T.L.B. process has been specially developed for the deposition in powder form of hard facing alloys, that is, alloys having heat, wear and corrosion resisting properties. In addition, a wide range of metal powders other than hard facing can be processed through the torch. The process is claimed to be more flexible than orthodox rod deposition and is consequently more adaptable to automatic deposition. With the usual oxy-acetylene process it is necessary for the operator to hold the welding torch in one hand and the filler rod in the other, the material being deposited is melted in the gas flame and the metals to be joined are fused by the same flame. The T.L.B. process needs only one hand to carry out the complete operation. Powdered metal is fed into the gas stream passing through the torch, and is carried along by the gas mixture to the welding nozzle or tip. The metal powder is heated as it travels through the flame, and is melted by the time it reaches the base or parent metal, producing a true bond. A great variety of materials can be welded or deposited with this new process, depositions of up to ⅛-in. thick in a single run can regularly be achieved. It was explained at the demonstration that this process does not require a supply of compressed air, the whole of the mechanism is embodied in the T.L.B. mixing chamber which is incorporated in a standard oxy-acetylene gas torch. The Dewrance T.L.B. attachment has been added between the oxy-acetylene valves and the extension piece. The powder mechanism consists mainly of the powder control valve assembly, the powder hopper, and the T.L.B. powder nozzle. Due to the gas stream emerging from the T.L.B. nozzle, a slight vacuum is created at the tip of the nozzle, which sucks powder into the powder chamber. This draws powder from the hopper through the powder control valve into the gas stream. The powder then flows with the gas stream to the tip of the welding torch where the gases are ignited in the normal manner. The flame adjustment and preheating are stated to depend very much on the base material and on the composition of the material being deposited. Flux is not normally required, and the depositing is carried out in one stage without floating after depositing. The finished deposition is much smoother than when deposited by the conventional rod method, and the thickness of deposit can be controlled more readily, thereby effecting a saving in the amount of material used, and reducing the machining to a minimum after depositing. The welding of mild steel to mild steel has been carried out, but this welding has been limited to relatively thin sheets as the process is not competitive with the normal electric welding using heavy rods for this duty. Brasses, bronzes and copper can also be readily deposited, and the process is particularly suited where dissimilar metals are being used. Some typical applications shown at the demonstration included a number of inlet and exhaust valves which had a nickel boron silicon hard facing deposited on E.N.53 and E.N.54 valve steel. The valves were subsequently tried in service and found to be entirely satisfactory. The process has been developed to use a wide range of Dewrance cast metal alloys; a few of these with approximate analysis are as follows :—

Alloy 650/18 Contains	Alloy 650/40 Contains	Alloy 650/52 Contains
Chromium	Nickel	Nickel
Cobalt	Chromium	Silicon
Nickel	Tungsten	Boron
Molybdenum	Cobalt	
Iron		

The process is claimed to be equally successful using stellite, colmonoy, or any other powdered hard facing material.—*Engineering and Boiler House Review, July 1955, Vol. 70, p. 219.*

Doxford Diaphragm Engine in Service

The Doxford engine of the m.v. *Stylehurst* has recently been modified by fitting diaphragms to isolate the lower pistons from the crankcase, and as the modification was effected to a standard type of engine which had been some twenty-six months in service, it is of particular interest to those owners contemplating a similar change to their existing Doxford machinery. This was, in fact, the first Doxford engine to enter sea service with diaphragm chambers. It has four cylinders with a bore of 670 mm., a combined piston stroke of 2,320 mm. and develops in service 4,400 b.h.p. at 115 r.p.m. The m.v. *Stylehurst* was built and engined by Swan, Hunter and Wigham Richardson, Ltd., at their Neptune yard, and was completed in October 1952. The vessel had operated continuously on heavy fuel and the owners, the Grenehurst Shipping Company, decided to adopt Doxford's latest recommendation to obviate the possibility of contamination of the lubricating oil by deposits which might be swept into the crankcase by the lower pistons. The opportunity was taken to effect the modification concurrently with other work carried out on the engine. The new parts had to be manufactured during the course of the modification and, in addition, jigs and fixtures for alignment purposes had also to be made. It is estimated that a similar modification could be effected within thirty days if all the necessary parts were prepared in advance. During the trials, which were of eight hours' duration in very poor weather, full revolutions were maintained, and it is interesting to note that a decided improvement in balance and smooth running of the engine was noted. The engine " critical " was observed to be 4 r.p.m. higher than formerly, but this presented no difficulties. During the trials the deposits from the diaphragm chambers were collected, and these did not exceed one half-pint, although extra lubrication had been given to the pistons during assembly. This experience has established that similar modifications would present no difficulties and practically all the required parts can be manufactured prior to commencing work on the engine. Means are provided to facilitate normal overhauls and the cleaning of lower piston heads in a manner similar to previous practice except that a portable stool is now provided to support the lower piston rod and head when it is in the position for inspection and overhaul on the top platform. The lower piston rod scraper rings may be inspected and overhauled and replaced quite conveniently when the piston rod is removed but in addition, means are provided for lowering the complete gland on the rod with the rod in position if this is desired. The lower end of the piston rod is now bevelled to enable it to be passed through the gland without damaging the scraper rings. To protect the piston rod from damage during crank-chamber overhauls it is suggested that a copper- or lead-lined sheathing should be provided. An analysis of the deposits and liquids drawn from the diaphragm chambers shows as follows:—

Specific gravity at 60 deg. F. ...	0·916
Open flash point, deg. F.	360
Redwood viscosity at 100 deg. F., seconds	375
Fuel, per cent	4
Sediment, per cent	2·85
Neutralization value KOH	1·98
pH value	3·4
Mineral acidity mgm/KOH	0·11
Water	Trace
Other impurities	Trace

It will be noted that the sample contains approximately 4 per cent fuel, together with a high percentage of sediment in the form of combustion soot and some mineral acid which is representative of the type of material which the modification is intended to prevent from entering the crankcase.—*The Motor Ship, July 1955; Vol. 36, p. 151.*

Japanese Cargo Liner

The most recent addition to the fleet of Nippon Yusen Kaisha, Tokyo, is the fast cargo motorship *Sagami Maru.* This ship, which has been built by the Mitsubishi shipyard at Yokohama, recently visited London on her maiden voyage. A sister ship of the *Sagami Maru* is the *Sanuki Maru.* These two vessels, in addition to being the latest, are also the largest and fastest in their owner's fleet, which now numbers not far short of fifty ships. They have a deadweight tonnage of over 11,000 tons and a service speed of 18 knots. The principal particulars of the *Sagami Maru* are as follows:—

Length, feet	510
Breadth, feet	64
Depth, feet	40
Tonnage :	
Gross	9,415
Net	5,376
Deadweight	11,111
Power, b.h.p.	12,000
Service speed, knots	18

The six cargo holds, which are equipped with steel hatch covers on the weather deck, are fitted with a Cargocaire installation. This installation which is of the ordinary (i.e., not the Waterwinner) type, is notable chiefly for the fact that the dampers can be controlled from the bridge. This control is by means of a compressed air system. A feature of the ship, as is often the case with Japanese ships, is the radio equipment, which includes six receivers. Less usual is the fitting of an automatic weather chart printer. This machine, which has been manufactured in Japan, is the first to be fitted in any Japanese merchant ship. It allows weather charts prepared in Japan to be transmitted by radio and printed in the ship on special forms. It is, indeed, probably the first of its sort to be fitted in any merchant ship, though the transmitting end of similar British apparatus, intended in this case for the transmission of photographs by radio, was fitted in the Shaw Savill liner *Gothic* during the Royal Tour of Australia and New Zealand. At the present time the use of facsimile reproduction equipment of this sort at sea is largely confined to aircraft carriers, where it is used for the reproduction of weather charts. The main engine in the *Sagami Maru* is a 10-cylinder Mitsubishi-M.A.N. supercharged Diesel engine. It develops its 12,000 b.h.p. at 118 r.p.m., and is direct-coupled to the single screw. It is a two-stroke engine, with cylinder bore of 780 mm. and stroke of 1,400 mm. The engine is of the KZ78/140 type.—*The Shipping World, 13th July 1955; Vol. 133, p. 49.*

Cylinder Liner Wear

All of the many publications dealing with operation of slow-speed Diesel engines on boiler fuels have made mention of the need for additional centrifuging immediately before the fuel is used. This additional fuel treatment is solely concerned with the provision of more efficient means for the removal of adventitious solid and liquid matter from the fuel. In the marine world in particular the centrifuge has performed this task in a satisfactory manner for many years, and in its " purifier " version it has been highly successful in removing water and solids from conventional Diesel oils. The introduction of the high-viscosity fuels, however, presents a greater problem if fuel cleanliness standards are to be maintained. The more viscous materials will obviously make the task of the centrifuge more difficult, and it has become standard practice where engines are using high-viscosity fuels to fit an additional centrifuge, with means of heating the fuel before centrifuging. The purpose of the heater is to reduce the viscosity of the fuel and, from practical consideration, it is usual in open systems to limit the heating temperature to 180 deg. F. If the average viscosity of marine Diesel oils is considered to be 50 sec. Redwood 1 at 100 deg. F., then heavy fuels whose viscosities exceed 200 sec. Redwood 1 at 100 deg. F. will have viscosities greater than marine Diesel fuel, even though heated to 180 deg. F., and this will result in a reduction in centrifuge efficiency. The provision of the second centrifuge increases the overall efficiency of the treatment to some considerable degree and thus helps to offset the viscosity effect. The majority of slow-speed Diesel engines employ separate cylinder and crankcase lubrication systems, the cylinder lubrication being on a total loss basis. The rate at which oil is supplied in the separate cylinder lubrication system is approximately 1½ gallons per 1,000 i.h.p. per day, and it will be clear that at this small flow the heavy-duty lubricating oil is badly handicapped in comparison with its use in the trunk-piston engine, which relies on splash for cylinder lubrication. It is therefore not surprising that the use of detergent oils is of doubtful value in the large-bore, slow-speed Diesel engine. It is well known that in the high-speed Diesel engine detergent oils are most effective and give considerable reduction in cylinder liner wear. In the high-speed engine, deposits usually take the form of varnish-like films which are well bonded to the piston, whereas in the slow-speed engine the deposits are almost entirely oil-wetted, black carbonaceous materials, the majority of which can be wiped from the piston surface. It is thought that the inability of the detergent lubricating oil to reduce wear in the slow-speed engine may therefore be associated with the nature of the deposits which occur in this type of engine.—*Paper by C. L. Bailey and J. G. Withers, read before the Fourth World Petroleum Congress at Rome, June 1955; The Motor Ship, July 1955; Vol. 36, p. 165.*

High Powered Japanese Ship

The *Sagami Maru* is the forerunner of a new class of cargo liners ordered by the Nippon Yusen Kaisha (N.Y.K.) from the Mitsubishi Nippon Heavy Industries, Ltd., and built at the Yokohama Shipyard and Engine Works. This ship, of 10,936 tons d.w.c. (11,111 metric tons), is propelled by what is claimed to be the highest-powered single-screw Diesel installation in a cargo ship. It is a Yokohama-M.A.N. pressure-charged single-acting two-stroke engine—the first of this type to enter service—having 10 cylinders with a bore of 780 mm., a piston stroke of 1,400 mm. and a maximum continuous output of 12,000 b.h.p. at 118 r.p.m. For scavenging air purposes, there is a vertical double-acting pump at the forward end of the engine, while the undersides of the working pistons act as air pumps. Up to an output of about 90 per cent of the maximum continuous rating—10,800 b.h.p.—the cylinders are scavenged only by these pumps but if a higher output is desired, the engine is supercharged by an independent electrically-driven blower operating in parallel with the scavenging air pump. Thus, the engine is generally similar to the standard single-acting two-stroke unit with after-charge scavenging and employing a rotary valve in the exhaust manifold. With the electric blower brought into service to give the full normal rating, the air passes through an air cooler between the scavenging air receiver and the rotary exhaust valves which prevent leakage of additional air in the initial compression. With the motor-driven blower acting in parallel with the scavenging pumps, the engine develops 12,000 b.h.p. at 118 r.p.m., while without it the economical output of 10,800 b.h.p. (90 per cent) at 114 r.p.m. is attained. Notwithstanding the general trend to use exhaust-gas-driven turboblowers for pressure-charging, the method described has been adopted for the *Sagami Maru* for the following reasons:—

(a) At the output mostly required at sea—10,800 b.h.p.—the main engine can be operated simply as a non-supercharged engine, i.e., without the blower.
(b) Starting and dead-slow running are no different from that of a non-supercharged engine.
(c) The initial cost is lower.

The diameter of the engine piston rod is 230 mm. and the engine-driven double-tandem double-acting scavenge pump has one cylinder with a bore of 1,225 mm. and a stroke of 1,260 mm., the diameter of the piston rods being 95 mm. and 85 mm. Made by the Hitachi Manufacturing Company, the supercharging blower is designed for a normal output of 130 $m.^3$ per min. of free air at a delivery pressure of 2,850 mm. Hg. when driven at 3,600 r.p.m. by a 180 h.p. 440-volt three-phase 60-cycle motor. It may be noted that the air-suction valve is actuated by a ½-h.p. electric motor. The bore of the air suction and outlet of the blower is 400 mm. The air cooler is of the copper-fin tube type with a total tube length of 1,548 mm., a tube diameter of 16 mm. and thickness of 1·2 mm., the fin diameter being 28 mm. and the thickness 6·4 mm. For the air inlet to the cooler, the diameter is 1,400 mm., and the outlet 1,010. The keel of the *Sagami Maru* was laid in October 1954, the launching was on 24th January 1955 and the trials were run in March, the ship leaving Yokohama on her maiden voyage on 25th April. Her principal characteristics are as follows:—

Length overall	509ft. 5in.
Length b.p.	475ft. 9in.
Breadth	64ft.
Depth	40ft. 4in.
Draught, loaded	28ft. 11in.
Corresponding deadweight, tons	10,936
Gross register, tons	9,415
Net register, tons	5,376
Cargo hold capacity (bale) ...	596,000ft.^{3}
Cargo hold capacity (grain) ...	650,300ft.^{3}
Service speed at 10,200 b.h.p., knots	17·8

Constructed under the double special survey of Lloyd's Register and the Nippon Kaiji Kyokai (Japan Marine Corporation), the ship is of almost all-welded construction. Longitudinal framing has been adopted at the upper deck and at the bottom, while the side framing and the remaining decks are constructed on the transverse system. Compartments in No. 4 hold arranged for the carriage of cargo oil or water ballast have a total capacity for 1,419 metric tons of oil. The total amount of fuel oil carried is about 1,650 tons (sufficient for 18,000 miles), while the fresh-water capacity is 346 tons and the ballast capacity is 3,000 tons.—*The Motor Ship, July 1955; Vol. 36, pp. 138-145.*

German Engines-aft Motorship

The Atlas Levante Line of Bremen, one of Germany's best-known Mediterranean-trading companies, has taken delivery of the 3,500 tons deadweight cargo motorship *Adria* from the Seebeck yard of the A. G. Weser. This shelter deck ship has her machinery, bridge and all accommodation aft and represents a new type of vessel for these owners. The two holds are closed by an equal number of long hatches. There is a stump mast at the break of the fo'c'sle, a tall unstayed mast carrying heavy-lift derricks and having a masthouse at its base between the hatches and a goal post mast against the bridge front. These carry a total of eight 3/5-ton derricks, one of 25 tons and one of 15 tons. *Adria* has been built to the highest class of the Germanischer Lloyd and has the following principal particulars:

Length between perpendiculars ...	285ft. 5¼in.
Breadth	45ft. 11⅛in.
Moulded depth to main deck ...	20ft. 0⅛in.
Deadweight capacity, tons ...	3,500
Corresponding draught	19ft. 10¼in.
Cargo capacity (grain)	217,000 cu. ft.
Cargo capacity (bale)	196,000 cu. ft.
Gross tonnage	1,992
Net tonnage	953
Speed, knots	13

The main machinery consists of two Deutz BV6M366 single-acting four-stroke turbocharged engines. These have a bore of 16½ inches, a stroke of 26 inches and an output of 1,250 b.h.p. each at 250 r.p.m. The combined output is transmitted to the propeller through Vulcan couplings and gears. At the above rating (50 per cent pressure-charging, by B.B.C. turboblowers), the b.m.e.p. is 116·6lb. per sq. in. and the fuel consumption is 0·347lb. per b.h.p. per hr. The engines are reversible and started by compressed air, only 35 cu. ft. of air being necessary for each start. The engines are cooled indirectly by a closed fresh water circulation system. Both fresh and sea water pumps are electrically-driven. Auxiliary power is provided by three 120 kW Diesel alternator sets, each driven by a 150 h.p. at 600 r.p.m. trunk-piston A4M 428 Deutz engine.—*The Marine Engineer and Naval Architect, June 1955; Vol. 78 pp. 216-217.*

Paddle Tugs in Dockyards

In a statement by the Admiralty concerning the seven Diesel-electric paddle tugs which are to be built for use in H.M. dockyards, it is observed that paddle tugs have been found by experience to be the most suitable for moving aircraft-carriers and other large warships through dock entrances and in the confined waters of dockyard basins. To enable them to operate under the overhanging sides of aircraft-carriers, the new tugs are being built with hinged masts, squat funnels and raked stems. They will be 145 feet in length (b.p.) with an extreme beam over paddle sponsons of 58 feet, and a moulded depth of 15 feet. Four Paxman Diesel generators will produce the power for two driving motors coupled to the paddle shafts through Renolds chain gearing, with a reduction ratio of eight-to-one. The main generators, motors and control gear are being supplied by the British Thomson-Houston Co., Ltd. Four standard Foden Diesel generators are being fitted for auxiliary services. The paddles will be of the feathering type. Each will have independent drive, but a coupling is arranged on the paddle shafts to enable the paddles to be coupled together when required. Machinery control will be direct from the bridge. The tugs and their machinery are being constructed to the rules and regulations of Lloyd's Register of Shipping to class 100A1 for towing and salvage work. The fitting-out of the tugs will be generally to Ministry of Transport requirements. There will be accommodation for a complement of twenty-one. To enable each tug to operate as a fire-fighting craft a monitor is being fitted on the forecastle deck, and it will be supplied from a 150-ton per hour pump in the motor room. Arrangements have been made to facilitate unshipping of all the machinery and auxiliaries so that these can be readily removed for overhaul ashore and replaced by similar units from a stock of spares. In this way a Paxman or Foden Diesel, or a pump or other unit can be replaced in a very short time. The Paxman and Foden engines will be identical to those being fitted in screw tugs and other craft building for the Admiralty.—*The Journal of Commerce, Shipbuilding and Engineering Edition, 16th June 1955; p. 8.*

Refrigerated Twin Screw Cargo Motorship

The latest addition to the fleet of the Port Line, Ltd., of London, is the twin-screw refrigerated and general-cargo motorship *Port Sydney*, which has been built by Swan, Hunter and Wigham Richardson, Ltd., at their Wallsend Shipyard. The vessel completed successful trials towards the end of February, and is now in service. Built under the special survey of officers of Lloyd's Register of Shipping, for the classification ✠ 100 A.1, but with freeboard corresponding to a complete-superstructure ship with tonnage openings, the design and construction of the *Port Sydney* are also in accordance with the requirements of the Ministry of Transport and Civil Aviation. The principal dimensions and other leading characteristics of the vessel are given below:—

Length b.p.	498ft. 0in.
Breadth moulded	70ft. 0in.
Depth moulded to upper deck ...	42ft. 0¼in.
Draught	28ft. 9⅞in.
Corresponding deadweight, tons ...	11,120
Gross tonnage	9,992
Cargo capacities—	
Insulated, cu. ft.	334,500
General, cu. ft.	295,800
B.H.P.	13,200
Corresponding r.p.m.	112/113
Speed, knots	17

In accordance with modern practice, electric welding has been extensively used in the construction of the hull, riveting only being used for the bilge and sheerstrake, seams and bottom-shell longitudinals, as well as for connecting the frames to the shell and the beams to the deck. There are six cargo holds, three of which, together with the lower 'tween decks, are insulated for the carriage of refrigerated cargo. General cargo is carried in Nos. 1, 5 and 6 holds and lower 'tween-deck spaces, and all the upper 'tween-deck spaces and forecastle. The *Port Sydney* is propelled by twin screws, the power for which is provided by two Wallsend-Doxford six-cylinder, opposed-piston, reversible, balanced-type oil engines, constructed by the Wallsend Slipway and Engineering Co., Ltd. With a cylinder diameter of 670 mm. and a combined stroke of 2,320 mm., the engines are capable of developing a total of 13,200 b.h.p. at about 112/113 r.p.m., and are designed to operate on heavy-grade fuel oil only. Each main engine is fitted with three lever-driven scavenge pumps and a detuning flywheel, of the Doxford-Bibby type, at the forward end of the crankshaft. Distilled water is used for cooling the pistons and the cylinders. The thrust shafts are bolted to the after end of the main crankshafts, with solid couplings and single thrust collars, and are arranged to take the flywheels. The thrust blocks are of Mitchell type, arranged for forced lubrication. The whole of the crank, thrust, intermediate and propeller shafting is of forged ingot steel.—*The Shipbuilder and Marine Engine-Builder, July 1955; Vol. 62, pp. 437–441.*

Corrosion Resistance of Low Alloy Steels

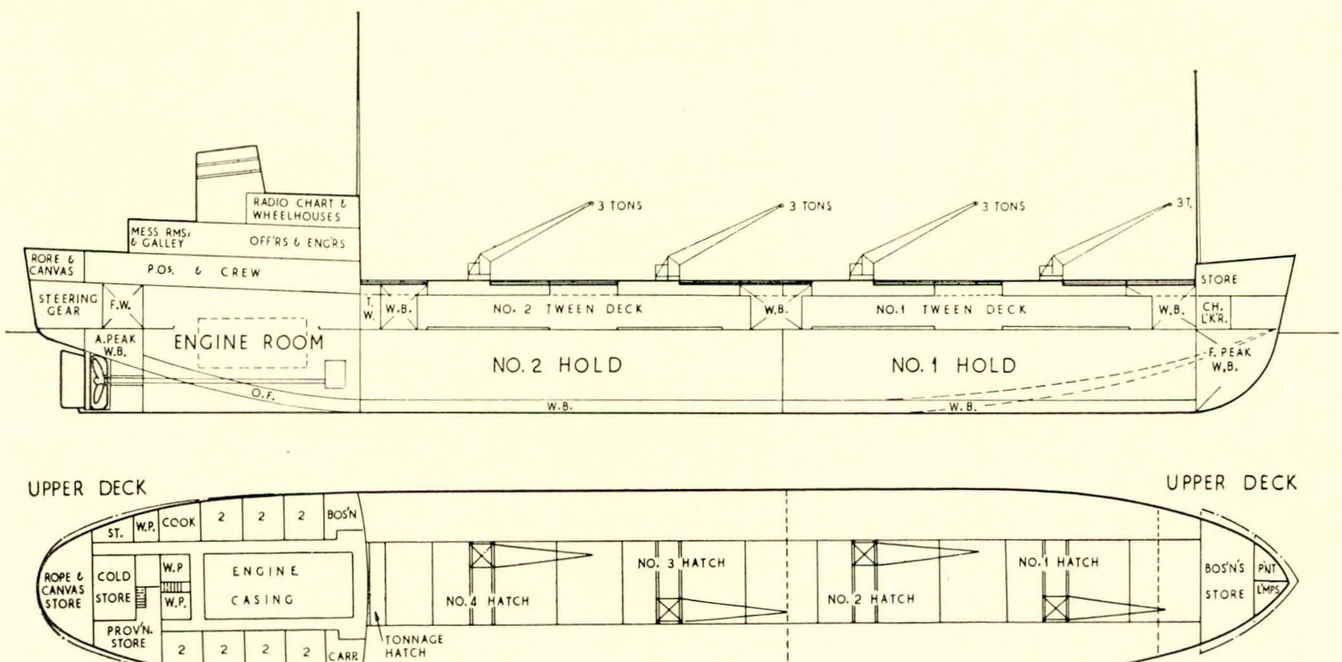
Low-alloy additions are a valuable means of reducing the rusting of steel exposed to the atmosphere outdoors. Under these circumstances the most resistant low-alloy steels rust during the first year or two of exposure at only one-third the rate of unalloyed mild steel. With longer exposure, they appear to still greater advantage. The most useful alloyed elements for this purpose are chromium, copper, and nickel. Aluminium and beryllium may also be of value. Otherwise, none of the wide variety of alloying elements tested was found to have had a marked effect on the resistance of steel to atmospheric corrosion. When immersed in sea-water low-alloy steels show to less advantage, but, under the more or less stagnant conditions described in the paper, the addition of about 3 per cent of chromium to mild steel roughly halved its wastage by corrosion over five years. The corrosion rates of plain carbon steels, over periods of several years in air or in sea water, are not markedly influenced by differences in their carbon contents. The results indicate that high-carbon steels are less corrodible in the atmosphere than low-carbon steels, but more so in the sea. The differences are not great, however, and are about 20 per cent in both cases. Heat-treatment is not a major factor in determining the severity of corrosion of low-alloy steels over long periods of exposure to air and sea water, but it may have a pronounced influence in laboratory tests extending over a few weeks only. There is little difference in the general corrosion of cast irons and plain carbon steels when immersed in sea water, although the former may be seriously weakened by local attack and graphitization. No general conclusion about the respective merits of the two types of material when exposed to the atmosphere can be drawn from this investigation. Laboratory tests of the intermittent spray type yield only fair correlation with the results of outdoor exposure. This is probably because the rusting of the steels in the laboratory test does not proceed sufficiently far for the differences in the properties of the rusts on the various steels to become fully operative. The investigation has demonstrated conclusively that marked improvements in the corrosion resistance of structural steels can be brought about by small additions of suitable alloying elements. This is particularly true for atmospheric exposure.—*J. A. Pope and A.K. Mohamed, Journal of the Iron and Steel Institute, July 1955; Vol. 180, pp. 271–284.*

Standard Ship

Scandinavian, and in particular Swedish, shipyards are the principal exponents of the reduction in ship prices by the adoption of standard ship designs. In Great Britain designs for tankers have been standardized by a number of yards, but for dry cargo vessels the only designs coming into this category have been for general purpose tramp ships in the 10,000-tons class. Some interest therefore attaches to a standard design for a small dry cargo ship of 3,000 to 3,500 tons deadweight which is being offered by the Atlantic Shipbuilding Co., Ltd., the new shipyard at Newport, Mon. This yard is completing the second of two dry cargo vessels of about this size, and is currently constructing another, all for Canadian interests. The standard design, which has been prepared for the firm by Burness, Kendall and Partners, Ltd., embodies a number of unusual features. These, together with careful attention to the specification, have enabled the price to be kept down. The design is intended to meet the requirements of a wide range of trades, particularly in the European market, and it is understood that it has already aroused a good deal of interest among Norwegian and other shipowners. Among the unusual features incorporated in the design, the principal one is the adoption of the naval architects' Hydroconic form of hull construction. This is a double chine form of hull in which no furnacing of plates is needed, while all frames and beams are in straight lengths of steel section. Its use cheapens the first cost of the hull very considerably, and also simplifies and cheapens repairs. It has been used extensively in tugs, and is also being adopted for the third of the three ships for Canada mentioned above. Principal particulars:—

Length overall, feet	305
Length b.p., feet	285
Breadth moulded, feet	45
Depth moulded to upper deck, feet	28
to second deck, feet	20
Draught as closed shelterdecker ...	23ft. 6in.
as open shelterdecker, feet ...	19
Deadweight as closed shelterdecker, tons	3,500
as open shelterdecker, tons ...	3,000

Another feature of interest is the patented design of hatch cover and cargo crane. Each hatch is served by one crane, which is mounted on transverse rails on one section of the hatch cover. The crane provides the power for opening and closing its hatch, while its ability to traverse from one side of the ship to the other allows it to have a shorter jib than would be needed for a centreline crane serving both sides of the ship, giving a cheaper, lighter and lower-powered machine. As can be seen from the accompanying drawings, the ship has a small forecastle and a single deckhouse right aft. This design, together with the use of deck cranes, gives a long clear



General arrangement of the French collier Penavel, *fitted with geared Diesel machinery* area for deck cargo. The hull is welded throughout, and the hatch coamings are continuous fore and aft between the erections, adding considerably to the strength of the hull girder. There are two holds with associated tweendecks, each served by two hatches. Each hold is about 100 feet long, and is clear of pillars and other obstructions. The total cubic capacity (grain) is about 180,000 cu. ft. No deep tank is necessary. Sufficient water ballast is carried in the fore and aft peaks, double bottoms, and in three tweendecks ballast tanks. These latter are arranged beneath the stowage positions of the hatch covers, and can be used for light grain cargoes if required. The provision of ballast spaces so high in the ship reduces the metacentric height in the ballast condition, and gives a more comfortable ship with a slower roll. The vessel is designed for a service speed of 13 knots. The machinery arrangement recommended is twin Diesel engines coupled to a single propeller. With the engines supercharged or not as desired, this gives a most flexible installation which can be adapted to a variety of trading conditions. If the ship is at light load draught as an open shelterdecker, both engines run unsupercharged will give the 13 knots service speed, while one engine will give about 11¼ knots. As a closed shelterdecker, one engine unsupercharged will give about nine knots, two engines without supercharging about 10½ knots, and both engines supercharged 13 knots.—*The Shipping World, 29th June 1955; Vol. 132, pp. 688-689.*

Contamination in Lubrication Systems

Responsibility for preventing contamination of lubricating oil systems rests entirely on the ship's company during normal operating periods and when the machinery components are secured. Four ways of contamination entry are described as follows: 1. *Contamination entry.* Fresh water contaminates the lubricating oil system from gland steam and from condensation on the inside of machinery housing. *Prevention.* Careful regulation of gland steam and continuous use of gland exhausting systems will reduce risk of water contamination from this source. Condensation is a problem with large machinery components such as propulsion reduction gears. As the temperature changes in engineering spaces, air within machinery casings "breathes"; that is, it expands and contracts, and moist air is drawn in. When machinery components cool after being secured, the moisture condenses on inner casings' surfaces. If this water of condensation is not removed promptly, ferrous surfaces in the machinery component will rust. Appearance of rust in strainers indicates that water contamination is causing serious corrosion which, if not immediately arrested, will destroy the machinery component. Therefore, at the earliest possible opportunity the system should be cleaned and the oil purified. Filters and strainers will not remove water. To prevent serious water contamination of lubricating oil systems, centrifugal purifiers on a steam turbine installation should be operated each day while the vessel is underway until there is no indication of water in the oil. The centrifugal purifiers are to be operated as separators, and not as clarifiers; that is, the bowls should be primed with fresh water before the oil to be purified is admitted. With forced-feed lubricating oil systems which are not connected to the centrifugal purifier, periodic checks must be made by drawing off oil samples. To determine the presence of water, allow samples to settle. Water is just as harmful to a feed pump lubricating oil system as it is to a propulsion turbine and gear lubricating system. 2. *Contamination entry.* Salt water from leaking coolers contaminates lubricating oil systems. *Prevention.* Zincs should be kept in good condition on the salt-water side of oil coolers. The overboard discharge and inlet to the lubricating oil cooler is to be shut when the unit is secured, and the water side of cooler drained. Entrance of salt water to contaminate the lubricating oil system is the result of a defective tube or tube sheet somewhere in the system, and this defect must be remedied. Sea water corrosion is more active and harmful than fresh water corrosion. Complete cleaning of the lubrication system is required when a large volume of salt water contaminates it. 3. *Contamination entry.* Damaged and improperly cleaned strainers and filters. *Prevention.* Strainer baskets should be carefully inspected during each watch. A hole the size of a pencil in a fine wire-mesh basket will allow contaminant to circulate and recirculate through the lubricating oil system, leaving a relatively clean strainer and with it a false sense of security. When strainer baskets are opened and removed for inspection, it must be done carefully so as not to let dirt fall into the strainer housing. Wiping off the strainer cover flange with dirty rags invariably scrapes dirt into the lubrication system. The cleaning of strainer baskets in a contaminated bucket of kerosene or Diesel oil is also ineffective. Dirt may be unwittingly transferred to the lubricating oil system if the strainer cover or strainer basket is set on a dirty surface before replacement. Not long ago, during an inspection of a casualty to a feed pump attributable to dirt in the lubricating oil system, some shipyard mechanics were seen to be painstakingly cleaning the strainer baskets in flushing oil. After placing the clean baskets in a large fuel oil drip pan beneath a boiler front, one of the mechanics began wire-brushing the interior of the strainer body casting to remove the adhering core sand. The strainer baskets, which had been forgotten, began rolling around in the drip pan through dirt and fuel oil loaded with abrasive and metal filings. The exterior of the basket and its frame picked up more contaminant from careless handling than was removed in the beginning by the careful washing. 4. *Contamination entry.* Contaminants may be added to a lubricating oil system during replenishing or the addition of make-up oil. *Prevention.* Solid particles have been introduced into lubricating oil systems at the time when new, clean lubricating oil is being added to a circulating system. Most of the contaminant gets into the new oil while it is being transferred from drums to storage tanks and struck down from storage tanks to the circulating systems. Contamination may come from the cover or top of standing oil drums. Dust, dirt, soot, rust, and water are usually present. The drum top should be carefully wiped clean before the drum is opened and vented. The transfer pump, if used, should be wiped clean, particularly the pump suction pipe which is inserted in the drum. If the transfer pump has been used with other liquids, the first lot of lubricating oil transferred will be contaminated by the liquid adhering to the inside of the pump and piping, and this lot therefore should be discarded. The transfer of oil by application of air pressure to oil drums should be discouraged, for dirt and moisture will be blown into the oil.—*G. C. Ambler, Bureau of Ships Journal, May 1955; Vol. 4, pp. 9–13.*

Alignment of Shafts and Drives

The Philadelphia Naval Shipyard has simplified the alignment of propulsion shafting and drive mechanism by the use of an alignment telescope. Alignment in new construction is relatively simple, but in instances where structural misalignment is due to major repair or modifications near the stern of a vessel it poses a more difficult problem. In these cases, the main motor flange, on submarines, or the reduction gear flange, on other installations, is the only established plane for determining proper alignment. To square off tight wire from these small surfaces and thereby obtain true alignment for positioning or machining the stern tube and strut is difficult, time consuming, and produces doubtful results. By mounting an alignment telescope on the motor or reduction gear flange, this shipyard found that the problem is greatly simplified and the end result very accurate. The alignment telescope used is of commercial design and is equipped with horizontal and vertical adjustment to line of sight as an integral part of the instrument. The telescope can be adjusted to bear on targets from eighteen inches to infinity in a true line, and it has an eyepiece adjustable to 90 degrees from line of sight. In use the telescope is mounted in a spigotted fixture bolted directly to the flange. The fixture is rigidly constructed so that a minimum of deflexion exists. After installation of fixture and telescope, the main motor (or reduction gear) is revolved in 90-degree increments and a target is located in the line of sight and at any extreme locations desired. Four points located 90 degrees apart are determined on the target. These points are then bisected to find a tentative centre and the telescope adjusted to bear on this established centre. Again the main motor is revolved and this centre checked at 90-degree increments to ensure that the established centre is true. Intermittent targets can be established by telescopic adjustment without further movement of motor or reduction gear. Alignment of the finished machine strut and stern tube is checked in a similar manner prior to installation of shafting.—*Bureau of Ships Journal, May 1955; Vol. 4, p. 45.*

The Combustion Gas Turbine as Prime Mover for Ships

The General Electric Company believes the time has arrived for the gas turbine to assume its proper role in the marine field. In an industry as conservative as the marine industry, such a statement cannot be safely made without considerable thought and comprehensive experience in the design, manufacture, installation and operation of gas turbines. Ninety-eight General Electric large land units have now accumulated over 600,000 operating hours (over seventy continuous machine years). Users of General Electric gas turbines (ranging from 4,500 h.p. to 7,000 h.p.) have reported that their machines were installed at equal or lower cost than competitive prime movers, and have lower annual operating cost. The gas turbine which this company is now manufacturing for the Maritime Administration for re-engining a Liberty ship is similar to the forty-one gas turbines operating on gas pipelines, but with modification for marine service. These gas pipe-line units have accumulated over 310,000 hours of commercial service, or the equivalent of over thirty-five continuous machine years. It appears that the engine-improvement phase of the Liberty ship conversion programme by the Maritime Administration has real merit, not only for mobilization purposes, but also to produce a new marine propulsion plant which may be installed in any type of commercial vessel. The fuel rate of this vessel will be competitive with modern steam plants in marine service. Some sources have said that the gas turbine produces excessive engine-room heat. This is not applicable to gas turbines in general. Some of the Naval gas turbines rated at less than 500 h.p. have thin sheet-metal cases, which do not permit lagging and may therefore give off objectionable heat. Just as the marine industry recognizes the need for properly insulating steam turbines and steam lines, so it is possible and less difficult to properly insulate marine gas turbines. Use of residual fuel has until recently been a serious obstacle in the path of gas turbines. Ash from such fuel is often very corrosive to high temperature turbine parts and also tends to form slag deposits that reduce turbine efficiency. Vanadium and sodium are two of the most corrosive ash constituents usually encountered, and methods of removing or inhibiting these undesirable elements have been developed by General Electric. Sodium is removed from residual fuels by washing and centrifuging. A water solution of magnesium sulphate thoroughly mixed with the oil to form an emulsion is an economical additive which is satisfactory for counteracting the vanadium. Residual fuel treated in this manner will be used as the fuel in the Maritime Liberty ship conversion.—*A. A. Hafer, The Oil Forum, June 1955; Vol. 9, pp. 229–231.*

32,000-ton Tanker

Designed for carrying oil in bulk and completed twenty-one months after her keel was laid, the *British Victory* was built under Lloyd's Register Rules and Survey to obtain its Class✠100A1 carrying petroleum in bulk certificate. There are thirty cargo tanks (three abreast in the width of the vessel) separated from the cargo hold and oil fuel tank forward by a cofferdam and pumproom, and from the fuel oil and settling tanks aft by another cofferdam port and starboard, and the cargo pumproom on the centreline. The structure of the *British Victory* is all-welded with the exception of the seams of the bilge strake and the upper deck stringer bar, which are riveted. Scalloped longitudinal framing has been adopted in the cargo oil tanks with scalloped transverse framing at the ends. Twin longitudinal bulkheads are fitted throughout the oil tanks and these, together with the transverse bulkheads, have longitudinal stiffeners with a series of webs in each tank. In the wing tanks between the shell and the longitudinal bulkhead at each transverse web, three horizontal struts are arranged in the tank depth. The *British Victory* is propelled by a single screw, driven by a set of geared turbines of Parsons' type Pametrada design, capable of developing 12,500 s.h.p. in service and a maximum power of 13,750 s.h.p. with propeller r.p.m. of 112 and 116 respectively. The astern turbines are capable of developing 60 per cent of the service ahead power. Steam is supplied to the turbines at 420 lb. per sq. in., superheated to 740 deg. F. The h.p. ahead turbine is of the impulse type, and l.p. turbine of the double-flow all-reaction type. The h.p. astern turbine consists of a two-row impulse wheel carried on an integral extension at the forward end of the h.p. rotor with an entirely separate cast steel casing. The l.p. astern turbine comprises a two-stage impulse wheel incorporated in the forward end of the l.p. ahead casing. Gearing is of the double-reaction articulated type, with forged nickel steel pinions, the secondary pinions being made hollow to accommodate the quill shafts. The gear wheels are of the built-up type, with plates bolted to flanges on the rim and wheel shaft. The gear case is of fabricated construction. The main condenser is of Weir's regenerative type, and capable of maintaining a vacuum of 28½ inches at 12,500 s.h.p. with sea temperature of 75 deg. F. The thrust block is of Michell pivoted type, independent of the main gear case, and lubricated from the forced lubrication system; tunnel bearings are of Michell pivoted type, self-lubricated. The propeller is 4-bladed with boss and blades cast solid, of manganese bronze, 21 feet in diameter. All shafting is of solid mild steel, the intermediate shafting being 20 inches and propeller shafting 22¼ inches in diameter. Steam is supplied by two watertube boilers of Foster Wheeler's "D" type with cast iron gilled economizers at 450 lb. per sq. in. working pressure and 750 deg. F. temperature at superheater outlet. The boilers are arranged to burn oil fuel only under a balanced system of forced draught. Suitable electrically-driven forced and induced draught fans are fitted for this purpose, together with an automatic fuel-air control system. Air heaters of the bled steam type are fitted to each boiler.—*Shipping Times, June 1955; Vol. 1, pp. 64–66.*

Rotary Regenerative Heaters for Shipboard Installations

Modern economic practice requires that a marine boiler be fitted with some form of heat-recovery device external to the boiler proper, such as an economizer or air preheater, or both. Such boiler heat-recovery devices, without exception, introduce the hazard of fire caused by heavy deposits of soot and other products of combustion on the heat-transfer surfaces. Fires in recuperative air heaters are not uncommon, particularly under conditions of low load where the dew point is reached and heavy deposits of combustible soot are quickly built. Similarly, economizer fires, though more rare, have been experienced. Danger of such fires can be minimized, of course, by the installation of air bypasses and by keeping the heat-transfer surfaces as clean as possible with the installation of soot-blowing equipment. However, the prime factor in preventing any such fire is in the engineering judgement and experience and care given by the operating personnel. The Military Sea Transportation Service considers that regenerative-type preheaters have, from a design standpoint, provided the means to reduce these fire hazards to a minimum. A fire has occurred in one of the six air preheaters used. This fire occurred in the port boiler in the USNS *Barrett* on the night of 9th–10th February 1954. The fire caused extensive damage to the preheater, requiring almost complete rebuilding. Subsequent to the fire, a thorough investigation was made to determine the cause thereof. This investigation disclosed that the fault was that of the operating personnel rather than that of the equipment. The circumstances leading up to this fire were as follows: From five to eight days preceding the fire, the boiler was operating at very low steam rates. For five days prior to the fire the air preheater had not been cleaned by use of either the gas- or air-side soot blowers. Inspection had been lax, as it was determined that for two days prior to the fire no licensed engineer had examined the cleanliness of the air preheater, although this is a simple procedure. On the afternoon and evening of 9th February, the automatic combustion controls installed became inoperative and the fires in the port boiler were extremely smoky and undoubtedly incomplete combustion obtained. It has been determined that the fire existed at a low level within the heater for probably 4½- to 6-hr. before the engineering personnel were sure of the cause of their high stack temperatures. During this time the boiler was secured and relit, even though the trouble with the automatic combustion control continued. These series of circumstances inevitably led to the fire which the Military Sea Transportation Service experienced. All those investigating this casualty came to the conclusion that the air heater was in this case the victim of circumstances rather than a causative agent. Instructions were issued to operating forces to observe carefully at all times the conditions of the preheaters in order to prevent a recurrence of such a casualty. Since that time no trouble has been experienced, and it appears that if elementary precautions, particularly relating to the cleanliness of the air preheaters, are followed, similar disasters will not occur.—*Paper by R. P. Giblon, G. G. Sharp and C. E. Hoch read at 1955 ASME Spring Meeting, Paper No. 55-S-32.*

Reduction Gear for Marine Turbines

In most turbine-driven merchant ships the turbine revolutions vary from 2,000 to 8,000 per minute, but the propeller speed is only between 80 and 200 r.p.m. The reduction ratios required are, therefore, very large and may be as much as 40 to 1. Electrical transmission is reliable, but it has the disadvantage of being heavy, complicated and bulky. The usual practice of modern designs, therefore, is to utilize double reduction articulated gears. With a compound turbine coupled to double reduction gearing, the efficiency of transmission ranges from 97 to 98½ per cent at full power. Turbine revolutions are increasing, since smaller turbines resist distortion better at higher temperatures, are lighter, and in general are more efficient. On the other hand, propellers tend to be run at lower revolutions in order to achieve higher efficiency with minimum erosion and minimum production of noise and vibration. These factors tend to increase the size of gear-boxes and to make even greater demands on the accuracy and reliability of gearing. Involute teeth cut as double helices on the wheels and pinions are usually employed in mechanical gearing for marine turbine sets. As a rule they are generated by hobbing, and post-hobbing processes such as lapping and shaving are being increasingly employed. In recent years considerable progress has been made both in accuracy of generation and alignment and in instruments for measuring the accuracy obtained. Tests have shown that with improvements in gear cutting and with the adoption of harder materials, greater loadings can safely be employed and that without any other change smaller and lighter gear-boxes are possible. Since its formation in 1944, the Parsons and Marine Engineering Turbine Research and Development Association (Pametrada) has made major contributions to the achievement of a still higher standard of accuracy and reliability in gearing for manoeuvring and transmission systems. The association is studying the basic problems of gear tooth lubrication and materials and the design of gear teeth. In addition, it is working on problems associated with marine gears, such as scuffing, pitting, and other forms of wear. Much fundamental work on the mechanism of tooth contact, profile correction, and evaluation of temperature flash is being carried out through the medium of small-scale disc tests, which simulate under continuous conditions the instantaneous contacts between gear teeth, thus allowing theories to be tried out at minimum cost. By offsetting the horizontal or vertical spindles carrying the discs, it is possible to simulate sliding, rolling, or any desired combination of sliding and rolling. Scuffing of discs has been examined by passing a heavy electrical current through the oil film between the discs and measuring the variation of electrical resistance with slide roll numbers, sliding velocity, electrical current and load. Tests are also in progress to determine the exact effect of oil viscosity and oil quality on pitting. The problem of temperature flash between gear teeth has been examined by means of a detailed study of temperatures involved in disc contact. The next stage is to carry out trials of specially designed gears and of normal full-sized marine gears. This may be done incidentally during the course of machinery trials at sea, or the tests may be made on shore by using turbines to drive the gears and absorbing the power through a brake. The latter system has several advantages, the most important being that gears are driven nearly according to design and service conditions can be very closely reproduced. On the other hand, there are two main disadvantages. In the first place, losses in the gears must be obtained by measuring the input and output torques and subtracting the second from the first. Since these losses amount, at the most, to some 2 per cent of the transmitted power, errors in measurement as small as ±½ per cent can seriously affect the accuracy of the results. The other disadvantage is that the expenditure on power is very large, since the full power required must be generated by the driving turbine and absorbed by the brake. A third alternative is to use the back-to-back principle, which appears to have been first employed as a method of testing gears as long ago as 1910 by a designer in the United States. In back-to-back tests two similar gear-boxes are arranged as a power-circulating system, so that the driving turbine has only to overcome the frictional losses of the system. Whereas a 30,000 h.p. gearbox, tested by the more conventional method, would require a 30,000 h.p. turbine, the use of a power circulation system makes it possible to test two 30,000 h.p. gear-boxes up to full power by using a turbine of about 1,500 h.p. Back-to-back tests of high-powered gearboxes are being carried out at Pametrada to check performance and to measure the effect on efficiency of variations in such factors as viscosity and quantity of the lubricant applied. They are also used—often to destruction—for determining the effects of materials, type of tooth generation, helix angle correction, and other basic factors. —*Shipbuilding and Shipping Record, 2nd June 1955; Vol. 85, pp. 702–704.*

Sea Conditions and Ship Speed

The experimental and theoretical study of ship motions is making significant progress toward further understanding of the above phenomena of pitch and heave in actual sea conditions. The new technique has been developed largely by Mr. M. St. Denis of the David Taylor Model Basin, in co-operation with Professor Pierson of New York University. Significant contributions are also being made by Fuchs and MacCamy at the University of California. The distinctive feature of these methods is the hypothesis that motion in a confused sea need not be considered as a succession of transients, but as the linear summation of responses to the very large number of regular component waves. This theory has not yet been fully confirmed, but it offers real promise. It has given added impetus to the theoretical and experimental determination of ship responses in regular waves of a wide range of frequencies. Such studies are important not only as an approximation of ocean swell conditions, but, more important, because they form the building blocks from which it appears that the response to complex sea patterns can be constructed by calculation. These techniques are beginning to be applied, and model tests in irregular waves are now being used to verify the theory. The studies must first be done in long-crested seas meeting the ship or model head on, but in time the work can be extended to different headings of the ship, to short-crested seas and finally to cross seas representing the superposition of a storm sea and a swell, for example. For experimental work of this type, a square or rectangular tank is needed, with wavemakers along two sides. A particularly important aspect of ship motions which can also be studied best by model experiment in a square tank is the interactions or couplings between the different motions—since they rarely if ever occur independently of one another. Pitch and heavy coupling are considered in ordinary model tests in a straight tank. The most important other cases are yaw-heel and roll-pitch, which occur when the waves are not head-on. These coupling effects have been dealt with quite thoroughly by Grim in Germany. He has shown on the basis of both theory and experiment that if the periods of angular motion—pitch, roll and yaw—are multiples of one another, the effects are very pronounced. For example, a model having pitching period half the rolling period was found to roll heavily in regular head seas. Understanding of the ocean and of the motions of ships has greatly increased. Research has not yet revealed specific means of reducing pitching and heaving significantly and thus permitting higher speeds in bad weather, but it does suggest several lines of attack. Changing the ship's pitching period is a possible approach in the design stage, since it is strongly influenced by ship proportions, as well as by weight distribution. Another promising direction is to increase the damping of motion in order to reduce the amplitudes when resonance occurs or to apply vertical forces at bow and/or stern to cause the ship to follow the wave more closely—rising to the crests instead of diving into them. There are two possible ways of accomplishing these ends: (1) radical modifications of hull forms, perhaps in the direction of using more V-shaped sections, and (2) use of controlled retractible fins at one or both ends of a ship. Both of these ways will be explored in the near future at the experimental Towing Tank at Stevens. Both theoretical and experimental methods of dealing with ship speeds and motions in actual storm conditions have been developed to the point where various possible means of improvement can be evaluated. Hence, higher year-round speeds for surface ships can, it is believed, be confidently expected in the not so distant future. —*E. V. Lewis, Journal of the American Society of Naval Engineers, May 1955; Vol. 67, pp. 303–319.*

Dutch Built Cargo Liner

The *Toreador* is one of the latest dry-cargo vessels built in Holland for Norwegian owners. The ship is designed for the carriage of general cargo, and has been delivered as an open shelterdecker with a draught of 26ft. 7in. but has been designed for a draught of 29ft. 9in. if operated as a closed shelterdecker. The leading characteristics are as follows:—

Length, o.a., feet	471.7
Length, b.p., feet	440
Breadth, moulded, feet	61.5
Depth to shelterdeck, feet	40.7
Deadweight (open shelterdeck), tons ...	8,745
Displacement at summer freeboard, tons	13,575

The vessel is constructed to Lloyd's Register of Shipping classification 100 A1. The ship is of welded construction except for the deck stringer angle steel which is riveted. Longitudinal framing has been applied in the ship's bottom and longitudinal girders are fitted under the shelterdeck. The ship is stiffened for navigation in ice by fitting heavier plating in the forebody, and one heavy stringer. There are six cargo holds, three forward and three aft of the amidships deckhouse. Deep-tank space has been arranged in the after part of one hold for the carriage of fuel oil, water ballast, or vegetable oil. A total of 25,120 cu. ft. of refrigerated cargo space is available. The cargo-handling gear includes one 25-ton derrick, four 10-ton derricks, eight 5-ton derricks, and six 3-ton derricks. The main propelling machinery consists of one 6-cylinder Doxford engine developing 7,500 b.h.p. at 116 r.p.m. which gives the vessel a speed of 17.5 knots. Electrical power is supplied by three Diesel-driven generators, each of which supplies 200 kW D.C. at 220 volts.—*Holland Shipbuilding, No. 12. 1955; Vol. 3, p. 18. Journal, The British Shipbuilding Research Association, May 1955; Vol. 10, Abstract No. 10,208.*

Stress Distribution in Hull During Launching

A number of experiments have been tried to ascertain the strength of a vessel by means of measuring the actual stresses induced on her hull by the load she receives. Such a measurement, however, is by no means easy to carry out, various difficulties accompanying it, let alone the vast amount of time and expense it requires. But as it is known that during the launching of a vessel, a large sagging moment nearly equal to the maximum one which will be experienced throughout her life is induced, we can obtained various kinds of data for the study of the ship structure by taking this opportunity of measuring the stresses on her hull in the course of launching. Of course the load condition in this case is different from that caused by the sea waves; it is also necessary to take into account the difference in rigidity between a vessel on the building berth, in its various stages of construction, and a completed one; and besides, it is extremely difficult to find out how great is the bending moment induced by the load. But, for the purpose of finding out local stress distributions on the hull of a vessel, these matters may safely be left out of consideration except at some particular parts. With a view to obtaining data for the study of the ship structure, the authors have carried out a series of measurements of stresses induced on several parts of the hulls of five vessels launched from the shipyards of the Hitachi Shipbuilding and Engineering Co., Ltd., in the course of 1952 with the aid of wire strain gauges, and have obtained many useful data for the study of the structural strength of the ship's hull. The authors are going to continue similar experiments availing themselves of every opportunity, together with model experiments and theoretical analyses.—*M. Kinoshita and K. Nishimaki, International Shipbuilding Progress, No. 7, 1955; Vol. 2, pp. 134–149.*

Mechanism of Cavitation Inception

The majority of hydrodynamic problems which arise in the design of devices that are submerged in flowing liquids may be treated by means of the classical aerodynamic techniques for incompressible fluids. The application of such methods to the design of ship and torpedo propulsive systems, and for the experimental determination of force coefficients for underwater missiles, is well known. However, a feature which is peculiar to the flow of liquids is the phenomenon of cavitation; that is, when the static pressure at any point in a liquid flow is low enough, boiling occurs and a two-phase flow system results. Aside from its adverse effect on hydrodynamic performance, cavitation is often undesirable because of the noise and physical damage which it produces. The present-day emphasis on the design of machinery with better performance at higher speeds accentuates these problems. Thus it is clear that cavitating flow is of increasing technical importance, for today the designer is confronted with the cavitation problem, more often than has been the case in the past. As with single-phase liquid flows, specific design problems usually are too difficult to permit a theoretical solution even when the basic knowledge is at hand, so the required design information must be obtained experimentally. Furthermore, the prototype is often a large machine and full-scale information is too expensive or difficult to obtain. Consequently, the desired experiments are made on small models. Such test results are useful to the designer only if they can be interpreted in the light of known laws of similitude. Although much is known concerning the modelling of single-phase flow, the onset of cavitation involves properties of the liquid not previously considered. For example, such factors as vapour pressure, surface tension, dissolved-gas content and surface conditions may affect the inception of cavitation. In order to learn more about the physical laws which govern the scaling of cavitating flows, an experimental programme was initiated in the High Speed Water Tunnel at the California Institute of Technology. This paper presents a summary of the progress to date and indicates the direction of the most recent researches. An experimental investigation of incipient cavitation is described and the relationship between these experiments and current theories is discussed. Experiments have indicated that the cavitation number for inception varies with free-stream velocity and body size for both streamlined and bluff bodies. Preliminary studies of the mechanism of cavitation were conducted in order to understand the reasons for the variations in cavitation performance. Photographic investigations have disclosed the effect of the boundary layer on cavitation inception for both streamlined and bluff bodies. In addition, this study has shown that tensions exist in the flow of ordinary water at incipient cavitation.—*R. W. Kermeen, J. T. McGraw, and B. R. Parkin, Transactions, A.S.M.E., May 1955; Vol. 77, pp. 533–541.*

Modern Ore Carriers

The purpose of this paper is to describe the developments in design in recent years of the modern ore carrier. Considerable impetus to the construction of ore carriers in recent years has resulted from the discovery and development of ore deposits outside the United States such as the iron mines in Labrador, Venezuela, and Liberia, and the bauxite mines in Surinam (Netherlands Guiana), British Guiana, Jamaica, and Haiti. This paper is devoted mainly to a discussion of vessels designed to carry these two materials as they seem to be the only ore cargoes available in sufficient tonnage to justify construction of specially designed vessels. Ships designed exclusively for bulk products other than ore, such as grain, sugar, limestone, sulphur, coal, gypsum, alumina, etc., tend to resemble one or another of the basic types of ore carriers and it is not uncommon to carry such products in combination with or alternatively with ore. Combination ships, including the most important version represented by the iron-ore carrier which is fitted for carriage of petroleum products, are included in this discussion. Until recently the bulk of the ore-carrier construction in the United States took place in the Great Lakes region and even today the size and importance of this fleet is not generally appreciated. The basic differences between Great Lakes vessels and ocean vessels are briefly described and the principal characteristics of some recent American and Canadian designs as well as conversions of Victory and C-4 type vessels are tabulated. Over twenty new large vessels have been placed in this trade since the war. It is of interest to note that the general type of self-unloading bulk carrier developed on the Great Lakes for carrying limestone, etc., is being used for ocean transportation of both gypsum and bauxite. The principal American construction of ocean ore carriers since long before the war consisted of ten vessels of 24,000 deadweight tons each for the Bethlehem fleets, starting with the SS *Venore*, delivered in July 1945, and finishing with the SS *Baltore* in 1946. The only American flag ore carriers which have gone into service since that time are the three conversions of C-4 vessels for operations of Ore Transport Inc. in the Labrador trade, which are the SS *Hawaiaan*, SS *Texan* and SS *Californian*. —*Paper by J. J. Henry, read at the Spring Meeting of the Society of Naval Architects and Marine Engineers, 19th May 1955.*

Removal of Rust from Iron

Recent studies by the Engineering Experiment Station, Annapolis, Md., have indicated that rust can be readily removed from iron or steel parts by the use of a solution of citric acid and ammonium chloride. This removal is predicated on the fact that the rust forms an iron-ammonium complex which is quite soluble. The iron and steel of the base metal also react, but very much more slowly, to form the same complex. The mild attack on the base metal can be reduced if ammonium hydroxide is added to the cleaning solution. The cleaning solution is composed of equal parts of citric acid and ammonium chloride dissolved in water. Ordinary commercial quality chemicals are adequate. The two ingredients may be purchased from any available source in drums or barrels such as are used for shipment of boiler compound. The most effective cleaning method consists of immersing the object in a container of the rust remover solution at or near its boiling temperature. The metal should be exposed until the desired degree of removal is obtained. Inspection should be made periodically until the rust has been removed. A 30-per-cent solution should be used. This should be kept in a crock or other acid-resistant vessel. If necessary, a steel tank might be used for a short ime without serious damage. In order to reduce the slight attack of the cleaning solution on the base metal, about 1 pint of concentrated ammonium hydroxide per gallon of the above solution should be added. This requirement is, however, not mandatory because the attack on the base metal will be no greater than that of a well-inhibited, sulphuric acid solution. After the rust is removed, the part should be thoroughly rinsed in clean water and dried. Care should be taken to dry thoroughly in order to prevent the formation of a fresh film on the clean surfaces.—*Bureau of Ships Journal, May 1955; Vol. 4, p. 29.*

Avoidable Accident

One of the primary rules of safety in occupations involving machinery, either afloat or ashore, is: "Never wear loose clothing or allow clothing or other materials such as cleaning rags to come in close proximity to moving machinery." As obvious as this rule may seem, the need for its constant repetition is also apparent. The inherent danger of moving machinery seizing clothing or rags is a danger which is easy to overlook; "familiarity breeds contempt." The possibility of a shirt tail or tattered end of a rag catching in moving machinery parts is probably apparent to the most calloused worker, but the instantaneous and dangerous results which may follow are probably not given much thought. That a loose cuff dangling perilously close to a roller could result, in less than a split second in one's arm being wrapped two or three times around the machinery with compound fractures, torn flesh, and spurting blood, may easily be overlooked. A loose dangling necktie could lead to decapitation or, at least, an awfully sore throat. A first assistant engineer serving aboard a Liberty ship was seriously injured and suffered excruciating pain when he neglected the above principles. It seems that the custom had grown on this vessel of shining sections of line shafting by means of coca mats suspended where they would rub on the revolving shaft sections. The avowed purpose of these mats was to keep the shaft shiny and rust-free. While the first assistant was hanging the coca mat over the line shaft near a shaft coupling, a small piece of the mat jammed in the narrow space between the revolving shaft and the coupling guard. Instantaneously, the mat was drawn under the guard as the shaft turned and the first assistant's right hand was jammed under the guard. Alerted by the man's screams of pain, the engineers on watch stopped the main engine. It was necessary for them to use crowbars on the coupling guard to extricate his hand. The hand and wrist were found to be severely lacerated, with compound fractures, and the vessel was diverted to the nearest port in order that the engineer had medical attention as quickly as possible.—*Proceedings of the Merchant Marine Council, United States Coast Guard, May 1955; Vol. 12, p. 89.*

Cargo Handling

Hatch covers have long been a matter of concern to the designer and ship operator. The time lost in opening and closing hatches and handling covers is costly. So too is cargo damage caused by leakage and injuries to crew and stevedores from some of the difficulties encountered with these units. However, great strides have been made in recent years in designing better and more efficient hatch covers. The mechanically actuated multiple-section covers removed many of the ills associated with the wood and pontoon covers. The latest mechanical covers require no cargo falls, blocks, padeyes, etc., to fold the covers back. Hydraulically operated units which feature accumulators assure quick, simple operation. These modern covers can be used for 'tween-deck as well as weather-deck openings. While there have been improvements in rigging, blocks, winches, safety provision, etc., the basic cargo-handling rigs have undergone little change. For example, the rate per gang on a Liberty ship, with its antiquated steam winches and gear, compares favourably with recent vessels having electric winches. The burtoning of cargo by means of married falls suspended from an in-shore and an off-shore boom is still standard practice. This type of rig is efficient for handling light drafts of cargo. Of considerable importance is the fact that it is understood by stevedores in ports wherever ships call. The conventional burtoning rig has a limitation in that spotting of cargo in the hatch square area is not readily done without considerable adjustment of booms and vangs. The Farrell gear, Fig. 2, consists essentially of a split topping lift (so arranged that part of the topping-lift tackle produces a component tending to pull the boom inboard)

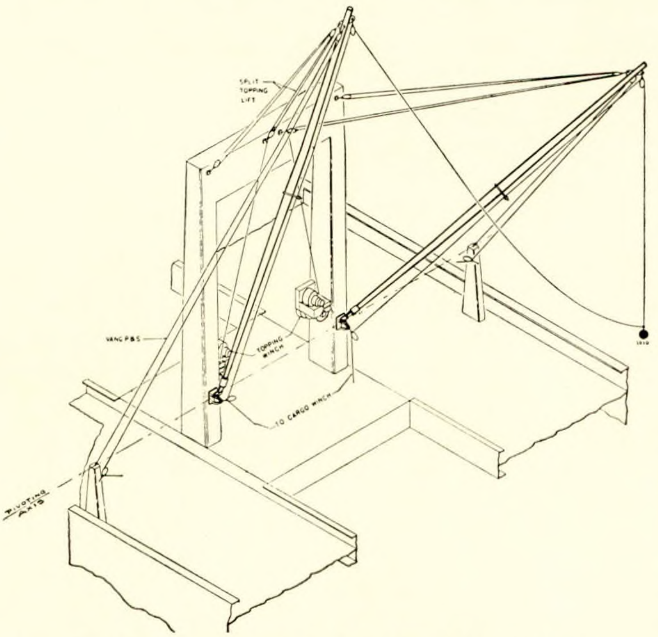


FIG. 2—*Farrel gear simplifies spotting of cargo*

plus an outboard vang to resist this inward pull. The heel of the vang tackle is mounted on a post at the bulwark in a transverse horizontal line with the heel of the boom. This arrangement forms a triangle while the base of the triangle consists of a straight line drawn between the boom heel and the bottom of the vang pendant. Thus, rotation can take place about the base with the head of the boom moving fore and aft on a line parallel to the ship's centreline as the topping lift is raised or lowered. The operator of the vang sets the position of the boom head with respect to the centreline. Power operation of the vangs and topping lift provide rapid movement of the boom head in any direction. By this means it is possible to spot the boom over any portion of the hatch

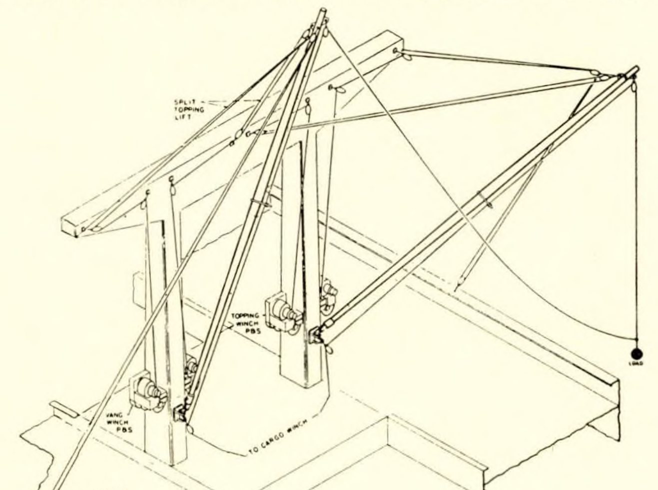


FIG. 3—*Advantages of Ebel gear include a built-in safety feature*

with a minimum of time and effort. Another method of rigging is the "Ebel" gear installed on the *Schuyler Otis Bland*, with satisfactory results (see Fig. 3). This rig requires a transverse member between the tops of the kingposts which extends outboard to a point just short of the bulwark line. The inboard topping lift is run from a winch on the inboard face of the kingpost to a block on the centreline of the cross bracing and from there to the boom head. This portion of the topping lift acts also as the inboard vang. The outboard topping lift runs from a winch on the outboard face of the kingpost to the outboard end of the transverse member and then to a block on the boom head and from there to a block located on the bulwark. This part of the topping lift thus acts as an outboard guy. The boom may be positioned rapidly by manipulation of the two winches and thus affords greater flexibility than the conventional rig. An interesting feature of the rig is the automatic safety control provided by the outboard topping lift and guy arrangement. When the tension in the married falls becomes excessive the boom tends to "ride up." The outboard guy thus warns the operator that the tension in the falls is too great.—*J. McDougall and D. T. Mallett, Marine Engineering, May 1955; Vol. 60, pp. 54–59.*

Boiler Waterside Deposits

The regular inspection of boiler watersides and the correct interpretation of the conditions observed are an essential part of boiler care. By careful examination a boiler operator can usually distinguish between harmful scale and normal sludge deposits. He can also recognize evidence of faulty blowdown, oil contamination, overheating, and corrosion. A partial inspection can be made in the drums and headers, but, for an accurate analysis of waterside conditions, a sample tube should be removed and split. The following descriptions of boiler waterside deposits can help personnel to make the most of boiler inspection. The waterside of a steam boiler in use is never metal bright. The first few hours of steaming produce a thin, shiny, brownish-black, protective metal-oxide finish on all wetted surfaces. This oxide coating can be seen in any boiler tube by brushing away the loose deposit. Surfaces exposed to vapour, such as the upper part of the steam drum, develop a soft, satin-black protective deposit. Both of these coatings are normal and help protect the tubes against corrosion. It is also normal for a small amount of loose deposit to settle on submerged surfaces of the tubes and drums. This deposit includes the unavoidable corrosion products from the feedwater and boiler systems and the reaction products of the boiler compound with traces of sea salts. When sea water leakage and corrosion are kept low and the blow-down is properly proportioned, this deposit is seldom more than 0·005 inch thick at any place in the boiler. Like all sedimentary deposits, the sludge will be thicker in the water drum and lower tube ends. The colour of this sludge ranges from reddish brown at the lower drum ends to reddish black in the steam drum ends of the tubes. When wet it is slimy and easily removed by water washing. When dry it forms a soft non-adherent powder which is easily removed by brushing. The waterside sludges mentioned above can cause serious damage if inadequate blowdown or infrequent mechanical cleaning permits an excessive accumulation. Accumulation of sludge begins in the water drum, but the sludge feeds into the lower ends of the hot generating tubes and bakes into adherent deposits. Heat blistering is the inevitable result, and it is a major cause of naval boiler tube failure. Sludge fouling can be identified by concentrations of deposits in the lower drum and lower tube ends when the upper tube ends are relatively clean. Thick sludge can be distinguished from true salt scale by its mudlike microstructure and also by the absence of a uniform crystalline pattern. The examination of a fragment cross section with a reading glass or other low-power magnifier will generally show this difference. It is desirable to be able to distinguish sludge from scale because the presence of sludge indicates faulty blowdown or cleaning while the presence of scale indicates improper boiler water treatment.—*F. E. Clarke, Bureau of Ships Journal, May 1955; Vol. 4, pp. 23–26.*

Patent Specifications

Metropolitan-Vickers Thermal Power Plant

The power plant shown diagrammatically in Fig. 2 has an application in equipment wherein the primary heating source is a nuclear reactor. The apparatus works in a closed cycle.

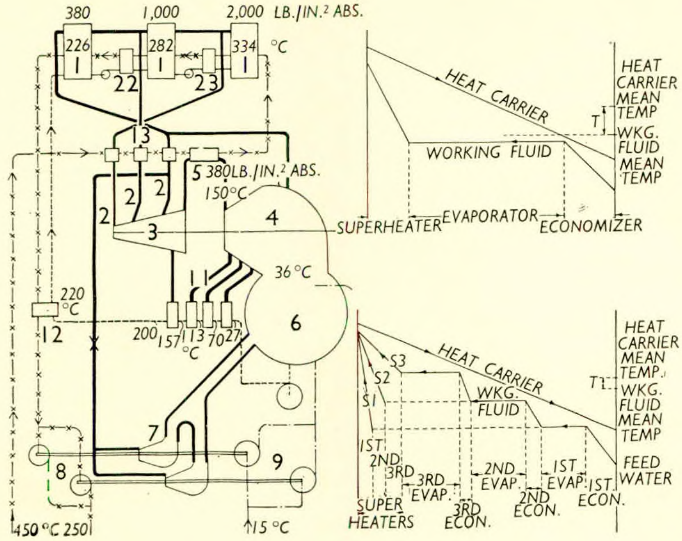


FIG. 2

The heat carrier is circulated through a series of boilers generating steam at successively lower temperatures and pressures, so that in each individual boiler the temperature difference between the heat carrier and the working medium (steam) is reduced below that obtainable with a single boiler. Thus there is only a small temperature change between the heat carrier and the steam at all parts of the system, although the overall temperature change is large. Two of the diagrams show graphically the heat distribution in a plant with a single boiler and with multiple boilers respectively. Three boilers (1) are arranged in series as regards the flow of the heating medium or heat carrier. Steam is supplied at progressively lower temperatures and pressures to the inlets (2) of the successive stages of a turbine (3). The exhaust steam, after passing through a reheater (5), is fed to a second turbine (4) and after expansion is exhausted to a condenser (6). Part of the steam is tapped and fed to auxiliary turbines (7), driving fans (8) for circulating the heating medium, and pumps (9) for the condenser cooling water. The condensate is circulated through fuel heaters (11) and an economizer (12) to the low pressure boiler, and thence through a second economizer (22) to the intermediate pressure boiler, and finally through a third economizer (23) to the high pressure boiler of the series. The heating medium, in its passage from the source of heat, is first passed through superheaters (13).—*Patent No. 727,659, W. Eccles, Metropolitan-Vickers Electrical Co., Ltd., London. The Motor Ship, August 1955; Vol. 36, p. 219.*

Horizontal Opposed-piston Supercharged Engine

An opposed-piston engine in which the exhaust ports are opened prior to the opening of the inlet ports, with the pistons not arranged out of phase, is illustrated in Fig. 3. The pistons operate two crankshafts (5, 6) and at the ends are arranged spiral-bevel gears (2, 4). These gears mesh with bevel wheels (9, 1) operating the driving shaft (7). The gearwheel (1) also meshes with additional spiral-bevel gears (10, 11) secured to shafts (12, 13), which are arranged to drive the fuel injection and lubricating oil pumps. The exhaust ports (20) are closer to the centre of the cylinder than the inlet ports (30) and the former are uncovered by the piston (2) when the crankshaft is rotated 105 degrees from the inner dead-centre. A rotary valve (40) is driven by the shaft (7) and is provided with openings (42, 43, 44). These open the passage (26) between the exhaust chamber (25) and the valve chamber (27) when the crankshaft has rotated 95 degrees. Thus, when the exhaust

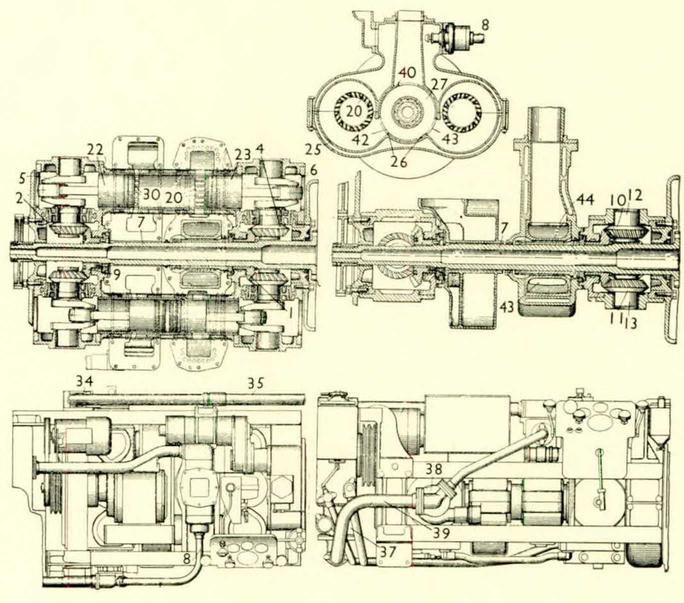


FIG. 3

ports are uncovered, the passage (26) is already open. Exhaust continues until the crankshaft has rotated 128½ degrees. At that point the piston (22) uncovers the scavenging-air ports (30). For the next 77½ degrees of crankshaft rotation the scavenging air passes through the cylinder in an uninterrupted spiral and thence through the exhaust ports. When the crankshaft has rotated 206 degrees the valve (40) is turned to the point where the passage (26) is closed. Thereupon exhaust ceases, although the exhaust piston has not closed the ports (26). For the next 25½ degrees of crankshaft rotation scavenging air continues to be introduced and the cylinder is thereby supercharged. The sea water cooling pump (34) draws its supply through an intake (35) and delivers it to a heat exchanger (37). Fresh water is drawn from the exhaust manifold through a pipe (8) by a pump (38) and is delivered through a pipe (39) to the heat exchanger (37).—*Patent No. 727,233, Research Engineering Corp., La Porte, Indiana. The Motor Ship, August 1955; Vol. 36, p. 219.*

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Marine Engineering and Shipbuilding Abstracts

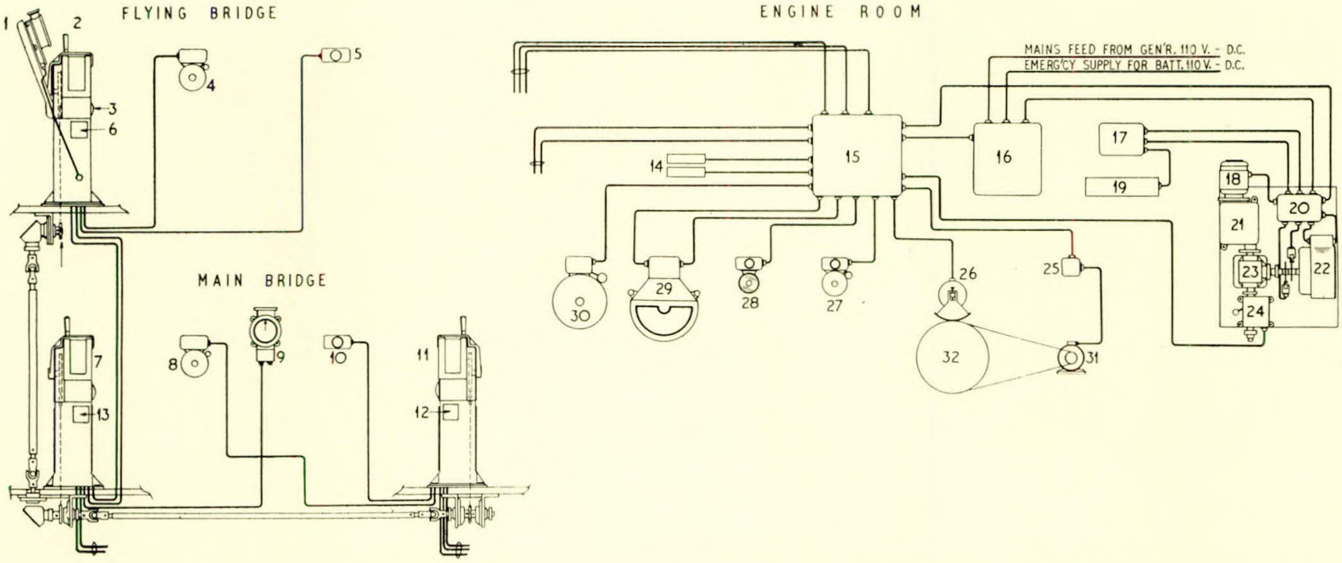
Volume XVIII, No. 11, November 1955

Review of Welded Ship Failures

This report presents a critical review of the available knowledge on structural failures of welded ships which are analysed and classified. A table is included which compares the failures suffered by a number of classes of riveted and welded ships, a total of about 6,000 vessels built since 1938 being considered. The known origins of major failures are listed, and a further table indicates the sea and loading conditions of the ships at the time of these failures. Finally, a survey which has been made of specimens of the steel taken from fractured ships is described. The findings are then summarized and followed by a discussion in which notch toughness, the influence of welding on notch toughness, and crack propagation are considered. The conclusions drawn are as follows:—(1) Brittle cleavage failures in ships were the result of a combination of circumstances. From a practical viewpoint, however, the two main causes of failure were (*a*) design and fabrication notches, and (*b*) a steel which tended to be notch sensitive at the lower operating temperatures. (2) A moderate increase in notch toughness of steel plate over that of wartime steel plate would very substantially reduce the probability of failure. The classification societies and the U.S. Navy have taken action to achieve this. (3) The present situation is that the main hull failures due to fabrication faults far outweigh those due specifically to design faults. Failures from both types of faults have occurred in pre-war and war-time built ships constructed with steel of pre-war quality. It remains to be seen if the improved post-war steels and present fabricating practices are sufficient to eliminate serious failures. (4) A reasonably rigid control and supervision of fabrication must be adopted by repair yards as well as building yards. (5) It is now time that broader and more fundamental aspects of design and construction be considered. The characteristics of brittle fractures in ships have been clearly established, and problems such as those presented by hatch corners and square cut-outs may now be considered to be solved. The improvements in the details of other structural members, such as bilge keels, connexion of tanker longitudinals at bulkheads, bulwarks, etc., have not been wholly successful and further study of these items appears desirable. (6) One of the immediate problems is the ability of various steels and weldments to resist rapid crack propagation, especially in thick plates. The notch toughness of the hull plate must be relied upon as the main line of defence against brittle fractures. (7) Stresses resulting from heating fuel oil or cargo oil have hitherto not been considered as a primary cause of failures. But since several serious fractures have occurred in way of hot oil, these thermal stresses may be more important than at first thought. Appendices describe investigations into residual welding stresses and some investigations on the thermal stresses arising in an LST and cargo ships.—*H. G. Acker, Welding Research Council, New York, Bulletin No. 19. Journal, The British Shipbuilding Research Association, May 1955; Vol. 10, Abstract No. 10,191.*

Synchrostep Remote Control

The two tugs building at W. J. Yarwood and Sons, Ltd., for the Geelong Harbour Trust Commissioners, have been equipped with Chadburns' "Synchrostep" remote control system, which provides electrical control of the main engines from both the wheelhouse and flying bridge positions. These two Diesel tugs have been specially designed for handling and berthing ships, particularly the larger vessels using the port, and including oil tankers which will be operating from the new Shell refinery in Geelong. The tugs, which will be the largest Diesel-engined harbour tugs operating in Australian ports, are also equipped to undertake inter-State tows, having a cruising range of about 5,000 miles. The main propelling machinery consists of a two-stroke British Polar Diesel engine developing 1,310 b.h.p. at 300 r.p.m. The *Spencer Nall*, which is the first of the tugs to be completed, underwent trials on 29th April. Her principal dimensions are 110ft. o.a., 28ft. moulded breadth and 13ft. 3in. depth moulded. The arrangement of the remote control system is shown in the drawing, and it will be seen that the transmitters consist of two single dial instruments in the wheelhouse and one single dial instrument on the flying bridge, all three instruments being mechanically interconnected by means of the latest Admiralty type rod shafting and gearing units. The instruments are of the non-reply type, and the reply indication, in the form of actual propeller shaft revolutions and direction, is given on the main bridge and on the flying bridge by means of two "Record" electrical direction and revolution indicators. The system is supplied with direct current at 110 volts from the vessel's generators, but, should this supply fail, an automatic current failure alarm gives indication of this both in the engine room and in the wheelhouse, and on the flying bridge, and at the same time automatically changes the supply on to an emergency 110-volts battery supply. Incorporated with the system is a normal Synchrostep non-reply engine room receiver which is used, should it be required, to control the engines locally in

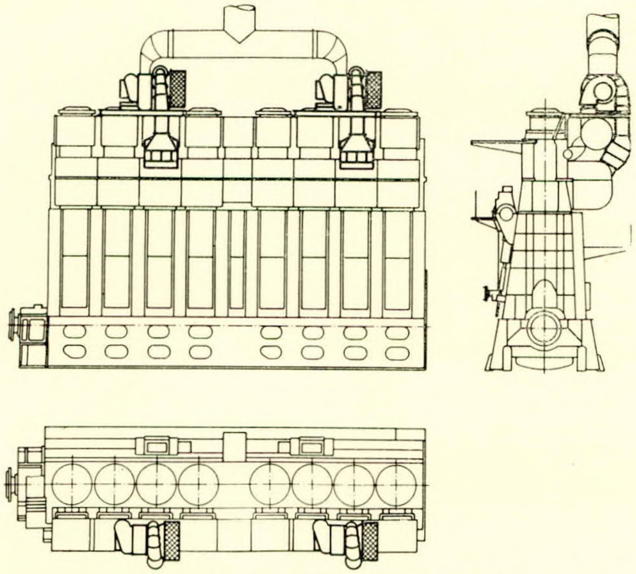


1 Revolution indicator
2 Dummy transmitter
3 Red bullseye lamp
4 Wrong way alarm
5 Green bullseye lamp
6 Buzzer
7 Transmitter bridge controller
8 Wrong way alarm
9 Revolution indicator
10 Green bullseye lamp
11 Transmitter engine telegraph
12 Red bullseye lamp
13 Buzzer
14 Dropper resistance
15 Alarm relay box
16 Supply failure relay and fuse box
17 Reversing contactors
18 Power motor
19 Rectifier unit
20 Junction box
21 Reduction gear unit
22 Hunter unit
23 Three-way reversible gear-box
24 Disengaging clutch
25 Revolution indicator
26 Wrong-way alarm
27 Wrong-way alarm red lamp
28 Green lamp and klaxon
29 Non-reply telegraph receiver
30 Alarm bell
31 Revolution indicator transmitter generator
32 Propeller shaft

the engine room instead of from the bridge. A clutch is provided on the remote control unit to disconnect the bridge control system when required, and red warning lights are incorporated in the bridge transmitters to indicate whether the system is on bridge or local control. The system can therefore be used as (*a*) a straightforward telegraph installation or (*b*) a remote control system. When used as an engine telegraph the hand-operated disengaging clutch on the remote control unit must be disengaged, and while it is in this position the lamps in the bridge control transmitters are not illuminated. On moving the lever of any bridge transmitter to the desired amount of throttle, in the required direction (indicated on the transmitter dials), the order pointer of the engine room receiver will move to the corresponding position, while the alarm bell in the engine room and the alarm buzzer situated inside each column of the bridge transmitter will start to function, and will continue to do so for five seconds, when all alarms will cease to function. When used as a remote control from the bridge, the hand-operated disengaging clutch on the remote control unit must be engaged. This will cause the remote control circuit to be completed, and will illuminate the lamps in the bridge transmitters to show that the clutch is engaged. A movement of the order lever of any bridge transmitter from neutral to the gears engaged position (indicated on the dial) in ahead or astern direction, will cause the remote control unit to move the throttle control on the engine to the ahead or astern position, thus engaging the gears for the direction required. Any further movement of any transmitter lever in the same direction will then cause the remote control unit to open the throttle to the desired amount. If any transmitter lever be moved from say full throttle ahead to say half throttle astern, the remote control unit will first of all close the throttle. When this movement has been completed, and after a delay of three seconds, the remote control unit will change the gear setting from the ahead to the astern direction, and when this operation has been completed, will open the throttle to half throttle position. If any transmitter lever is then moved to the neutral position indicated on the dial, the remote control unit will close the throttle and, when closed, the unit will disengage the gears and position them in neutral.—*The Shipping World, 25th May 1955; Vol. 132, p. 569.*

German Supercharged Two-stroke Diesel

Reports have been coming in for some time that the German firm of Maschinenfabrik Augsburg-Nurnberg A.G. and their licencees have been taking orders for supercharged two-stroke Diesel engines, but up to the present very little information has been made available regarding the design of these engines. M.A.N. have, however, recently released preliminary details of the progress of their development work on this new type of engine and stated that they, and their licencees, have now twenty-six supercharged two-stroke marine Diesel engines on order for various shipowners, three being allocated for the Blue Star Line vessels under construction at the Bremer Vulkan shipyard. As interest in supercharging is concentrated mainly on the engines having high outputs, M.A.N. have concentrated on the supercharging of the type KZ78/140 engine which has a normal output of 900 h.p. per cylinder at 115 r.p.m. (m.e.p. 5·25 atm., m.i.p. 6·2 to 6·3 atm.) Two methods of supercharging have been tried out; one using the scavenge pump and blower operating in series. With this method exhaust gas enters the turbine at an even pressure from a common exhaust pipe. The blowers direct the air into the scavenge pumps which compresses it up to the full charge pressure. With the second arrangement which operates on the pulse system, the blowers are so placed that the exhaust gases take the shortest possible path, thereby making use of the kinetic energy of the gases. The blowers deliver direct to the engine and are assisted only to a small degree by the under side of the piston which acts as a supplementary scavenge pump. Both these arrangements have proved reliable. The cylinder output of the engine can be increased without difficulty up to 1,200 h.p., equal to 33 per cent supercharge, leaving a sufficient margin of safety for a further load up to 1,300 h.p. per cylinder. Most of the engines now on order have been initially designed for 1,125 h.p. per cylinder which is equal to 25 per cent supercharging. There are advantages and disadvantages to both systems. When using the first arrangement of scavenge pump and blower in series, the saving in fuel is not so great as with the pulse system. Furthermore with the first system, the use of scavenge pumps is necessary, and this means an increased cost in production. However, as the greater part of the scavenging air for the M.A.N. engine is supplied



Outline of M.A.N. supercharged engine showing position of blowers

by the underside of the piston, the added cost of production is not too great. The advantages of the first system are that with this arrangement the manœuvring characteristics of the engine remain equal to those of a normally aspirated engine and, in the event of the blowers being out of action for any reason, the engine output is still about 60 to 70 per cent of the supercharged output. Therefore no emergency blower is required. With the second arrangement a lower fuel consumption is achieved and the engine is substantially simplified in construction. By using two or more pistons as supplementary scavenge pumps, the manœuvring characteristics are very favourably influenced. Supercharging is not only restricted to the type KZ78/140 and the smaller KZ70/120 engines, but is also used for the smaller M.A.N. Diesel engines two-stroke naturally aspirated down to 2,000 h.p. output with speed up to 225 r.p.m.—*The Shipping World, 8th June 1955; Vol. 132, p. 621.*

Götaverken Turbocharged Engine

In 1938 Gotaverken developed their two-stroke single-acting Diesel propelling engine and since that time they and their licensees have constructed this class of machinery for installation in two-hundred-and-fifty ocean going vessels, whilst orders are in hand for the engines of sixty ships. Two years ago these builders produced a turbocharged engine designed exclusively for fast Naval vessels, and this unit operates with the highest mean effective pressure of any two-stroke engine. It is termed the "TOP," an abbreviation of turbocharged opposed piston engine, has a cylinder bore of 185 mm., a piston stroke of 230 mm. for each piston, and with 10 cylinders develops 3,250 b.h.p. at 975 r.p.m. The scavenging pressure is 1·75 kg. per sq. cm. (2·5lb. per sq. in), and the corresponding mean effective pressure is 12·15 kg. per sq. cm. or 172lb. per sq. in. The experience gained with this highly supercharged engine has been valuable in the evolution of a new turbocharged two-stroke slow-running design for mercantile ship propulsion and, as a result of the research work that has been carried out for some time past, Gotaverken have now developed their existing mercantile engine as a turbocharged unit for commercial work. It is based on the normal unsupercharged engine, which has cylinders longitudinally scavenged, the scavenging air entering ports which encircle the lower part of the liner (Fig. 5). In each cylinder there is one exhaust valve in the centre of the cover. The fuel-injection system is arranged with two valves for each cylinder. The cylinders are cooled by fresh water and the pistons by lubricating oil. Some of the fundamental characteristics of the engine have been gradually modified. The scavenging system has been developed from the normal scavenging air cylinder of the tandem type at the forward end of the engine, driven by the crankshaft, to under-piston scavenging and finally to the present method with two small double-acting scavenge pumps for each working piston, built into the upper part of the engine entablature and driven from the crosshead. This system is now

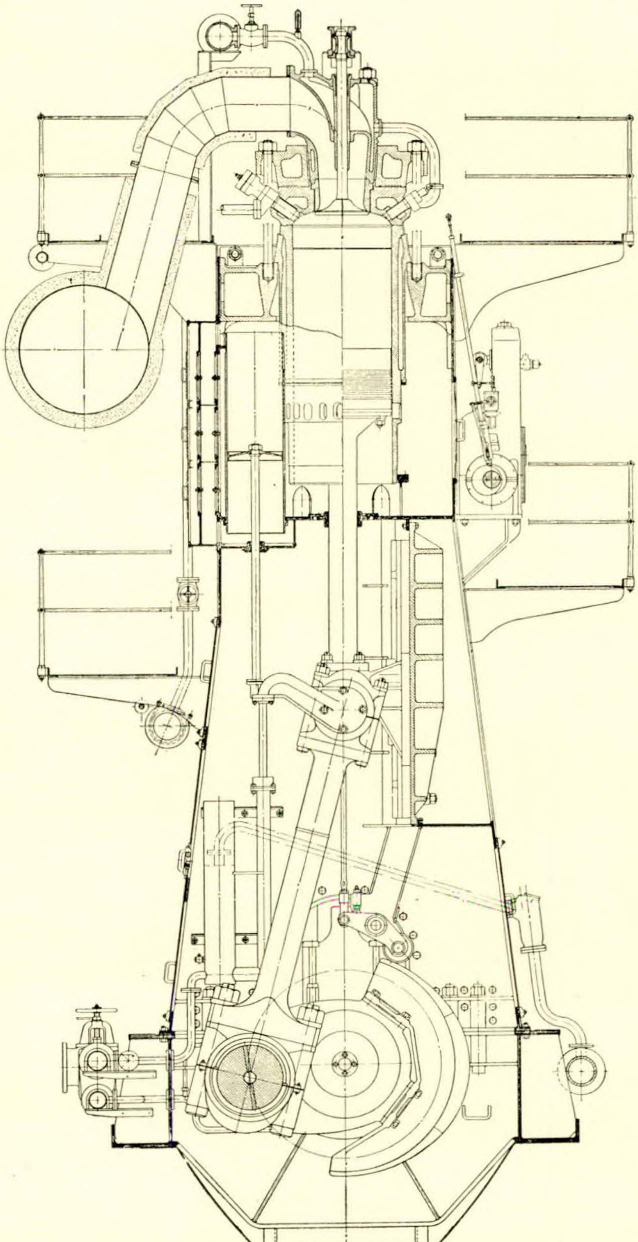


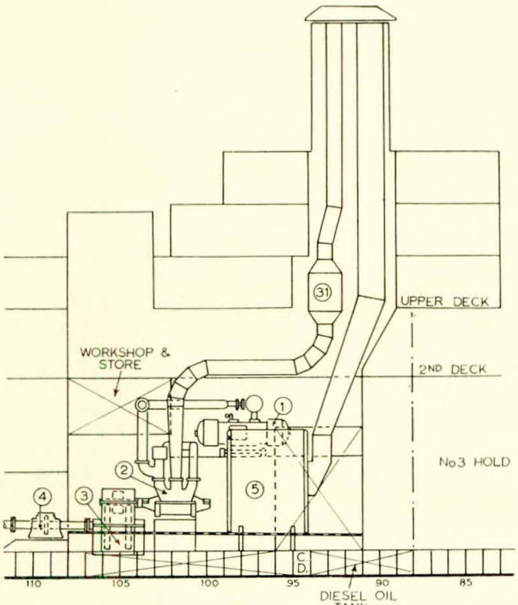
FIG. 5—*Sectional elevation of the Götaverken two-stroke engine. There are two small scavenge pumps for each cylinder driven from the crosshead*

standardized on the non-supercharged engine and, as explained later, is employed in the turbocharged design. The engine has a welded bedplate and entablatures, with cast-iron guides, and the Michell thrust bearing is incorporated in the after part of the bed-plate. The welded entablatures are designed as one box-type unit for each engine cylinder and, through them, the ignition forces are transmitted from the cylinder head to the bedplate without the stay bolts required in a cast-iron structure. In the upper part of each entablature is inserted a cast-steel frame, to which the cylinder and cylinder covers are attached by studs. The entablatures are bolted together, and in their upper parts, adjacent to the scavenging-air belt, they are in communication, thus forming a scavenging-air receiver throughout the complete length of the engine. Around the piston rods and the scavenging-air pistons, stuffing boxes are fitted to prevent contaminated cylinder oil falling into the crankchamber. The Gotaverken designers have aimed at a somewhat higher m.i.p. than is normally used in supercharged two-stroke marine engines. The supercharge is about 35 per cent and the corresponding m.i.p. is 8·8 kg. per sq. cm., or 125lb. per sq. in. The existing scavenging pumps remain as in the non-supercharged engine and are coupled in series with the exhaust-gas turbocharger. Some of the main reasons for this arrangement are of interest. The scavenging pumps (Fig. 5) are of compact design, so that in retaining them there is no sacrifice of space. At the same time, they render it possible for the normal unsupercharged output level to be maintained in all circumstances, and they guarantee sufficient scavenging air, even under slow-running conditions. Should the turbocharger stop, there is a bypass valve enabling the engine to run unsupercharged. The engine revolutions would then drop about 10 per cent, depending upon the hull and propeller design. With this system, available turbocharging units are large enough to limit the number required to two, even in the biggest engines projected. Where only one unit is needed, it may be employed without sacrificing reliability. There is a certain freedom as to the location of the turbocharger, which may be fitted in the most convenient position in the engine-room.—*The Motor Ship, May 1955; Vol. 36, pp. 64–66.*

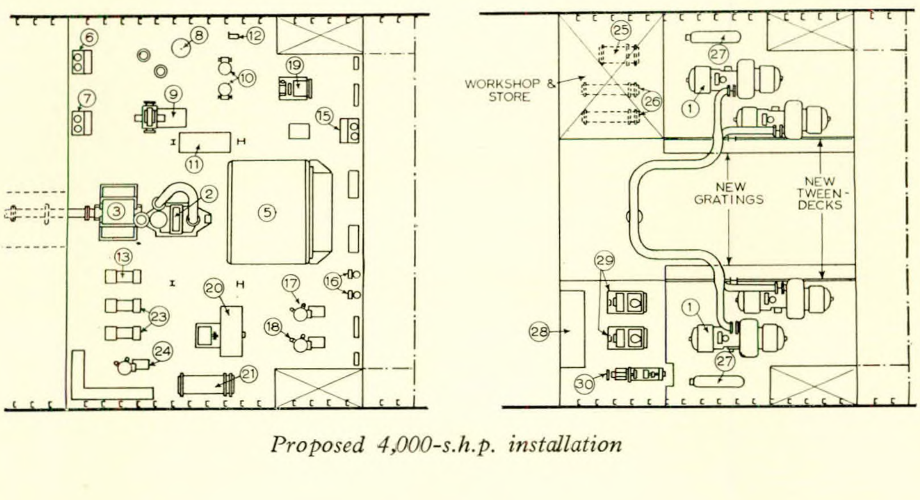
Free-piston Machinery for Liberty Ships

A proposal for the re-engining of Liberty ships, and one which has been carefully considered and presented, is that put forward by the Amsterdam Dry Dock Company, in close connexion with the Pescara-Sigma works. This proposal is to reduce fuel and maintenance costs of the Liberty vessels with the standard triple-expansion engine and two watertube boilers, and to give increased cargo and deadweight capacity by the installation of free-piston gas-turbine machinery. The installation comprises four gas generators placed on new 'tweendecks at the sides of the engine-room and delivering to a single gas turbine developing 4,000 s.h.p. at 5,500 r.p.m. This turbine has five wheels for ahead and one for astern, all housed within the same turbine casing. Reduction gearing brings the speed of 5,500 r.p.m. to a tailshaft speed of 105 r.p.m. For an installation such as has been proposed the operating characteristics at normal full power would be:—

Total power at shafting, maximum continuous, s.h.p.	4,000
R.p.m. of turbine	5,500



1.—Gas generator. 2.—Gas turbine. 3.—Reduction gear. 4.—Thrust block. 5.—Donkey boiler. 6.—Fire and bilge pump. 7.—Ballast and G.S. pump. 8.—Evaporator. 9.—S.W. circulating pump. 10.—Feed pump. 11.—Filter tank. 12.—Evaporating feed pump. 13.—F.W. cooling pump. 15.—Fuel transfer pump. 16.—Fuel pumps. 17.—Fuel purifier. 18.—Fuel clarifier. 19.—Air compressor. 20.—Boiler fan. 21.—Auxiliary condenser with air and circulating pumps. 23.—Cooling oil pumps. 24.—Lubricating oil purifier. 25.—F.W. cooler. 26.—Oil coolers. 27.—Starting air vessels. 28.—Switchboard. 29.—20-kW. steam generator. 30.—20-kW. Diesel generator and compressor. 31.—Exhaust-gas boiler.



Proposed 4,000-s.h.p. installation

R.p.m. of propeller shaft	105
Turbine efficiency (including astern wheel, per cent	85
Efficiency of reduction gearing with thrust bearing, per cent	96
Gas pressure losses in main lines, per cent ...	2
Effective gas pressure, kg. per cm.^{2}	3·16
Gas temperature at turbine inlet, deg. C. ...	440
Specific fuel consumption, gr. per s.h.p. per hr.	182
Service power, s.h.p.	3,600
Corresponding r.p.m. at propeller	100

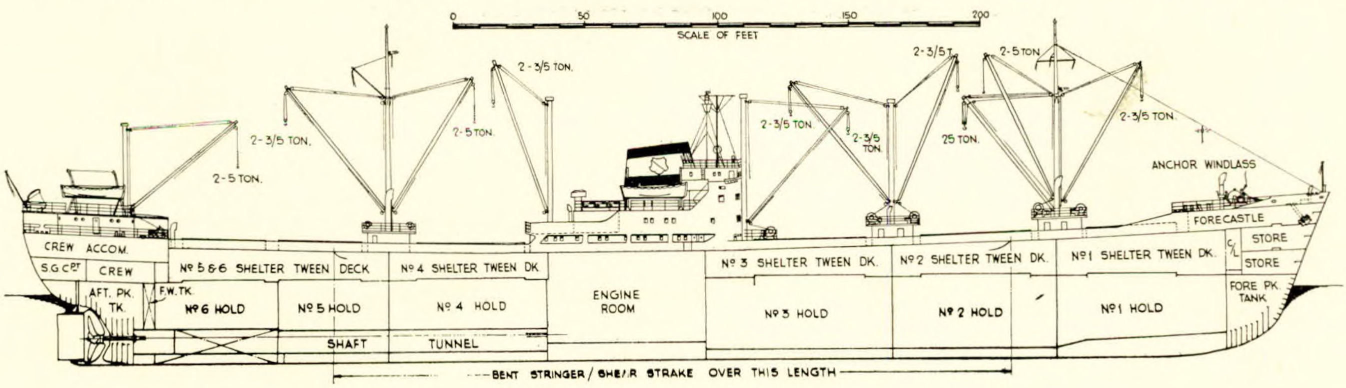
Each gas generator can, if desired, be overhauled in turn during the voyage, the three remaining generators giving 2,800 s.h.p., while two only would ensure 1,800 s.h.p. It is stated that the existing shafting, including the thrust bearing and intermediate shaft bearings, are acceptable as they stand for the increased power by Lloyd's Register, Bureau Veritas and the American Bureau of Shipping. A new propeller to suit the differing conditions will, however, be required. While Lloyd's Register and Bureau Veritas will accept the existing rudder, upper rudder stock and steering gear for the higher speed, the American Bureau of Shipping require a lower rudder stock of increased diameter.—*The Motor Ship, July 1955; Vol. 36, pp. 154–155.*

Norwegian All-welded Motorship

An interesting vessel has recently been delivered by Kockums Mek. Verkstads to Norwegian owners. She is the 11,750 tons d.w. motorship *Sitanja*, owned by the old-established firm of Tschudi and Eitzen, Oslo, for whom the Malmo yard have already built five vessels. Among these is the motor tanker *B.P. Newton* which was the prototype for the standard Kockums 16,000-tons d.w. tanker. The *Sitanja* is an open shelter decker of almost entirely welded construction with longitudinal framing in the bottom and decks and extra strengthening for bulk ore cargoes. A feature of particular interest is that the sheerstrake and deck stringer over some 280 feet of the ship's length amidships are formed from bent high-tensile steel plate. This method of ensuring a stress-free structure, suitable for welded construction has been adopted by the yard as standard practice for tankers and ore carriers, but has not hitherto been applied to normal dry cargo vessels. The *Sitanja* has the following principal dimensions:—

Length overall	496ft. 1½in.
Length between perpendiculars ...	460ft.
Moulded breadth	63ft.
Moulded depth to shelter deck ...	41ft. 6in.
Moulded depth to main deck	31ft. 6in.
Deadweight capacity, tons	11,750
Corresponding draught	27ft. 3½in.
Cargo capacity (grain), cu. ft.	763,740
Cargo capacity (bale), cu. ft.	715,170
Deep tank capacity at 40 cu. ft. per ton (included in above), tons	419
Total bunkers, tons	1,711
Trial speed, fully loaded, knots ...	15

The new ship is a high class general trader, having six holds, three forward and three aft of the machinery, and a pair of deep tanks arranged on each side of the thrust recess. Cargo is handled by eighteen 3/5-ton derricks and a 25-ton derrick is fitted at No. 2 hatch. The entire outfit of deck machinery is electrically-operated and includes fourteen 3-ton and four 5-ton winches all mounted on raised platforms, the after pair on the poop deck being arranged for a warping lead aft. There are four 24-ft. lifeboats. The main engine is arranged to operate on heavy fuels and the *Sitanja* has been equipped with a 370 sq. ft. oil-fired donkey boiler and one 1,675 sq. ft. exhaust-heated boiler. These produce steam for heating the double-bottom and day tanks. The *Sitanja* is propelled by a seven-cylinder single-acting two-stroke K7Z 78/140A Kockums M.A.N. engine developing 6,300 b.h.p. at 115 r.p.m.—*The Marine Engineer and Naval Architect, May 1955; Vol. 78, pp. 175–176.*



General arrangement of the Sitanja's *accommodation and cargo spaces*

Heat Transfer Problems of Liquid Cooled Gas Turbine Blades

This paper opens with a brief survey of the various methods of cooling gas turbine rotor blades, followed by a more detailed account of the different techniques of liquid-cooling, the use of one of which is considered to be inevitable if turbines are to operate as gas temperatures appreciably higher than those are to operate at gas temperatures appreciably higher than those method of liquid-cooling is the closed thermo-syphon system, in which a quantity of liquid is enclosed within each rotor blade to act, through the agency of centrifugal convective forces, as a conveyor of heat from the working-blade span to a more easily cooled root projection in the turbine disc or drum. This method obviates the difficulties of providing the continuous cooling passages within the blades, necessary in forced-convection systems, and the possibility of failure due to blockage of the blade passages by centrifugal deposition of solid impurities in the coolant. This latter problem is present with all open cooling systems, whether operated with forced- or free-convection heat transfer. With maximum gas temperatures in turbines sufficiently high to make the complication of liquid-cooling worth while, it is suggested that, in the closed thermo-syphon system, evaporative heat transfer offers a better prospect of obtaining the necessary rates of heat flow than pure free convection. Experimental investigations of the characteristics of the closed thermo-syphon system are described and these can be conveniently considered in two principal parts. The first consisted of an examination of the operation of the closed thermo-syphon using an apparatus (Fig. 1) in which conditions similar to those of a turbine rotor were approached. The thermo-syphon elements of simple cylindrical form were mounted at their mid points in a wheel which was enclosed in casing forming two concentric annuli, the outer one being fed with hot combustion gas and the inner with cold air. The rotating elements were thus able to pick up heat over their outer halves in the manner of cooled turbine blades and reject it over their inner halves to the cold air representing the blade root projections and secondary coolant. The measured rise in temperature and rate of flow of the air were then used to assess the rate of heat transfer through the thermo-syphon elements. An important observation made during these experiments was that the rate of heat flow through the thermo-syphon under conditions of the rotating rig remained constant, while the quantity of internal coolant, which was distilled water for the large majority of the tests, was varied from that sufficient to fill the whole of each element to less than two per cent of the volume. This is explained by the fact that, provided there is sufficient water present in the tube to fill the space with covered only by a film, and also because of doubts concerning the stability of the film when subjected to Coriolis forces in turbine blades, a series of experiments was made with the heated length full of distilled water, and separately, some organic coolants. The observations on the heat-transfer processes under these conditions are analysed and discussed and, by dimensional analysis, the results are correlated and a criterion is established for the heat flux in the thermo-syphon at which film boiling occurred. This condition sets a limit to the degree of cooling possible with the thermo-syphon. The results from the static apparatus indicate that the internal heat-transfer coefficients in the thermo-syphon for which evaporation is allowed to occur will be effectively infinite in comparison with the

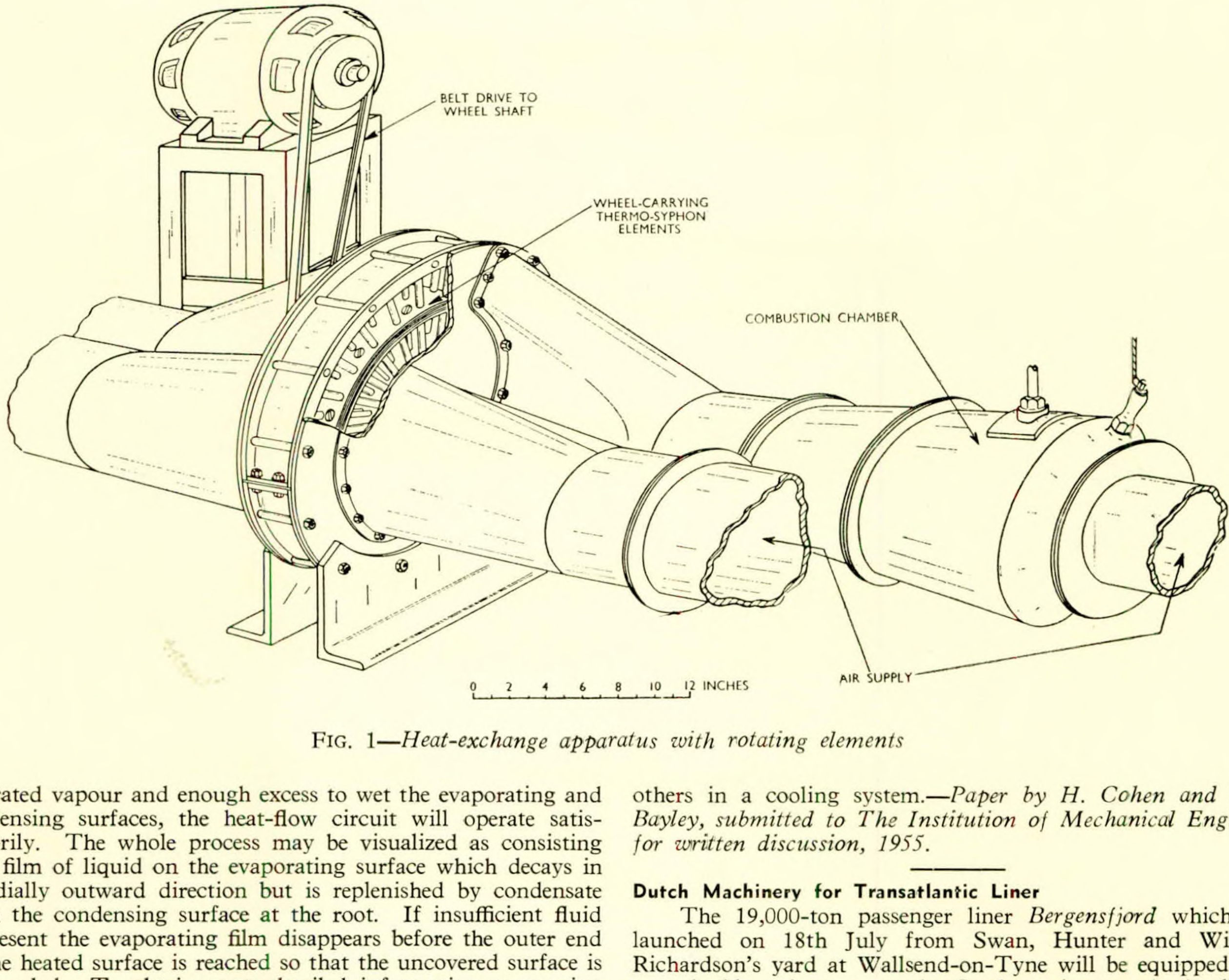
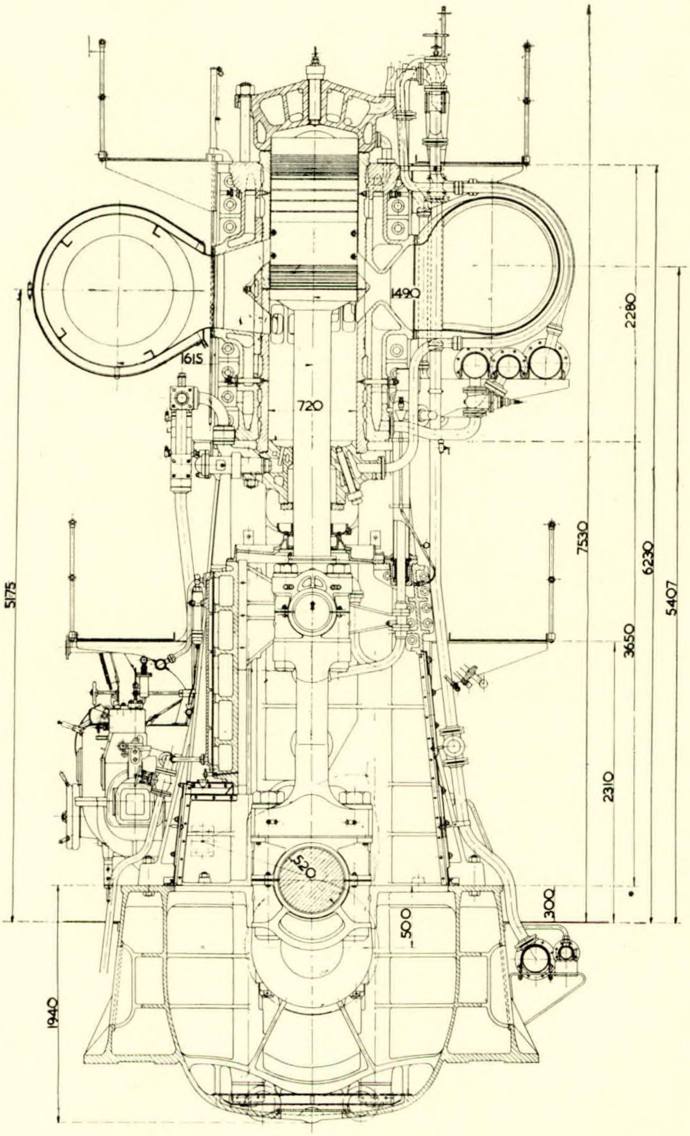


FIG. 1—*Heat-exchange apparatus with rotating elements*

saturated vapour and enough excess to wet the evaporating and condensing surfaces, the heat-flow circuit will operate satisfactorily. The whole process may be visualized as consisting of a film of liquid on the evaporating surface which decays in a radially outward direction but is replenished by condensate from the condensing surface at the root. If insufficient fluid is present the evaporating film disappears before the outer end of the heated surface is reached so that the uncovered surface is not cooled. To obtain more detailed information concerning the heat-transfer characteristics of the closed thermo-syphon, the second part of the experimental programme was undertaken, which consisted of a series of tests on a static apparatus in which gravitational acceleration replaced the centrifugal accelerations of the rotating-element apparatus and turbine-cooling systems. The first experiments provided adequate confirmation of the observation concerning the effect of filling quantity, and results were obtained in which rates of heat flow up to 80,000 B.t.u. per sq. ft. per hr. were transferred through the thermo-syphon elements when these contained quantities of internal coolant of only 7 per cent of the volume of the heated end. The results obtained are shown to be in agreement with a simple theory of the film-evaporation process. Because of the interest aroused by the similarity between the heat-transfer rates in the rotating-element apparatus when the heated walls were submerged in coolant and when they were others in a cooling system.—*Paper by H. Cohen and F. J. Bayley, submitted to The Institution of Mechanical Engineers for written discussion, 1955.*

Dutch Machinery for Transatlantic Liner

The 19,000-ton passenger liner *Bergensfjord* which was launched on 18th July from Swan, Hunter and Wigham Richardson's yard at Wallsend-on-Tyne will be equipped with two double-acting two-stroke Stork engines, each developing 9,300 b.h.p. The *Bergensfjord* is being built for the regular service of the Norwegian American Line between Oslo and New York as well as for cruising, and will be completed next year. The *Oslofjord,* a rather smaller liner (16,500 gross tons), was built in Holland at the De Schelde yard in 1949 for the same owners and she was also equipped with Stork double-acting engines. They have proved eminently satisfactory and economical. The engines for the *Bergensfjord* are in the main, similar to those in the *Oslofjord,* except that they have eight cylinders instead of seven, to give the additional power required for the larger ship with the higher speed of 21 knots. The cylinders are 720 mm. in diameter with a piston stroke of 1,100 mm. and the speed for the output mentioned is 130 r.p.m. The brake mean effective pressure is 4·94 kg. per sq. cm. (70 lb. per sq. in.) and the mean piston speed 4·77 m. per sec. (935 ft. per min.). The scavenging air is supplied by separ-



Sectional elevation of one of the 9,300-b.h.p. Stork engines for installation in the Bergensfjord

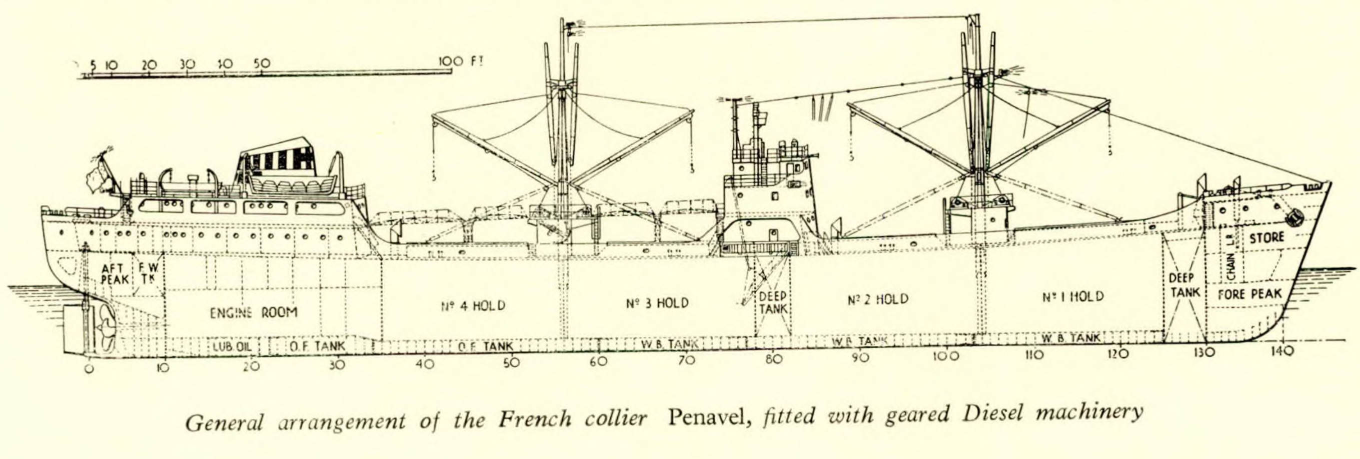
ate electrically-driven blowers and the engine mechanical efficiency is, therefore, high—87 per cent—the mean indicated pressure being 5·67 kg. per sq. cm. (80 lb. per sq. in.), which is normal for Stork double-acting engines. The fuel consumption on test was 154 gr. per b.h.p. per hr. (0·34 lb.) to which has to be added the equivalent consumption of the motors driving the scavenging blowers. The engines are to run on boiler oil, whilst those in the *Oslofjord* operate on Diesel fuel. The bedplate is dished to lower the height of the engine in order to meet the owners' requirements for the engine-room, which is totally enclosed, as in the *Oslofjord*. For the same reason the stroke of the engines was decreased from the normal 1,200 to 1,100 mm., the speed correspondingly being increased from the normal 120 to 130 r.p.m., which meets propeller requirements for passenger liners with a service speed of over 20 knots. At the same time the cylinder blocks were lowered by dishing the top cylinder covers in order to decrease the height required for removal of pistons and piston rods. The results of the *Oslofjord* have proved that this engine type is very suitable for the passenger liner service and accordingly the construction of the *Bergensfjord* engines is the same as in the case of those installed in the *Oslofjord* with only two exceptions. The columns of the *Bergensfjord* engines are welded together from steel plates in order to eliminate the possibility of transverse vibration by an increase of natural frequency. This was not necessary for the seven-cylinder engines of the *Oslofjord*. The shop trials proved the effectiveness of this measure, as the engine on test was remarkably steady. As the owners specified that it should be possible to remove a fuel pump when the engine is running, the pump blocks were not combined in units of four pumps as usual, but each pump has its own separate block. It was proved during the course of the shop trials that a fuel pump block could be replaced on the running engine within the space of half-an-hour.—*The Motor Ship, July 1955; Vol. 36, pp. 138-139.*

French Collier with Geared Diesels

A recent delivery from the shipyard of the Ateliers et Chantiers de Bretagne, Nantes, was the motor collier *Penavel*, built to the order of the Compagnie Nantaise des Chargeurs de l'Ouest. Of the self-trimming type, the vessel is provided with geared Diesel machinery arranged aft. Built to the requirements of Bureau Veritas, the vessel has one continuous deck, a forecastle and long poop, raked bow and cruiser stern. Eight watertight bulkheads are provided which divide the vessel into nine compartments, including the forepeak, two deep tanks, four holds, machinery compartment and after peak. The hull was completely welded. The principal particulars are:—

Length overall	342ft. 6in.
Length b.p.	320ft. 11in.
Breadth moulded	47ft. 7in.
Depth to main deck	24ft. 11in.
Draught	19ft. 4in.
Gross tonnage	4,350
Deadweight tonnage	4,350
Capacity of holds	215,300 cu. ft.
Speed loaded, knots	13

Cargo handling is effected by eight 5-ton derricks fitted to two bipod masts. The cargo winches are arranged on tabernacles



General arrangement of the French collier Penavel, *fitted with geared Diesel machinery* round the masts. Other items of deck machinery include an hydraulically-operated windlass and hydraulic steering gear controlled by telemotor from the bridge. There are also two capstans which can be used for lifting the lifeboats slung in oscillating Loe davits. All holds are fitted with removable MacGregor steel hatch covers and are provided with CO_2 extinguishing apparatus. Propelling machinery consists of two supercharged 6-cylinder M.A.N. four-stroke, single-acting Diesel engines each developing 1,250 b.h.p. at 250 r.p.m. The two engines are coupled to a single shaft, through Vulcan fluid couplings and reduction gearing, the propeller speed being 143 r.p.m. Electric current is provided by two 150 kVA Diesel generators and one 55 kVA generator all of which incorporate an hydraulic pump for the supply of power to deck machinery. —*Shipbuilding and Shipping Record, 26th May 1955; Vol. 85, p. 669.*

Aspects of Indirect Diesel Drives

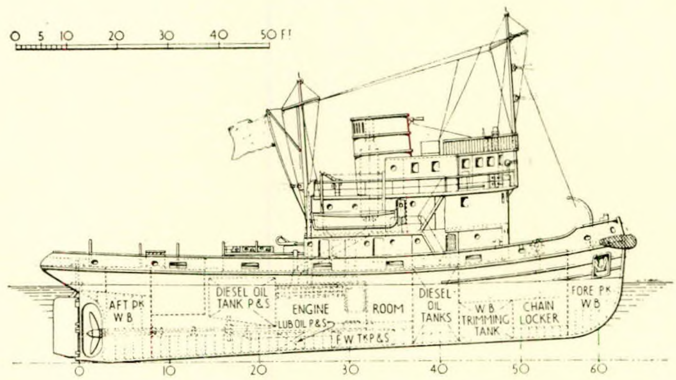
It is often thought that the amount of power obtainable in a multiple engine installation is proportional to the number of engines in operation. This is not so. Let it be assumed that there are for instance three engines, all coupled to one propeller shaft, through gears of 4·7-1 ratio, each developing full power at 750 r.p.m. and capable of giving full torque (full b.m.e.p.) all the way down to idling speed. The propeller will of necessity be designed to absorb full power at 160 r.p.m. and will have a pitch requiring the full torque available from the three engines to turn it. Now, if one engine is cut out, the torque available is only ⅔ of full torque and therefore the propeller cannot turn at 160 r.p.m. any longer without increasing the torque from each remaining engine, i.e. burn more fuel, increase its b.m.e.p. and overloading them. So the propeller r.p.m. will drop and so will the engine r.p.m. until it reaches a value which will only require a torque equal to that capable of being developed by the two engines. Now because power = torque × r.p.m., while the torque from each engine has remained constant, the r.p.m. has decreased from the full r.p.m. of 750 and therefore the power available from each engine is less than its rated value. Fig. 9 shows the power required by the propeller in free water and that available from the engines at various r.p.m. and it is seen that with two engines only 81 per cent of the power of each engine is available, and with one engine only 59 per cent is available. A curve of propeller h.p. with the ship in bollard is also shown and here it is noted that even with all engines on, full power is not available, as it must be limited at 80 per cent, whilst with two engines 65 per cent is obtained from each engine, and with one engine alone only 42 per cent of the engine power is obtainable. The curves also show the saving in fuel obtained by cutting out engines at reduced power.—*J. D'Ottavio, International Shipbuilding Progress, No. 9, 1955; Vol. 2, pp. 203-216.*

Powerful Motor Tug

The illustration shows the single screw motor tug *Vanquisher*. Before construction commenced, model tests were carried out at the National Physical Laboratory. The principal particulars are:—

Length b.p., feet	105
Breadth moulded	28ft. 3in.
Gross tonnage	294
Static pull, tons	14¾
Speed on trials, knots	11

Built to the requirements of Lloyd's Register and the M.O.T.



General arrangement of motor tug Vanquisher

regulations for seagoing duties, the vessel is divided by watertight bulkheads into a number of compartments comprising fore peak, crew accommodation, Diesel fuel tanks, engine room and after peak. The propelling machinery comprises a British Polar Diesel engine developing 1,280 b.h.p. at 250 r.p.m. complete with air compressor, two lubricating pumps, cooling water

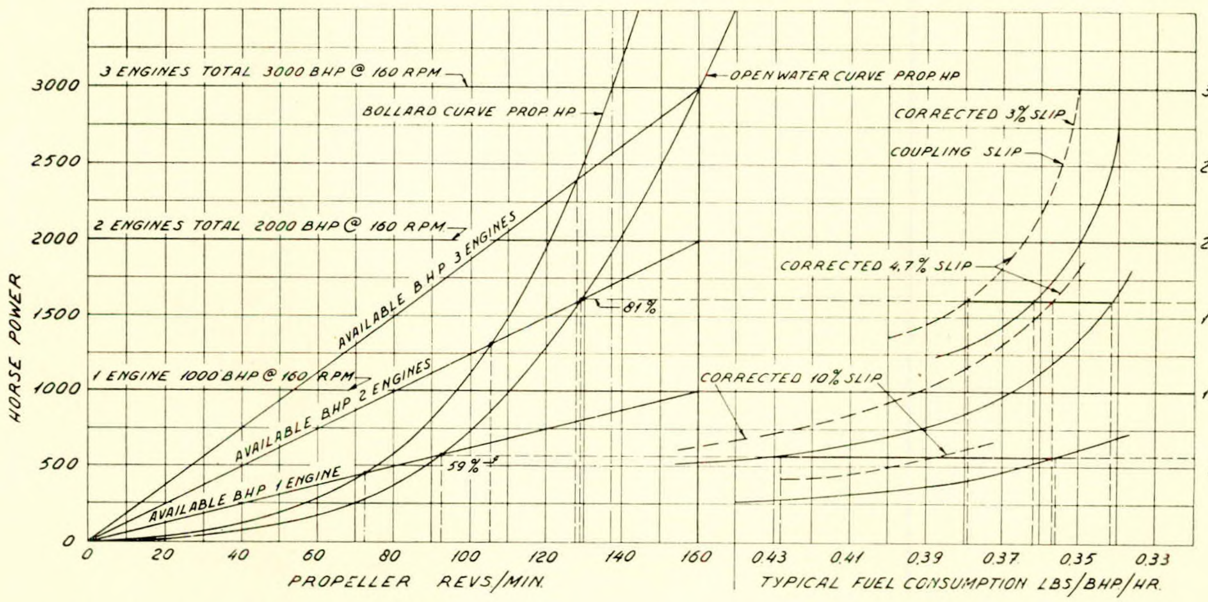
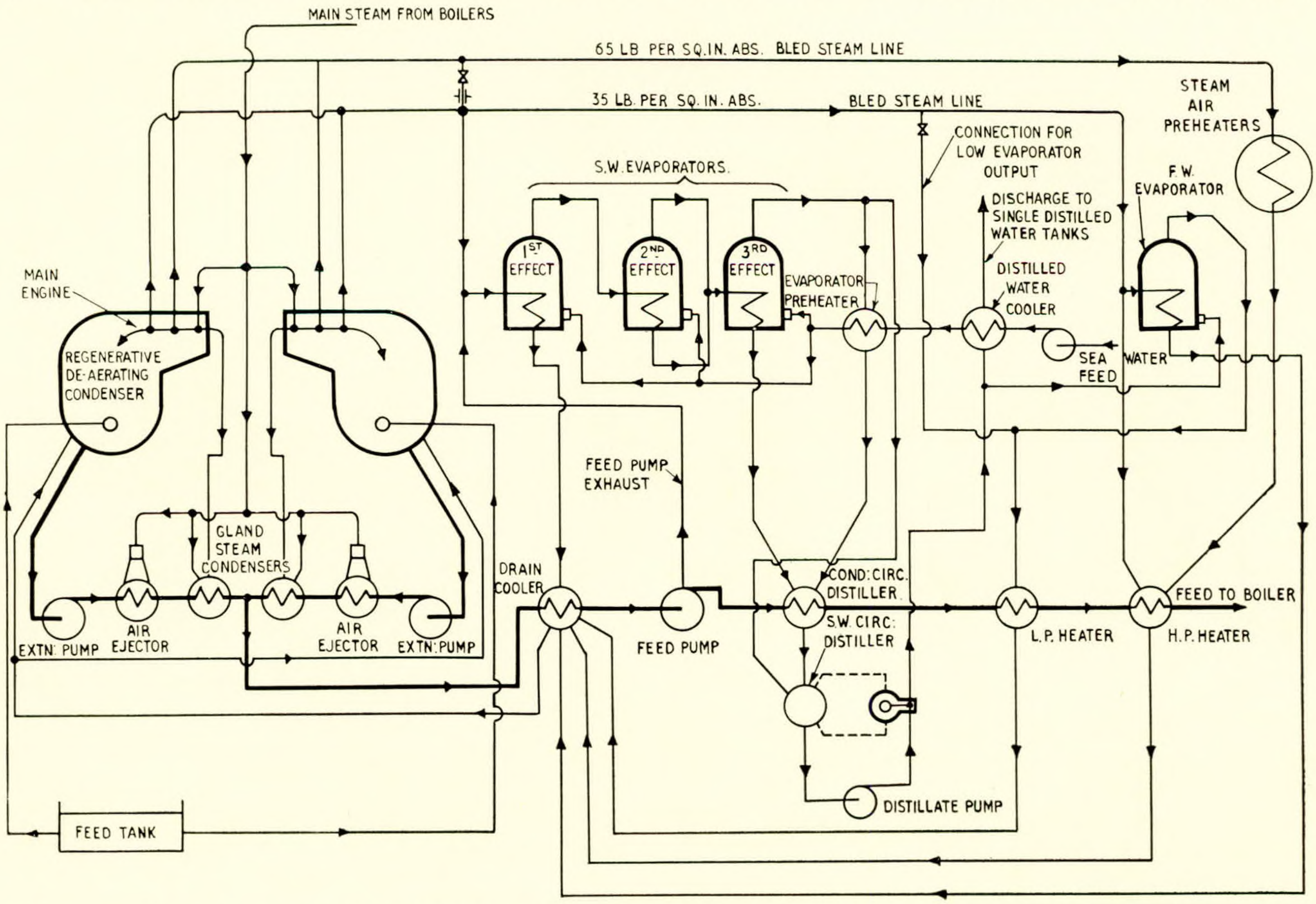


FIG. 9—*Curves of propeller h.p., engine h.p. and fuel consumption for a three engine geared drive*

circulating pump and bilge pump. Two 50 b.h.p. Russell Newbery Diesel engines are coupled to 30 kW 220 cycle d.c. generators, the second set being also clutch coupled to a 284 tons per hour fire and salvage pump. The remaining auxiliaries are electrically driven.—*Shipbuilding and Shipping Record, 30th June 1955; Vol. 85, pp. 842-843.*

Distilling Plant of the *Southern Cross*

The trim of the *Southern Cross* is preserved by transferring fresh water between forward and after tanks and stability is maintained by filling the empty fuel tanks with distilled water. At first thought this latter operation might seem a very expensive one, but the method employed actually enables distilled water to be produced at a price between 6d. and 1s. per ton. Single distilled water for domestic purposes, fuel tank filling and feed for the make-up evaporators is provided by a Weir sea water evaporating plant, the installation consisting of three evaporator units arranged to operate normally in triple effect. Under normal operating conditions, primary steam to the first effect is bled from the second tapping on the turbines at full power or from the first tapping at service and lower powers. Heating steam to the second effect evaporator is the vapour from No. 1 and the heating steam to No. 3 evaporator is the vapour from No. 2 evaporator. Feed regulators are provided. The feed to the evaporator is arranged so that a specific density can be maintained in each shell, the surplus brine overflowing over the Weir ebullition level control to the independent Drysdale brine extraction pumps and discharged overboard. The vapour from the final effect is led through an evaporator preheater, where it heats incoming feed water from 75 to 152 deg. F. before it enters the evaporator shell. The vapour and drains from this preheater then pass to the condensate-circulated distiller. The drains from this condenser then pass to the sea water-circulated distiller. When the vessel is operating at low power and there is insufficient feed passing through the condensate-circulated distiller, a proportion of the vapour is bypassed direct into a sea water-circulated distiller. The single distillate then passes from this heat exchanger to an extraction pump, where it is cooled in an evaporator feed preheater before passing to the liming and chlorinating plant and hence to the domestic tanks. A portion of this distilled water is used as feed to the f.w. evaporators where the water is double-distilled for main boiler make-up feed. The vapour from these evaporators is condensed in the main l.p. feed heater. As stated above, three evaporators normally operate in triple effect, but they are so connected that Nos. 1 and 2 or 2 and 3 can be operated in double effect should this be found necessary, and in addition to the bled steam supply connexions, connexions are provided for using auxiliary boiler steam whereby No. 1 evaporator or No. 2 evaporator can be operated in single effect for harbour duty, the vapour from the evaporator in this case being condensed in the sea water circulating distiller condenser supplied for this purpose. Steam is generated in the high pressure boilers primarily for generating power in expanding to the main turbines, the measure of the power available in the steam being the heat potential for useful work between the condition of the steam at the turbine throttle and the exhaust to the main condenser wherein the latent heat is entirely lost. Where an evaporating plant is supplied with direct boiler steam, this potential for useful work is completely lost and consequently results in fresh water being produced at a very high cost. In the case of the evaporating plant in the *Southern Cross,* the primary steam is taken from a bled point from the main turbines after it has done useful work and the vapour



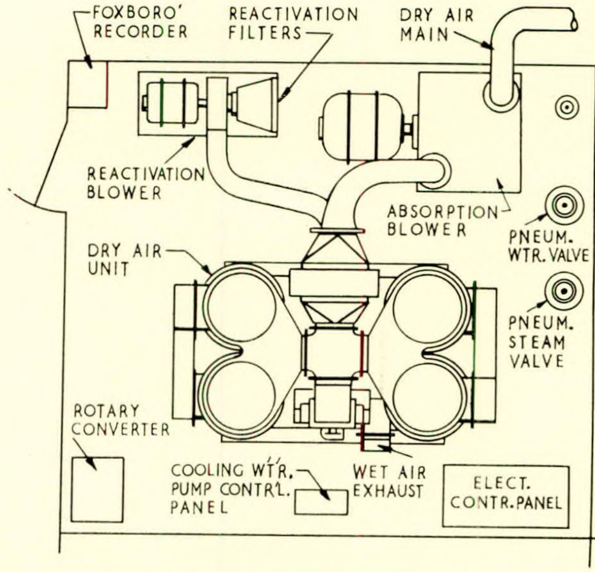
Diagrammatic arrangement of the closed feed system installed in the Southern Cross *showing how the distilling plant is an integral part of the steam cycle* generated in the third effect evaporator together with the distillate drains from first and second effect evaporators are led to the condensate heater wherein the latent heat in the vapour and the sensible heat in the drains are reclaimed in the main condensate and returned to the boiler. Thus, the fresh water produced is only at the expense of the small heat potential for useful work in the steam between the primary steam bleed point and a corresponding lower bleed point which would otherwise be required to maintain the necessary degree of feed heating should evaporator vapour not be available. Virtually, the only heat loss involved is that contained in the brine which is continuously pumped overboard at a rate determined by the density carried in the evaporator shells.—*The Marine Engineer and Naval Architect, June 1955; Vol. 78, pp. 232-234.*

Japanese Motor Tanker for Finland

The single-screw motor tanker *Wipunen,* delivered from the Nippon Kokan Kabushiki Kaisha, Tsurumi Shipyard, Yokohama, is not only the first ship to be ordered by Finland from Japan, but she is also the largest vessel to be owned by Finland. The *Wipunen* is an oil tanker of 19,433 tons deadweight built for Mr. Antii Wihuri of the Suomen Tankkilaiva Oy., Helsinki. As a result of the behaviour of the Cargocaire installation in two previous tankers belonging to Mr. Antii Wihuri, the *Wiikinki* and *Wirakel,* both built in Holland, the owners decided to have a similar installation in this new vessel which was then under construction in Japan. On the first dry docking of the *Wiikinki,* about three days' time was saved in gas-freeing the vessel which at the then existing freight rates went a long way towards paying for the cost of the installation. On dry docking the *Wirakel* after eighteen months' service, all the tanks were in perfect condition, with the exception of one tank which was in very bad condition. The reason for this was that the valve on the Cargocaire dry air line leading to this particular tank had been broken, and no dry air had been able to enter the tank for some considerable time. The Cargocaire unit in the *Wipunen* is installed in the after part of the main engine casing, and a dry air pipe is led to each cargo oil tank and to the cofferdam. Butterworth tank-cleaning apparatus is also installed. The principal particulars of the *Wipunen* are as follows:—

Length o.a.	566ft. 7in.
Length b.p.	541ft. 4in.
Breadth	74ft. 6in.
Depth moulded	38ft. 9in.
Draught (summer)	28ft. 8in.
Deadweight, tons	19,433
Gross tonnage	12,893
Net tonnage	7,233
Engine output, b.h.p.	7,375
Speed, knots	15
Cargo tank capacity, cu. ft. ...	933,089
Dry cargo capacity, cu. ft. ...	17,159

The *Wipunen* has been built to the highest class of Lloyd's Register for carrying petroleum in bulk, and is strengthened for navigation in ice. She has been built on the longitudinal system of framing, and electric welding has been used extensively in her construction. The bilge and sheerstrakes and deck stringers are riveted. Deck equipment includes two steel masts, one forward and one on the navigation bridge and two derrick posts amidships One 5-ton derrick is fitted to the fore mast, one 4-ton derrick and one 2-ton horizontally moving I-beam are fitted to each of the amidships derrick posts for handling the cargo hoses, and there are two 5-ton derricks fitted to the funnel for handling engine parts. Two steam-driven winches are fitted on the main deck, a steam windlass on the forecastle and a steam winch aft. The propelling machinery in the *Wipunen* consists of a single-screw Mitsui-B. and W. Diesel engine having eight cylinders of 740 mm. bore and 1,600 mm. stroke, developing 9,200 i.h.p. at 115 r.p.m. The engine was manufactured by the Mitsui Shipbuilding and Engineering Co., Ltd., Japan. The cylinders are fresh water cooled and the pis-



Layout of the Cargocaire unit

tons oil cooled. Three electrically-driven cooling water pumps of the horizontal centrifugal type, each with a capacity of 275 tons per hour, are installed. Lubricating oil is supplied by two electrically-driven Imo pumps, each having a capacity of 275 cu. metres per hour. One fresh water cooler and one oil cooler are provided. Three Diesel generator sets are installed for the supply of electricity, each set comprising one five-cylinder single-acting four-stroke Diesel engine of Hitachi-B. and W. type, with cylinder dimensions of 240 mm. bore and 400 mm. stroke, developing 280 b.h.p. at 500 r.p.m.; the output of each generator being 180 kW at 230 volts d.c. In addition there is one steam engine driven 60-kW, 230-volt generator, and for emergency purposes one Diesel engine driven 4-kW, 115-volt generator. Cooling and lubricating oil pumps are electrically driven at 220 volts.—*The Shipping World, 6th July 1955; Vol. 133, pp. 19-21.*

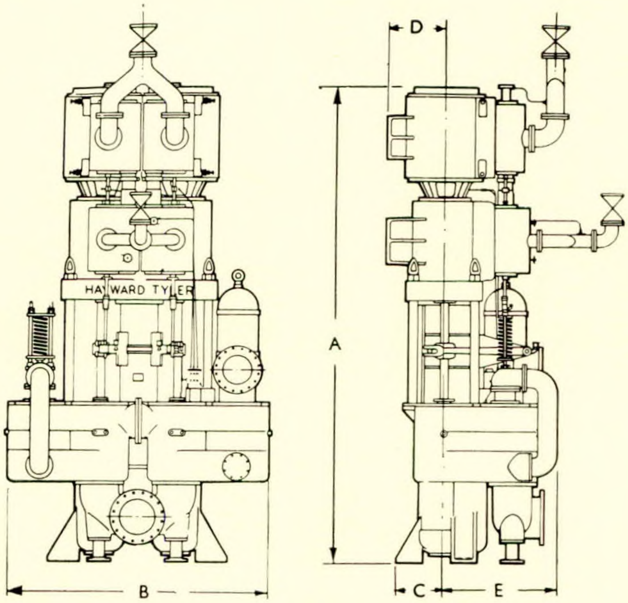
Industrial Xeroradiography

This paper presents a status report on the new radiographic process, xeroradiography. The two basic elements of xerography are the xerographic plate and the developer powder. The plate normally used has a photoconductive layer of amorphous selenium deposited on a stiff metal backing plate, usually of aluminium, although possibly of brass or other materials. In the dark, this surface can accept and hold an electrostatic charge, but when it is exposed to light or penetrating radiation, the charge will decay in an amount somewhat proportional to the radiation. Thus, after exposure, a charge pattern remains on the plate surface in the form of a latent electrostatic image. This image is then developed by dusting it with a finely-divided powder that has been given an electrostatic charge of opposite sign by frictional electricity. The powder will be attracted to the charged portions of the plate, more powder being attracted to the areas of greater charge concentration. The electrostatic charge is usually applied to the selenium surface by passing over it a fine wire assembly held at a high potential so that corona current is emitted. The charging unit is made up of three parallel wires which are maintained at a positive electrical potential well above the corona threshold, and a screen, or grid, of fine wires between the corona wires and the plate. This screen, maintained at a positive potential of a few hundred V. aids in producing a uniform charge of about the same positive potential on the plate. This assembly is driven across the plate in about 10 seconds. The whole assembly is contained in a light-tight box, so that the plate is charged in the dark. The plate, thus sensitized and ready for exposure, is then covered by a dark slide. The plate is exposed in a radiographic set-up in a manner entirely like that used for film. To develop the image, the plate is suspended face downward in a large box, and a cloud of finely-divided white powder is sprayed into the box. The powder is charged by turbulent action in the nozzle of the spray gun. Development time is in the order of 20 to 30 seconds. The negative charge on the powder forms a virtual electrode in front of the plate at some distance from the plate. This virtual electrode creates a weak field between it and charged areas of the plate, so that the powder will be attracted to the plate, areas of higher charge attracting more powder than those of lesser charge. The image can be viewed immediately after the completion of development. Photographic reproductions of a xeroradiograph are uniquely superior to such reproductions of a film radiograph, for the following reasons. In a film radiograph, optical densities from near 0 to over 3 are found. To make a photographic reproduction of this demands more photographic range or latitude than can be recorded on usual photographic emulsions, and any subsequent reproductions from that film are apt to have lost quality. On the other hand, a xeroradiograph does not display its information over a large range of densities, but provides highly visible abrupt changes in density at discontinuities. Over the entire X-ray spectrum, the xeroradiographic plate has a speed about the same as Type A radiographic film.—*R. G. Vyverberg, H. E. Clark, and J. H. Dessauer, Nondestructive Testing, May/June 1955; Vol. 13, pp. 35-39.*

Vertical Compound Cargo Oil Pump

Prompted by the desire for installation that not only takes up less space but is also easy to maintain, some tanker owners are installing vertical cargo pumps in place of the conventional horizontal type. To meet this demand Hayward Tyler and Co., Ltd., Luton, have increased their range of low consumption steam-driven duplex cargo pumps, and are now manufacturing a range of vertical compound steam cargo pumps having capacities of 400, 500 and 750 tons per hour. In the case of tankers which have been fitted originally with horizontal simple pumps, greatly increased pumping capacity can be obtained from existing pump room and boiler installations by conversion to the vertical compound type, provided that the pipeline layout permits this to be done. The pump manufacturers, who are often consulted on tanker pipeline layouts, stress the importance of providing adequate suction line sizes from the pump end and adequate exhaust line sizes from the steam end. This is important when considering the fitting of vertical compound duplex cargo pumps in existing tankers, since too small an exhaust line would exert a back pressure back on the steam end and partially nullify the effect of compounding.

The steam cylinders are made of close grained cast iron and are designed for a maximum steam pressure in the h.p. cylinders of 200 lb. per sq. in. The liquid ends of these pumps are all designed for a maximum working pressure of 150 lb. per sq. in. The construction of the pump is simple and greatly facilitates maintenance, and easily accessible lubrication points and stuffing boxes are fitted. The arrangement of the high and low pressure steam cylinders enables the piston rods to be withdrawn with a minimum of space and trouble, and steam sealing between the cylinders is effected by means of a labyrinth instead of a stuffing box. The piston rods and pump rods are connected by longitudinally-split muff couplings, precision machined to fit grooves in the rods. By this method of construction there is no possibility of the rods becoming unscrewed, and they are easily uncoupled for inspection or replacement. On the liquid end the removal of the buckets and rings, and nests of suction and discharge valves, is simply effected through the large covers provided, and the bucket is so arranged that bucket rings can be changed without removing the bucket and rod.—*The Shipping World, 10th August, 1955; Vol. 133, p. 143.*



The Hayward Tyler vertical compound cargo oil pump

STEAM CONSUMPTION SCHEDULE FOR RECIPROCATING VERTICAL CARGO PUMPS

Size of pump	Capacity (tons water per hr.)	Discharge (lb. per sq. in.)	Steam pressure 100 required for 100 lb. per sq. in. delivery pressure	Steam consumption (lb. per hr.) at 125 lb. per sq. in delivery	140 lb. per sq. in. delivery	140 lb. per sq. in. delivery	Steam consumption in lb. per hr. per w.h.p.
14 and 24 × 14 × 24	400	200	105	5,100	6,400	7,150	45
16 and 25 × 16 × 24	500	200	105	6,500	8,150	9,100	45
18 and 30 × 18 × 28	750	200	105	9,800	12,250	13,750	45

Largest Dutch Merchant Vessel

Recently the largest vessel in the Dutch merchant service sailed from the yard of Wilton-Fijenoord on official trials. This was the twin-screw factory ship *Willem Barendsz* of 26,152 tons d.w.c., with a displacement of 44,300 tons, ordered by the Nederland Maatschappij voor de Walvisvaart N.V., under the management of Vinke and Co., Amsterdam. The principal dimensions are:—

Length o.a., metres	205
Length b.p., metres	190
Breadth, on blubber deck, metres ...	27·5
Depth to blubber deck, metres ...	19·0
Deadweight, tons	26,152
Corresponding draught, metres ...	10·7
Displacement (approximately), tons	44,300
Speed, knots	13-14

The keel was laid on 2nd November 1953, and the vessel was launched on 20th November 1954. The keelplates and adjacent plates have a length of 9·5 metres, a breadth of 2 metres and a minimum thickness of 30·5 mm. Once the keel strake was constructed, building was continued with prefabricated sections, these having an average size of 10 metres by 12 metres and with weights from 20 to 70 tons. The extra-heavy sections were put in place by two cranes each capable of lifting 40 tons. The Wilton-Fijenoord yard is laid out to facilitate the use of the most modern shipbuilding methods: many sections were pre-fabricated in the welding shops and then transported by heavy-duty trucks to the building berth. Particular care is essential in lowering these units into position, as once in place, only a slight degree of movement is obtainable by the use of stretching screws. Two cranes were required to lift the bow assembly into place, a specially constructed yoke being used. At the time of launching, the weight of the ship was estimated at 12,000-13,000 tons, the strength and blubber (or flensing) decks being completed and all the factory plant installed. Specially constructed for navigation in ice and to the requirements of Lloyd's Register, the ship is of welded construction throughout with the exception of the sheerstrake, bilge and framing fore and aft. Longitudinal framing has been adopted with transverse framing forward and aft of the tanks. The factory plant, by Kieler Howaldtswerke, extends over practically the whole length of the ship. There are nineteen boilers, three of them for blubber and the remainder are bone boilers, rotated at a speed of 3 r.p.m. by an electric motor running at 720 r.p.m. and driving the boilers by chain. The factory requirements place a great demand on the electric- and steam-generating plant. There are eight Scotch-type oil-fired boilers placed on a flat above and aft of the main engines, while on the port and starboard sides of the machinery space is an extensive array of three-stage sea-water evaporating units. The boilers, arranged in lines of four athwartships, are of Marshall and Anderson manufacture, each with three furnaces, a length of 3,759 mm., a diameter of 4,950 mm. and capable of producing a maximum of 10,000 kg. per. hr. of steam at 15·8 kg. per $cm.^2$ Each has a heating surface of 350 m^2. The four Weir-type fuel pumps have a capacity of 50 m^3 per hr. and are steam driven. Forward of the boiler room are the two main engines, each a Wilton-Fijenoord-M.A.N. six-cylinder unit with a bore of 780 mm. and a piston stroke of 1,400 mm. The output of each is 5,250 b.h.p. at 112 r.p.m. These engines are generally of standard M.A.N. design with after-charging and with rotary exhaust valves. Surprisingly, for an installation of such a relatively high output, no arrangements are made for the burning of boiler oil in the main engines. Lips-type three-bladed bronze propellers are fitted, each with a diameter of 5,300 mm. and a surface area of 7·659 m^2, while the cast-steel spare propellers are of the same make. When operating as a tanker only, i.e. with no factory load, electrical requirements are met by two Smit 300-kW., 220/380-volt, 568-amp., three-phase generators, each connected to a 5·5-kW., 55-volt, 100-amp. generator, and driven at 375 r.p.m. by a M.A.N. seven-cylinder Diesel engine developing 440 b.h.p. The cylinders have a bore of 285 mm. and the piston stroke is 420 mm. With the factory in operation, the additional load is met by four turbo-generators, each comprising a 600-kW., 220/380-volt Smit generator driven by a Borsig back-pressure steam turbine designed to operate with steam at a pressure of 15 kg. per cm^2 and 200 deg. C., the exhaust steam at 5 kg. per cm^2 and 158 deg. C., then passing to the factory for further use. These turbines have a maximum speed of 7,000 r.p.m. but are geared down to 1,500 r.p.m.—*The Motor Ship, August 1955; Vol. 36, pp. 201-204.*

New Bulk Carrier

The first of a fleet of six ships being built by various shipbuilders in the United Kingdom for the Sugar Line, Ltd., has been completed and delivered from Scotts' Shipbuilding and Engineering Co., Ltd., Greenock. This vessel, the *Crystal Cube*, and her sisterships, have been designed by Captain F. H. Formby, technical director of the Athel Line, Ltd. Although primarily intended for the carriage of sugar in bulk, the vessels have been planned with a view to carrying any type of bulk cargo, from heavy ores to light grain cargoes up to 58·7 cu. ft. per ton, including timber (with deck cargo), as well as bagged and bale cargoes. The principal particulars of the *Crystal Cube* are as follows:—

Length o.a.	460ft. 8½in.
Length b.p., feet	430
Breadth moulded, feet ...	61
Depth moulded to upper deck	37ft. 3in.
Draught loaded	24ft. 6¾in.
Deadweight, tons	9,740
Gross tonnage	8,679
Net tonnage	4,886
Service speed, knots	12½

To give the maximum available cubic capacity with a full form, the machinery in the *Crystal Cube* has been placed aft, and the hull has been designed with two longitudinal bulkheads running from the engine room to the forward bulkhead of No. 2 hold. The bulkheads have been stopped at this point, in view of the fining of the forward body interfering with suitable stowage space in what would be the wing spaces of No. 1 hold. The longitudinal bulkheads have been so placed that the full bulk cargo of about 9,000 tons can be carried in the Nos. 2, 3, 4 and 5 centre compartments plus No. 1 hold, at a storage factor of up to 48·7 cu. ft. per ton. The main propelling and auxiliary machinery is installed at the after end of the *Crystal Cube*. The main engine is a four-cylinder Scott-Doxford opposed-piston Diesel with a bore of 600 mm. and 2,320 mm. total stroke, and two side lever scavenge pumps. The No. 1 scavenge drive also operates a group of three plunger pumps for water and lubricating oil services. The engine is directly coupled to the shafting and develops 3,300 b.h.p. at 110 r.p.m. The necessary arrangements for burning heavy marine fuel are fitted although the owners intend to run at first on Diesel fuel. The engine is of the builders' normal design incorporating their special arrangement of ring main for the h.p. fuel injection and circulating system and an improved form of scraper gland for the lower pistons. The cylinder jackets and upper pistons are cooled by inhibited fresh water. The engine was constructed for using fresh water cooling of the lower pistons also, through the usual link gear, and the shop tests were carried out on that system, but during fitting out in the ship, by special request of the owners, the pipe systems, auxiliaries, and engine details concerned were rearranged to enable the lower pistons to be cooled with oil in the forced-lubrication system. This involved converting the engine-driven sea water pump to a forced-lubrication pump, using the original engine-driven forced-lubrication pump as a supplementary sea water pump, providing a larger independent lubrication oil pump and an additional oil cooler. Passages through the link gear, piston rods, pistons and inlet and outlet piping were increased to the fullest extent possible to enable a sufficient quantity of cooling oil to be circulated without excessive pressure. Exhaust gas from the main engine is passed through a diverter valve to a silencer or a Cochran composite boiler 8 feet in diameter, and 19 feet 6 inches high, operating at a working pressure of 150lb. per sq. in., the exhaust gas section of the boiler having a surface of 640 sq. ft., and the oil-fired section a surface of 610 sq. ft. The oil-burning unit is of the low air pressure gravity feed type and was supplied by the Wallsend Slipway and Engineering Co., Ltd. Steam is also supplied by a three-furnace cylindrical boiler.—*The Shipping World, 27th July 1955; Vol. 133, pp. 91-93.*

New Dutch Wave Tank

Experiments with ship models in the model basin may, in general be divided into two groups: (a) those for the purpose of determining the optimum form of the hull, the propeller and the rudder for each special case; and (b) those concerned with predicting the behaviour of the ship under special conditions at sea or on inland waters, and studying the loss of speed caused by skin roughness, wind and waves, restricted depth of water. These experiments also deal with ship movements to be expected in a rough sea, such as rolling, pitching and heaving. Both types of experiment aim with continually increasing efficiency at giving the shipowner and shipbuilder advice as to the best hydrodynamic form of their ships. The first type of experiment has up to now nearly always been carried out in still water, and has attained a high degree of perfection. It may safely be said that for any ship in this ideal condition it is possible with the aid of model experiments to determine the optimum form of hull, propeller and rudder to a high degree of accuracy. However, in rough water this is not the case. Though most model tanks have equipment for generating waves which travel straight down the tank, and therefore meet the model from directly ahead or astern, this condition is an idealized one and does not correspond very well with that of a ship at sea. The wave pattern at sea is a great deal more complicated, and it will seldom happen that a ship moves in a direction at right angles to the wave front. In most cases the direction of movement of the waves will differ from that of the ship. The design of the sea-keeping basin of the N.S.M.B. differs considerably from that of existing basins in which waves are generated in one direction only. It consists of a rectangular tank 330 feet in length, 80 feet in breadth and 8 feet deep. The breadth exceeds that of an ordinary model basin of the same length and depth owing to the fact that wave generators are provided along two adjacent sides of the basin for a length of 330 feet. These wave generators consist of a large number of plates (about 160) which are placed side by side in the water and can be moved to and fro. By oscillating these plates with phase differences relative to each other, waves from all possible directions can be generated; the ship model, towed along under the towing carriage at the middle of the basin over the 330 feet length, thus moves in a sea of waves having different directions. In this way, full-scale conditions can be imitated more naturally than in an ordinary model basin. Along the other two sides of the basin, opposite the wave generators, a beach has been provided since the waves, after travelling across the basin, have to be damped as much as possible.—*W. P. A. van Lammeren, The Shipping World, 27th July 1955; Vol. 133, pp. 97-98.*

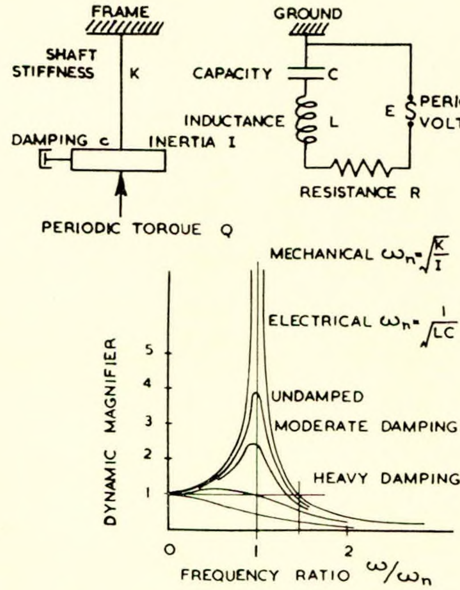


FIG. 6—*Vibratory system of one degree of freedom*

Propeller Shaft Vibration Problems

The simplest vibrating system is one incorporating the lumped elements of inertia, stiffness and damping. In the case of torsion, this combination can be represented as in Fig. 6. Such a system has a natural frequency as shown in the figure, and will oscillate when excited with amplitude depending upon the ratio ω/ω_n where ω is the angular frequency of the exciting torque applied at the inertia. The mechanical elements of inertia and stiffness are analogous in electrical theory to inductance and the inverse of capacitance, and the same dimensionless graph applies to the circuit in the figure, with respect to electrical charge instead of angular displacement. In fact, this electrical circuit may be built physically and excited with a voltage of fixed amplitude but variable frequency proportional to time. If the condenser voltage is applied to the YY plates of a cathode ray oscilloscope and the exciting voltage is triggered and geared to a linear XX time-base, the curves shown can be reproduced on the C.R.O. screen as envelopes of a modulated carrier of variable frequency. The dangers of subjecting such a system to impulses of frequency ω_n are all too apparent. In a mechanical system large amplitudes mean large stresses, and in conjunction with the inherent repetitions of amplitude and stress associated with vibrations, conditions are just those under which fatigue failure is likely to occur. This is especially so with marine propeller shafts where corrosive conditions are present. Two small auxiliary schooners suffered a series of three propeller shaft failures since Diesel engines were installed in 1950. The engines four-cylinder two-stroke exhaust-pulse-charged, were situated well aft, and the whole arrangement of engine and tailshaft was seated in a comparatively rigid framework. The elements of the system susceptible to torsional vibration were as shown in Fig. 9. Due to the comparatively large crankshaft stiffness and flywheel inertia, the node is located very close to the flywheel. The one node of vibration may be studied by a consideration of the system shown in Fig. 6. Here the energy pulses initiating vibration can arise from the engine torque fluctuation, which is not completely smoothed out by the flywheel, or from variations in propeller resistance torque due to the passage of blades through a variable slip stream. The engine, being four-cylinder, two-stroke, will produce four pulses per revolution, whilst the propeller, being three-bladed, would produce three or six pulses per revolution. Exciting frequencies of three or four times the number of revolutions per second are therefore possible. There will consequently be certain engine speeds for which the frequency of the pulse equals the natural frequency of the dynamic system. Following the loss of three shafts and one propeller, the latest shaft to break was examined. It showed obvious signs of fatigue, evidenced by 45 degrees crosses at many pits near the propeller taper as well as a major fracture through a keyway and other defects. Fig. 9 shows diagrammatically, the shafting systems of the two vessels. A calculation based on rough linear measurements of shafting and flywheel revealed

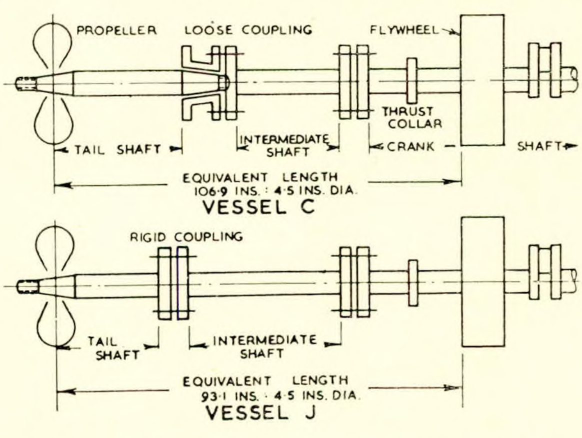
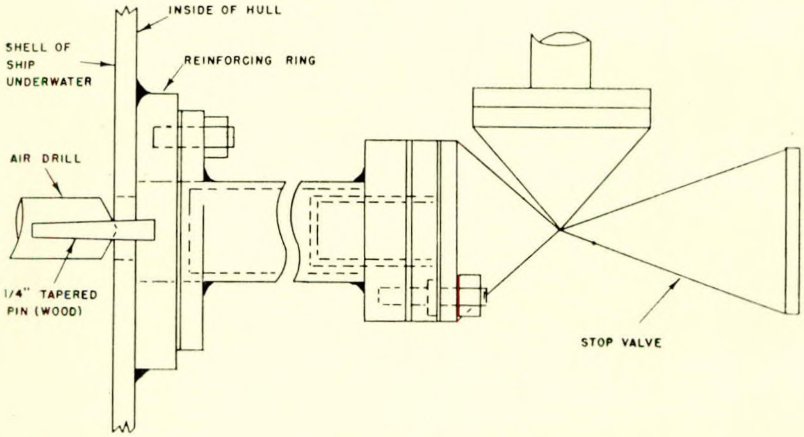


FIG. 9—*Dynamical systems of vessels "C" and "J"*

that the one node natural frequency of the system as it had been installed was 1,430 cycles per minute, and when more accurate data on the engine came to hand the figure was 1,400 cycles per minute. From what has been said above it will be apparent that shaft speeds of 358-350 r.p.m. could result in the system's resonating to engine pulses which were the obvious major source of excitation. As the designed engine speed was 340 r.p.m., it is not surprising that fatigue failures resulted from this near-resonant condition. Subsequent experimental investigations resulted in the observation, under seagoing conditions, of a resonance curve similar to that discussed.—*J. P. Duncan, Journal of the Institution of Engineers, Australia, 1955; Vol. 27, No. 3, pp. 57-63.*

Installation of Sea Chest Without Docking

A method of installing a below-the-waterline sea chest for boiler blowdown discharge without drydocking the ship has been tried out successfully at Pearl Harbour Naval Shipyard. This method is for use when the only purpose for drydocking the vessel is to install the sea connexion. The prevailing method of installing a sea chest below the waterline requires drydocking. Under the new plan, the chest can be installed without drydocking the vessel by completing the following steps: (1) *From inside the ship,* shipyard workers (a) Weld reinforcing ring on the inside of the shell instead of on its outside. (b) Drill ¼-in. pilot hole. (c) Plug pilot hole with wooden taper pin painted either red or yellow to facilitate identification by the diver. (d) Project wooden taper pin 1½ inches out from the shell, where the diver can see it. (e) Install sea chest body and sea valve. (2) *From outside the vessel,* the diver drills the proper opening in the shell of the vessel, using the coloured taper pin as a guide.—*Bureau of Ships Journal, June 1955; Vol. 4, p. 46.*



Large Shipyard for Dominican Republic

On the Rio Haina, eight miles west of Ciudad Trujillo, Dominican Republic, work is going forward on a shipyard planned to drydock the world's largest ships, and a free port. Both will have nearly three miles of wharfage frontage on both sides of the river. Known as Astilleros Dominicanos Gibbs, C. por A., (Dominican-Gibbs Shipyards) the shipyard is being built with United States and Dominican capital and operated as a private concern. The largest of five floating drydocks planned for the yard will be 1,105 feet long and 145 feet wide, designed primarily to handle the monster supertankers and ore carriers now in use, and also large enough to lift aircraft carriers of the *Forrestal* class and transatlantic liners such as the *Queen Mary,* the *United States* and the *America.—Marine News, July 1955; Vol. 42, pp. 64-65.*

Life Tests on Small High Duty Diesel Engine

In this paper the "life" tests on a high-duty Diesel engine are described, but the greatest weakness of the engine lies in the high rate of wear of the piston rings and chromium plate cylinder liner under conditions of moderate load. It appears that when the temperatures of the internal metal surfaces fall below 130 deg. C. condensation of the acids of sulphur occur which gives rise to an increased wear of ring, but which may or may not give rise to a serious increase of wear in the liner. Where hardened piston rings are used the rate of wear of the chromium bore does not increase measurably, but when soft rings are used the rate of liner wear under the cooler conditions may rise to very high values. A possible explanation of this behaviour (wear of the bore) is that corrosion of the chromium occurs both when hard and soft rings are used and that the corrosion debris breaks away. The hard debris becomes embedded in a soft ring which then acts as a lap so that the greater part of the wear of the liner is brought about by abrasion. When a hard ring is used it is assumed that the debris is not so readily embedded in the ring. Carry-over of wear has been observed mainly when the same soft ring has been used in a test following one giving a high rate of wear, and this observation fits the suggested explanation of wear. If the temperature of deposited chromium be raised to a sufficiently high value, the nature of the chromium is changed. Entrained hydrogen will be released with the attendant risk of cracking or flaking of the deposit, and the chromium itself will soften. The critical temperature at which this occurs is about 300 deg. C. From the tests at high temperature it is apparent that at 240 deg. C. the chromium surface is still satisfactory even when the engine is running at high outputs. It appears likely that the upper temperature limit of operation of a chromium bore will be set by lubrication rather than by metallurgical considerations.—*B. W. Millington, Diesel Engine Users Association, Report S240, 1955.*

Patent Specifications

Improvements in Tanker Vessels

The invention concerns tankers (vessels for carrying petroleum in bulk) and refers in particular to a construction suitable for large tankers. In Fig. 1 the numeral (1) denotes the trunk deck, (2 and 3) longitudinal bulkheads between which are situated centre tanks (8), while on the outside of the longitudinal bulkheads are situated the wing tanks (9).

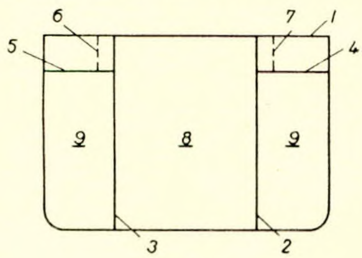


FIG. 1

The main deck (4 and 5) is situated a distance of, for example, eight feet under the trunk deck (1). The total depth is made greater than usual so that the centre tanks get a larger cubic volume than they otherwise would. The compartments (spaces) over the main deck (4 and 5), are solely used for water ballast. The wing tanks become consequently somewhat smaller than is usual.

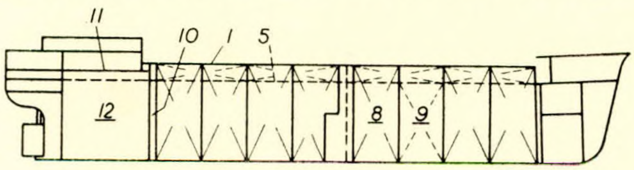
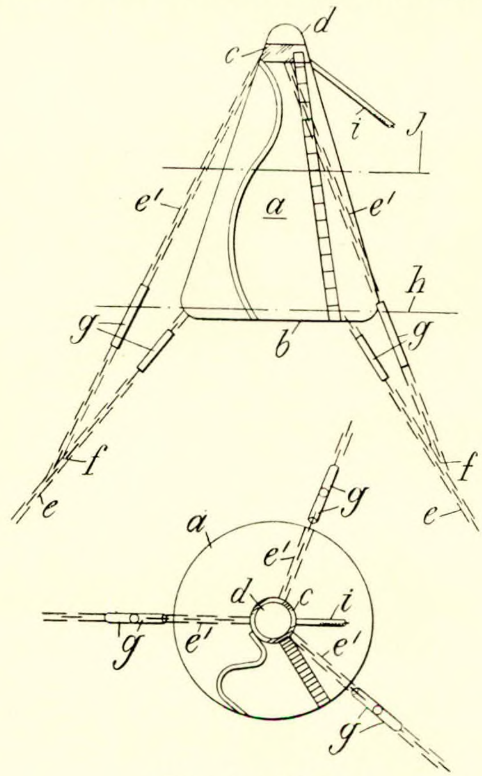


FIG. 2

Trunks (6 and 7) are fitted through the ballast tanks leading down to the wing tanks. These trunks can be fitted anywhere inside the horizontal limits of the tanks, either adjacent to the vertical sides, at the centre or any other suitable position preferably in a central position. In Fig. 2 is shown how the main deck (4 and 5) in the wing tanks run forward to the stem and aft to the after cofferdam (10). From here they can run into a raised deck (11), a so-called quarter deck, and aft of same in line with deck (4 and 5) or as a continuous deck from forward to aft with or without sheer. The engine room is shown at (12).—*(British Patent No. 735,621, issued to C. F. Christensen. Complete Specification published 24th August 1955.)*

Buoy

Buoys as at present used are subject to the drawback—often serious—that their actual position varies with rise and fall of the tide level. The object of the present invention is to provide buoys that may be used to meet a large variety of requirements including the giving of visible and/or audible notice of the presence of rocks or shallows at sea. The buoy illustrated comprises a hollow steel body (a) of truncated conical shape, the base (b) being some 40 feet in diameter and the height being some 60 feet. The body is surmounted by a lantern compartment (c) incorporating an electric lamp or lamps and a compartment (d) embodying a device for giving continuous or intermittent audible warning. The base is adapted to be connected at three equidistantly spaced points at or near the base to three or more, preferably three, chains or flexible cables (e) carrying anchors (not shown) at their lower ends, further chain or flexible cable lengths (e^1) being connected

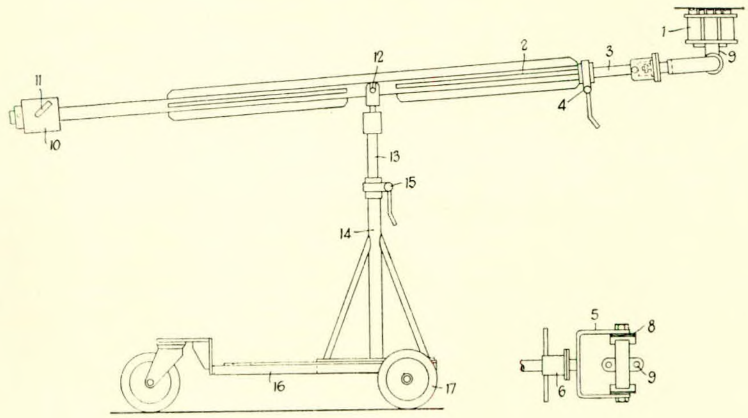


FIGS. 1 *and* 2.

between a point at, or points near, the top of the body and intermediate points (f) on each of the aforesaid chains or cables (e) respectively. Length-adjusting devices (g) are preferably provided in each of the six cable or chain lengths. Wooden fenders shaped somewhat as a letter S are provided at intervals on the surface of the body to protect it from floating objects or the like.—*(British Patent No. 736,652, issued to W. Wicks. Complete Specification published 14th September 1955.)*

Movable Descaling Apparatus

This invention relates to pneumatic descaling apparatus in which the active surface area of a descaling device as compared with the known apparatus is increased and which is specially suitable for descaling the keel or under surfaces of ships or like surfaces, where the surfaces to be descaled are downwardly directed and may be horizontal or at some arbitrary inclination to the vertical. In the form of the invention shown in Figs. 1 and 2, the descaling device (1) is mounted at the end of a bar or tube (3) which is slidable in at one end of a lever (2) and can be secured in a desired position by a clamp (4). The outer end of the bar (3) is formed with a socket (6) in which a fork (5) is mounted so that it can be rotated over a definite angle about the axis of the bar (3) and the lever (2). The descaling device (1) is secured on a member (9) which is pivotally secured

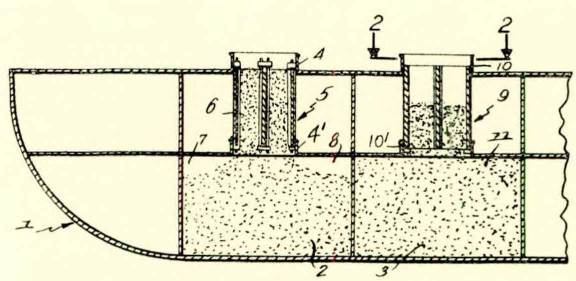


FIGS. 1 *(above)* and 2 *(lower right)*

between the ends of the fork (5) with interposed friction rings (8). It will be clear that in this form of execution the pneumatic descaling device (1) can be rotated over a distinct angle about two mutually perpendicular axes so that the active surface of the device can be adapted to each position of the surface to be descaled. At the end of the other arm of the lever there is a displaceable weight (10) which can be secured by a clamp (11). In this way by displacing the weight (10) and by changing the length of the other lever arm (2 and 3) the pressure with which the descaling device (1) is pressed against the keel surface can be adjusted. In the middle the lever is provided with two pivoting gudgeons (12) supported by a fork on the displaceable bar (13), which can slide in the standard (14) and which after adjustment to the desired height may be secured by the clamping device (15). The standard is supported by struts welded to an underframe (16) which is mounted on wheels (17) so that the scaling device can be moved under any desired part of the ship's keel.—*(British Patent No. 735,153, issued to W. Hilliers' Scheepsclassificeer Verfbedrijf N.V. Complete Specification published 17th August 1955.)*

Reusable Grainfeeder

The purpose of a grainfeeder on an ocean going ship is to feed grain to the lower hold of the ship when slack occurs or space appears at any place therein by reason of the grain in the hold settling after the ship rolls or pitches. Normally a grainfeeder is merely an oblong wooden box built to prescribed specifications. These specifications are quite severe, as grain, of course, has a static pressure like water. Especially heavy strain occurs when the ship is rolling. Apart from double planking, heavy uprights have to be used. It is a very costly matter to make such grainfeeders, as it causes delay in the ship's sailings and they must very frequently be built and installed on Saturdays and Sundays, when "double time" must be paid. Such a grainfeeder can be used only once, as, after the discharge of the cargo, usually some other commodity has to be loaded into the ship and therefore such grainfeeder is knocked apart, removed, and discarded, as it is in the way. Substantial amounts of money are in effect thrown away or lost



each year by such proceedings by an average shipping company. The grainfeeder of the invention, on the contrary, is capable of being put together in a few hours by the crew, while after discharge of the grain cargo, it can be taken apart in an hour and stacked in the side of the ship for re-using. Every part of it can be readily handled manually. Fig. 1 is a diagrammatic vertical section of a portion of a seagoing ship, showing positioned in hatches thereof two grainfeeders of the invention, each positioned above the hold of the ship, and indicating how, consequent upon rolling of the ship, grain is automatically fed from the grainfeeders into the portion of the hold directly underneath.—*(British Patent No. 736,946, issued to Albesco Inc. Complete Specification published 14th September 1955.)*

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Marine Engineering and Shipbuilding Abstracts

Volume XVIII, No. 12, December 1955

Automatic Radar Monitor

An automatic, patented, radar monitoring device, known as the Raytector, which operates in conjunction with any radar search equipment to provide visual and audible alarms whenever an object appears in a predetermined zone, has been developed. This automatic monitoring equipment, which frees the radar operator for other duties, enables fuller use of the radar equipment by reducing operational costs and increasing dependability of search. Extensive field tests, including one of the Socony tanker *Mobiloil* between Beaumont, Texas, and Providence, Rhode Island, have proven the adaptability of the Raytector to any radar set and its ability to meet a wide variety of navigational and traffic requirements. Simple front panel controls, easily operated by non-technical personnel, can be used to adjust the zone of operation, vary the range of the equipment and select a "crowded waters" or "uncrowded waters" mode of operation. This equipment also contains built-in test monitoring circuits which initiate an alarm signal in the event of failure of either the radar equipment or the Raytector. In the event of such a failure, built-in simulated radar signals, which are controlled by front panel controls, can be employed to quickly localize trouble to either the radar set or the Raytector.—*Marine News, July 1955; Vol. 42, p. 59.*

Gas Turbine Driven Generator

A 350-kW gas turbo-generator for marine standby and emergency duty, shown for the first time by W. H. Allen, Sons and Co., Ltd., at the Engineering, Marine and Welding Exhibition, is one of eight similar machines being supplied for installation in each of eight new ships for the Blue Funnel Line. They are believed to be the first emergency gas turbo-generators to be installed in vessels of the mercantile marine, and were chosen by reason of their lightness and compactness, resulting in great saving in weight and space. The gas turbine set will be housed in a separate compartment on the upper deck, and two 450-kW Allen Diesel-driven generators will be installed in the engine room in each of these eight vessels, thus saving the engine room space which would be required by a third Diesel engine. The unit comprises a single shaft open cycle type engine consisting of three main components, a single-stage centrifugal compressor, reverse flow combustion chamber, and a two-stage axial flow turbine. The turbine drives both its compressors direct at a speed of 15,000 r.p.m., and also an Allen d.c. generator; the latter through epicyclic reduction gearing, running at a speed of 1,500 r.p.m. In the 350-kW gas turbine, air is drawn into the compressor which has a compression ratio of 2·7 : 1 and is then diffused into the annular air casing which completely envelops the turbine casing. From this casing the air passes into the casing surrounding the combustion chamber flame tube. The working temperatures of the flame tube and turbine casing are thus reduced. Some air enters the flame tube at the top via the swirler, where it mixes with the fuel atomized by the burner. The products of combustion are cooled to the designed working temperature by the excess air which enters the flame tube via the dilution holes in the periphery of the flame tube. The gases are then discharged through the stator and turbine blading where the heat energy is converted into mechanical energy. When finally installed, the exhaust will be led out of the engine room via the exhaust trunking. The compressor is of the single-stage centrifugal type, of aluminium bronze (Narrmac), built in two sections, a compressor inducer consisting of a number of curved vanes integral with a small diameter hub arranged to mate up with the second section which has radial vanes, to form the main compressor rotor. The turbine is of the two-stage axial-flow type. The rotor blades are of aerofoil section, each machined integral with its root from Nimonic 90 bar., and fitted in axial fir-tree type slots in the periphery of the H 46 rotor wheels and then secured axially by a locking ring. The whole compressor and turbine rotor assembly is clamped together by means of a manganese molybdenum bolt passing through the centrally-bored wheels. The single reverse flow combustion chamber designed and made by J. Lucas (Gas Turbine Equipment), Ltd., is mounted vertically above the shaft, midway between the compressor and turbine rotors, and is fitted with a Lucas fuel oil burner. The flame tube is made of Nimonic 75. For starting purposes ignition of the fuel and air mixture is carried out by a high energy surface discharge ignition unit, incorporating a torch ignitor and operated on a 24-volts d.c. supply. The machine is designed for automatic starting in the event of a fall of voltage of the main electricity supply, and for hand starting in the compartment. The speed reduction gear is of the Allen-Stoeckicht double-helical, planetary, single-reduction type, designed to reduce the turbine speed of 15 000 r.p.m. down to the generator speed of 1,500 r.p.m.—*The Shipping World, 10th August 1955; Vol. 133, p. 145.*

Pametrada Progress

In his foreword to the recently issued progress report, for 1954, of the Parsons and Marine Engineering Turbine Research and Development Association, Sir Philip Johnson, B.A. (Chairman), referred to the wide range of research undertaken by the Association and to the substantial progress attained. Preliminary tests of sodium-potassium filled rotor blades, in a specially designed rig, indicated that a satisfactory rate of heat transfer could be attained. Preliminary tests on ceramic nozzles showed that casing blades of adequate strength and ability to withstand thermal shock could also be obtained. It had not been possible to continue research into burning heavy fuel in the 3,500-h.p. gas turbine, as the plant had been damaged by an explosion. As a result of the research on high-speed bearings, marked improvements in design had been made. In particular, a bearing had been evolved in which the rotor could continue to operate after a failure of lubricating oil, without causing damage. There had been some concern about the number of cases of shrouding stripped from first-stage blading. While, in most instances, this was attributable to the effects of water carry-over, it was felt that greater attention should be given to methods of ensuring reliable attachment between blading and shrouding. This formed the subject of a new investigation by the research department. An h.p. rotor failure was a subject of particular interest. The rotor was of the solid-drum reaction type, with a first-stage impulse wheel of Curtis type. Examination of the rotor, after nearly three years in service, revealed cracks extending, through each of the steam balancing holes in the disc, radially inwards towards, and in some cases into, the body of the adjacent dummy, and outwards, in some instances, as far as the wheel periphery itself. The rotational stresses in the disc were re-examined and found to be low, and, even taking into account the stress concentration around the balance holes, they were no higher than 8 tons per sq. in. It is important that the holes were, in fact, well-rounded at the edges. Attention was then turned to possible thermal stresses. It was found that, if priming occurred when running at full power, it would be possible for a combined maximum tensile stress of about 36 tons per sq. in. to occur at the edge of the balance holes. Conversely, if hot steam were suddenly admitted to the turbine when it was warmed through to a uniform temperature of about 180 deg. F., it would be possible to set up a maximum compressive stress of about 31 tons per sq. in., again on the edge of the holes. While no definite information was available to confirm that this cycle of stresses was in fact occurring, it was quite certain that the extremely arduous nature of the service involved frequent rapid manœuvring. The failure emphasized the undesirability of having holes of any kind through h.p. turbine first-stage discs, a position in which they are never required for normal designs. Three design inquiries were dealt with by the gas-turbine section, in addition to work in connexion with a number of research schemes. The two most important inquiries were for main machinery of 8,000 s.h.p. and 6,400 s.h.p., respectively, for merchant ships in the medium tonnage class. In connexion with the 3,500-s.h.p. marine gas turbine, the blading of the l.p. compressor was completely redesigned, and was expected to give significant improvements in output and thermal efficiency.—*The Shipbuilder and Marine Engine-Builder, August 1955; Vol. 62, pp. 510-511.*

Welded Reinforcement of Openings in Structural Steel Tension Members

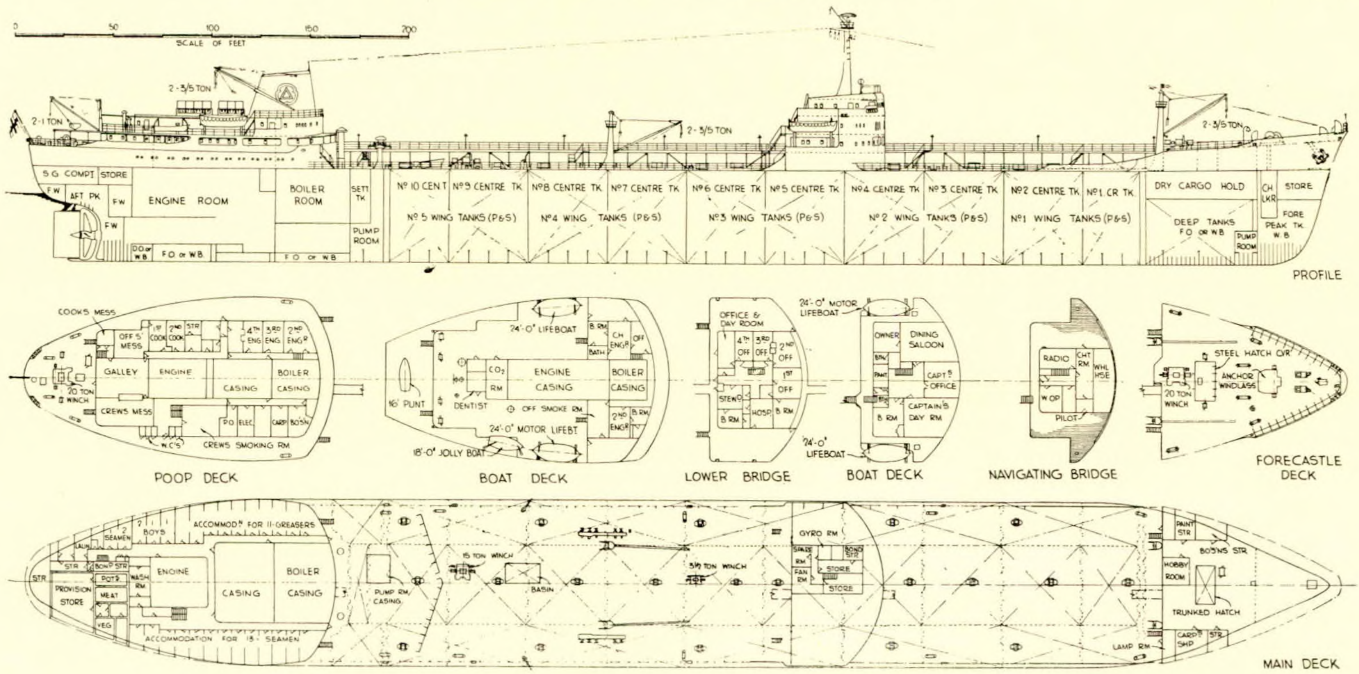
In this investigation some of the geometric factors affect the performance of plates with reinforced openings, such as the shape of the opening, the type and amount of reinforcement, and the width and thickness of the body plate, have been considered. Some of the tests were repeated at low temperatures to bring in the factor of cleavage fracture. In the course of the investigation, a considerable amount of work was directed towards determining the nature of the plastic flow that precedes the initiation of fracture, and the conditions that precipitate fracture. As a result of this work, some recommendations in regard to the design of openings and their reinforcement are made, as follows: (1) These tests indicate that the best possible practical design of an opening and its reinforcement will assure the development of the yield strength and the ultimate strength of the steel—but only about one-fourth to one-third of its potential energy-absorbing capacity—and these properties only when the conditions of loading and temperature are favourable. (2) In the reduction of stress concentration, the shape of the opening was found to be more important than the amount or type of reinforcement. The tests indicated that a corner radius equal to or greater than one-eighth of the width of the opening is desirable. (3) Although reinforcement can somewhat increase the load-carrying capacity of a member with an opening, it does not appreciably improve its energy-absorbing capacity. (4) Increasing the cross-sectional area of the reinforcement for an opening brings about increased ultimate strength and energy absorption only when the width of reinforcement in the direction of the body-plate thickness is kept small. (5) The optimum percentage of reinforcement was about 35 to 40 per cent for a face bar, 95 to 100 per cent for a single doubler plate, and between 30 and 60 per cent for an insert plate. (6) Above the optimum percentage of reinforcement, failure with reduced strength and energy absorption occurred in the weld at the outer edge of the reinforcement or in the body plate. (7) The optimum percentage of reinforcement increases as abrupt changes in the dimensions of the reinforcement in the direction of the body-plate are reduced. (8) The square of the radius of gyration of the net cross-section of the plates with openings about the transverse centreline is a satisfactory empirical parameter for determining the best distribution of the reinforcement.—*D. Vasarhelyi and R. A. Hechtman, U.S. Ship Structure Committee, Report No. SSC-75. Journal, The British Shipbuilding Research Association, June 1955; Vol. 10, Abstract No. 10,296.*

Europe's Most Powerful Tanker

Uddevallavarvet is one of Europe's most modern and outsanding shipyards. It has recently delivered to the Rederiaktiebolaget Comitia of Uddevalla their first turbine-driven ship and the largest vessel they have yet built. This ship, the *Josefina Thorden*, has the distinction of being the highest-powered European-built single-screw vessel and is also Scandinavia's largest steamship. She has been built to the highest class of Lloyd's Register of Shipping and is largely of welded construction, having two longitudinal bulkheads and longitudinal framing to the deck bottom and side plating. The longitudinal bulkheads are stiffened by horizontal corrugations and vertical webs, while the transverse bulkheads have vertical corrugations and horizontal webs. The ship is heavily constructed with a keel plate 1·22 inch thick and bottom shell plating 1·40 inch thick. The bottom centre girder is 10 feet deep and the centre line deck girder is 6ft. 11in. deep. As with vessels constructed in certain other Swedish yards the connexion, over the parallel body, between the sheer strake and the deck stringer consists of a rounded plate butt-welded to the shell and deck plating. In view of the vessel's high service speed, the forward part of the ship is heavily strengthened by means of stringers and webs. The principal particulars are as follows:—

Length overall	667ft. 7½in.
Length between perpendiculars ...	630ft.
Moulded beam	84ft.
Moulded depth	46ft. 9in.
Draught on summer free-board ...	34ft. 10½in.
Deadweight capacity, tons	32,230
Gross tonnage	20,144
Net tonnage	12,165
Dry cargo capacity (bale), cu. ft. ...	46,923
Service speed fully loaded, knots ...	18

There are ten centre and five wing tanks on each side. Eight of the ten centre tanks have the same capacity of 76,892 cu. ft. Oil fuel bunkers totalling 3,914 tons can be carried in a deep tank forward above which there is a large dry cargo hold. There are also wing bunker tanks forward of the engine room



General arrangement of the Josefina Thordén

and three double-bottom tanks beneath the engine room. There are two pump rooms, a small forehold pump room for fuel oil, situated aft of the fore-peak, and the main pump room immediately forward of the engine room. The main pump room contains three two-stage centrifugal pumps, each with a capacity of 1,100 tons of water per hour against a head of 142lb. per sq. in. pressure. The turbines are situated in the engine room and drive the pumps through special glands in the transverse bulkhead. The *Josefina Thorden* is the first ship to be delivered by a European shipyard to have single-screw machinery rated at 20,000 s.h.p. In contrast to other large contemporary steam tankers the boilers are arranged forward of the turbines. There are two Foster Wheeler D-type boilers which provide steam at 620lb. per sq. in. and 810 deg. F. These units were made by the Foster Wheeler Corporation of America, who also provided an l.p. steam generator for supplying the heating, tank cleaning and deck machinery lines. The main machinery consists of two-casing cross-compound De Laval impulse steam turbines designed to deliver 18,000 s.h.p. at 103 r.p.m. and 20,000 s.h.p. at 106 r.p.m. The h.p. turbine has thirteen single impulse stages and the l.p. turbine has eight single-row wheels. The astern turbine, consisting of a two-row Curtis wheel followed by a single impulse stage and designed for an output of 10,800 s.h.p. at 87 r.p.m., is incorporated in the forward end of the l.p. casing. The normal power running speed of both h.p. and l.p. ahead turbines is 3,044 r.p.m. and the gearing is of articulated double-reduction type, carried within fabricated gearcases.—*The Marine Engineer and Naval Architect, May 1955; Vol. 78, pp. 182–187.*

Fatal Accident

The point at which a shipboard tool changes from a valuable friend and ally to an unfriendly instrument of injury is difficult to define, or anticipate. A good case in point occurred recently aboard a dredge which was operating on the Inland Waters. Two men were using a wrench to set up on some heavy nuts when the inevitable slip, which they were not expecting, occurred and both went sprawling. One was injured so seriously that he died the next day. The cause of the accident was so simple and predictable that the only surprising aspect was the fact that both men were totally unprepared for what happened. The Chief Engineer and an oiler were engaged in making repairs to the bearings of the starboard swing winch. This winch was used to swing or veer the dredge by means of heaving in or paying out on a cable and anchor. They had adjusted and aligned bearings, using shims, and were replacing the bearing cap nuts using a ratchet socket wrench. These cap nuts were $1\frac{5}{8}$in. o.d. The socket used was for 1 11/16in. nuts, or 1/16in. too large. The oiler later claimed that the only proper sized socket aboard had been locked up by someone who had gone ashore and that they did not know where the key was located. The socket wrench handle was 18 inches long and the two men were using a 3 foot length of $1\frac{1}{2}$in. steel pipe as a "cheater", or extra lever arm. There was a good snug fit between the pipe and the wrench arm. After setting up all the cap nuts but one with considerable tension, and with no trouble, the two men placed the socket on the last cap nut and commenced to set it up. Just at the final tension was being applied, with both men leaning heavily on the pipe extension, the over-sized socket slipped off the nut and both men sprawled across the deck. The oiler was struck on the face with the pipe handle but was not seriously injured. However, the Chief Engineer struck the door of the cage-type electrical enclosure in the small of his back, on the right side. Unable to get up, the injured man could only groan and ask for the Master. As soon as the Master arrived, the Chief Engineer insisted on crawling on to a stretcher by himself. He was removed from the ship on the stretcher and taken ashore by motor launch where he was transferred to a hospital as rapidly as possible. Unfortunately he died the next morning from complications arising from a rupture of the spleen, fractured ribs, and liver injuries. Here was a needless death. While the use of the oversize socket was either poor judgment or just plain laziness, this might have been overcome if a little more care had been exercised, since all of the cap nuts had been se up without accident until the last nut. Knowing the socket was slightly loose, the two men heaving on the pipe lever should have been more cautious, even to the point of expecting the socket to slip, and being prepared for it.—*Proceedings of the Merchant Marine Council, U.S. Coast Guard, July 1955; Vol. 12, pp. 124-125.*

Dutch-built French Motor Trawler

The trawler *Geneviève le Borgne* has been completed by Gebr. Pot of Bolnes, Holland, for the joint ownership of the Soc. Fecampoise de Peche and the Soc. Havraise de Peche. She will be engaged in cod fishing in the Barents Sea and off Newfoundland Grand Banks as well as on other West Atlantic grounds. The ship has been built to the highest class of

Bureau Veritas for this type of vessel, and has main dimensions as follows:—

Length between perpendiculars ...	213ft. 11in.
Breadth moulded	36ft.
Depth	19ft. 10in.
Displacement, tons	2,660
Deadweight, tons	1,520
Speed, loaded, knots	12
Gross tonnage	1,396
Net tonnage	597

The total capacity of the holds is about 45,000 cu. ft. The space is divided into four compartments by a steel bulkhead and two detachable wooden ones. A cod liver oil boiling plant and twelve tanks totalling 70 tons are provided. The fish is separated from the steel hold structure by a detachable closed ceiling and sparring. The shell and tanktop are coated with tasteless bitumastic solution. Extensive cooling and refrigerated provision stores for the crew include nine wine tanks of 1,000 litres each and four of 250 litres capacity. The double bottoms carry 535 tons of oil fuel, 205 tons of fresh water and 80 tons of feed water. The deck machinery consists of a 30 h.p. electric windlass and a 150 h.p. Ward-Leonard electric trawl winch, having two drums, each containing about 1⅓ miles of 25 mm. wire. The main engine is a five-cylinder single-acting trunk piston two-stroke M.A.N. unit. The cylinder bore is 520 mm., the stroke is 700 mm., and the output 1,050 b.h.p. at 140 r.p.m. with a maximum of 1,650 b.h.p. at 196 r.p.m. The mean piston speed is 804ft. per minute, and mean effective pressure is 79·5lb. per sq in. at service power. A fresh water cooling installation with heat exchanger is fitted. The bronze propeller was designed and manufactured by the Lips Propeller Works of Drunen for a speed of about four knots when trawling and 12·5 knots when running free on trial trip in loaded condition. Two M.A.N. auxiliary Diesel engines, each developing 315 b.h.p. at 375 r.p.m., are coupled to a 121 kW Laurence Scott Ward-Leonard generator for the trawl winch, a 63 kW 220-volt d.c. compound wound Laurence Scott generator and an air compressor. The 63 kW machine serves as generator for the ship's main circuit as well as exciter for the Ward-Leonard scheme. The harbour service set consists of an 80 b.h.p. at 600 r.p.m. M.A.N. auxiliary Diesel engine coupled to a 40 kW Laurence Scott generator and a starting air compressor. An oil fired Cochran auxiliary boiler type, with a working pressure of 100lb. per sq. in. is fitted to provide steam used for accommodation heating and the cod liver oil plant. The nautical outfit of the ship includes a wireless installation, direction-finder, Kelvin-Hughes-Kingfisher echo-sounder, a Sperry Minor gyrocompass, a radar set, electric whistle with two different tones and automatic relay and a Sperry Loran installation.—*The Marine Engineer and Naval Architect, July 1955; Vol. 78, pp. 275-276.*

Aluminium Switchboards

In the design and construction of power and lighting switchboards for naval vessels, as for all naval apparatus, a prime consideration is economy of weight without sacrifice of effectiveness or reliability. To this end, the use of aluminium for buses and framework has long been under consideration. Aluminium busbars were probably first used in a marine application by the Cunard Line on the *Aquitania* in 1914. During the middle 1930's, aluminium was used for both the buswork and the structure of the switchboards on the aircraft carriers U.S.S. *Yorktown* and U.S.S. *Enterprise.* However, further naval use of aluminium for this application was curtailed by the scarcity of this metal during World War II. It is also of interest to note that on the postwar liner U.S.S. *United States* the structures of the switchboards and power panels are of structural aluminium. The Electrical Branch of the U.S. Bureau of Ships was engaged upon an evaluation programme that involved the design, manufacture, and testing (electrical, shock, and vibration) of an all-welded, all-aluminium switchboard unit. To accomplish this task, the Newport News Shipbuilding and Dry Dock Company, under the direction of the Bureau of Ships, designed and fabricated a generator unit utilizing a welded-aluminium framework and aluminium buswork which was suitable for use on the CVA-59 class aircraft carriers. This unit was chosen because it incorporated the Navy's largest circuit breaker (3,200-ampere, high-interrupting capacity—type ACB) and therefore required the greatest busbar current-carrying capacity. It was felt that if this unit proved to be practical, any similar type of unit would also be feasible. The generator unit was designed and engineered to incorporate the requirements and advantages of aluminium. A complete review the current-carrying capacities of aluminium buswork was also included in the overall evaluation programme. The design stage included many detailed studies and tests on such aspects as framework corner design and shape, alloys, and welding techniques. These studies indicated that an aluminium unit could be built with framing members that approximate the steel members of existing designs and also meet shock and vibration requirements. The busbar study proved that by proper selection of material the Bureau's present current-carrying capacity specification requirements could be safely exceeded by from 25 to 30 per cent. The board successfully withstood high-impact shock tests and proved to be far better from a vibration standpoint than similar steel units. Electrically, the switchboard was entirely adequate, and short-time current-carrying tests of 100,000 amperes were successfully completed. The resultant structure and buswork, exclusive of circuit breaker, transformers, and other electrical equipment, showed a weight saving of 57 per cent. However, the total weight saving with all components installed was only 24 per cent because the circuit breaker in this particular unit weighed approximately twice as much as the framework. The average savings for all types of switchboards and power panels will be approximately 35 per cent. The size and volume of the prototype aluminium switchboard are identical with the former steel-copper designs. Present figures indicate that the costs of aluminium switchgear and similar steel-copper types are now approximately equal, but it is thought that in the future the cost of the aluminium units will decline. From a conservation standpoint the all-aluminium unit is far better than the steel-copper switchboard.—*R. J. McCaffery, Bureau of Ships Journal, July 1955; Vol. 4, pp. 19-21.*

East German Shipbuilding Research Station

According to an article by W. Henschke in Die Technik, a new State Shipbuilding Research Station is under construction at Potsdam, in Eastern Germany, and the first section has recently been opened. The normal type of towing tank has the disadvantages that the tank must be 600 to 2,000 feet long, that the rails for the carriage must be very accurately laid, and that the electrical equipment required is very complicated. In order to reduce the initial cost and the construction time, a simpler solution was adopted in this case. The model is towed by an electric winch placed at one end of the tank, and a second line attached to the stern of the model passes over leading blocks and hoists a weight suspended in a shaft; this second line serves merely to keep the model on its course. The tension in the main tow-line is measured by leading the line over a block suspended from a weighing machine; this tension is the sum of the resistance of the model and the tension in the after line due to the weight. It is said that this tank can be used not only for resistance measurements, but also for self-propulsion tests and for the investigation of the behaviour of a model in a seaway, although a wave-maker does not appear to have been installed as yet. The tank is 260 feet long, 29 feet wide, and 14½ feet deep.—*Journal The British Shipbuilding Research Association, June 1955; Vol. 10, Abstract No. 10,285.*

High Temperatures Cause Failure

Recently several electronic installations became temporarily inoperative because the interconnecting cables failed. Investigation showed that failures resulted from exposure to excessive ambient temperatures, even though some of the cables were heat-resistant types. In one instance, the cables were found installed next to a steam line. Generally, cables should not run through high temperature locations such as machinery spaces, laundry spaces, galleys, stacks, and areas exposed to stack gases. In such locations, cable runs along the overhead are particularly undesirable since temperatures are normally higher there than near the deck. However, if it is necessary to run a cable through a high temperature area, it must be installed a suitable distance from any high-heat radiator such as the steam line mentioned.—*Bureau of Ships Journal, July 1955; Vol. 4, p. 33.*

Inert Metal-arc Overlay Process

An investigation was made of the welding variables which affect the penetration and dilution of overlays deposited by the inert-gas metal-arc-welding process with a consumable electrode. The influence of each of the welding variables was shown by depositing comparable single-arc bead welds where all the other variables were held constant. In order to obtain increased deposition rates and still maintain minimum dilution and penetration, multiple-arc techniques were developed that incorporated the addition of an auxiliary filler wire to each arc and oscillation of the whole welding unit. In general, single-arc operations with straight-polarity direct current or alternating current produce less dilution and penetration than reverse-polarity direct current. Minimum welding current, in conjunction with longer arc lengths, produces a less forceful arc plasma which is conducive to less penetration. Increasing the stick-out of the electrode wire and the use of argon as a shielding gas aid in producing minimum dilution. The a.c. series arc technique which employs multiple-arcs with activated electrode wires, the addition of auxiliary wires and oscillation of the whole welding unit produced overlays that had minimum dilution and penetration at very high deposition rates. Satisfactory multiple-arc overlays, with minimum dilution and penetration were made with copper base electrode wires using reverse-polarity direct current.—*C. R. Felmley, The Welding Journal, June 1955; Vol. 34, pp. 542-550.*

Air Outlets for Hollow Equipment

A proposal to inspect all portable davits, chocks, bitts, booms, and other hollow equipment that must be heated and/or drilled, to make sure that vent holes are provided, is suggested by an employee at the Charleston Naval Shipyard. Hollow equipment of this type that is not provided with vent holes before it is welded or heated, during modification or repair, builds up considerable air or gas pressure. When a hole is drilled, to install the data label plate, the air or gas will escape with enough force to endanger the driller. Some of this equipment contains a resin-like substance which spurts out of the drilled hole if the enclosed structure is heated before drilling. On one occasion, a sheetmetal helper at the Charleston Naval Shipyard was drilling a hole in a 3-in. pipe boat davit to install a test data plate. At the instant the drill pierced the inside, either the heat of the drill bit or a spark ignited the gas that had built up within the pipe during the assembly, and the driller received a first-degree burn on his hand. In another instance, an oxy-acetylene welder was brazing the flanges on a small piston. The welder had finished one end and was getting up to get a drink of water before doing the other end when the piston exploded and injured him to the point that his left leg had to be amputated. Subsequent investigation revealed that the piston was hollow and that it had not been vented before the brazing work.—*Bureau of Ships Journal, July 1955; Vol. 4, p. 45.*

The Modern Tanker

The deck layout of a typical oceangoing tanker is shown in the accompanying illustrations. The detail section shows a fairly common type of tank lid, the vapour lines leading from the coaming to a main 6 in. line along the gangway and thence up the mainmast, the deck connexions to the cargo pipelines, the housing at the top of the after pumproom, the port samson post and the mainmast, and the method by which these are stepped on the deck. The mast or samson posts of a tanker do not, of course, have to support the heavy derricks, working gear and slings of cargo which are so much a part of the working life of a dry-cargo ship. These structures do not have to be so heavily built, nor do the masts need to be continued below the main deck and stepped on some lower deck. Therefore they are stepped on the main deck itself and supported at the heels by webs which buttress them very efficiently. Tank lids shown in the drawing are typical but of course, there are many designs.

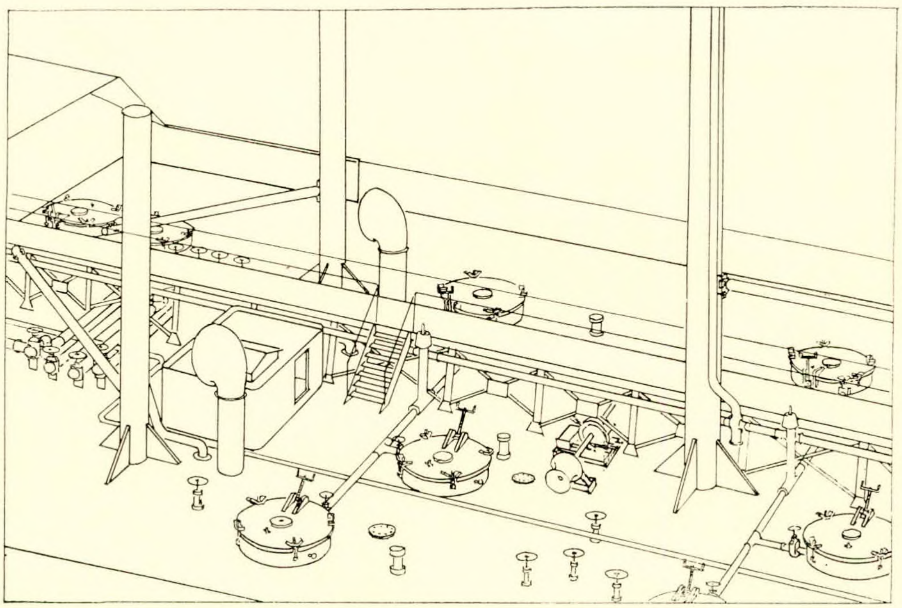


FIG. 1—*Detail of section in Fig. 2, indicated by pecked lines—showing tank lids, valves, cross-deck loading/discharge connexions, pump room top, ventilators, derricks, mainmast, after flying bridge, vapour lines, pressure/vacuum valves, bunker transfer line, ullage plugs, sighting ports, Butterworth openings and after warping winch*

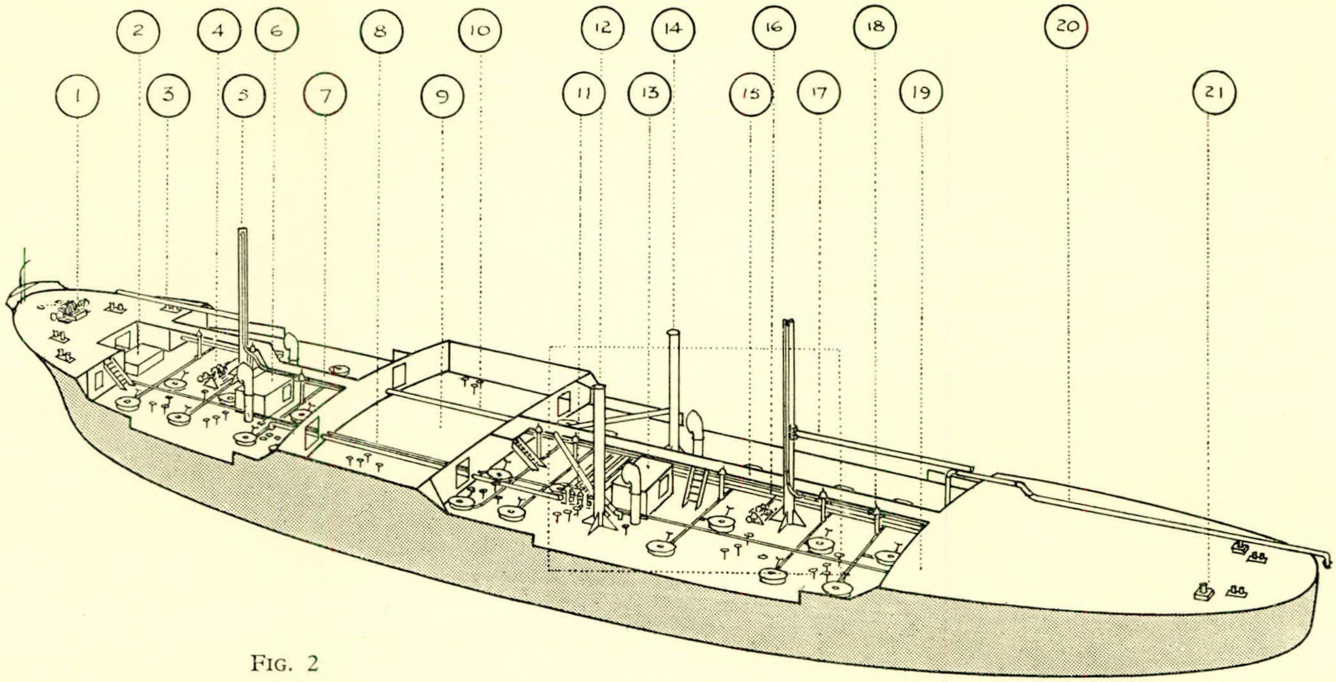


FIG. 2

1 Windlass
2 Fore hatch
3 Foremast derrick
4 Warping winch
5 Foremast
6 Forward cargo pump room top
7 Forward flying bridge
8 Port overdeck load/discharge line and bunker line running between fore deep and cross bunker tanks
9 Centre-castle space (store rooms at sides)
10 Starboard overdeck load/discharge line
11 Cross-deck connexions and gate valves
12 Port samson post with derrick
13 After cargo pump room top
14 Starboard samson post with derrick
15 After flying bridge
16 Warping winch
17 Derrick on mainmast
18 Pressure/vacuum valve where No. 9 tank branch vapour line joins main gas line running to mast
19 Poop deck
20 Stern load/discharge line
21 Mooring capstans, both sides poop

The inside face of a tank lid has a channel around its periphery, and this channel is packed with greasy hemp of about 1½ in. square section. When the wing nuts around the outside edge of the lid are screwed down the greasy packing ensures an oil-tight joint through which there should be no liquid or vapour leak. There are two small cocks on each tank lid—one on top of the lid itself and another on the side of the coaming. The first is for fitting a pressure gauge when it is necessary to know the pressure in the tank; the second is for coupling up the air salvage hoses. A sighting port is also fitted on the top of the lid and this is sometimes used when gauging, or measuring the contents. For the purpose of gauging, however, ullage pipes are usually fitted at the crown of each tank. Since the oil tanker's cargo is handled by pumps situated in special pumprooms, there is no need for the large number of winches found on a cargo ship's deck. The deck machinery is limited to windlass, warping winches on the fore and after decks where they can be used to operate the derricks or in mooring the ship, and a warping winch or two capstans aft. The deck machinery is usually steam-driven because electrically-operated machinery would carry with it the grave risk of fire and explosion. The venting of cargo tanks and the removal of all vapours are of utmost importance, and to achieve this a system of 4-in. pipes runs from the crowns of the tanks to a common 6-in. line secured along the flying bridge. The lengths of 4-in. line from the three compartments comprising one complete tank are joined at their junction with a short 6-in. length rising to the main gas line. Where this vertical length meets the common line, a pressure-vacuum valve is set into the line. This operates to ensure an equalization of pressure inside and outside the tanks. If pressure builds up in any of the three compartments, the pressure side of the gas valve opens and allows free passage of the gas, causing the pressure along the main line and thence to the vertical pipes on the masts. If, for any reason, a vacuum begins to form in the tank, the other side of the valve operates to allow the pressure inside and outside to be equalized. This is unlikely to happen with cargo in the tank, but may easily occur when steaming out during the tank cleaning process. Pressure equalization is therefore automatic once the valves have been correctly set.—*G. A. B. King, Shipping Times, June 1955; Vol. 1, pp. 82-93.*

Boiler Fuel Contamination by Sea Water

When sea water contaminated fuels are burned, saltlike deposits form on the boiler firesides. These accumulations act as insulation on the firesides and reduce the heat transfer between the combustion gases and the water or steam in the generating and superheating tubes. Consequently the operating efficiency of the boiler and the cruising range of the ship are lowered. When the deposits are heated to a sufficiently high temperature, a normal occurrence in naval boilers, they become fluid and move to lower sections of the banks. In this state they tend to fuse and solidify in relatively inaccessible locations where they form rocklike masses which are difficult to remove. In some instances it has been necessary to replace the tubes in order to remove these formations. But even if the accumulations are small, they make maintenance of the firesides difficult. Water in boiler fuel also tends to enlarge the flame angle and is one of the factors responsible for the deposits found on the diffusers and bladed cones of the burners. Accumulations of this type, which may be carbonaceous as well as saltlike, definitely interfere with combustion processes and are reflected in a reduction in combustion efficiency. The life of boiler refractory materials is markedly diminished when sea water contaminated fuels are burned. The inorganic salts present in sea water combine with the refractory material and form compounds which have melting points considerably lower than the original refractory. In addition, water and fuel oil form viscous emulsions which can cause pumping difficulties as well as make the determination and maintenance of optimum oil viscosity troublesome. There are two quick test methods that can be employed to indicate the presence of water in boiler fuels. However, these methods will not give the percentage of water present. In the first method, a frosted electric light bulb is immersed in a fuel sample and allowed to drain. (Instead of dipping, a half teaspoonful of the fuel can be poured on to the bulb surface.) If water is present, bubbles will extend beyond the thin residual oil film. In the case of fine water dispersion, the remaining oil film may have a chocolate brown appearance. In the second method, a drop of fuel is placed on a hot metal plate. If any water is present it will produce crackling sounds or spattering. Fuel that does not contain water will spread rapidly over the hot surfaces and may vaporize if the temperature of the plate is high enough. It should be remembered that each of these methods furnishes only an indication of the presence of water. However, an abundance of bubbles in the first method or severe crackling sounds and spattering in the second can be construed as evidence of a high degree of water contamination.—*A. Howarth, Bureau of Ships Journal, June 1955; Vol. 4, pp. 19-20.*

Laying Out Shell Plate with Transit

The San Francisco Naval Shipyard saves considerably in costs and time by using a transit for laying out shell plate. Before the transit was adopted by the San Francisco shipyard for laying out shell plate, the plates had to be set in place, pulled in to all the longs and stringers, scribed, and then returned to the dock for cutting and bevelling. After the plates were bevelled and cut, they were again hauled by crane back to the proper position on the hull of the vessel and pulled in place. The method proposed for laying out shell plate enables the shipfitters to make the final layout and neatly cut the plate to the exact size before positioning it on the hull of the vessel (see Fig. 1). This is accomplished by setting up a surveyor's transit dock-side and establishing a vertical line at the approximate centre of the open area of the hull to be covered by the blanket plate (two or more steel plates welded together). A horizontal line is then established in the same area in the same manner. After the horizontal and vertical lines are established, a batten (a strip of template wood ¼-inch thick, 4 inches wide, and any desired length) is clamped across the upper portion of the hull opening to be covered. The fore and aft edges of the existing shell plates are marked on the batten, the location of the vertical centreline being determined with the transit. The face of the batten is then reversed and clamped to the bottom of the opening. The fore and aft edges of the existing shell plate and the centre vertical transit line are marked on the batten. The same batten is applied to one of the two vertical edges of the hull opening, and the depth of this opening as well as the point where the horizontal centreline intersects the batten is marked. These steps are repeated to obtain measurements of the other vertical edge of the opening in the hull. The measurement markings on the batten are then applied to the blanket plate, which has a vertical and horizontal line scribed on its surface to simulate the lines on the hull opening by the transit. The batten markings are then transferred to their equivalent positions on the blanket plate. The required root opening for welding is subtracted and the edges are cut and bevelled to fit the opening in the hull. The blanket plate is now ready to be installed. Laying out shell plate with a transit is reported to have these advantages over the standard methods: (a) At least one crane lift for each blanket plate is eliminated, since the need for trial fittings is eliminated. (b) The installation and removal of a 28- by 30-ft. staging for each blanket plate is eliminated. (c) A very accurate fitting of the blanket plate is assured. (d) The final layout can be made on the dock, and the plate can be cut before it is placed in position on the ship. (e) All the necessary readings can be made with the aid of the transit in about half an hour.—*Bureau of Ships Journal, July 1955; Vol. 4, pp. 37-38.*

VERTICAL AND HORIZONTAL LINES established with transit.

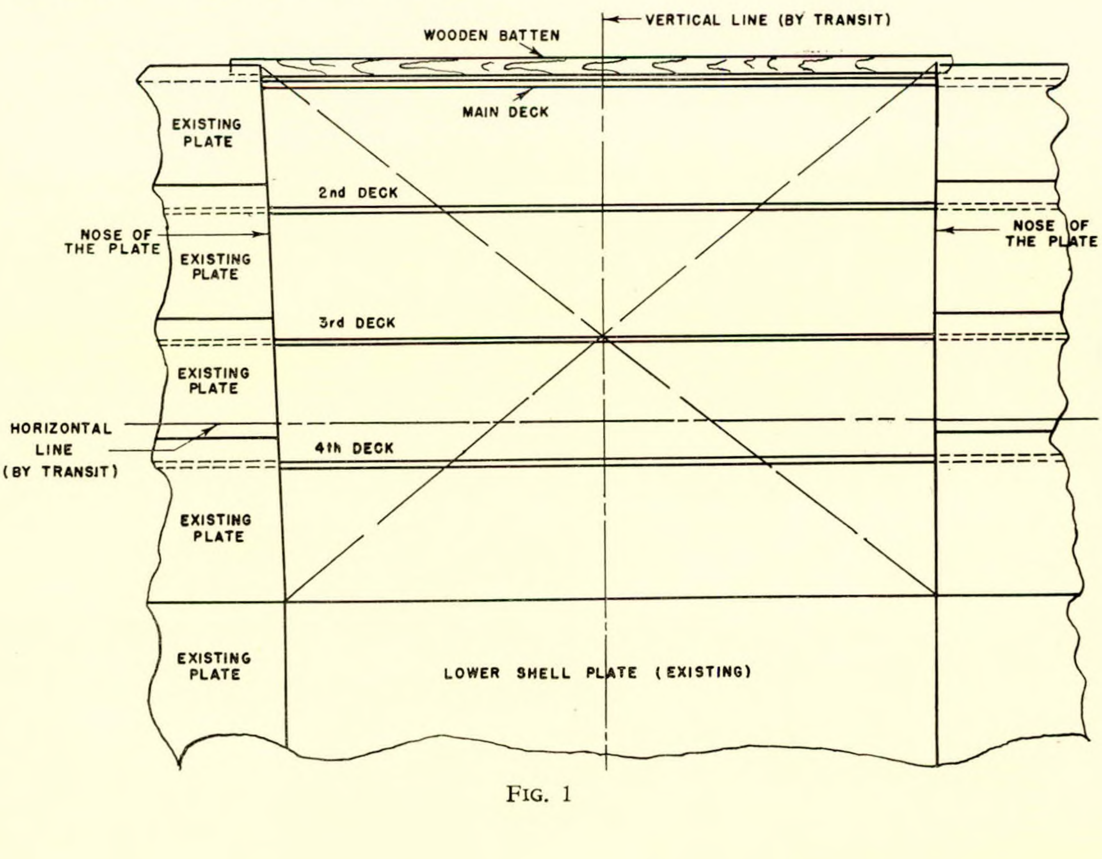
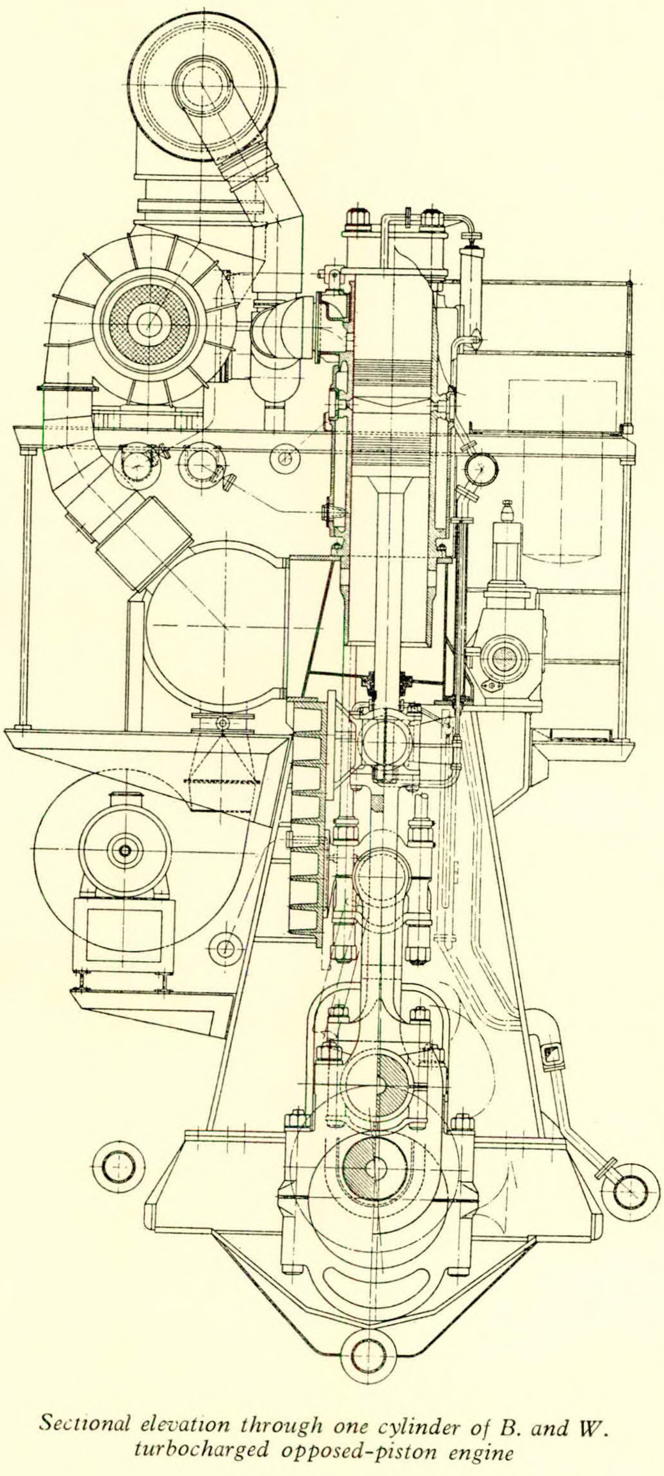


FIG. 1

Swedish-built Motor Cargo Ship with Danish Engine for Norway

Installed in the 3,000-ton 15-knot cargo ship *Balzac,* recently completed by Oskarshamns Varv, Sweden, is the first Copenhagen-built B. and W. turbocharged opposed-piston engine in which the scavenging and charging air at all loads and speeds is supplied solely by the exhaust-gas-driven turbochargers. It has five cylinders with a diameter of 500 mm. and a combined stroke of 1,500 mm., namely, 1,100 mm. for the main piston and 400 mm. for the exhaust piston. The B. and W. turbocharging system for the engine is based upon the variable pressure principle, and is in general, similar to that adopted with the engines which have exhaust poppet valves.



Sectional elevation through one cylinder of B. and W. turbocharged opposed-piston engine

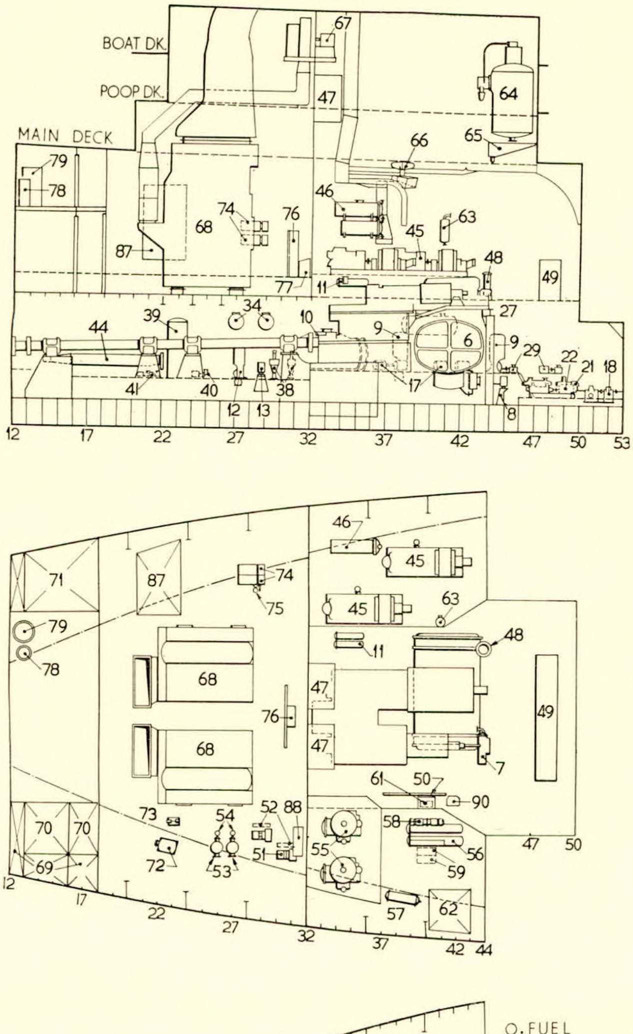
The port timing is effected solely by the pistons and there is no additional control gear. As in the case of the B. and W. unsupercharged engine, there is a phase angle between the movement of the exhaust and the main pistons, in order that the best running conditions may be ensured. There are minor modifications to the port timing, the piston setting and the fuel injection pumps, but apart from these, and the fact that there are, of course, no chain-driven blowers, the design of the engine is similar to that with normal induction. Arrangements are made for operation on boiler oil. The entablatures, the A frames and the bedplates are of fabricated steel. The standard unsupercharged engine of the dimensions given has an output of 2,900 b.h.p. with a corresponding m.i.p. of 6·5 kg. per sq. cm. (93lb. per sq. in.) but with turbocharging the continuous rating is 3,900 b.h.p. at 170 r.p.m. with an m.i.p. of 7·9 kg. per sq. cm. (112lb. per sq. in.). This is approximately equivalent to a b.m.e.p. of 99 to 100lb. per sq. in. The increase in power is 35 per cent. In the case of the *Balzac,* however, for continuous service a more conservative rating is adopted, namely 3,100 b.h.p. at 145 r.p.m. with an m.i.p. of 7·5 kg. per sq. cm. or 105lb. per sq. in. Located at the back of the engine are two Brown-Boveri turbochargers of the VTR 500 type. The exhaust gases from cylinders Nos. 1, 2 and 3 are supplied to one turbine and those from the other cylinders to the second turbocharger. The energy available in the exhaust gas at any load according to the propeller law is sufficient to ensure provision of adequate charging air volume and pressure at low speeds. The scavenging and charging air is drawn from the engine room into the turboblowers through combined filters and silencers. After compression it is discharged to the scavenging air manifold through coolers in which the air temperature is reduced to about 10 degrees above that of the cooling water. Salt water is circulated round the coolers. For use if both turbochargers should be out of commission at the same time, there is a small emergency blower driven by an electric motor. With this in service the engine can run on the three forward cylinders at about 65 per cent of the service speed. A special system of gravitation lubrication is provided for lubricating the ball and roller bearings of the turbocharger, instead of gearwheel pumps driven direct by the rotor shaft. The oil is pumped from a sump tank below the turbochargers by means of an electrically-driven pump. It passes through a filter, and up into a tank which is arranged at a distance of five or six metres above the centre line of the turbochargers. From this tank the oil gravitates down through the bearings, back into the sump tank, sight glasses being provided to observe the oil flow.—*The Motor Ship, August, 1955; Vol. 36, pp. 198-199.*

Large French Tanker

The first of a group of three tankers of 33,000 tons d.w. ordered from Société des Ateliers et Chantiers de France, Dunkirk, has been delivered. This tanker, the *Chambord,* is similar in many respects to the 32,000 tons d.w. class designed and built for the British Tanker Co., Ltd., in the United Kingdom. The French building programme includes two other tankers, the *Chenonceaux* and *Cheverny,* both of which are building at Dunkirk. The principal particulars of the *Chambord* are as follows:—

Length o.a.	669ft. 4in.
Length w.l.	644ft. 8in.
Length b.p.	630ft. 9in.
Breadth moulded, feet ...	86
Depth	46ft. 8in.
Draught loaded	35ft. 6in.
Speed, knots	17
Deadweight, tons	33.200
Displacement, tons	42,440
Gross tonnage (about) ...	21,800
Tank capacity (about), cu. ft.	1,493,613

The *Chambord* has been built, like most tankers, on the longitudinal system. The bow is raked above the waterline but



General arrangement of machinery in the turbine tanker Chambord

1 L.P. turbine
2 H.P. ahead turbine
3 H.P. astern turbine
4 Reduction gearing
5 Turbine gear
6 Main condenser
7 Main condenser ejector
8 Main extraction pumps
9 Main circulating pumps
10 Auxiliary condenser
11 Auxiliary condenser ejector
12 Auxiliary circulation pump
13 Auxiliary extraction pump
14 Atmospheric condenser
15 Atmospheric condenser and L.O. cooling circulating pump
16 Drain pump
17 Butterworth and fire pump
18 Cargo pump turbines
19 Distilling condenser
20 Electric stripping pump
21 Turbo feed pumps
22 Electric feed pump
23 General service pump
24 Turbo bilge pump
26 Brine tank
27 Condenser
28 Electric F.O. transfer pump
29 F.O. pressure pump
30 F.O. suction strainer
31 Condenser strainer
32 Pre-mix tank
33 Electric condenser pumps
34 Oil coolers
35 Lubricating oil suction strainer
36 Lubricating oil strainer
37 Electric L.O. pump
38 L.O. separators
39 Dirty water, hydrophor
40 Sanitary pump
41 Refrigerator pumps
42 Domestic pumps
43 Drinking water pump
44 Spare shaft
45 Turbo alternators
46 Butterworth heater
47 L.O. gravity tanks
48 Emergency start ejector
49 Switchboard
50 Manoeuvring platform
51 Air compressors
52 Compressed air cooler
53 Air bottle
54 Compressor drain
55 Boiler evaporators
57 Distilled water cooler
58 Raw water heater
59 Distilled water pump
61 Distilled water tank
62 Feed tank
63 Desuperheat
64 Deaerator
65 Desuperheater
66 10-tons crane
67 Ventilator fans
68 Boilers
69 Cofferdam
70 Reserve oil tank
71 Diesel oil tank
72 Domestic water heater
73 Drinking water pump
74 Fuel oil heaters
75 Fuel oil strainer
76 Bailey control panel
78 Drinking water, hydrophor
79 Domestic water, hydrophor
80 Air conditioning pumps
87 Reserve tank
88 Observation tank
89 Separator heaters
90 Telephone cabinet

below it is vertical and terminates in a bulbous forefoot. The hull is entirely welded with the exception of the gunwale bar, two longitudinal seams on the deck and six longitudinal seams on the hull for three-quarters of its length. Prefabrication has played an important part in the construction of this vessel, some of the sections weighing as much as 60 tons. Oil cargo is carried in thirty tanks having a total capacity of about 1,493,613 cu. ft. The main pump room is arranged in the cofferdam immediately ahead of the engine room. There are three main cargo pumps installed, each having a capacity of about 1,300 tons per hr. These pumps are mounted horizontally and each is driven by a single-stage steam turbine installed in the engine room and driving through a gas-tight gland in the bulkhead. The pumps are of the centrifugal type naving large diameter outlets which are arranged at the upper part of the volute in order to allow them to prime ready for starting and to prevent the formation of gas pockets during operation. The pumps are driven through a simple gear train. In addition to the main cargo pumps the vessel is equipped with two steam-driven pumps of about 250 tons per hr. capacity. The *Chambord* is propelled by a group of turbines driving a single propeller through reduction gearing. The turbines are of the Parsons type built at the Atelier de Penhoet and developing a normal output of 13,935 h.p., at about 105 r.p.m., with a maximum continuous power of 15,200 h.p. at 108 r.p.m. On her trials the *Chambord* attained a speed of 17·467 knots fully loaded. The engine consists of one h.p. turbine and one l.p. turbine in series. There are two astern turbines, one incorporated in the l.p. ahead casing and the other a separate h.p. astern turbine, designed to develop together at 50 per cent of the normal ahead revolutions (about 52·5 r.p.m.) power of roughly 80 per cent of the normal ahead power at 105 r.p.m. with a moderate steam consumption. They are also capable of operating for one half-hour at 70 per cent of the normal ahead revolutions. The l.p. turbine exhausts into the condenser located directly below it. Steam is supplied by two Penhoet P.41 type watertube boilers installed on the flat aft and designed for an effective maximum pressure of 665 lb. per sq. in. The normal pressure at the superheater outlet is 600 lb. per sq. in. at 850 deg. F. Each boiler has three oil fuel burners of the Penhoet mechanical pulverization type designed to burn heavy Bunker C fuel. Bailey automatic control is provided. With 3,000 tons of fuel on board the *Chambord* will be able to make the round trip from Dunkirk to the Persian Gulf and back, the average daily consumption being 76 to 80 tons.—*The Shipping World, 22nd June 1955; Vol. 132, pp. 661-665.*

Flexibility of Shafts

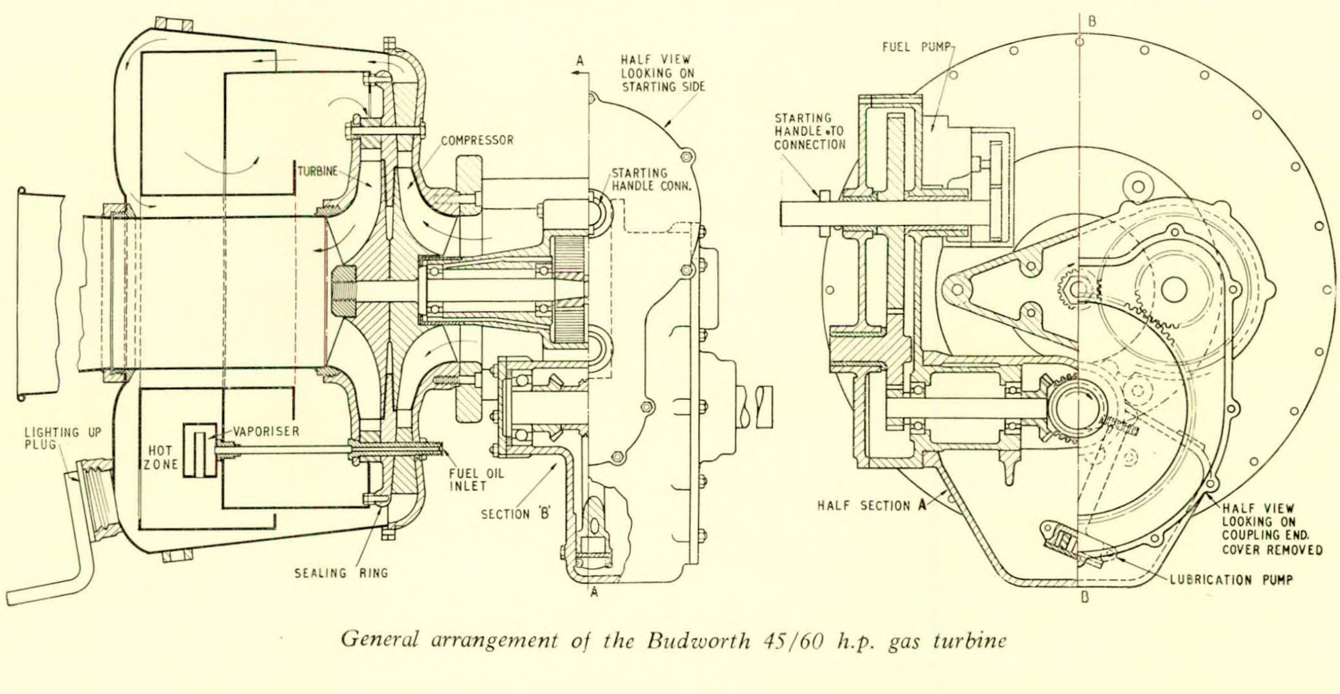
When assembling a composite propeller shaft, two different methods of alignment can be used. Method 1, generally used in French merchant shipbuilding, is based on adjusting the levels of the shaft bearings so that the mating flanges of the shafts are co-axial and parallel when brought together for bolting. On the other hand, the principle of method 2, which predominates in the French Navy, is to have all the shaft bearings in exact alignment, irrespective of whether this makes the shaft flanges before bolting parallel and co-axial or not. The author makes a general theoretical analysis of the bending moments and deflexions introduced into a shaft system by these two methods of assembly, and illustrates this with examples. These include calculations of slopes and deflexions in shaft systems of a number of cargo vessels, and photographs of shaft models so constructed that the deflexions are exaggerated and can easily be seen. The effects of making the shafts hollow, and of a general slight curvature of the shafting caused, for example, by a mishap at sea, are also considered. In general, the author advocates method 2 because it gives the greatest reduction in the maximum values of the bending moment in the shaft, and the best division of the total load among the bearings. The author suggests that method 1 should be confined to cases where there are two bearings for each shaft section, and he gives a formula for the recommended distance between the bearings in such cases. The one exception for which method 1 is advocated in preference to method 2 is when aligning the flanges of the shaft and engine, particularly a reciprocating engine; it is better not to introduce any external bending moment into the crankshaft so as to avoid crank-web deflexions.—*Paper by J. Legris read at the 1955 Meeting of Association Technique Maritime et Aéronautique, Paris. Journal, The British Shipbuilding Research Association, August 1955; Vol. 10, Abstract No. 10,577.*

New Lightweight Gas Turbine

The Budworth 45/60 h.p. gas turbine has been designed for portable and auxiliary power requirements by Mr. David Budworth, Harwich, Essex. It is of the simplest possible design and weighs only 60 lb. when coupled to the 15:1 reduction gear shown in the drawing. As an alternative, a 20:1 bevel reduction gear can be supplied which, for some applications, is a more convenient layout. The turbine has a speed of 45,000 r.p.m., which gives an output shaft speed of 3,000 r.p.m. with the one reduction gear and 2,250 r.p.m. with the bevel gearing. The power/weight ratio is 1 lb. per b.h.p. with a 2,250 r.p.m. shaft output speed, and better for higher final shaft speeds. The turbine unit consists of an annular combustion chamber surrounding a radial compressor and turbine mounted back to back on the same shaft. On leaving the diffuser, the air is split into three streams indicated by arrows on the drawing. The outermost stream of air keeps the casing of the combustion chamber cool as well as cooling the exhaust outlet and the face of the turbine nozzle ring. The middle stream provides air for primary combustion, while the inner one cools the face of the turbine rotor. Most of the engine and reduction gear is made of aluminium. Suitable heat-resisting materials are used for the turbine and exhaust pipe, and the outside of the combustion chamber is made of mild steel, dull chromium plated in order to preserve it against corrosion. A feature of the turbine is the simplicity of the starting procedure. This can easily be carried out by hand by one man. Ignition is started by inserting a lighted piece of rag previously soaked in fuel oil through the aperture provided. This contains a quick-release plug as shown on the drawing. After a very short space of time, immediately if paraffin is used as a fuel, the turbine will start. There is no electrical gear to get damaged by water, and no batteries to add to the weight of the unit. A low-pressure fuel system is used, the fuel pressure being only a few pounds per square inch higher than that of the combustion chamber. The pressure inside the combustion chamber has only to rise a very little with speed for the fuel supply to be cut off completely; and conversely with a slight drop in speed the fuel supply is increased. This method of governing has been found to be extremely simple, and is sufficiently accurate for most power generation applications. There are, however, a few exceptions which can be overcome by fitting a centrifugal governor. The turbine can be run on any distillate fuel and the estimated fuel consumption is in the region of 1·5 to 1·6 lb. per b.h.p. per hr. Lubrication of the gearing is by means of jets supplied with oil from a gear pump in the sump.—*The Shipping World, 14th September 1955; Vol. 133, p. 267.*

A Study of Cracks in Stern of Cargo Vessel

A study of cracks appearing in service in the stern of a cargo vessel is reported upon in a paper by H. de Leiris, read at the 1955 Annual Meeting of the Association Technique Maritime et Aéronautique. In an open shelterdecker of some 840 tons d.w. and 209-ft. length o.a., propelled by a supercharged four-stroke eight-cylinder Diesel engine of 1,160 h.p.



General arrangement of the Budworth 45/60 h.p. gas turbine situated aft, very extensive cracks appeared in the stern portion of the hull during service. The severity of the cracks was so unusual that an extensive laboratory examination of specimens containing a total of over 100 cracks was made; the results of this are described in this paper. The cause of the cracks seems to be excessive vibration of the stern which had been observed after the trials of the vessel but had not been thought severe enough to warrant the expense of stiffening the structure. The cracks were first noticed after the vessel had encountered heavy seas on a passage from Casablanca to Bordeaux. They were found in several strakes aft, in the aftermost watertight bulkhead, and in frames inside the after peak. It was also discovered that the longitudinal angle stringer which stiffens each side at about 4 feet below the strength deck had been left out, owing to an error in assembly, between frame 1 and frame 5 both on the port and on the starboard side. This omission had lessened the stiffness of the stern. The fractures in the plating could be classified into five types, and those in the framing also into five types. The various types are discussed and illustrated. It could be established that all developed progressively and were initiated by cyclic stresses. Since fatigue cracking in ships' hulls is rare, comparison with other similar failures is limited. Among those examined, certain types of cracks have not been observed in ship structures before, but three-quarters of the total number of cracks can be recognized as of a type found round cut-outs in members subjected to cyclic loading, i.e. cracks arising from the geometry of the part and not from unsatisfactory welding or oxy-flame cutting. Some cracks, however, could be attributed to these last two causes. The quality of the steel, although not beyond criticism, did not seem to have played any decisive role in the failures.—*Journal, The British Shipbuilding Research Association, July 1955; Vol. 10, Abstract No. 10,474*

Canadian Pallet-carrying Ship

An interesting vessel completed in May by Collingwood Shipyards, Ltd., of Collingwood, Ontario, is the cargo liner *Fort Henry*. This ship is a new addition to the Great Lakes cargo fleet of Canada Steamship Lines, Ltd., and unlike other recent additions to this fleet, which have been bulk carriers of normal Great Lakes type for the carriage of grain and iron ore, she is intended for the palletized cargo service which Canada Steamship Lines run on the Great Lakes. The vessels running this service are known in Canada as "package freighters," and the *Fort Henry* is the first new vessel of the type to be built for more than twenty-five years. The fork-lift trucks enter and leave the ship through five 8 feet wide by 10 feet high watertight gangway doors which have been arranged on the tweendeck, to port and starboard. These gangway doors have been spaced to suit the dock facilities, and have electric hoists for opening. Four 18-feet long by 9-feet wide hydraulically-operated elevators transport the trucks to and from the lower hold and tweendeck. The tank top and main deck are flush and level, without sheer or camber. They are covered with 1½-inch asphalt. The ship has also been arranged to carry grain in bulk and has, therefore, been fitted with six large flush hatches on the spar deck and main deck. In addition to these six hatches, the main deck has a series of flush trimming hatches fitted to port and starboard to permit the grain in the lower hold to be properly trimmed. The hatches in the spar or upper deck were made flush so that automobiles could be easily and efficiently carried on this broad expanse of deck. The principal particulars of the vessel are as follows:

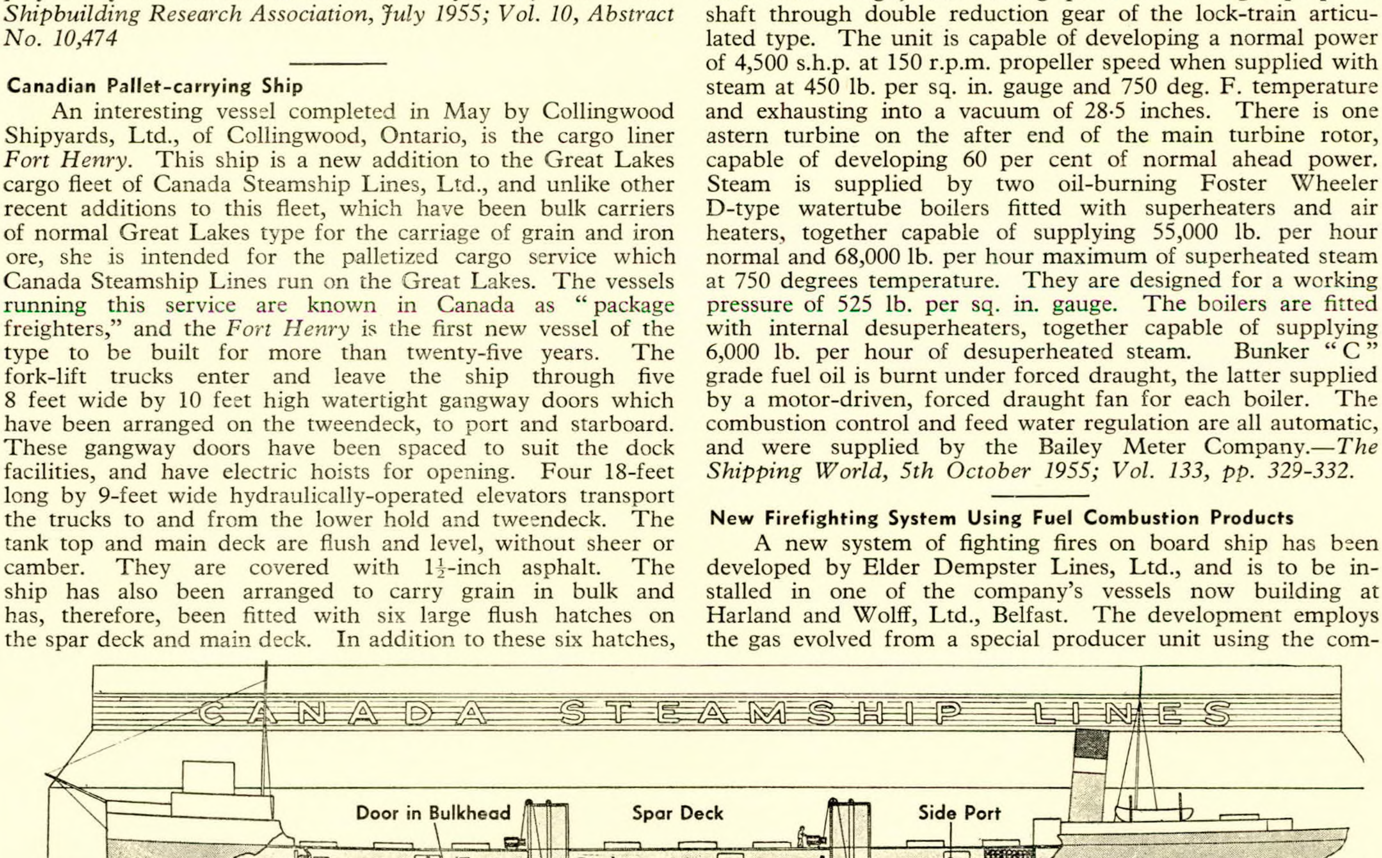
Length o.a.	461ft. 6in.
Length b.p., feet	440
Breadth moulded, feet	56
Depth of tank top, feet	5
Depth of main deck	18ft. 6in.
Depth to upper deck, feet ...	32
Summer draught, full	23ft. 9in.
Package freight draught, feet, aft	18
forward	11ft. 6in.
Full deadweight at summer draught, tons	8,420
Full deadweight at summer freight draught, tons ...	3,550
Shaft h.p.	4,500
Trial speed, knots	17·5

Hold capacities:	*Length*	*Grain cu. ft.*	*Bale cu. ft.*
Tweendeck ...	292ft. 6in.	214,718	198,122
No. 1 hold ...	90ft.	48,874	42,300
No. 2 hold ...	90ft.	66,166	58,810
No. 3 hold ...	112ft. 6in.	81,958	73,074
Total ...		411,716	372,306

The main propulsion machinery consists of a set of Inglis-Parsons steam turbine machinery of the single rotor type to Pametrada design, transmitting power to a single propeller shaft through double reduction gear of the lock-train articulated type. The unit is capable of developing a normal power of 4,500 s.h.p. at 150 r.p.m. propeller speed when supplied with steam at 450 lb. per sq. in. gauge and 750 deg. F. temperature and exhausting into a vacuum of 28·5 inches. There is one astern turbine on the after end of the main turbine rotor, capable of developing 60 per cent of normal ahead power. Steam is supplied by two oil-burning Foster Wheeler D-type watertube boilers fitted with superheaters and air heaters, together capable of supplying 55,000 lb. per hour normal and 68,000 lb. per hour maximum of superheated steam at 750 degrees temperature. They are designed for a working pressure of 525 lb. per sq. in. gauge. The boilers are fitted with internal desuperheaters, together capable of supplying 6,000 lb. per hour of desuperheated steam. Bunker "C" grade fuel oil is burnt under forced draught, the latter supplied by a motor-driven, forced draught fan for each boiler. The combustion control and feed water regulation are all automatic, and were supplied by the Bailey Meter Company.—*The Shipping World, 5th October 1955; Vol. 133, pp. 329-332.*

New Firefighting System Using Fuel Combustion Products

A new system of fighting fires on board ship has been developed by Elder Dempster Lines, Ltd., and is to be installed in one of the company's vessels now building at Harland and Wolff, Ltd., Belfast. The development employs the gas evolved from a special producer unit using the com-



Arrangement of a typical package cargo vessel of Canada Steamship Lines. The internal layout of the Fort Henry *is generally similar to this* bustion products of Diesel oil. The main advantage of this new system is that it has a self-contained generating unit and need not rely on storage limited in capacity by space and weight; and the supply of extinguishing gas is virtually inexhaustible. The prototype has been approved by the Ministry of Transport subject to final acceptance trials. As a means of providing the inert gas, the installation employs a water-jacketed refractory lined combustion chamber incorporating a blower. This combustion unit is supplied by W. C. Holmes and Co., Ltd. The fire engineering layout and technique will be under the direction of the Pyrene Co., Ltd. In the system now being installed at Belfast, the gas producing unit is a modification of a machine now largely used for rendering inert the atmosphere in chemical process plant and gas undertakings for the purpose of preventing fire and explosion. The use of combustion products for this purpose is well established and once the plant is acquired and installed, inert gas can be produced relatively cheaply. The composition of the gas, as determined by standard tests, is as follows:

Oxygen, per cent	0—1
Carbon monoxide	Nil
Carbon dioxide, per cent ...	15—14
Unburnt hydrocarbons ...	Less than 0·1 per cent
Oxides of nitrogen, per cent	0·01—0·015
Nitrogen	Remainder 85 per cent

In the plant now being erected, the gas is passed direct from a contact cooler, from which it emerges at a temperature slightly above that of the cooling water. There is no carry over of water droplets, and therefore the cargo will not be damaged by moisture. The generator itself is about 9 feet long, by 4 feet wide, and can be arranged to fit between decks. It consists of a horizontal combustion chamber, water jacketed, and refractory lined. At one end is the burner and air mixing arrangements, and the other opens directly to the foot of the cooling chamber, the cooled gas leaving near the top of this chamber through a side connexion. On the top of the combustion chamber is mounted a Diesel engine driving the air blower, the water service pump and the fuel pump. Pressure relief valves are fitted on the air and gas side and air control valves on the burner are marked with start and running positions. The fuel is Diesel oil and is kept at constant pressure by means of a governor valve, the quantity used being controlled. The engine uses the same fuel, and the total consumption to meet Ministry requirements for a 10,000-tons cargo ship would be about 25 gallons per hour. The pump is arranged to supply the cooling water, and alternatively, can supply the two independent jets required by the Ministry of Transport to meet the regulations as an alternative to the engineroom pumps. The inert gas can further be led into the engine room in support of these water jets. A control panel is fitted giving visual and audible indications if the system becomes out of balance, and if the carbon dioxide content of the inert gas should drop from the normal 14 per cent to 15 per cent. This control panel is also fitted with a continuously reading carbon dioxide indicator, which will, under operating conditions, enable the carbon dioxide to be kept to a maximum. It is not proposed to use salt water cooling except in case of fire. A fresh water supply will be arranged for use when testing, and the generator itself can be tested on atmospheric discharged without opening the inert gas to the holds, while the system as a whole can be tested conveniently when the holds are free of cargo.—*The Shipping World, 17th August 1955; Vol. 133, p.161.*

Gas Turbine with Geared Drive

Some technical details of the new geared-drive gas turbine unit to be installed in the Shell tanker *Auris* have now been released. The new set will consist of an open cycle gas turbine with intercooler and heat exchanger. Air drawn in from the atmosphere passes through the l.p. compressor, the intercooler, the h.p. compressor and the heat exchanger, to the combustion chamber wherein it is heated by the burning fuel. The combustion products expand first through the h.p. turbine which drives the h.p. compressor only. They then pass through the l.p. turbine which, in addition to driving the l.p. compressor, produces the propelling power. From the l.p. turbine the products of combustion flow through the heat exchanger, thence to an exhaust gas boiler in which steam at 50 lb. per sq. in. is generated, and to the atmosphere. The principal particulars of the installation are as follows:—

Design ambient temperature, deg. F. ...	75
Turbine inlet temperature, deg. F. ...	1,200
Design output, b.h.p.	5,500
Heat rate, B.Th.U. per b.h.p. per hr. ...	9,100
Waste heat boiler	
Steam pressure, lb. per sq. in	50
Evaporation, lb. per. hr.	5,000
H.P. turbine shaft	
Speed, r.p.m.	5,800
No. of compressor stages	16
No. of turbine stages	5
L.P. turbine shaft	
Speed, r.p.m.	3,840
No. of compressor stages	12
No. of turbine stages	8

The output and heat rate ignore the power obtained from the waste heat boiler. Power to the propeller shaft is transmitted through Cammell Laird articulated double-reduction gearing in conjunction with an hydraulic manœuvring system of Pametrada design. In this gearbox a small tooth-type flexible coupling connects the l.p. turbine with a quill shaft on which are keyed the driving wheels of the ahead hydraulic coupling and the astern torque converter, arranged at the after and forward ends of the first reduction pinion. The ahead coupling consists of a driving and a driven wheel; in addition, the astern converter has a fixed blade cascade between the two wheels. The driven wheels are connected to the ends of the first reduction pinion. Manœuvring is effected by filling one or the other of the two couplings with oil. This is controlled by means of a single piston type oil valve which supplies oil to the couplings as required. This valve is interlocked with the fuel supply to the combustion chamber in order to prevent mishandling. The ahead coupling has an efficiency of about 98 per cent. A direct-drive friction clutch is provided in the same casing in order to eliminate the two per cent loss when the transmission system is operating at full power in the ahead direction.—*The Shipping World, 7th September 1955; Vol. 133, p. 235.*

Fishery Research Trawler

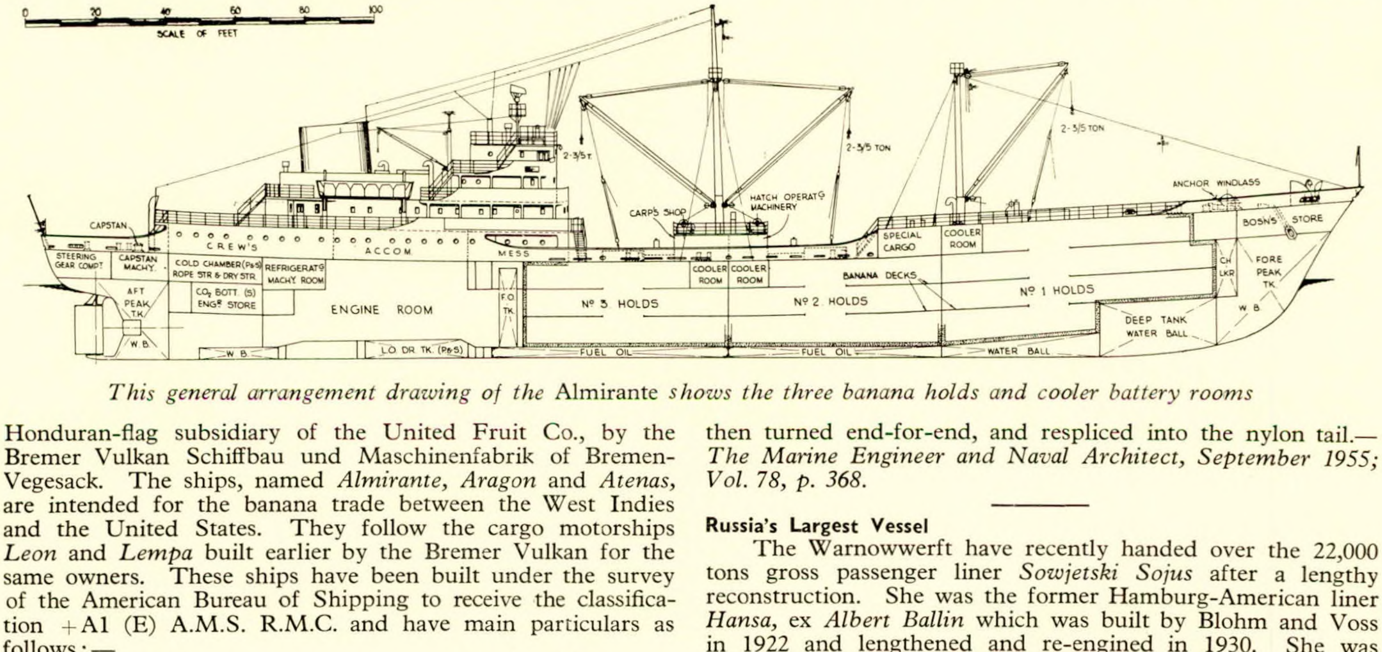
The research trawler *Sir William Hardy*, which is now in service, has been built for the Department of Scientific and Industrial Research by Hall, Russell and Co., Ltd., Aberdeen. She is specially equipped for investigations into the preservation of fish as food, and is also intended to provide raw material for work at the Torry Research Station at Aberdeen. The design was drawn up by Torry Research Station staff acting with the advice of the N.P.L. Ship Division, the British Shipbuilding Research Association, the government departments concerned and representatives of the fishing industry. In addition to the normal facilities for trawling, the equipment provided includes laboratories, experimental freezing plant, an insulated hold for iced fish and a liver oil plant. The *Sir William Hardy* is notable as the first British trawler to have Diesel-electric propulsion. It is also notable in that arrangements have been made for the major overhaul of the four main propulsion generator units by their removal in turn from the ship, a spare unit being provided for replacement purposes. The trawl winch is electric, and this and the propulsion motor are connected in the same constant-current circuit with the four main generators, so that power can be transferred from the propulsion motor to the trawl winch as required. The four main Diesel engines are Meadows six-cylinder type engines, each with a twenty-four-hour rating of 200 b.h.p. at 1,400 r.p.m. They are direct coupled to 125-kW constant current generators through flexible couplings. Each Diesel-generator set is mounted on a combined bedplate secured to the seating by Silentbloc flexible mountings and is fitted with its own fresh and salt water circulating pumps, heat exchangers, and oil coolers. The freshwater circulating systems are thermostatically controlled, and immersion heaters are fitted in the sumps to enable engines not in service to be kept warm. The C.A.V. fuel pumps incorporate a special governor, which can be set to operate at the normal speed of 1,400 r.p.m., or for special duty at 1,000 r.p.m. The generators are provided with top-mounted combined isolating-and-transfer switch cubicles, so that each unit can be used independently. The propulsion motor is a 600 s.h.p. 200 r.p.m. double-unit machine. It is equipped with a combined thrust-and-journal bearing of the Michell type at the after end; the Michell bearing is watercooled by means of a shaft-driven pump. The motor is ventilated by duplicate centrifugal fans, taking their air supply from above deck and exhausting into the engineroom; either fan can deliver sufficient air for continuous operation of the motor at full speed. Excitation is provided by exciter sets each of which consists of a 17 h.p. 1,500 r.p.m. motor driving two metadyne exciters; one exciter controls the speed and output of the propulsion motor, and the other the constant current system of propulsion. Duplicate exciter sets are provided, one acting as a standby.—*The Shipping World, 20th July 1955, Vol. 133, p. 67-69.*

German-built Banana Carriers

Three fruit-carrying motorships of unusual design have been delivered to the Empresa Hondurena de Vapores, the Honduran-flag subsidiary of the United Fruit Co., by the Bremer Vulkan Schiffbau und Maschinenfabrik of Bremen-Vegesack. The ships, named *Almirante, Aragon* and *Atenas,* are intended for the banana trade between the West Indies and the United States. They follow the cargo motorships *Leon* and *Lempa* built earlier by the Bremer Vulkan for the same owners. These ships have been built under the survey of the American Bureau of Shipping to receive the classification +A1 (E) A.M.S. R.M.C. and have main particulars as follows:—

Length o.a.	377ft. $3\frac{3}{8}$in.
Length b.p.	343ft. 3in.
Breadth moulded, feet	52
Depth to main deck	27ft. 3in.
Gross tonnage	3,677
Deadweight capacity on 21ft. $6\frac{3}{4}$in. draught, tons ...	3,430
Bin capacity, cu. ft.	155,000
Speed on banana draught, knots	$15\frac{1}{2}$

The hull and superstructure are entirely welded except the seams of the shell's side strakes, the stringer bar and the side framing. There are three continuous decks, a long forecastle extending over number 1 hatch, and a combined bridge and engine casing aft. The three banana holds are forward of the machinery and all three holds are of the same capacity. Each hatch is served by two 3/5-ton derricks. The cargo winches were supplied by AEG-Hatlapa, the windlass by AEG-Atlas-Werke, and the steering gear by John Hastie and Co., Ltd. The holds are vertically ventilated and the Freon-12 refrigerating plant was supplied by York Corporation. The holds are insulated by Fibreglass. Insulation is employed and the air ducts are of aluminium alloy. The main machinery consists of two seven-cylinder, single-acting, two-stroke Bremer Vulkan-M.A.N. Diesel engines developing together 3,920 b.h.p. at 190 r.p.m., and driving the single propeller shaft through Vulcan hydraulic couplings and reduction gear. The engines are of the same trunk piston design and cylinder dimensions as the five-cylinder engines of the twin-screw cargo motorships *Leon* and *Lempa.* Auxiliary power is provided by four 140 kW Diesel generators.—*The Marine Engineer and Naval Architect, August 1955; Vol. 78, pp.303-304.*



This general arrangement drawing of the Almirante *shows the three banana holds and cooler battery rooms*

Nylon Berthing Wire Tails

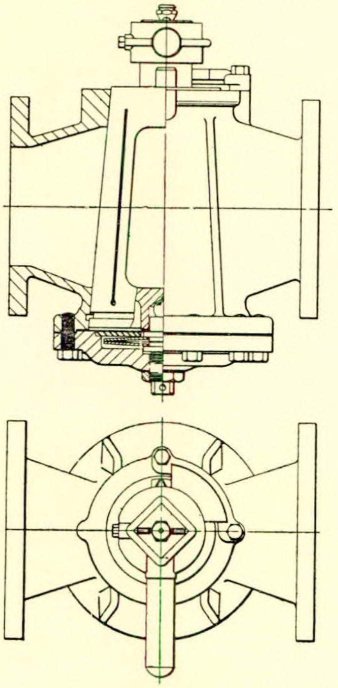
Additional security for vessels made fast in harbours subject to a heavy swell or tidal flow has been achieved by the use of Viking nylon tail combinations with fibre rope. When towing with steel wire rope, it is common practice to attach a short length of fibre rope to the wire rope and thus overcome the lack of elasticity. This principle has been applied by British Ropes, Ltd. to their fibre rope mooring lines to eliminate their vulnerability to shock loads. The Viking nylon tail combination comprises a six fathom length of nylon rope incorporated in the shore end of mooring lines, to which it gives additional elasticity and assists in passing the line to the shore or tug. When the manila portion of the line wears, it can be cut clear of the eye splice in the nylon tail and then turned end-for-end, and respliced into the nylon tail.—*The Marine Engineer and Naval Architect, September 1955; Vol. 78, p. 368.*

Russia's Largest Vessel

The Warnowwerft have recently handed over the 22,000 tons gross passenger liner *Sowjetski Sojus* after a lengthy reconstruction. She was the former Hamburg-American liner *Hansa,* ex *Albert Ballin* which was built by Blohm and Voss in 1922 and lengthened and re-engined in 1930. She was sunk during the war at Warnemunde where she was seized by the Russians. For the last few years she has been in the hands of this post-war yard for complete rebuilding and modernization. Certainly few would recognize her as she appears above from the somewhat solid looking former four-masted Hamburg-America liner. It is understood that she will now operate from Vladivostok. The rather similar ex-*Hamburg* of the same line was mined at Sassnitz in 1945, and is also undergoing modernization for Russian ownership under the name of *Yuri Dolgoruky.* The *New York* was broken up in this country after the war and the *Deutschland* was bombed and sunk at Lübeck.—*The Marine Engineer and Naval Architect, September 1955; Vol. 78, p. 361.*

New Vent Valves for Tankers

A number of the latest Shell tankers are now being fitted with a new type of valve in the vent system between the tank crown and the atmosphere. This valve is a modification of the lubricated taper-plug type, which is used very extensively throughout the oil industry. These valves are impervious to the effects of sea water corrosion, they ensure a vapour-tight seal, are easy to operate and require only routine lubrication. The Audco lubricated plug valve, as it is known, was approved by Shell Tankers, Ltd. on grounds of positive action and shut-off. The robust plug shank and 90 degree turn gives confidence to the operator that, when the plug is rotated, the vent is indeed open or shut as intended. Rotational stops are fitted and a pointer indicates clearly the port position. A wrench head with a short stub of non-sparking bronze is permanently fixed to the plug square and the valve is operated by a short length of pipe slipped over the stub.—*The Marine Engineer and Naval Architect, September 1955; Vol 78, p. 367.*



Audco taper-plug valve as used for tank vent lines

16-knot Motor Ship with Turbocharged Engine

The *Demodocus,* which completed her trials last month, is the first of six similar vessels ordered by Messrs. Alfred Holt and Co. (the Blue Funnel Line). She was built by Vickers-Armstrongs, Ltd., on the Tyne, whilst the second ship, the *Dolius,* was recently launched by Harland and Wolff, Ltd. The third, the *Diomed,* is being constructed at the Caledon yard and the other three by Vickers-Armstrongs, Ltd. The following are the main characteristics of the *Demodocus*:

Gross register, tons, about ...	7,800
Deadweight, tons, about ...	9,000
Length overall, feet ...	487
Breadth, moulded, feet ...	62
Depth, moulded, feet ...	35ft. 2in.
Loaded draught, feet ...	28
Machinery, b.h.p. ...	8,000
Service speed loaded, knots ...	16

She has the first two-stroke turbocharged engine built by Harland and Wolff (the other seven ships will have machinery of similar type), whilst a new electrical system of distribution has been employed in which there are two separate circuits, one for the engine-room and the second for all the other ship's services. So far as the commercial efficiency of operation is concerned, it is estimated that this vessel, maintaining over 16 knots at sea, fully laden, will require no more than 30-31 tons of boiler oil daily for all purposes, without the disadvantage of any higher cylinder liner wear than might have been anticipated with the engine running on Diesel fuel. The *Demodocus,* and her sister ships, are based, in design, upon the owners' *Bellerophon,* the dimensions being practically similar, but in place of seven-cylinder, non-turbocharged two-stroke machinery of 7,200 b.h.p. is a six-cylinder turbocharged Harland-B. and W. engine of 8,000 b.h.p., an output which, in the two final ships, may be raised to about 8,500 b.h.p. The reduction in engine length is considerable. Other alterations include the electrical distribution system as stated, some differences in deep tank arrangement, slight changes in the accommodation, and some hull details. The Archaouloff injection system is fitted to the engine and although this has now been applied to the machinery of the earlier ships, it was not originally installed in them. In this system the fuel pump, instead of being mechanically driven, is operated by gas from the engine cylinder, so that no camshaft is required. It is claimed that this arrangement, apart from saving weight due to the absence of the camshaft, enables a reduction in engine length of about 18 inches to be obtained (due to the absence of camshaft driving gear), is simpler in maintenance than the normal system, provides equally good or slightly better fuel consumption, needs less starting air, ensures perfect combustion at the right time, and leads to a reduction in cylinder liner wear. The fuel consumption on trials on Diesel oil was found to be 155 gr. per b.h.p. per hr. or 0·34 lb., whilst on boiler oil the figure is somewhat higher owing to the lower calorific value. In general design the engine is of the standard type which the builders have constructed for many years as an unsupercharged unit. In supercharging, the two Roots-driven scavenge blowers have been dispensed with and scavenging air is supplied from two exhaust-driven Napier turbo-blowers operating on the pulse system. The continuous speed is about 6,400 r.p.m. and the blowers are designed for a maximum inlet temperature of 1,200 deg. F., or about 650 deg. C. The propelling engine is, as stated, the first of the Harland and Wolff opposed-piston turbocharged type to be installed, and of this class, apart from the eight ordered by the Blue Funnel Line, six will be fitted in ships for the Bank Line, in addition to installations in other vessels now under contract. It has six cylinders and is designed to develop 8,000 b.h.p. at 112 r.p.m. in service, to suit the lines of the ship, but has a maximum continuous output of 8,500 b.h.p. At 8,000 b.h.p. the mean indicated pressure is the moderate figure of 100 lb. per sq. in. The engine has a cylinder diameter of 750 mm., the stroke of the main piston being 1,500 mm. and of the exhaust piston 500 mm., a total of 2,000 mm. In this ship, the standby and emergency generating plant comprises a Paxman vee-type engine driving a 280-kW. generator, but in the vessels following, including the *Dolius,* an Allen exhaust-gas turbine is to be installed, and this unit will come into operation automatically and so quickly, when required, that the generator will be on load within 30 seconds.—*The Motor Ship, October 1955; Vol. 36, pp. 276-282.*

Patent Specifications

Sea Water Evaporating Plant

This patent concerns evaporating and distilling plants in which sea water or estuarine water is evaporated for the purpose of removing undesirable solids which are present in the water. The vapour so produced is subsequently condensed, the condensate being suitable for consumption as drinking water or for use in high pressure steam boilers. In Fig. 5 is shown a double-effect evaporating plant with preheaters, with an arrangement whereby the chemical specified by the invention is added to the feed water in an open tank from which the feed water is subsequently pumped through the feed-heaters into the first-effect evaporator shell. The chemicals stipulated are ferric chloride or sulphate, or aluminium chloride or sulphate, or zinc chloride or sulphate, or a combination of such salts, or an aqueous solution of one or more of these salts to prevent the formation of calcium carbonate scale and magnesium hydroxide scale. Heating steam is supplied to the first-effect evaporator by way of the pipe (1) to the heating surface (2) of the first-effect shell (3). Sea water is drawn by the pump (5) through the pipe (4) and discharged through the air-ejector-condenser (6) and condenser (7) to the discharge pipeline (8). Some of the water is rejected by way

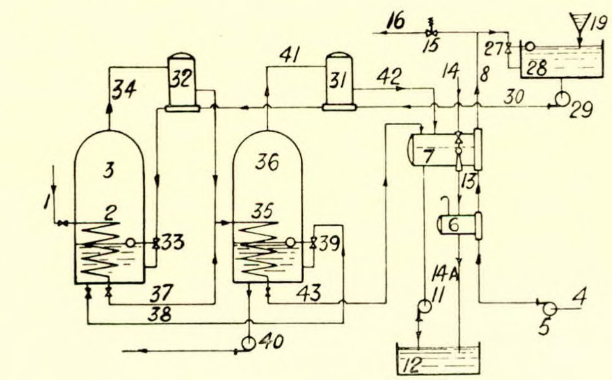
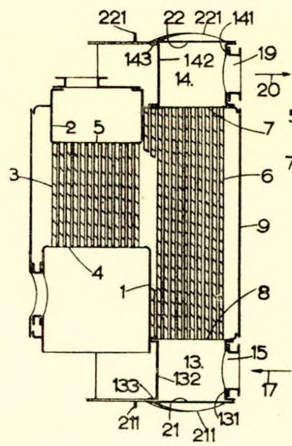


FIG. 5

of the spring-loaded valve (15), while the remainder of the water passes by way of a regulating valve (27) into a tank (28). The tank (28) receives all the sea water which is being fed into the evaporator shell (3). The chemical compound is added by way of the funnel (19) to the sea water in the tank (28), whence it is withdrawn by a pump (29) and discharged through a pipe (30) and preheaters (31) and (32) and a feed regulator (33) into the first evaporator shell (3). Vapour formed by the transfer of heat across the heating surface (2) leaves the evaporator by way of a vapour outlet pipe (34) and passes into the shell of the preheater (32). In this preheater, a small amount of vapour is condensed by the feed water passing through the preheater (32), the resultant drainage and any uncondensed vapour passing to the heating surface (35) of the second-effect shell (36). Drainage resulting from the condensation of the heating steam in surface (2) passes through pipe (37) to the heating surface (35) of the evaporator (36). Brine from the evaporator (3) passes by way of pipe (38) and feed regulator (39) into the shell of the second-effect evaporator (36). Brine from the evaporator (36) is withdrawn by a pump (40) which discharges the brine to waste. Vapour resulting from the transfer of heat across the heating surface (35) leaves the evaporator (36) through a pipe (41) and passes into the preheater (31) through which the feed to the evaporating plant is passing. A certain amount of vapour is condensed in heating the feed water. The non-condensed vapour and the drainage which is formed, leave the preheater (31) through pipe (42) and enter the condenser (7).—*British Patent No. 734, 755, issued to G. and J. Weir, Ltd., and F. B. Ling. Complete specification published 10th August 1955.—Engineering and Boiler House Review, October 1955; Vol. 70, pp. 355.*

Combined Directly-fired and Waste-heat Water-heater or Boiler

This invention relates to improvements in combined directly-fired and waste-heat water-heaters or steam boilers in which a nest of vertical fire tubes having a furnace for direct firing together with a nest of vertical fire tubes for heating by waste-heat gasses are located in a shell common to both nests. The invention has particular reference to water-heaters or steam boiler in which the waste-heat recovery section has helically grooved vertical fire tubes fed with the pulsating high speed flow of hot gases discharged by Diesel and other internal combustion engines. It is another object



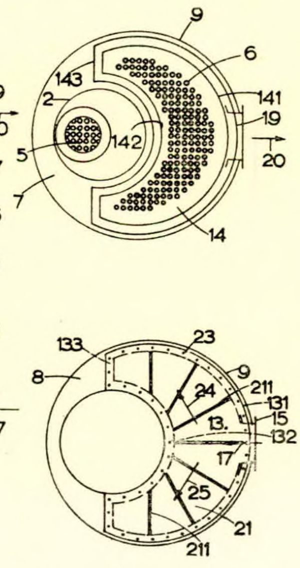


FIG. 2 FIG. 3—*(top)*, FIG. 4—*(bottom)*.

of the invention not only to silence the exhaust from such engines, but also to alleviate the "drumming" that occurs with some waste-heat recovery devices. In Figs. 2, 3 and 4, there is a firebox (1), a flue-box (2) and the helically-grooved tubes (3), the lower ends of which are expanded into the lower tubeplate (4). Their upper ends are expanded into the upper tube-plate (5) to form the directly-fired section of the composite water-heater or steam boiler. The waste-heat recovery section comprises the helically grooved fire tubes (6), the upper ends of which are expanded into the lower tube-plate (8), the upper tube-plate (7) and the lower tube-plate (8) forming the crown plates of the shell (9) common to both sections. The helically-grooved vertical fire-tubes (6) are arranged in uniformly-spaced relation to form a crescent shaped nest, and the tube-plates (8) and (7) respectively are provided with a crescent-shaped inlet box (13) and a crescent-shaped outlet box (14). The inlet box (13) has a side branch (15) in its convex wall (131) for connexion to the hot waste gas supply line indicated by the arrow (17), whilst the outlet box (14) has a side branch (19) in its convex wall (141) for connexion to the exhaust-gas delivery line indicated by the arrow (20). The axis of the inside surface of the concave wall of the crescent-shaped box and the axis of the inside surface of the convex wall of that box lie in a plane containing the longitudinal axis of the side branch connected to the hot waste-gas supply line. The inlet-box (13) has a horizontal flange (133), for the coverplate (21) stiffened by the external ribs (211), and the outlet box (14) has a horizontal flange (143) for the cover-plate (22) stiffened by the external ribs (221). Both the cover-plates (21) and (22) are secured by bolts (23) and joined for purposes of easy removal when the tubes (6) need cleaning or replacing. In this way the waste-heat recovery section can be serviced and retubed without any disturbance of the hot waste-gas supply line or of the exhaust-gas delivery line.—*British Patent No. 732,752, issued to Spanner Boilers, Ltd. Complete specification published 29th June 1955. —Engineering and Boiler House Review, September 1955; Vol. 70, pp. 314-315.*

H. and W. Constant Combustion Device

In Fig. 1 is illustrated a constant pressure combustion device, comprising one or more expansion cylinders in connexion with the compression space of the engine cylinder. The piston of the expansion cylinder is loaded to the desired combustion pressure by air from the starting system. The diagram applies to a two-stroke engine of the opposed piston type, with a branch (4) communicating with the combustion space between the main pistons (2, 3). In the expansion cylinder (5) is a piston (6), integral with a larger diameter piston (7). Air enters through the inlet (8). An escape port (10) is provided, and there is a pneumatic cushion for the pistons at the end of the upstroke. The combustion pressure drives down the expansion piston, and as the main pistons (2, 3) move apart on the power stroke, the piston (6) returns the contents of the expansion chamber (5) to the main cylinder and remains at the top of its stroke until compression and fuel ignition repeat the engine cycle. The pressure applied by the air admitted through the port (8) is such that the piston (6) is held at the top of its stroke with sufficient pressure to allow the compression in the engine cylinder to reach that required for ignition. Before the end of compression, when the piston (6) is driven down by the combustion pressure, the effect is to ensure substantially constant pressure as the compression stroke is completed and the power stroke commences. (*Patent No. 726,337. W. A. Harper, C. C. Pounder, and Sir F. E. Rebbeck, Harland and Wolff, Ltd., Belfast.*)—*The Motor Ship, June 1955; Vol. 36, p. 118.*

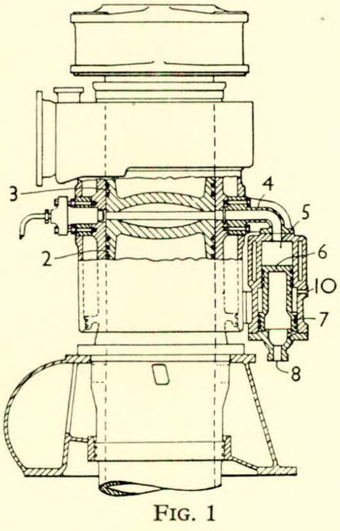
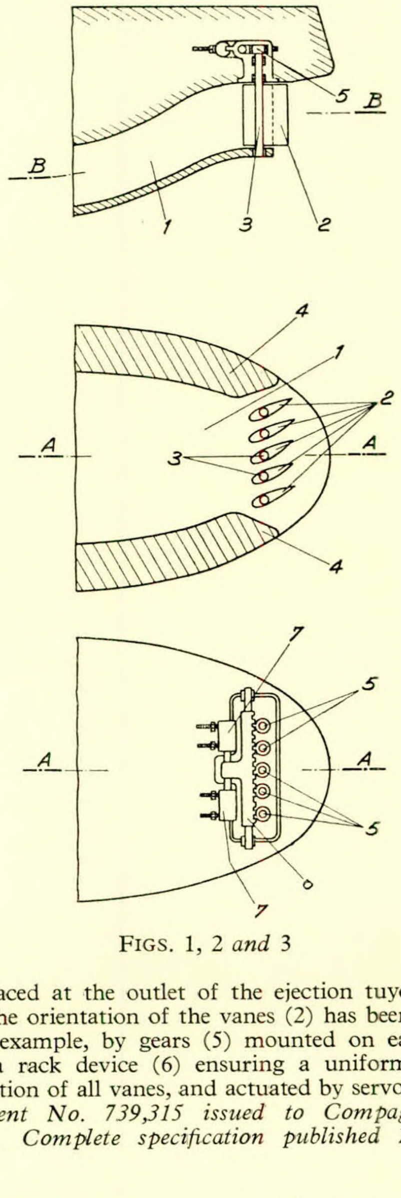


FIG. 1

Control Arrangement for Jet Propelled Marine Craft

This invention relates to control gear arrangements for jet propelled marine craft. In Figs. 1 to 3 the liquid ejected by a jet propulsion plant (not shown in the drawings) strikes a vane grid (2), the vanes of which are movable about their vertical



FIGS. 1, 2 *and* 3

axes (3), placed at the outlet of the ejection tuyere (4). The control of the orientation of the vanes (2) has been represented, by way of example, by gears (5) mounted on each vane and driven by a rack device (6) ensuring a uniform and simultaneous rotation of all vanes, and actuated by servomotors (7).—*British Patent No. 739,315 issued to Compagnie Electro-Mecanique. Complete specification published 26th October 1955.*

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