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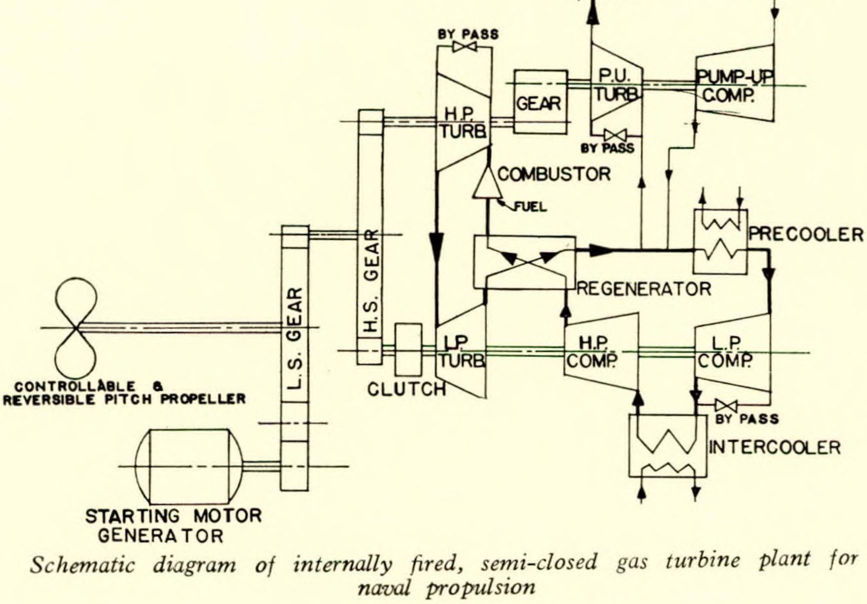
Marine Engineering and Shipbuilding Abstracts

Volume XX, No. 1, January 1957

*Patent Specifications

Internally Fired, Semi-closed Cycle Gas Turbine Plant

This paper describes an internally fired, semi-closed cycle gas turbine plant built for the U.S. Navy by the Westinghouse Electric Corporation for a special propulsion application. During initial plant tests the Navy determined that the special propulsion application was no longer necessary to its overall programme. Review of the cycle and plant configuration showed that, with minor modifications, the plant could be used to demonstrate the feasibility of the internally fired closed cycle unit for man-of-war applications. Tests were therefore conducted which confirmed the following expected advantages of this cycle: (1) essentially constant efficiency over a wide load range (20 per cent to 100 per cent) by variable pressure level; (2) low air consumption resulting in small inlet and exhaust ducting due to recirculation of cycle gases; (3) low space and weight requirements due to supercharging of the main components; and (4) ease of control, because the plant is basically a two-shaft unit. The gas turbine unit is designed to operate as follows: The recirculated gas is compressed in the low pressure compressor, intercooled, and further compressed in the high pressure compressor. The gas then passes to the regenerator to be heated by the turbine exhaust gases. It is raised to turbine inlet temperature in the combustor and enters the high pressure turbine, which is the power turbine. The



Schematic diagram of internally fired, semi-closed gas turbine plant for naval propulsion gases pass to the low pressure turbine which drives the compressors. Exhaust from this turbine after regeneration is ducted to both the pump-up turbine and the precooler. The pump-up turbine flow is made up by the pump-up compressor at a rate sufficient to provide oxygen for combustion at all operating conditions. This make-up flow is cooled with the recirculated gases in the precooler to complete the circuit to the low pressure compressor. The pump-up set is not self-sustaining, and a small amount of power is supplied to it at all conditions by a mechanical connexion to the power turbine. A bypass valve around the pump-up turbine is used for both part load control and to avoid surge of the pump-up compressor at reduced speed. Bypass valves are also provided for the low pressure compressor and the power turbine for plant starting, and the low pressure turbine is clutched to the gear to obtain simultaneous cranking of all equipment used throughout the plant. Fouling and corrosion of the internally fired, semi-closed cycle gas turbine were evaluated by the tests. Conventional gas turbine components are satisfactory for low sulphur fuel operation, and with additional precooler equipment development it is expected that high sulphur fuel operation will be achieved.—*S. H. DeWitt and W. B. Boyum; 1956 A.S.M.E. Gas Turbine Power Conference; Paper No. 56-GTP-16.*

Tank Cleaning by Vapour Injection Emulsion Method

The selection of a cleaning technique for the removal of oil contamination and scale from cargo tanks depends on many factors. The more important considerations include the extent and nature of the contaminant, extent and degree of scale formation, type of repair or alteration, and the use for which the cleaned tanks are intended. Several different methods are used for tank cleaning, and each has its own limitations and advantages. The Butterworthing method using hot sea water is the present accepted U.S. Navy process for cleaning cargo tanks. Steam cleaning, both with and without additives, and hot oil cleaning are widely used, also. Once the tank surfaces are rid of the oily contaminants, other methods can be used to advantage if further cleaning and scale removal are needed. A method which has been developed for removal of oil residues in tanks covered with heavy rust scale is the vapour injection emulsion cleaning process described in detail in this article. The emulsion cleaning of fuel oil tanks by the vapour injection process is based upon the diffusion of finely divided particles of cleaner by the movement of steam to the relatively cold surfaces of the tank. Condensation takes place then, and an emulsion is formed. Preheating by steam lowers the viscosity of Diesel oil—the solvent component of the emulsion cleaner—and aids in the penetration of the emulsifying agent. Oversteaming, however, should be avoided, since it raises the temperature of the side walls too high, lowering the condensation rate and decreasing the cleaner efficiency. Six steps in the procedure for cleaning a cargo tank by the emulsion method are: (1) Ventilate the tank with a suction blower. The tube is extended to the bottom of the tank through the manhole. The displace air is replaced by entrance of fresh air through the manhole. Ventilation should be continued until the space is declared safe for entrance. (2) Remove sludge by pumping. Any type of sump pump that will lift the heavy sludge is satisfactory. The pump should be placed in the tank and the suction hose extended to the lowest point to evacuate as much sludge as possible. After the sludge has been removed, the pump and all hoses should be removed from the space. (3) Open the tank vent to make sure pressure cannot build up when steaming takes place later. (4) Place on the manhole and bolt down the chemical injector (Figs. 1 and 2) which is mounted on a portable manhole cover made of

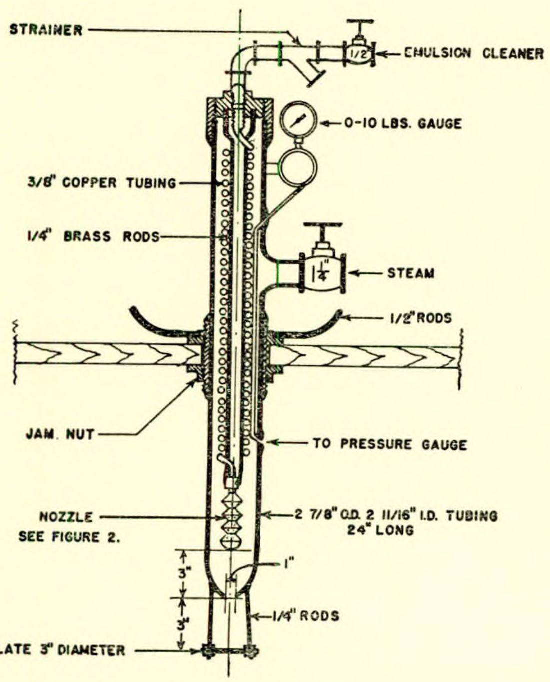


FIG. 1—*Chemical injector*

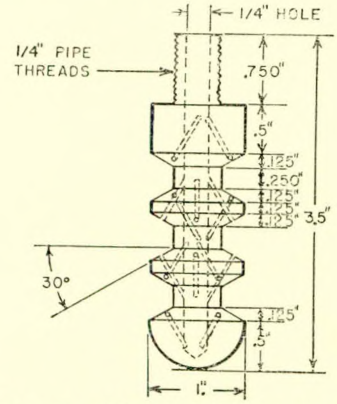


FIG. 2—*Chemical injector nozzle*

Drill twenty-four $\frac{1}{32}$-in. holes, four holes 90-deg. on each surface. Holes in opposite surfaces aligned exactly. Three groups of four pairs, each group 30-deg. offset

waterproof plywood with the same dimensions as the regular manhole cover. Note that the lower end of the injector is extended below the level of the deck to obtain best results. A steam line of 1½ inches or larger is connected to the valve provided on the chemical injector. The steam pressure on the line should be 100lb. per sq. in. (5) Close all other openings except the vent, which should be left open at all times, and admit steam at full pressure for one or more hours. The pressure gauge on the injector should show no pressure when steaming. If it does, shut off the steam immediately and check the tank vent. When the temperature, as indicated by the thermometer in the manhole cover, reaches from 175 to 180 deg. F., the tank is ready for the injection of the emulsion cleaner. (6) Supply the emulsion cleaner to the injector by a chemical injection pump. The pump should be of the positive displacement type with a capacity of approximately 5 gals. per min. at 100lb. per sq. in. The exit side of the pump should be equipped with an adjustable pressure bypass back to the intake of the pump. The pipe to the nozzle of the injector should be equipped with a gauge so that the proper nozzle pressure can be determined and kept constant by adjustment of the bypass. The pump discharge should be connected to the chemical injector, the pump should be started, and the emulsion cleaner pumped through the nozzle of the injector. The solution is forced through the nozzle at the rate of 3 gallons a minute. The amount of cleaner required will depend upon the size of tank being cleaned. Fifty gallons will clean a 1,200-barrel tank, and 150 gallons will clean a 6,000-barrel tank. (7) During the injection of the cleaner, the steam pressure should remain on the injector. (8) After the emulsion cleaner has been injected, run cold water through the pump and lines to the nozzle for 15 to 30 minutes while steaming. A wet steam will be produced, causing excessive sweating on the walls of the tank, which emulsifies the oil and washes the emulsion to the bottom of the tank. (9) As soon as the tank is cool enough to enter, wash it down with a jet of high pressure fresh water. The sump pump should be lowered into the tank and the suction line placed in the lowest point. The pump should be started and the emulsified oil and water discharged from the tank. If the tank is washed down before it becomes completely cold, no wiping is necessary to remove the final traces of oil which cling to the lowest point of the tank. The emulsion cleaner is made up of an amine-oleate soap dissolved in Diesel oil. The composition of the cleaner is: Two-and-a-half gallons of oleic acid (red oil); one gallon of triethanolamine or half-gallon monoethanolamine; forty-five gallons of Diesel oil. In compounding the formula, the oleic acid is dissolved in the Diesel oil until a homogeneous solution is formed. Then the amine is added slowly with vigorous stirring in the given proportions.—*Bureau of Ships Journal, August 1956; Vol. 5, pp. 17-19.*

Dutch Tanker

The 24,815-ton d.w. motor tanker *Astrid Naess,* built by the Netherlands Dock and Shipbuilding Company, Amsterdam, has now entered the service of her owners, the Nederlandse Norness Scheepvaart Maatschappij, Amsterdam. The principal characteristics of this vessel are as follows:—

Length overall	187·35 m.
Length b.p.	176·80 m.
Breadth	23·32 m.
Depth	12·975 m.
Draught	10·00 m.
Deadweight, tons	24,815
Displacement, tons	32,350
Gross tonnage	16,278
Net tonnage	9,432
Cargo capacity, cu. ft.	567,694
Dry cargo capacity, cu. ft.	27,104
Fuel oil capacity, tons	2,225
Lubricating oil capacity, tons	69
Boiler fuel capacity, tons	72
Fresh water capacity, tons	240
Feed water capacity, tons	50
Main propelling machinery: One N.D.S.M.-Stork Diesel engine of 8,400 b.h.p. at 115 r.p.m.	
Trial speed, knots	15·56

The *Astrid Naess* is of the normal tanker type with poop, bridge and forecastle erections, raked plate stem and cruiser stern. The ship has one mast while a combination signal and radar mast is fitted on the top deck of the bridge deckhouse. Longitudinal framing has been employed in the construction of the cargo section, transverse framing being employed forward and aft of this section. The ship is of welded construction with the exception of the bilge lap-joints, keel plate and stringer angle, which are riveted. Aluminium is employed in the construction of the bridge deckhouse insofar as the plating is within a 13-ft. radius of the steering compass. The bulkheads are of the corrugated type, the two longitudinal bulkheads being of the horizontally, the transverse bulkheads of the vertically corrugated, type. The cargo section is subdivided by these bulkheads into ten triple tanks, all of them fitted with cast-iron heating coils. The main propulsion machinery consists of an 8-cylinder, single-acting, two-stroke supercharged Stork-N.D.S.M. Diesel engine of 8,400 b.h.p. at 115 r.p.m. The cylinder liners are Porus Krome hardened. Also fitted are two auxiliary sets, each consisting of an 8-cylinder four-stroke Stork-Ricardo Diesel engine coupled to a D.C. generator of 200 kW, 220 volts. The cylinder diameter of the engines is 200 mm., the stroke 300 mm. They have an output of 300 h.p. at 565 r.p.m. One of the engines is connected by a friction coupling to a starting air compressor at an end-pressure of 25 kg. per sq. cm. In addition, there is one "Reader" vertical enclosed two-crank compound steam engine of the forced lubricated type. This engine drives a 75-kW generator, while a Reavell starting air compressor is driven by another "Reader" steam engine. A Caird and Rayner evaporator with a capacity of 20 tons per 24 hours is fitted to increase the boiler feed water capacity. There is a Stork exhaust gas boiler designed to deliver 62,000 kg. per hr. with an entrance temperature of 295 deg. C.—*Holland Shipbuilding, August 1956; Vol. 5, pp. 32-34.*

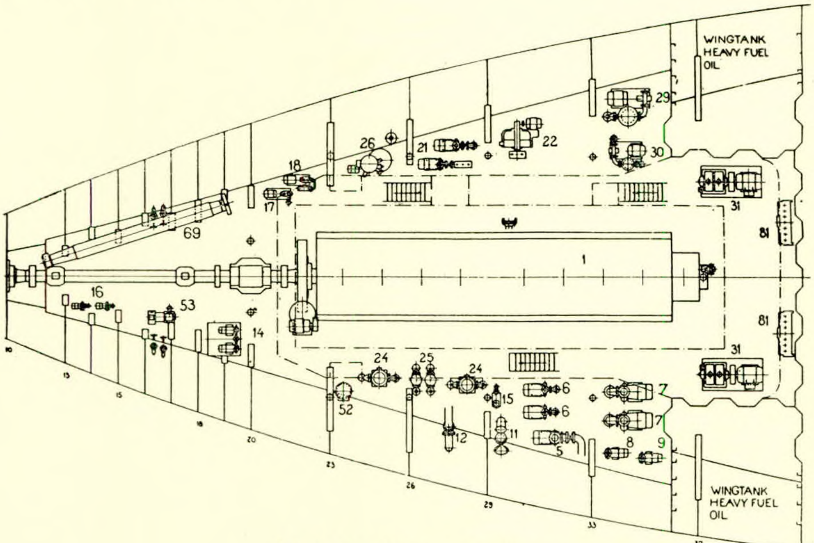
Marine Auxiliary Generator Installation

A 350-kW gas turbine set, the first of a number ordered by Alfred Holt and Company, for installation in a group of new motor ships built or building for the Blue Funnel Line, is now installed is the m.v. *Dolius,* a 16,800 gross tons displacement cargo liner which completed its trials in January of this year. Since then, two voyages to the Far East have been completed in addition to coasting in home waters. According to Ministry of Transport regulations, an emergency generator set must be located outside the main engine room and above the bulkhead deck. In the *Dolius* it is mounted on the boat deck immediately behind the funnel. This latest Allen G.T. to go into service is of the simple, open-cycle, non-recuperative type specially designed for shipboard emergency generation. It is a robust one-shaft machine with single-stage, centrifugal compression, a single combustion chamber and a two-stage axial turbine. The turbine rotor directly drives the compressor through Hirth-type couplings and also the alternator via a 10:1 epicyclic reduction gear. Both G.T. and alternator are mounted on a common bedplate and, apart from fuel and starting-air connexions, etc., the set is completely self-contained. Arrangements are provided for fast automatic starting in the event of failure of the ship's electrical supply; for instance, the fuel system is kept constantly warm with electric heaters and the bearings are always primed with a continuously running, motor-driven, lubricating oil pump. Originally it was intended that the set should automatically come into operation as soon as the ship's voltage fell below a minimum level, but this first machine has been arranged for manual starting only. Provided all trips and other devices are already set to the starting position, this operation is confined to the pressing of a button. While the *Dolius* was docked in Hamburg recently, a new compressor-turbine unit was fitted, incorporating a modification to the compressor diffuser. This resulted in an increase in power output and thermal efficiency. Cartridge starting has been replaced by a compressed air system. An exhaust temperature trip has been fitted to limit outlet temperature to 630 deg. C., and extra lagging has been fitted. Initially, starting was by a twin-breech Rotax cartridge starter, the turbine of this device applying a torque to the free end of the main shaft. Teething troubles, however, led to the conversion, in the *Dolius,* to a compressed air starting system. A Rotax air turbine is now mounted on top of the reduction gearbox and drives via a self-releasing clutch and through the main gears, to the engine shaft. Air normally at 350lb. per sq. in. gauge is supplied from two receivers installed in the engine room. Ordinarily these bottles are recharged from the ship's main compressed air supply but in the event of failure of this service, a Petter air cooled Diesel-driven reciprocating compressor can be used and this machine is installed also in the engine room exclusively for this task. On pressing the button, full speed

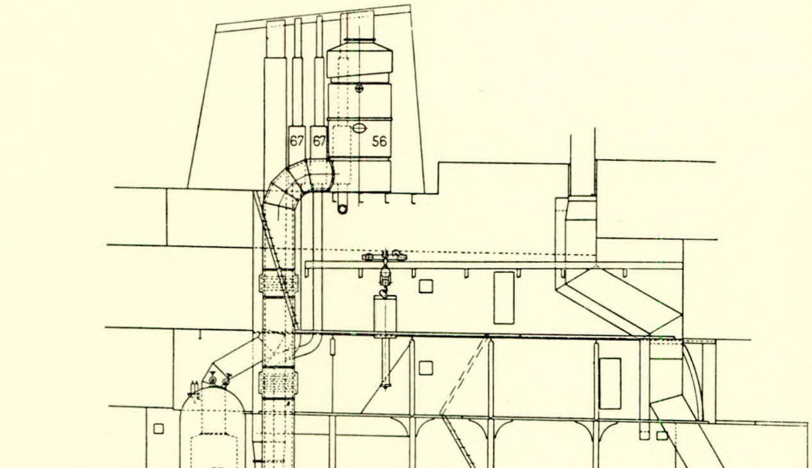
(15,000 turbine r.p.m.) is reached in 36 seconds, with a starting air pressure of 260lb. per sq. in gauge. Full load is attainable in a further 11 seconds, this time lag being a function of voltage build-up in the generator rather than rate of load acceptance by the turbine. Minimum permissible starting air pressure is 250lb. per sq. in. gauge. One receiver only is used and, for the start mentioned above, a drop of 30lb. per sq. in. gauge occurred. Thus, it is evident that at least six starts can be obtained if necessary from the two receivers before recharging.—*The Oil Engine and Gas Turbine, September 1956; Vol. 24, pp. 195-196.*

New Design of Cargo Ship

Requirements for cargo ships are changing owing to the increasing demand for carrying bulk cargoes, and the adoption of higher speeds for tramp ships. New types of vessel are, therefore, being developed, and one of the most interesting of these is the *Ingwi*, which was completed recently by Kockums Mek. Verk. for Kommandittselskapet Harwi (Rolf Wigand), Bergen. This vessel has many features of particular interest. She is intended to be employed in the tramp trade, but has been built to carry bulk cargoes and ore more efficiently than the normal dry-cargo ship. The vessel is convertible between an



Plan at engine room floor.



Elevation looking to port.
Engine room of the Ingwi

(1) Main engine; (2) Auxiliary engines; (3) Generators; (4) Harbour generator; (5) Fresh water pump for main engine cylinders; (6) Fresh water pumps for main engine pistons; (7) Sea water pumps for main engine; (8) Fresh water pump for auxiliary engines; (9) Sea water pump for auxiliary engines; (10) Fresh water coolers; (11) Cooling water inlet for main engine; (12) Cooling water outlet for main engine; (13) Cooling water strainer for main engine; (14) Double cooling aggregate for fuel valves; (15) Drain pump for main engine telescopic pipes; (16) Cooling water pumps for refrigeration machinery; (17) Sea water hydrofor pump; (18) Fresh water hydrofor pump; (19) Sea water hydrofor; (20) Fresh water hydrofor; (21) Fire extinguishing pumps; (22) Bilge pump; (23) Lubricating oil separating tank; (24) Lubricating oil pumps; (25) Lubricating oil strainers; (26) Lubricating oil purifier; (27) Lubricating oil coolers; (28) Lubricating oil magnetic strainer; (29) Transfer pump, heavy oil; (30) Transfer pump, Diesel oil; (31) Compressors; (32) Emergency air compressor; (33) Starting air receivers; (34) Low pressure air receiver; (35) Starting air bottle for auxiliary engines; (36) Heavy oil circulating pump for main engine; (37) Turbo filter; (38) Strainer for fuel oil; (39) Diesel oil purifier; (40) Water heater for fuel oil purifier; (41) Clarifier; (42) Purifier; (43) Steam heaters for fuel oil purifier; (44) Water tank for fuel oil purifier; (45) Steam heater for heavy oil; (46) Fan for purifier room; (47) Storage tanks for machine oil; (48) Storage tanks for cylinder oil; (49) Storage tanks for compressor oil; (50) Storage tanks for kerosene; (51) Storage tanks for trichlorethylene; (52) Trichlorethylene evaporator; (53) Feed water pumps; (54) Feed water strainer; (55) Boiler; (56) Exhaust-gas boiler; (57) Main steam box in engine-room; (58) Observation tank; (59) Condensed water tank; (60) Expansion vessel; (61) Vacuum pump for fresh water generator; (62) Condensed water pump for fresh water generator; (63) Pump for fresh water generator; (64) Heat exchanger for fresh water generator; (65) Fresh water separator; (66) Fresh water heater for accommodation; (67) Silencers for auxiliary engines; (68) Refrigeration compressors; (69) Spare tailshaft; (70) Emery grinding machine; (71) Drilling machine; (72) Lathe; (73) Workbench; (74) Main switchboard; (75) Electrical store; (76) Engine store; (77) Spare piston; (78) Spare cylinder liners; (79) Spare cylinder covers; (80) Sludge ejector; (81) Valve chests; (82) Fine filters; (83) Fan for engine room; (84) Electric heater for lubricating oil purifier

open and a closed shelterdecker. She has machinery aft, is equipped with Diesel-driven alternators for the supply of current throughout the vessel, and has hydraulically operated winches. The *Ingwi*, like her sister ships, of which there are four, for four different Norwegian owners, is a wholly welded vessel and has been constructed to the highest class of Det Norske Veritas. She has been completed as a closed shelterdecker with a deadweight capacity of 13,400 tons on a draught of 31ft. 2¾in. The main characteristics are:—

Length overall	486ft. 6¾in.
Length b.p.	450ft. 0in.
Breadth moulded...	61ft. 0¼in.
Depth to main deck	31ft. 1in.
Depth to shelter deck	40ft. 0½in.
Cargo capacity, cu. ft. grain ...	678,060
Ballast capacity, tons	3,639
Bunker capacity, tons	1,117
Gross register (closed), tons ...	8,991
Net register (closed), tons ...	5,083
Deadweight capacity (closed), tons	13,400
Deadweight capacity (open), tons	11,200
Draught (closed)...	31ft. 2¾in.
Draught (open)	26ft. 11in.
Trial speed, knots	14¾
Daily fuel consumption, tons ...	18-20
Machinery, b.h.p. at 125 r.p.m.	5,340

The propelling engine, although a standard M.A.N. type, is the first of its class built at Kockums. It is of the single-acting two-stroke design with eight cylinders 700 mm. in diameter, with a piston stroke of 1,200 mm., the rated output being 5,340 b.h.p. at 125 r.p.m. It is arranged for operating on boiler oil which is separated on the purifier/clarifier system, the machines being of the De Laval type. On trials, when light, the vessel averaged 16·22 knots with the engine developing 4,855 b.h.p.—*The Motor Ship, September 1956; Vol. 37, pp. 216-220.*

Influence of Geometry upon Ultrasonic Defect Determination

The shape of a test object, the test standards used, and the surface condition of the test object are among the factors that affect ultrasonic flaw detection in its ability to indicate the true size of a defect or flaw. In the testing of large steam-turbine rotor and generator field forgings, two such factors that somewhat dominate attempts to correlate the magnitude of ultrasonic indications and the defect sizes involved are the defect orientation and the geometric configuration of the test forging, i.e. the outside and bore diameters. Methods are presented to measure the angle at which the defect is ledged, along with the correction factors that must be applied to an indication to eliminate this effect. Also presented is a study of the forging geometry and its influence upon the testing sensitivity, with its subsequent effects upon defect size determination.—*S. Serabian, Nondestructive Testing, 1956; Vol. 14, No. 4, pp. 18-21.*

Effect of Model and Tank Size on Resistance Results

The effect of model and tank size on resistance results is discussed in general terms. A method is developed for determining viscous and blockage effects in resistance tests of a series of geometrically similar models carried out in more than one tank. The method is applied to two series of tests, one for a fast steamer and the other for a cruiser, made originally at the Hamburg Model Experiment Establishment and supplemented by work at the National Physical Laboratory.—*Paper by G. Huges, submitted to the Institution of Naval Architects for written discussion, 1956.*

Marine Gas Turbines in Naval Service

After some 33,000 hours of operating experience with marine gas turbines in naval vessels, the reliability, economy, ease of operation, and quick replacement of this type of prime mover may be considered to be established. Naval applications in the United States of marine gas turbines have increased rapidly during the past few years, not only as main propulsion units but also to drive emergency and auxiliary generators. At the present time, a total of fifty-five vessels of the U.S. Navy are in operation with gas-driven engines installed. These include a destroyer, landing craft, and a number of minesweepers. The number of engines installed include 200 individual units. At the beginning of November 1955, the total installed horsepower was 32,564 and the operating time of the engines, most of them driving auxiliary generators, was 33,327 hours. One reason for the wide applications of gas turbines is their economy. For instance, a comparison of the maintenance costs shows: gas turbines on tanker *Auris*—0·06 cent per kW hr.; Brown Boveri gas turbines (based on service life of 1,000 hr.)—0·23 cent per kW hr.; General Electric Belle Isle gas turbine—0·004 cent per kW hr.; Diesel engines—0·30 cent per kW hr.; steam turbines (without boilers)—0·49 cent per kW hr. Details of experience gained in the operation of the various units in service are given in the paper.—*W. M. M. Fowden, Jr. and J. W. Sawyer, A.S.M.E. Paper No. 56-GTP-7, read at Gas Turbine Conference, Washington, 16th-17th April 1956. Journal, British Shipbuilding Research Association, July 1956; Vol. 11, Abstract No. 11,804.*

Diesel Engines or Steam Turbines for High-speed Merchant Vessels?

This article gives a comparison of Diesel engines and steam turbines for the propulsion of Spanish high-speed merchant vessels. The weights, fuel consumption, cost of fuel, cost of repair work and maintenance, depreciation of the ship, and other factors are compared in considerable detail. The example relates to a 10,000-ton ship, having a service speed of 17 knots, and machinery developing 8,000 s.h.p. It is found that a modern turbine-driven ship of this size is less expensive than a motor ship, that the operating costs are also lower, and that it carries a greater deadweight; all other conditions being equal, it has more space available in its holds, so that the space available for the cargo is considerably increased, the vibration is much less than motor ships, it is much more stable in operation, and there are various other advantages. On a voyage of 7,000 miles, the steam ship can carry 500 tons more than the motor vessel, and the difference between the annual running costs of the steam ship and of the motor vessel amount to about 2½ per cent of the total cost of the ship in favour of the steam ship. It is also noticed that if Velox boilers are used, the fuel economy is even greater than quoted in the example. A list of Spanish manufacturers making suitable boilers and turbines is given.—*J. de Tellaeche, Dyna, April 1956; Vol. 31, p. 329. Journal, The British Shipbuilding Research Association, July 1956; Vol. 11, Abstract No. 11,816.*

Recent Trends in Ship Design and Construction

The purpose of the paper is to relate the present trends in ship design to the fundamental principles of improving economy, efficiency, and safety, both in the construction and operation of merchant vessels. Modern trends in design toward reducing the initial and subsequent operating costs are considered, together with the greatly improved standards of safety currently being adopted. The initial cost of a vessel depends largely upon the owner's requirements, statutory requirements, the current cost of materials and labour, and the shipbuilder's efficiency. Design trends intended primarily to reduce the initial cost are directed toward the utilization of the properties of plates and sections to their best advantage, consistent with ease of construction, and are seen in the current adoption of longitudinal deck and bottom framing, continuous hatch side coamings and girders, and corrugated bulkheads. Improved operational efficiency is shown to result from the following recent developments: (1) Placing the machinery aft. (2) Improving living conditions on board. (3) Improved hatch design and cargo handling methods which reduce the time in port. (4) Improved deadweight-displacement ratios. (5) Reduced maintenance costs. (6) The evolution of new ship types. The improved safety features are considered and are shown to be dependent on statutory and classification requirements which are subjected to frequent modifications as wider experience is gained through thorough investigation into the reasons for ship structural failures and losses of life and cargo. An example is quoted of this evolutionary process with regard to the development of notch-tough steel. The employment of optional navigation equipment, such as automatic steering systems and position finding devices, is also discussed and it is finally suggested that the rate of design development has never been greater than at present, due in no small part to the greatly improved facilities for the exchange of technical data and experience currently available through such societies as this and the world-wide technical press.—*Paper by J. D. Hearshaw, abstracted in S.N.A.M.E. Bulletin, July 1956; Vol. 11, p. 43.*

Rolling and Pitching of a Ship at Sea

An analysis is made of the relation between the rolling and pitching of a ship (r.r.s. *Discovery II*) at sea and the waves recorded simultaneously by a shipborne wave recorder. Most of the observations were made with the ship steaming at 6½ knots on a series of twelve courses in a long-crested wave system, but good results were also obtained with the ship head-on to short-crested storm waves. The method is based primarily on spectral analysis by means of an automatic Fourier analyser, but an important extension is made to show the connexion between the spectra of ship motions and their statistical properties. The practical results are compared with theoretical relations derived by methods which ignore certain effects but are simple to apply, and advanced methods which estimate some terms with greater precision but at the cost of greater computational labour. The theoretical results are in general quite close to, and never very different from, the practical results, the differences being due largely to statistical variations. Attention is drawn to the fact, often not realized, that all measurements at sea are subject to considerable statistical errors which can only be reduced by increasing the duration of recording. This does not matter for the purpose of prediction, but it is important in the interpretation of records.—*Paper by D. E. Cartwright and L. J. Rydill, submitted to the Institution of Naval Architects for written discussion, 1956.*

Work Carried out in Model Tanks

Modern experiment tanks are now usually equipped with a wavemaker, and by driving a model through manufactured storms of different intensities, a start has been made in finding out what happens to the propulsive efficiency and the behaviour of a ship after her acceptance trials, when she goes into service. Among the many troubles experienced by ships at sea in bad weather conditions, are "slamming", poor course keeping, unseakindliness, and vibrations, and the way in which experiment tanks are tackling these problems serve to illustrate their usefulness to the shipping industry. "Slamming" is the name given to the blow a ship sometimes receives in rough weather, which is accompanied by a loud report and causes a shudder to pass through the hull structure with a quick period vibration. If the magnitude of the impulse causing the slam exceeds the elastic limit of the plating and framing of the hull, permanent damage occurs to the bottom plating and floors close to the keel, usually between one-eighth and one-tenth the ship's length from the fore perpendicular. There are at least three different kinds of slamming:—viz. "panting", which occurs in the outer bottom plating close to the L.W.L. forward when a very fast ship is driven through a swell, and this is cured by fitting "panting stringers". "Pounding" takes place when a high-speed skimming vessel encounters waves, or when a seaplane porpoises on the water surface, and it is minimized by V-ing the floors forward. Tankers "flying light" with all tanks empty during cleaning operations sometimes suffer from this form of slamming. There is also a third kind of slamming which takes place with the forefoot of the ship always completely immersed, and it is this kind of slamming which causes most of the slamming damage.—*J. L. Kent, European Shipbuilding, 1956; Vol. 5, No. 3, pp. 58-65.*

Design Considerations for Naval Gas Turbines

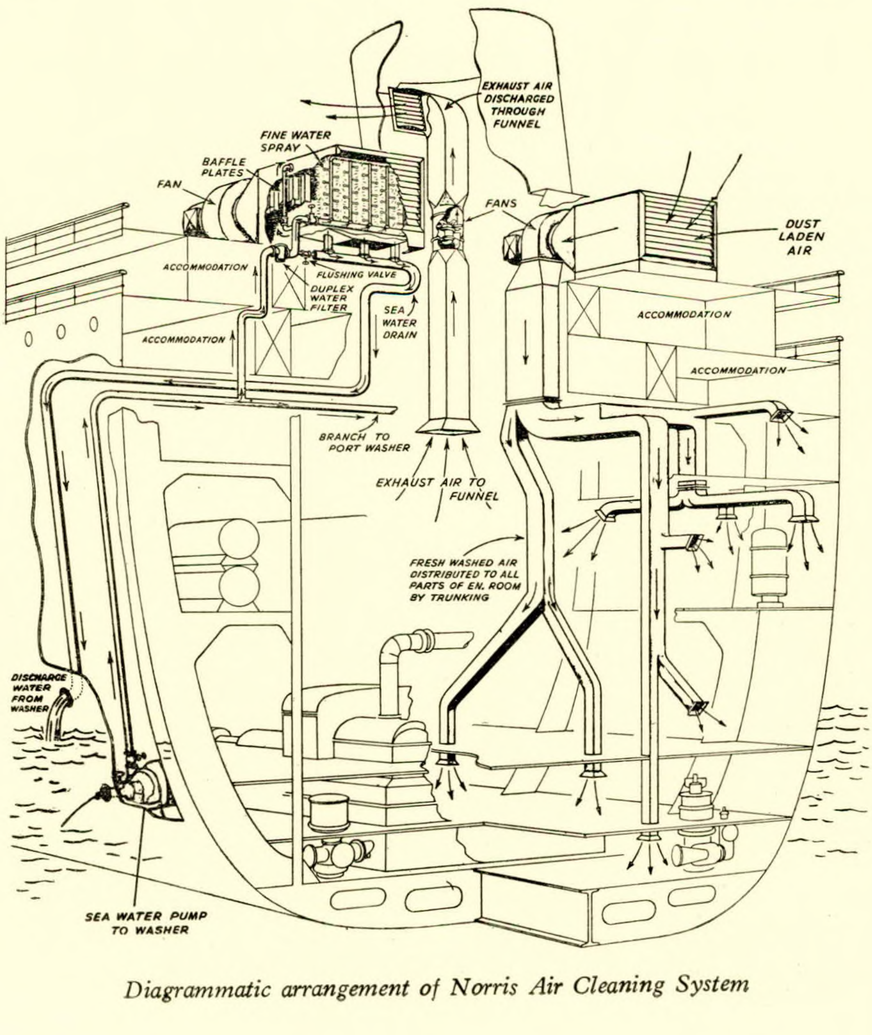
This paper discusses design consideration for a naval gas turbine in terms of duty-profile operating requirements and the basic parameters of engine design. In considering duty-profile requirements for naval combatant vessels, the main advantage of the gas turbine as in the case of emergency and booster engines is its low specific weight. It remains to be determined how much of this weight saving can be used for increased fuel tankage or other equipments. The maintenance advantages of a combatant naval gas turbine are rather difficult to evaluate at this time. Reliability and maintenance are related; however, it is possible to design for good maintenance with long periods between overhauls and yet not have good reliability. Reliability might be defined as inversely proportional to the number of forced outages. Maintenance might be defined as proportional to forced outages costs plus overhaul costs. Good reliability is dependent on having minimum number of engine accessories and components in series; therefore simplicity of design is desirable. Items like intercoolers and other accessories have to be weighed in terms of complexity added to the engine versus performance gains. Reliability within the design overhaul life of the engine is the important consideration. The overall reliability of an engine is only as good as the product of reliability factors for engine components and accessories in series. Components and accessories should be kept to the minimum necessary for push-button operation with good part load fuel consumption. The reliability of components for a given overhaul life is something that cannot be compromised. Neither can the operating characteristic requirements be compromised below a certain minimum. For those applications where part load performance is important, it is necessary to have good part load component efficiencies. To achieve producibility and good mechanical reliability, aerodynamic and thermodynamic compromises are necessary. Both compressor and turbine component efficiencies are important; however, the turbine component efficiency is relatively more important than compressor efficiency and therefore leaves very little room for compromise. Good maintenance is dependent on the concept of excellent reliability within the operating overhaul period, as well as designing for rapid disassembly and replacement of complete component sections (such as turbine rotors). Down time can be kept to a minimum, resulting in increased availability of the ship and decreased maintenance personnel requirements. By replacing complete sections and shipping small sections to a repair base for overhaul, the level of training of ship's service personnel will be a bare minimum. Small sections could be stored on board and periodically deposited at a repair base. Auxiliary engines designed for 1,000 hours between overhauls would require more overhauls than other engines; however, the ease of overhaul, small size of parts, and the small number of parts subject to failure should make up for this. It is believed possible to obtain better results than 1,000 hours at full power between overhauls at some time in the future. The duty profile or required operating characteristics determine the thermodynamic *régimes* for which a power plant is designed. The compromises in compressor efficiency to obtain pressure ratio or the compromises in turbine efficiency to obtain higher turbine inlet temperatures are dependent on the duty profile and thermodynamic considerations. On paper, gas turbine engines look as if they can be designed for maximum simplicity and reliability. The proof of the paper studies can only be determined by obtaining fleet operating experience.—*G. L. Graves, Jr., 1956 A.S.M.E. Gas Turbine Power Conference; Paper No. 56-GTP-1.*

Dry Dock at Naples

A new dry dock in the port of Naples was completed earlier this year. This dock is capable of accommodating the largest ships afloat (including the *Queen Elizabeth* and the *Universe Leader*). It has been financed by the Italian Government, and its management has been entrusted to Societa Esercizio Bacini Napolitani. The dock is among the largest in Europe. It has a maximum length of 1,150 feet, working breadth of 148 feet, and depth over the sill of 44·3 feet. There is an entrance at either end and it is equipped with a total of three caissons, so that it can when required be divided into two docks which can be used independently and simultaneously. The pumping equipment for emptying the dock comprises three principal pumps of vertical centrifugal type, each with a capacity of 700,000 cu. ft. per hr. and driven by electric motors of 1,100 h.p. taking electricity at 2,000 volts; three draining pumps of similar type with a capacity of about 81,000 cu. ft. per hr.; and three smaller pumps of 1,750 cu. ft. per hr. capacity. Electrical supply for the pump motors and for lighting is provided from two transformer houses, where the 9,000-volts supply voltage is reduced to 2,000, 380, 260 and 220 volts for the various services. Six capstans of 5 or 10 tons capacity, an electric crane of 12·5 to 25 tons capacity, a fresh water distribution system and a central compressed air plant complete the equipment of the dock.—*The Shipping World, 26th September 1956; Vol. 135, p. 274.*

Air Cleaning System for Ore Carriers

The ever-increasing demand for aluminium, iron and steel has necessitated the transport over great distances, by sea, of bauxite and iron ore. While some cleaning can be dealt with prior to the loading of the vessel, unfortunately these materials, by their very nature, cause clouds of dust to envelop the ship during the loading and unloading periods and, in consequence, to find their way into the accommodation and engine room. This can cause considerable damage to machinery, and it also necessitates frequent painting and cleaning to keep the vessel habitable. With the loading and unloading of bauxite, this position is aggravated even more than with iron ore; and, unless steps are taken to prevent this situation, then the ship's personnel are forced to accept very uncongenial conditions, and the owners, high maintenance costs. A well-known firm of shipowners in the United States of America were very much aware of this problem, and when they considered building specially designed bauxite-carriers, they felt it most desirable that some system of air-cleaning should be provided. In conjunction with this American company, the Norris Warming Co., Ltd., of Newcastle-on-Tyne, developed air cleaning devices to protect the accommodation and engine rooms of their new building programme. The vessels fitted with this equipment have now been in service for a number of years, and the results have fully justified the additional expenditure involved. The principle of the system is illustrated by the diagram reproduced herewith, and it will be noted that it consists of maintaining the engine room and accommodation under a slight air pressure, to prevent any leakage of dust. All air introduced into the spaces for ventilation purposes, and to maintain this slight air pressure, is passed through a Norris specially designed marine air washer. The air washer consists of a stainless steel chamber, which houses, at the inlet end, a number of atomizing nozzles. At the other end, a series of baffle plates are provided, and are arranged in two separate sections; the first section is wetted and flushed down, and the second section is dry. The dust laden air entering the air washers passes through a concentrated mist of moisture, and, at this point, a percentage of the dust is removed. The remainder then comes into contact with the first set of baffle plates, where the balance of dust particles are arrested and are washed down into the collecting tank at the bottom. The final set of baffle plates arrest any free moisture entrained in



Diagrammatic arrangement of Norris Air Cleaning System the air, and prevent it from passing over into the trunking system. As comparatively large volumes of water are required to feed the air washer, it is obvious that the only supply readily available is sea water. Most vessels have sea water pumps with a suitable margin to handle these volumes of water at the correct pressure, and a direct pipe connexion is made from these to the washer. Prior to the water entering the washer, it is advisable to fit suitable strainers to prevent foreign matter clogging the atomizing nozzles. All water pumped into the washer, plus the collected dust, is then led overboard by an open drain.—*The Shipbuilder and Marine Engine-Builder, October 1956; Vol. 63, pp. 612-613.*

The Alignment of Main Propulsion Shaft Bearings

This paper deals with the alignment of main propulsion shaft bearings in ships. The topic is defined as positioning bearings to support the static weight of the shaft system, prevent undesirable vibration or buckling of the shaft column, and to maintain acceptable working stresses in the shaft. The paper does not deal with shaft stresses. Factors considered to influence shafting alignment are movements of the hull, bearing movements relative to the hull, distribution of the mass of the system and designed longitudinal positioning of the bearings. To illustrate movements of the hull, data are presented showing variation in curvature of hull girders attributable to changes in temperature and ship's speed, and between a docked and an afloat condition. These variations appear as a system of faired curves. Total movements are in the order of 2 or 3 inches. Bearing movements relative to the hull are attributed, among other reasons, to bearing wear, thermal expansion of reduction gear casings, and inaccuracies in the positioning of bearings. Data are presented illustrating measured thermal expansion of a reduction gear casing averaging 0·026 inch. Analysis of shafting alignment problems confronting a repair activity and the procedures used in their resolution are discussed. Included is a technique of employing the continuous beam calculation of the shaft system as the fundamental comparator for evaluating bearing placements and appreciating the sensitivity of the system to variations in vertical alignment of the bearings. Conclusions predicted by the use of the continuous beam calculations were subsequently substantiated by physical measurements aboard ship. Detailed shipboard alignment measuring techniques are described, including a method of evaluating alignment by the direct measurement of bearing loading with a hydraulic jack. Four shaft alignment investigations are described to illustrate the influence of longitudinal positioning of bearings and other factors on the alignment problem, and to highlight the value of continuous beam calculations of the shaft system to evaluate bearing loading and the sensitivity of the system to variation in vertical alignment. During post repair trials ship A experienced a working of the shaft and stuffing box which pumped water into the shaft alley as the ship approached full power. Continuous beam calculations of the shaft system revealed that with all bearings positioned on a straight line the forward stern tube bearing would be completely unloaded. Shipboard bearing load measurements showed that the forward stern tube bearing was unloaded. The operational difficulty had been precipitated by the effective raising of the after stern tube bearing, when this bearing (only) had been renewed. Corrective action was to redock the ship, and renew (raise) the forward stern tube bearing, although bearing wear had not been excessive. In cases B and C, shipboard observations revealed that the forward bull gear journals lifted from the bearings as the gears were brought to operational temperatures. Ship B had experienced reduction gear tooth failures, wiped bearings, and a rumble at low shaft speed. Calculations of both shaft systems indicated that thermal expansion of the reduction gear casings below the measured expansion of 0·026 inch, or small wear limits of the forward stern tube bearings resulted in bearings becoming unloaded. The calculations also led to what is considered as the most desirable corrective action, which in one case was to eliminate a spring bearing between the reduction gear and the forward stern tube bearing, and in both cases to replace the forward stern tube water lubricated bearing with a babbitted spring bearing. Ship D illustrates a case where bearings had to be added to remedy the existence of a shaft lateral vibration critical within the range of propeller blade excitations.—*Paper by R. E. Kosiba, J. J. Francis, and R. A. Woollacott, abstracted in S.N.A.M.E. Bulletin, July 1956; Vol. 11, pp. 48-9.*

Some Aspects of Proposed Control Surfaces

There is special emphasis today on increased speeds and improved seakindliness for ships. Atomic engines have greatly extended the amount of power that can be installed in ships. At these expected higher speeds, the behaviour of the ship may well be the governing factor in determining the maximum speed attainable. Thus, studies of improvements in control surfaces are appropriate. Use of control surfaces for stabilizing roll has been found to be feasible and there are indications that they may have useful applications for the reduction of pitching motion. In all cases, it is desirable to improve manœuvring characteristics. From a manœuvring point of view, the control surfaces may be rudders on ships and submarines or stern and bow planes on submarines. Each has a specific function and each requires somewhat different considerations. Common to all, however, are such criteria as high lift with minimum area, low torque, sufficient strength and simplicity of design. This paper proposes certain types of control surfaces that may be useful for future high-speed ships. In particular, emphasis is placed on a new type of control surface that has merit for present use. This control surface, called "flapped all-movable", appears to have its origin many years ago in the aircraft field, but has not been adopted until recently for ship use. This flapped all-movable surface is compared to the simply-all-movable surface, which, to date, is the most efficient type of rudder or diving plane. Also, external nose and trailing edge flaps, as well as doubly all-movable flaps are briefly discussed. In addition to these proposed control surfaces, this paper includes a discussion of some of the principles guiding the design of a control surface.—*Paper by B. L. Silverstein and L. F. Whicker, abstracted in S.N.A.M.E. Bulletin, July 1956; Vol. 11, p. 52.*

Hydrodynamic Towing Basin

The "state of the art" of hydrodynamics leaves much to be desired. The reason for the deficiency is the lack of adequate test facilities which would supply the much needed information demanded by modern design trends. This applies to both subsurface and surface vehicles. Recognizing this need, Convair is initiating construction of a facility which will offer here in San Diego the world's most modern hydrodynamics laboratory. The laboratory will be capable of: (1) Complete hydrostatic investigations. (2) Hydrodynamic stability tests. (3) Resistance tests in the cavitation as well as subcavitation speed ranges. (4) Force tests, both steady state and impact in smooth and variable sea states. Designed principally for the high-speed hydrodynamics problems associated with water based aircraft, it is equally capable of solving the many problems connected with the maritime industry.—*Paper by F. L. Thornburg, abstracted in S.N.A.M.E Bulletin, July 1956; Vol. 11, p. 57.*

Fin Stabilizers for Ships

The Bureau of Ships has been conducting studies and experiments for a number of years to find an improved method of stabilizing ships. Recently, the demand for a steadying device has increased, and as a result of continuous research, the U.S. Navy has completed an activated hydrofoil fin type stabilizer installation. This stabilizer was successfully shop-tested in November 1955. The fin type stabilizer will greatly reduce the amount of ship maintenance that is necessary now as a result of damage inflicted by the pitch and roll motion of the vessel. The hydrofoil fin type ship stabilizer is essentially a powerful steering gear, with horizontal rudders which are automatically and continuously controlled by modern sensing elements and electro-hydraulic servo-mechanisms, much in the same way that automatic steering controls the rudders to keep a ship on course. A stabilizer of this type has been installed in the U.S.S. *Gyatt* (DDG-712). It is the first fin stabilizer to be built in America, the first for the U.S. Navy, and first installed on a large vessel in the U.S.A. The second fin stabilizer, scheduled for delivery in September 1956, will be installed in the *Garden State Mariner* (EAG-153) under "Project Jupiter". The following parameters were set up for the design of the fin stabilizer for the H.S.S. *Gyatt:* (1) The stabilizer shall be of the retractable, activated hydrofoil fin type, consisting of two fins, one on each side of the ship, located at about the turn of the bilge. (2) The stabilizer shall be designed for installation in a ship having the following characteristics: (a) Displacement, 3,409 tons; (b) Metacentric height, 3·95 feet; (c) Period of roll, 9 seconds; (d) Centre of gravity above base, 14·9 feet. (3) The fins shall have sufficient capacity to incline the ship in a smooth sea as follows:

Ship speed in knots	*Wave slope or angle, deg.*	*Maximum restoring moment, ft. tons (2,240lb.)*
20	5½	1,290
25	7	1,640
30	7	1,640
35	7	1,640

(4) The average residual roll over a minimum period of fifteen minutes, with the stabilizer operating and with the ship speed and wave slope as specified in paragraph 3, shall not exceed 3 degrees out to out. (Paragraphs 3 and 4 actually mean that the stabilizer shall have enough capacity to reduce the rolling motions of the ship when the deck edge is dipping in the water (26 degrees or more from the vertical) to an average of 3 degrees out to out, or 1½ degrees either side of the vertical.) (5) The stabilizer shall be designed for minimum weight and loss of buoyancy which shall not exceed the following: (a) Weight, 68 tons (2,000lb.); (b) Loss of buoyancy, 10·6 tons (2,000lb.). Total, 78·6 tons (2,000lb.). The stabilizer system can be divided into three main divisions: (1) Control system which senses the rolling motion of the ship. (2) Fin positioning or tilting machinery. (3) Hydrofoil fins which apply the righting moment to the vessel. Each hydrofoil fin designed for the U.S.S. *Gyatt* has an area of 45 sq. ft. including the full-span tail flap. The tail flaps are needed to increase the hydrodynamic efficiency of the fins. The fins are 4·5-ft. wide and 10-ft. long. The fins are tilted in opposite directions as ordered by the control systems. As the ship tends to roll to port (Fig. 5), the port fin is elevated and the starboard fin is lowered so that the water exerts a force on the fins in the direction indicated by the small arrows marked "F". These forces exert a righting movement on the ship equal to, but opposite to, the disturbing movements produced by the wave

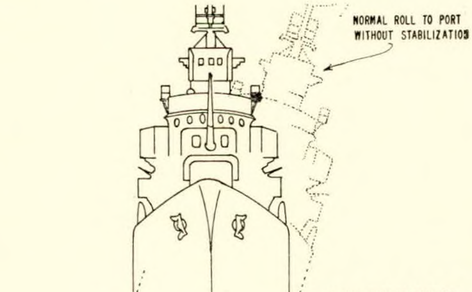


FIG. 5—*The force "F", due to the forward motion of the ship in the water, acts as shown to produce a righting moment when the ship rolls to port*

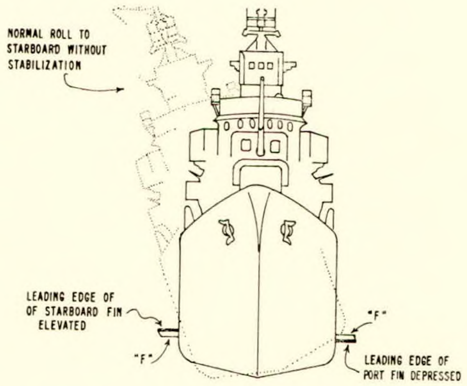


FIG. 6—*The force "F", due to the ship's forward motion in the water, acts as shown to produce righting moment when the ship rolls to starboard*

action. The ship in Fig. 6 tends to roll to the starboard and the fins have changed position. The port fin is lowered and the starboard is elevated so that the righting movement is in the opposite direction to that in Fig. 5. The righting movement is produced by a combination of the ship's speed through the water, the fin angle of attack, and the action of the water on the fin. The fins can be retracted into the hull for docking purposes or when they are not being used. The positioning machinery designed for the *Gyatt* consists of the power plant, rams, cylinders, rapson slides, and linkage. This machinery can move the fin from hardover to hardover (46 degrees) in less than 1 second. The stabilizing control system on the *Gyatt* automatically and continuously controls the positioning of the fins by anticipating the rolling motions of the ship, and in turn orders the positioning machinery to tilt the fins to compensate for the rolling. The gyro (sensing) unit measures the roll angle and the roll velocity caused by the disturbing movements applied to the ship by the wave action. The combination of these two signals is transmitted to the hydraulic relay unit (hydraulic amplifier) which controls the fin positioning servo-system. The control system that is to be installed in the *Garden State Mariner* is similar to that on the *Gyatt* except that it is a voltage control system utilizing magnetic amplifiers and resolvers. The system used in the *Gyatt* is a torque control system with synchros and a hydraulic amplifier. —*M. D. Martin, Bureau of Ships Journal, September 1956; Vol. 5, pp. 15-20.*

Marine Shafting Alignment

The paper gives a brief review of current alignment procedures and apparatus as used by shipyards in England, France, U.S. Naval Shipyards, and Gulf Coast private shipyards. These procedures are composed of various combinations of four methods which are: coupling method, piano wire, beam of light, and optical telescope. Section III of the paper gives an evaluation of the above procedures and presents recent developments in apparatus for accomplishing specific measurements. Section IV suggests a reconsideration in procedures, to make use of alignment telescope. Appendix II offers an all-purpose method which includes an alignment telescope fitted with illuminated auto-reflection and auto-collimation, micrometer vertical and horizontal adjustment of sighting line, and a removable ball joint for pointing of telescope. Three lightweight self-centring, scroll-type lathe chucks are bored for mounting telescope, lighted displacement target, and lighted reflection (mirror) target. All chucks are provided with jaw extensions of length to suit bearing housings and couplings as required. The two chucks used for targets are adapted for back drive to permit insertion in stern tubes and long bearings. Illustrations of the telescope, targets, and chucks are given and are more completely described in the paper. Preparation for checking alignment consists of removal of shafting, fitting suitable jaw extensions to each chuck, and mounting telescope and targets in the chucks. The mirror target chuck is mounted on coupling boss, and the telescope and chuck are secured in aftermost outboard bearing housing. With telescope micrometers adjusted to zero, the telescope is pointed to place cross hairs on lighted mirror target. (*a*) *Checking Coupling Axis of Rotation.* With target dark and telescope lighted, adjust target so that reflection of telescope light into telescope remains stationary when coupling is rotated. The position of the reflected beam of light is on the coupling axis of rotation and the coupling opening is determined from the reflected light position, the distance to coupling, and the coupling diameter. (*b*) *Checking of Outboard Bearing Positions.* Insert displacement target and chuck into bearing housing nearest telescope, darken telescope, and light the target. Adjust the micrometers on telescope so that cross hairs centre on target centre, and record micrometer readings. Move target and chuck to consecutive positions including inner ends of stern tube bearings and record readings. (*c*) *Checking of Inboard Bearing Positions.* Provide cylindrical target mount of outside diameter the same as that of journal and use the same lighted displacement target as fitted to chuck, taking micrometer readings in the same manner as for outboard bearings. In order to check inboard alignment of shafting in place, a suitable bracket is required for mounting telescope to rotate with coupling, and a right-angled eyepiece might be needed. A legible scale is mounted in line of sight to rotate with shaft. Scale readings can be recorded for various positions of rotation and various distances. If the shaft is not in a tunnel, pipe nipples and plugs can be provided in bulkheads. The author believes that the advantages to be found in the recommended procedure will justify its use in all except the smallest repair and building yards.—*Paper by D. M. Beard, abstracted in S.N.A.M.E. Bulletin, July 1956; Vol. 11, pp. 46-47.*

Investigation of Cavitation Erosion in Diesel Engine Coolant Systems

One of the faults repeatedly observed during rigorous testing of Diesel engines is the cavitation erosion of surfaces exposed to the jacket coolant. This type of damage generally occurs on the outer surface of the cylinder liners, or in the cylinder block bores at the lower seal area. It is usually greatest on the thrust side of the cylinder, but may also be serious on the diametrically opposed side, or even around the full circumference. Cavitation erosion is most evident in engines with wet type cylinder liners, and especially in lightweight, high-speed engines. It is not confined to engines of any particular design, and is often closely related to corrosion effects. The first part of the present paper deals with cavitation tests on various materials, carried out with a magnetostriction apparatus. The cavitation effects produced by this apparatus are greatly accelerated, so that factors such as galvanic action and minor variations in the immersing fluid have no significant effect on the amount of damage. The results show that the loss of weight in a given time increases rapidly with increasing amplitude of vibration. The cavitation resistance of aluminium alloys tends to increase with hardness. Coatings of neoprene or nylon have very good resistance. Damage due to cavitation erosion can be produced on a stationary surface separated from the vibrating surface by a liquid. The amount of damage is related to the distance between the surfaces. In the second part of the investigation, liner and block vibrations were measured in a type of engine known to be subject to cavitation erosion. In addition, the resonant frequencies of the cylinder liner were determined under a variety of conditions. The results suggest that the frequencies observed during engine tests correspond to the major resonant frequencies of the cylinder liners or other components. These frequencies are excited by certain impacts, notably the firing pulses and piston slap. The experiments are being continued in order to determine more precisely the variation of cavitation damage with vibration characteristics.—*A. R. Schrader, S.A.E. Paper No. 759, 1956. Journal, The British Shipbuilding Research Association, August 1956; Vol. 11, Abstract No. 11,949.*

Hydrodynamic-type Gas Bearings

This paper deals with the properties and applications of gas-lubricated bearings of the hydrodynamic, or self-acting, type as distinct from the pressure-fed, or hydrostatic, type. Although the idea of using gas as a bearing lubricant is by no means new, it is only within recent years that requirements have arisen for the development of machines employing such

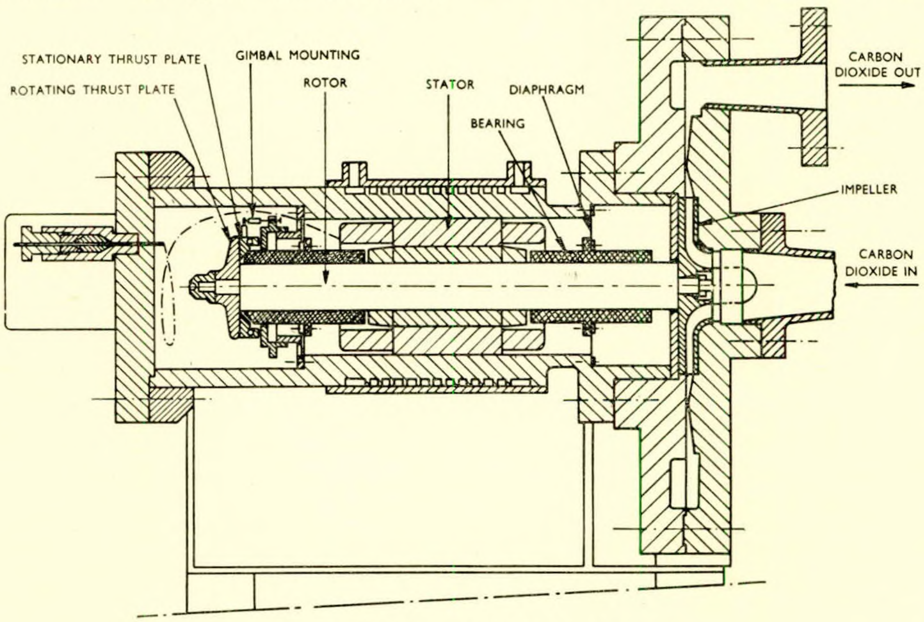


FIG. 1—*Carbon dioxide circulator*

Carbon dioxide pressure, 100lb. per sq. in. gauge; carbon dioxide temperature, 150 deg. C. (300 deg.. F.) for flow through impeller—static gas in motor barrel is much cooler; speed range 2,000-13,000 r.p.m.

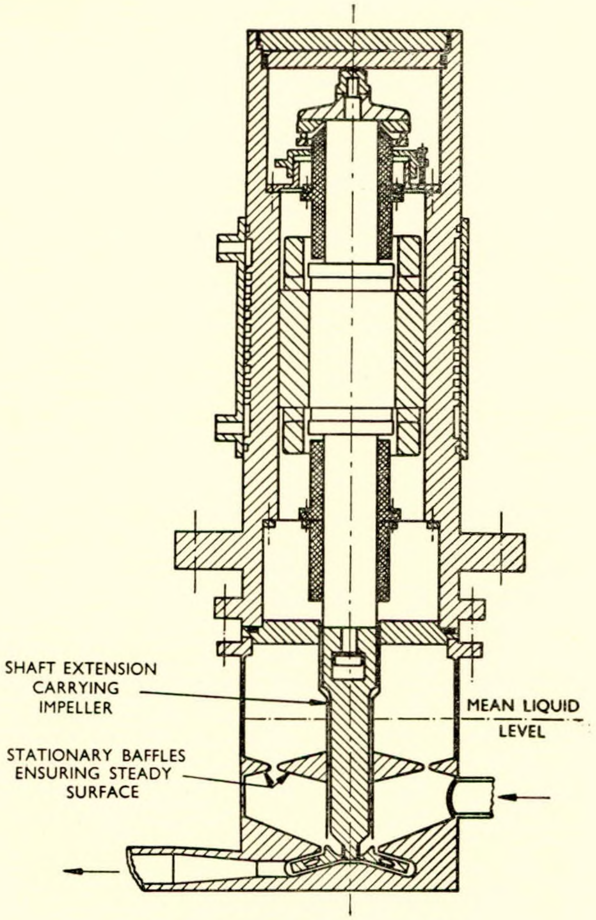
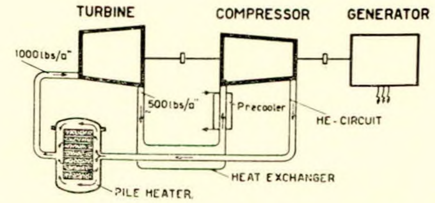


FIG. 2—*Molten-bismuth pump*

bearings. The two machines described in some detail (Figs. 1 and 2) are applications to atomic energy problems for which it would have been much more difficult to provide solutions using conventional liquid or grease lubricated bearings. The machine shown in Fig. 1 is used to circulate carbon dioxide coolant gas through a test circuit in a nuclear reactor. The gas is contained at a mean pressure of 100lb. per sq. in. gauge, passes through the impeller at 150 deg. C. (300 deg. F.), and is required to be free from contamination by oil. The gas journal bearings are 5 inches in length and 2 inches in diameter, and have a mean diametral clearance of 0·0012 inch. The shaft is of chromium-plated nickel alloy and the bushes of cast nickel alloy, in a combination chosen to give freedom from corrosion, good anti-galling properties, similar thermal expansion of shaft and bush, and low thermal conductivity along the shaft which limits transfer of heat from the impeller to the motor unit. The speed range for this machine is 2,000-13,000 r.p.m., being driven from a variable frequency three-phase alternator, and the rotating parts, which weigh about 20lb., are balanced dynamically and statically to within 5×10^{-2}oz. in. per oz. mass. The thrust-bearing plates are of similar materials to those used for the journals. The machine shown in Fig. 2 is used for pumping liquid bismuth at about 400 deg. C. (750 deg. F.), and is suitable for use in test circuits in which the bismuth is radioactive and must be preserved from contamination by liquid lubricants and by the surrounding air. The motor unit is normally filled with an inert gas, either helium or argon, which also lubricates the bearings. The pump is so designed and positioned in the circuits that it also acts as a surge tank, and the pump bowl contains baffles to prevent undue turbulence of the liquid surface in order to reduce gas entrainment in the liquid metal. A notable feature of this application is that nuclear radiations emitted by the liquid in the pump bowl have no effect upon the gas bearings, whereas organic lubricants deteriorate seriously in radiation fields. In very strong radiation fields it would be necessary to provide extra shielding against nuclear radiation for the motor windings, as has indeed been done in later designs, unless radiation-proof windings are developed, insulated, for example, by inorganic materials such as ceramics.—*Paper by G. W. K. Ford, D. M. Harris and D. Pantall, presented at a General Meeting of The Institution of Mechanical Engineers on 26th October 1956.*

Closed Cycle Gas Turbine Plant with Nuclear Reactor

The author considers the use of a closed helium circuit in a thermal power plant employing atomic energy and concludes that it is feasible. Since helium is neutral to the material of the pile and in addition has advantageous characteristics from the viewpoint of nuclear physics in that, like graphite, it acts as a neutron moderator, the pile could be brought in direct contact with the gas. The helium issuing from the compressor of the circuit could be supplied to the pile under a high pressure of 700-1,400 p.s.i.—for increasing the heat transmission figures and reducing the dimensions—and would afterwards flow in direct contact as a cooling medium around the uranium rods, as also through bores in the graphite itself. In



Scheme of a gas cycle reactor combined with closed cycle gas turbine in one group

this way the pile would give up the heat which it produces by direct convection to the helium. Under the above-mentioned assumptions (increased pressures) the large quantities of heat produced in the pile will, on account of the good heat transmission, be capable of being usefully taken up. In this manner no tubes would be required for the helium heater. Calculations show that if a pile temperature of 1,500 deg. F. is assumed as admissible, only a very small surface is needed to heat up the circulating helium to 1,300 deg. F. American Turbine Corporation and Escher Wyss have been developing designs particularly adapted to United States requirements. The simplicity and the high efficiency of the closed-cycle power plant system makes this plant especially desirable for nuclear application. Accordingly, a new 10-15 MW set has been designed which meets the requirements for a wide range of possible applications. Axial machinery was chosen for plants above 10 MW because of its compactness and high efficiency. Pure nitrogen is used as a working fluid when employed in a nuclear plant.—*C. Keller, 1956 A.S.M.E. Gas Turbine Power Conference; Paper No. 56-GTP-15.*

Proposed Oceanographic Research Vessel

The problem of designing a ship specifically for oceanographic research is far from simple. Should she be large like the Russian hydrographic ship *Witjas,* purported to be of 5,500 tons displacement, or small like the 380-ton *Atlantis,* should she be a 12- or a 16-knot ship, and should it be attempted to provide for all types of acoustical work—these are only a few of the difficult questions that must be answered. The *per diem* cost of an oceanographic expedition is quite high and is one of the more important factors which put an upper limit on the size of the research ship. The ship must be large enough to carry sufficient personnel and equipment to make an expedition scientifically profitable, and yet her requirements as to crew, rations, and fuel—not to mention maintenance cost between cruises—must be modest. Seaworthiness is

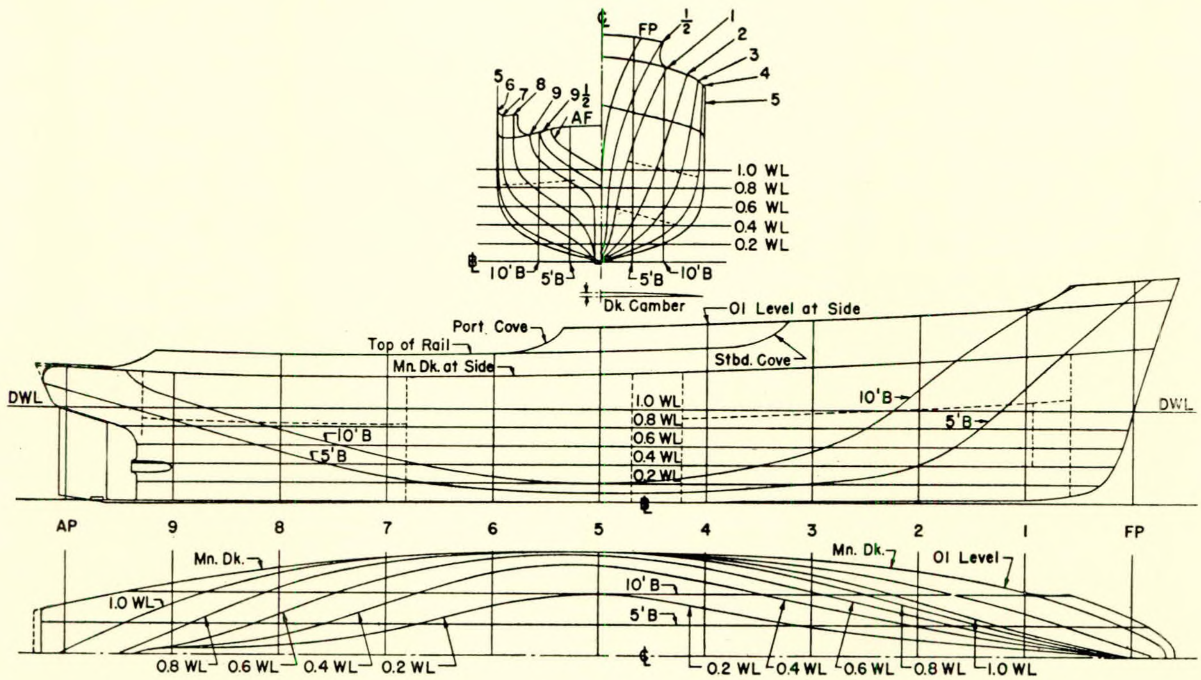


FIG. 1(a)—*Preliminary lines*

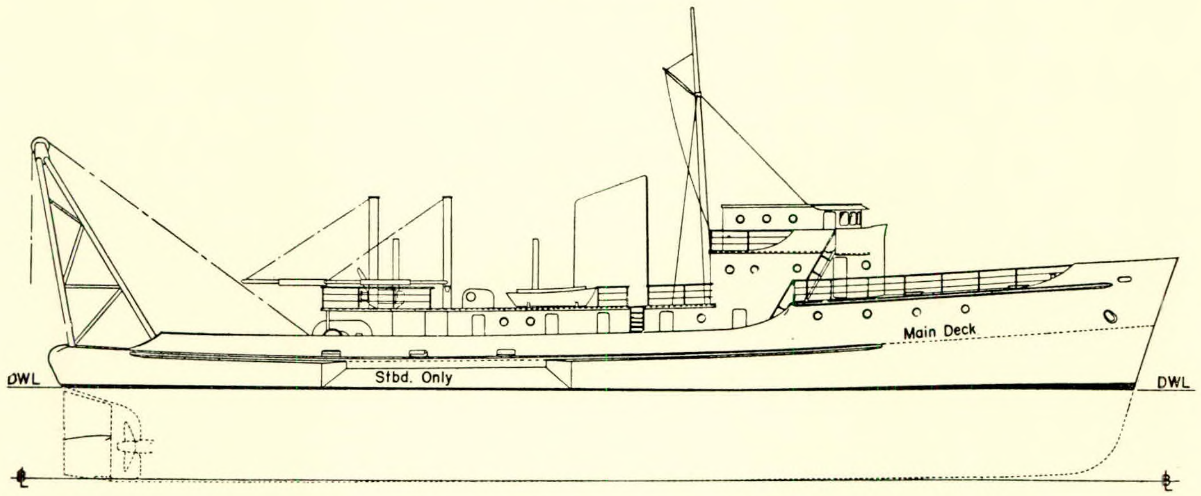


FIG. 1(b)—*Outboard profile (Rev. 2)*

FIG. 1—*Oceanographic research vessel*

of course a basic requirement of any vessel intended for long periods of blue water sailing, but more is desired of the research ship than mere ability to survive heavy weather. It is desirable to reduce the sea-excited motion of the ship as much as possible. Excessive motion not only means misery and consequent inefficiency for personnel but adds to the difficulty of handling gear and, most important of all, hampers the conduct of even the most routine scientific work. In addition, it might be mentioned that for certain types of work it would be a great advantage to be able to control the heading of the ship at speeds below steerageway and even while laying-to. Precise criteria for satisfactory performance do not exist, but there is obvious benefit in a vessel which will permit operations which have previously been prevented by a state 5 sea. A hull which has been proposed to meet the many and diverse requirements of oceanographic research was designed, the lines and outboard profile of which are shown in Fig. 1. Pertinent design particulars are listed below:

Length overall, ft.	181
Length, waterline, ft.	170
Length between perpendiculars, ft. ...	163
Draft (design waterline), ft. ...	14·75
Displacement (design waterline) (salt water) tons	1,000
Design speed (still water), knots ...	12

Within the limitations of the tests conducted, the model of the proposed oceanographic research vessel rode easily, was reasonably dry and showed motions which were on the average somewhat less than those observed on models of other types of vessels.—*David W. Taylor Model Basin, Report 1055, 1956.*

Marine Air Conditioning

There is a modern tendency to design air conditioning systems for marine use on a basis of extremely low air changes

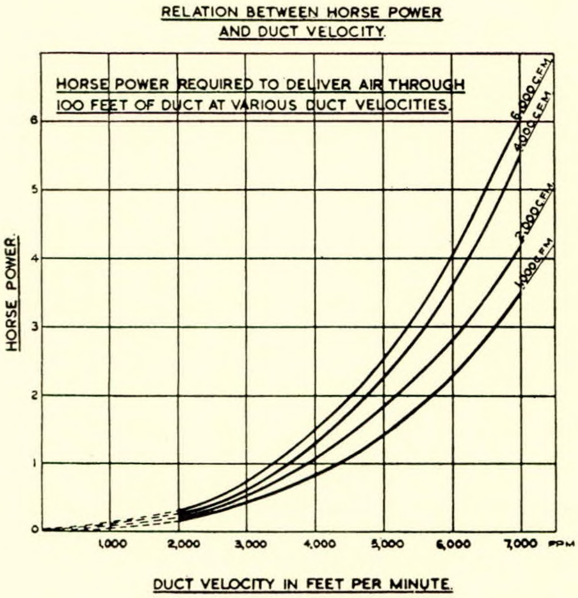


FIG. 7

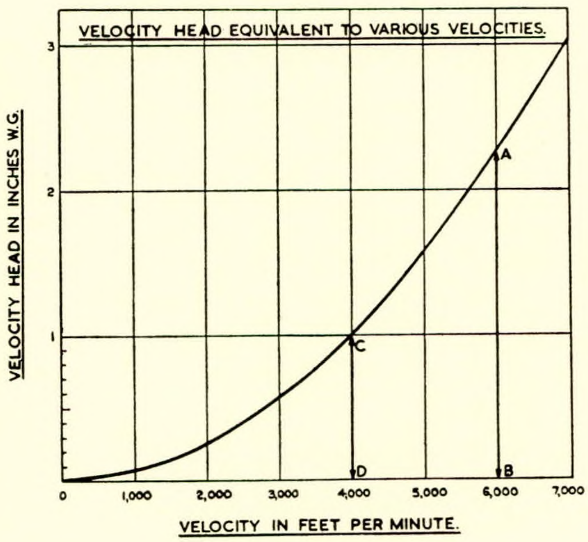


FIG. 8

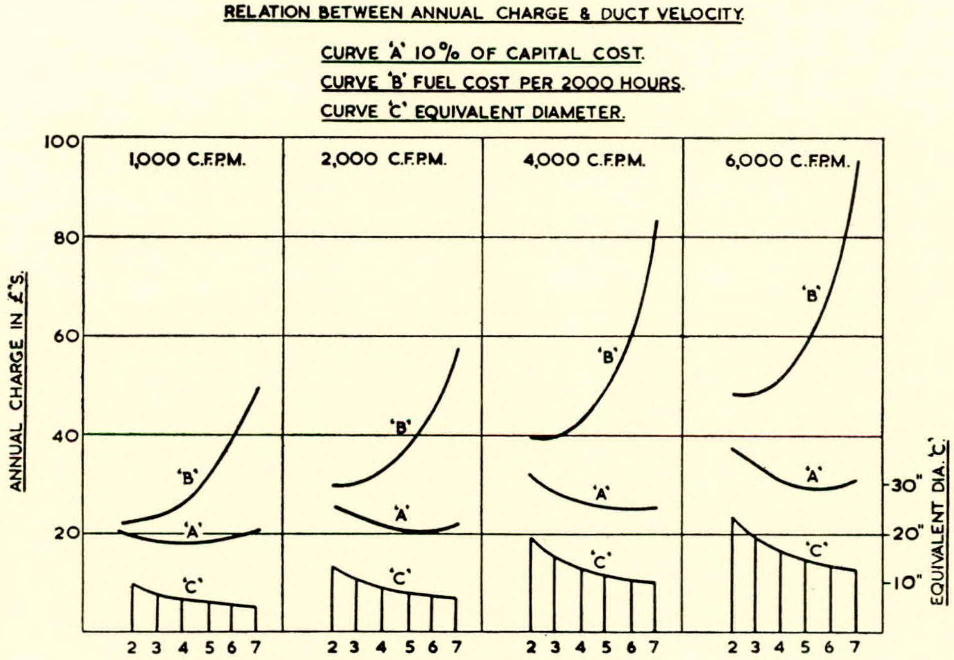


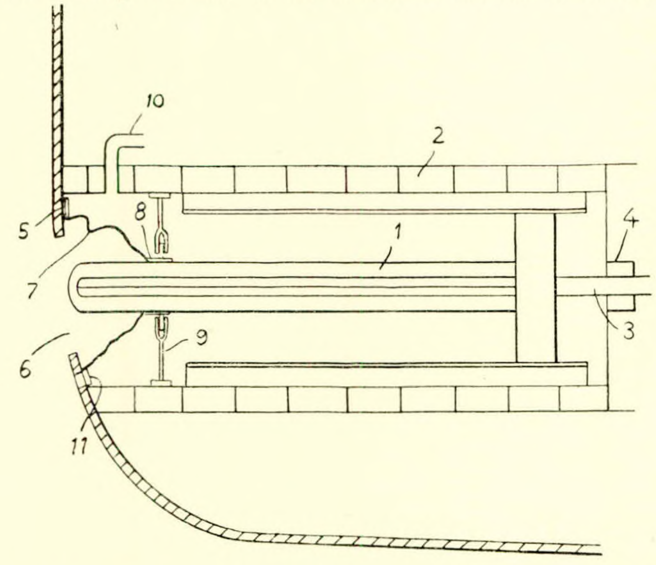
FIG. 9

varying from three to six times per hour and while even with conventional trunk velocities these systems must have ducts of relatively small cross sectional area as compared with systems using more orthodox rates of air changes, it is somewhat surprising to note that it is to this type of installation that high velocity principles have recently been applied. In other words, having reduced the duct sizes by reducing the circulation air, the system is further complicated by introducing high velocities and resulting high pressures to achieve even smaller cross sectional areas, the value of which may be more apparent than real. The effect of increased velocity on pressure loss and also on horsepower is illustrated in Figs. 7 and 8. Fig. 9 gives the result of a computation of annual charges for systems based on varying velocities. It must be stressed that high velocity in a system has no virtue other than the reduction in duct dimensions effected and this must be paid for continuously in power consumption. This does not mean that it should never be used but only where its use enables the necessary air ducts to be accommodated. There is no virtue or economy in reducing duct sizes below what can be readily accommodated merely for the purpose of maintaining a high velocity which does not facilitate installation and involves higher motor costs and higher power consumption. The approach to the use of high air velocity should therefore be discriminating, it being applied only where it is of material value and not likely to involve uneconomic power consumption.—*J. K. W. MacVicar and S. T. Fairweather, European Shipbuilding, 1956; Vol. 5, No. 3, pp. 66-71.*

Patent Specifications

Stabilizer Fin Chamber

In ships incorporating stabilizing systems which operate through the medium of fins or vanes, these are normally housed within cavities in the hull when the system is not in use and can be extended from these cavities to project from the hull when the stabilizing system is in use. Such cavities are normally constituted by a housing which projects into the interior of the hull. The loss of buoyancy due to such housings filling with water can amount to a considerable proportion of the total weight of the stabilizing equipment. According to the invention, means are provided for feeding a gas, for example, compressed air, exhaust or other vapours, to a housing, whereby the water which enters, or would enter the housing, will be expelled from or stopped from entering the housing. In this way the housing will become or remain at least partly filled with the gas whereby the buoyancy of the vessel will be increased. If the supply of gas is fed continuously to the housing, the excess gas which escapes from the outer end of the housing will rise to the surface of the sea as bubbles.



Referring to the drawing, the stabilizing fin (1) is situated in a housing (2) into which it is retracted when not in use. The fin is moved by a shaft (3) which passes through a sealing gland (4) in the inner end of the housing and is actuated by a mechanism not shown in the drawing. The fin (1) can be extended outside the hull (5) by passing it through an orifice (6) in the hull which is shaped to allow the fin to be inclined to effect the stabilization. The orifice (6) is provided with a sealing device comprising a flexible diaphragm (7) which is secured around its outer edge adjacent the edge of the orifice (6). The sealing diaphragm is provided with an opening near its centre through which the fin (1) can slide in a substantially watertight manner. For example, this opening may be formed as a sleeve (8) which conforms with and slidably fits the cross-section of the fin. The sleeve is restrained from movement in the direction of movement of the fin by support members (9) carried from the housing (2) and engaging a part of the sleeve (8). Compressed air or gas is admitted to the housing (2) through the pipe (10) and expels any water which leaks into the housing through the valve (11) situated near the bottom of the housing.—*(British Patent No. 760,792, issued to Research Interests, Ltd. Complete specification published 7th November 1956.)*

Reciprocating I.C. Engines with Exhaust Turbines

This invention relates to compound power plants of the kind comprising a reciprocating internal combustion engine, a turbine arranged to be driven by the exhaust gases from the engine, the turbine being arranged to drive a compressor for compressing the gaseous charge supplied to the engine. In such arrangements problems arise in relating the power outputs of the reciprocating engine and turbine to the power absorbed by the compressor and propellers. It is an object of the invention to provide an improved power plant of this type, including two independently rotatable propellers, which will provide a simple and effective manner for the transference of power between the engine and the turbo-compressor assembly, in the manner required to give satisfactory operating conditions over the whole operating range of the power plant. Thus a power plant according to the invention includes two independently rotatable propellers, at least one of which is of the variable pitch type, and differential transmission mechanism.

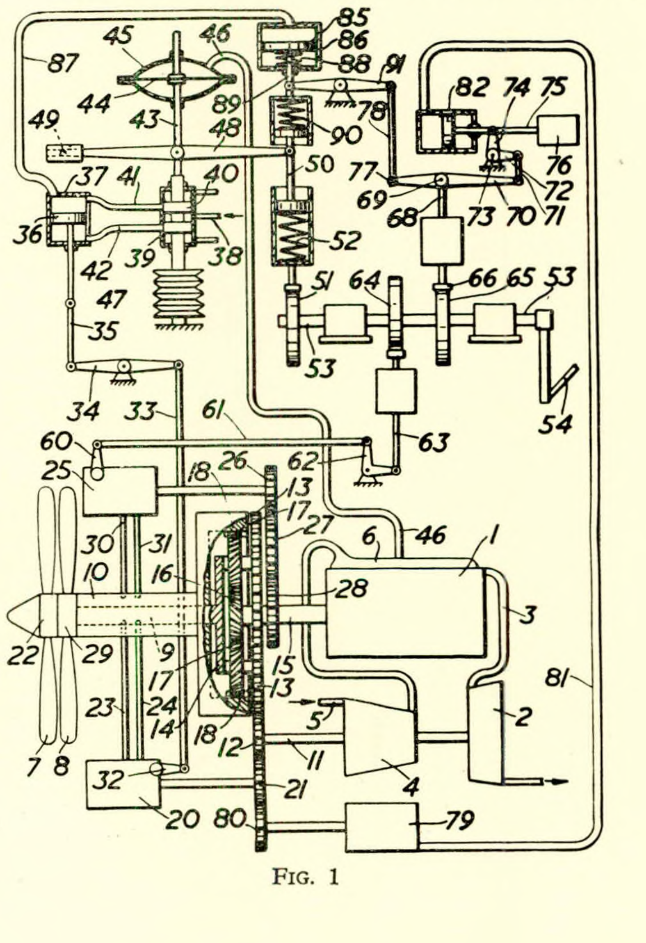


FIG. 1

One prime mover is arranged to drive one of the propellers directly while the second prime mover is connected to the input element of the differential mechanism, the output elements of which are connected respectively to the second propeller and to the assembly comprising the first prime mover and associate propeller. In Fig. 1 the power plant comprises a reciprocating compression ignition engine (1), an exhaust driven turbine (2) to which the exhaust gases from the engine are led through a duct (3), and an axial flow compressor (4) mounted co-axially with the turbine (2) and mechanically coupled to it. The compressor receives air from the atmosphere through an intake (5) and delivers compressed gaseous charge to the engine through a boost duct (6). Two contra-rotating variable pitch propellers (7) and (8) are mounted on inner and outer concentric propeller shafts (9) and (10) arranged co-axially with the crankshaft (15) of the reciprocating engine (1).—*(British Patent No. 760,005, issued to D. Napier and Son, Ltd. Complete specification published 31st October 1956.)*

Variable Pitch Propeller Control System

In recent years the widespread application of Diesel engines and gas turbines to the propulsion of ships has emphasized the advantages of controllable pitch propeller systems. Inasmuch as power plants of these types function most efficiently at constant speed and in one direction, the advantages of changing propeller pitch instead of engine speed to effect changes in ship's speed or reversal in ship's direction of travel are readily apparent. Heretofore a variety of controllable pitch propeller systems have been proposed. In many instances such systems comprise a variable pitch or adjustable blade propeller mounted on a hollow drive shaft actuated by a hydraulic piston within the drive shaft. The piston is connected through shafts and/or linkages to the propeller blades, and is positioned by an external hydraulic pump and valve system through hydraulic rotating joints. In other instances, the piston is positioned by an hydraulic pump housed within the drive shaft and driven by an electric motor which is also mounted in the drive shaft and energized through electrical slip rings from an external electrical power source. According to one aspect of this invention a variable-pitch propeller drive includes within the propeller shaft means for converting a portion of the energy of rotation of the shaft into liquid pressure energy and means within the propeller shaft arrangement to use the liquid pressure energy to vary the pitch of the propeller (12). The converting means may comprise a hydraulic pump (16) while the utilization means may include a piston (14) in a cylinder (15) supplied from the pump and a piston rod (13) coupled to vary the pitch of the propeller blades. There need thus be no need for hydraulic or electric slip rings for supplying external power to vary the propeller pitch.—*(British Patent No. 760,568, issued to The Sperry Corporation. Complete specification published 31st October 1956.)*

Foldable Mast

The foldable mast shown in Fig. 1 has an upper part (1) which is pivoted on the shaft (12) and the rate of folding is regulated by the restraining force in the wires (18) which are guided in channels and are fixed to the platform on the upper

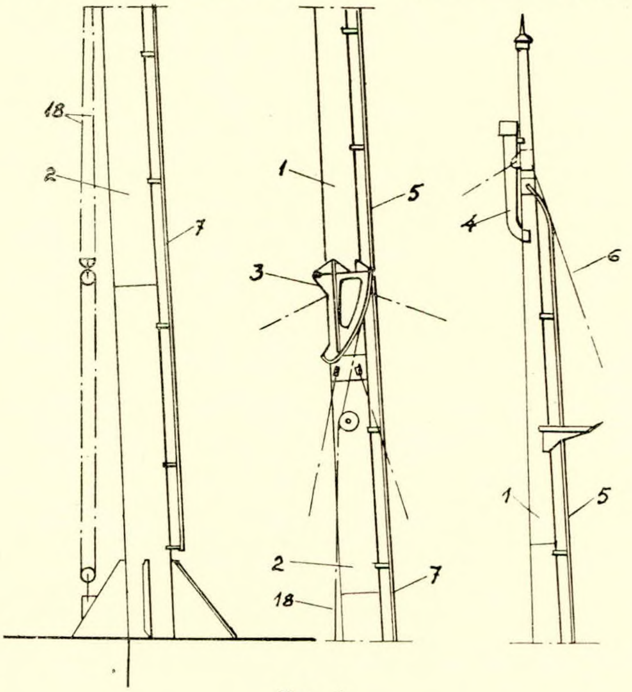
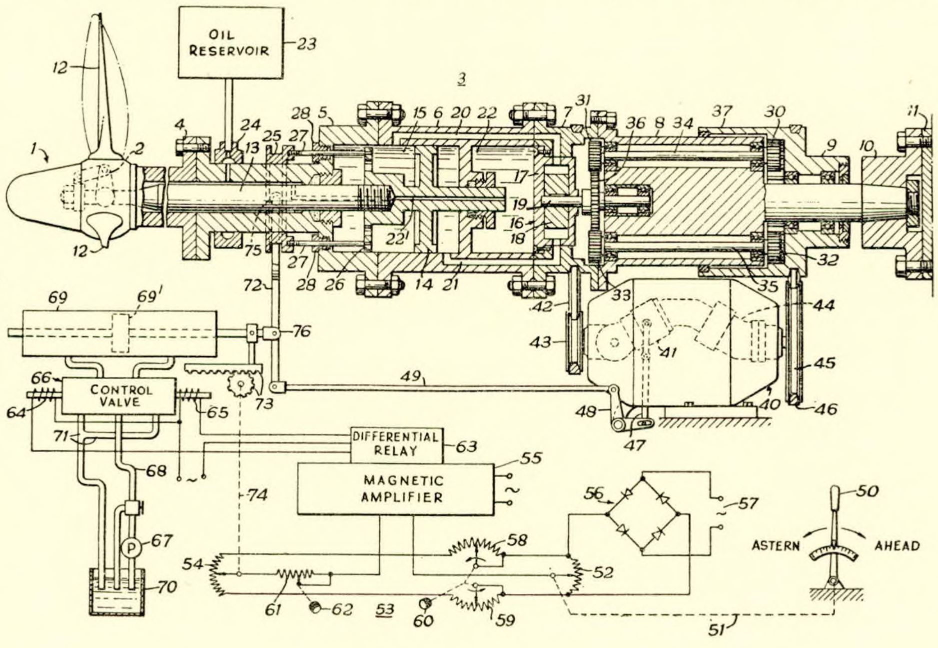


FIG. 1

mast. The quadrant-like lever (3) in its normal position has greater length in a vertical direction than in a horizontal direction. According as the upper mast (1) folds, the moment which is required to counteract the moment in the upper mast



on its pivot will increase. However, owing to the shape of the quadrant-like lever, the radial distance of the contacting point of the wire in the channel from the pivot centre will increase as the upper mast is lowered, and the device works in such a way that the two counteracting moments are substantially equal during the folding action of the mast. It will also be obvious that owing to the shape of the quadrant-like levers, the force required for raising the mast will be substantially constant during raising.—*(British Patent No. 759,483, issued to Uddevallavarvet A.B. Complete specification published 17th October 1956.)*

Jet Propulsion Device

According to this invention a power driven impeller is adapted to draw water in through a forward aperture and to discharge the water under pressure through a rear aperture, so as to cause movement of the hull, comprises an elongated rotor within a stator. In Fig. 2 a single rotary impeller (6) is positioned within the hull (7) of a ship and extends longitudinally of the hull. The impeller (6) is driven by a power unit (8) by means of a shaft (9). The intake of the impeller is connected by a pair of tubes (10) extending forwardly and each communicating with an aperture (11) positioned on each side of the bow of the ship below the water line. The discharge end of the rotary impeller communicates through a duct (12) with an aperture (13) positioned in the stern of the ship forwardly of the rudder (14). Thus water is drawn in by the impeller from the bows of the ship through the tubes (10) and is discharged under pressure through the duct (12) in the form of a rearwardly directed stream.—*(British Patent No. 759,500, issued to E. Taylor and R. Shipp. Complete specification published 17th October 1956.)*

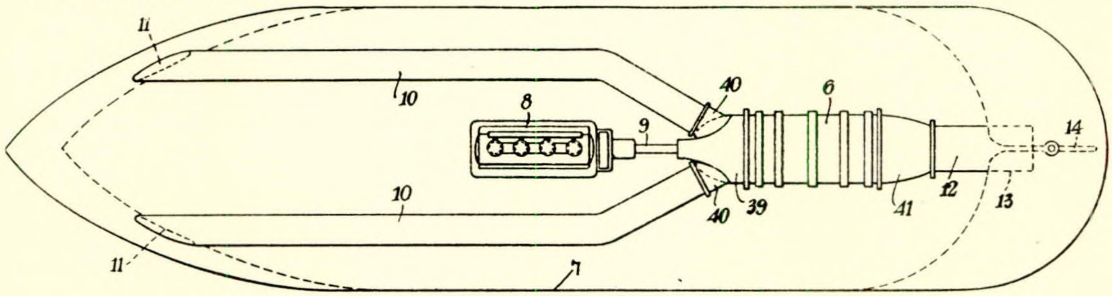


FIG. 2

Ship's Draught Gauge

When a ship is loading in an open roadstead or in rough surfaced water, the wave motion of the water prevents the accurate determination of the ship's draught by observation of the load lines painted on the hull of the ship and as a consequence the ship may be overloaded or underloaded. It is an object of this invention to provide a draught gauge which gives a stabilized indication of the draft of the ship, i.e. an indication which is substantially unaffected by the wave motion in the water surrounding the ship, or which at least is sufficiently steady to enable a mean reading of the draught to be taken with substantial accuracy. In Figs. 1 and 2 the apparatus comprises a pair of lower diaghragm chambers (1) and (2) adjacent the bottom (3) of the ship and a pair of upper diaphragm chambers (4) and (5) at a level corresponding to the loaded water line indicated at (6). The light water line is indicated at (7), and it is necessary that the lower diaphragm chambers (1) and (2) be positioned below the line (7). The diaphragms are horizontally arranged and the lower side of each of the lower chambers is connected by a pipe (8) to a sea cock (9) for admitting water from the outside of the hull to the lower side of the respective chamber. A handwheel (10) on the deck of the vessel is connected by a spindle (11) to the sea cock (9) so that the cock can be opened and closed from the deck. The upper side of each of the lower chambers (1) and (2) is connected to the lower side of the upper diaphragm chambers (4) and (5) by means of a small bore tube (12), the tube being filled with sea water. The upper side of each upper diaphragm chamber is connected by tubing (13) to an indicating device (14). This device comprises a pair of pointers movable over a common scale, each of the pointers being actuated by an air operated bellows. The tubing (13) from each of the upper chambers (4) and (5) is connected to the bellows of one of the indicating pointers so that one of the pointers will move in response to deflexion of the diaphragm of one of the upper chambers and the other pointer will move in response to deflexion of the diaphragm of the other of the upper chambers.—*(British Patent No. 759,355, issued to R. Malone and The Malone Instrument Co., Ltd. Complete specification published 17th October 1956.)*

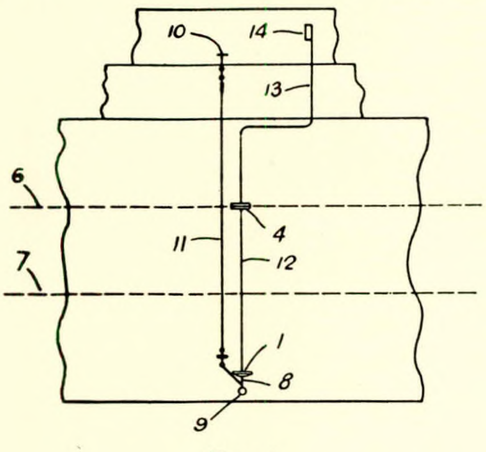


FIG. 1

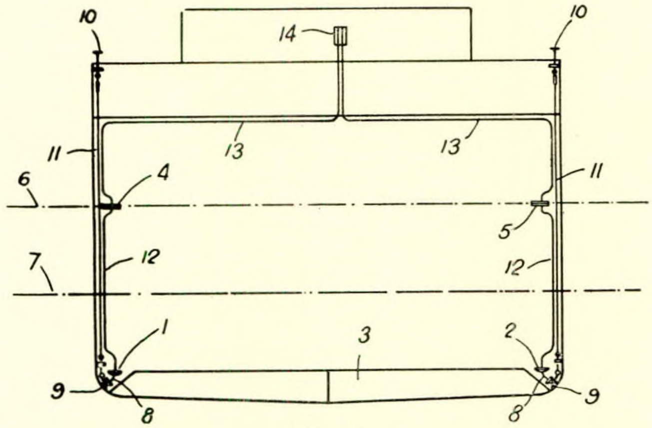


FIG. 2

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Marine Engineering and Shipbuilding Abstracts

Volume XX, No. 2, February 1957

* Patent Specifications

Cavitation Control through Diesel Engine Water Treatment

Cavitation erosion on the water side of cylinder liners in Diesel engines is attributed to piston induced vibration of the liner. It is always greatest at the side on which the piston bears during the power stroke. The extent of the attack is unaffected by the location of the water inlet and outlet, or by variation of water velocity within the range usual for Diesel engines. The use of a magnetrostriction cavitation tester, combined with close control of temperatures and pressures, has made it possible to reproduce the operating conditions encountered in an engine. Tests have been carried out on buttons machined from the cooling surface of two liners, made of the same cast iron alloy. The loss in weight after a given time was measured for temperatures between 50 deg. F. and 190 deg. F., and for pressures between atmospheric and 30lb. per sq. in. The coolant used was distilled water, to which various cavitation inhibiting substances were added. The effect of each additive was investigated over a range of concentrations. The results are given in tables and curves. It appears from the results that cavitation erosion in the water system of Diesel engines can be reduced by increasing the water pressure, whatever water treatment is used. Changes in operating temperature will have less predictable effects. Potassium chromate is an effective cavitation inhibitor if the concentration is kept between 1,000 and 2,000 parts per million. The protection against cavitation afforded by a soluble oil corrosion inhibitor is similar to that of potassium chromate, but is less sensitive to variations in concentration. The boron-nitrate type of corrosion inhibitor does not provide such protection up to a concentration of 5,000 parts per million. Gelatine inhibits cavitation at atmospheric pressure and room temperature, but seems useless under operating conditions. A chemical wetting agent (aerosol) intended to increase adhesion between the liquid and liner, seemed to lose its effectiveness after a short period. Other types of surface tension depressants may give better results and should be investigated. Treatments of this last type do not of themselves provide protection against corrosion. It is emphasized that tests that do not reproduce the temperature and pressure conditions encountered during operation may give misleading results.—*W. Margulis, J. A. McGowan and W. C. Leith, S.A.E. Paper No. 757, 1956. Journal, The British Shipbuilding Research Association, August 1956; Vol. 11, Abstract No. 11,947.*

Influence of Decreasing Boiler Pressure upon Natural Circulation

In an article in the journal "Energie", R. Loewenstein analyses the effect of decreasing boiler pressure on the pressure conditions and steam formation in downcomers and risers and hence on the water circulation. The inquiry is restricted to boilers with natural circulation and unheated downcomers. It is shown that, by selecting a favourable velocity, steam formation at the downcomer outlet may be avoided. A formula is derived for the quantity of steam formed per second in the downcomer due to the decreasing boiler pressure. This shows that the steam formation for a given rate of pressure fall increases sharply at pressures below 550lb. per sq. in. A particular boiler is chosen as an example and a detailed quantitative analysis made. Curves are given which show the pressure due to the water column in a downcomer as a function of the water velocity at the downcomer outlet, both for normal operating conditions and for a boiler pressure decreasing at the rates of 0·2 and 0·6 atmospheres per second (3 and 9lb. per sq. in. per sec.). It is found that, when the boiler pressure is decreasing, there is a certain critical velocity (between 1 and 1·5ft. per sec.) at which steam accumulates in the tube and the circulation is completely interrupted. In considering the effects that decreasing boiler pressure will produce in the risers, the steam bubbles arriving from the downcomer and the lower drum must be taken into account. It is desirable that those risers in which the suction effect is strongest should be connected to the highest point of the lower drum. It is not advisable to use a few downcomers of large cross-section, since if the boiler pressure is decreasing large quantities of steam will be conducted to a few favoured risers. This will cause a great increase in the upward force in these risers, which will deprive the others of water. If the ratio of the total downcomer cross-section to the total riser cross-section is too large, there is a risk that the water velocity in the downcomer will fall to the critical value. Since, however, in marine boilers this ratio is usually less than 0·5, a sudden reduction in pressure under full load conditions is not likely to stop the circulation. If this happens at all, it will probably be under partial loading or when the boiler is inclined through ship motions, heel, etc.—*Journal, The British Shipbuilding Research Association, August 1956; Vol. 11, Abstract No. 11,896.*

New Cargo Vessel with Turbocharged Engine

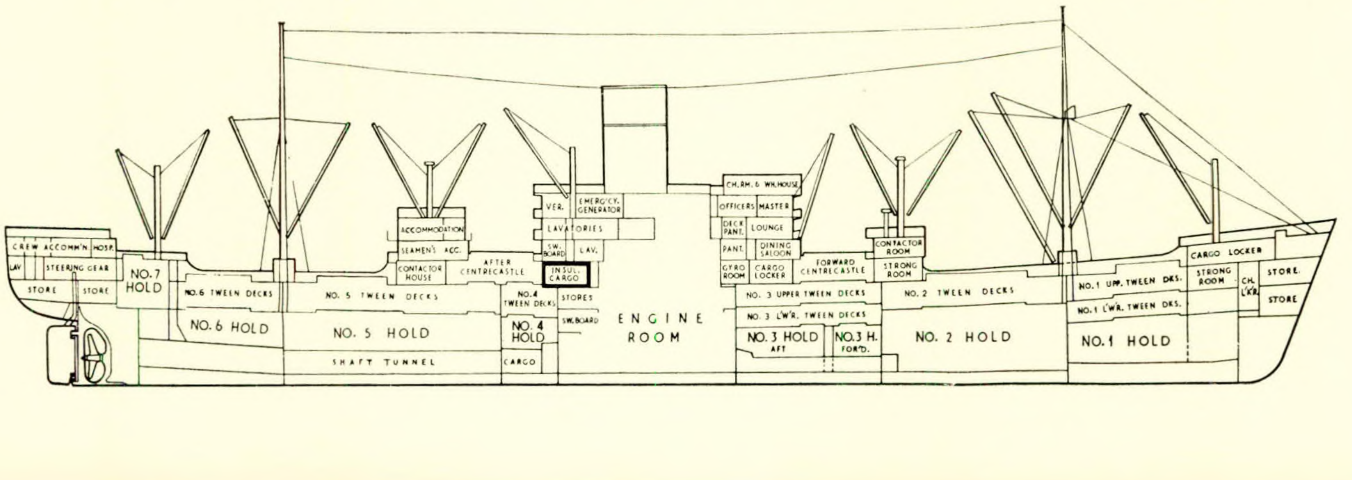
The third of a new group of single-screw cargo vessels ordered by Messrs. Alfred Holt and Company for their Far East trade has been completed. This vessel, the *Diomed,* 9,200 tons d.w., has been built by the Caledon Shipbuilding and Engineering Co., Ltd., Dundee. The *Diomed* was launched on 12th April 1956, and completed her trials on August 23rd. The first of this new group of motor vessels was the *Demodocus,* built by Vickers-Armstrongs, Ltd., at their Naval Yard, Walker-on-Tyne, last year. This vessel was the first to be fitted with the new Harland and Wolff single-acting opposed-piston turbocharged engine. This engine is a six-cylinder unit with cylinders of 750-mm. bore and 2,000-mm. combined stroke, and has a service power of 8,000 s.h.p. at 112 r.p.m. Both the *Demodocus* and the second ship *Dolius* are equipped with this type of propelling machinery, and a feature of the *Dolius* is that she has a gas turbine-driven emergency generating set fitted in a special compartment on the boat deck. The results from this unit have been very satisfactory and subsequently four other vessels in the group will be fitted with these sets. The *Diomed,* however, is fitted with a Diesel engine driven emergency generating set. The propelling machinery in earlier Diesel engined vessels is rated at 6,800 s.h.p. The new turbocharged engine develops 8,000 s.h.p. and occupies less length of engine room. The rating of 8,000 s.h.p. is, incidentally, extremely modest as a service output of 9,000 s.h.p. is within the capacity of this engine. The principal particulars of the *Diomed* are as follows:—

Length overall	491ft. 6in.
Breadth moulded...	62ft. 4in.
Depth moulded to upper deck ...	35ft. 3in.
Draught	28ft. 5in.
Deadweight, tons...	9,200
Gross, tons	8,200
Machinery output, s.h.p. ...	8,000
Service speed, knots	15

The *Diomed* is of the two-deck type with poop, long centre-castle and forecastle, and has a raked stem and cruiser stern. She is fitted with six cargo holds and tweendecks, and an additional tweendeck space aft which, like the lower holds of Nos. 2 and 3, is arranged for the carriage of liquid cargoes, giving a total of seven deep tanks in all. The hatches are served by 27 derricks, 18 of which are arranged for 5-ton lifts and eight for 10-ton lifts, with, in addition, a heavy-lift derrick of 70 tons capacity. A 7-ton derrick is also fitted on the funnel for the purpose of serving the engine room. The main propelling machinery consists of a turbocharged single-acting opposed piston Diesel engine built by John G. Kincaid and Co., Ltd., Greenock. The ship is fitted with what is in fact the first large exhaust gas turbocharged Diesel engine to be built in Scotland. The turbocharging system for the engine is based on the variable pressure principle. The port timing is effected solely by the pistons and there is no additional control gear. As in the case of the non-turbocharged engine there is a phase angle between the movements of the exhaust and main piston 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 no chain driven blowers the design of the engine is similar to that with normal induction. The entablatures, "A" frames, and bedplates are of fabricated steel and the crankshaft of fully built construction. Located at the back of the engine are the Napier turbochargers. The energy available in the exhaust gases at the higher loads is sufficient to ensure adequate charging air volume and pressure. At low speeds, when manœuvring the blower, rotors are rotated by an electric motor transmitting torque to the rotor shaft by means of an Airflex coupling. With one blower out of action the ship can easily attain half service speed. The scavenging and charging air is drawn from the engine room into the turbo-blowers through combined filters and silencers. After compression the air is discharged to the scavenging air manifold through coolers in which the air temperature is reduced to about 10 deg. F., above that of the cooling water. Sea water is circulated through the cooler tubes.—*The Shipping World, 3rd October 1956; Vol. 135, pp. 293-296.*

Gas-shielded Metal Arc Cutting Process

Ever since the oxygen cutting process first proved capable of severing mild steel sections without mechanical force, researchers have tried to adapt it to the cutting of other metals. Unfortunately, among the common metals of construction only mild steel appears to have the precise combination of physical and thermic properties which permit the oxygen jet to pierce and sever by oxidation alone. Adding other materials to the oxygen jet, such as fluxes or combustible materials, permits the oxygen severing of numerous other metals, but the associated problems are such that these techniques are generally used only in the oxygen cutting of stainless steels. The heat of the arc will also accomplish the severing of all metals used in common structural practice, but the action is purely a melting one. Such is the function of the arc in the common methods of carbon arc, arc gas and covered electrode cutting. Using standard gas-shielded metal arc welding apparatus, a process



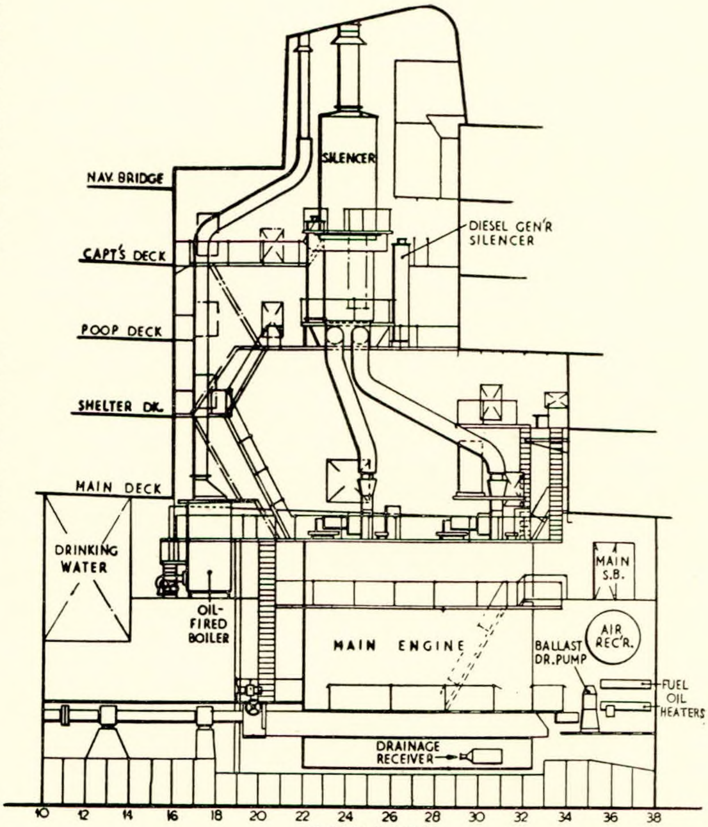
has been developed which permits the cutting of numerous materials that resist oxygen cutting to such a degree that most of the severing of them is generally carried on by mechanical means. This new cutting process is capable of both manual and machine operation and produces cut surfaces which in many cases are suitable in the as-cut condition for further processing, such as welding. The necessary apparatus is likely to be available in any well-equipped shop doing fabrication of non-ferrous materials, and the art is readily mastered by men trained in the field of gas-shielded arc welding or oxygen cutting.—*E. H. Roper, The Welding Journal, September 1956; Vol. 35, pp. 915-919.*

French-built Vessel for British Owners

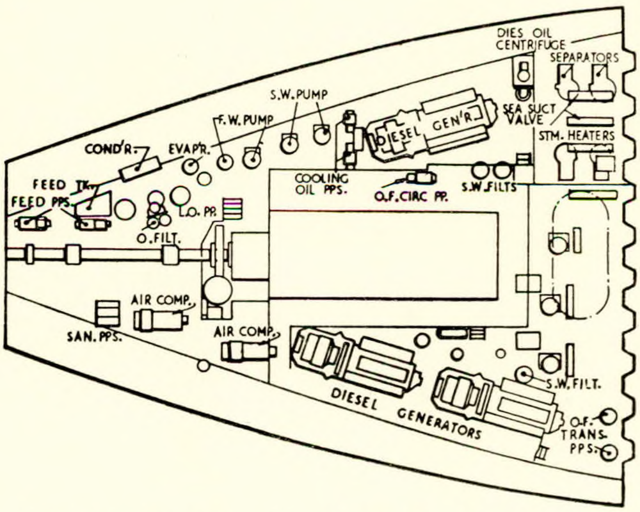
The first of a new class of vessels being built in France for a British owner has now been delivered. This vessel, the *La Pradera,* 11,366 tons d.w., has been built by the Chantiers de l'Atlantique Penhoet-Loire, St. Nazaire, for Buries Markes, Ltd., London. The vessel is of interesting design and has both her machinery and navigating bridge aft. The *La Pradera* has been built with a Maierform-type hull, and is powered by a Burmeister and Wain turbocharged Diesel engine. Her principal particulars are as follows:—

Length overall	454ft. 0in.
Length b.p.	444ft. 0in.
Breadth moulded...	61ft. 1in.
Depth to shelterdeck	40ft. 6in.
Depth to main deck	31ft. 0in.
Draught, summer	27ft. 3in.
Deadweight, tons...	11,366
Gross, tons	6,980
Net, tons	3,740
Machinery output, b.h.p. ...	4,600
Service speed, knots	$14\frac{1}{4}$
Block coefficient	0·74
Cargo capacity:	
Grain, cu. ft.	709,401
Bale, cu. ft.	615,183

The *La Pradera* is an open shelterdeck type vessel with a raked straight bow and cruiser stern, and has been built under the survey of the Bureau Veritas. A feature of the vessel is the radius connexion between the sheerstrake and the stringer plating, and also between the hatch coamings and deck plating. This has been done to ensure against a sudden break of section. The material used is of specially toughened steel. The vessel is divided as follows: Forepeak, No. 1, No. 2 and No. 3 hold, which has a deep tank under the main deck, Nos. 4 and 5 holds, the machinery compartment and the afterpeak. The shelterdeck is continuous and the main deck is lowered aft for the accommodation. There are seven watertight bulkheads, and bulkheads are arranged in way of the centre line between the hatches. Bunkers and double bottoms for fuel have a total capacity of about 45,200 cu. ft., for Diesel oil 5,300 cu. ft., and fresh water 8,300 cu. ft. The forepeak is used for ballast water and the after peak for fresh water. Nos. 3, 4, 5 and 6 double bottom tanks are for Diesel oil or fuel oil, and the forepeak, Nos. 1 and 2 tanks are for water ballast. The double bottoms and shelter deck are built on the longitudinal system; the shell, main deck and after and fore sections on the transverse system. The stem is of welded steel plate. The bottom plating is welded, and the centre girder is welded to the shell plating and the ballast tank top. Welding is also used for all deck plates, and a welded composite stern post is fitted. The hatch end beams are strengthened in order to compensate for the absence of pillars at the corners of the hatches. The propelling machinery in the *La Pradera* consists of a Burmeister and Wain six-cylinder type 662-VTBF-140, two-stroke, single-acting, turbocharged Diesel engine of 620 mm. bore and 1,400 mm. stroke, developing 4,600 b.h.p. at 115 r.p.m. The main engine is fitted with two exhaust turbochargers fitted with intake silencers and air coolers. In addition to the exhaust gas-driven turbochargers, an electrically-driven emergency blower is fitted in the engine room. The engine can run on the emergency blower only, on the three forward cylinders and with the after three cylinders cut out. Under these conditions the engine can achieve 60 per cent r.p.m. and 35 per cent of its power. The main engine m.i.p. is about 112·4lb. per sq. in.—*The Shipping World, 31st October 1956; Vol. 135, pp. 385-387.*



ELEVATION



PLAN VIEW OF ENGINE ROOM FLOOR

Machinery arrangement in the La Pradera

Ultrasonic Inspection

This paper summarizes the experiences of ultrasonic testing equipment in oil and marine industries and includes supporting data, types of equipment involved, etc. Types of equipment covered are those used for non-destructive thickness measurements and flaw detection. Thickness measurements are made to determine corrosion rates in pipelines, storage tanks, refinery equipment, oil tank ships, etc. Cost savings as compared with "drilling and rewelding" are described. Advantages resulting from large volumes of accurate data are explained. The major activity is the measurement of thickness of structural members, hull plating and bulkheads of seagoing oil tankers. Comparative costs of testing techniques are included in the paper. Non-destructive flaw detection activities described include testing of oil field drilling equipment, compressor engines and preventive maintenance tests. The paper also includes a brief description of the advantages and disadvantages of ultrasonic weld testing, the capabilities of present instrumentation for flaw detection and the equipment needed for specific oil industry applications.—*Paper by D. J. Evans, abstracted in Nondestructive Testing, September/October 1956; Vol. 14, p. 16.*

Measurement of Crack Depths by Direct Current Conduction

The nondestructive testing of materials and finished components, with its obvious advantages, has greatly increased in scope and application during the past twenty-five years. In particular, the detection of flaws and cracks has received marked attention, and many new methods relying upon the magnetic or electrical characteristics of the materials or the use of ultrasonics have been developed, while existing methods have been improved. Little has been published, however, concerning the accurate measurement of the depth of surface cracks. The combined efforts of many investigators in the past have produced several highly successful techniques for flaw detection in ferrous and non-ferrous materials, among them being the magnetic particle method, the dye penetrant method, eddy current induction method, X-ray method, ultrasonic method, and alternating current and direct current methods. Some of these methods have also been used to estimate crack depths, but difficuties are encountered in their application. Magnetic particle and dye methods have negligible value in determining the crack depth. X-rays and ultrasonics, in addition to the initial high expenditure, have many other drawbacks. Alternating current conduction and eddy current induction methods have been used successfully in certain cases for crack depth measurement, but in both methods the phenomenon of the "skin effect" poses problems of calibration for various materials, and limits the depth of crack that can be measured satisfactorily. The present article develops the underlying theory of the direct current conduction method, and describes both the use to which it has been put in the past and an apparatus that is capable of measuring the depth of cracks of the above type, on any metal, magnetic or non-magnetic, that is reasonably homogeneous and isotropic.—*J. G. Buchanan and R. C. A. Thurston, Nondestructive Testing, September/October 1956; Vol. 14, pp. 36-39.*

Ultrasonic Testing by Immersion Method

In the immersion method of ultrasonic testing the searching unit does not directly contact the test part, but is usually separated from it by 1 inch or more of water. Other liquids may be used, but water has been found most satisfactory. The first maximum (or peak) in the reflected pattern shows the initial pulse, the second is the top surface reflection, the third is the reflection from the discontinuity and the fourth, the bottom surface reflection. These are followed by second order reflections from discontinuity, bottom and top surfaces. The equipment necessary for ultrasonic testing by the immersion method includes: (1) pulse-echo type instrument, such as the Immerscope or Reflectoscope; (2) tank with suitable couplant in which the part can be immersed; (3) searching unit; (4) holder for searching unit; (5) traversing mechanism. In positioning the searching head, at least 1 inch of water must be used for every 4 inches of aluminium or steel. If this is not done the second order reflections will appear between the first order front and back surface indications, hiding discontinuities. The searching unit must be aligned properly with the test piece so that a maximum amount of energy can search the part. Data relating to frequency used in inspecting under various conditions are given. Although it is almost impossible to determine quantitatively the size of a defect by ultrasonics, comparison with reference blocks is the method most widely used. Advantages of this method of inspection include: (1) It permits inspection of parts with relatively rough surfaces. (2) In contact testing, rough surfaces can cause excessive wear of searching units, and, in some cases, sufficient ultrasonic energy may not enter the part for a proper test. (3) It permits the use of high test frequencies, which are required for close-to-the-surface inspection. The use of high frequencies with the contact method is limited. (4) It permits easy scanning of the article since a mechanized installation can be used. (5) It permits angulation of the ultrasonic beam, so it can be directed normally into fillets (often inaccessible with the contact method) and, since the beam can be readily angulated, maximum energy can be reflected from defects lying on an angle other than parallel to the surface.—*J. B. Morgan, Steel (U.S.A.), 27th August 1956; pp. 94-96; p. 28. Abstracted in The Abstract Bulletin of Aluminium Laboratories, Ltd., September 1956; Vol. 27, pp. 588-589.*

Propeller Boss Fairings

In a recent issue of "Marina Italiana", E. Castagneto discusses the results of open-water experiments on propellers with shaft bearings of different diameter, or in the presence of a model rudder; and of propulsion experiments with and without a rudder. It is concluded that the presence of an obstruction modifies the wake fraction and can contribute to improved propulsive efficiency. The experiments were carried out with model propellers attached to shafts, some of which were used bare, while others were encased in a thin sheath. Three different propellers were used, one of which had adjustable blades. In each case the propeller was removed from the shaft and the test repeated with a solid body representing the boss so as to deduce the thrust and torque coefficients of the propeller blade. The tests were carried out both at constant speed of advance with different rates of shaft revolution and *vice versa*. The results showed that increasing the diameter of the shaft or the sheath increased the thrust, torque, and efficiency of the propeller in open water. The increase in thrust remained practically constant, but the increase in torque decreased with increase of load. The increase in efficiency for a shaft and boss of equal thickness ranged between 2 and 3 per cent. The effective pitch at zero thrust also increased with shaft diameter. Under given conditions, the thrust and torque are lower when the shaft is unsheathed. The results show the necessity for standardizing experimental procedure in open water propeller tests. In the presence of an obstruction, such as a rudder on single screw ships, the hydrodynamic phenomena show substantial variations which are not easily reduced to simple theory.—*Journal, British Shipbuilding Research Association, October 1956; Vol. 11, Abstract No. 12,075.*

Weld Joint Flaws as Initiating Points of Brittle Fracture

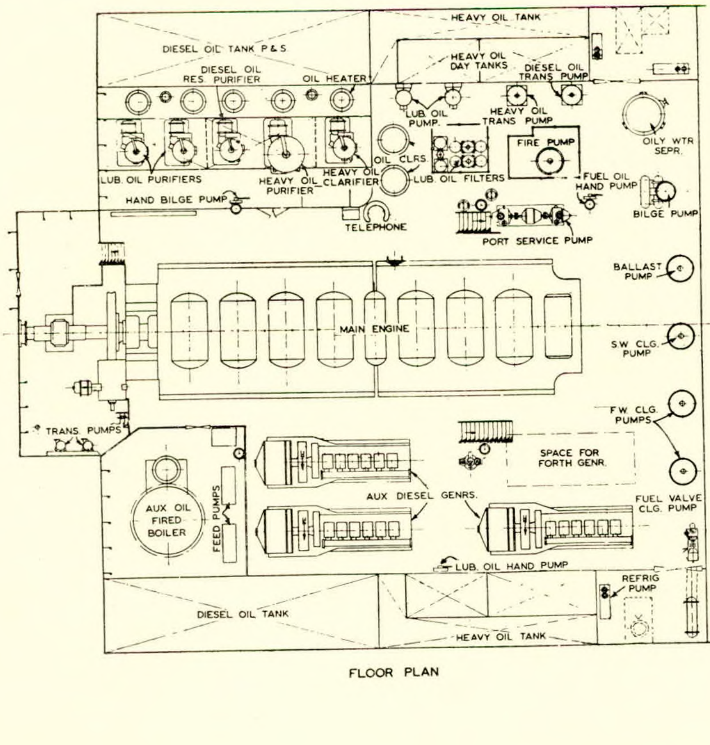
A literature survey was made to determine the fundamental factors and circumstances that are known about brittle fractures in ship steels and similar materials. The survey was the initial part of this investigation for the Ship Structure Committee under a Bureau of Ships contract on the evaluation of flaws in weld joints. Various testing methods and specimens used in previous investigations involving brittle fracture were reviewed. Preliminary studies were made to determine the best method of introducing flaws into weld joints to simulate the flaws found in service failures. A major portion of the effort on the project has been involved with determining: (*a*) what kind of test specimen and apparatus should be used to evaluate weld-joint flaws; (*b*) what kind of loading or types of loading are needed to simulate service conditions in ships or other large structures; and (*c*) what nominal stress is required to initiate a brittle fracture from large weld cracks and other flaws such as lack of weld fusion. A significant result has been that brittle fractures initiate from the weld defects in the laboratory specimen under conditions very close to the reported conditions involved in some service failures of ships.—*D. C. Martin, R. S. Ryan and P. J. Rieppel, Ship Structure Committee Report No. SSC-86, 1956.*

German Vessel with A.C. Auxiliaries

The *Cläre Hugo Stinnes,* built by Nordseewerke Emden G.m.b.H. for the Brennstoff-Chemikalien-und Transport, AG, Mühlheim-Ruhr, is one of ten vessels to be constructed by Nordseewerke having the same principal dimensions but propelled by different types of machinery. The other nine ships, of which the *Roland* was completed earlier this year, are for Norwegian, Panamanian and Greek owners, and have slight differences in equipment and general arrangement. As an open shelterdecker the new ship would have a deadweight capacity of 10,932 tons, but as she is completed as a closed shelterdecker the deadweight capacity is 13,465 tons. Constructed to the requirements of Lloyd's Register of Shipping + 100 A1, "Strengthened for Navigation in Ice", the main characteristics of the *Cläre Hugo Stinnes* are as follow:—

Length overall	153·33 m. (503ft. 1in.)
Length b.p.	141·16 m. (463ft. 1¾in.)
Breadth, moulded...	18·9 m. (62ft. 0in.)
Depth to upper deck	11·97 m. (39ft. 3in.)
Depth to second deck	9·23 m. (30ft. 3in.)
Draught (summer), closed ...	9·1 m. (29ft. 10in.)
Corresponding deadweight, tons	13,465
Corresponding displacement, tons	18,537
Gross register, tons	9,269·2
Net register, tons...	5,397·3

The vessel has two continuous decks and five cargo hatches, the latter having an average width of 8 m. (26ft. 3in.) and lengths varying from 9·8 m. (32ft. 1¾in.) to 13·6 m. (44ft. 7½in.). All the winches are mounted on deckhouses, thus keeping the decks free for hatch covers and cargo. All fourteen winches have motors of the A.C. type manufactured by Siemens-Schuckert, who were also responsible for the anchor windlass and warping winch driving units. A 40-ton heavy-lift derrick is arranged at No. 2 hatch. Although the winches are served by alternating current, the steering gear is driven by a Siemens rudder engine supplied with current at 190 volts d.c. by a generator driven from an 18-kW 380-volt three-phase motor fed from the engine room switchboard. A Ward Leonard transformer controls the supply to the steering motor. The steering motors and their equipment are duplicated. The main engine is a seven-cylinder M.A.N. single-acting two-stroke unit having a cylinder bore of 780 mm. and a stroke of 1,400 mm. It develops 6,300 b.h.p. at 119 r.p.m. An exhaust-driven turbocharger is fitted, as well as a scavenge pump. The engine is designed to operate on heavy oil and the fuel consumption is 24 tons per day, bunker capacity being provided for 800 tons of heavy oil, 263 tons of Diesel oil and 34 tons of lubricating oil. Three 275-b.h.p. six-cylinder four-stroke M.A.N. auxiliary engines, each having a cylinder bore of 235 mm. and a stroke of 330 mm., drive Siemens-Schuckert 225-kVA 400-volt 50-cycle generators at 500 r.p.m. The entire ship's electrical installation, including all engine room auxiliary motors, is served by three-phase alternating current.—*The Motor Ship, November 1956; Vol. 37, pp. 319-322.*



FLOOR PLAN

True Motion Radar

Until now all merchant ship radar sets have retained one common basic principle—one's own ship is in the centre of the screen. Described in another way, one's own ship remains fixed on the P.P.I. while all targets, moving vessels, landmarks, buoys, and stationary vessels move relatively. Now, however, the Decca Radar Company, in the new True Motion TM 46, have made a fundamental departure in practice by introducing "true motion", in which landmarks and fixed targets remain steady while all moving targets, as well as one's own ship, are actually seen moving across the screen. At a glance, by reference to the afterglow or "wake" of any vessel (the trace persisting as the moving target progresses) the direction of that vessel is immediately obvious; for the same reason, the track of one's own ship is also clear. By the very nature of its presentation it gives the observer a bird's eye view of his surroundings and comes nearer to normal daylight conditions than any of its predecessors. The display of true motion is obtained by feeding compass and speed information regarding one's own ship into a small resolver in the Trackmaster unit mounted on or alongside the viewing unit. The speed and direction of the ship are here converted into movements east-west and north-south. These movements control the amount of current in the off-centring coils of the display itself and affect the position of the electrical centre, that is, the position of one's own ship about which the trace rotates. On a compass-stabilized display, when the electrical centre moves in harmony with the course and speed of one's own ship, the effect on the P.P.I. is to present this true motion quality. The amount of movement imparted to the display depends on the radar scale in use and is automatically adjusted when the range scale is altered. The speed of one's own ship is fed into the Trackmaster either by hand or automatically from the ship's log, and the accuracy requirement here is not high. The bearing of one's own ship comes from a transmitting compass. In finding the bearing of other ships very accurately, an electronic bearing cursor, with parallel lines, can be employed. Where less accurate information is needed, the afterglow trails (or wakes, as mentioned above) give a sufficient indication. As one's own ship is continually moving over the face of the tube or screen in the direction of her course, the range of warning ahead is steadily being reduced and would ultimately be ineffective unless one's own ship's position were reset. Resetting is accomplished simply and speedily by shift controls, one east-west and the other north-south, and one's own ship can be positioned anywhere on the screen. After resetting, the true motion picture is painted again almost instantaneously. How often the position would have to be reset must be dependent on the range scale being used and on the speed of own ship. Obviously, therefore, a slow ship on a 10-mile range would take longer to make the passage across the screen than a fast ship on a 5-mile range scale. For example, because a 12-knot ship covers six miles in 30 minutes, there would be more than 10 miles' warning remaining ahead after 30 minutes on the 10-mile scale, provided one's own ship was started nearly fully off-set on the reciprocal of her course. A rapid assessment of the speed of other ships can be obtained by timing the movement of their echoes as observed on the screen. Incorporated in the TM 46, in addition to the true motion display, are the conventional ship's head-up picture, and the north-up picture, and one can be changed to the other by the manipulation of a switch.—*Shipbuilding and Shipping Record, 1st November 1956; Vol. 88, pp. 570-571.*

Novel Electrical Cable Reel

A new and unique cable winch design has been devised to maintain physical and electrical continuity of the cable without a slipring unit. The basic design, sketched isometrically in Fig. 1, consists of a compact, four-drum assembly. The cable is paid out or retrieved by rotating the outer drum in the proper direction. The internal three-drum assembly has a differential action so that one end of the cable can be fixed by transferring cable to or from a stationary drum as the outer drum rotates. The outer drum and drum A (Fig. 1) are integral and rotate as a unit. Drums A and C are of equal diameter and are axially aligned. Drum C is stationary. Drum B revolves about drums A and C and also rotates about its own axis in a direction opposite to its direction of revolution about A and C. All drums are spirally grooved to accommodate cable. The proper rigging and drum winding of the cable is shown in Fig. 1. Cable from the outer drum is led to and wrapped around drum A. The cable is transferred from drum A to drum B and then to drum C with appropriate wrapping on each drum. In operation, the outer drum and drum A rotate in a clockwise direction to pay out cable (see Fig. 1). Drum B revolves about A and C clockwise at half the rotative speed of A. As drum B revolves about A clockwise, it rotates about its own axis counter-clockwise, simultaneously removing cable from drum A and laying cable on drum C. When the outer drum rotates in a counter-clockwise direction to retrieve cable, the transfer of cable on the inner drum assembly moves from C via B to A in the same manner as before but in the opposite direction. The cable used on the internal drum assembly automatically spools itself while transferring from one drum to the next. The transfer of cable occurs without any twisting or sliding on the drums. Some of the more obvious advantages of the new proposed design over conventional winch and slipring unit assemblies are: (*a*) The elimination of costly slipring units. (*b*) The elimination of continual slipring maintenance. (*c*) The elimination of the sealing problem with increased reliability, especially in corrosive media such as sea water or salt water spray. (*d*) The elimination of contact resistance and hazardous brush arcing. A solution to other design problems requiring a transfer of electrical power, or a signal to a rotating member of a finite number of turns without sliprings, lies in the proper adaptation of the design.—*F. T. Grubelich, Bureau of Ships Journal, November 1956; Vol. 5, pp. 37-38.*

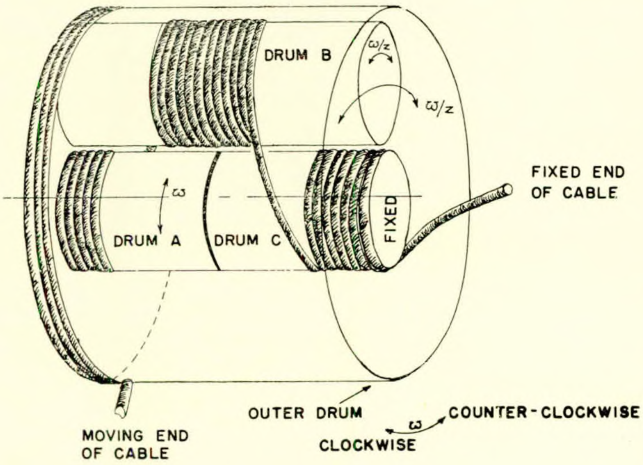


FIG. 1—*Isometric sketch of the cable reel, showing the four drums and the direction in which the cable revolves around them*

Supercharging Large Two-stroke Engines with Transverse Scavenging

In a paper read at a meeting of the Association Technique Maritime, R. De Pieri describes the development of supercharging for Fiat two-stroke engines with transverse scavenging. Engine driven piston pumps normally supply scavenging air. The first studies in 1942 were carried out on a 4,500-h.p. eight-cylinder engine used to drive a 3,200-kW alternator. Two Brown-Boveri turboblowers running at 8,000 r.p.m. were employed, with an intercooler between them and the air pump. Later an aftercooler was added. A constant-pressure turbine proved preferable to an impulse turbine since it was easier to construct and less easily damaged in operation. This installation gave an engine output of 6,900 h.p. and an m.e.p. of 7·12 kg./cm.^{2} (101·3lb./sq. in.). The consumption, although not extremely low, was less than that of the naturally-aspirated engine. Endurance runs at an m.e.p. of 6 kg./cm.^{2} (85·3lb./sq. in.) left the engine in completely satisfactory condition. Trials on a more modern 4,800-h.p. engine with a similar supercharging cycle resulted in a power output of 9,000 h.p. The power increment increased appreciably with load. The Fiat policy was to provide more air than necessary to reduce the mean temperature of the working cycle, which results in a lower efficiency but an enhanced working life. The air circulating through the engine remains abundant even at high powers. For medium-power engines, two-thirds of the power required to compress the scavenging air is provided by the turboblower and the remaining one-third by the engine-driven pumps. With a high degree of supercharge the specific consumption was found to increase slightly, probably because the injection system had not been designed for so high an m.e.p. Another series of tests was carried out with the engine-driven scavenge air pumps arranged in parallel with the turboblower, the latter again providing about two-thirds of the total quantity of air required. At low and medium speeds the engine did not run satisfactorily because the turboblower was operating below its delivery speed. Above an m.e.p. of 4 kg./cm.^{2} (57lb./sq. in.) the running became regular, satisfactory power was developed, and consumptions approximately equal to those of the earlier tests were obtained. Despite the greater simplicity of construction of the parallel arrangement, its less satisfactory behaviour at low speed made it appear preferable to keep to the arrangement with the piston pumps in series with the turboblowers. Tests now under way suggest that the Fiat supercharging principle can be applied equally well to smaller, higher speed engines. A supercharged Fiat marine engine has given over 1,100 hours' trouble-free running. Two more eight-cylinder engines are being installed in two new ships in the Cantiere Navale Breda. New Fiat engines of the type developing 750-800 h.p./cylinder are now designed with a view to supercharging, with multiple scavenge-air pumps replacing the single pump.—*Journal. The British Shipbuilding Research Association, October 1956; Vol. 11, Abstract No. 12,127.*

Performance of Trawlers During Fishing

A paper by M. Jourdain read at a meeting of the Association Technique Maritime presents an analysis of trawler performance, taking into account the effects of current parallel to the ship's route, and the influence of bad weather. The speed and r.p.m. relationships are plotted and examples are given of the construction of such graphs from practical speed measurements during trawling. For a given trawl it appears that the efficiency of a trawler may be considered to be proportional to the speed of the net through the water. In calm weather, this is virtually constant, no matter what the speed of the current or the apparent speeds of the trawler relative to the bottom of the sea, provided that the propeller torque remains constant. In bad weather the speed of the trawl net through the water is reduced owing to the increase in the resistance of the trawler due to the effects of wind and sea. The efficiency of a trawler while fishing depends therefore on the torque available, the hydrodynamic resistance of the net, and the sensitivity of the trawler to the effects of sea and wind. The most profitable improvements are likely to result from studies of the resistance of the trawling equipment—a domain that has scarcely yet been explored.—*Journal, The British Shipbuilding Research Association, October 1956; Vol. 11, Abstract No. 12,169.*

Organic Moderated Reactor Experiment

A small experimental reactor, the Organic Moderated Reactor Experiment (OMRE), is being built for the U.S. Atomic Energy Commission at Idaho by Atomics International, and is expected to operate early next year. An organic compound, diphenyl, serves both as moderator and coolant. OMRE will produce about 16,000 kW in the form of heat. The reactor core will consist of plate-type fuel elements, containing uranium 235, and the organic coolant, which will be in the liquid state while the reactor is working. Heat from the fission process in the reactor core is absorbed by the diphenyl which circulates through heat exchangers, and through a purification still. Advantages of diphenyl are: (1) high hydrogen content, which makes it an excellent moderator; (2) relatively high boiling point, so that there is no need to maintain it under high pressure; (3) negligible corrosion with standard materials of construction; (4) it does not react readily with uranium; (5) it becomes only slightly reactive on exposure to nuclear radiation.—*Atoms for Peace Digest, 14th July 1956; Vol. 2, p. 11. Journal, The British Shipbuilding Research Association, October 1956; Vol. 11, Abstract No. 12,135.*

Ocean Waves and Wave-induced Ship Stresses

Little is known about the frequency of occurrence of the various magnitudes of ocean waves. Even less is known about the severity of the wave-induced motions and stresses which ships experience in service. The intent of this research effort is to show that, by utilization of statistical methods, it is possible to describe and predict service conditions for ships in an orderly and relatively simple manner despite the general complexities of the problem. Wave observations taken continuously over a period of six years at several weather stations in the Atlantic Ocean were studied. Wave-induced motions and stresses in ships obtained under a wide range of operating conditions were studied for seven different ships. On the basis of an analysis of voluminous experimental data it is concluded that the probability distributions of wave height, wave length, wave-induced pitch, roll, and heave motions of ships, and wave-induced ship stresses may all be approximated by a one-parameter-type distribution when the environmental conditions are steady, whereas these variables will tend to follow the two-parameter logarithmically normal distribution when the environmental conditions are allowed to vary over a wide range. It is desirable, from the standpoint of savings in time, and cost, that the actual distribution applicable to particular ships be determined in the future by means of either theory or model tests. There are far too many unknowns to make the purely theoretical approach practical. Model tests would seem to offer the most direct method that could be developed into a useful tool within the immediate future. It will first be necessary to compare the distribution obtained from model tests with those obtained from full-scale sea trials under comparable conditions.—*Paper by N. H. Jasper, read at the Annual Meeting of the Society of Naval Architects and Marine Engineers, 15th November 1956.*

American Tanker for Arctic Service

The first of two new ocean-going tankers designed to withstand the rigours of Arctic travel was launched recently at the Bethlehem Steel Company's Staten Island Yard. Main propulsion for the new vessel, the U.S.N.S. *Alatna* (T-AOG 81) and for its sister ship, the U.S.N.S. *Chattahoochee* (T-AOG 82) will be provided by two sixteen-cylinder Diesel engines. The ships are being built by Bethlehem for the U.S. Navy's Military Sea Transport Service under a U.S. Maritime Administration contract. They are described as "twin-screw, bulk petroleum products carriers, ice-strengthened". The 3,450 deadweight-ton ships are designed to carry Diesel oil, aviation gasolene and motor gasolene on Arctic supply runs for the U.S. Department of Defense. They are fabricated with a heavy band of steel encircling the ship at the water line to give them limited ice-breaker capabilities. Because the ship hulls have been designed similar to ice-breaker hulls, they may ride up on sheet ice and then use their weight to break through. The new bulk petroleum carriers are 302 feet long, with a 60 feet beam and a depth to the main deck of 26½ feet. They have a designed draft of 19 feet and will operate at a designed sea speed of 12 knots. The Alco model 251 Vee-type, sixteen-cylinder engines for the twin-screw ships are 2,400-h.p. units that will develop 2,000 h.p. at 1,000 r.p.m. to motivate two propulsion generators. The engines have a 9-inch bore and 10½-inch stroke, and weigh 42,000lb. each when dry. Their output is bolstered by an Alco water cooled turbo-supercharger designed and built as an integral part of the new engines. The engine base supplied for marine service with the 251 sixteen-cylinder power plant has a deep sump to provide positive lubrication under 15-deg. inclination and list conditions. The engine is provided with a bronze engine driven salt water pump to circulate salt water to shell-and-tube jacket-water cooler, and an interchangeable-jacket-water pump and opposite rotation for twin-screw applications.—*Marine News, October 1956; Vol. 43, p. 89.*

Shock Testing Equipment

It is reported that a new device has been evolved which makes it possible to simulate mechanical shock conditions accurately and reliably. Acceleration values several hundred times as great as gravity can be obtained in times measured in milliseconds. The device has a rated output thrust of 12,000lb., produced from the release of stored energy of a gas compressed to a pressure of 2,000lb. per sq. in., the sudden release of this energy being accomplished by the principle of over-balancing thrust loads. The apparatus is basically a cylinder separated by means of an orifice plate into two chambers, one of which is open ended. A piston moving in the open-ended chamber is connected to a piston rod, which constitutes the thrust column reacting against the test specimen, while on the underside of the piston, in line with the orifice plate opening, is mounted a small sealing ring. When the open-ended chamber is pressurized to some low value, the piston is forced down against the orifice plate, and the sealing ring serves to isolate the underface of the piston from the closed-end chamber, except for the small area inside the sealing ring. To obtain a balanced thrust condition on the piston, the closed-end chamber must be pressurized to an extremely high value, since the pressure in this chamber can act only upon the small area of the piston inside the sealing ring. However, as soon as the pressure in the closed-end chamber is increased above the balanced condition, the piston is forced away from the orifice plate. When this occurs, the sealing ring loses contact with the plate, and the high pressure in the closed-end chamber is instantaneously released, reacting on the entire underface of the piston and producing extremely high acceleration of the piston column and test specimen.—*The Engineers' Digest, November 1956; Vol. 17, pp. 456–457.*

Naval Propulsion Progress in the Last Ten Years

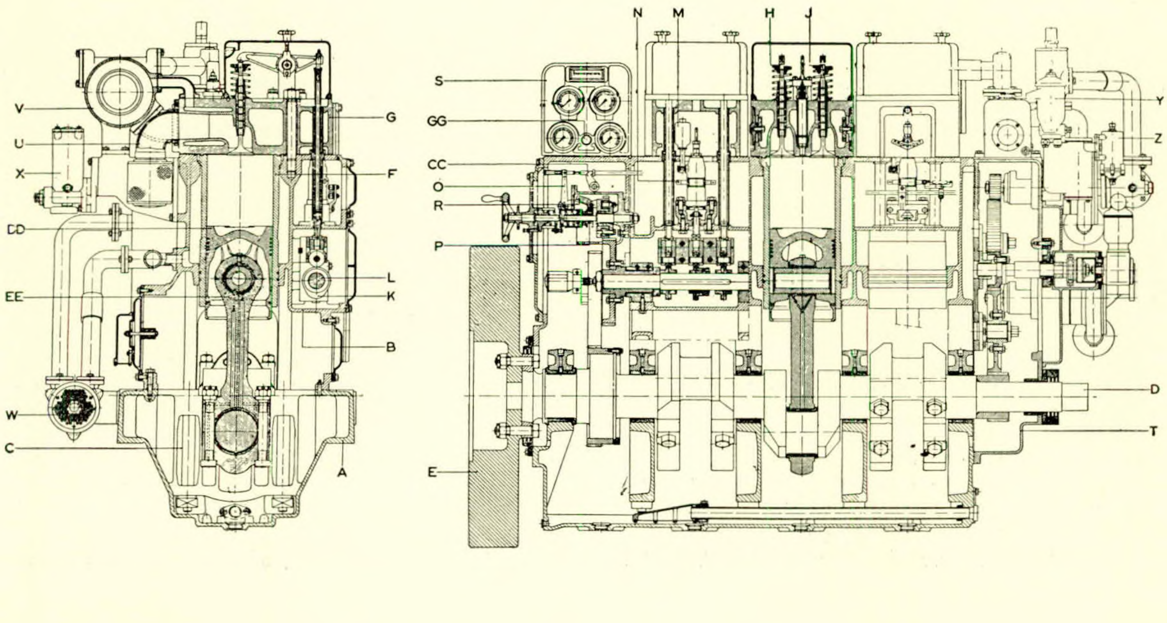
The end of the war in 1945 marked the start of a period of intensive research and development in the fields of steam turbines, Diesel engines, and gas turbines for naval propulsion. This has continued through the period and the fruits of this development work are now at sea in new classes of ships, which have recently joined the Fleet. At the same time the development of new types of weapons has resulted in great changes in the military characteristics of ships and this in turn has led to a further review of the requirements of propulsion machinery. The paper outlines three phases of machinery development; that based on analysis of the lessons of World War II and using engineering achievements up to 1950; that needed for ships influenced by new armament and the possibility of nuclear attack, and using engineering achievement up to 1955; and the final phase resulting from the possibility of using nuclear energy for propulsion. The general principles and the Admiralty establishments involved in the development and testing work are outlined, and the types of machinery evolved are described briefly. The first phase resulted in steam turbine, Diesel and gas turbine machinery developed along more or less conventional lines. The steam installations have been taken to a point where further development is subject to the law of diminishing returns for rapidly increasing effort. Diesel development has resulted in the Admiralty Standard Range of engines and the very light-weight Deltic engine for small craft. The Admiralty developments in gas turbines have resulted in several interesting engines of differing characteristics. These have given rise to definite conclusions on the type of gas turbine appropriate to naval use. These types of installation are briefly described and a fuller account of the development of each component is given in an appendix. The second phase demanded further reduction in the weight of machinery and fuel and also in the height of machinery spaces. This resulted in the adoption of steam turbine machinery for a proportion of full power with gas turbines geared to the same shaft to give the higher powers. The advantages of this type of machinery to meet nuclear warfare are mentioned. The possibilities of nuclear propulsion are briefly mentioned and the paper concludes with the lessons learned by the Admiralty in this period of development, the need for careful planning, the need for manufacturing excellence, and the importance of unconsidered trifles.—*The Twenty-first Parsons Memorial Lecture, given by Vice-Admiral Sir Frank T. Mason, K.C.B., at a Meeting of the North East Coast Institution of Engineers and Shipbuilders on 2nd November 1956.*

New Medium Speed Diesel Engine

The latest design development announced by Kromhout Motoren Fabriek D. Goedkoop Jr. N.V., is their F240 series of medium-speed four-stroke Diesel engines for marine and stationary applications. With cylinder dimensions of 240-mm. bore by 260-mm. stroke, the design is available with three, four, six and eight cylinders to give outputs of 190, 250, 375 and 500 b.h.p. respectively at 750 r.p.m. As it is seen from the drawing, the crankcase (A) and cylinder block (B) are both individual one-piece castings, this applying to all engines irrespective of the number of cylinders. The cylinder block and crankcase are bolted together by the long through-bolts (C) which relieve the cylinder block of the vertical combustion stresses. They also enable the cylinder block to be cast with the minimum thickness. Wet-type cylinder liners (F) are fitted in the cylinder block. The substantially-dimensioned crankshaft (D) is carried in the base plate and its end float is controlled by thrust bearings fitted near the flywheel (E). The connecting rods (EE) are one-piece forgings that carry the aluminium pistons (DD) and are fitted with replaceable big end bearings. An improved feature from the servicing point of view is that camshaft (L) driven by gear wheels from the crankshaft can be removed from the cylinder block laterally. The valve mechanism follows conventional lines with the inlet valves (H) and exhaust valves (J) in the cylinder head (G), operated from the camshaft through rocker arms, push rods and tappets. The camshaft, in addition to operating the valve mechanism, operates also the fuel pumps (M) and incorporates an air-starting cam which operates the air regulating slide valves during starting. The individual fuel pumps are fitted to each cylinder and the centrifugal governor (P) is operative over the entire speed range. An instrument panel (S) is located above the governor and all control instruments except the exhaust gas temperature meters are fitted to this panel. Various auxiliaries are mounted at the forward end of the engine and driven by a gear train from the crankshaft. These include lubricating oil and cooling water pumps, an air compressor and the cylinder lubricators. The lubricating oil system comprises the sump formed in the lower part of the bed plate, suction strainers, the oil pump, pressure filters and the oil cooler. A closed fresh water cooling circuit is normally employed, there being a centrifugal pump for promoting circulation with a thermostat included in the system. On the marine engine, the salt water and bilge pumps are of the plunger type arranged transversely and driven by an eccentric from the layshaft.—*Gas and Oil Power, September 1956; Vol. 51, pp. 195, 197.*

German Train Ferry with 10,000-b.h.p. High Speed Machinery

A ferry ship building at Kieler Howaldtswerke for the German State Railways for delivery in May, to be employed on the Grossenbrode-Gjedser run, will be the most remarkable vessel of her class yet built. She will be a combined car and train ferry, having a capacity for either ten express railway cars of 86ft. $7\frac{1}{2}$in. each, or thirty railways goods cars or, alternatively, she will carry 250 motorcars. She will run in conjunction with the existing German ferry *Deutschland* and the Danish train ferries *Danmark* and *King Frederick IX* and have similar bow and stern to these vessels. The new ferry will, however, have a second deck above the railway deck to



increase the motorcar capacity. Trains and cars enter and leave the ship through the bow and stern. During the construction the possible service of the ferry on the projected Puttgarden (Fehmarn)-Rödby route has been taken into consideration. The overall length is 446ft. 3in., the length b.p. being 426ft. 6in., whilst the beam is 56ft. 6in. moulded and 58ft. 0¾in. overall. The depth to the main deck is 24ft. 1¼in. and the draught 15ft. 8½in. The propelling machinery represents a complete novelty for such vessels. Aft there will be two propellers and, forward, a Voith-Schneider propeller. All propellers are driven by electric motors, current being supplied by Maybach four-stroke supercharged engines running at 1,400 r.p.m. and each driving a d.c. generator and three-phase alternator. The total capacity of the engines is about 10,000 b.h.p. Alternating current is supplied to the three propelling motors and direct current for the ship's service. The speed will be 16 knots, with a maximum of 17½ knots, and in normal operation the two propelling motors, situated in the after part of the vessel, will have an output of 6,000 b.h.p. when running at a speed of 150 r.p.m.—*The Motor Ship, October 1956; Vol. 37, p. 247.*

Hungarian-built Ocean-going Ships

The Gheorghiu Dej (formerly Ganz) shipyards at Budapest are now building a class of 2,500 tons d.w. twin-screw ocean-going motorships. These are of all-welded construction with aluminium alloy superstructures. The principal particulars of these ships which are being constructed under the supervision of Lloyd's Register of Shipping are as follows:—

Length overall	292ft. 0in.
Length between perpendiculars...	271ft. 0in.
Moulded breadth...	40ft. 0in.
Moulded depth	24ft. 5in.
Loaded draught	17ft. 2in.
Cargo capacity, cu. ft.	46,816
Total power, b.h.p.	2,400

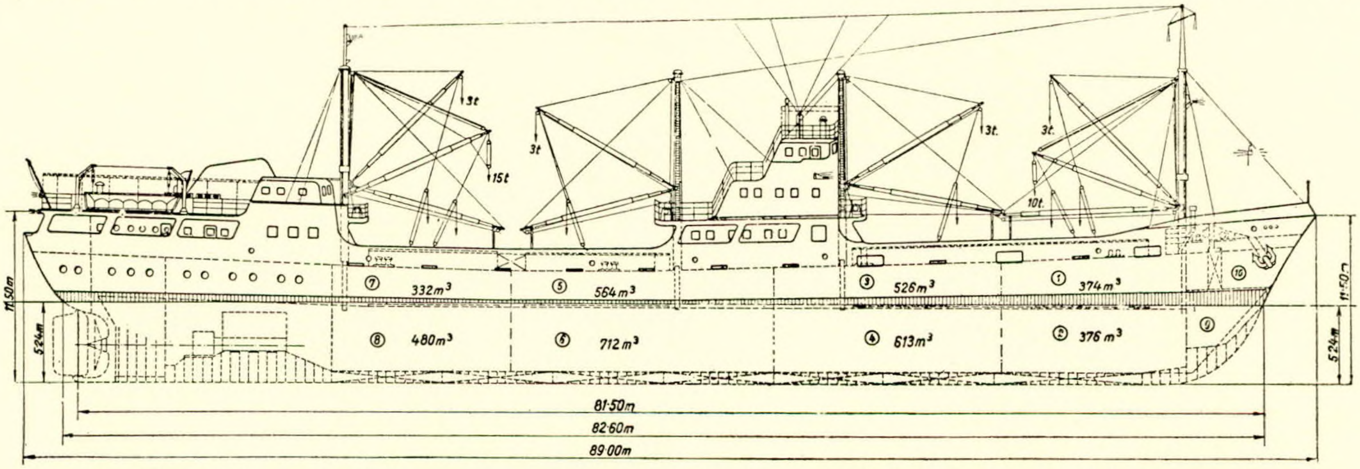
There are four holds, two forward and two aft of the bridge structure which is amidships, the machinery and remaining accommodation are aft. Cargo-handling equipment includes eight 3-ton derricks and one each of 10 and 15-tons capacity. —*The Marine Engineer and Naval Architect, October 1956; Vol. 79, p. 358.*

17½-knot Cargo Vessels

Most of the tramps or general-purposes ships which are being built in this country, with a few exceptions, are designed for a service speed of 12 or 13 knots, and the majority of those, representing perhaps 100-150, ordered from various Continental yards, are 14½-15-knot motor ships, the engine power for a 10,500-ton vessel being about 5,400 b.h.p. The two cargo ships which have recently been completed by Deutsche Werft, Hamburg, for Leif Höegh and Co. A/S, the *Höegh Cape* and the *Höegh Cliff*, destined mainly for liner services, but also to be employed as tramps—are of higher speed, being contracted for 17·25 knots on service, and in effect a speed of 19 knots was measured during the extensive trials of the *Höegh Cape* in the North Sea. This has involved the installation of an 8,750-b.h.p. B. and W. two-stroke turbocharged engine running on heavy oil. The fuel consumption is expected to be in the neighbourhood of 36 tons per 24 hr. In addition to carrying general cargo there are refrigerated cargo holds, also vegetable tanks and latex may also be loaded when required. The main particulars of the two ships are:—

Gross register, tons	9,191
Net register, tons	5,286
Length overall	496ft. 0in.
Length b.p.	455ft. 0in.
Breadth, moulded	64ft. 0in.
Depth, moulded to 1st deck ...	38ft. 6in.
Depth, moulded to 2nd deck ...	28ft. 6in.
Deadweight capacity, tons ...	10,400
Corresponding draught	26ft. 7$\frac{1}{16}$in.
Light displacement, tons ...	5,191
Corresponding draught	10ft. 5in.
Machinery, b.h.p.	8,750
Service speed, knots	17¼

The *Höegh Cape* and her sister ship have been constructed to comply to the rules of Det Norske Veritas +1A1. She is wholly welded and has a raked stem and cruiser stern. There are six cargo holds with a total capacity of 697,299 cu. ft. (grain) or 632,370 cu. ft. (bale). The refrigerated cargo hold on the second deck is divided into four compartments, including one section for the provision rooms and the total refrigerated capacity is 21,422 cu. ft. (grain). In No. 3 hold are four vegetable oil tanks, totalling 48,276 cu. ft., and these may be used alternatively for latex or water ballast. A standard Copenhagen-built Burmeister and Wain two-stroke turbocharged engine is installed in the machinery compartment amidships. It has seven cylinders with a diameter of 740 mm. and a piston stroke of 1,600 mm., the output at 115 r.p.m. being 8,750 b.h.p. The engine drives a four-bladed bronze propeller with a diameter of 5·75 m. (18ft. 10¼in.). The exhaust-gas blowers supply all the scavenging air required and there is an emergency electrically driven blower which can be brought into operation in the event of failure of the exhaust-gas blowers. The engine runs on boiler oil, and the centrifugal purifying plant is installed in a compartment at the starboard side of the engine room. In it are four De Laval separators, three of which are for purifying the boiler oil, and the other for Diesel oil or heavy oil if wanted. One of the purifiers is of the self-cleaning type. There is a fifth machine for the lubricating oil and this may be utilized as a spare heavy oil



Profile of the 2,500 ton d.w. twin-screw motorships which the Gheorghiu Dej shipyard are building on the Danube or Diesel oil separator. The generating plant comprises three six-cylinder 360 b.h.p. four-stroke B. and W. engines running at 500 r.p.m., each driving a 240-kW Thrige dynamo supplying current at 230 volts throughout the ship.—*The Motor Ship, October 1956; Vol. 37, pp. 253-254.*

Projected 25,000-ton Russian Tankers

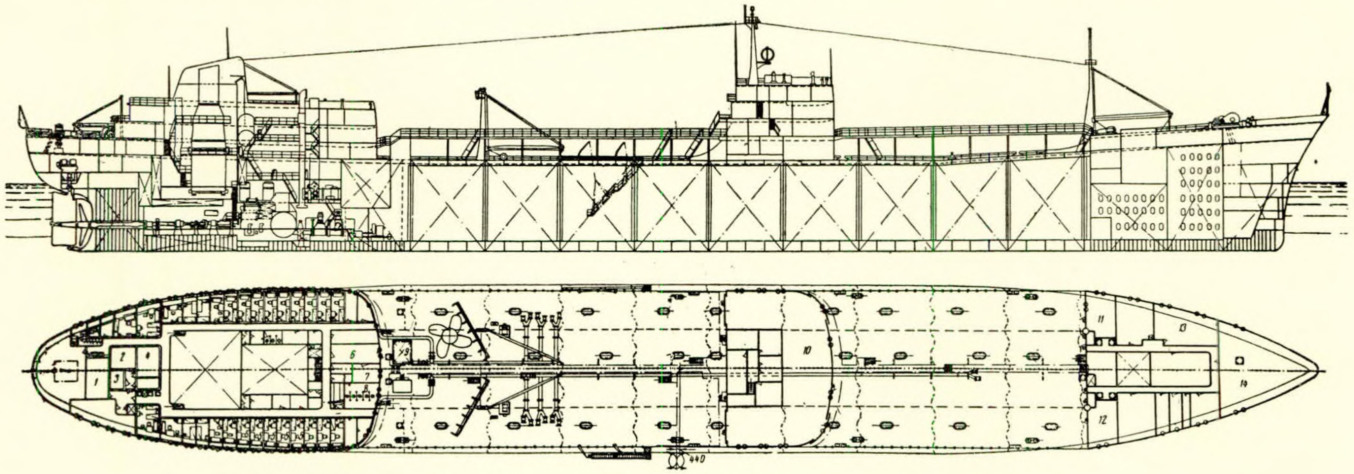
The sixth five-year plan (1956-1960) of the Soviet Union provides for augmenting the Ocean fleet by more efficient vessels and, in particular, improving the performance of the larger tankers by at least 25 per cent. The design office of the Ministry of Shipbuilding has, on behalf of the Ministry of the Sea Fleet, developed an 18-knot, 25,000 ton d.w. tanker design incorporating the following principal particulars:—

Length between perpendiculars...	620ft. 0in.
Moulded breadth...	86ft. 6in.
Depth	44ft. 0in.
Draught	33ft. 9in.
Deadweight capacity, tons	25,000
Total deadweight, tons	28,970
Displacement, tons	38,000
Designed speed, knots	18·5
Machinery power, s.h.p.	19,000

These ships are designed for transporting high grade fuel but can carry simultaneously three different grades of oil. They are to be largely welded with a bare steel weight of 6,000 tons corresponding to 5·7lb. per cu. ft. This figure corresponds with values for up-to-date vessels built elsewhere. In the worst condition—no cargo and 15,750 tons of ballast—the metacentric height will be 7ft. 7½in. The vessel will remain afloat with the engine room flooded in which case the main deck will be 10 inches above sea level, a figure in excess of the requirements of the Sea Register of the U.S.S.R. The cargo tanks are designed for alternative grain transport and equipment for rapid cleaning is thus required. The cargo spaces have been based on a specific gravity of 0·73. The ship's company of seventeen officers and forty-one men is to be accommodated in single and double-berth cabins amidships and aft, all being air conditioned with equipment capable of maintaining a temperature of 70 deg. F. and an ambient temperature of minus 3 deg. F. The main pump room is arranged immediately forward of the machinery and contains three 13¾in., 750-ton pumps with a discharge pressure of 150lb. per sq. in. The after tanks are intended for heavier grades of oil and with heating coils to ensure a temperature of 95 deg. F. which can be increased to 113 deg. F. before unloading. The propelling machinery consists of two water tube boilers rated at 41 tons per hour with a maximum of 55 tons per hour, delivering steam at 650lb. per sq. in. at 850 deg. F. to a cross-compound double-reduction geared turbine set. As a result of model tests carried out at the AN. Krilov tank, a five-bladed propeller has been chosen as this shows an improved efficiency at high powers. Auxiliary power is supplied by three 600 kW, 400 volt turbo-alternator sets. The lighting services are supplied at 127 volts. There is also a 100-kW Diesel alternating set. Deck equipment includes a gyro compass, echo sounder, automatic course plotter, radar and radio direction-finder. A streamlined balanced rudder with an area of 295 sq. ft. will be fitted. The table below compares this new design with a typical 16,000-ton 16-knot tanker and the 10,000-ton tankers of the *Casbeck* class.

	16,000-ton tanker	*Casbeck*	25,000-ton tanker
Transporting capacity, per cent	100	50	176
Cost of transport, per ton, per cent	100	143	104
Capital outlay, per ton, per cent	100	152	81
Work output of crew, per cent	100	49	133

—*The Marine Engineer and Naval Architect, October 1956; Vol. 79, p. 361.*



General arrangement of the 18-knot, 25,000 ton d.w. Soviet tankers

Service Damage in Rolling-type Bearings

There are many ways in which rolling-type bearings can be mechanically damaged before and during mounting. When a race, cage, and roller assembly is forced axially on the other race or in a cocked position, damage from seizure and smearing in cylindrical bearings may occur. If the parts are lined up properly and a lubricant is used, such damage will be prevented. It is advisable to have a film of oil or grease on the parts before assembling them. To mount bearings on a shaft by applying blows or pressure to the outer race will usually cause denting. If the pressure is great enough, the race may crack. With less pressure, the dents will be farther inboard in the raceway, and if the balls roll over them, the bearing will be rough and noisy. As operation continues, flaking will develop at the individual dents. Metal chips and dirt are certain to get into bearings when they are placed on dirty benches or floors. This foreign matter can produce noise, denting, locking of one or more rolling elements, and a host of troubles that ultimately cause failure. To mount bearings by striking the outer race with a hammer is to invite denting or brinelling. When a bearing is applied so that the inner race is the rotating member and the outer race is stationary (the most common type of mounting), the inner race must be pressed or shrunk on the shaft with an interference appropriate to the application. There should be enough interference to prevent the inner race from slipping on the shaft under service conditions. If the interference is such that all radial looseness is taken out of the bearings, there will be line to line contact, with all of the ball or rollers contacting both raceways before the service load is applied. In some applications requiring great rigidity, line to line contact may be desirabe. However, in such a case, a bearing with small initial looseness would be used to avoid excessive mounting interference. If the interference is excessive, all balls and rollers may still be under load when the service load is applied, and the total load may be such as to reduce the fatigue life of the bearing. Where this condition has existed, the ball or roller path will be seen all the way around the outer raceway. However, if the outer race has slipped or crept in the housing, the diagnosis cannot be made by examination of the bearings alone. Excessive interference introduces high tensional stresses in the race, and fatigue failures may manifest themselves in the rolling surface as axial cracks that progress as flakings or as cracks. On the other hand, insufficient interference may permit the race to slip on the shaft, and result in wear and scoring. In extreme cases, the bore smears and develops rubbing cracks that eventually cause the race to break. The elastic deflexions at the contact areas between rolling elements and races in normal operations result in a minute amount of translatory motion, or sliding, in addition to the rolling. Furthermore, the weight of the cage must be carried on either the rolling elements or on some surface of the races or on a combination of these. In some types of roller bearings, a thrust component will cause the rollers to slide against a flange. Therefore, effective lubrication is needed at all times. The quantity of lubricant required at any one time is usually rather small but the supply must be of such a nature that enough is constantly available. Too little lubricant usually causes a rise in temperature, although excessive amounts can also raise the temperature because of churning. In the selection of a lubricant for an application, careful consideration must be given to several factors such as the type of bearing, degree of loading, speed, and ambient temperature. The first visible indication of trouble is usually a fine roughening, or waviness, on the surface. Later, fine cracks develop, followed by flaking. Eventually, the temperature may rise to a point where discoloration and softening of the hardened parts result. If the lubricant deteriorates, or if the quantity present gradually becomes reduced below the minimum required for effective lubrication, seizure of the rolling elements in the cage pockets and on the races will result in smearing. A lubricant with insufficient viscosity and film strength may also lead to similar failures. Where high speeds are involved, centrifugal forces become important, and the best lubricant is demanded. Inertia forces, acting on the rolling elements at high speed and with sudden starting or stopping, can result in high pressures between rolling element and cage. The rapid acceleration produced in an unloaded bearing by subjecting it to a high pressure air jet for cleaning will often develop smearing. The finely finished surfaces of ball and roller bearings, except those made of stainless steel and special materials, are readily subject to corrosion by water, acids, and other agents. However, none of these should be present in a bearing, and it is essential that the user prevent their entrance. In certain applications, such as electrical equipment, there is a possibility of electric current passing through the bearing. When the current is broken at the contact surfaces between the races and rolling elements, arcing or sparking will result. High temperature will be produced, and damage will occur at localized points. Overall damage to the bearing is obviously in proportion to the number and size of the individual points. Each time the current breaks in its passage between race and roller, a pit is produced in both members. Moderate amounts of electric pitting do not necessarily result in failure.—*R. C. Case, Bureau of Ships Journal, October 1956; Vol. 5, pp. 6-12.*

Lignum Vitæ as Bearing Material

Up to the present time, quantitative tests do not appear to have been made of the wearing properties of *lignum vitæ* when used for underwater bearings. Since a study of the limited strength data available did not reveal any significant difference in hardness between the end grain and long grain of the timber, an investigation was carried out to assess the difference in wearing properties between these two surfaces. For the experiments, two model stern bushes of the individual strip type were made up. These bushes were to scale in most respects, except length, which was deliberately shortened to increase the pressure on the bearings. Each bush was in two halves of cast bronze, into which were set bronze-retaining strips forming dovetail grooves for the wood strips, and also permitting waterways between them. The *lignum vitæ* strips were 6 inches long, $\frac{1}{2}$ inch thick and $\frac{7}{8}$ inch wide at the base, bevelled on each side at an angle of $12\frac{1}{2}$ degrees to give a bearing face approximately $\frac{5}{8}$ inch wide. Each strip was a moderately tight fit in its groove. Before being mounted in the test apparatus, each bush was bolted to a flange and the wood strips machined out to suit 4-in. diameter mild steel shafting. The bushes were then mounted in a galvanized tank, so that they could be immersed in water during the test. The general arrangement of the test apparatus is shown in Fig. 2. A 3-h.p. motor was used to drive the shafting through suitable gearing at 460 r.p.m. This speed gave a rubbing speed at the bearing surface of approximately 8ft. per sec. The temperature of the water in the tank was kept constant

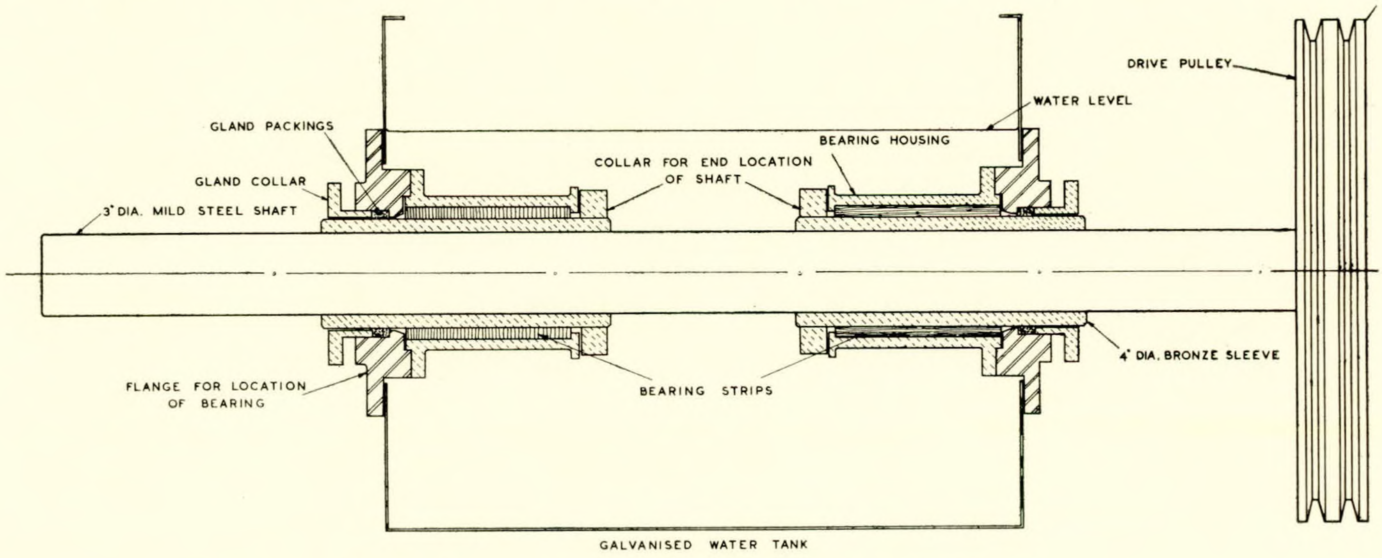
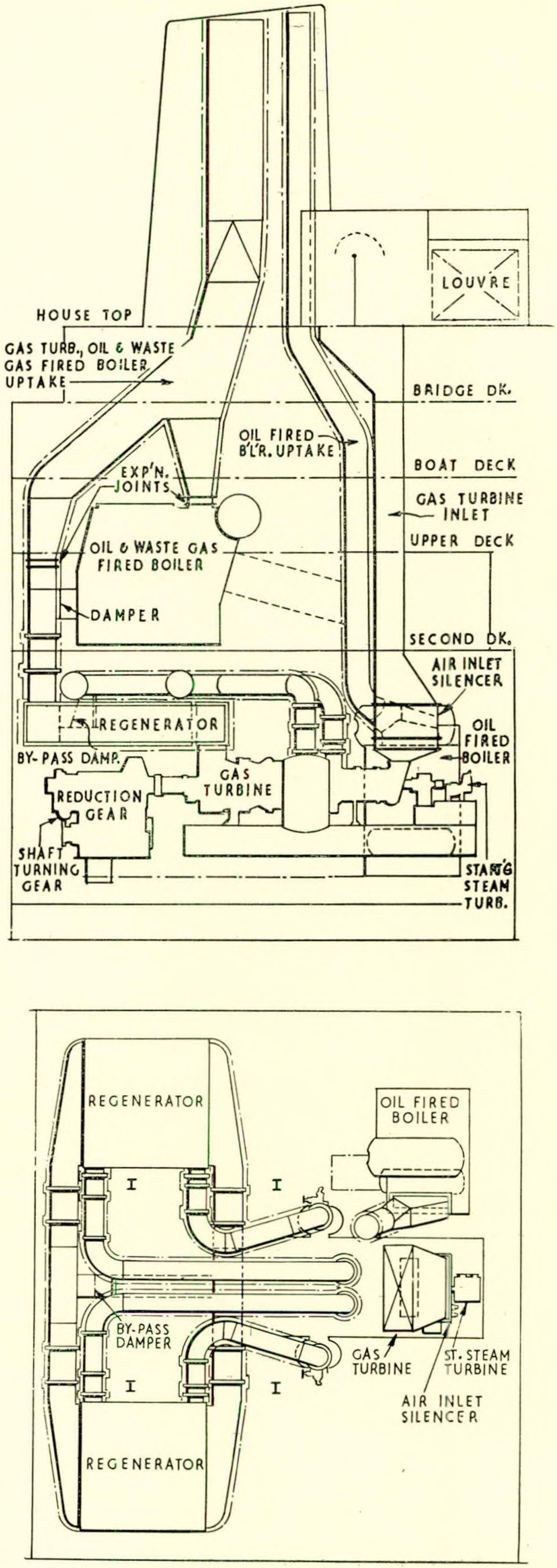


FIG. 2—*Arrangement of model stern-tube bearing apparatus*

at about 18 deg. C., by means of a cooling coil. After assembly, the tank was filled with water and the apparatus run for some hours to bed-down the bearings. The bushes were then removed for the purpose of taking initial measurements. Each test consisted of 1,000 hours' running, in about ten spells of 100 hours' continuous running. The figures obtained for wear shows that, in the most heavily loaded bearing strip, the wear of the long grain material is two to three times that of the end grain. It cannot be foreseen whether longer runs or greater loads would accentuate this difference. While there is no doubt of the superiority of end grain under the conditions of test, the general resistance to wear of both forms appears to be high. Moreover, the figures obtained for *lignum vitæ* on bronze are comparatively small, and the difference may not be significant, in view of the fairly large clearances allowed in fitting stern tube bearings in practice. The greater wear with mild steel is no doubt due to the corrosion of this metal when immersed in salt water, and is borne out by the fact that *lignum vitæ* has been found unsuitable for use with rope tackle when steel ropes are employed.—*W. T. Hide, The Shipbuilder and Marine Engine-Builder, November 1956; Vol. 63, pp. 644-645.*

American Gas Turbine Ship

The *John Sergeant* is powered by a regenerative rotary compressor open-cycle gas turbine with reduction gearing and a controllable pitch propeller. The gas turbine is similar to a commercial type developed by the American General Electric Company but has been modified slightly in order to make it more suitable for marine use. The unit is an open-cycle two-shaft machine utilizing a combustion system, a single-stage high-pressure turbine driving a multi-stage compressor and a single-stage low pressure turbine driving the propeller. Although the vessel made her voyage from America to England on Diesel fuel, Bunker C oil, suitably treated to control the sodium, vanadium and calcium content, will be burned at all times except when manœuvring is being carried out. On the voyage across the Atlantic the fuel average consumption for the whole voyage was 0·51lb./s.h.p.-hr., giving a fuel economy of about 10 to 20 per cent better than that of a C2 vessel of comparable size. The average speed was 16·819 knots. When running on Bunker C oil, the average fuel consumption has been estimated at 0·533lb./s.h.p.-hr. at an engine output of 6,600 s.h.p. It is understood that the unit can be operated for at least 15,000 hours before overhaul. Up to the time of arrival at Southampton the gas turbine had run for 341 hours. During her speed trials the gas turbine developed 7,575 h.p., giving the vessel a maximum speed of 18·3 knots. The use of a controllable pitch propeller, with which this vessel is fitted, dispenses with the necessity for reversing the gas turbine. It allows a large degree of control and enables the gas turbine to be used to the best advantage and under the most suitable conditions of speed and fuel consumption. The turbine consists of four basic components; the compressor, a combustion system, high and low pressure turbines, and the regenerator or heat exchanger. Air is compressed in the compressor and then directed to the combustion chambers where the fuel is injected. The heated gas leaving the combustion chamber is passed through the turbines to develop power. The h.p turbine operates the compressor, while the l.p. turbine produces the useful power. The exhaust from the turbines enters the heat exchanger and air, on leaving the compressor before it enters the combustion system, is passed through the heat exchanger and heated by the exhaust gases. Both h.p. and l.p. turbines are housed in the same casing but are mechanically independent. A unique feature of the unit is the variable-angle, second-stage turbine nozzle which allows a proper energy distribution between the h.p. and l.p. turbines over a wide range of ambient temperatures. This results in an improved thermal efficiency and a more rapid manœuvring ability. The gas turbine will be operated on Bunker C fuel oil, and to counteract the formation of deposits on the turbine nozzles and blading, as well as corrosion, special equipment has been installed. Sodium, calcium and vanadium are present



Elevation and plan of ductwork in the John Sergeant in widely varying amounts of residual fuel oils and, combined with the high inlet temperature, the responsible for corrosion and deposition. In addition, any contamination of the fuel oil by sea water will introduce sodium. The principal General Electric specification for fuel oils which can be burned in the gas turbine with acceptable corrosion are being adhered to in this installation and are as follows:—The ratio of the weight of sodium in the ash to the weight of vanadium in the ash should not be greater than 0·3: The ratio of the weight of magnesium to the weight of vanadium in the ash should not be less than 3·0: The sodium content of the oil should not exceed 10 p.p.m., and a value of 5 or less is preferred: The calcium content should not exceed 10 p.p.m., and a value of 5 or less is preferred. In order to meet these specifications it is usually necessary to remove the sodium. This is done by mixing intimately about 5 per cent fresh water and a demulsifying agent, such as Tretolite, with the heated oil. The water and the dissolved sodium are then removed by centrifuging. Magnesium sulphate, better known as Epsom salts, is added to the water to ensure that there is sufficient difference in specific gravity between the water and the oil to permit satisfactory centrifuging. The water washing system will also remove calcium. It has been determined that a minimum of magnesium sulphate solution (S.G. 1·005 at room temperature) is essential for the reduction of calcium. To obtain the required magnesium to vanadium ratio, a magnesium sulphate-water solution is mixed intimately with the fuel at a rate equal to 1 or 2 per cent of the fuel flow. It is injected into the fuel ahead of combustion in order to prevent settling out. The *John Sergeant* is equipped with a four-bladed, 17ft. 6in. diameter hydraulically-operated controllable-pitch propeller designed to absorb 6,000 s.h.p. when rotating at 110 r.p.m. It has been manufactured by the S. Morgan Smith Company of York, Pennsylvania. The blades are actuated by means of a connecting rod attached to a servo-motor located inboard within a section of shafting. Both the propeller and the gas turbine are controllable from the bridge as well as from the engine room, the two units being integrated so that the speed is controlled by a single lever.—*The Shipping World, 24th October 1956; Vol. 135, pp. 359-362.*

Rotating Beam Channel and 30-inch Water Tunnel

Early in 1949 the Admiralty embarked on an extensive programme of research into the behaviour of underwater weapons. Review of the then existing research facilities revealed the need for large-scale research plant to measure forces, moments, pressure distribution, shaft speeds and torques, and temperatures, and to make visual, photographic and acoustic observations on models during long runs at high speed and at varying ambient pressures. Detailed study of the requirements led to the decision that both a rotating beam channel and water tunnel would be necessary. The design and construction of these two plants were undertaken by the Ministry of Works in collaboration with the Royal Naval Scientific Service. The rotating beam channel in its domed roof building consists of an annular channel of reinforced concrete construction, of 136 feet maximum diameter, 34 feet wide and 15 feet deep, containing about one million gallons of water, over which the beam rotates on a central hub. Models which can be fitted to either arm of the beam at radii of 45, 50 and 55 feet are supported at a depth of 5 feet below the water surface. A specially designed model handling gear is provided, for fixing to the arm models, of up to 3,000lb. in weight and 21 feet long. The beam is operated from a control room having a commanding view of the annular channel area and is driven through a gearbox by a 1,510-h.p. d.c. motor with Ward Leonard control having an electronic servo holding the set speed to within ± 0·1 per cent of the maximum speed of 150ft. per sec. The various forces to which the models are subjected such as drag, centrifugal force, pressure distribution, etc., are converted into electrical signals inside the model and transmitted along the beam by cables to a number of mercury troughs, thence to the control room where the signals can be displayed and recorded. Visual and photographic observations of the models are made from two observation rooms below water level. The water is kept to a high degree of optical clarity by continuous filtration and chemical dosing. The water tunnel consists basically of a closed water circuit built of sections of large-diameter steel pipe containing some 200,000 gallons of water. The water is circulated by an axial-flow variable-pitched pump driven by an 850-h.p. d.c. motor supplied by a Ward Leonard system equipped with an electronic speed servo capable of holding the speed of the pump to within ± 0·1 per cent of the maximum speed. The upper limb of the tunnel is located on the first floor of the building which forms the tunnel laboratory and contains the working section in which models are mounted for test and observation. The working section is of novel design in that the 30-inch diameter water jet is encased in a slotted wall of perspex surrounded by an annular chamber. This permits the testing of much larger models than a conventional closed-throat type working section of the same diameter would allow with the same degree of tunnel interference. Visual and photographic observation of the model is provided by toughened-glass windows in the outer shell. The elimination of bubbles formed under cavitating conditions is effected by a device known as a "resorber". This is a part of the water circuit below ground and consists of four vertical passes having a maximum depth of 75 feet below the working section. As the water circulates through the resorber it is subjected to an additional static pressure for a sufficient period to reabsorb any air bubbles that may be present back into solution in the water.—*Paper by E. H. Lever, H. Ritter, M. Woolfson and C. T. Wright, read at a meeting of the Institution of Mechanical Engineers on 9th November 1956.*

A Concrete Floating Dock

The Seattle Division of Todd Shipyards Corporation has completed A.F.D.L.-48, the largest concrete floating dry dock ever built in the United States. It was delivered to the U.S. Navy for use at the San Diego Naval Base. The dock is 400 feet in length, by 96 feet overall width and 68 feet inside width. It has a lifting capacity of 4,000 tons and is entirely self-contained in that it can be operated without dependence upon shore power. The motor driven pumps are powered by Diesel generators mounted in machinery compartments in the wing walls. Six separate pump rooms house the dewatering pumps, each pump room so situated that it will serve four ballast compartments. In addition to the pumping machinery, pumps and flooding and discharge valves, Todd also installed fire pumps, air compressors both for starting the Diesels and for ship service, general service pumps for sanitary water and engine cooling, and a distilling unit with a capacity of 12,000 gallons per day for use at remote locations. Two travelling cranes with capacities of 5 tons at 46-feet radius and 16 tons at 16-feet radius are fitted on the walls. Accommodation for two officers and a crew of 34 men has been provided including a ward room, staterooms, crew quarters, dining rooms, galley and infirmary.—*The Marine Engineer and Naval Architect, September 1956; Vol. 79, pp. 313-314.*

Patent Specifications

Twin Screw Propulsion Unit with Fluid Operated Friction Clutches

This patent covers a marine propulsion unit of the twin-screw type with two prime movers, each coupled to a propeller through a fluid operated clutch, in which the fluid to operate each clutch is normally derived from a pump driven by the respective prime mover, and including means for starting one prime mover, as by an auxiliary starter motor, and means for starting the second prime mover indirectly from the first, including means for connecting the hydraulic fluid circuit associated with the first prime mover and its clutch to the clutch of the second prime mover so as to engage the clutch of the second prime mover so that torque is imparted to the propeller of the second prime mover by reason of the ship's motion and drives the second prime mover in a direction to cause it to start. The installation shown in Fig. 1 comprises two high powered compression ignition engines (1) and (2), each connected to a separate propeller (3) and (4), through alternative "ahead" and "astern" parallel transmission paths, each path including a separate hydraulically operated friction clutch (5, 6, 7, and 8). The pressurized operating fluid for the "ahead" and "astern" clutches of each engine is derived from a gear pump (9, 10) driven directly by the respective engine. The output side of each pump is connected to the input port of a reversing type shuttle control valve (11, 12) and is also connected through a non-return valve (13, 14) to the input port of a pilot two-way piston type selector valve (15, 16) the piston of which is mechanically controlled by a cam (17, 18) arranged to be operated manually by a member which constitutes a main "ahead-astern" manœuvring control for the particular engine. The two outlet ports of the pilot selector valves 15, 16 are connected respectively via conduits 19, 20, 21, 22 to pressure chambers at opposite ends of the shuttle valves 11, 12, while the two outlet ports of the shuttle valve are connected respectively via conduits (23, 24, 25, 26) to the hydraulic operating rams of the ahead and astern clutches (5, 6, 7, 8). The moving part of each shuttle valve (11, 12) is mechanically connected to a centralizing unit (27, 28) including two opposed springs (29, 30, 31, 32) acting on a flange member (33, 34) which acts to hold the moving part of the shuttle valve in a central position in which the supply of hydraulic fluid to both clutches 5 and 6, or 7 and 8 is cut off. Thus, when engine (1, 2) is running, and its gear pump (9, 10) is delivering hydraulic fluid under pressure to the input port of the respective pilot selector valve (15, 16), movements of this pilot selector valve into one or other of its two operative positions causes the pressure fluid to be delivered to one end or other of the shuttle valve (11, 12) and the shuttle valve jumps across to connect the operating ram of either the ahead or astern clutch (5, 6, 7, 8), as the case may be, to the source of hydraulic pressure.—*British Patent No. 763,101 issued to D. Napier and Son, Ltd. Complete specification published 5th December 1956.*

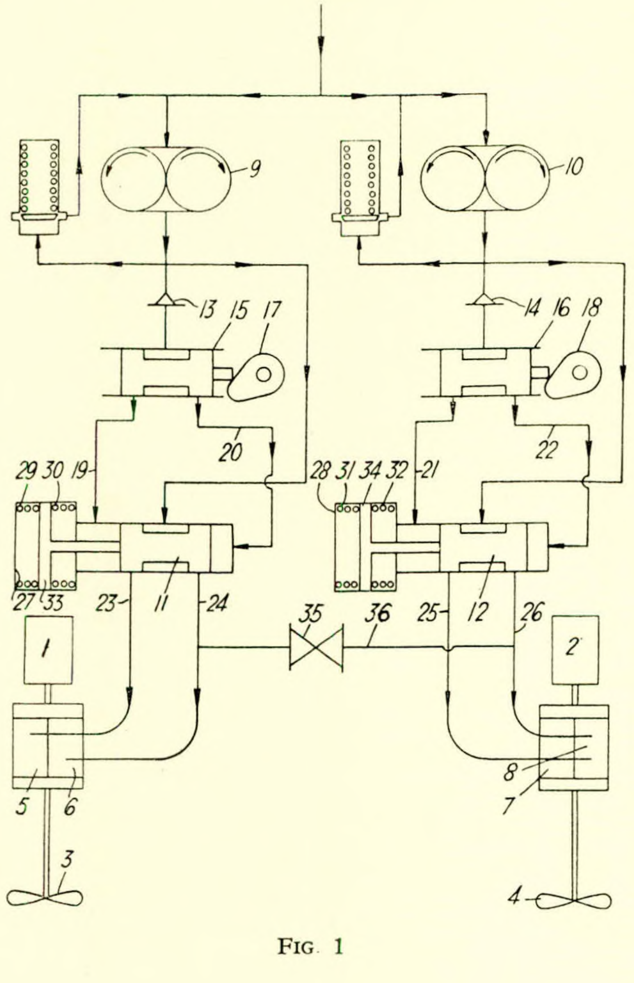
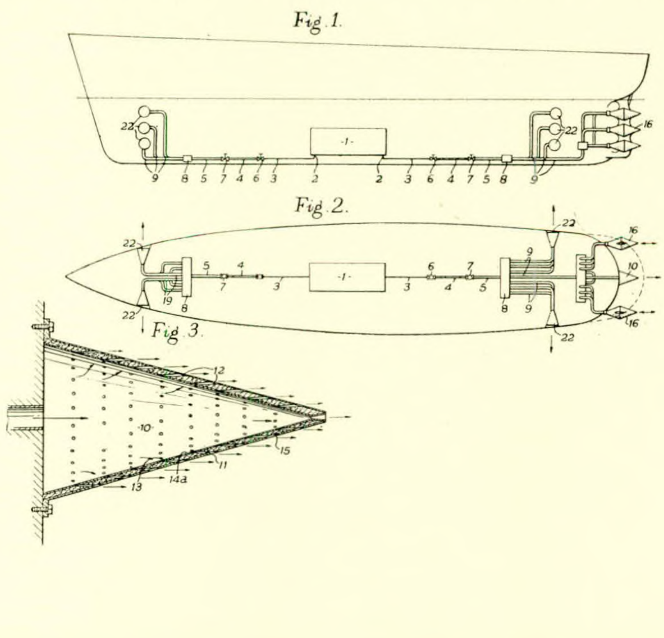


FIG. 1

Jet Propulsion Plant

This invention consists in the provision of a jet propulsion plant for ships consisting of a boiler or boilers operating in conjunction with reduction chambers and propulsion cones. In Figs. 1 and 2 is shown a boiler (1) which is similar to conventional ships' boilers in use at present with the exception that the working fluid is drawn from the bottom of the boiler (at 2); the pipes (3) conveying the working fluid are passed



through the combustion chamber of the boiler in order to obtain maximum heat. In this manner a mixture of steam and water is obtained. The boiler (1) is fed with sea water to which is added a small quantity of alkalis. The steam/water mixture is then passed through pipes (4) and (5) and a reducing valve or valves (6 and 7) to a reduction chamber (8), which is preferably of cylindrical shape. In this chamber the pressure of the mixture of steam and water is reduced. The mixture of steam and water enters at the top of the chamber and is delivered from the bottom of the chamber. The mixture is then passed to the propulsion cones (10 and 16) by piping (9). The propulsion cones are installed on the exterior of the ship's hull, below water line and as deep as possible. In Fig. 3 is shown one type of propulsion cone. This is a single cone and is provided with a series of conical holes (11), these holes being in alignment with the axis of the cone; the wide mouth (12) of the hole (11) is on the exterior surface and the narrow mouth (13) is on the interior of the cone. The cone (10) is constructed of two shells (14, 14a) and the space in between the shells is packed with insulating material (15), such as asbestos. In operation the mixture contained inside the cone escapes through the conical holes and in doing so exerts pressure against the resistance of the surrounding water and thereby propels the ship forward.—*British Patent No. 762,325, issued to D. K. Mountanos. Complete specification published 28th November 1956.*

Jet Propulsion Unit

According to this invention a hydraulic jet propulsion unit for water craft consists of a free piston driving motor and a double-acting piston pump with nozzle-like shaped pistons. The free piston motor is built as a double-acting motor with two pistons, one travelling in opposite direction to the other. The piston pump has a number of pistons travelling in opposite directions, one half of which number is rigidly connected with one driving piston. The other half is connected with the other driving piston so that two piston systems are provided which reciprocate in opposite direction. Fig. 1 is an axial section of the jet-propulsion unit, in which the operating engine is a double-acting two-stroke internal combustion engine having oppositely moving reciprocating pistons and two piston compressors for the combustion air. The water pump is arranged

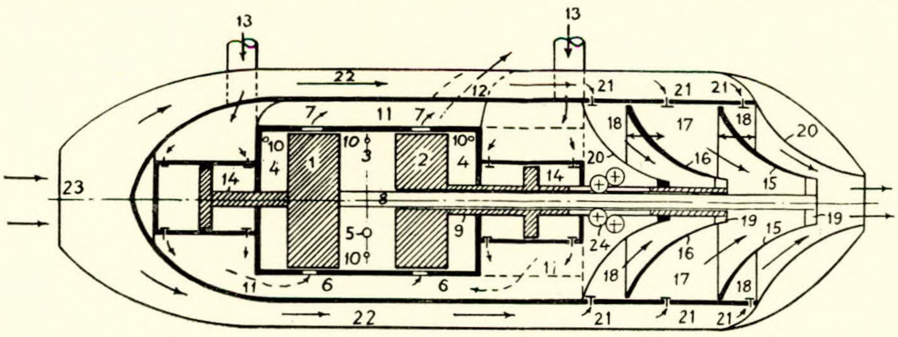


FIG. 1

to the rear of the engine and has pump pistons formed as nozzles, which likewise move in opposite direction to each other.—*(British Patent No. 759,630, issued to J. Endres. Complete specification published 24th October 1956.)*

Improved Costa Bulb

It has hitherto been believed that best results will be obtained with a Costa bulb if the headpiece of the bulb is arranged at a more or less great distance from the cap like part of the propelling screw. It was believed that the medium leaving the hub zone should carry out first a certain twist in order that it may thereafter be drawn off somewhat accelerated along the bulb shaped part. This invention is based on the realization that the effect of the bulb shaped part in a vessel in motion may in all respects be improved further if the bulb

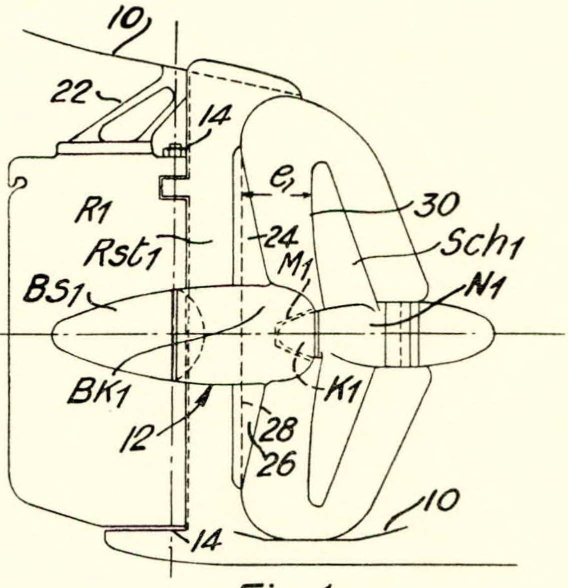


Fig. 1.

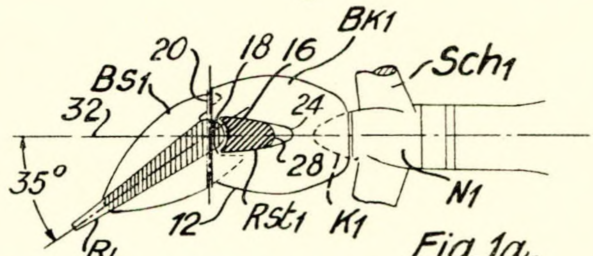


Fig. 1a.

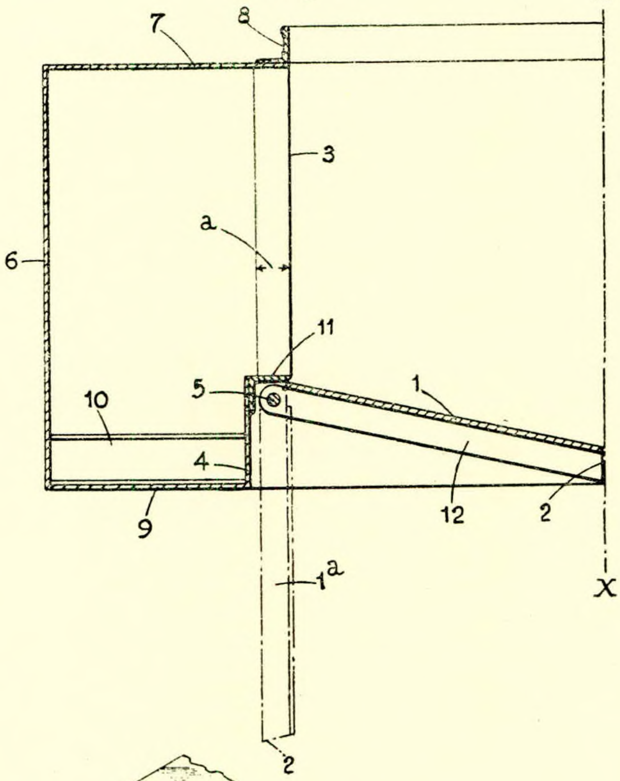
shaped part is pushed up to the propelling screw. In Figs. 1 and 1a the propeller (Sch_1) projects from the hull (10) towards the rear and is driven by the engine (not shown) of the vessel. The bulb (12), comprising the front part or head piece (Bk_1) and the rear part (Bs_1), does not participate in the rotation of the propeller (Sch_1). However, the distance (e) between the forward edge (28) of the rudder post (Rst_1) and the trailing edge (30) of the blades of the propeller is reduced by arranging the cap like part (K_1) in the recess (M_1) of the head piece (Bk_1) of the bulb (12) so that the wake caused by the propeller (Sch_1) is smoothed by arranging the bulb shaped part (12) directly behind the propeller. By this means it is obtained that the vortices produced by the turning blades of the propeller, and consisting mainly of air and turbulent water, will be led undisturbed backwards along the surface of the bulb (12). The bulb (12) and the propeller screw are separate parts, the bulb (12) not being supported by the propeller and not participating in its rotation. The rear part (Bs_1) of the bulb is rigidly connected with the rudder blade

(R_1) and angularly displaceable therewith with respect to the head piece (Bk_1) rigidly connected to the rudder post (Rst_1). The rudder blade (R_1) is shown in Fig. 1a in the extreme position at an angle of about 35 degrees to the median 32, the rudder blade (R_1) being swingable by the connecting link (22). The bulb (12) has a smaller diameter than the propeller, and the largest diameter of the bulb is approximately in the plane of the leading edge (28) of the rudder post (Rst_1).—*British Patent No. 762,445, issued to L. Costa and E. R. F. Maier. Complete specification issued 28th November 1956.*

Drop Bottom Barge

Drop-bottom barges which are employed for conveying and depositing dredgings, ballast, "overburden", and like material are designed with a deep walled interior which forms a hopper for containing a considerable quantity of the material, but due to the slope of the walls converging to the bottom doors there is a tendency for the material to cling to the walls and for stones and similar objects to get between the hopper walls and the hinge of the respective doors so as to obstruct or hinder the operation of the doors, particularly when closing. The object of the invention is to provide an improved drop-bottom hopper barge which minimizes these undesirable characteristics and achieves a design which is efficient and reliable for use with ballast and like material. Referring to the drawings (Figs. 1 and 2), the line x—x represents the longitudinal central section line of a barge of which the right-hand side has been omitted for convenience. The barge bottom includes two hinged doors, one of which (1) is shown, the other being similar and disposed on the opposite side of central line x—x and facing the one illustrated. The door (1) drops open in the nature of a flap as shown in broken lines (at 1a), and when closed is held in a horizontal or slightly downwardly inclined position as illustrated in Fig. 1 with respective edges (2) of the two doors, which are remote from the hinges, in meeting or interlocking engagement. In such a preferred embodiment the hopper has at least two opposite walls (3) vertically arranged and each with a recess (4) at the bottom end in which the hinge (5) of the respective door (1) is accommodated. The barge is diagrammatically represented by the outer skin (6), the deck (7), the coaming (8), the bottom (9) and the floor frame (10). The walls (3) of the hopper are vertically arranged so that when the doors are opened to deposit a load of ballast, any tendency for this material to cling to the walls is minimized and lodgement in the hinge space at (5) is prevented or minimized.—*British Patent No. 760,841, issued to E. C. Jones and Son (Brentford), Ltd. Complete specification published 7th November 1956.*

Fig.1.



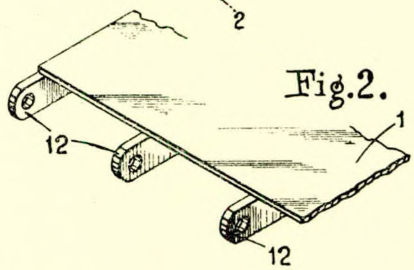


Fig.2.

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Marine Engineering and Shipbuilding Abstracts

Volume XX, No. 3, March 1957

* Patent Specifications

Launch with Opposed Piston Engine

The Rootes-Lister 77, which is powered by a marine version of the Rootes opposed-piston two-stroke cycle Diesel engine. The craft is the first marine application of this unusual engine which was introduced as a road vehicle power unit just over four years ago. The launch has a Deborine fibre-glass hull and was designed and constructed by Halmatic, Ltd., of Portsmouth. The engine has the type designation Rootes-Lister TSM3GR and provides a high power/weight ratio, the net weight, including gearbox, being only 1,585lb. The cylinders have a bore of 3¼ inches and each piston has a stroke of 4 inches, giving a swept volume of 199 cu. in. The marine rating is 77 b.h.p. at 1,800 r.p.m. on a 12-hr. basis and the fuel consumption is given as 0·38lb. per b.h.p. hr. at full load. The engine, moreover, is comparatively compact, the overall height being only 33½ inches, which allows it to be considered for installation under the cockpit floor of a craft where conventional in-line type engines would be too high. Other engine design features include direct injection and through scavenging on the Kadenacy principle. This latter feature, of course, depends upon a rapid early opening of the exhaust valves—which in the case of the Rootes engine are a series of circumferential ports—and the rapid discharge of the burnt gases to the exhaust in the form of a "slug" giving rise to the term "ballastic discharge". In this manner, a partial vacuum is created in the cylinder so that when the air port is opened, there is complete through scavenging of the combustion spaces. The disadvantages of the normal two-stroke engine with valves as compared with the Rootes-Lister design are that the former requires a camshaft to run at engine speed, i.e. twice the speed of a four-stroke type and, as a result, the valve springs have to be stronger to operate at this speed. The valve, moreover, must have a large area to allow rapid passage of the exhaust gases and, opening early, they do so against the high back pressure still remaining in the cylinders. All these points necessitate a heavy camshaft drive and severe mechanical losses. The Rootes-Lister design has no camshaft, the horizontal piston arrangement controlling both inlet and exhaust ports at opposite ends of the cylinder barrels and the ports are so arranged that the exhaust opens in advance of the inlet. In addition to the ballastic discharge effect of the exhaust gases, a Rootes-type blower mounted on the front end cover provides a low pressure positive through scavenge, the efficiency of which is emphasized by the comparatively low fuel consumption. The six pistons are opposed in pairs in three cylinders and there are none of the conventional cylinder heads, gaskets or valve gear to require maintenance or replacement. Ease of maintenance is, indeed, a strong feature of the engine, for example, decarbonizing at the exhaust ports is carried out simply by removing one exhaust manifold branch placed conveniently at the top of the engine. Quiet running and the absence of vibration during trial runs were noticeable. The first is achieved by good combustion arrangements which reduce knock to negligible proportions and is aided by the specially tuned air intake silencers. Vibration is eliminated by the balanced equal-and-opposite action of the pistons and their linkage working on the centrally positioned crankshaft. By dispensing with cylinder heads, heat losses are reduced and the improved thermal efficiency leads to greatly improved combustion and easy starting. A 12-volt electrical system is adequate for all normal temperatures. The engine in the Rootes-Lister 77 drives through an oil operated reverse and 2 to 1 reduction gear, made by the Self-Changing Gear Company, and the gearbox is fitted with sequent control which allows control through a single lever with finger-tip operation—*Gas and Oil Power, August 1956; Vol. 51, pp. 181-182.*

New Pye Underwater TV Camera

Pye Ltd. have produced a new hand-held underwater television camera which will enable divers to be effectively supervised from above water. It will be the smallest and cheapest underwater TV camera yet to be produced. A free-swimming non-suited diver carrying his own self-contained breathing apparatus is able to perform underwater operations far more efficiently than the heavily-clad, slow-moving conventional diver. With practically unrestricted movement a free-swimming diver is independent and necessarily cut off from the ship. This method presents certain difficulties in that personnel on board ship are unable to tell what has happened under water until it is related to them by the diver. Consequently, incorrect information given by the diver and mis-direction by officials may result in absolute confusion. Time lost by this method of relaying information is an important factor and undoubtedly a large amount of money is at present being wasted. With the new camera, instead of having to rely on a diver's report, a number of expert observers may view the underwater scene displayed on large-screen picture monitors. A record of the picture reproduced on the monitor screen is easily made by photographic means, which obviates the difficulty of taking photographs under water. With accurate visual information available on the display monitor, it is possible for expert direction to be given to the diver and an accessory for this equipment, the underwater loudspeaker, provides a simple method. Two divers, one holding the camera and the other concentrating on a complex project, may be directed by observers above water to derive the utmost advantage from favourable weather conditions. The camera is intended for operation down to 250 feet, but to provide an adequate safety margin the container has been designed to withstand a water pressure of 220lb. per sq. in., corresponding to a depth of 500 feet. The unit is buoyant in water and weighs 38lb. in air.—*The Marine Engineer and Naval Architect, 1st September 1956; Vol. 79, p. 324.*

Norwegian Nuclear Tanker

A Dutch-Norwegian research group at Kjeller in Norway, has been investigating for some time the possibility of constructing a large nuclear powered tanker. The scheme is now approaching fruition and there is planned an 18-knot, 32,000 tons d.w. tanker which would probably be built at the Rosenberg yard, Stavanger. The De Laval Turbine Company, Stockholm, the General Dynamic Corporation in America and the Bergesen group (owners of this yard) are said to be interested in the project. The propulsion set is based on a heavy water moderated and water cooled reactor of 64 MW heat output. The plant and protection would weigh 1,000 tons but would be able to operate for four years, 6,000 MW-days, without refuelling. The 15 tons of uranium necessary would compare against the many thousands of tons of bunkers for an orthodox ship.—*The Marine Engineer and Naval Architect, September 1956; Vol. 79, p. 316.*

Performance of Largest Motor Tanker

Owned by Fearnley and Eger, Oslo, the *Ferncrest* is of 34,800 tons d.w.c., and is propelled by an Eriksberg-B. and W. turbocharged two-stroke engine of the poppet valve type which develops 12,500 b.h.p. (13,900 i.h.p.) at 115 r.p.m. The ten cylinders have a bore of 740 mm., and a piston stroke of 1,600 mm. There are three exhaust gas driven turboblowers, all of Rateau manufacture. Before considering the fuel consumption figures it should be noted that there are two 330-kW Diesel generators, also a turbine driven generator with an output of 220 kW. The latter is driven at sea by steam generated by an exhaust gas boiler with a heating surface of 350 sq. m. which also supplies steam for oil and accommodation heating, etc. In addition, there are three three-furnace Scotch boilers, each with a heating surface of 250 sq. m. which are required for cargo pumping and heating. On leaving Gothenburg on January 19th, Diesel oil only was used for the main engine, but since July 12th, the engine has run on heavy oil. This was bunkered at Mena al Ahmadi and has a specific gravity of 0·94. It is as yet too early to give details of the fuel consumption on this oil, but in any case figures for the latest voyage would be of little value as the ship was running a much reduced output owing to the weather conditions which have prevailed. Generally, the engine output has been restricted to some 11,500 i.h.p. and the average daily mileage has been some 364·6—a speed of about 15·2 knots. The mean fuel consumption per day—in ballast or loaded condition—is 38 tons of Diesel oil for the main engine only. With the Diesel driven generators in operation, there is an added daily consumption of 1·9 tons, bringing the total fuel consumption to about 40 tons. These totals are equivalent to a specific fuel consumption of 0·34lb. per s.h.p. hr. for propulsion and under 0·36lb. per s.h.p. hr. for all purposes. However, with steam being generated by the exhaust gas boiler and with the turbo-generator in operation, the fuel consumption is reduced by at least 0·9 tons—to some 39 tons. Now that the engine is running on boiler oil, and with the present price differential between that and Diesel oil being 80s. 6d. per ton, there is a saving in the fuel bill of nearly £160 per day, or assuming 300 days per annum operation, of £48,000. These figures do not take into account, however, the increase in consumption to be encountered on the heavy oil of lower calorific value.—*The Motor Ship, September 1956; Vol. 37, p. 198.*

German Motor Trawlers

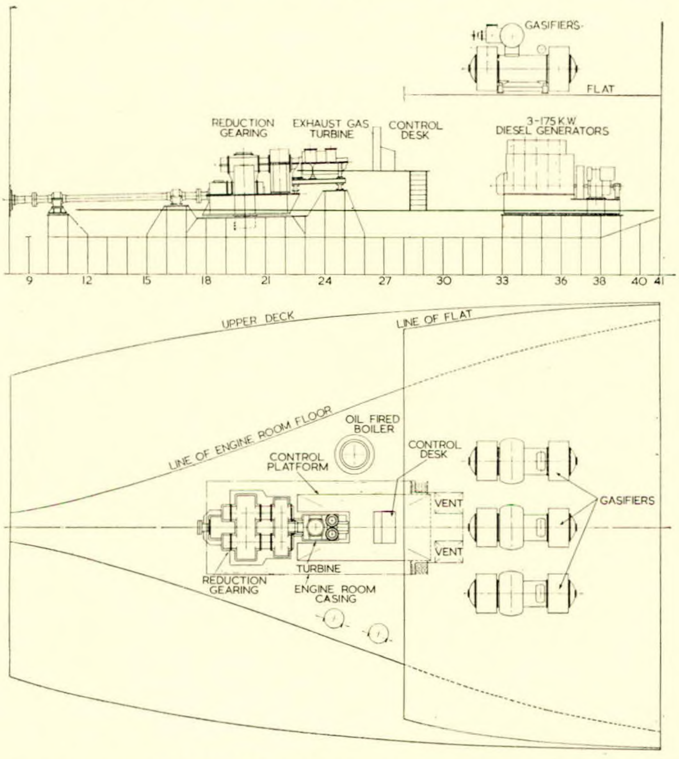
The highest powered German motor trawler has been launched by H. C. Stülcken Sohn for Cranzer Fischdampfer A.G. She is named *Zephyros* and will be driven by three six-cylinder Maybach Diesel engines having an output of 600 b.h.p. each at 1,500 r.p.m. and driving an A.E.G. generator. These generators supply current to two A.E.G. propelling motors of 720 b.h.p., which drive a single propeller through reduction gear. The speed of the vessel will be 14 knots. She will have a crew of 30 and a gross register of 650 tons. The ship is 208ft. 8in. overall and 182ft. 5in. b.p. and has a beam of 29ft. 10¼in. The draught is 14ft. 1¼in. In addition to the usual fish holds, the vessel will have a deep-freeze hold with a capacity of 1,313·7 cu. ft. and will also be equipped with a fish meal and liver oil plant. At present, two more vessels are being built at German yards, to be driven by the same type of high-speed Maybach engines, namely, a train ferry for the German State Railways at Kieler Howaldtswerke (6,000 b.h.p., 17 knots) and a coastal passenger vessel for the Hamburg-Heligoland run at Norderwerft (6,000 b.h.p., 19 knots).—*The Motor Ship, September 1956; Vol. 37, p. 233.*

Details of the New French Liner

The new C.G.T. liner will be called *France.* It will have a length of about 985 feet, a beam of about 108 feet and a depth of 79 feet. The gross tonnage will be 38,000 tons and the displacement 55,000 at a mean draught of 33 feet. There will be nine decks and 14 watertight bulkheads in order to conform to the requirements of the 1948 Convention, particularly those concerning stability after damage. The machinery will consist of quadruple-screw geared turbines developing a total of 150,000 s.h.p. Each shaft will be driven by a three-casing set of C.E.M.-Parsons-Penhöet single reduction turbines taking steam from eight boilers at 1,000lb. per sq. in. and 900 deg. F. The turbines and their boilers will be arranged in two "units" similar to the disposition employed in the latest naval vessels, so that underwater damage amidships will not deprive the vessel of propulsion. The steam conditions have been chosen in order to realize the maximum economy and result from experience obtained from the *Flandre* and *Ville de Tunis* class of ships. All water will be distilled on board in four groups of low pressure evaporators, having an output of 300 tons per day each. These evaporators will be integrated in the steam cycle. Generator capacity totalling 12,000 kW will be provided in two widely separated compartments. Generation will be at 440 volts 60 cycles. Two pairs of Denny-Brown ship stabilizers will be fitted, following upon the success of this equipment in the *Queen Elizabeth.* As opposed to the *Normandie,* there will be only two classes and the theme will be one of less luxury and more comfort. Over 2,000 passengers and 8,600 tons of fuel will be carried.—*La Revue Nautique, August 1956. The Marine Engineer and Naval Architect, September 1956; Vol. 79, p. 317.*

Ore Carrier with Free-piston Machinery

An ore carrier of about 9,400 tons d.w.c., to be propelled by free-piston gas turbine machinery, has been ordered from Lithgows, Ltd., by Scottish Ore Carriers, Ltd. (managers, J. and J. Denholm, Ltd.). This vessel is to have a length b.p. of 407 feet, a moulded breadth of 57 feet and a depth to the upper deck of 32ft. 3in. The principal engineering contractors are Rankin and Blackmore, Ltd., and the three free-piston gasifiers will be of the GS. 34 type, each of 1,000 s.h.p., and built by the Free Piston Engine Co., Ltd. The turbine will be constructed by Rankin and Blackmore, Ltd., to the designs of Power Jets (Research and Development), Ltd., and will give 2,500 s.h.p. in service. It will have six rows of ahead blading and two rows of astern blading. This unit will



Arrangement of 2,500-b.h.p. free-piston machinery installation in a 9,400-ton ore carrier

transmit through articulated, locked train double-reduction gearing, built by the Fairfield S.B. Co., Ltd., giving a final propeller speed of 115 r.p.m. It is stated that the gasifiers will be designed to start on Diesel oil but run on boiler oil. For the sake of simplicity at this stage, it has been decided not to drive the electric generators by exhaust-gas turbines supplied by any of the gasifiers.—*The Motor Ship, November 1956; Vol. 37, p. 307.*

German Standard Motor Tramps

The 15,145 tons deadweight cargo motorship *Hadjitsakos* was recently delivered from the A.G. Weser shipyard, Bremen, to the Ocean Cargo Line Ltd., of Monrovia, one of the Livanos companies. The *Hadjitsakos* is the first of a group of eight similar ships, ordered by different owners and is the prototype of a new standard design developed by the A.G. Weser. The ship has been constructed to the highest class of Lloyd's Register of Shipping, and is of full scantling type so that she can operate either as a closed or open shelterdecker. She has the following principal particulars:—

Length overall	521ft. 10in.
Length between perpendiculars...	485ft. 0in.
Moulded breadth...	65ft. 3in.
Moulded depth to upper deck ...	41ft. 6in.
Moulded depth to second deck ...	32ft. 6in.
Deadweight capacity as closed shelterdecker on 30ft. 4$\frac{5}{16}$in. draught, tons	15,145
Deadweight capacity as open shelterdecker on 27ft. 8$\frac{5}{16}$in. draught, tons	13,078
Measurements as closed shelterdecker:—	
Gross, tons	10,583
Net, tons	6,363
Grain capacity, cu. ft.	818,296
Bale capacity, cu. ft.	758,679

External features are a short forecastle, a well raked, slightly curved stem and a cruiser stern. The double bottom is arranged for the stowage of water ballast, fuel oil and fresh water. Fuel oil can also be carried in the engine room port and starboard wing bunkers. The streamlined rudder is operated by Donkin electro-hydraulic four-ram steering gear. Two 28-ft. lifeboats (one fitted with a Diesel engine), a cutter and a gig are carried under gravity type davits. The *Hadjitsakos* is propelled by a six-cylinder Hawthorn-Doxford engine of the 670-mm. bore by 2,320 mm. stroke type. The output of the engine is 6,000 h.p. at 115 r.p.m., which is sufficient for a service speed of 14½ knots. The four-bladed, solid manganese bronze Zeise propeller has a diameter of 5,355 mm. and a pitch of 4,240 mm. A speed of over 16 knots was easily obtained on trials, with the ship in ballast condition.—*The Marine Engineer and Naval Architect, August 1956; Vol. 79, pp. 273-274.*

Engines-aft East-Asiatic Liner Launched

Burmeister and Wain have launched the 10,200 tons d.w. cargo motorship *Bogota* for the East-Asiatic Company, Copenhagen. She is the first of a new series of liners in which the propelling machinery is arranged aft. The main particulars of the vessel are:—

Length between perpendiculars...	455ft. 2in.
Moulded breadth...	62ft. 6in.
Depth to upper deck	38ft. 2in.
Depth to second deck	29ft. 10in.
Draught, approximately... ...	27ft. 6in.
Speed on loaded trials	17½ knots

There are five holds, all forward of the engine room; the after part of No. 2 and the forward part of No. 3 holds consists of deep tanks for liquid cargo. No. 5 upper 'tweendecks are arranged as two refrigerated cargo rooms with a total capacity of 8,200 cu. ft. Two bipod masts and 11 kingposts carry ten 3-ton and ten 5-ton derricks. There is a 20-ton heavy derrick on the foremast and a 60-ton derrick on the mainmast. The superstructure amidships contains rooms for the deck officers and two owner's cabins. The aft superstructure contain quarters for the crew, provisions room, etc., on the upper deck, rooms for assistants and petty officers, galley, officers' and crew's cafeteria and the recreation room on the poop deck. Accommodation for the chief and assistant engineers, apprentices and the officers' smoke room are in the deckhouse above. The main engine is an eight-cylinder single-acting two-stroke unit of the 874-VTBF-160 type with cylinders 740 mm. bore by 1,600 mm. stroke, developing 10,000 b.h.p. at 115 r.p.m. The auxiliaries comprise two three-cylinder and two five-cylinder four-stroke B. and W. trunk Diesel engines with a bore of 245 mm. and a stroke of 400 mm. These will be arranged on flats in the wings of the engine room.—*The Marine Engineer and Naval Architect, September 1956; Vol. 79, pp. 315-316.*

Model and Ship Trials in Shallow Water

This paper presents the results of an extensive research into model test technique and the extrapolation problem in shallow water. From the research described the following conclusions are drawn: Concerning the extrapolation problem—for the prediction of full-scale trial performance in shallow water from model tests, a method of extrapolation is necessary in which the frictional drag of the ship form is determined by the non-wave making part of the resistance curve. Concerning the wall effect in a model basin having restricted water depths—resistance tests made with models according to the method of images give good results for the determination of wall effect in the model basin. A three-dimensional extrapolator makes it possible to divide this wall effect into two parts; viz. the influence on frictional resistance and the influence on wave making resistance. A comparison of the results of model tests for the determination of wall effect in shallow water and the application of the Kreitner equation given by Schuster show a satisfactory agreement for the usual values of the restricted depths of water. Concerning the effect of slots between the walls and an adjustable bottom: A comparison of the results of model tests in shallow water in a basin with a pressure-tight concrete bottom and a basin with an adjustable bottom shows that the influence of the slots between walls and adjustable bottom cannot be accounted for by a virtual increase of the tank width only, but that there is also a virtual increase in water depth. Therefore, any research concerned with the correlation of the results of model tests and ship trials in shallow water must be carried out in a pressure-tight model basin. However, the determination of the optimum hull form and the best type of propulsion for shallow water is still possible from comparative tests in a tank with adjustable bottom. Concerning the correlation problem—the results of propulsion tests with models in a pressure-tight model basin at a restricted depth of water and the results of the corresponding full-scale trials in still water show a satisfactory correlation if a three-dimensional extrapolator and a wall effect correction are applied. The correlation of the results of model tests in shallow water and full-scale trials in a river is hampered by the difficulties in determining ship speed and water depth.—*Paper by J. D. van Manen and W. P. A. van Lammeren, read at the Annual Meeting of the Society of Naval Architects and Marine Engineers, 15th November 1956.*

Resistance of Trawler Hull Forms

This paper covers the development of a fishing-trawler hull of 0·65 prismatic coefficient and 300 displacement-length ratio to serve as a parent form for future series investigations. An average American trawler hull was first designed and tested. Three variations, covering the effects of water plane coefficient, midship section coefficient and shape of section area curve, were investigated to find desirable values from a smooth water resistance standpoint and the indicated changes incorporated into two new hull designs. The better of the two proved to have almost 12 per cent less resistance than the first form tried. Since these forms are extremely susceptible to laminar flow, an extensive investigation of the turbulence stimulation problem became necessary. An appendix describes the development of the stimulators used in these tests.—*Paper by C. Ridgely-Nevitt, read at the Annual Meeting of the Society of Naval Architects and Marine Engineers, 15th November 1956.*

Propagation of Fatigue Cracks

The growth of a fatigue crack is visualized as follows: If the crack were in a purely elastic solid then the applied stress would generate a high stress at the tip of the crack, as a result of the large stress concentration there. In a metal this high stress would be above the yield stress and so a plastic region would be created ahead of the tip of the crack. This has three consequences: (*a*) As the plastic region yields, it work hardens. (*b*) The plastic region is constrained by its elastic surroundings which, as yielding occurs, take up some of the load, thus reducing the stress in the plastic region. (*c*) This increase in stress in the surrounding regions will cause some parts, which in the purely elastic case carried a stress less than the yield stress, to be subjected to a large enough stress to become plastic. On reversing the applied stress, reverse plastic flow will occur, again governed by the three effects mentioned. Continual application of an alternating stress will cause a continual work hardening at the tip of the crack, leading to one or other of the following possibilities: 1. If the peak stress at the tip of the crack in an elastic medium is less than the fracture stress then the work hardening will tend to a limit after which the applied stress causes elastic deformation only. In this case the crack does not spread 2. If this peak stress under purely elastic deformation is greater than the fracture stress, then it is impossible for the material to work-harden sufficiently to produce a purely elastic state. Fracture in the neighbourhood of the tip of the crack will occur after a number of cycles. The crack then starts to spread with the same process being repeated at the new tip of the crack. Consider the history of a small volume of metal in the line of advance of the crack. When the crack is far away the stress in the volume is essentially equal to the applied stress and the deformation is elastic. As the crack approaches, the stress rises. When it rises to the initial yield stress the volume becomes plastic and then describes a hysteresis loop, gradually work hardening. When its ductility is exhausted, it fractures and becomes part of the crack.—*A. K. Head, Journal of Applied Mechanics, September 1956; Vol. 23, pp. 407-410.*

Ship Steels and Their Relation to Ship Failures

A combination of three conditions is required for brittle fractures of ships in service: (*a*) The existence of stresses from all sources probably in excess of about 10,000lb. per sq. in. (*b*) The presence of a severe notch introduced by unsatisfactory design details or poor welding practices. (*c*) The use of ship steel with low notch toughness at the operating temperature. It is not feasible to operate ships at stress levels from all sources that are low enough to eliminate failures. A substantial reduction in failure rate has been achieved through elimination of severe notches by improvements in design details and in fabrication practice, but complete elimination of failures by flawless design and construction does not appear to be feasible. It appears that brittle fractures in ships would be virtually eliminated by utilizing steel with a mean Charpy V-notch 15ft.-lb. transition temperature of 10 deg. F. and, at the same time, using good designs and fabrication techniques. Ships built of plate with the low notch toughness of World War II steel will almost certainly be subject to failure due to brittle behaviour (particularly if operated in a future emergency) unless there is extensive use of steel of greater notch toughness for the plating in critical locations or operational use of the ships only in tropic waters. Ships built of steel made to American Bureau of Shipping specifications ABS-A, ABS-B and ABS-C should be virtually free from service failure due to brittle behaviour, even if operated in a future emergency involving a 10 deg. F. lower operating temperature than during World War II, except that some fractures may occur in the sections made of the ABS-B grade produced from 1947 to 1/31/56. Under conditions of a future emergency involving large-scale ship construction programmes, ships' steel will probably be restricted to semikilled grades. This will necessitate using a semikilled alternative for plates over one inch in thickness where killed steel made in conformance with ABS-C now applies. A semikilled grade containing 0·2 per cent maximum carbon and 1·00 per cent to 1·35 per cent manganese appears to be a satisfactory alternative for ABS-C steel in plates over 1 inch and at least up to 1½ inches in thickness. This steel has about the greatest notch toughness that can be achieved in an as-rolled, semikilled grade made to current practice and to strength levels now specified. Although completed research points to a solution to the brittle failure problem in ship steels at current strength levels, there are important investigations in progress that should be continued. Consideration of the relation between mill practice and notch toughness may suggest other ways of achieving improvements. Basic studies of the influence of metallurgical structures on the mechanics of fracture can be expected to contribute valuable information. Studies of the notch toughness of ship steels at levels of strength other than those now in use may be necessary in anticipation of the use of such materials.—*W. J. Harris and C. Williams, Ship Structure Committee, Report SSC-80, 1956.*

German Built Cargo Liner for British Owners

The *Newcastle Star* is propelled by an 11,250-b.h.p. Bremer Vulkan-M.A.N. ten-cylinder engine—the highest-powered German-built M.A.N. unit yet completed—with a particularly interesting pressure-charging arrangement. Of the single-acting two-stroke type, the engine is designed so that the scavenging air is delivered by three exhaust-gas turbo-blowers to a first-stage belt. The lower sides of the working pistons operate in parallel with a crankshaft-driven Roots-type blower, and draw air from the first-stage section; it is delivered to a second-stage belt which supplies the working cylinders with the supercharged air. Turbocharging of the M.A.N. engine can be carried out either on (1) the constant-pressure system, or (2) the impulse system, the former normally requiring a scavenging air pump and entailing some slight increase in fuel consumption. As, however, the greater part of the second stage scavenging air for this type of M.A.N. engine is delivered from the undersides of the working pistons, the power absorbed by the scavenging pump supplying the small amount of second-stage air is not heavy. With the impulse system, however, apart from the lower fuel consumption, the construction of the engines is claimed to be simpler. The Bremer Vulkan-M.A.N. engines utilize the constant-pressure system but have the Roots-type blower in place of the double-acting scavenging air pump fitted at the forward end of the engine as in the case of the *Sagami Maru*. This Japanese-built engine also had ten cylinders with a bore of 780 mm., and is stated to have an output of 12,000 b.h.p. at 118 r.p.m. As in the case of the Bremer Vulkan-M.A.N. engine, the undersides of the working pistons act as air pumps. In that instance up to an output of 90 per cent of the figure mentioned—10,800 b.h.p.—the cylinders are scavenged only by the underpistons and the double-acting pumps but, if a higher output is desired, an independent electrically driven blower is brought into action to operate in parallel with the pumps. The *Newcastle Star* is of the open shelterdeck type built under Lloyd's special survey to Class + A1 + LMC + RMC and is designed for a mixed trade of refrigerated cargoes of frozen, chilled fruit and dairy products, with general cargoes of wool, etc. The principal characteristics of this class of ship are as follows: —

Length o.a.	158·317 m. (519ft. $5\frac{1}{16}$in.)
Length b.p.	143·253 m. (470ft. 0in.)
Breadth, moulded	21·336 m. (70ft. 0in.)
Depth, moulded to upper (2nd) deck	10·592 m. (34ft. 9in.)
Depth, moulded to shelter (1st) deck	13·945 m. (45ft. 9in.)
Draught, summer freeboard ...	9·137 m. (29ft. $11\frac{3}{4}$in.)
Corresponding deadweight ...	12,150 tons
Capacity of refrigerated spaces	326,010 cu. ft.
Total general cargo capacity (Grain)	11,242 m.^{3} (397,030ft.^{3})
(Bale)	9,885·7 m.^{3} (349,120ft.^{3})

Essentially the engine differs but little from the standard M.A.N. design except for the turbocharging, and for the fact that the bedplate is wider than usual and that it and the frames are of cast iron construction. There are ten cylinders with a bore of 780 mm. and a piston stroke of 1,400 mm., the rated output at 115 r.p.m. being 11,250 b.h.p. There are three Brown Boveri turboblowers, type VTR 630, designed for a maximum speed of 7,500 r.p.m. and a temperature of 660 deg. C. They draw air through filtered air inlets at starboard side of the engine room, below the blowers themselves and deliver to the first-stage manifold at a pressure of about 4lb. per sq. in. From there the air is pumped via the Roots-type blower, which is driven through a hydraulic coupling and by chain of the crankshaft, to the second-stage manifold, the pressure then being 5-5·8lb. per sq. in. The undersides of the working pistons act as air pumps and they operate in parallel with the chain-driven blower. A rotary exhaust slide valve is fitted to each cylinder, giving the usual post-charging effect. They are actuated by roller chain from the manœuvring shaft and, as a safety measure to prevent possible jamming by broken piston rings, the shaft drive is carried by two pins which are designed to fracture and stop the shaft in the event of jamming of an exhaust valve. The pistons are constructed in two parts, the upper unit, of cast steel, being water cooled and secured to the upper flange of the piston rod. The lower part, of cast iron, is attached to the lower flange of the 230-mm. diameter piston rod and works as an air pump. Each combustion space has a special Bosch-type fuel injection pump, double filters being fitted in the supply line to the pump. Regulating of the fuel supply is by the manœuvring wheel at

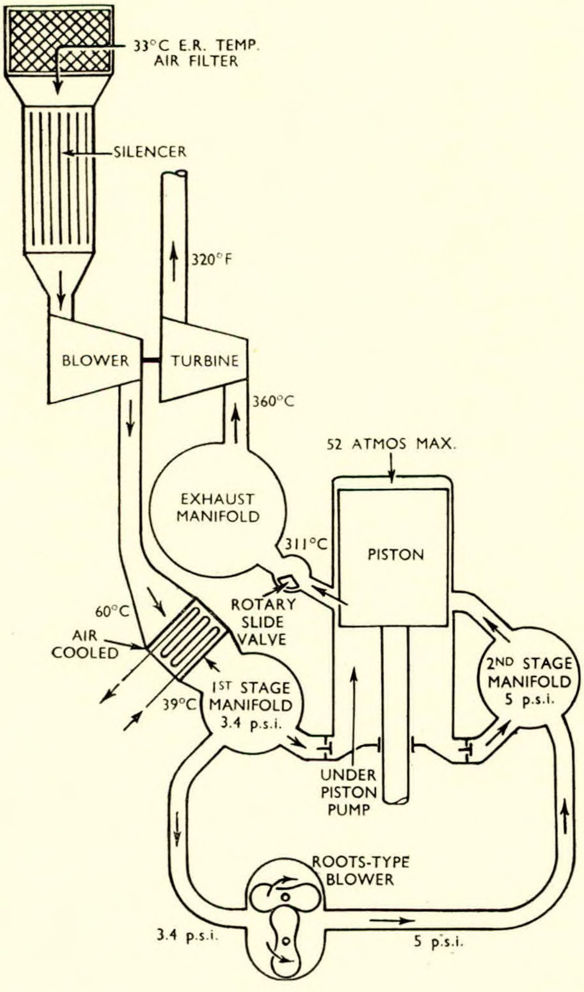


Diagram of the Bremer-Vulkan-M.A.N. pressure charging system the control position, rods from the wheel leading to the regulating shaft of the fuel pumps. Movement of this shaft gives earlier or later re-opening of the fuel delivery port, thus controlling the engine speed.—*The Motor Ship, December 1956; Vol. 37, pp. 336-342.*

New Werkspoor Two-stroke Engine

A new uniflow-scavenging two-stroke engine is shortly to run trials at the Amsterdam works of Werkspoor N.V. It is to be known as the type TE.450 and it has the following design particulars:—

Cylinder bore	450 mm. (17¾in.)
Stroke	700 mm. (26½in.)
Speed	250 r.p.m.
M.E.P. (normal)	5 kg. per sq. cm. (71lb. per sq. in.)
M.E.P. (turbocharged) ...	6·5 kg. per sq. cm. (92·5lb. per sq. in.)
Piston speed...	5·83 m. per sec. (1,150ft. per min.)

It is intended to build this engine in six- and eight-cylinder form with outputs of approximately 1,800 and 2,400 b.h.p. (or 2,400 and 3,200 b.h.p. when pressure charged). The TE.450 engine has been designed to extend the output range covered by the T.M. four-stroke engines which have been built by Werkspoor for some 25 years. In furtherance of Werkspoor's aim to provide reliable service and accessibility at a reasonable price it has been considered advisable to change from the four-stroke to the two-stroke design for the new engine. Bedplate and frame are welded and carry a cast iron cylinder block containing the scavenging air receiver and camshaft housing. A separate reciprocating scavenging air pump driven by lever from the running gear is provided for each cylinder. A built-up design of piston is employed and the cylinder head carries two exhaust valves. A longer stroke (900 mm.) version with a maximum speed of 200 r.p.m. is also under development.—*The Marine Engineer and Naval Architect, October 1956; Vol. 79, p. 356.*

Three Royal Mail Liners for Belfast

Royal Mail Lines, Ltd., have placed an order with Harland and Wolff, Ltd., Belfast, for three 20,000-ton gross passenger and cargo motorships to be named *Amazon, Aragon* and *Arlanza* for their London to South America service. The keel of the first ship is to be laid next June and delivery is expected in September 1959. This is the largest order ever placed by the Royal Mail at one time, the total cost of each ship being over £5m., which is approximately eight times the original cost of one of the *Highland* ships which they will eventually replace. The separate bridge structure, such a prominent feature of the *Highland* and *D*-class ships, is being retained to allow No. 3 hatch to be centred over the hold. Accommodation is to be provided for 100 first-class, 100 cabin-class and 270 tourist-class passengers, all in wholly air conditioned spaces. Denny Brown ship stabilizers will be fitted. Space will be provided in five main cargo sub-divisions for 4,000 tons of chilled meat. A proportion of this space will be available for fruit and dairy produce as seasonal demands may require. Two chambers will be capable of maintaining a temperature of −5 deg. F. The principal dimensions are as follows:—

Length between perpendiculars, feet	450
Length overall, feet...	583
Beam, feet	78
Depth, feet	41
Insulated cargo capacity, cu. ft. ...	445,000
General cargo capacity, cu. ft. ...	45,000
Service speed, knots...	17½

The machinery is to consist of two six-cylinder opposed-piston 750-mm. bore by 1,500-mm. plus 500-mm. stroke Harland and Wolff turbocharged single-acting two-stroke engines burning heavy fuel and designed to develop 8,500 b.h.p. each. The new ships will continue to use the owners' customary London discharging berth in Victoria Dock, but their beam is only two feet less than the Canal Cutting between the Royal Albert and Victoria Docks. By the time they are completed the P.L.A. will have widened the Cutting.—*The Marine Engineer and Naval Architect, October 1956; Vol. 79, p. 355.*

Australian Passenger Cargo Motorship

The West Australian State Shipping Service has a fleet of five vessels operating from Fremantle to North West ports of Western Australia and Darwin. The latest to be added is the Koojarra, a vessel of about 2,700 gross tons and 4,400 tons loaded displacement. She was constructed at the State Dockyard, Newcastle, New South Wales, being somewhat similar to the *Kabbarli,* built at the same yard for this service, but with increased passenger accommodation and other modifications. Whilst constructed in Australia, however, most of the machinery and equipment was supplied by British manufacturers, including the two propelling engines, the winches, windlass, capstan, chain cable, generating machinery, compressors and exhaust-gas boiler. The vessel has a raked bow, streamlined funnel and superstructure, and was built to Lloyd's classification 100 A1, the main particulars being:—

Length overall	297ft. 3in.
Length b.p.	270ft. 0in.
Moulded breadth...	46ft. 0in.
Moulded depth to upper deck ...	21ft. 6in.
Draught, loaded	18ft. 0in.
Loaded displacement	4,400 tons
Machinery...	1,820 b.h.p.
Service speed	12 knots

Accommodation is provided for sixty-one passengers in one class. The *Koojarra* is the first Australian-built ship equipped with an air conditioning plant for the ventilation of all passenger and crew accommodation and public rooms. The system is designed to supply air to each room at a fixed temperature, irrespective of the climatic conditions prevailing, and is independent of the forced and natural draught ventilation system installed for service areas. Two five-ton and six three-ton Stothert and Pitt, and two two-ton Laurence, Scott electric winches are provided on deck, 10 Jennings derrick topping winches being also included in the cargo-handling equipment. Of the derricks two are for 10-ton lifts and six for five tons, two being for two tons. The cargo is carried in three holds forward of the engine room and two aft, the whole of No. 3 and part of No. 2 'tweendecks being fitted for the transport of refrigerated cargoes. The hatches are fitted with MacGregor covers, Nos. 1, 2 and 5 being of the folding type and Nos. 3 and 4 of the hinged flush-deck type. The double bottom is arranged for oil fuel, fresh water or water ballast and tanks forward of the engine room take either fresh water or water ballast, whilst the corresponding tanks aft are fitted for oil fuel or water ballast. The total fuel capacity is 320 tons and the fresh water capacity 520 tons. Two British Polar 910 b.h.p. propelling engines are installed, having six cylinders with a diameter of 340 mm. and a piston stroke of 570 mm. They drive Stone's Heliston propellers through reduction gear and B.T.H. electric couplings, the propeller revolutions being 130 r.p.m., corresponding to 300 r.p.m. at the engines. A Siemens-Ford torsionmeter is fitted to the main propeller shafting. The generating plant comprises three Mirrlees-engined 150-kW dynamos, and there is a 20-kW emergency generator on the boat deck and a 75-kW motor generator in the engine room to enable shore power to be used on the ship's supply. A Spanner exhaust gas boiler is fitted and supplies steam to a 300-gal. fresh water calorifier, which is also provided with electric heating elements. The gases from the engine may be bypassed direct to the funnel.—*The Motor Ship, November 1956; Vol. 37, p. 323.*

Pulsation Errors in Manometer Gauges

Many instruments are available for measuring a steady gas pressure differential with a high degree of accuracy. Little information is available, however, for designing a manometer capable of indicating accurately the time mean of a periodic pulsating pressure drop. Such pulsating conditions are encountered by a manometer employed for measuring the head across a pressure-inferential meter installed in a duct leading to or from a reciprocating engine or compressor where the damping capacity between the meter and the source of pulsation is inadequate. It would appear that manometer pulsation errors can arise from one or more of the following sources:

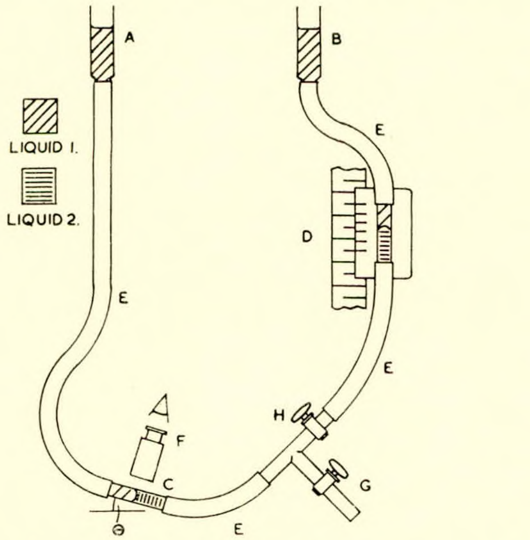


FIG. 6—*Diagram of two-liquid differential manometer*

1. Wave action and attenuation in the manometer connecting leads resulting in distortion at the manometer of the original pressure-pulsation pattern. 2. Volumes included in the manometer connexions or forming part of the manometer itself. Errors occur whether the volumes are symmetrical on either side of the manometer liquid column or not. 3. Restrictions or throttles in the manometer leads. 4. Non-viscous damping of the manometer-liquid oscillation. Although the inclusion of "viscous" damping elements in the connecting leads should ideally eliminate errors resulting from pressure pulsations of moderate amplitudes, the use of short pads of felt or similar material can result in a partial improvement only in manometer accuracy. It is concluded that a manometer gauge must satisfy the following requirements if pressure-pulsation errors are to be eliminated: 1. The length of the "gas" filled leads from the source of pressure to the manometer liquid surface to be as short as possible, and the cross section of the leads to be uniform. 2. Damping to be confined to the manometer liquid or liquids and to be viscous in nature. A type of manometer which satisfies these requirements is shown. (Fig. 6.)—*T. J. Williams, Trans.A.S.M.E., October 1956; Vol. 78, pp. 1,461-1,469.*

Coupled Torsional-horizontal Bending Vibrations in Ships

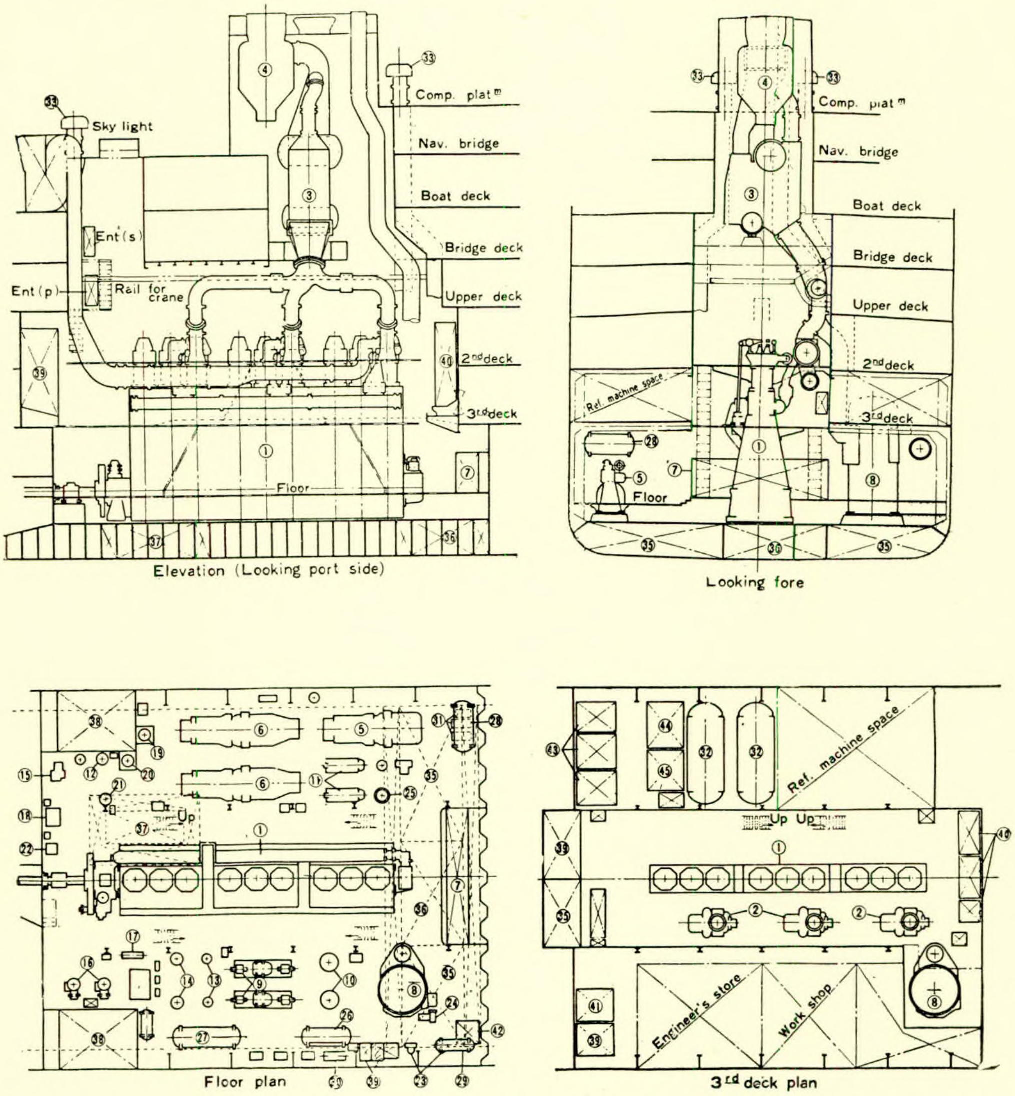
When a ship is excited by a horizontal or torsional vibratory generating force, the vibratory motion of the hull is generally represented as a coupled torsional-horizontal bending vibration because the location of the centre of gravity along the length of a ship deviates from that of the shear centre or the torsion centre in the vertical plane, including the centre line of the hull. Thus, when one compares the critical frequencies of horizontal hull vibration (calculated without reference to bending) with the corresponding natural hull vibration frequencies, one must correct the natural frequencies to make allowance for the above mentioned coupled interference. Strictly speaking, coupled torsional bending hull vibrations should be considered as vibrations of a curved bar; the profiles of the location of the centre of gravity along the ship's length and of the torsion centre of the element of hull length are both slightly curved. The present paper, however, is based on a study of the coupled torsional-bending vibration of a ship-like beam of variable cross section, with straight axes of the centre of gravity and of the torsion centre.—*T. Kumai, European Shipbuilding, 1956; Vol. 5, No. 4, pp. 95-98.*

New Japanese Diesel Engine

The *Sanuki Maru* was laid down in November 1954 and delivered in May 1955, after trials on which a speed of 20·6 knots was recorded. Built by the Mitsubishi S.B. and Engineering Company and having the first large turbocharged Mitsubishi Nagasaki U.E.C. Diesel engine, the vessel is classed with Lloyd's Register ✠ 100A1 and has the following principal dimensions and service particulars:—

Length, overall	156·7 m. (514ft. 0in.)
Length, b.p.	145·0 m. (476ft. 0in.)
Breadth, moulded...	19·5 m. (64ft. 0in.)
Depth, moulded	12·3 m. (40ft. 6in.)
Summer draught...	8·75 m. (28ft. 8in.)
Gross tonnage	9,307
Deadweight, metric tons... ...	11,040
Cargo capacity	16,585 cu. m. (585,000 cu. ft.)
Trial speed, knots	20·6
Service speed, knots	17·8
Service power, b.h.p.	10,000
Service r.p.m.	111
Maximum firing pressure, kg./cm.^{2}	54
Compression pressure, kg./cm.^{2}...	42
M.E.P., kg./cm.^{2}...	7·5
Exhaust temperature not greater than	350 deg. C.
Propeller (4 blades), diameter, m.	5·6
Propeller pitch, m.	5·15
Fuel consumption	150 gr./h.p./hr. 35·3 tons/day
Type of fuel (medium fuel oil)	Sp.G. 0·95 at 800 sec. Red. No. 1

The ship is a flush decker, with the main deck and bottom constructed on the longitudinal framing system, the second deck extending fore and aft. Care has been exercised in the arrangement of the cargo spaces in order that loading and unloading may be easily carried out at the numerous ports of call. All the hatches are fitted with MacGregor-type steel covers. A refrigerated chamber, silk room, strong room and mail room are provided and each hold is served by the Cargocaire dry hold system. It is interesting to note that the vessel's fuel capacity is sufficient for 18,600 sea miles at service speed. After extensive development work by Mitsubishi Nagasaki on a small three-cylinder unit had indicated the success of the design, the large U.E.C. engine installed in the *Sanuki Maru* was erected, shop tested and transferred to the ship to be subjected to sea endurance tests lasting six days and nights. The nine-cylinder two-stroke engine having a cylinder bore of 750 mm. and a stroke of 1,500 mm. develops a maximum continuously rated output of 12,000 b.h.p. at 120 r.p.m. when operating with the three exhaust-driven turbochargers which are located at top platform level. The new engine is of the uniflow-scavenging type, three exhaust valves being arranged in each cylinder cover, thereby providing a larger valve opening than would be available with the conventional design. Fitted in the centre of each head is a spring loaded fuel valve, to which the fuel is delivered by a system incorporating a number of unusual features. The fuel oil, after leaving the purifier, passes through a colloidal treatment unit and is then pumped up to the daily service tank from which it is drawn by the four plungers of the engine-driven fuel pump. From the pumps the oil passes through a common discharge to the camshaft-operated fuel-control valves, which are virtually individual timing valves regulating the discharge to each fuel valve. By means of a control rod attached to a lever on top of each fuel injector the spring pressure of all the fuel valves may be immediately adjusted to suit different operating conditions. It is, therefore, possible for the three factors governing correct injection, namely, fuel pressure, timing and valve opening pressure, to be adjusted independently. A point of technical interest is the use of Gamlen fuel additive, which is mixed with the fuel oil in the ratio of one litre per two kilo litres. The additive is used with the heavier grades of fuel and we are informed that it gives excellent results. Each of the three exhaust-gas-driven turbochargers is connected to a bank of three cylinders and, as shown by one of the accompanying illustrations, a small electrically driven blower is arranged in the scavenge air line for use when starting the engine. This blower was not required during shop trials but has been installed as a precautionary measure. The discharge from each main turboblower passes through a sea water-circulated air cooler before entering the cylinders. Both the main pistons and cylinder jackets are fresh water cooled. Compared with the non-pressure-charged type of engine having the same cylinder dimensions developing the same power it is stated that there is an 8 per cent increase in a cargo-carrying capacity and an approximate £3,500 saving on the annual fuel bill due to a higher mechanical efficiency.—*The Motor Ship, October 1956; Vol. 37, pp. 266-268.*



Engine-room plans of the Sanuki Maru

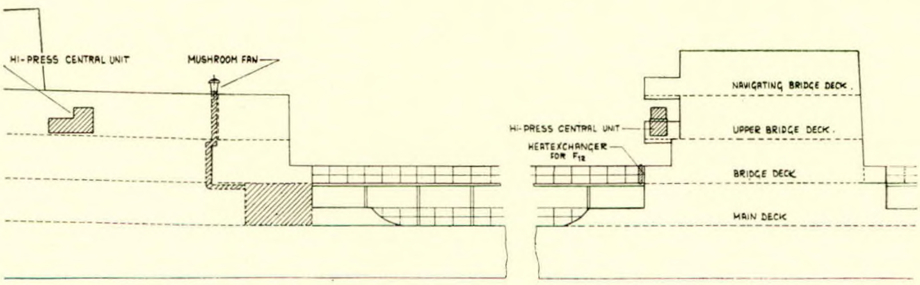
(1) Main engine; (2) Turbocharger; (3) Exhaust gas boiler; (4) Spark arrester; (5) Main dynamo; (6) Main dynamo and air compressor; (7) Main switch board; (8) Donkey boiler; (9) Cooling F.W. pump; (10) Sea water circulating pump; (11) L.O. pump; (12) L.O. transfer pump; (13) F.O. service pump; (14) F.O. transfer pump; (15) L.O. purifier; (16) F.O. purifier; (17) Delivery pump for colloidol filter; (18) Bilge pump; (19) Bilge and ballast pump; (20) G.S. pump; (21) F.W. pump; (22) Sanitary pump; (23) Feed water pump; (24) Forced draft fan for donkey boiler; (25) L.O. cooler; (26) Piston cooling F.W. cooler; (27) Jacket cooling F.W. cooler; (28) Jacket cooling F.W. cooler for aux. engine; (29) Aux. condenser; (30) Cargocaire cooling pump; (31) Ref. machine cooling pump; (32) Starting air reservoir; (33) Ventilating fan; (34) Donkey boiler funnel; (35) F.W. tank; (36) F.W. collecting tank; (37) L.O. drain tank; (38) F.O. tank; (39) F.O. settling tank; (40) F.O. service tank; (41) Jacket cooling F.W. tank; (42) Cascade tank; (43) L.O. settling or service tank; (44) Cylinder oil tank; (45) L.O. reserve tank.

Electrolytic Descaling

Electrolytic descaling is a clean, economical, and quiet method for cleaning rust, corrosion products, and black oil residues from large rectangular cargo tanks. Before the cleaning is started, the tanks should be gas freed and the oil residues removed. The inner tank walls are made the cathode in an electrical circuit. The sea water filling the tank serves as the electrolyte. When sea water is not available, a 1½ per cent salt mixture can be substituted. Information in this article is based on tests made using a sodium chloride solution as the electrolyte. There is no reason why other electrolytes could not be used satisfactorily. Suspended fabricated steel grids or curtains are used as anodes. Electrical energy is supplied by one or more low-voltage (about 10 volts) high-amperage (about 1,200 amperes) d.c. welding generators, so that a current density of about 0·5 to 1·0 ampere a square foot on the tank surface is maintained. The current passes from the anode through the electrolyte to the cathode. The length of time needed to loosen the scale varies with the thickness of the scale and the current density. Time required is longer with heavier scale and shorter with greater current density. Best results are obtained with a current density between 0·5 and 0·7 ampere per sq. ft. for the side and top surfaces, and a current density of about 1 ampere per sq. ft. for the bottom of the tank. Tanks can be descaled in 24 to 48 hours. For example, seven welding generators will clean a tank of 500,000-gal. capacity. One generator would be connected to the anodes for each bulkhead, one to the overhead anodes, and two in parallel to the bottom anodes. The anodes consist of curtains or grids of ½-in. to ⅝-in. diameter steel rods with hooked ends inserted into holes in angle iron framing, or hooked over 1-in. diameter rods. Current should never be applied when the tank is only partially full of water. Apply a generator load not to exceed 1,200 amperes a generator at about 10 volts. Adjust the current from 0·5 to 0·7 amperes a sq. ft. for the bottom anode. Run the current at this load for 16 hours minimum. Then drain down the tank and flush out the scale, making sure to remove all scale accumulations on the bottom of the tank. Refill with salt water, reapply the current, and continue descaling for 16 hours minimum. Reverse the flow of electricity at half the former load for five minutes at the end of the final descaling. If the ship is in drydock, the tank is drained by opening the 10-in. gate valve installed at the bottom. Flush the entire interior surface of the tank with salt water from the ship's fire mains, or from the shore connexion, leaving the drain valve open during the flushing operations. Continue electrolytic descaling if it is needed. If descaling is complete, hose down the interior tank surfaces with fresh water. Precautions must be taken during descaling to prevent explosions from hydrogen gas accumulations. Gas test readings should be taken frequently in spaces near the tank. "No smoking" and "No welding" signs are posted in the vicinity of the tank. Where a vessel is to be electrolytically descaled while afloat, with generators located at the pier, the size of copper ground-return cables should be at least 1,000,000 circular mils for each 1,000 amperes. This precaution will prevent possible stray current damage to the hull. The electrolytic method of descaling rust from tanker compartments has proved to be quieter, cleaner, quicker, and more economical than other conventional cleaning techniques.—*Bureau of Ships Journal, October 1956; Vol. 5, pp. 17-19.*

Air Cooling Plant Installed at Sea

On 20th July 1956 a new Hi-Press air cooling plant was started up and tested on board the Norwegian tanker *Hornfighter* of 13,350 tons d.w., while the vessel was in the Atlantic on a voyage from Curaçao to Rio de Janeiro. The whole plant had been installed at sea while the ship was in normal service. Such a thing had never been done before and many considered it impossible to solve the problems in connexion with such an installation in a satisfactory manner. The fact that it is now possible to make such an essential improvement without taking the ship out of service is of course a development of great importance to shipowners. The *Hornfighter,* owned by Jakob Kjode of Bergen, was built in 1954 by Frederiksstad Mek. Verksted. The ship was equipped with a ventilating plant of the conventional type, and a steam heating plant with radiators in all rooms. As the vessel sailed mainly in tropical waters, the want of an air conditioning plant soon made itself felt, and in the winter of 1956 the owners decided to have the vessel equipped with a Hi-Press air cooling plant as this plant met all the owners' requirements and could, without difficulty, be installed while the ship was sailing. The plant is intended only for the cooling of outside air as the original heating plant is used for the heating of the rooms. A plant of this type consists of central units, pipe systems, cabinets and refrigerating plant. The central unit contains an air filter, an automatically controlled air cooler for the direct expansion of freon, a high-pressure centrifugal fan, and a steel-sheet sound trap for erection on the open deck. The pipe system, of usual Hi-Press design, consists of thin walled steel pipes in standard dimensions of 45 to 108 mm. outside diameter, assembled by means of standard fittings, made specially for this purpose, and special clamping bands, including a rubber packing ring.—*K. Hemmingsen, Shipbuilding and Shipping Record, 15th November 1956; Vol. 88, pp. 639-640.*



Positioning of the central units: there are units port and starboard in the position shown aft. The mushroom fan is serving the refrigeration plant

Factors Influencing Shaft Alignment

Shaft alignment consists of positioning bearings to support the shaft system to allow it to propel the ship in conjunction with the propulsion machinery. It is considered in four parts: (1) static support of the weight of the shaft system; (2) prevention of buckling of the shaft column; (3) prevention of undesirable vibration; and (4) the maintaining of working stresses within design criteria. The composite stress in the shafting is basically another problem. Appreciating the environment of the shafting system is a prerequisite to the understanding of shafting alignment. To this end, studies were made by the shipyard to establish the response of a ship's hull in way of shafting to such influences as temperature, loading, speed, etc. The factors considered to influence shafting alignment are the position of the bearings, the distribution of the supported mass, and the forces imposed by dynamic action of the propeller, hull and attached machinery. The various effects which establish the position of the bearings are separated for this discussion into movements of the hull and movements relative to the hull. It is well known that temperature variations cause the hull girder to deflect both vertically and horizontally. Measurements have revealed hull deflexions between midships and the main strut of 1½-in. for a destroyer-type ship afloat. Undocking changes also produce deflexions of the hull girder. Measurements taken while undocking a destroyer hull revealed that the mainstrut section moved vertically relative to midships by 3in. Temperature and loading essentially were constant. Changes in loading will produce similar hull deflexions. It is apparent that in repair alignment needless time and money may be expended repositioning bearings to place them on a straight line, if the effects mentioned will make any straight line relative to the hull conform to the deflexion curve of the hull. It is considered that plot of the displacement of each bearing relative to the hull would be erratic. The need for shafting alignment work manifests itself in a variety of operational phenomena. Appreciation of this need, and the determination of the scope of investigation and the amount of work required in time to permit efficient and effective repair is one of the most difficult and important phases of the work. The principal sources of data for initial evaluation are contained in reports from ship's personnel, ship's-machinery logs, inspections and measurements made by repair personnel and observations on sea trials. The following would be considered as evidence indicative of the need for investigating shaft bearing alignment: 1. Repeated wiping of bearings. 2. Excessive wear rate, or uneven bearing wear. 3. Overheating of bearings. 4. Damaged shaft, bearing or hull in way of shaft. 5. Unusual noises in the shaft system such as thumping or squealing. 6. Poor seating of shafting in bearings as shown by clearance readings. 7. Leaking at stern tube stuffing box or excessive working of stuffing boxes.

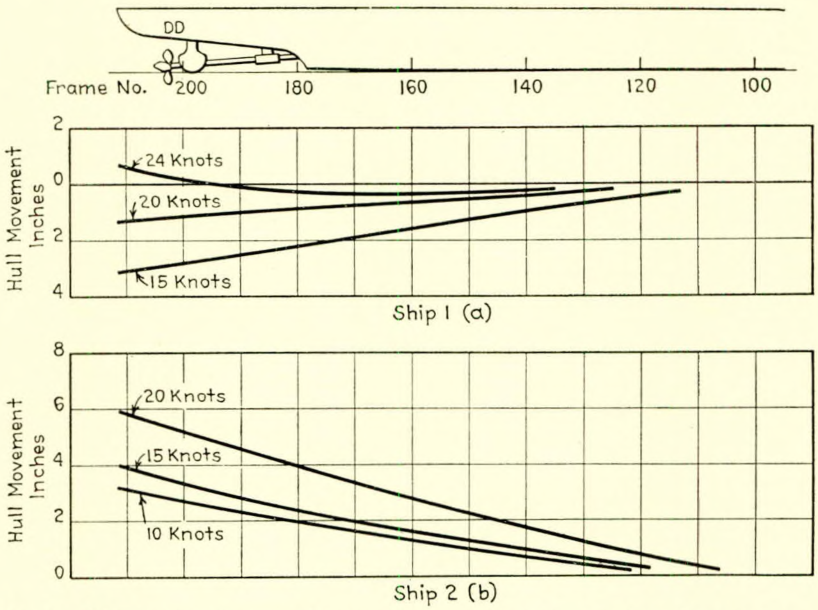


FIG. 1—*Movement of stern due to changes in ship's speed: reference zero is arbitrary and not zero speed; only differences are significant*

the hull deflects as a fair curve. In the operation of a ship, bearing wear produces an effective movement of the shaft support point relative to the hull. The magnitude of wear experienced varies in individual cases, but acceptable limits should be established by considering the requirements of the bearing manufacturer and the effect on the shafting system and associated equipment. Inaccuracies in installing and moving bearings constitute an effective bearing movement. A design which requires positioning accuracy not attainable is certain to result in unsatisfactory operation and expensive maintenance. Temperature also causes movements of bearings relative to the hull. The obvious offender in this category is the reduction gear casing which has a lubricating oil sump built integral with the base. Measurements of one casing showed a vertical lift of 0·025in. of the bull gear bearings relative to the hull girder. Other movements encountered result from repair to bearing housings, bearings, or associated structure. Inadvertent but significant movement is possible when a stern tube bearing housing is restored to dimension after excessive erosion or corrosion. Similarly, renewing one worn bearing may result in disastrously disturbing a system which has accommodated itself to misalignment by helpful wear down. In appreciating the effects it is apparent that the bearing movements relative to the hull are characterized by a randomness. Unlike the fair-curve deflexion of the hull, a 8. Whipping of shaft or vibration of shaft bearing pedestal. 9. Misalignment of associated main propulsion machinery. Evaluation of these factors will determine the need for further investigation and procedures to be followed.—*R. E. Kosiba, J. J. Francis and R. A. Woollacott, Marine Engineering/Log, October 1956; Vol. 61, pp. 83-84, 117.*

Automatic Mooring Winches

Automatic mooring winches are generally classified according to the type of driving unit supplying power for the winch, such as a steam engine, electric motor or electro-hydraulic transmission. Considering each of the steam, electric and hydraulic winches separately, the author discusses the steam automatic line positioning and tensioning mooring winch. A typical automatic steam mooring winch consists of a 12-in. × 14-in. double cylinder, reversing, steam engine with piston type valves, spur gearing a rope drum which is mounted between and supported by side housings. Winch heads, with or without whelps, are mounted either close up or on extended shafts conected by couplings to the end of the main shaft and supported by outboard bearings. When used in conjunction with warping heads, the welded steel rope drum is provided with a hand wheel operated clutch and brake. Warping heads can handle up to 12-in. circumference manila line and the brake is capable of holding 66,000-lb. or

107,000-lb. line pull based on the seventh layer of 1¼-in. diameter wire rope on the drum. The operation of the automatic tensioning winch can best be understood by reference to Fig. L. The winch operator, after the line is secured to the dock, inhauls the mooring line until it is taut. The clutch and pawls on the automatic device are then engaged and the control and reverse valve levers set to wind in. The winch will not take in the wire until the live pull is just balanced by the steam pressure in the cylinders. If the equilibrium is upset, say by the vessel being lifted by a wave, the tension on the wire will increase and the drum will pay out. When this takes place the automatic line positioning

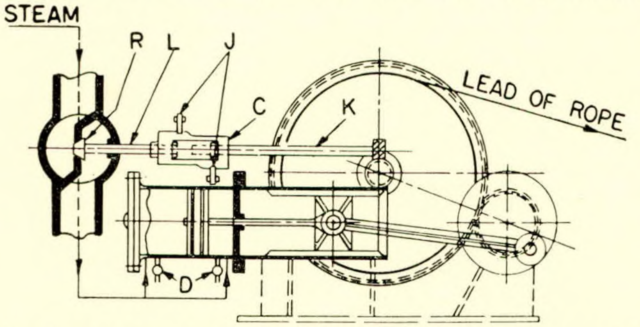


FIG. L—*Schematic diagram of automatic tensioning winch*

control device comes into effect. While the drum is paying out wire, a shaft (K), connected to the drum through a helical gear, is rotated. This shaft is threaded into an automatic sleeve nut (C) which moves up and down the shaft as it turns. On the other end of the sleeve nut (C), and free to rotate in it, is a shaft (L) which is coupled through linkages and a clutch to a steam valve (R) controlling the admission of steam to the winch cylinders. When the automatic pawls (J) are engaged, the sleeve nut (C) is prevented from turning and causes shaft (L) to rotate and actuate the steam valve (R). The mechanism and valve is arranged so that as the wire pays out, the steam valve is opened and more steam is admitted to the cylinders, thus resisting the paying out forces on the wire and driving the winch drum to recover the wire and return to its original starting position. In addition any sudden shock or severe overload on the winch system is instantly relieved through relief valves (D) on the steam cylinders. While the steam pressure on the cylinders of the steam automatic mooring winch effectively indicates the line tension, the electric automatic mooring winch must be provided with a resilient mechanical means of measuring tension. The electric control mechanism is operated by the movement of this resilient member to control the motor drive.—*W. Munch, Bulletin, The Society of Naval Architects and Marine Engineers, November 1956; Vol. 11, pp. 1-13.*

Trends in Application of Oil Engines

Although considerable attention is being devoted to bulk carriers and their strategic deployment, developments continue in interesting ways of solving propulsion problems within and outside the conventional framework. Not only is the internal combustion engine being applied to relatively high powered ships, but it is being fitted in ships of a size hitherto too big for construction in certain yards. Many yards, British and Continental, are becoming interested in larger ships, for dry, bulk and oil cargoes. Furthermore, a new type of motor tramp is emerging. A tanker of 45,370 tons deadweight to be built at Dunkirk is to have two B. and W. oil engines which, according to a contemporary report, constitute "a more powerful installation than any yet fitted in a motor tanker". These units will be turbocharged, two-cycle engines, each having nine cylinders, 740-mm. diameter and 1,600-mm. stroke, and each developing 11,250 h.p. at 113 r.p.m. The total output on the twin screws will thus be 22,500 b.h.p. Although this power is allocated to a single screw of a large turbine driven tanker, the French tanker owner is perhaps more cautious in outlook and indeed follows the French liking for twin screws in big tankers; in those which are intended to transit the Suez Canal, it is considered that the twin screws give better manœuvrability. The second happening of interest is the delivery by the Moss Vaerft and Dokk in Norway of their largest ship constructed to date. She is the ore carrier *Sunbreeze,* 11,000 tons, and is a single-screw motorship with machinery aft. She is a self-trimmer, capable of carrying bauxite, alumina, grain coal or iron ore. She has steel hatch covers, three cargo holds, of which No. 1 is the largest, accommodation aft and the navigating bridge forward of amidships. She has no cargo handling gear. The machinery is of single-acting two-cycle airless injection type, direct coupled and direct reversing.—*A. C. Hardy, The Journal of Commerce, 29th November 1956; No. 40,260, p. 5.*

Health Aspects of Welding

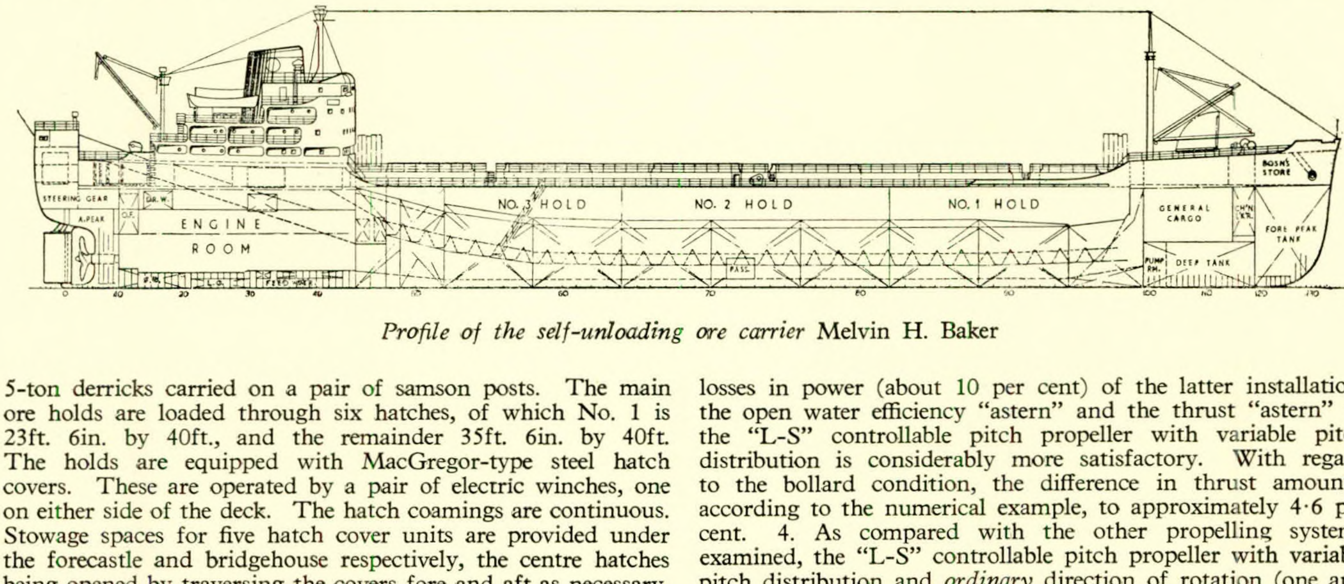
In submerged arc welding, the arc is covered with a rather heavy layer of granulated flux which normally shields the arc. These fluxes also contain fluorides. Tests have shown that, when the flux is heated to the melting point, hydrogen fluoride (hydrofluoric acid) or other gaseous fluorine compounds are produced. The amount of hydrogen fluoride produced depends on the amount of flux heated and the temperature reached. Little difficulty has been noted during machine welding where the size and position of the arc is carefully controlled. There have been complaints of nose and throat irritation during manual welding, particularly where large amounts of metal were being deposited. In the further interest of improving welds and extending the application of welding, the inert gas, metal arc welding process has been developed. In this process, oxygen is kept away from the welding arc by means of a mantel of inert gas, originally helium or argon, and more recently the less inert carbon dioxide. This process has introduced some very new and unusual potential hazards. The ultraviolet radiation is increased approximately tenfold, depending on the current density, the type of metal being welded and the particular inert gas used. This calls for additional precautions, both in protecting the skin and eyes of the welder and in shielding nearby personnel. Dark—preferably wool—clothing must be worn to cover *all* exposed skin surfaces. Tests by several authorities have shown that adequate eye protection from the ultraviolet is given in the use of the welding lenses that are recommended for the various amperages of ordinary arc welding. The increased brightness from visible light, however, may make it necessary for some people to use lenses that are one or two shades darker in order to prevent eye strain and depletion of the visual purple which is the cause of snow blindness during the summer months in the arctic region. It is worthy of note that intense ultraviolet light may cause the disintegration of cotton clothing. Ozone, which may be hazardous, of course, in any type of welding under conditions of inadequate ventilation, is produced in larger amounts by the inert-gas metal-arc welding process. The amount produced will vary with the type of metal and gas used; also with the current density and the atmospheric humidity. Fluxes containing fluorides are frequently used in welding and in brazing brass, aluminium or other alloys. In this case the heat of the flame may produce hydrogen fluoride. The concentration of hydrogen fluoride in the fumes is frequently high enough to be irritating, indicating that it is over the maximum allowable concentration (3 ppm). The concentration produced in the general room air will depend on the room size, general ventilation and the amount of welding or brazing being done. If the flux penetrates the outer skin surface through nicks or cuts, or around the fingernails, it may produce severe irritation. Skin contact should be avoided. Some brazing alloys may contain cadmium which is readily volatilized on heating. The brown cadmium fumes so evolved are extremely dangerous, producing severe and even fatal lung damage on short exposure. The maximum allowable concentration for cadmium is 0·1 mg. per cubic meter of air.—*B. L. Vosburg and J. J. Ferry, The Welding Journal, October 1956; Vol. 35, pp. 1,015-1,020.*

German Built Swedish Ore Carrier

A self-unloading ore carrier has been delivered from the Bremen yard of A. G. Weser to Scandinavian Ore Carriers Inc. A.B. (Nordstrom & Thulin A.B. and others), Stockholm. This vessel, the *Melvin H. Baker,* is about 17,000 tons deadweight, and is notable for the type of self-unloading gear with which she is fitted. This gear avoids the use of a boom-type discharge arm, the cargo being discharged by means of a horizontal conveyor which can be run on either side through ports in the stern. The vessel is a motorship, the main engines being Nohab-Polar five-cylinder engines. The *Melvin H. Baker* is intended for the trade between Nova Scotia and U.S. Atlantic ports. The self-unloading gear was specified on account of the high cost of unloading in these ports. The principal particulars of the ship are as follows:—

Length o.a.	524ft. 9in.
Length b.p.	507ft.
Breadth moulded	73ft.
Depth moulded	39ft. 5in.
Draught, summer	30ft. 1in.
Tonnage:—	
Deadweight	17,165 tons
Gross	10,883 tons
Net	6,386 tons
Machinery output	7,360 b.h.p.
Speed	15 knots
Cargo capacity:—	
Ore holds	598,056 cu. ft.
General cargo hold	58,255 cu. ft.

The ore carrying space has an overall length of 316ft., and is divided by transverse bulkheads into three holds. A hold for general cargo is arranged forward of this space, access being through a hatch 13ft. 6in. by 20ft. Cargo is handled by two 5-ton derricks carried on a pair of samson posts. The main ore holds are loaded through six hatches, of which No. 1 is 23ft. 6in. by 40ft., and the remainder 35ft. 6in. by 40ft. The holds are equipped with MacGregor-type steel hatch covers. These are operated by a pair of electric winches, one on either side of the deck. The hatch coamings are continuous. Stowage spaces for five hatch cover units are provided under the forecastle and bridgehouse respectively, the centre hatches being opened by traversing the covers fore and aft as necessary. In cross section, the hold bottoms are arranged in the form of a "W", and similar contours in the fore-and-aft direction allow the ore to be fed to two sets of hopper doors, one on either side of the centreline, each with thirty-one doors. Each set discharges ore on to a conveyor belt beneath the holds. The two conveyor belts, each 4ft. wide, carry the ore aft. They pass through the engine room in passages inclined at an angle of 18 degrees to the horizontal, and terminate in the discharging compartment aft in the poop. Here the ore falls on to a transverse conveyor belt extending from one side of the ship to the other. Beneath this is arranged a second transverse conveyor belt, and this can be traversed sideways through either side of the ship to project up to 11ft. 6in. beyond the maximum beam of the ship. The ore is carried out on this second belt, and then ashore. The various conveyor belts are driven by electric motors at speeds of 23ft. per sec., giving a cargo discharge rate of 1,500 tons per hour. The main propelling machinery consists of a pair of Nohab-Polar type M610T two-stroke Diesel engines, each developing 3,680 b.h.p. at 230 r.p.m. The engines are coupled to a single propeller shaft through reduction gearing.—*The Shipping World, 7th November 1956; Vol. 135, p. 419.*



Profile of the self-unloading ore carrier Melvin H. Baker

Results of Lips-Schelde Propeller Tests

The results of the investigations concerning the open-water efficiency "astern" and the thrust "astern" of the Lips-Schelde "L-S" controllable pitch propellers with constant pitch and variable pitch distributions, and the "T" series propellers, enable the following conclusions to be drawn: 1. When sailing "astern" the open water efficiency and the thrust of the "L-S" controllable pitch propeller with variable pitch distribution is considerably superior (between $\lambda = 0{\cdot}1$ and $0{\cdot}4$ abt. 18 per cent) to those of the "L-S" controllable pitch propeller with constant pitch distribution. With regard to the bollard condition, the difference in thrust amounts, according to the numerical example, to approximately 18·5 per cent. 2. When sailing "astern" the open water efficiency and the thrust of the "L-S" controllable pitch propeller with variable pitch distribution is considerably superior to those of the "T"-series propeller, with constant pitch, with an ordinary propelling installation. With regard to the bollard condition the difference in thrust amounts, according to the numerical example, to approximately 19·6 per cent. 3. When sailing "astern" it is shown that in comparing the "L-S" controllable pitch propeller, with variable pitch distribution, and the "T" series propeller with fixed blades, combined with a Diesel electric propelling installation (constant power, varying torque and number of revolutions), and in paying regard to the higher losses in power (about 10 per cent) of the latter installation, the open water efficiency "astern" and the thrust "astern" of the "L-S" controllable pitch propeller with variable pitch distribution is considerably more satisfactory. With regard to the bollard condition, the difference in thrust amounts, according to the numerical example, to approximately 4·6 per cent. 4. As compared with the other propelling systems examined, the "L-S" controllable pitch propeller with variable pitch distribution and *ordinary* direction of rotation (one and the same direction of rotation for sailing "ahead" and "astern") yields the optimum backing power. 5. When sailing "ahead", the "L-S" controllable pitch propeller with variable pitch distribution, yields over the usual range of advance coefficients ($\lambda = 0{\cdot}3$-$0{\cdot}5$), *the same open-water efficiency* as do the "L-S" controllable pitch propeller with constant pitch, and the "T"-series propellers, which cover the same range of pitch ratios.—*International Shipbuilding Progress, 1956; Vol. 3, No. 26, pp. 517-527.*

Bearings for Marine Geared Turbines

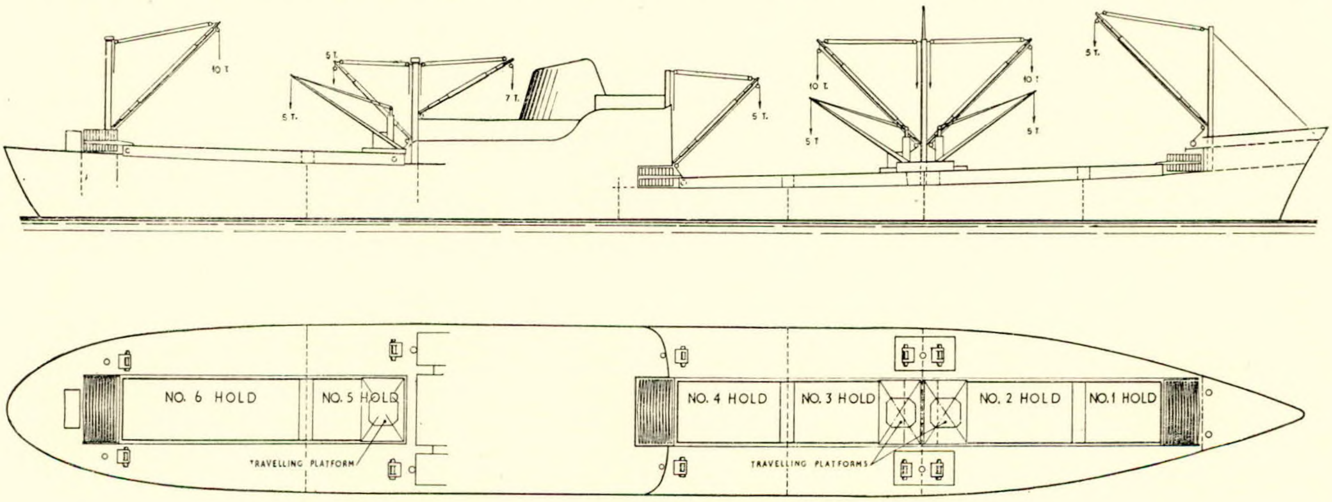
The prime requirements of bearings for marine geared turbines, reliability and efficiency, are considered. It is shown that the correct choice of length/diameter ratio and clearance ratio is essential in order to attain these requirements, and it is indicated that for the conditions which occur in most marine turbine and gear bearings, relatively short bearings with length/diameter ratio between two-thirds and one-third, and clearance ratios between one and two thousandths of an inch per inch diameter, give the best results. Bearing and shell materials are discussed. The advantages of high-tin white-metal as bearing material are shown and the relative merits of non-ferrous and ferrous shell materials are considered. It is concluded that the best general design is one of thin white-metal on a steel shell; the shell may be of normal thickness or it may be of thin shell type, and the advantages of this design are considered. The requirement of utmost reliability in a marine bearing involves if possible the ability for limited operation under a condition of lubricating oil failure. The shortcomings in this respect of present bearings are mentioned, and an improved design which overcomes them is described.—*International Shipbuilding Progress, 1956; Vol. 3, No. 26, pp. 528-539.*

Cargo Liner with Crane-operated Hatch Covers

The *Burrard* and *Bolinas* are the latest additions to the ships maintaining the Fred Olsen service between Northern Europe and Canadian and American North Pacific ports. Two 5-ton electric cranes are mounted on platforms which are effectively enlarged hatch covers, and serve as such when the hatches are closed: these platforms run on rails inside the coamings. The two cranes can thus serve any one of the four holds, as required, or alternatively, of course, can work separate holds. In addition to the cranes, each hold is served by a pair of derricks of 5- or 10-tons capacity, mounted on samson posts. The pair of posts between Nos. 2 and 3 holds is set well out on the side decks, so as to allow the cranes to pass through. The hatch coamings are unusually high, and a good deal of longitudinal stiffening has been worked into them. The propelling machinery consists of an Akers.-B. & W. six-cylinder turbocharged Diesel engine developing 8,600 b.h.p. It is of the opposed piston type.—*The Shipping World, 21st November 1956; Vol. 135, pp. 460-461.*

Forced Circulation Boilers and Some Marine Applications

For a number of years, especially since the end of World War II, the engineering literature has contained an increasing number of references to forced circulation boilers. Most of these articles extol the virtues of some one kind of boiler or tend to justify the conventional natural circulation principles. The range of boiler types covered by the term "forced circulation" is much broader than is usually realized, and since some of these types are basically quite different the existing confusion on this subject is understandable. In this paper an attempt is made to classify and describe the more common types of forced circulation boilers among a brief history of the basic types and the factors which have stimulated their development. A brief comparison with natural circulation brings out the advantages of forced circulation and the disadvantages. It will be noted that advantages far outweigh the disadvantages. This is not intended to minimize the importance of natural circulation, but rather to show that forced circulation represents a basic step forward in steam generation similar to that of the water tube over the fire tube boiler. The latter part of this paper is devoted to pioneer marine installations of various types of forced circulation boilers and the results of these installations.—*Abstract of a paper by W. L. Marshall. Bulletin, The Society of Naval Architects and Marine Engineers, November 1956; Vol. 11, p. 33.*



Arrangement of holds, hatches and cranes in the Burrard

The principal particulars of the *Burrard* are as follows:—

Length o.a.	507ft. 9½in.
Length b.p.	465ft.
Breadth moulded	65ft.
Depth to upper deck	40ft.
Depth to second deck	30ft.
Draught, loaded	30ft.
Tonnage:—	
Deadweight	11,150 tons
Gross	9,358 tons
Net	5,572 tons
Horsepower	8,600 b.h.p.
Service speed	17 knots

The arrangement of the hatches and cargo handling gear can best be understood by reference to the accompanying drawing. Nos. 1 to 4 holds are served by what is in fact one long hatch, with coamings extending the length of the four holds.

Patent Specifications

Hydraulic Steering Gear

This invention relates to hydraulic steering gear for ships, of the type in which a reversible pump, operated by rotation of a steering wheel, is adapted to pump hydraulic fluid through one of two pressure conduits to one or the other of two hydraulic rams arranged one on each side of a tiller bar at the upper end of the rudder post of the ship, so that rotation of the wheel in one direction deflects the rudder to the other side. With hand hydraulic steering gears, when the rudder is taken over towards the hardover position, the tendency of the rudder to self-centre causes a pressure on the hydraulic rams which is transmitted back to the reversible pump so that the steering wheel must be held against this pressure. If released, the wheel would be rotated and allow the rudder to return to a midships position. Also, similar pressures occur in heavy seas and waves hitting the rudder can cause the wheel to kick in the helmsman's hands in such a manner that it is difficult to hold the wheel. Trawlers fitted with hand hydraulic steering gears frequently require, when trawling, to steam with the

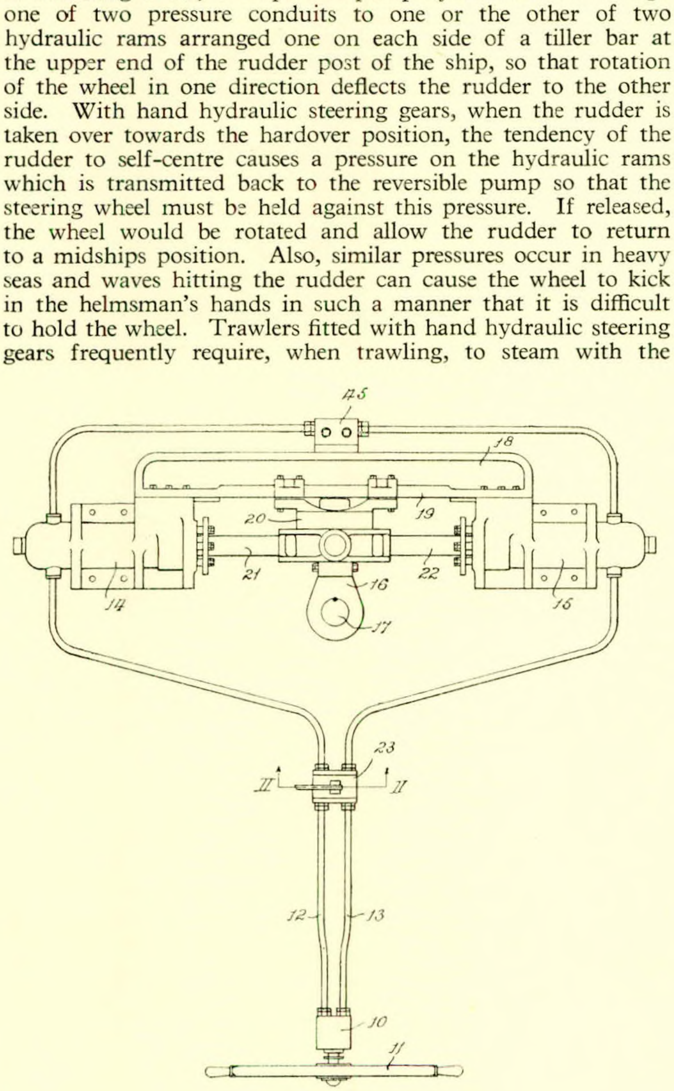
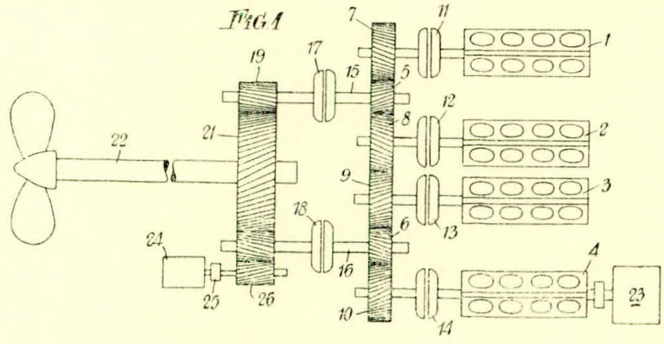


FIG. 1

rudder over at an angle, and with known steering gears it must be held in this position by the helmsman. An object of the invention is to provide a steering gear in which these disadvantages are overcome. The invention provides a steering gear in which there are two non-return valves, one in each of the pressure conduits so as normally to prevent passage of fluid from the hydraulic rams to the reversible pump, these valves being coupled together so that when hydraulic fluid is pumped through one pressure conduit to one ram, the valve in the other conduit is opened to permit fluid in the other pressure conduit and the other ram to return to the pump, and *vice versa*. The hydraulic steering gear as shown in Fig. 1 comprises a reversible pump (10), operated by rotation of a steering wheel (11), adapted to pump hydraulic field through one of two pressure conduits (12, 13) to one of two hydraulic cylinders (14, 15), arranged one on each side of a tiller bar (16) arranged at the upper end of a rudder post (17). A girder-like member (18) extends between the two cylinders (14 and 15), and this provides a guideway (19) for a crosshead (20), which is formed at the junction of two rams (21, 22) which work in the cylinders (14, 15). The crosshead (20) connects also to the tiller bar (16). In each pressure conduit (12, 13) is a non-return valve.—*British Patent No. 760,863, issued to Donkin and Co., Ltd. Complete specification published 7th November 1956.*

Ship Propulsion Plant

The propulsion plant illustrated in Fig. 1 comprises the four non-reversing internal combustion engines (1, 2, 3, 4), the engines 1 and 2 being coupled together to form a pair via the small intermediate wheel (5) and the engines 3 and 4 being coupled to form a pair via the small intermediate wheel (6). Disposed between the crankshaft pinions (7, 8, 9, 10) and intermediate wheels (5 and 6) on the one hand and the internal combustion engines (1, 2, 3, 4) on the other hand are the fluid clutches (11, 12, 13, 14) which enable the internal combustion engines to be brought into and out of operation at choice without interfering with the running of the engines. The shafts (15 and 16) associated with the intermediate wheels (5 and 6) respectively drive via the fluid clutches (17 and 18) respectively the pinions (19 and 20) respectively which mesh with the large gear wheel (21) on the propeller shaft (22).



The two engines (1 and 2), forming the port pair, rotate oppositely to the two engines (3 and 4) forming the starboard pair, the crankshaft pinions (8 and 9) of the inner engines (2 and 3) of the two pairs meshing with one another. The form of plant illustrated can be used for single-screw craft or can be doubled for craft having two screws. For travel ahead, the port fluid clutch (17) is filled and the starboard fluid clutch (18) is emptied, and *vice versa* for travel astern. Since the intermediate wheels (5 and 6) have fewer teeth than the crankshaft pinions (7, 8, 9 and 10), the fluid clutches (17 and 18) rotate more rapidly than the internal combustion engines (1, 2, 3 and 4), so that the clutches can be relatively small. The clutches (17 and 18) are of such size and design that the inner internal combustion engines can be placed so close to one another that only a catwalk remains free between them without the clutches coming into contact with one another. In the space thus gained at the outside of the engines, auxiliary engines such as pumps or the like can be arranged. The size of the fluid clutches (17 and 18) is determined by the transmission ratio between the crankshaft pinions (7, 8, 9 and 10) and the intermediate wheels (5 and 6). A very slow speed stage is obtained by the electric motor (24) which is supplied, for example, from the generator (23), and which through the medium of the clutch (25) directly drives the large gearwheel (21) via the pinion (26). In this case the fluid clutches (17 and 18) must of course be empty.—*(British Patent No. 764,005, issued to Aktien-Gesellschaft Weser. Complete specification published 19th December 1956.)*

Oil-fired Boiler

During normal operation of the boiler shown in section in Figs. 2, 2(a) and 2(b), oil fuel supplied to the burners (49) is burnt in the combustion chamber (19), part of the flue gases flowing through pass (140) containing the superheater (105) and the tube bank section (61), and part through pass (141) containing the tube bank section (63), to the damper controlled outlet and from there to the economizer and the smoke stack.

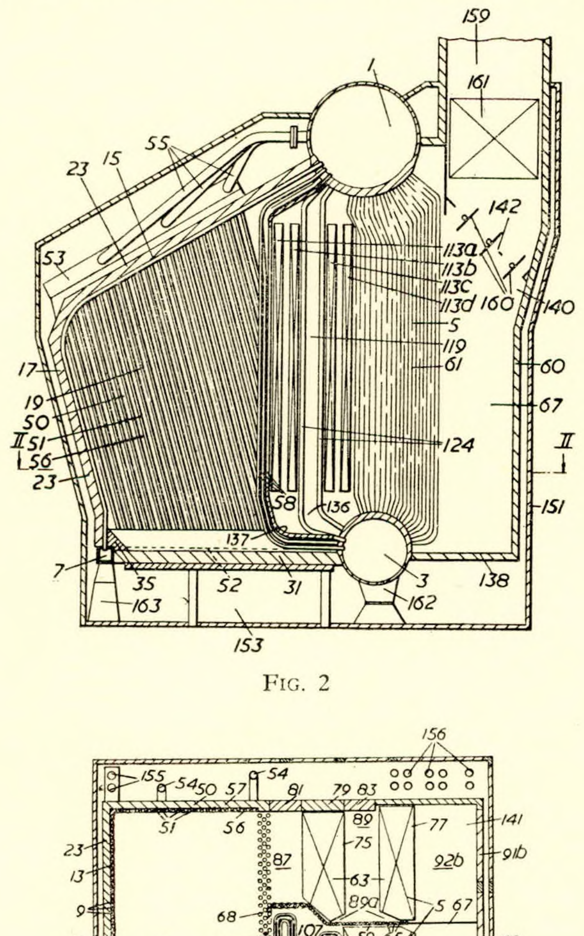


FIG. 2

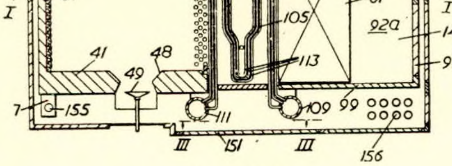


FIG. 2(a)

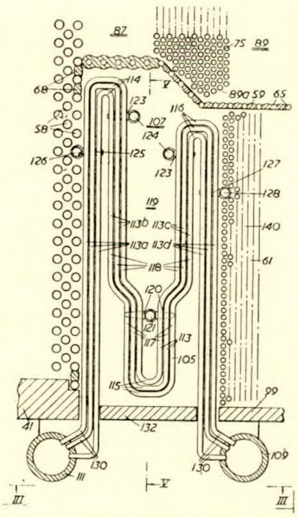


FIG. 2(b)

Normally, the dampers controlling the flow of gases from the combustion chamber over the superheater will be kept closed until steam is generated in the boiler. This steam passes from drum (1) through the superheater and then to the point of consumption, and once such a flow of steam has commenced, the damper controlling the flow of hot gases over the superheater may be opened to the required degree. The amount of superheat imparted to the steam may be varied by adjustment of the two sets of dampers controlling the gas flows respectively through pass (140) and through pass (141). Thus, a moderate steam temperature is obtained over a wide load range with both sets of dampers open; a higher steam temperature may be obtained by partial closure of the dampers associated with gas pass (141) and a lower steam temperature by partial closure of the dampers (160) associated with gas pass (140). It will be appreciated that the arrangement of the tube bank (5) and the superheater (105) provides excellent facilities for cleaning the tubes of the tube bank, and of the superheater, while the baffle wall separating the two gas passes is readily accessible for inspection and repair. As a result of the upward inclination of the axis of each tube (113) from the headers (109 and 111) to the central return bend (115), proper drainage of each tube is ensured when the headers (109 and 111) are emptied, despite the upright disposition of the headers. Individual superheater tubes may be readily removed and new tubes inserted, since the end portions (130) of each tube are inclined outwardly to the associated headers (109 and 111), and by cutting through the end portions the tubes may be released from the headers and withdrawn through the space between the headers. There are three tubes in each nest, and if necessary these may be withdrawn simultaneously in order to effect tube renewal. The arrangement of the fingers which support the tubes (113) is such that upon a short outward movement of any tube (113) it disengages from the fingers (120) and the return bend (115) is free to fall to a certain extent, so facilitating withdrawal of the tube. The support fingers (120 and 123) on the tubes (121 and 124) may easily be renewed from within the access space (119) and the support fingers (125) on the tube (126) of the tube screen (58) may be reached from the furnace chamber by cutting away a few of the screen tubes (58). The remaining support fingers (127) are in a cooler zone and are less likely to require renewal. —*(British Patent No. 760,502, issued to Babcock and Wilcox, Ltd. Complete specification published 31st October 1956.) Engineering and Boiler House Review, January 1957; Vol. 72, pp. 32-33.*

Propulsion Plant with Auxiliary Engine

This invention is primarily applicable to craft having a powerful main engine capable of propelling the craft at high

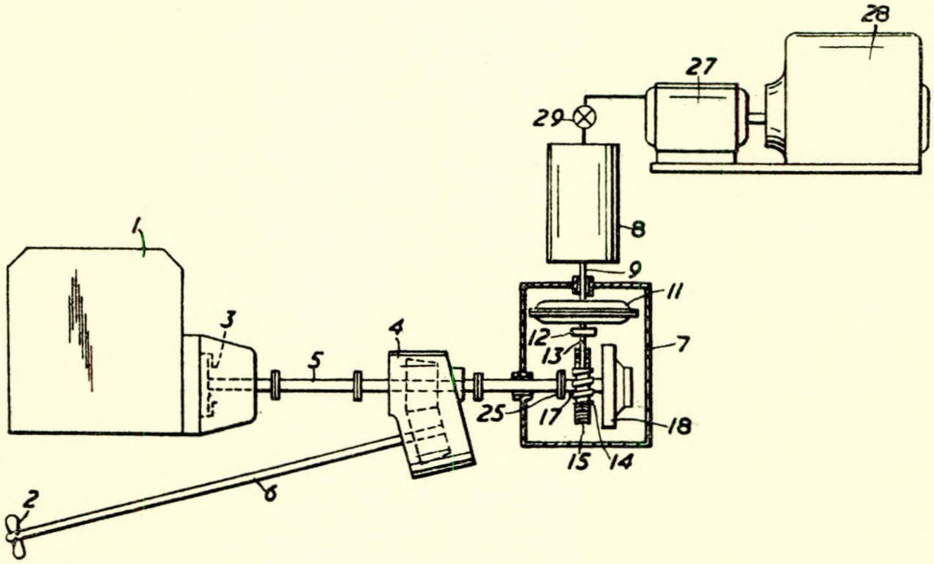


FIG. 1

speeds, for example high speed motor launches. Such craft are difficult to manœuvre at low speeds, for example when entering harbour, coming alongside a jetty or quay, or picking up a buoy. Moreover, the fuel consumption of the main engine is very high, so that if it is used for low speed manœuvring the endurance of the craft at sea will be greatly reduced. By making use of an auxiliary engine to provide the power for manœuvring at low speed, for example three or four knots, manœuvring is made much easier and the available supply of fuel is conserved for use when high speed is required. In Fig. 1 the main engine (1) of about 2,000 h.p. drives the propeller (2) through a main clutch (3) and a V-drive transfer box (4) which connects the horizontal forward-facing clutch output shaft (5) to the sloping propeller shaft (6) with a 1 to 1 ratio. The clutch output shaft (5) is continued through the transfer box (4) to an auxiliary drive unit. The auxiliary drive unit comprises a pedestal (7) at the top of which is mounted a 20 h.p. electric motor (8) with its axis vertical and its shaft coupled to a hydraulic coupling (11) situated in the pedestal (7). The output shaft of the hydraulic coupling (11) is connected through reduction gearing (12) to a layshaft (13) carrying a worm (14). This meshes with a worm wheel (15) mounted in the pedestal with its axis in line with the shaft (5) of the transfer box (4). The worm wheel (15) is mounted on a ball bearing to turn freely on the output shaft (17) of a clutch (18). —*British Patent No. 763,091, issued to Saunders-Roe (Anglesey), Ltd. Complete specification published 5th December 1956.*

Resilient Fender

Fig. 1 shows a fender according to the invention in plan view, partly broken away to illustrate the constructional principle, as mounted upon a vertical surface. In use the fender is arranged either against the side of the vessel or a quay or between them upon the surface of the water, and the resilient cushioning members, such as the tyres (5), effectively absorb and resist considerable crushing forces between the vessel and quay or between two vessels located side by side. Upward

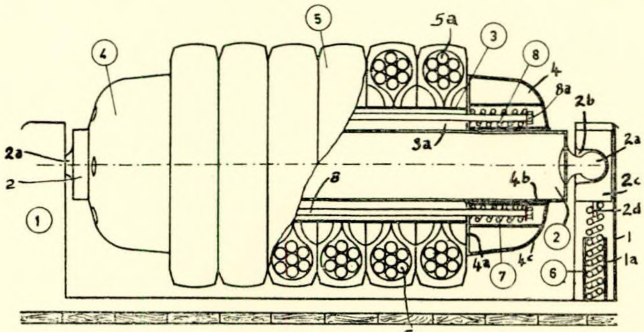


FIG. 1

and downward movement of a vessel on the water is taken up by the fenders, so that the fender continues to prevent contact between the vessel and a quay or other vessel alongside. The fender shown in Fig. 1 absorbs crushing forces by means of the tyres (5) and also by means of the springs (6), and upward and downward movement of a vessel in contact therewith causes the fender to rotate in its bearings without being damaged or ceasing to provide the requisite cushioning effect.—*(British Patent No. 764,276, issued to M. Rolando. Complete specification published 19th December 1956.)*

Marine Engineering and Shipbuilding Abstracts

Volume XX, No. 4, April 1957

* Patent Specifications

New Antifouling Device

A group of Amsterdam scientists has developed a new method aimed at combating the fouling of ship's hulls. This is the outcome of investigations started on request of the Koninklijke Hollandsche Lloyd, Amsterdam. The new method consists in the creation of a layer which prevents the fouling of the ship's hull by its toxic and repellent properties. The application of this coating is not done in the form of paint, but by blowing the vaporized compounds under the hull with the aid of the exhaust gases of the auxiliary engines, during the stay of the ship in the fouling ports. Experiments conducted on two ships of the K.H.L. have shown that with the aid of transportation by gas the protective layer can be spread over the whole of the ship's shell and bottom plating as well as the rudder and propeller. The results obtained with this new method are claimed to be promising. The m.v. *Gooiland,* one of the experimental vessels, for example, after four voyages to Brazil and fourteen months in the water turned out to be completely free from fouling. The m.v. *Gaasterland's* shell and bottom plating were free from fouling after four voyages to Brazil. The new method has been patented in the Netherlands.—*Holland Shipbuilding, November 1956; Vol. 5, p. 24.*

Review of the Pitot Tube

A Pitot tube or similar velocity-measuring device consists essentially of three pstyd; namely, the head or instrument section, the pressure connecting lines between the head and the pressure indicating device, and the indicating device. The discussions of this presentation are limited to the head or instrument section. It is recognized that the connecting lines and the pressure measuring device are important elements of the system, but in many respects they are adequately treated elsewhere. Although the matter of measurements in pulsating flow is of extreme importance in many applications, this subject does not receive more than a scant mention in this paper, since the effect of pulsations cannot be adequately treated without a detailed consideration of the flow in connecting pressure lines and the dynamic characteristics of the pressure measuring device. A large variety of geometrical shapes are possible and actually have been used. This complicates the problem when one attempts to analyse and correlate available data for presentation in summarized but adequate form. Any suitable opening in a reasonable geometrical shape will provide a pressure approaching closely the total or impact pressure. Also, any difference in pressure between two pressure openings can be calibrated in terms of velocity or, if the tube is installed in a pipe line, against the rate of flow. This paper attempts to correlate similar phenomena associated with probes. Several of the more common forms of probes are described, and their pressure performance is discussed in detail for normal and for a few special flow conditions. Experimental calibrations for a Pitot-static tube with a short tip, especially designed for measurements of quantity rate of flow of liquids in pipes, demonstrate the suitability of the probe. Necessary corrections to probe calibrations in a large cross-section uniform flow field for use in pipes are developed, presumably to account for effects of velocity gradients and mutual interference of flow between the pipe wall and the probe. Quantitative information, presented in the form of curves, provides readily available data for designers and users to determine desirable forms for given applications and to interpret pressure measurements made with specific probes.—*R. G. Folsom, Trans.A.S.M.E., October 1956; Vol. 78, pp. 1,447-1,460.*

Hydraulic Hatch Covers

The *Mariposa* is the first commercially operated vessel to have the Greer hydraulic hatch covers. The *Mariposa* is equipped with eight hydraulic hatch covers which are automatic, quick acting and self-stowing. The hatch covers on the main deck are watertight and the 'tween deck covers are flush and non-tight. Hatch seven on the upper deck is equipped with a Greer flush and watertight hydraulic hatch cover. The hatch cover when closed will serve as a promenade area for passengers. The watertight hatch covers may be opened from a completely secured position, including undogging of the hatch, in approximately four minutes, or closed from an open condition to a completely secured condition including latching in approximately two minutes. These operations occur by having only one man push a button and the same man perform the latch operations. No ships' gear or tackle is required, no tarpaulins or strongbacks, no hatch boards or beams, and no battens or complicated screw-down dogs. The 'tween deck hatch covers are automatically controlled from a master "push button" control station located on the deck. Therefore, all the hatch covers may be opened in a hold in anywhere from six to eight minutes, depending on which of the three holds is being operated. The use on these vessels of the Greer hydro-hatch, hydraulically operated hatch covers, will facilitate loading and discharge of cargo, with consequent increase of operating efficiency and effect a reduction in cargo handling costs. Due to the quick, smooth action of the hatch covers, there need be no fear of changing weather conditions, especially in the service for which these vessels are intended. If a storm should suddenly occur all the weather deck hatch covers can be closed in several minutes by one man and reopened immediately when the storm has subsided, with little or no time lost due to the opening and closing of hatch covers. The safety of the personnel operating the hatches is further safeguarded by the use of the Greer hydro-hatch, hydraulically operated hatch covers, since there will be no swinging pontoons, loose hatch boards or strongbacks, or wire rope to injure the personnel. The source of power of the Greer hydro-hatch comes from a 5-h.p. pump and motor assembly with the Greer hydro-pneumatic accumulators. These accumulators, or hydraulic storage units, provide enough stored hydraulic energy to close all the hatch covers on the ship should there be no electrical energy (dead ship) available. These accumulators act as both pulsation dampeners and shock absorbers as well.—*Marine News, November 1956; Vol. 43, pp. 31, 45.*

Dropwise Condensation of Steam

If a steam condenser is coated with a very thin layer of a hydrophobic substance, the condensate is deposited upon it in discontinuous drops rather than as a continuous film. The rate of heat transfer, and consequently the condensation rate, are considerably increased. The British Coal Utilisation Research Association has carried out tests of coatings consisting of various types of coal extract; two of these have given good results. These extracts, in the form of 1 per cent solutions in methylcyclohexanone (Sextone B), were applied by painting, dipping, or electrophoretic deposition to cupro-nickel condenser tubes of ⅝-in. outside diameter. The tubes were drained and afterwards baked for thirty minutes at about 240 deg. C. (465 deg. F.). They were then placed in a single-tube test condenser. In one case, the treatment increased the overall heat transfer by 55 per cent. These coatings last considerably longer than many of those previously tried; the latter either have too short a life or contaminate the steam. Further tests are in progress.—*R. L. Bond and R. Holland* et al., *Nature, 25th August 1956; Vol. 178, p. 431. Journal, The British Shipbuilding Research Association, November 1956; Vol. 11, Abstract No. 12,220.*

Craze-Cracking of Metals

Metal components sometimes fail by a form of cracking which is known variously as thermal fatigue, craze-cracking, heat checking, and network or mosaic cracking. This type of failure occurs when a surface is repeatedly heated and cooled. The cracks which are formed in the heated layer are usually shallow and may either be oriented at random or have a preferred direction. The surface may withstand many thousands of thermal cycles before cracking develops, or on the other hand cracks may be found after the first few cycles if the conditions are severe and the material is brittle. For many applications slight cracking can be tolerated but usually a stage is reached when the cracks propagate sufficiently either to fracture or to render the component unsatisfactory. When a surface of a piece of metal is rapidly heated thermal expansion is to some extent prevented by the cooler underlying metal and compressive stresses are produced. If the elastic limit is exceeded, plastic flow takes place in the direction perpendicular to the surface, and on subsequent cooling this region is left in tension. This tensile stress is the primary cause of the cracking, although other factors such as external stresses and phase changes may complicate the picture. The object of the investigation was to develop a suitable test equipment and procedure to study the cracking obtained in steels, and later to evaluate different metals and alloys for their tendency to craze cracking.—*L. Northcott and H. G. Baron, Journal of the Iron and Steel Institute, December 1956; Vol. 184, pp. 385-409.*

Motor Vessel with Ultrasonic Anti-barnacle Protection

Ultrasonic equipment for repelling barnacles and marine growth is the feature of a new vessel recently completed for Denholm Line Steamers, Ltd. The principal particulars of the *Glenpark* are as follows: —

Length o.a.	468ft.
Length b.p.	430ft.
Breadth	61ft. 9in.
Draught	26ft. 10¾in.
Deadweight	10,410 tons
Net tonnage	3,344 tons
Gross tonnage	6,151 tons
Speed	13 knots

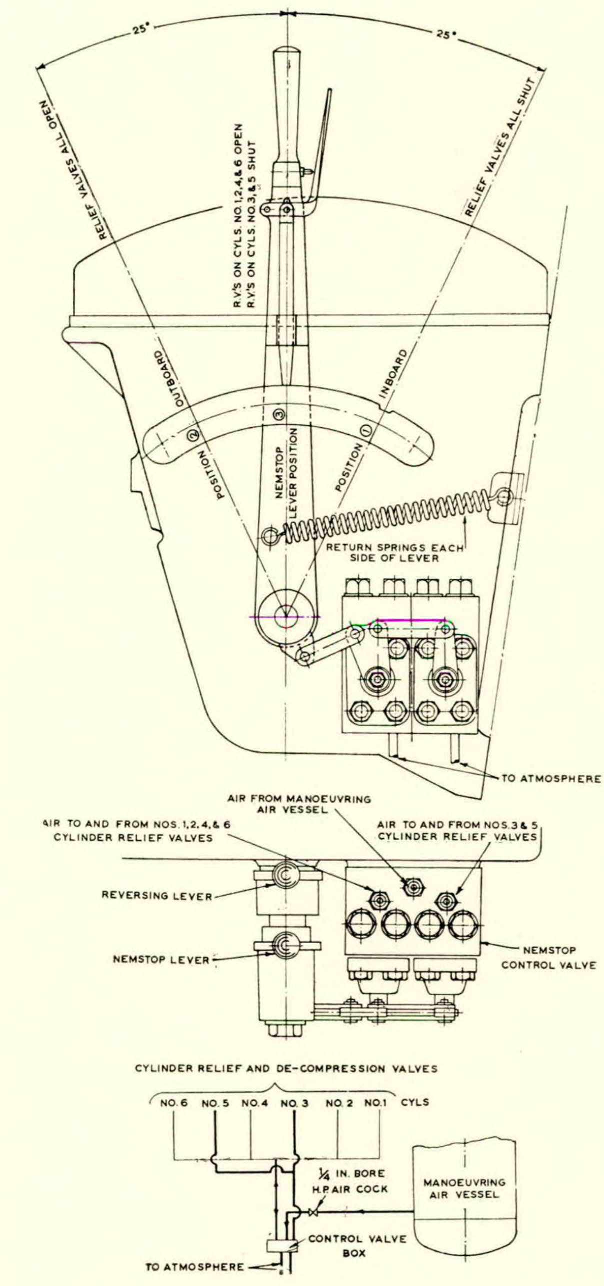
The *Glenpark* is equipped with a Marconi ultrasonic vibrator which passes ultrasonic waves along the ship's bottom and thus prevents the adherence of barnacles and marine growth. With this equipment fitted bottom painting is not necessary, and only a high-gloss paint is used. The *Scotstoun* was the first ship in the company to be thus equipped, and it is understood that she has now been at sea for over six months and is maintaining her original speed. It is hoped that this method of hull protection will enable the vessel's owners to dispense with docking their ships at frequent intervals, and permit a constant speed to be maintained. The *Glenpark* is powered by a four-cylinder diaphragm-type Doxford engine of 600-mm. bore and 2,320-mm. combined stroke, and having an output of 3,500 b.h.p. The lower pistons are oil cooled. The main engine has been built and installed by Barclay, Curle and Co., Ltd., Glasgow. Electricity for power and lighting is supplied by four Ruston and Hornsby Diesel engines, each coupled to a 175-kW generator. All the auxiliary equipment is electrically driven. An oil-fired Cochran auxiliary boiler supplies heat for domestic use.—*The Shipping World, 21st November 1956; Vol. 135, p. 457.*

Motor Ship with Decompression Braking System

Trials were held recently on the Newbiggin Mile, off the Northumberland coast, of the ore carrier *Tritonica*—launched as the *Tritonia,* but renamed since there proved to be already a ship of that name—a vessel of 19,495 tons d.w.c. on a summer loaded draught of about 31ft. $9\frac{7}{8}$in. This deadweight includes cargo, fuel, fresh water and feed water, stores and the crew and their effects. Built at the Sunderland shipyard of Sir James Laing and Sons to the order of the Dingwall Shipping Co., Ltd., of Canada, associated with Scandinavian Ore Carriers, Ltd., and Nordstrom and Thulin of Stockholm, the ship is managed by Mann and Sons, Ltd., London. Of the single-deck type built under the special survey of Lloyd's Register for their class 100 A1, strengthened for navigation in ice and in accordance with their latest rules for the shipment of bulk cargoes, the *Tritonica* has the following characteristics: —

Length b.p.	500ft. 0in.
Breadth, moulded	70ft. 6in.
Depth, moulded to upper deck	44ft. 3in.
Camber to upper deck and all erections and house tops	1ft. 6in.
Sheer at forward perpendicular (upper deck)	7ft. 6in.
Sheer at after perpendicular (upper deck)	1ft. 6in.
Rise of floor	$3\frac{1}{2}$in.
Deadweight	19,495 tons
Corresponding draught summer load	31ft. $9\frac{7}{8}$in.
Machinery	6,800 b.h.p.
Service speed	14 knots

The hull is longitudinally framed between the fore peak and the engine room forward bulkhead, while transverse framing is adopted for the ends and the side passages. The tank top longitudinals are formed of toe welded angles, closely spaced for strengthening purposes, the tank top plating being 18 mm. thick in the cargo holds and flush welded on the seams and butts to facilitate grab discharge. In the wing portion of the double bottom, above tank top level, and in the side passageways and topside tanks, transverse webs are fitted and spaced 12ft. apart, with intermediate vertical frames on the shell and longitudinal bulkheads extending to the underside of the topside tanks: longitudinal stiffeners are fitted to the boundaries of these tanks. The transverse bulkheads in the centre cargo hold portion of the vessel are of both longitudinal and vertical corrugated construction, while the longitudinal bulkheads—arranged to form passageways on each side of the ship between the double-bottom tanks and topside tanks—are of flat plate construction, as are also the wing tank divisional bulkheads. A feature of the engine, which is otherwise of the standard North Eastern-Doxford design, with diaphragm chambers and oil-cooled lower pistons, is that it is equipped with the new Nemstop braking system. This was first fitted on the motor tanker *British Vision* and has since been specified for a number of Doxford-engined ships. It is a system developed to overcome the difficulty in getting a large engine quickly over from ahead to astern owing to the heavy "drag" on the propeller. The method incorporates the use of partial decompression and comprises shutting off the fuel supply to the engine and allowing part of the air charge in the cylinder to escape through a special relief valve during the compression and expansion strokes. This gives a net braking effect during each cycle of engine operation, due to the fact that during the expansion stroke there is less air in the cylinders to expand and counteract the braking effect of the compression stroke. In the course of the acceptance trials a "crash-stop" was ordered and, by means of the Nemstop gear, the engine was brought to a standstill from 104 r.p.m. ahead in 13 seconds without the use of starting air. Within 30 seconds of the telegraph ringing for the crash stop, the engine was running astern at 60 r.p.m.—a very commendable result. A diagram of the Nemstop system is shown on this page and the operation of it is as follows: (1) *Engine running and normal manœuvring.* The Nemstop lever should be in position 1, when all relief valves are shut and function normally. (2) *Using single-stage Nemstop for stopping the engine.* The fuel is first shut off, then the Nemstop lever placed in position 2, whereupon all relief valves are held open and, due to the continuous escape of air on both compression and expansion strokes, a net braking effect is obtained. This quickly stops the engine even though the compression pressure which resists rotation of the engine is reduced. (3) *Using two-stage Nemstop for stopping the engine.* This is used when the engine has a tendency to crawl with the gear, as before, in position 2. When this occurs, the Nemstop lever is moved to position 3, when full compression is restored in cylinders 3 and 5, the immediate increase in resistance during the compression being sufficient to stop the engine. (4) *Use of the Nemstop gear for scavenging combustion spaces after long standby periods.* The usual procedure is to open the indicator cocks, and give the engine a turn on starting air. Nemstop gear can be used for this pur-



Diagrammatic arrangement of the Nemstop braking system pose by placing the lever in position 2 and applying starting air. The engine will then turn very slowly and a good scavenging action is obtained through the relief valves. (5) An additional advantage claimed for the gear is that the engine can be quickly stopped to prevent consequent damage should any defect develop in the running gear and services when running at full speed.—*The Motor Ship, November 1956; Vol. 37, pp. 290-294.*

Italian Bulk Ore Carrier

The motor ship *Giovanni Agnelli* is the third of six similar vessels under construction at the Cantieri Ansaldo, Spezia, Italy, for the Soc. di Nav. Carbosider, which is associated with the large Italian industrial Bibolini group. The fourth vessel, the *Giovanni Ansaldo*, has just been completed. The *Giovanni Agnelli* is classified with, and has been built to the rules of, the Registro Italiano Navale, Lloyd's Register of Shipping and the American Bureau of Shipping. The principal characteristics are: —

Deadweight capacity	15,800 tons
Gross register	11,230 tons
Length o.a.	165·46 m.
Length b.p.	153 m.
Maximum breadth	20·90 m.
Depth	12·20 m.
Draught fully loaded	8·60 m.
Machinery	5,500 b.h.p.
Speed	14 knots

The total volume of the cargo holds is 20,200 cu. m., or 713,262 cu. ft. There is a general cargo hold between frames 180 and 194, of which the capacity is 600 cu. m., or 21,186 cu. ft. Ballast water is carried in longitudinal tanks, also in double bottom tanks under the cargo holds, the tank space amounting to 6,007·7 cu. m., or 212,133 cu. ft. Fuel oil is carried in double bottom tanks under the machinery space, also in deep tanks, the total quantity being in the neighbourhood of 1,000 tons (38,200 cu. ft.). The machinery is installed aft and the propelling engine is a Fiat two-stroke unit of 5,500-b.h.p. service rating running at 120-125 r.p.m., the maximum output being 7,350 b.h.p. at 135 r.p.m. It has seven cylinders with a diameter of 750 mm., the piston stroke being 1,320 mm. It is arranged to operate on boiler oil and the exhaust gases are supplied to a La Mont boiler, which has a capacity of 600 kg. of steam per hr. at 7 kg./cm.^{2}; in addition, a Clarkson oil-fired boiler is installed supplying 900 kg. of steam per hr. at the same pressure. The steam is employed for heating the accommodation and the fuel and for the purifiers. Four 125-kW Diesel engined dynamos are provided for the current required throughout the ship and for the auxiliaries, the deck machinery, as well as the engine room pumps, being electrically driven. These pumps include the following: —

Two 260-ton lubricating oil pumps.
One 70-ton transfer pump.
Two 270-ton sea water circulating pumps.
One 270-ton spare pump.
One 90-ton fire pump.
One 130-ton general service pump.
One 10-ton refrigerating circulation pump.

The two centrifuges for dealing with the fuel oil have a capacity of 4,000 litres per hr. and that for the lubricating oil, 400 litres per hr.—*The Motor Ship, November 1956; Vol. 37, p. 312.*

Doxford Engines in 17-knot German Cargo Liners

The *Geestemünde* and *Cuxhaven*, the first of which was recently completed by Nordseewerke for the Hamburg-American Line's service to the West Coast of South America (although owned by the Bugsier Reederei und Bergungs A.G. Hamburg) are each equipped with a six-cylinder 8,600 b.h.p. Wilton-Doxford engine. Each has six cylinders, 750 mm. in diameter with a stroke of 2,500 mm. The ships are of 12,250 tons d.w. (9,390 gross tons) completed as closed shelterdeckers, but could be adapted to open shelterdeckers, in which case the capacity is 9,850 tons. They are also designed with special strengthening and may be employed as tramps for bulk cargoes at a later date. Each ship is equipped with 20 squirrel-cage a.c. Siemens-Schuckert cargo winches and alternating current is used for practically all purposes throughout the vessel. The cargo hold capacity totals 702,765 cu. ft. (grain). In service the engine will develop 7,740 b.h.p. and maintain 17 knots loaded, and it is recorded that the Doxford engine was chosen "because of the good service results of two Doxford-engined vessels acquired by the owners after the war". The oldest of them is the *Westsee*, purchased in 1951, but built in 1924 by Doxfords at Sunderland and named the *Westmoor*.—*The Motor Ship, November 1956; Vol. 37, p. 305.*

Coasting Motorship

After the completion of successful trials, the single-screw coasting motorship *Eskwater*, constructed by T. Mitchison, Ltd., of Gateshead-on-Tyne, has been handed over to her owners, the Eskgarth Shipping Co., Ltd., of London. The *Eskwater* has been built to the design of Messrs. Burness, Corlett and Partners, Ltd., and incorporates this firm's hydroconic system of construction. It is claimed that, by adopting this type of construction, there is a considerable saving in initial cost, and that the problems associated with repairs are simplified. The principal dimensions and other leading characteristics are given in the accompanying table: —

Length, b.p.	175ft. 0in.
Breadth moulded	30ft. 0in.
Depth	13ft. 5in.
Gross tonnage...	600
Deadweight, tons	868
B.H.P.	600
Corresponding r.p.m. ...	300
Service speed, knots	10

The vessel has been built in accordance with the requirements of Lloyd's Register of Shipping, while the latest recommendations of the Ministry of Transport are also implemented. Constructed and equipped for the carriage of timber cargoes, the *Eskwater* has been designed for coasting voyages. There is one hold, which extends from the forepeak bulkhead to the bulkhead forming the forward boundary of the main machinery space. The hold is suitable for carrying long lengths of timber or steel, and wood division boards are provided, so that the hold can be subdivided into smaller compartments, when necessary. The hold is served by three 3-ton derricks, two of which are supported on the main mast and one on the fore mast. There are two hatchways, each of which is fitted with hatch covers of the type developed by MacGregor and Co. (Naval Architects), Ltd. The covers at No. 1 hatchway are arranged for individual pull, while those at No. 2 hatchway are of the single-pull design. The *Eskwater* is propelled by a Crossley type-H.R.N. 8/30 Diesel engine, manufactured by Crossley Brothers, Ltd. The main engine is capable of developing 600 b.h.p. at 300 r.p.m. and giving the vessel a service speed of 10 knots, in the laden condition.—*The Shipbuilder and Marine Engine-Builder, December 1956; Vol. 63, p. 708.*

A New Cylinder Lubricant

A new Diesel engine cylinder lubricant Tro-mar DX 130 has been developed and with this is claimed a notable reduction in cylinder liner wear is achieved, when operating on residual fuels. It consists of a high quality mineral base oil incorporating special additives and is described as a dispersion type lubricant in which the additives incorporated, combating corrosive and abrasive wear, are carried as a stable dispersion in an oil base. No water is used in its manufacture. The consumption per b.h.p.-hr. is no more than with conventional cylinder lubricating oils. No alteration is required to the mechanical lubricator settings when changing over from one oil to another. Comparative service performances have been provided by the manufacturers, showing the cylinder liner wear reduction with various classes of engines, including Doxford, Fiat, B. and W., Harland and Wolff, and M.A.N. machinery; the wear reduction varies from 72 per cent to 90 per cent, although in one instance it was only 30 per cent, but this is because the comparison is with an alkaline detergent oil. Ring wear also appears to be substantially reduced. The figures given refer to operation under normal seagoing conditions. It is stated that, although primarily intended for use in engines operating on residual fuels, there is evidence of comparable performance in engines using marine Diesel fuel. The new lubricant may be stored in barrels or in bulk in the normal manner without deterioration, and may be regarded as a heavy conventional mineral oil and handled as such.—*The Motor Ship, December 1956; Vol. 37, p. 370.*

Bulbous Bows for Trawlers

Tests conducted at the Ship Division of the National Physical Laboratory with trawler forms have revealed that compared with good conventional hulls, substantial reductions in ship resistance can be achieved by the introduction of a bulbous bow. Between the speed-length ratios 0·80 and 1·30 the reduction in resistance in calm water is 10 to 20 per cent, depending on the designed speed. Additional experiments have been carried out to determine the seakindliness of these forms, by towing them head to sea through a series of waves generated in the tank. Pitch and heave have been recorded in wave conditions covering a range of wave lengths from 140 to 280 feet for a mean wave height of 6·0 feet on the full-scale. Ciné films, shot at full-scale simulation speed, have also been taken to observe the extent of shipped water and the wave behaviour at the forward end of these vessels. Reductions in pitching amplitude of about 10 per cent in heavy pitching conditions have been recorded in favour of the bulbous bow forms, the heave remaining unaltered. The *Cape Columbia* is the first of two sister ships which incorporate the bulbous bow. They have been built by Cook, Welton and Gemmell, Ltd., Beverley, Yorkshire, for Hudson Bros. of Hull. She has a registered length of 188ft. 6in. with a 32ft. 6in. moulded breadth, and 17ft. 6in. moulded depth. The gross tonnage is about 800 tons. The main machinery consists of a triple-expansion engine developing about 1,400 i.h.p. at 132 r.p.m. on loaded trials. Loaded speed trials were carried out on the Newbiggin measured mile, and the vessel has fulfilled the performance predictions from the model experiments, attaining a speed just short of 14·0 knots. A highly successful voyage has now been completed, and the results fully justify the confidence placed in her capabilities by those concerned. The innovation of the bulbous bow marks a significant step forward in the development of the trawler, and the experience gained in the handling of these ships and the service performance data accumulated will be keenly studied for future designs. As speeds increase the benefits of the bulbous bow will become increasingly important.—*The Marine Engineer and Naval Architect, January 1957; Vol. 80, p. 26.*

Direct Current Propulsion

Direct current drives for ships of various sizes and for a variety of purposes have been supplied to an increasing extent since about the end of 1950. The comparatively large number of equipments commissioned in so short a time indicates not only a considerable increase in the utilization of direct-current propeller drives, but also of electrical propulsion in general. The growing popularity of d.c. may be attributed to the facility for a wide control of speed, either manual or automatic, as for example with the Ward-Leonard system, the simplicity of which cannot be attained with alternating current. Sensitive and rapid speed regulation adjustable over a wide range practically without losses, combined with small and reliable control organs which can be installed anywhere in the ship, is one of the most valuable characteristics of direct-current propulsion. This is of the greatest importance particularly as regards manœuvrability. In connexion with the development and choice of machines for this Ward-Leonard propeller drive the differentially compounded Krämer generator was selected by the AEG as being particularly suitable. Its employment enables the control and regulating actions to be influenced in an advantageous and simple manner by the generator or the propeller motor in such a way that considerably greater propeller torque can be obtained without overloading the driving plant. By means of the Krämer machine the full available power is automatically utilized either at full speed with low thrust, or low speed and high thrust according to the ship resistance. During manœuvring the drop in propeller speed, which takes place automatically with increase of torque, ensures contact between the propeller and the surrounding water without excessive slip and at the same time prevents overloading of the Diesel engine. To obtain the full manœuvring effect, however, a definite time sequence is necessary which on the one hand ensures the most rapid manœuvring effect and on the other prevents too rapid an acceleration or retardation of the propeller. The Krämer machine fulfils this requirement ideally, in that its electrical characteristics provide the desirable time lag by means of which the times of reversal of the propellers are influenced. A further point of importance is that the propeller motor, which during reversal acts for some time as a generator, then maintains the direction of the current in the self-excited winding of the Krämer machine. The duration of this effect is the longer the more vigorous the manœuvre. It is thus possible to suit the speed of reversal of the propeller to the severity of the manœuvre, whereby the total time required for the reversal can be shortened. Both the change in torque and also the adjustable automatically controlled time lag are produced without any special controller elements, so that complete reliability of operation is attained. —*AEG Progress, No. 4, 1956; pp. 307-308.*

German-built Ship with Foreign Auxiliaries

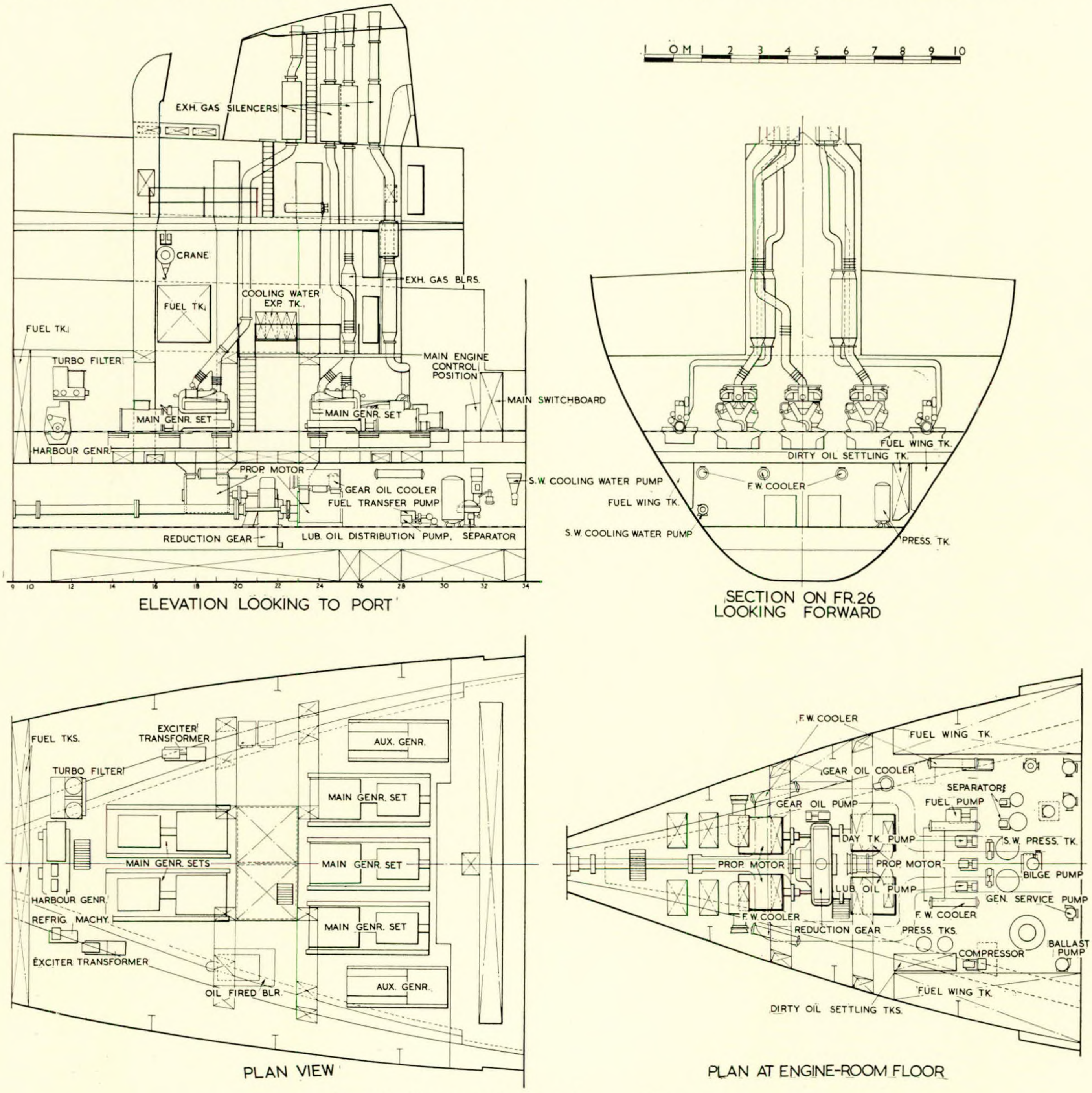
The *Frankrig*, recently completed by Rickmers Werft for A/F Det Dansk-Franske, D/S, Copenhagen, is an example of a ship built in Germany to foreign account, with much of the equipment from the country of ownership. She is of 5,820 tons d.w. with a length of 385ft. 10in. b.p. and is propelled by a 4,000-b.h.p. B.V.-M.A.N. engine, giving a service speed loaded of 14·65 knots. Among the foreign equipment are the three 150 kW Thrige generators, a Hi-Press heating and ventilation plant for the officers' and crew's accommodation, ten Thrige winches and two self-cleaning Titan Superjector separators, as well as a Titan Diesel oil separator and a lubricating oil separator. British equipment includes a Kelvin Hughes echo sounder, MacGregor single-pull hatch covers, Decca radar, and Walker Trident electric log.—*The Motor Ship, January 1957; Vol. 37, p. 429.*

German Diesel Electric Vessel with High Speed Engines

Although Diesel electric ships, engaged on short services for passenger carrying, have been constructed in Germany, the *Tinnum* is the first ocean-going cargo ship engined on similar lines. She is a vessel of 6,400 tons deadweight, equipped with five 12-cylinder vee-type 885 b.h.p. Daimler-Benz engines running at 1,300 r.p.m. Each drives a 600-kW direct-current generator. The *Tinnum* has been built by Werft Nobiskrug G.m.b.H., for Zerssen and Co. The *Tinnum* will be employed in her owners' services to the Mediterranean, the Gulf of Mexico and Florida. The ship has been built in accordance with the requirements of the highest class of Germanischer Lloyd and has the following main particulars:—

Length overall	113·75 m. (373ft. 2¼in.)
Length b.p.	103·50 m. (339ft. 7⅜in.)
Moulded beam	15·00 m. (49ft. 2½in.)
Depth to upper deck	9·45 m. (31ft.)

The *Tinnum* can be employed both as a closed shelterdecker and an open shelterdecker. The three cargo holds have MacGregor single-pull hatch covers on the upper deck and wooden covers on the second deck. Hallén bipod masts have been chosen for handling the derricks. There are 10 five-ton derricks, a 20-ton derrick and a 50-ton heavy-lift derrick. The wheelhouse has an Atlas-Raytheon 500 radar, an Atlas echo sounder, a Walker's Trident electric log, a Plath visual direction finder and a rudder indicator. A large control desk combines the Siemens-Anschütz self-steering installation and gyro compass with the main engine controls and indicators. There is also an optical cargo hold ventilation control board and an extensive wireless installation. The *Tinnum* is the largest vessel built in Germany with Diesel-electric direct current propulsion and with high speed engines placed on a platform deck above the propeller motors. As stated, the five main engines have been built by Daimler-Benz AG. They are 12-cylinder units in vee-form, having cylinders with a diameter of 175 mm. and a piston stroke of 205 mm. They are fresh water cooled and the working temperature is 40 deg. C. (about 104 deg. F.). The normal speed is 1,300 r.p.m., corresponding to an output of 885 b.h.p. They drive their own cooling water and lubricating oil pumps. Rubber mountings are placed between each Diesel generator set and the deck to decrease vibration. Noise-absorbing insulation, consisting of glass-wool, has been fitted to all the engine room walls and to the exhaust pipes, and the effect of this is stated to be very satisfactory. The main engines have exhaust gas turbocharging, BBC blowers having been fitted. They run at a speed of about 34,000 r.p.m. The



Engine room plans of the m.s. Tinnum

complete engines can be taken out for repair at the engine works when required and replaced by new engines, about 18-20 hours being needed for replacing an engine. The propelling machinery consists of four 694-kW Siemens motors having each an output of 900 b.h.p. at 1,200 r.p.m. and driving a single propeller through a Tacke reduction gear. The propeller speed is 125 r.p.m. at 3,600 b.h.p.—*The Motor Ship, December 1956; Vol. 37, pp. 356-359.*

Quick-opening Hatches on Ore Carrier

The *Gypsum Empress* and her sister ship, *Gypsum Duchess,* are recent additions to the Panama Gypsum Company's ore fleet. The *Empress,* first of these new 441-ft. (overall) ore ships of 10,677 deadweight and 5045 gross tons, was delivered in April 1956 by Deutsche Werft Shipyard, Hamburg, Germany. Equipped with 3,300-h.p. General Electric steam turbines and Foster Wheeler boilers, the *Gypsum Empress* and her sister ship cruise at 13 knots between Nova Scotia and various East Coast ports. Greer (Marine Corporation) hydro-hatch covers were installed on the seven hatches aboard each ship. This system permits all hatches to be opened automatically by pushbutton control in about five minutes. Constructed in three watertight sections, the covers measure 21ft. in length by 34ft. in width. In the open position two hinged sections of the cover fold upright and move aft with the exception of the folding sections on Hatch No. 7, which move forward. The remaining section on each individual hatch swings upright at the opposite end of the hatch. The Greer hydro-hatch system incorporates two hydraulic cylinders in each hatch-cover pontoon, or a total of six cylinders in each hatch cover. Hydraulic pressure of 3,000lb. per sq. in. actuates the covers. This pressure is developed in a power station located forward of No. 1 hatch. The power station comprises two electric-motor-driven 15-h.p. hydraulic pumps (connected in parallel), a bank of four accumulators (which retain the pressure for possible emergency closing), an oil reservoir, attendant piping, valves, controls, filters, etc. An additional bank of four accumulators is located aft in the fidley. Four control stations are provided which permit operation of the hatch covers by merely pressing pushbuttons. Watertight and with flush-mounted pushbuttons, these stations are so located that complete visibility of the covers is maintained during their operation. The pushbuttons control solenoid valves located in watertight control valve units mounted at the forward and after end of each hatch coaming. Actuated by the pushbutton, the solenoid valve admits oil under pressure to the proper ports of the two hydraulic cylinders located in each hatch-cover pontoon. One pushbutton governs operation of the single section (which always is opened first and closed last); a second pushbutton governs operation of the double-section cover. The design of the system permits operation of each hatch cover within two minutes for opening time and one minute for closing time. This is in addition to time required for unlatching or latching operations. All covers may be opened completely in approximately five minutes by use of both pump and motor units. Using only one pump all covers may be opened completely in seven to eight minutes. Any two covers may be opened completely in a period of only two minutes.—*Marine Engineering/Log, November 1956; Vol. 61, pp. 74-75.*

S.G. Iron in Tankers

This paper reviews the results of laboratory tests and service experience with S.G. iron pipe in tankers. Results or corrosion tests under various conditions are given in full, confirming general experience that S.G. iron has approximately the same corrosion-resistance as grey cast iron, and that in most cases its rate of corrosion is substantially less than that of steel. Moreover, with cast irons the corrosion rate tends to slow down with time, thus widening the differential between these materials and steels. The results reported in this paper cover tests in sea water under various conditions, and in distilled water, as well as in mineral and organic acids. The account of service tests in tankers, however, is of major importance. These tests have shown that after five years' service S.G. iron pipes are still in good working condition and that corrosion appears to have completely ceased. Under similar conditions steel pipes always require replacement. S.G. iron steam-heating coils, carrying steam at 150 deg. C., are absolutely intact, showing no trace of serious corrosion, whereas steel pipes in the same system have had to be removed on account of severe deterioration. Reference is made to the fact that forty tankers are now equipped with S.G. iron piping, and service reports are entirely satisfactory.—*M. Paris and B. de la Brunière: "Ductile Iron: Corrosion-Resistance in Sea Water and Petroleum-Tanker Service". Nat. Assocn. Corrosion Engineers, Preprint, March 1956; 13pp. The Nickel Bulletin, December 1956; Vol. 29, p. 217.*

Modernizing a Windlass

A conversion was recently carried out which may be of interest to those still using a gipsy chain drive from a winch to drive a windlass. This was undertaken on the m.v. *Kenrix,* owned by The Rix Shipping Co., Ltd., Hull, and subsequently arranged so that the windlass was driven by a 12-h.p. Armstrong air-cooled Diesel engine transmitting through 60:1 worm reduction gearing and by a system of chain and sprockets. The engine and worm gear were installed aft of the windlass, the drive being taken to the middle of the countershaft between the two centre cheeks. Both the power unit and reduction gear were mounted on a channel iron bed bolted to the deck for most of its length. The drive is by a triple chain to the countershaft sprocket, thence to the jockey sprocket. As it was not possible to fit the latter on the slack run of chain, the seating and adjusting gear for it are heavier than would normally be the case. Service experience has proved this arrangement most satisfactory, the only necessary precaution being to ensure that no attempt is made to adjust the chain with the winch in gear. The slow running speed of the Armstrong engine is said to enable the hoisting of the anchor into the hawse pipe to be undertaken very gradually, thus allowing ample time for the declutching of the engine before the anchor is in position.—*The Motor Ship, January 1957; Vol. 37, p. 395.*

Advanced Steam Conditions for Turbine Ships

The turbocharged marine Diesel engine is now being built in a power range of up to 12,000/13,000 s.h.p. and although not yet commonplace a number of such sets of machinery are in service and on order. Single-screw engines of this type burning low grade fuel are being installed in fast cargo ships, which have hitherto required twin-screw machinery. Turbine installations of comparative power cannot compete with such Diesel plants in terms of fuel economy, but they may, however, prove to be less expensive in overall running costs within the next few years if labour and material charges continue to rise. There is room for further improvement in the fuel consumption of turbine driven ships and the figure of 0·5lb. per s.h.p. per hr. should be considered a maximum. The following machinery design proposal for 12,500 s.h.p. is suggested as being well within the capabilities of the present day engine builder. The proposed steam conditions are 1,000lb. per sq. in. (abs.) and 1,000 deg. F. at the superheater outlet, with reheating to 1,000 deg. F. Three-casing turbines would be employed and reheating would take place between the h.p. and i.p. cylinders. The steam conditions at the h.p. turbine stop valve would be 975lb. per sq. in. (abs.) and 990 deg. F., exhausting to the reheater at 300lb. per sq. in. (abs.) and 650 deg. F. This would give an effective heat drop of 123 B.t.u. per lb. of steam (h.p. turbine efficiency—78 per cent). After reheating, the steam would enter the i.p. turbine at 300lb. per sq. in. (abs.) and 1,000 deg. F. and finally exhaust from the l.p. turbine to the condenser at a vacuum of 28·5 in. Hg. The effective heat drop through the i.p. and l.p. stops would be 474 B.t.u. per lb. of steam (l.p. turbine efficiency—88 per cent).

The total turbine heat drop would thus be 597 B.t.u. per lb. of steam. Hence the steam consumption for 12,500 s.h.p. (assuming a 2 per cent gearing loss) will be 54,400lb. of steam per hour. The steam consumption of the auxiliary machinery is taken to be 5,300lb. of steam per hour (see detail following) and the total boiler output required would be 59,700lb. of steam per hour at 1,000lb. per sq. in. (abs.) and 1,000 deg. F. The total heat required is thus:—

Boilers—	
59,700lb. per hour at 1,504 B.t.u. per lb. ...	$89{\cdot}78 \times 10^6$
Reheater—	
54,400lb. per hour at 183 B.t.u. per lb. ...	$9{\cdot}95 \times 10^6$
	$99{\cdot}73 \times 10^6$

Assuming a net calorific value of the fuel to be 18,000 B.t.u. per lb. and a boiler efficiency of 88·5 per cent, the total fuel consumption would be 6,250lb. per hour, 0·5lb. per s.h.p. per hour. The highest gains in cycle efficiency are obtained by raising the initial steam temperature. The present practical limit is 1,200 deg. F. and this temperature if used would considerably improve upon the figure of 0·5lb. per s.h.p. per hour. It is doubtful, however, if marine engine builders will recommend such a step until more experience has been gained with temperature-resistant materials and until epicyclic reverse-reduction gearing can be used to dispense with the need for an astern turbine and its corresponding reduction of steam temperature. The advantage obtained from the use of higher initial pressures is the higher saturation temperature obtained which reduces the superheater load and improves the efficiency. Reheating improves the efficiency and reduces the wetness of the steam in the later stages of the l.p. turbine and hence reduces trouble due to blade erosion.—*F. A. Powell, The Marine Engineer and Naval Architect, Annual Steam Number 1956; Vol. 79, pp. 407-408.*

Shipyard Use of the Analog Computer

This paper describes how the analog computer available today may be used to advantage as a tool to assist in the study and solution of engineering problems encountered in shipbuilding. The theory of operation of the analog computer is outlined briefly, as is the method of setting up a problem for computer study. An illustrative case of the application of the analog computer to marine engineering problems is an investigation made of the performance of a special type of condenser operating with a boiler and automatic regulating valve to control boiler pressure under certain abnormal operating conditions. Results obtained included plots of flow through the valve and condenser level, as functions of varying boiler heat inputs, and plots of valve opening against time with a certain change in boiler heat input. A calculation also was made concerning a system of two similar heat exchangers, each with its own turbine driven pump, and with the pump suctions and discharges and the heat exchanger outlets cross-connected. What was desired here were the flows in all parts of the system when one pump slowed down or stopped, and the determination of an effective means for detecting that one pump had stopped, without utilizing a mechanical connexion to the pump turbine shaft. The analog computer also has been used in the calculations involved with nuclear-radiation-shielding design and evaluation, in the study of various types of propulsion turbine throttle controls, and in a great number of other allied fields, such as submarine diving-plane-control-system design and the associated hull hydrodynamics simulation, as well as shaft vibration problems.—*Paper by J. R. Hunter, read at the Annual Meeting of The Society of Naval Architects and Marine Engineers, on 15th November 1956.*

Cleavage Strength of Steel

A certain property of steel called the "nominal cleavage of strength" can be determined by means of tensile test pieces on the edges of which a brittle alloy has been welded. On exposing these test pieces to an increasing tensile stress, cracks are continuously formed in the brittle layers of welding. At a certain critical stress a fracture will be initiated at one of the sharp crack fronts. A characteristic property of a steel, the nominal cleavage strength is defined as the highest nominal tensile stress a steel can withstand in the presence of a crack without the initiation of fracture at the crack front. The nominal cleavage strength decreases continuously as the temperature decreases, just as the conventional yield point increases continuously with decreasing temperature. If curves of these strengths are drawn, the point of intersection would appear to be a transition temperature for the steel, defined as the temperature above which plastic deformation can occur in a large volume of material despite the presence of a crack, but below which only minor local plastic deformation is possible immediately in front of the crack. By means of the testing method called "NC-testing" (NC = nominal cleavage) results have been obtained that show a certain correlation to the results of impact tests as well as to those of other brittle-fracture tests, e.g. Tipper, van der Veen, and others. Moreover, the critical stresses for initiation and propagation of brittle fractures are shown by means of NC-curves as well as the influence of residual welding stresses. It is also shown that the quality of a steel with regard to its brittle-fracture tendency cannot be simply expressed by an impact transition temperature, as the critical stresses quoted above are not directly connected with the position of this temperature.—*Paper by T. M. Norén, read at a meeting of the North East Coast Institution of Engineers and Shipbuilders on 30th November 1956.*

Pitch Distribution of Wake-adapted Marine Propellers

This paper contains the further development and completion of some conceptions on single-screw ship propulsion as introduced in the author's paper, "A Simplified Method for Preliminary Powering of Single Screw Merchant Ships". After a brief re-evaluation of the principle of the "substitute propeller" as presented in that paper, and a more elaborate derivation of the relations between the thrust-deduction and wake factors in terms of propeller loading, the present paper shows how these relations can be integrated over the propeller radius to obtain an elegant solution of the problem of the radial distribution of thrust deduction as a function of the radial wake distribution and propeller loading for propellers of minimum energy loss. It is then shown how this solution can be applied to the basic definition of radial pitch distribution for optimum propellers under all conditions of loading and radial wake distribution. Thereupon, a simple rational formulation is proposed for the hydrodynamic pitch angle, corrected for induced velocities, as a starting point for the circulation-theory design method of wake adapted propellers of minimum energy loss. It is finally shown that this formulation yields radial pitch distributions in very close agreement with those derived from the substitute propeller concept, and that it may be accepted as a definite solution in a field where hitherto rather conflicting conceptions have prevailed. In an appendix, the basic design of controllable pitch propellers is discussed in terms of the considerations presented in the paper.—*Paper by L. Troost, read at the Annual Meeting of The Society of Naval Architects and Marine Engineers, New York, 15th November 1956.*

Optimum Propeller with Finite Hub

The determination of the bound circulation for an optimum propeller was initially performed by Goldstein who was able to relate the circulation distribution of a propeller with a finite number of blades to one having an infinite number. Goldstein's solution was, however, performed for the case of a propeller having a zero hub diameter and when the vortex sheet extends to the propeller axis. Such a solution is considered sufficiently accurate for propellers with a hub diameter which is relatively small compared with the propeller diameter; in this case, the presence of the hub is assumed to have little effect on the circulation distribution along the radius. However, the increasing use of propellers with relatively large hubs has emphasized the need of a solution for such cases. The problem was originally investigated by Lerbs who, through the use of corrective "induction factors", was able to consider the presence of an infinitely long hub. A solution to the potential problem has also been given by McCormick for the effect of a finite hub but with simplified boundary conditions given at the hub radius. The effect of such assumptions is to produce a discontinuous change of circulation at the hub which is contrary to available experimental evidence. The problem is solved for a propeller having a minimum energy loss and for which the flow far behind the propeller can be considered to be the same as that formed by rigid trailing vortices moving backwards at constant angular velocity. The hub is assumed to be of constant diameter and extending from the propeller plane to infinity. The effect of the hub has been calculated for specific cases, showing that it becomes important for propellers with large hub diameters and small number of blades and increases with increasing pitch.—*A. J. Tachmindji, David W. Taylor Model Basin, Report 1051; 1956.*

Largest Swedish Icebreaker

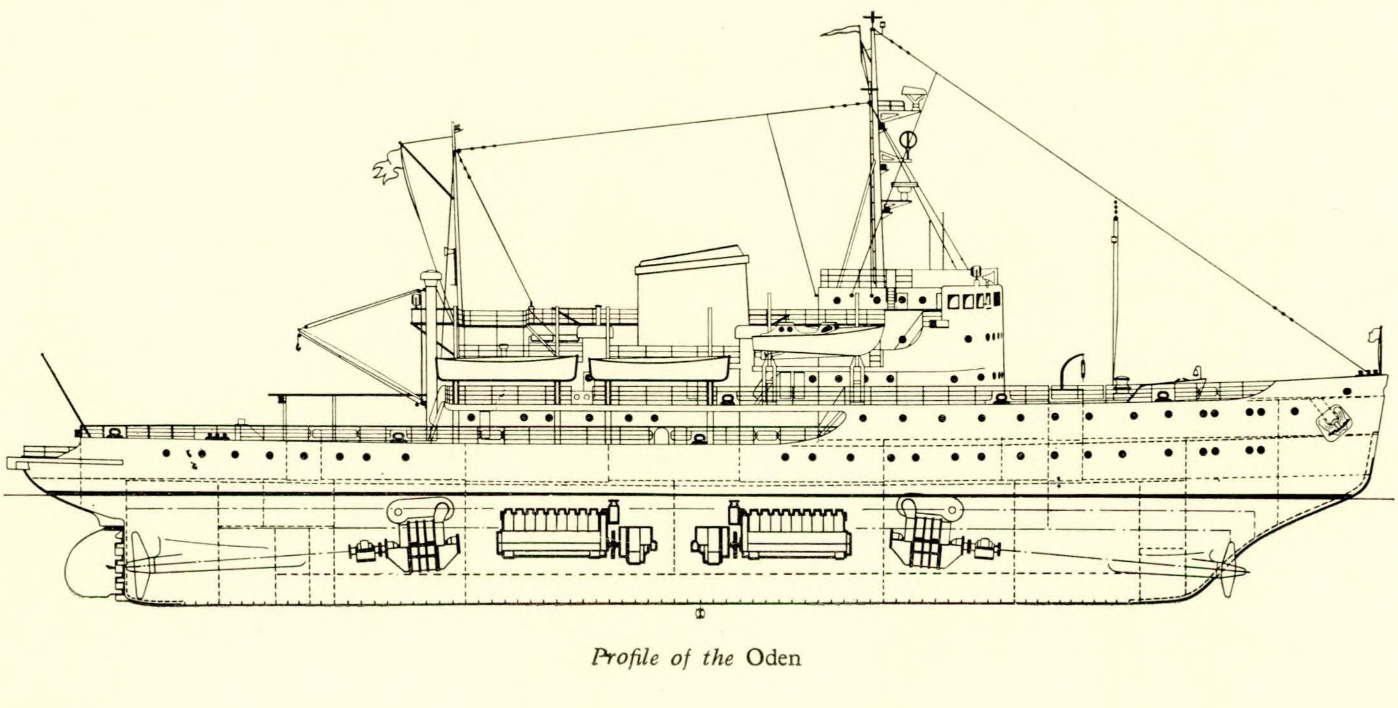
The icebreaker *Oden,* which was recently launched by the Wärtsiläkoncernen, Helsingfors, will be the largest and most powerful Swedish icebreaker. She is being built to the order of the Swedish Government and will be placed in service off the coast of Sweden. The main particulars are:—

Displacement	4,950 tons
Gross register	4,000 tons
Length overall	83·22 m. (273ft.)
Length on waterline	79·71 m. (260ft.)
Maximum beam	19·4 m. (63ft. 9in.)
Machinery (normal)	10,500 b.h.p.
Draught, mean	6·69 m. (22ft. 4in.)

There are four machinery compartments. In Nos. 2 and 3 amidships are the six Diesel-generators (three in each compartment), whilst in each of the machinery spaces Nos. 1 and 4 are two propulsion motors. The generating sets comprise nine-cylinder two-stroke Polar engines constructed by Nydqvist and Holm, coupled to d.c. dynamos running at 325 r.p.m. Each engine develops 1,700 b.h.p. normally, with a maximum output of 2,000 b.h.p. at 375 r.p.m. for six hours. The *Oden* will carry 2,062 tons of oil and water, including 850 tons of fuel and 600 tons of ballast water. The crew totals 73 men. The hull is divided into 10 watertight compartments. It is anticipated that the ship will be completed in December 1957. —*The Motor Ship, December 1956; Vol. 37, p. 362.*

Scoring Characteristics of Metals

This paper is concerned with the score resistance of elemental substances, chiefly metals, in high-speed sliding contact with steel. By score resistance is meant the ability of materials to slide against each other without welding or otherwise sticking together. The terms "galling", "seizing", and "metal transfer and pickup" are sometimes used in technical literature as synonyms for scoring. Score resistance is a property of great practical importance in bearings, piston rings, shaft seals, gears, and indeed all parts that rub together. Although it is property that has been studied for many years, very little is actually known about it, and what is known is entirely empirical and frequently contradictory. A bearing journal combination is an exceedingly complex thing. There are at least nine different layers of materials to be considered. Starting with the bearing metal, we have first a layer of oxides and, depending upon the character of the lubricant, possibly sulphides, chlorides, and other materials. Over these there may be a layer of soap, produced by reaction between the metal oxides and organic acids from the lubricant. Over this there may be an absorbed layer of oriented polar molecules from the lubricant. Finally, there is the lubricant itself. The thickness of these various layers is extremely small. The oxides and soap layers may each be of the order of a hundred Angstrom units, while the layer of oriented polar molecules is at most only a few molecules thick. The thickness of the lubricant layer may range from as large as a few thousandths of an inch to as small as a few millionths of an inch or less. As we move on toward the journal, we encounter the same layers in reverse order—first the oriented polar molecules, then the soap film, then the oxide layer, and finally the steel of the journal itself. Although a complete description of the events involved in the scoring failure of a bearing would take account of each of these layers, the most important aspect of the problem is what happens when bare metal slides against bare metal, because the layers of oriented polar molecules and soap films offer only weak resistance to penetration, while the oxide layer is un-



Profile of the Oden

doubtedly broken up and stirred about, exposing areas of bare metal, when the bearing and journal slide against each other at high speed and under heavy load. It is generally held by physicists that all metals will adhere to each other when actual metal-to-metal contact occurs. However, as the slider-bearing experiments show, this adhesion may range from relatively strong to relatively weak. In fact, the score resistance of a pair of metals is simply a measure of the ease with which their adhering surfaces may be separated by shearing and of the location of the plane of shearing with respect to the interfacial plane. With metal pairs that are soluble, scoring and welding readily occur. This suggests that at points of metal-to-metal contact there is diffusion or solution of one metal into the other. If the solid journal metal is soluble in the molten bearing metal, the resulting junction, when solidification occurs, will be an alloy of the two metals, and this will tend to give a strong weld. On the other hand, if the solid journal metal is not soluble in the molten bearing metal, diffusion of journal atoms into the molten bearing metal will be on a greatly reduced scale. Such a junction will be weaker, all other things being equal, than the junction between miscible metals.—*A. E. Roach, C. L. Goodzeit, and R. P. Hunnicutt. Transactions A.S.M.E., November 1956; Vol. 78, pp. 1,659-1,667.*

Cathodic Protection in the U.S. Navy

This is a comprehensive paper on the progress of cathodic protection in the U.S. Navy. It deals with the technical problems associated with the application of cathodic protection to ships, how many of these problems are being solved, and what further refinements are necessary. The scope of the Bureau of Ships' Research and Development Program of Cathodic Protection is presented in detail, summarizing the results of many experiments. Latest advances in materials, components, and systems are given and early prototype shipboard installations are discussed. A general guide to design of cathodic protection for ships is included, together with a more detailed outline of a step-by-step procedure in the appendix. Several aspects of the impact of cathodic protection on the marine industry are presented as well as expected results and a prediction of progress in the future. Rapid strides expected in the improvement of materials and equipment will result in reliable performance with a minimum of manual operation and maintenance. In the opinion of the authors, cathodic protection in the future will become integral with ship design.—*Paper by D. P. Graham, F. E. Cook, and H. S. Preiser, read at the Annual Meeting of The Society of Naval Architects and Marine Engineers, 15th November 1956.*

Local Vibration

In considering the diagnosis and cure of local vibrations, several important points have to be borne in mind. First, it is necessary to determine the extent to which main hull vibration is involved. Upon this will rest the decision as to whether the vibration can be reduced to an acceptable level by local stiffening or whether it is necessary, having located the source of excitation, to take such measures as are used to cure main hull vibration. These cover such methods as reducing the magnitude of the forces, neutralizing the forces or altering their frequencies. In this respect, however, it must be remembered that the higher mode hull vibrations tend to have progressively flatter resonant characteristics, in addition to which the critical frequencies may become more closely spaced. Thus, any attempts to avoid vibration by altering the frequency of the exciting forces becomes more difficult in the higher frequency ranges. When introducing stiffening to cure a local vibration, the effect is to raise the natural frequency of that part of the structure to a higher value than that of the exciting forces and so avoid resonant conditions. When the "amplitude/frequency" response characteristics and the pattern of the vibration have been clearly identified it is sometimes possible to make a reasonable estimate of the scantlings of the additional structural members that will give the required diminution of amplitude. In such cases it is desirable to introduce restraints at the positions of maximum vibration amplitude where the most effective use of material can be made. It has been observed that the situation can arise when the application of corrective measures to reduce one type of vibration can act adversely in other directions. Problems often arise that lend themselves to more accurate analysis. In considering propeller blade excited vibrations, for example, measurements on the hull may indicate that the amplitudes would be diminished by replacing a 4-bladed propeller with one having five blades. Here, however, the periodic forces acting on the propulsion system are involved and it would be necessary by measurement or calculation to determine the appropriate critical frequencies to ensure against resonant conditions before changing the propeller. The trends in design towards larger ships with more powerful propelling machinery may be expected to present new problems and to accentuate existing ones. On the problem of propeller blade excited vibrations in large single-screw ships, for instance, the authors have in mind that the time may be approaching when the traditional type of propeller aperture has to be discarded in favour of the "clear water" type. It is interesting to note than in the high-powered *Mariner* class of ships the clear-water stern arrangement was adopted after careful study which included considerations of vibration. This type of design permits greatly increased all-round clearances to be adopted and thus enables not only the hydrodynamic forces on the adjacent hull to be kept to a minimum but also those on the forward part of the rudder. In the conventional type of aperture, measurements on various ships have suggested that the pulsating hydrodynamic forces on the forward part of the rudder or fin are of such magnitude that they can give rise to serious vibration. In such a structure the whole of the rudder assembly and the fine after sections may act virtually as a cantilever appendage to the main hull. The resulting vibration will be related to the frequency of this structure and to the frequency and magnitude of the applied forces. The present trend is for the frequency and magnitude of the applied forces to increase with increasing powers and for the frequencies of the structure to diminish as the size increases. Until far more experimental data are obtained, the results of such changes are partly conjectural. Recent experience suggests, however, that as in other phases of vibration, resonant conditions will appear more frequently unless suitable measures are taken. While propeller-excited vibrations are fairly common in multiple-screw ships, it is nevertheless possible to design for large bossing and tip clearances and the arrangement does not suffer from the apparent limitations of the conventional single-screw type design.—*Paper by A. J. Johnson and P. W. Ayling, read at a meeting of the Institution of Engineers and Shipbuilders in Scotland on 20th November 1956.*

Investigation of Process of Expanding Boiler Tubes

The objectives of this work were to examine the tube expanding process in relation to as many variables as possible and to determine whether there would be any significant "size effect". For these investigations an apparatus has been constructed which enables detailed observations to be made of the behaviour of a specimen tube-plate joint during expanding. A typical record is given showing how the strains and mandrel torque develop during expanding. No appreciable size effect was found on the three geometrically similar systems tested. It is shown that the residual strains are much greater on the back than on the front of the plate, leading to lack of parallelism in the joint face, and that the plastic-elastic interface can extend much farther into the plate than was previously thought possible. Comparison is made between these experimental results and the predictions of a theory proposed by the authors, and it is concluded that higher seat pressures could be obtained if axial extrusion of the tube could be restricted. This finding was confirmed by axially compressing short lengths of tube into a plate, the axial compression causing the tubes to bulge and expand themselves into the plate seat.—*Paper by J. M. Alexander and H. Ford, read at a meeting of the Institution of Mechanical Engineers on 4th January 1957.*

Nuclear Reactor for Tanker

Ford Instrument Company, Division of Sperry Rand Corporation, will make an engineering study to investigate the economic and technical feasibility of putting a nuclear-powered main-propulsion system using closed-cycle gas turbines into a 707-ft. long supertanker with a capacity of 38,000 dwt. The previous AEC contract with the company called for a study of a closed-cycle, gas-cooled reactor for shore-based power. The second contract involves an extension of this work with emphasis on a different application. Tanker operators have shown considerable interest in this application, since it would not require, as do conventional tankers, depletion of its oil cargo while en route in order to fuel its propulsion plant. The resulting economy would be substantial. Generally, the closed-cycle, gas cooled concept appears to offer great used in all closed-cycle plants to date. The two low pressure (or power) turbines drive the propeller through a reduction gear train, and permit reversing of the ship by adjustment of the turbine guide vanes. The turbines exhaust directly into the recuperators. The recuperators are twin units so arranged as to allow for maximum space savings in the machinery areas. The precoolers are built as integral units with the recuperators. They serve to restore compressor inlet-temperature conditions. The steam generators use the hot exhaust from the recuperators to generate steam for driving pumping machinery, for cargo and space heating, and, in port, for cargo pumping.—*Marine Engineering/Log, November 1956; Vol. 61, pp. 68-69.*

Pitch Reduction with Fixed Bow Fins

Considerable attention has been given in recent years to the general problem of the motion of ships in both regular and irregular seas and to the operating limitations imposed on ships by such motions. The importance of reducing the pitching motion has been recognized and some attempts at solution have been initiated. It is, indeed, possible to design

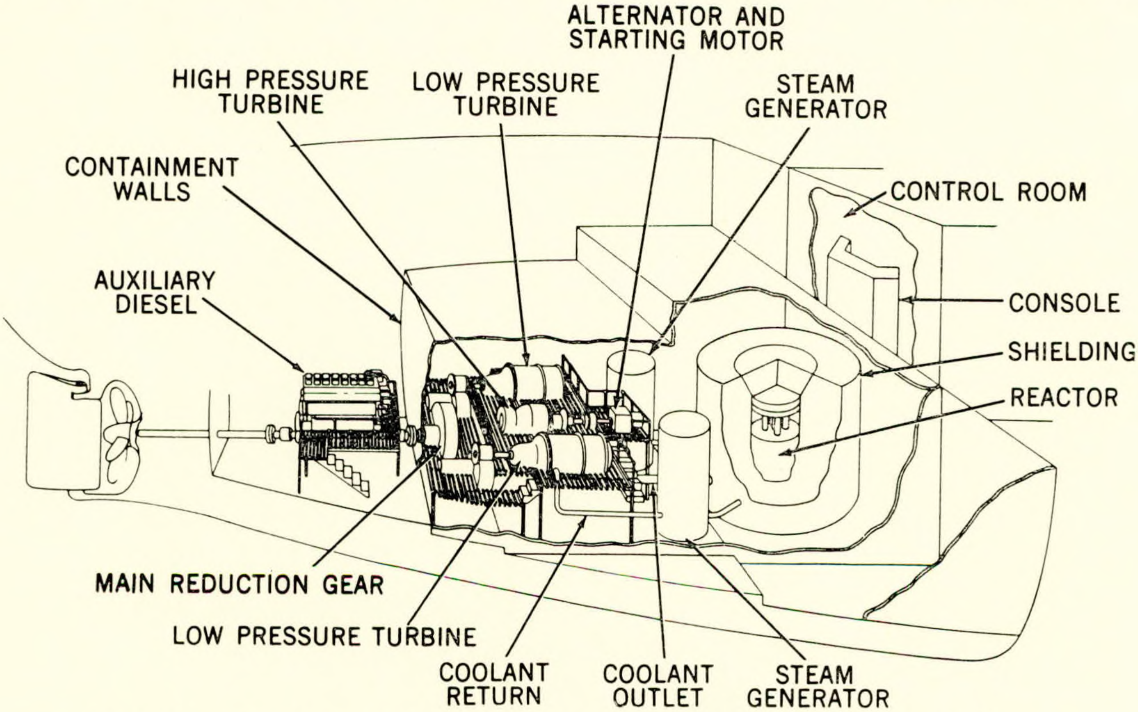


FIG. 1—*Nuclear plant layout as understudy for installation in a 38,000-d.w. tanker*

promise of economical power generation in the foreseeable future because it is expected to be simple, safe and to have low installation and operating costs. Most other reactor concepts now under study require the reactor to heat water or liquid metal which is then pumped through a heat exchanger. There the heat energy converts water to steam which in turn drives a steam turbine. The closed cycle, gas cooled reactor differs in that it heats an inert gas which drives a gas turbine directly. The major components of the proposed nuclear propulsion system are enclosed within the containment walls, as illustrated in Fig. 1. The gas cooled nuclear reactor is comprised of a pressure vessel with a cover which can be removed for refuelling. The cover seals the vessel and supports the control rod drive mechanisms used in control of the reactor. The reactor core, which contains the fuel elements, moderator section, control rods, and cooling ducts, is designed for efficient heat transfer and economical use of fuel. The compressor-turbine set is arranged in a single case that also incorporates the intercooler between the low pressure and high pressure compressor sections. The high pressure turbine and the axial-flow compressors of this set are of types which have been

a hull form having relatively high damping characteristics; however, such form would deviate significantly from what is known as a normal form. On the other hand, it is generally recognized that only radical deviations from normal form are sufficient to bring about a significant decrease in the motion. One possibility remains—that of increasing the damping characteristics of the ship by means other than modifications in the basic ship form, for example by the use of fins, or wing appendages, located at selected points along the hull. Some experimental work has been carried out in this direction. A noteworthy full-scale application has been the installation of anti-pitching fins on the Holland-America Line vessel *Ryndam*. The *Ryndam* experience has been somewhat unsatisfactory and discouraging. It has been reported that the ship suffered greatly by transverse vibrations attributed to the fins. These were installed at the bow of the ship not far below the static load waterline. After two days' operation they had to be removed before the vessel could continue her journey. Model test results, on the other hand, have given promise of large reductions in pitching amplitudes through the influence of anti-pitching fins, and further investigations were thought advisable despite the *Ryndam* experience. The Taylor Model Basin has been active for some time in the study of anti-pitching fins. The present report describes a series of tests in waves with a 10-ft. self-propelled model representing the 0·60 block coefficient, Series 60 parent form with and without anti-pitching fins. It is emphasized that scaling up of the data pertaining to the model with fins may be accompanied by serious errors resulting from possible scale effects which as yet have not been determined. The test results indicate that reduction of the pitching motion of ships can be effected by means of fixed anti-pitching fins. Further theoretical and experimental investigations are considered necessary before the useful range and limitations of anti-pitching fins can be adequately established. The pitch reduction attributable to the fins considerably improves dryness of the model in head seas. The practical speed range, as restricted by motions of undesirable magnitude, is also extended. Forefoot and forebody emergence occurring during the tests without the fins were not observed when the fins were installed.—*U. A. Pournaras, David W. Taylor Model Basin, Report 1,061, 1956.*

Automatic Control for Naval Boilers

Of prime consideration to the type of combustion control equipment applied to a boiler is the type of oil burner to be used. Original test work and early experimental work in combustion control for U.S. Navy boilers has established that for a combustion-control system to do the job it is designed to do, the oil burners must have a wide turn-down range so that it is not necessary for operators to be changing burners constantly on small load changes. Straight mechanical atomizing burners have a range that is no greater than the square root of the ratio of minimum to maximum pressure. A straight mechanical burner with a maximum of 1,000lb. per sq. in. and a minimum of about 100lb. per sq. in. would have no more than about 3½ to 1 turn-down and could not be considered satisfactory for good operation without excessive operator attention. A number of burner manufacturers have developed burners that have wide-range characteristics; that is, turn-down ranges from 7 to 1 to as high as 20 or 30 to 1, the higher turn-down ranges being achieved by some form of external assistance, such as steam atomizing or high pressure air. Burner turn-down range also can be extended by using a return-flow-type burner with a number of variations as to the arrangement of supply and return control. At the present time, this is the preferred type of burner on naval vessels. The early return-flow burners were provided with constant oil pressure at the burner supply header, and all control was obtained by a valve in the return line. While satisfactory performance was obtained, the oil pumping costs were high, larger coolers were required in the return line, and wear on the sprayer plates was excessive. One modification to alleviate these problems was a burner designed for a constant differential pressure between the supply and return headers, and another was a burner designed for a variable differential. Most return-type burners have critical pressure-flow characteristics, and care is necessary in selection of the proper control. A typical example of control for a variable differential-type burner is shown in Fig. 4. Experience in controlling oil burners showed that each different type of burner, and even different size burners within a general type, raised problems that are unique and individual. For example, one size of variable-differential, return flow burner was controlled fairly simply with a direct, self-powered, variable-differential valve in the supply header and a pneumatic control valve in the return header. Another burner of the same general type but of greater capacity required the substitution of an air operated, supply control valve for the direct variable-differential valve. Operation showed that many of the characteristics were not predictable and that final burner control adjustments had to be worked out during actual boiler firing operations.—*Paper by C. H. Barnard and H. D. Vollmer, read at the Annual Meeting of The Society of Naval Architects and Marine Engineers, New York, on 15th November 1956.*

Air Conditioning of Modern Tankers and Cargo Ships

For comparing the provision of air conditioning versus ventilation, a large supertanker with a crew of fifty-six was selected, the ventilation installation being of the very highest grade, with salient features as follows: In the midships house 9,000 c.f.m. of supply air and in the aft house 28,000 c.f.m. of supply air hold the temperature down to 8 deg. F. above the

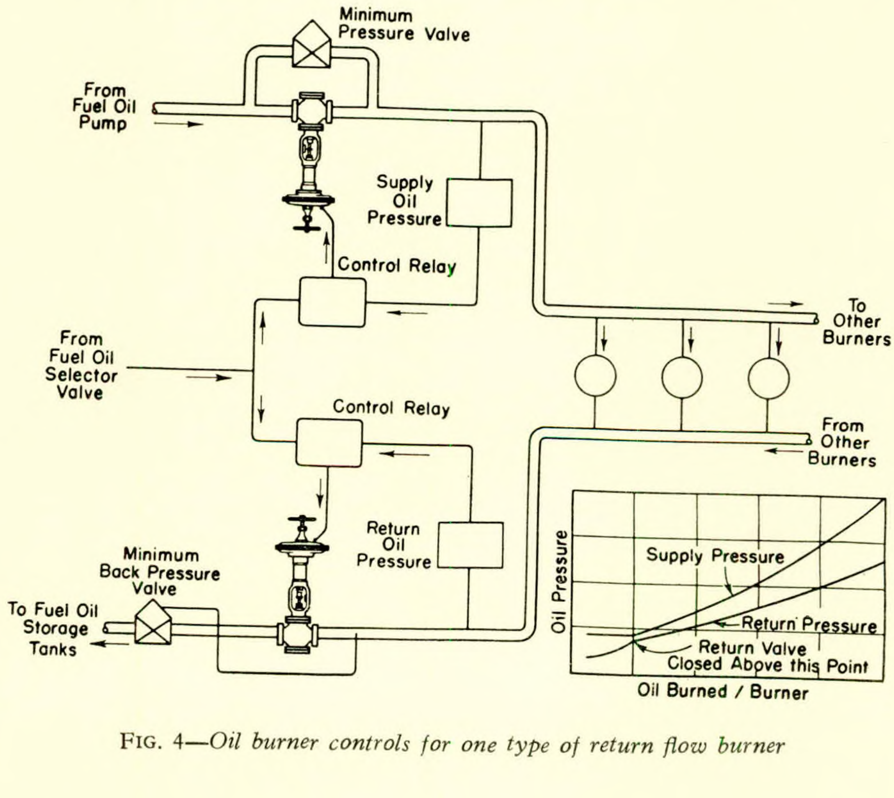


FIG. 4—*Oil burner controls for one type of return flow burner*

outside air. This is far better than the average tanker installation, but no ventilation installation can improve on outside air conditions, and with 95 deg. F. outside the inside temperature will be 103 deg. F. On this vessel several package air conditioning units are located in the various public spaces. The air conditioning proposal for this tanker is as shown in Fig. 17 and is based on the following design points: Retain normal supply ventilation with an 8 deg. F. temperature rise in the galley, pantries, laundry, butcher shop, clean linen lockers, drying rooms, and a storeroom. Provide air conditioning for all other living spaces to maintain 85 deg. DB, 71 deg. WB, 78 deg. F. effective temperature, with outside air at 95 deg. DB and 82 deg. WB. Provide one freon compressor plant in the amidship house; accept the hazard of a breakdown, and under that condition operate the supply fan as a ventilation fan which will hold inside conditions to a 15-deg. temperature rise. The complications and expense of two half-size compressor plants do not appear to be justified. Provide two freon compressor plants in the aft house, cross-connected so that with one plant broken down the other plant can serve the entire aft house. No recirculation of cooled air; the entire amount is exhausted overboard through exhaust systems or by natural exhaust through passages and up stairways. Thermostatic control is provided for the three cooling coils only. Each stateroom will have a manual damper in the supply duct; if too cool, the occupant can shut down on the supply damper. There is no change in concept for winter operation, which will be the same as on the ventilated ship. Room heating will be obtained by opening the manual control of the steam convector in the room and throttling by damper the supply of outside air. There are no outside air controls; the fans handle the same quantity of air at all seasons. The main physical changes from the ventilated to the air conditioned ship would be as follows: Add two freon compressors, condensers and receivers in the engine hatch at boat deck level, with necessary foundations, access gratings, and condenser cooling water. Add a new machinery room in the amidship house, cofferdammed about 16in. above the weather deck, in which one compressor, condenser, and receiver can be located; provide condenser cooling water from the firemain. The fans remain in the fan room three decks above. Eliminate the package air conditioning units. A direct-expansion cooling coil and filter bank is added in each of the three fan rooms. A thermostatic control is added for each cooling coil. Fan and duct sizes are all greatly reduced. At 95 deg. F. outside, on the ventilated ship enough air will be moved to maintain an 8 deg. F. temperature rise. At 95 deg. F. outside, the air conditioning system moves enough 65 deg. F. air to allow a 20 deg. F. rise to 85 deg. F. The fans and ducts for the latter system need be only 8/20 of the size of those for the ventilated ship. The outside covering on the insulation for the supply ducts will be vapour sealed on the air conditioned ship.—*Paper by G. B. Johnson and D. E. Phillips, read at the Annual Meeting of the Society of Naval Architects and Marine Engineers, New York, 15-16th November 1956.*

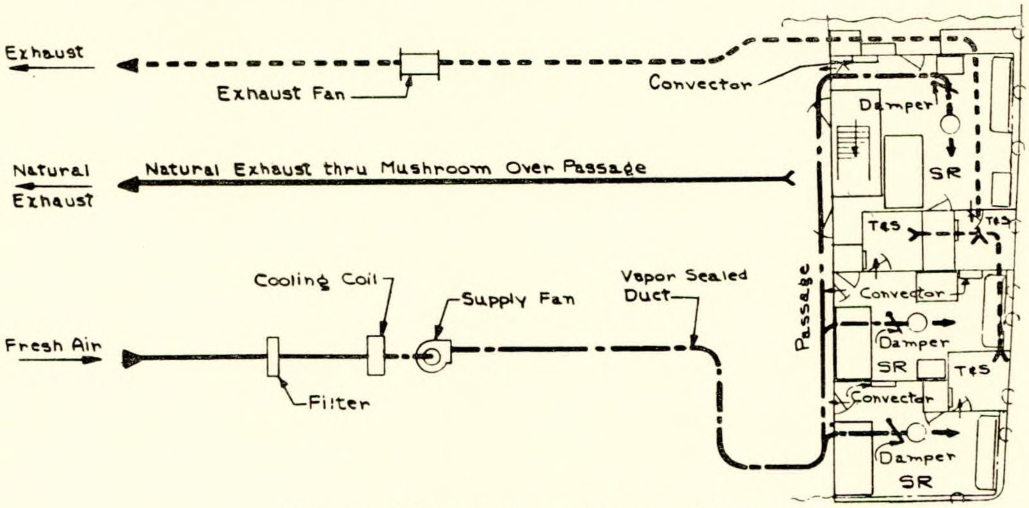


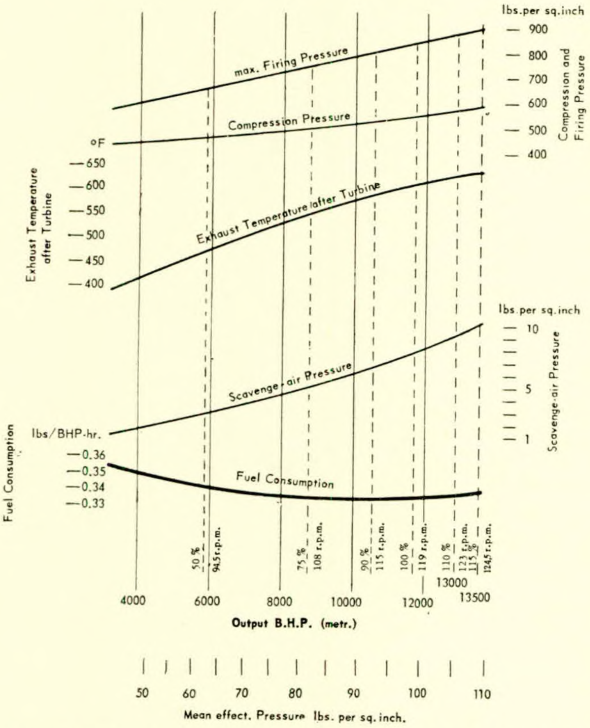
FIG. 17—*Simple system with manual control*

Microwave Course Beacon

The Elliott microwave course beacon system consists of a shore-based "X" band (3 cm.) transmitter and any number of pre-tuned shipborne receivers. The system makes use of the well-known Lorenz method of two overlapping beams, these being keyed by complementary audio Morse signals, and is specifically designed as a marine navigational aid for all types of craft. Subsequent to the initial development of this navigational system, operational trials proved the need for changes in design to provide reliable, unattended operation over extended periods in adverse weather conditions, together with greater reliability, range, efficiency, ease of assembly, installation and maintenance. Output power was increased to 8 kW peak and changes in the hard-valve modulator circuit gave the required increase in stability. The normal supply requirement for the transmitter is 90/115 to 190/250 volts, 56/60 c/s, but were mains power is not available the transmitter can be operated by employing suitable wind-driven generators. The maximum power required is 100 watts. To ensure that the transmitter is operating satisfactorily, it is essential that some form of failure alarm be provided. Although several methods are possible, it is considered that the most satisfactory way would be to install a simple monitor-receiver at a convenient position, passing the output by line or cable to a permanently manned point, such as the Harbour Master's Office or pilotage control, to provide a lobe signal verifying that the system is working satisfactorily. In the Beacon receiver the need for simplicity of operation and low power consumption, with the minimum of maintenance, has been met by the use of transistor amplification and printed circuit instruction. Redesign of the receiver has provided greater sensitivity and the manufacture of the outer casing and antenna horn in moulded glass fibre has provided light weight, without any loss in structural strength. Experience of the receiver over many months of use has determined the need for a hand-type unit with a built-in 3½-in. loudspeaker which can be held at eye level either within or external to the wheelhouse. Double headphone-operated receivers can be made if required, and a mounted type receiver has been considered. Although this navigational aid was devised for straight-path harbour entry, it is feasible to provide a system where the safe channel involves ships altering course at a marker buoy. In such circumstances two transmitters can be positioned on shore employing different lobe signals. One course path could be directed at the marker buoy covering the initial approach channel, while the second transmitter's course path would denote the correct channel from the marker buoy to harbour. Alternatively, two transmitters could be employed to give a definite fix at the point of intersection. These applications are being subjected to a practical test at a Scottish port with a view to determining the most satisfactory method.—*H. V. G. Bloodworth, British Communications and Electronics, December 1956; Vol. 3, pp. 651-652.*

Sulzer Turbocharged Engine Tests

The test bed trials of the first turbocharged two-stroke engine to be built by Sulzer Bros., Ltd., Winterthur, have now been concluded. This is the first of three similar 9RSAD76 engines for the Royal Rotterdam Lloyd and is a nine-cylinder crosshead type unit with a bore of 760 mm., a piston stroke of 1,550 mm., and a rated output of 11,700 b.h.p. at 119 r.p.m.,



Graphs showing test results with a Sulzer turbocharged engine

the maximum power being 13,500 b.h.p. or 1,500 b.h.p. per cylinder. The engine is cross-scavenged and turbocharged on the pulse system, the scavenging and charging air being supplied from three exhaust gas blowers of Sulzer design and manufacture. A 28-hour continuous run was carried out during the official acceptance test, when the engine ran for twelve hours at 11,700 b.h.p. and for two hours at 13,000 b.h.p.—*The Motor Ship, January 1957; Vol. 37, p. 434.*

Corrosion Protection

In protective coatings maximum wetting ability on the part of the sealing vehicle is mandatory—otherwise the seal will be superficial and broken in places—often in those locations where an attack is most deadly in its weakening effect on the structure. The electro-chemical action of oxidation at unprotected spots is actually accelerated by the presence of the seal on the readily accessible surfaces—current density is increased and deterioration at the unprotected spot is proportionately accelerated—the phenomenon is inescapable—hence the wetting ability of the vehicle determines the effectiveness of the seal as a whole. It is also necessary that the seal be durable and elastic enough when dry to stretch and contract to whatever degree the structure moves, expands or shrinks, under the worst weather that can be expected and load conditions. In olden times seafaring men had their own rust prevention processes; a barrel of raw fish oil mixed with graphite was used to cover all the exposed rustable metal surfaces, and this relatively crude, but quite effective measure, proven over decades, served to establish firmly the fact that wettability is all important. Fish oil, because of its low surface tension, drives out air and moisture from interstices so thoroughly that a protective coating can be established even on a damp surface. It is not even necessary to remove all rust; however, rust scale, mill scale and any previous coating not sticking tight, or under which rust may be forming, should be removed. Actually the fish oil then envelopes the particles of remaining rust and makes them a part of the primer coating. The foregoing is of the very essence of the technique of preservation of metallic surfaces, and actually it performs healing service for metal structures exactly similar to that which it does for the fish from which it was extracted. Furthermore, having regard to its source, it is not surprising that this oil expands and contracts without break in the film whenever environmental changes occur. As is natural, it takes longer for this type of oily vehicle to dry; however, the time is not excessive (varying from four to twelve hours) and is but a minute fraction of time, when it is considered that this valuable natural product will do its work reliably as a sealant for from four to seven years, which in most maintenance is about twice as long as the next best vehicle. A recently completed project, undertaken by the Battelle Memorial Institute, has demonstrated in an unusual way the remarkable wetting and penetrative properties of fish oil. A carefully rusted series of steel plates were coated with a rust inhibitor of the type referred to above. The vehicle had hitherto been impregnated with radioisotopes in known density—and after a suitable test interval the rusted and coated plates were honed progressively down to bare metal and a record kept of each honing, which proved conclusively that the selected vehicle consistently carried the protection all the way in through the rust down to the bare metal.—*R. E. W. Harrison, ASME Journal, November 1956; Vol. 68; pp. 713-717.*

Patent Specifications

Superheater

This invention relates to steam superheaters of a type in which the superheater elements are placed horizontally between banks of watertubes connecting a lower and upper drum. As shown in Fig. 7, the boiler comprises a lower water-drum (1) and an upper steam-and-water-drum (2), these being connected by banks (3 and 4) of steam generating tubes. The superheater elements (5) are placed between the tube banks (3 and 4). The superheater headers (6 and 7) are equipped with the usual drain valves. Each superheater element (5) is formed of two U-shaped portions joined by a connecting bar (9). Referring to Figs. 8 and 9, it can be seen that the connecting bar (9) is placed parallel with the boiler tubes (3 and 4) and is located between the group of inlets and outlets of the superheater elements (5) which are connected to the superheater headers (6 and 7). Each element comprises two U-loops arranged in series, providing a low temperature and a high temperature section. To connect the loops in series, the ends of those tubes, or limbs of two loops, which are to be joined are held by clamps (10) in contact with jointing surfaces in the connecting bar (9) at the end of the passages (11). Each clamp (10) engages collars or shoulders (12) on the two tubes and is tightened by a nut (13) on a stud (14) screwed into the bar (9). The passages (11) in the bar lead into a duct or ducts (15) extending longitudinally of the bar. At the lower end the duct is equipped with a drain connexion (16) controlled by a valve (17). The ends of the tubes (5) to be clamped to the connecting bar (9) are so arranged as to allow the connexion to be made at the front of the bar (9), i.e. the side remote from the boiler tubes. Where high-pressures and temperatures are concerned it is conventional practice to connect the superheater elements to the headers by welding to stub tubes, which are themselves welded to the headers. According to this invention, with the connecting bar (9) condensate from the appropriate portions of the elements (5), which are connected to it, will drain into the duct or ducts (15) in the bar prior to discharge through the drain valve (17). This arrangement reduces to a minimum the danger of corrosion to the internal surfaces of the element tubing. Further, the use of the connecting bar makes it possible to couple a compressed-air supply to the bar at the drain connexion so that the superheater can be blown-out to dry the internal surfaces of the elements and also to remove dust-like deposits.—*(British Patent No. 759,471, issued to The Superheater Co., Ltd., and H. Melhuish. Complete specification published 17th October 1956.) Engineering and Boiler House Review, December 1956, Vol. 71, p. 424.*

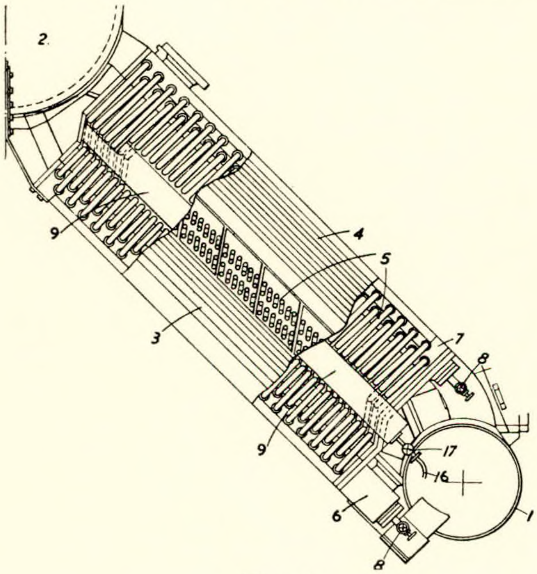


FIG. 7

Prevention of Rust on Decks of Tankers

The decks of tankers are much liable to rust formation. The rust preventive protective layer according to the invention can be applied in the following manner. The deck, after rust has been removed therefrom, is provided with a primer coating. A primer suitable for this purpose consists of a solution of tar in solvent naphtha, benzol, white spirit, turpentine, or some other organic solvent. On this primer coating a coating of at most 2 to 3 mm. thickness of a pitch having the proper plasticity range is applied, which coating, if desired, may be mixed with fillers. Since the coating must not flux in the

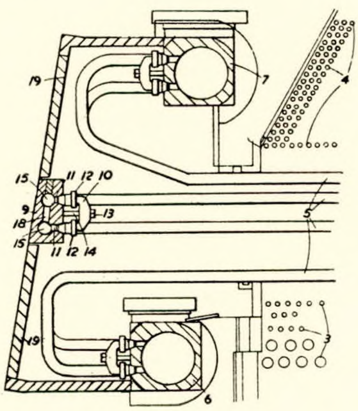


FIG. 8

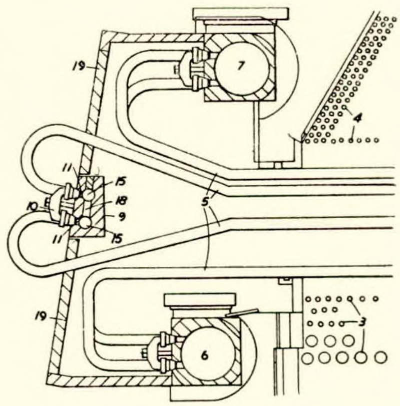
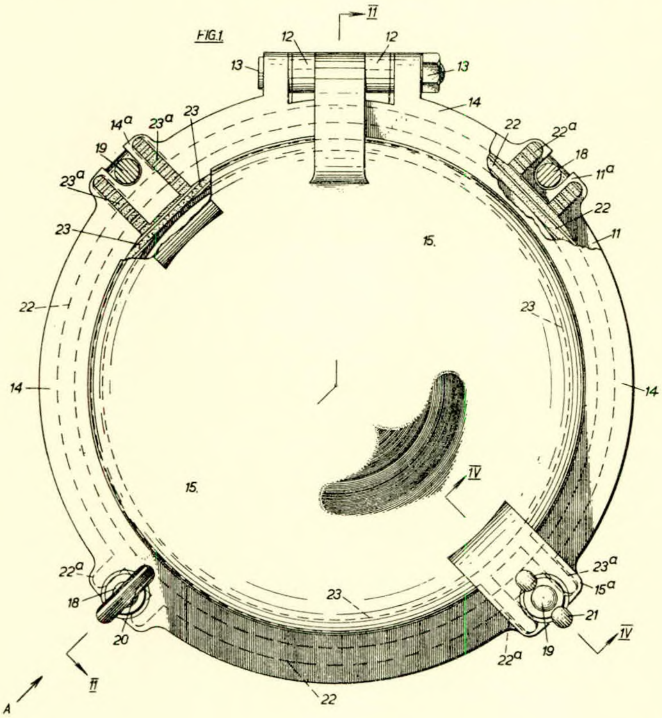


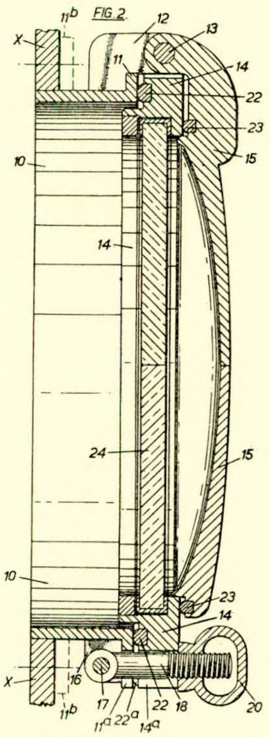
FIG. 9

tropics nor become brittle in the cold, the difference between the ring and ball softening point and the breaking point according to Fraas must be at least 60 degrees. This coating serves to preserve the deck. In order to protect the coating from mechanical effects there is applied to it a layer of felt impregnated with tar, reinforced, if desired, with a textile, glass or metal fabric. For the top layer use is likewise made of the pitch having the specified plasticity range. This pitch includes crushed slate, nibble or other rock and may be mixed with a filler such as limestone meal, schist meal, microtalc, infusorial earth, asbestos meal and cork meal.—*(British Patent No. 761,949, issued to Key and Kramer Asphalt Ruberoid N.V. Complete specification published 21st November 1956.)*

Ships' Windows or Side Scuttles

According to the invention and in one of its embodiments as shown in Figs. 1 and 2, the main frame of a scuttle comprises a spigot tube of steel or other ferrous metal capable of being welded within the window aperture of a ship's plating. To the spigot tube is secured a pair of angled lugs adapted to hold a common hinge pin or bolt for a glass holding frame (for convenience termed "light frame") and dead-light. Also secured to the tube are pairs of lugs—for example four—for securing to the tube swivel bolts adapted to engage lugs of the light frame and dead-light to hold them in closed position. The spigot tube is provided with support lugs in register with the lugs of the light frame and dead-light, the lugs being slotted to permit passage of the securing bolts and hinge-pin lugs. The support lugs may be formed on, or constituted by, an outwardly directed circumferential flange of the spigot tube. Interposed between the spigot tube and the light frame, and between the light frame and dead-light, are rubber or other resilient sealing rings. These sealing rings are recessed and secured in one or other of said frames. At the points where the swivel bolts engage the co-operating lugs of the light frame and dead-light, radial extensions of the sealing rings are provided, being located one on each side of the bolts. By these means, bending stresses imposed on the light frame and dead-light when the securing nuts or handles are tightened on the swivel bolts are obviated or reduced to a minimum. Avoidance of bending stresses is essential when aluminium alloys are used in the construction of the light frame and dead-light if permanent distortion of the parts and impairment of their water tightness is to be prevented.—*(British Patent No. 764,504, issued to T. Utley. Complete specification published 28th December 1956.)*



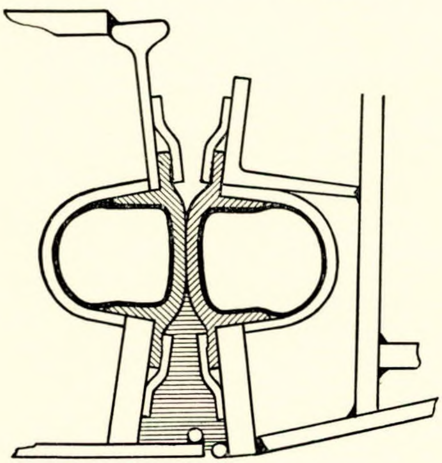


Marine Engineering and Shipbuilding Abstracts

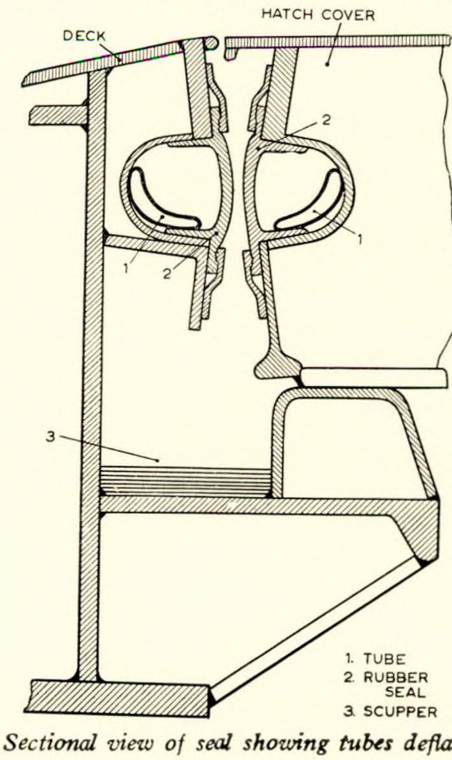
Volume XX, No. 5, May 1957

Flush Fitting Hatch Covers

A new type of hatch cover designed specifically for carrying deck cargo has been fitted on the 8,800-ton d.w. motor cargo ship *Rabenfels* built by H. C. Stulcken Sohn for D. D. G. Hansa. So far the new covers have been provided only for No. 1 hatch. The equipment has received the approval of both Germanischer Lloyd and the Seeberufsgenossenschaft, the equivalent of the British Ministry of Transport. Watertightness between the cover and the coaming is obtained by the use of rubber tubes filled with compressed air at a pressure of 3·5 kg./cm.^{2} and down to 1 kg./cm.^{2} the seal is stated to be effective. The air is supplied from the engine room system and the tube filling operation takes 30 minutes. It is possible for the seal to operate if the tubes are at different pressures and even should a tube become defective the inflated one would spread into the space not occupied. A scupper, arranged to



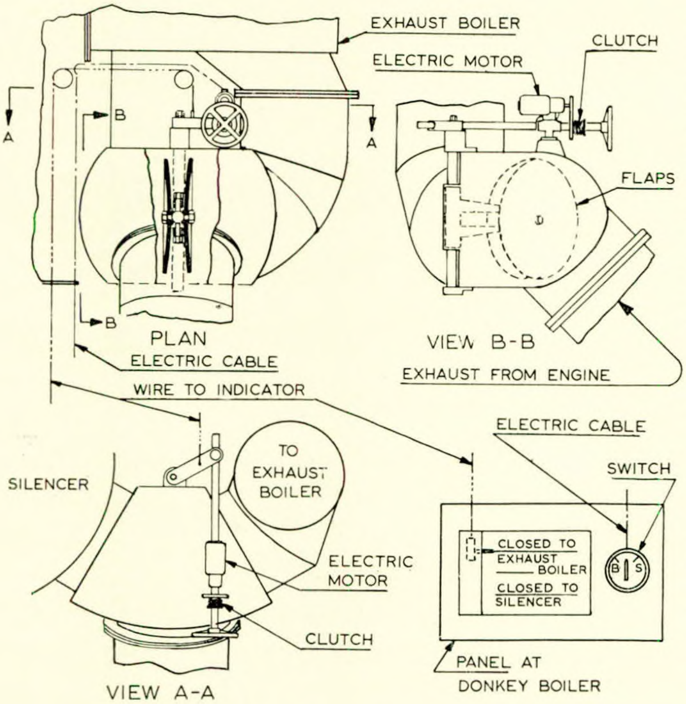
Tubes inflated and the seal in operation



Sectional view of seal showing tubes deflated drain to the bilges, is fitted underneath the sealing tubes and runs round between the coaming and the lower edge of the cover. No actual working load is taken by the seal as the base of the cover rests on steel plates supported from the deck. The accompanying illustrations show the cover in position with the tubes deflated, also with the tubes operating normally, a watertight seal being made between the two faces. On this class of ship the bridge is situated right forward, almost on the forecastle, so that the No. 1 hatch is just aft of the bridge. The hatch is covered by four iron covers each weighing 3·5 tons and designed for a maximum working load of 1·07 tons/$m.^2$ and arranged so that when in position they fit flush with the deck.—*The Motor Ship, January 1957; Vol. 37, p. 401.*

Electrically Controlled Changeover Valve

A new type of changeover valve in connexion with exhaust gas boiler installations has been developed by Akers Mek. Verk., Oslo, and its first application was on the motor ship *Burrard,* recently completed by these builders. The valve is situated in the funnel and is connected to the silencer and to the exhaust gas boiler, so that the gases may pass either directly to the

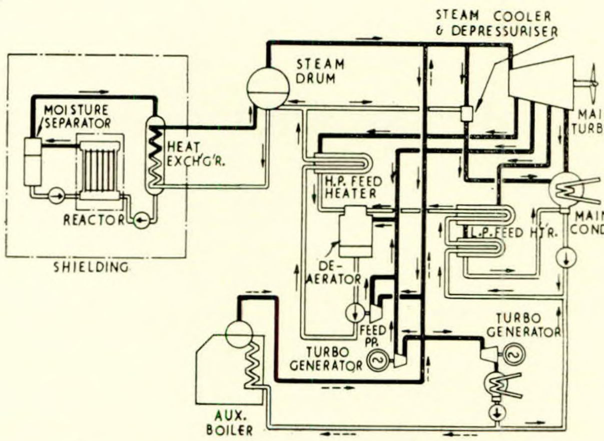


The Akers changeover valve

silencer or through the exhaust gas boiler, which also acts as a silencer. The valve may be actuated by hand, but, owing to the distance between the funnel and the boiler, an electric motor is provided, operating the valve and controlled by a switch from the boiler. Between the motor and the rod through which it actuates the valve is a clutch. This prevents overloading if the motor is not stopped immediately the flap valve has reached its seating. A light indicates when the motor is running and a simple form of indicator wire connected to the valve gear shows the position of the valve to the operator at the boiler.—*The Motor Ship, February 1957; Vol. 37, p. 470.*

Reactor/Steam Turbine Proposal

One of the many possible nuclear propulsion flow arrangements which have been studied by the de Laval Steam Turbine Company, Stockholm, is shown in the accompanying diagram. The reactor is of the heterogeneous, light water cooled and water moderated type. Condensate is pumped from the heat exchanger to the reactor core where it boils. The steam is



Diagrammatic layout of the de Laval scheme

passed through a moisture separator and kept below the pressure in the reactor core by a liquid level control pump which returns the recovered condensate to the reactor. The dry saturated steam goes to the heat exchanger, where it is condensed and slightly cooled. The reactor has a primary shielding to keep the induced radioactivity in the primary circuit low enough to permit maintenance work. Secondary shielding reduces neutron and gamma radiation to levels tolerable for the operating crew. The secondary steam generated in the heat exchanger is not radioactive. It is collected in a steam drum and moisture separator which supplies the main turbine, turbo-generators and turbo-feed pump. Steam to the turbine is regulated by means of a control valve and emergency trip valve. The steam can be bypassed round the turbine to the condenser to prevent excessive pressure rise. The condensate from the main condenser is pumped through a low-pressure feed water heater and a deærator, and is then pumped through a high-pressure heater into the steam drum. Steam for the feed water heaters and deærator is bled from various stages of the turbine. The turbo-generators, feed pumps, and a low-pressure steam circuit can be fed from a standby boiler when the reactor has to be run at a reduced load. The standby boiler can also run the main turbine at sea should the reactor have to be shut down at sea. The selection and arrangement of equipment, reactor safety and similar points are being investigated by de Laval in close co-operation with the Swedish Shipbuilding Research Foundation and with A.B. Atomenergi, the Swedish company concerned with atomic power stations. —*The Shipping World, 2nd January 1957; Vol. 136, p. 16.*

Analysing the Stepless Planing Boat

During recent years the David Taylor Model Basin has towed a number of models of planing craft in smooth water to determine resistance, trim angle, wetted lengths and wetted surface. In most cases each of these models was considered to represent a particular full-scale boat, and the data obtained were presented in dimensional form for specific boat dimensions and displacements. Each model, however, can represent a boat of any size. Therefore, when a new design is to be developed, all models of previous designs can be considered to represent boats of the size of the new design, and the data on their performance can be used for guidance. In order to do this easily the designer needs to have the information on the previous designs in suitable form. The purpose of this report is mainly to indicate appropriate methods of presenting and utilizing the accumulated information on hull forms and model test results for planing boats to guide the design of future boats. In this report the important planing hull parameters are defined and a convenient method of combining them in a hull-form characteristics sheet is shown. A plan for presenting model test results in a dimensionless form suitable for comparison and analysis is next given. The hull-form characteristics and model test results are at present being incorporated in a Taylor Model Basin design data sheet, an example of which is given. The effects on performance of variations in some of the primary parameters are then illustrated and discussed. Also, methods are proposed for improving the usefulness of future model tests for purposes of comparison and analysis. Finally, a step by step design method is proposed, and data are presented which it is believed will assist the designer in making design decisions quickly and with assurance of correctness.—*E. P. Clement, David W. Taylor Model Basin, 1955; Report 1,093.*

Improving the Locking Capacity of Nut and Bolt Connexions

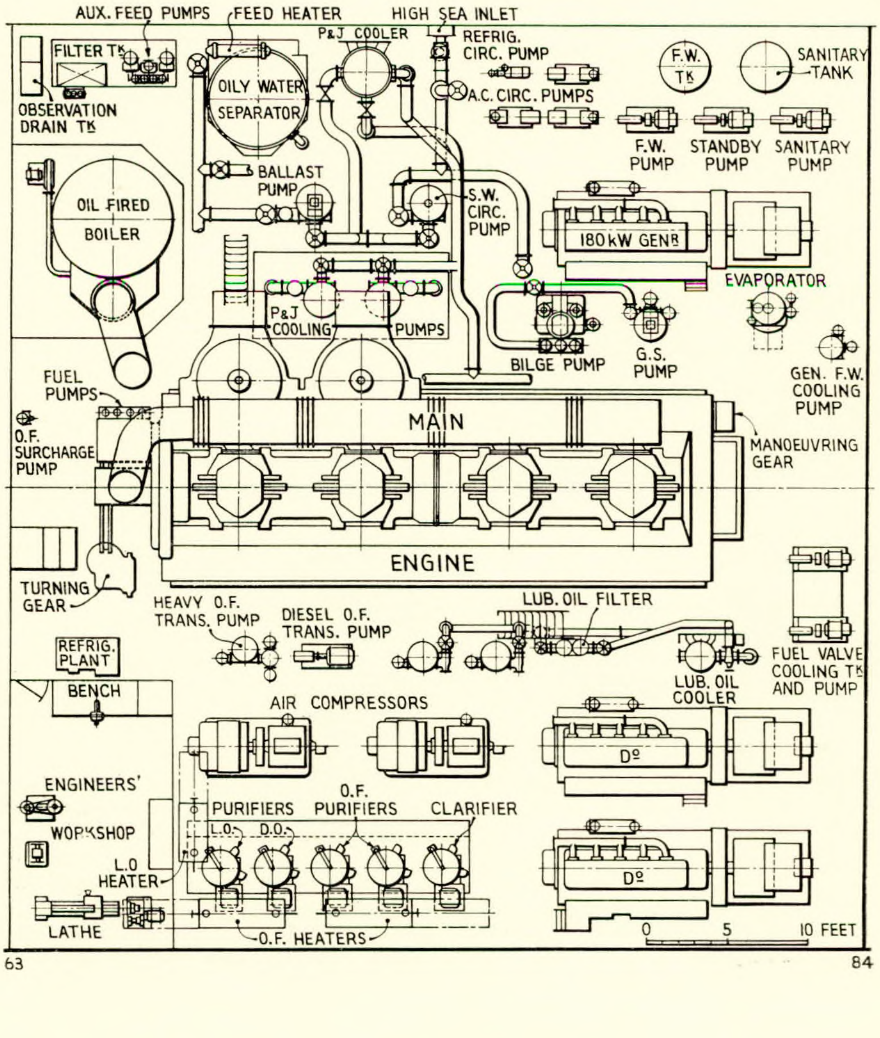
Investigations have been made to determine the effect of various types of spring washer on the locking capacity of nut and bolt connexions. New apparatus has made it possible to measure tightening and loosening torques, axial loads, and the elastic properties of the system. Specimen washers of each type investigated were subjected to different surface treatments before being tested under standard conditions. The results are presented in tables and curves. For maximum security of a bolted connexion, the moment required for loosening should assume its greatest value at an axial stress corresponding to the maximum load plus the initial load required to hold the parts together. The usual types of spring washer with smooth surfaces produce an appreciable increase in axial stress and also reduce the loosening torque, especially at high stresses, until the nut has been loosened through 15-20 degrees; at greater angles the loosening torque increases considerably. It is therefore recommended that the connexion should first be tightened to the greatest permissible axial stress and then loosened through 15 to 20 degrees. With washers having the recently developed crystalline-rough surfaces, the tightening torque needed to attain a given axial stress is about 50 per cent greater than it is with smooth high-tensile washers. The loosening torque exceeds the tightening torque by 50 to 90 per cent and has its maximum value at starting.—*W. Koennecke, Forsch. Ing. Wes., 1955; Vol. 22, p. 85. Journal, The British Shipbuilding Research Association, December 1955; Vol. 11, Abstract No. 12,345.*

Clyde-built Motorship for Norwegian Owners

A recent delivery from the yard of Alexander Stephen and Sons, Ltd., Linthouse, Glasgow, was the single-screw motorship *Crux,* built for the South American service of Det Bergenske Dampskibsselskab, Bergen. The *Crux* is an open shelter decker built to the requirements of Det Norske Veritas, highest class, her scantlings being for a closed shelterdecker with a corresponding increase in draught, tonnage and deadweight. The principal particulars are: —

Length b.p.	400ft.
Breadth moulded	57ft.
Depth moulded to shelter deck ...	35ft.
Draught loaded	23ft.
Gross tonnage, tons	4,504
Deadweight tonnage, tons ...	6,420
Service speed, knots	14½

Welding has been extensively employed throughout the vessel, large units weighing anything up to 38 tons being prefabricated before being placed in the building berth. The propelling machinery consists of a 4-cylinder opposed piston, two-stroke Stephen-Doxford engine having cylinders with a bore of 670 mm. and a combined stroke of 2,320 mm. Controls of the engine are arranged at the forward end, and the engine is of the diaphragm chamber with glands to prevent pollution of the



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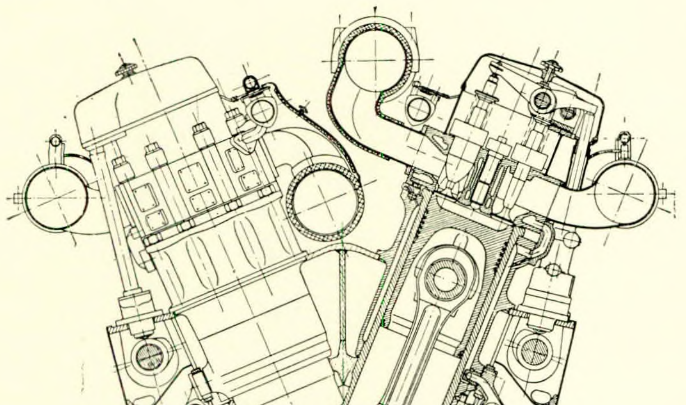
crankcase oil. The bedplate, columns and entablature are of normal welded design. A built-up crank is employed in two lengths with a single collar on an extension of the aft journal for a Michell thrust bearing. The flywheel of moderate proportions is fitted on the aft coupling of the thrust shaft, and has a toothed rim arranged to take the turning gear. At the end of the crankshaft a Doxford-Bibby detuner is fitted to eliminate vibration and torsional oscillations. For cooling the main engine water jackets and pistons, distilled water is used, and operates on a closed system which is kept primed by a header tank. The water is specially treated to prevent corrosion in jackets and pistons. For cooling fuel valves, etc., distilled water is also employed, but two supplies are arranged separately to avoid contamination of pistons and jacket cooling water. Arrangements are made to run on boiler oil, if desired. Doxford's fuel timing valve injection system is used in this engine in conjunction with a standard type of fuel pump; also included is a new type of air starting valve. These changes lead to a simpler control system.—*Shipbuilding and Shipping Record, 22nd November 1956; Vol. 88, p. 671-674.*

High Speed Engine for Cargo Ships

The Pielstick engine, built in France by the S.E.M.T., after employment for Naval vessels, is now being installed in a number of merchant ships. It is a relatively high speed unit (about 425 r.p.m.) and, when exhaust turbocharged, operates at a mean effective pressure of 10·35 kg. per sq. cm., or 150lb. per sq. in. It is stated to have a fuel consumption of 156 gr., or 0·35lb. per b.h.p.-hr. The Hamburg shipping company of A. Kirsten and Company have ordered from H. C. Stülcken Sohn three 5,000-ton motor cargo vessels for their Great Lakes service, and in each of these two of the Henschel-Pielstick engines will be installed, with eight cylinders, developing 2,200 b.h.p. at 428 r.p.m. Of the vee type the engine has a cylinder diameter of 400 mm. and a piston stroke of 460 mm., Brown-Boveri exhaust gas turbochargers being fitted, one for each group of four cylinders. The weight of the engine is about 15 kg. per b.h.p., the mean piston speed being in the neighbourhood of 6·6 m. per sec. or 1,300ft. per min. The engine can be built as a non-supercharged type, in which case the cylinder output is 155 b.h.p., compared with 275 b.h.p. for the turbocharged design. For reversing, ahead and astern cams are fitted side by side on the two camshafts, and the shafts are moved longitudinally by a servo motor which is controlled from the manœuvring platform. Fresh water cooling is used for the cylinders. The engine framework is of steel plate welded construction without tie rods. The cylinders are separate and engine units can be built with up to eighteen cylinders, the output being 4,500 b.h.p. The general construction of the engine may be noted from the sectional elevation. The cylinder covers are of cast iron, held by eight bolts. In each cylinder are two inlet and two exhaust valves and a safety valve. All valves have double springs and those for admission and discharge are actuated by two rockers through push rods. The rockers are enclosed in oiltight casings and are lubricated by the engine oil circulating system.—*The Motor Ship, March 1957; Vol. 37, p. 532.*



Sectional end elevation of the Henschel-Pielstick engine. Speed 428 r.p.m.

Titanium Condenser Tubes

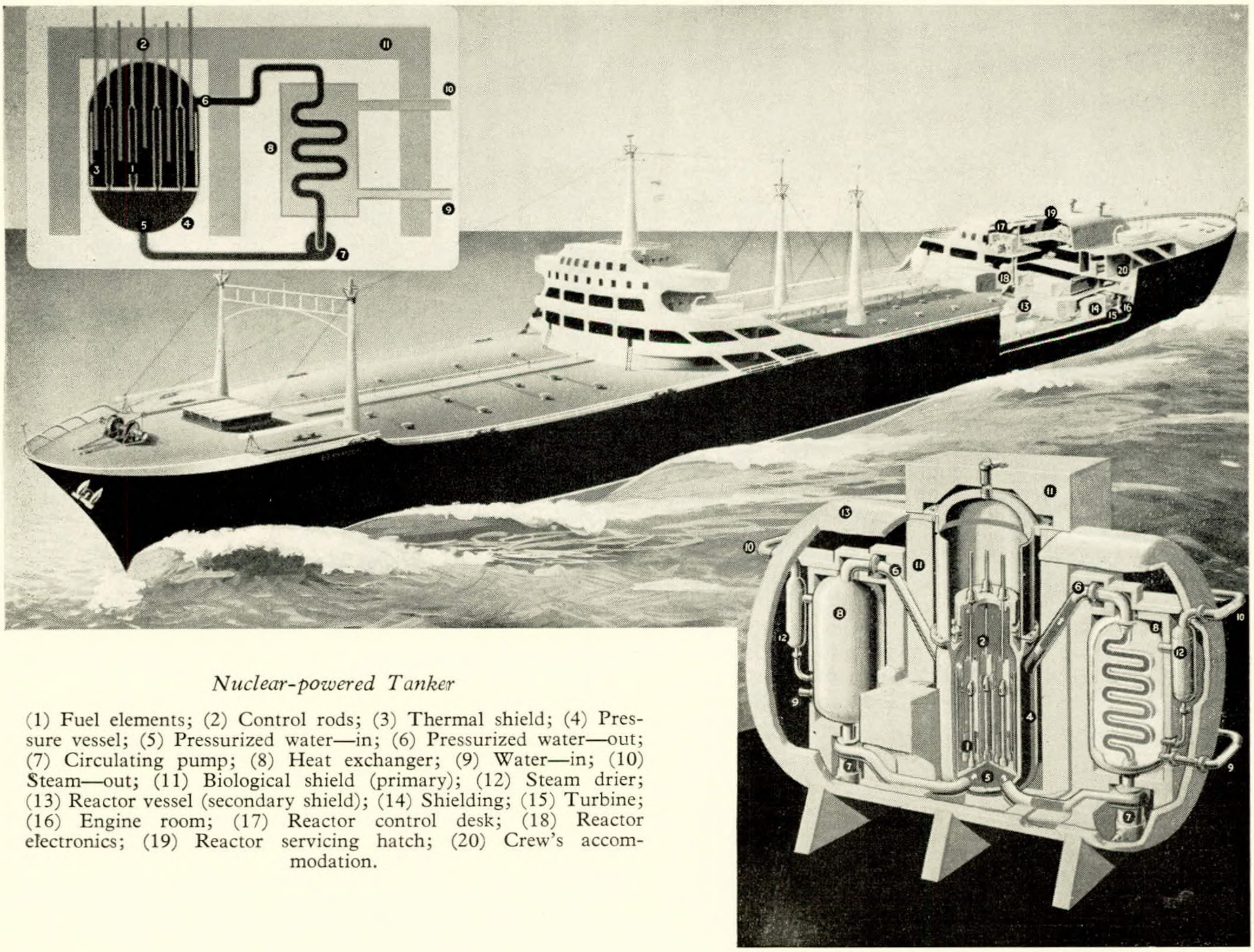
The first titanium condenser tubes ever to be fitted to condensers installed in a British power station are now undergoing service trials at Uskmouth Power Station in the C.E.A. South Wales Division. Imperical Chemical Industries, Limited, the pioneers and leading suppliers of titanium and its alloys in Great Britain, claim to have established by exhaustive laboratory tests that titanium has a phenomenal resistance to many corrosive media, particularly sea water. Now, in cooperation with the Central Electricity Authority, the company is carrying out long term trials in which a number of titanium tubes are being exposed, under normal service conditions, to a particularly aggressive estuarine cooling water from the River Usk. The tubes are solid drawn from I.C.I. Titanium 130. The condensers installed with each of the 60-MW generating sets at Uskmouth are twin units with a total surface of 47,000 sq. ft.; and designed for 28·9in. vacuum when supplied with 40,000 gal./min. of water at a temperature of 56 deg. F. Tests have shown that titanium has excellent resistance to attack by sea water, solutions of most metal chlorides, and even aqua regia. This outstanding resistance to sea water corrosion is particularly important; the metal resists marine atmospheres, sea spray, and immersion in or impingement by sea water; even when fouling organisms grow on it, no corrosion takes place under them. In all these respects it is said to be considerably better than traditional materials, including cupro-nickel and aluminium bronze; its behaviour is, in fact, comparable with that of the platinum group of metals. In crevice attack tests, made under differential æration conditions in sea water, titanium does not pit or etch, although there is some slight loss of metal. In the same conditions, stainless steel and certain copper alloys are significantly pitted. Thus there is every indication that titanium is not susceptible to crevice attack. Fatigue properties show no deterioration on immersion, and tests carried out under a constant stream of sea water have indicated a higher fatigue limit than in air. This unusual result is probably due to the slight cooling effect resulting from the evaporation of sea water. Resistance to stress corrosion is also outstanding. Only in the most corrosive conditions, such as exposure to red fuming nitric acid, is there any evidence of stress corrosion in titanium. It appears that considerable progress has been made in the development of methods for jointing titanium. It has been established that spot, flash, pressure and fusion welding techniques are all suitable, provided that modifications are made to allow for the peculiar chemical and metallurgical properties of the metal. Welding of commercially pure titanium containing only small amounts of impurities (i.e. Titanium 130) yield joints with excellent properties. Many titanium alloys are much less suitable for welding because of the constitutional effects resulting from the thermal cycle imposed. Considerable attention has, however, been paid to the development of alloys combining good mechanical properties with satisfactory welding characteristics.—*Engineering and Boiler House Review, March 1957; Vol. 72, pp. 75–76.*

Nuclear Powered Tanker Proposal

The illustration is an artist's impression of a nuclear propelled tanker as shown to shipowners and shipbuilders who attended a conference at Harwell last autumn. In this drawing a pressurized-water reactor is used. It has recently been announced that scientists at the British Atomic Research Establishment at Harwell are studying the organic liquid moderator for ship propulsion. With this type of reactor diphenyl or terphenyl is employed, allowing lower pressures to be used with consequently reduced corrosion problems. These organic liquids are used instead of water as a coolant and moderator, and the heat transfer properties are said to be better. They also permit a simplified design of shell and a higher safety factor for the reactor. No information is yet available as to the size and speed of any nuclear-powered tanker which may be built in the United Kingdom, or when the keel of such a vessel is likely to be laid. Through the General Council of British Shipping and the British Shipbuilding Research Association, representative shipowners and shipbuilders at a meeting in London on 23rd January had under further consideration the question of the possible application of nuclear power to merchant ship propulsion. No decision was reached for the immediate construction of a prototype tanker or other type of vessel using an atomic unit, although the practicability and desirability of such a step was discussed, taking into account the main requirement of the shipping industry that for eventual commercial use propulsion of merchant ships by atomic energy must be possible on an economic basis.—*The Shipping World, 30th January 1957; Vol. 136, p. 161.*

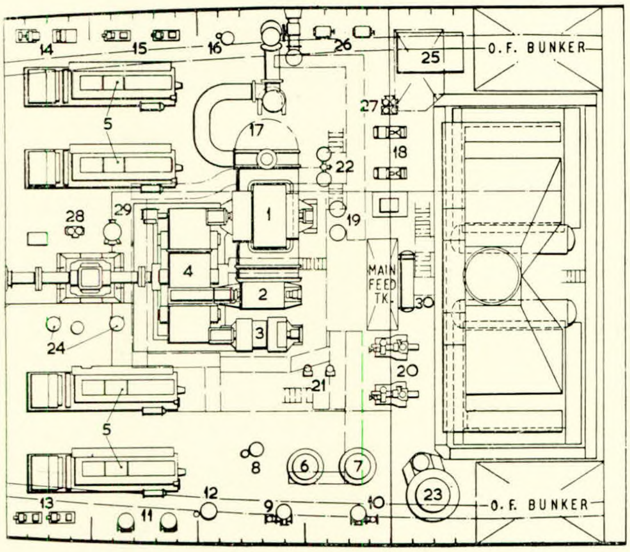
Refrigerated Cargo Liner

The first of a new class of refrigerated cargo liner ordered from the Greenock Dockyard Co., Ltd., by the Scottish Shire Line, Ltd., London, has now been delivered. This vessel, the *Argyllshire,* 11,250 tons d.w., completed trials off the measured mile at Skelmorlie on 18th October, the day before her sister ship the *Ayrshire* was launched. The Scottish Shire Line, associated with Clan Line Steamers, Ltd., has long felt the need for greater tonnage so far as the Australian trade is concerned, especially in the export from that country of chilled and frozen meat, fruit and dairy produce. The two new vessels will be employed in that trade. The *Argyllshire,* launched on 23rd May 1956 by the Countess Jellicoe, is the fifty-first vessel to be built for the Clan Line by the Greenock Dockyard Co., Ltd., and is also the largest ship to be built for them. In addition to refrigerated cargoes, the *Argyllshire* can also carry wool in bale in her 'tweendecks, the height of these decks having been arranged for maximum bale stowage. The propelling machinery consists of a set of Parsons steam turbines developing 10,500 s.h.p., which give her a speed in service of $16\frac{3}{4}$ knots. During her trials over the measured mile an average speed of 18·99 knots was attained. Accommodation of a very high standard is provided for twelve pas-



Nuclear-powered Tanker

(1) Fuel elements; (2) Control rods; (3) Thermal shield; (4) Pressure vessel; (5) Pressurized water—in; (6) Pressurized water—out; (7) Circulating pump; (8) Heat exchanger; (9) Water—in; (10) Steam—out; (11) Biological shield (primary); (12) Steam drier; (13) Reactor vessel (secondary shield); (14) Shielding; (15) Turbine; (16) Engine room; (17) Reactor control desk; (18) Reactor electronics; (19) Reactor servicing hatch; (20) Crew's accommodation.



Key to Machinery Arrangement

(1) L.P. ahead and astern turbines; (2) H.P. ahead turbines; (3) I.P. ahead and H.P. astern turbines; (4) Double reduction gearing; (5) Diesel-driven generators; (6) F.W. evaporator; (7) S.W. evaporator; (8) Bilge pump; (9) Ballast pump; (10) General service pump; (11) Refrig. pump; (12) San. pump; (13) S.W. pres. pumps; (14) Air compressors; (15) Dom. F.W. pres. pumps; (16) F.W. pump; (17) Main circ. pump; (18) O.F. transfer pumps; (19) Extraction pumps; (20) Main feed pumps; (21) Aux. boiler feed pumps; (22) Grease extractor; (23) Aux. boiler; (24) Lub. oil pumps; (25) Oil-burning unit; (26) Dom. refrig. circ. water pumps; (27) Fuel oil filters; (28) Lub. oil filters; (29) Lub. oil coolers; (30) Drain cooler.

sengers. The accommodation throughout is, in point of fact, very well planned and most attractively decorated. The principal particulars of the *Argyllshire* are as follows:—

Length o.a.	535ft.
Length b.p.	497ft.
Breadth moulded	69ft.
Depth moulded	41ft. 7in.
Draught	28ft. 3in.
Deadweight, tons	11,250
Gross, tons	9,400
Machinery output, s.h.p.... ...	10,500
Service speed, knots	16¾
Total cargo capacity (including refrigerated):	
Bale, cu. ft.	600,000
Refrigerated, cu. ft.	378,000

The vessel is powered by a set of Parsons reaction turbines consisting of three turbines, h.p., i.p., and l.p., working in series. For astern working, an h.p. astern turbine consisting of an impulse wheel is incorporated with the i.p. turbines, and separated from the ahead portion by a diaphragm. This is arranged to work in series, with the l.p. astern turbine of impulse reaction type situated in the exhaust casing of the l.p. ahead turbine. The service s.h.p. is 10,500 at 112 propeller revolutions per minute, and the machinery is capable of developing 11,550 s.h.p. maximum, at 115·5 r.p.m. The power developed by the main engines is measured by a Siemens torsionmeter fitted to the propeller shaft. Two Babcock and Wilcox steam generating units are fitted to supply steam for the main engines, each comprising one marine type single pass watertube boiler with side and rear water-cooled walls, integral type superheater, tubular type air preheater, and one set of mechanical soot blowers. The steam pressure is 415lb. per sq. in. and the temperature 750 deg. F. The boilers are worked on forced draught and are oil fired. The total evaporation is 94,000lb. per hr. and the total heating surface is 14,044 sq. ft. A Cochran auxiliary boiler is fitted for supplying steam for domestic purposes, to the evaporator, oil purifiers and the 100 tons per hour Comyn oily water separator.—*The Shipping World, 14th November 1956; Vol. 135, pp. 433-435.*

Large Japanese Tanker

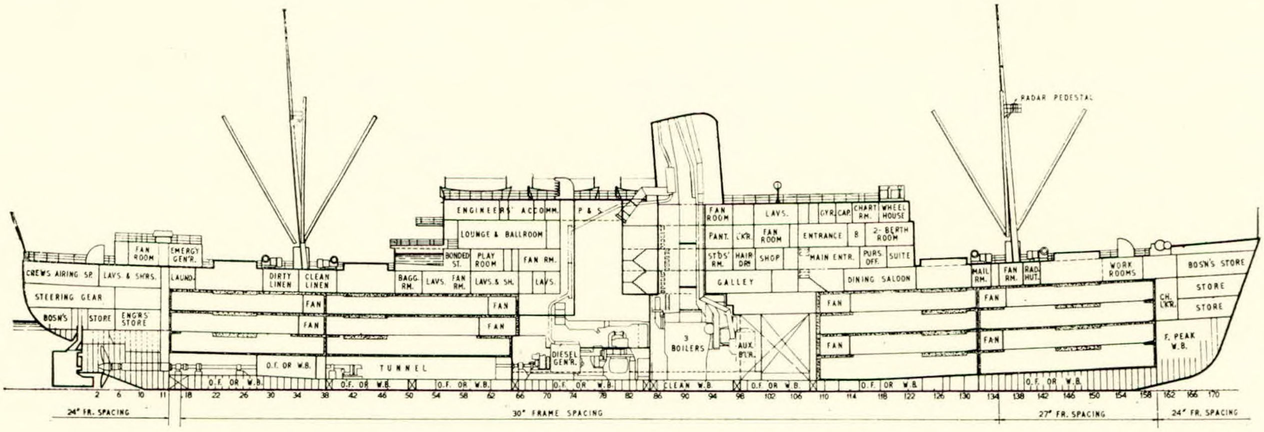
The largest Diesel-engined tanker built in Japan was recently completed to the order of the Nitto Shosen Kabushiki Kaisha at the Nagasaki yard of the Mitsubishi Zosen Kabushiki Kaisha (Mitsubishi Shipbuilding and Engineering Co., Ltd.). The hull was laid down in the middle of January 1956, and the ship was launched in the latter part of May. Trials were run towards the end of September. The main details are:—

Length overall	203·175 m. (666ft. 7in.)
Length b.p.	192·324 m. (631ft. 0in.)
Breadth, moulded	26·822 m. (88ft. 0in.)
Depth, moulded	13·716 m. (45ft. 0in.)
Draught, about	10·319 m. (33ft. 10¼in.)
Gross register, tons ...	20,496
Net register, tons ...	13,145
Deadweight capacity, tons	33,315
Service speed, loaded, knots ...	14¾
Cruising range, sea miles	29,300
Cargo tank capacity, cu. m. ...	43,900
Bunker capacity, m. tons ...	3,385
Fresh water capacity, m. tons...	620
Machinery—maximum continuous rating, b.h.p. ...	12,000
Normal service rating, b.h.p....	10,200

The hull is divided into thirty cargo oil tanks by two longitudinal and nine transverse bulkheads, their length being about 12 m. each. Aft of the cargo oil tanks is a group of bunker tanks, also the main pump room. A Mitsubishi-Nagasaki-U.E.C. single-acting two-stroke turbocharged engine is installed, designed to develop a maximum continuous output of 12,000 b.h.p. at 123 r.p.m. or a service continuous output of 10,200 b.h.p. at 117 r.p.m. The engine has nine cylinders, 750 mm. in diameter with a piston stroke of 1,500 mm., the mean piston speed being 6·15 m. per sec., the mean effective pressure 7·36 kg. per sq. cm. and the weight 510 tons, with cast iron construction. The overall length is 16·85 m., the overall height 9·3 m., and the width of the bedplate 3·6 m. There are two four-stroke supercharged Diesel-engined 450-volt generators of 240 kW. After passing through the turboblowers the exhaust gas is delivered to a boiler which supplies sufficient steam for all purposes when the vessel is at sea. All auxiliaries are driven by electric motors, except the Butterworth pump, the forced-draught fan, the feed pump and the deck machinery.—*The Motor Ship, December 1956; Vol. 37, p. 353.*

British Passenger Ship with Method I Fire Protection

A twin-screw passenger and refrigerated banana cargo liner embodying some new and interesting features is now in service between the United Kingdom and the West Indies. This vessel, the *Camito,* 8,735 tons gross, has been built by Alexander Stephen and Sons, Ltd., for Elders and Fyffes, Ltd., London. In general the new vessel is similar to the *Golfito,* built in 1949, but with the distinction that she is the first ship carrying over 100 passengers to be built in Great Britain according to Method I of the 1948 Convention. This is a method of fire protection which involves the use of internal divisional bulkheading of non-combustible B class divisions, without the installation of a sprinkler system in the accommodation and service spaces. All bulkheads, linings and ceilings have been made of incombustible Marinite, and this makes each cabin a fireproof unit. It will also reduce the estimated fire potential of each cabin from the 11¼ million B.t.u. associated with conventional timber construction to 5 million B.t.u.—most of



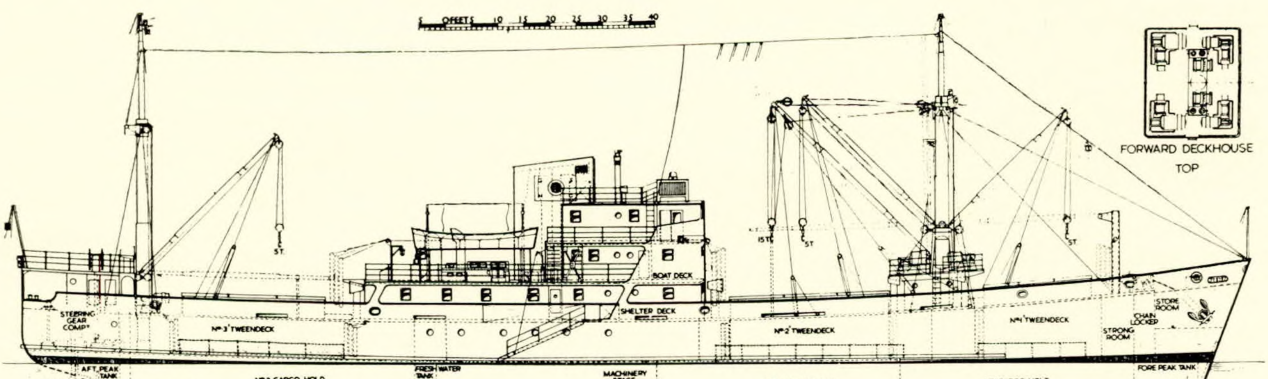
which arises from the furniture, curtains, bedclothes, carpets and the passengers' personal belongings. The *Camito* is propelled by steam turbines of Pametrada design which give her a speed in service of 17½ knots. Longitudinal framing has been adopted in the double bottom which extends forward and aft and is suitable for the carriage of fuel oil or water ballast. The tunnel extends across the vessel and contains the refrigerating machinery for the cargo holds and provision rooms, and there are also two Diesel oil tanks and two oil fuel or water ballast tanks at the ship's side. Deep oil fuel bunkers are fitted at the forward end of the boiler room. Seven watertight bulkheads, extending to the upper deck, form the fore peak carrying water ballast, two insulated cargo holds forward, the engine room and boiler room amidships, two insulated cargo holds aft, and the after peak also carrying water ballast. The principal particulars of the *Camito* are as follows: —

Length o.a.	447ft. 7in.
Length b.p.	415ft.
Breadth moulded	62ft. 2in.
Depth to upper deck ...	34ft. 9in.
Draught	26ft. 1in.
Gross tonnage	8,735 tons
Net	4,505 tons
Deadweight	5,800 tons
Machinery output	10,500 s.h.p.
Service speed	17½ knots
Passengers (1st class) ...	103

The propelling machinery consists of a two shaft arrangement of cross-compounded double-reduction geared turbines of Pametrada design. The turbines are designed to develop a combined shaft horsepower of 10,500 at normal service rating with propeller revolutions of about 125 per minute. Each unit has one h.p. ahead turbine and one l.p. ahead turbine each in its own casing. The astern turbine is incorporated in the exhaust end of the l.p. turbine. The reduction gears are of articulated type and all teeth have been shaved. The gear cases are fabricated and the thrust blocks are separate external fittings. Steam is supplied by three Babcock and Wilcox single-pass sectional header watertube boilers designed for 510lb. per sq. in. pressure and superheater outlet temperature of 750 deg. F. The oil fuel pressure system burners have been made by Hopkinsons, Ltd. Bailey automatic combustion control is fitted to the boilers. A Cochran boiler is installed in the boiler room to supply steam at 100lb. per sq. in. to the low pressure system for hotel services, fuel heating, etc. At sea, a steam generator of Weir's make is used for the same purposes thereby reducing the risk of contamination of the main condensate. There are two main condensers, one for each turbine unit and underslung from the l.p. turbine casings. They are designed to maintain a vacuum of 28·5in. with 75 deg. F. sea water when the turbines are developing the maximum designed shaft horsepower. The vessel is fitted with the Weir closed-pass system with two turbine-driven main feed pumps and the usual extraction pumps, air ejectors, etc. An electro-feeder is used for boiler feeding in port. A Weir's deærator is fitted in the condensate circuit for use in port and while manœuvring.—*The Shipping World, 9th January 1957; Vol. 136, pp. 71-74.*

1,000-ton d.w.c. Shelterdecker

The m.s. *Heron,* launched from the Bristol shipyard of Charles Hill and Sons, Ltd., is a short-sea trader of about 1,000 tons d.w.c. for the General Steam Navigation Co., Ltd., London. Of the shelterdeck type, this ship has a length b.p. of 215ft., a moulded breadth of 36ft. 6in. and a depth moulded to the main deck of 13ft., the depth of the shelter 'tweendecks amidships being 7ft. 6in. The ship carries her deadweight of 1,000 tons on a draught of about 12ft. 10in. and has a total bale capacity for the cargo holds, 'tweendecks and strong room



of about 70,000 cu. ft. All deck machinery is electrically driven, the 15-ton derrick at No. 2 hatch and the two 5-ton derricks for Nos. 1, 2 and 3 hatches being served by Laurence Scott winches with remote control arrangements. Designed for a service speed of 12 knots, the ship is propelled by a British Polar M48M engine, installed amidships and capable of developing 1,280 b.h.p. at 250 r.p.m. This is coupled to a Heliston bronze propeller.—*The Motor Ship, February 1957, Vol. 37, p. 451.*

20,000 s.h.p. Nuclear Propulsion Plant

The common misconception that nuclear fuel is cheap and thus efficiency is of secondary importance has become evident. Actually, the reverse is true and, hence, the general trend is towards the development of reactors for elevated temperatures and thus higher efficiencies. The closed-cycle gas turbine in combination with a gas cooled reactor offers not only an efficient, but also a simple, nuclear power plant. Detailed studies have been prepared for a marine power plant of 20,000 normal shaft horse power rating that would be suitable for installation in a tanker. This type of ship was selected for study due to the higher powered propulsion plant required for most of the present-day tankers and since they are at sea a greater percentage of the time than are most merchant ships. The features of high power and high percentage time at sea make the economics for nuclear propulsion more favourable. As nuclear energy is expensive, it is logical that an effort is being made to develop more efficient power plants to supplant the present nuclear plants that employ steam at low to moderate pressure. It is for this reason that the Atomic Energy Commission has recently contracted with Aerojet General Corporation for the development, building and operation of an experimental high temperature, gas cooled power reactor. In addition, design study contracts for closed-cycle gas turbine propulsion plants with gas cooled reactors have been placed with the Cleveland Diesel Division of the General Motors Corporation, Ford Instrument Company and General Dynamics. Fig. 1 shows the flow diagram of the proposed propulsion plant and gives the state points for an output of 20,000 s.h.p. The gas enters the l.p. compressor at 115 deg. F. and 170lb. per sq. in. abs. After compression, the gas passes through the intercooler where the temperature is decreased from 281 deg. F. to 115 deg. F. The gas leaves the h.p. compressor at 263 deg. F. and 737lb. per sq. in. abs. and enters the high pressure side of the recuperator where it is preheated to 773 deg. F. by the l.p. turbine exhaust gas before entering the reactor. From the reactor the gas enters the h.p. turbine at 1,300 deg. F. and 705lb. per sq. in. abs. where it gives up part of its energy to drive the compressor, and finally the remaining energy is converted to useful output by the l.p. or power turbine. After passing through the l.p. side of the recuperator where heat is transferred to the gas, leaving the last compressor stage, the gas is cooled to its initial temperature in a fresh water-cooled precooler. The system is closed and the process is one of continuous flow with cycle efficiency remaining substantially constant over a wide range of loads. From a consideration of power plant design, nitrogen has been selected as a working fluid in this study. Nitrogen is relatively inexpensive and is readily available. Also, it is chemically inert with respect to the materials in the reactor and turbine

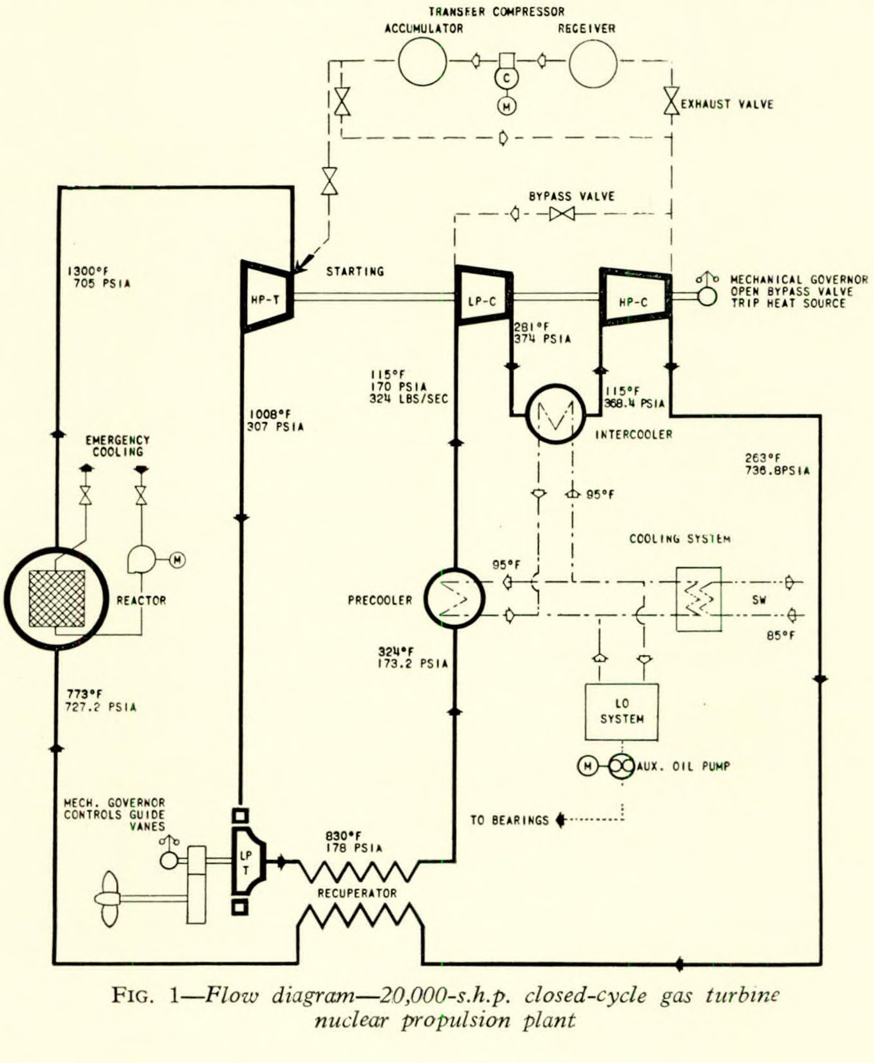


FIG. 1—*Flow diagram—20,000-s.h.p. closed-cycle gas turbine nuclear propulsion plant*

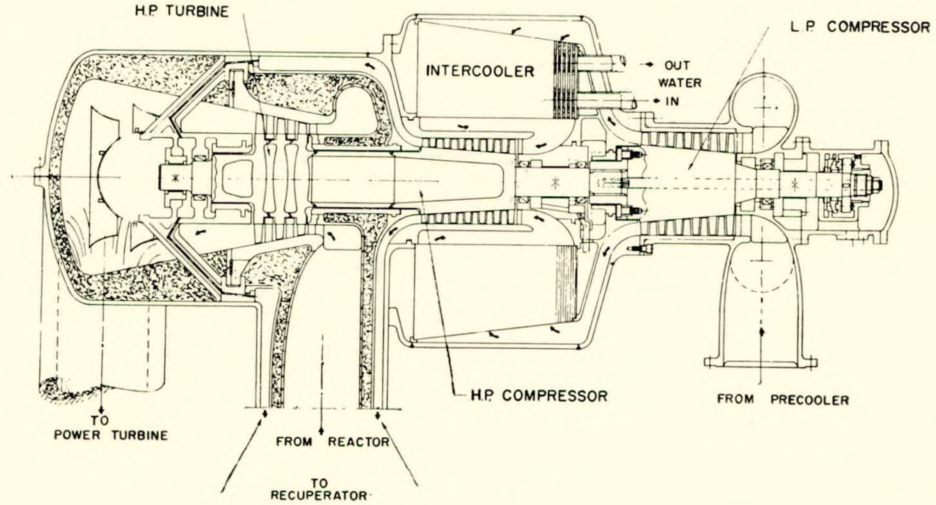


FIG. 3—*The compressor/turbine set*

unit. Its density is close to air which makes it ideally suited for the turbo-compressor unit. The radiation stability is considered good and if properly purified, conditioned and replaced periodically, it is believed that the gas can be circulated directly through the reactor and gas turbine without requiring secondary shielding of the turbo-compressor unit. The accidental leakage of the working fluid into the machinery space is not expected to expose the operating personnel to excessive radiation, as the engine room exhaust ventilation system would rapidly remove the gas. The thermal characteristics of nitrogen are such that the head requirements of the power turbine can be met with a single-stage radial inflow wheel resulting in a simple reliable system for reversing and control. Furthermore, a small nitrogen producing plant could be provided on board ship. The compressor/turbine set is illustrated in Fig. 3. It is a three-bearing assembly consisting of the low pressure com-

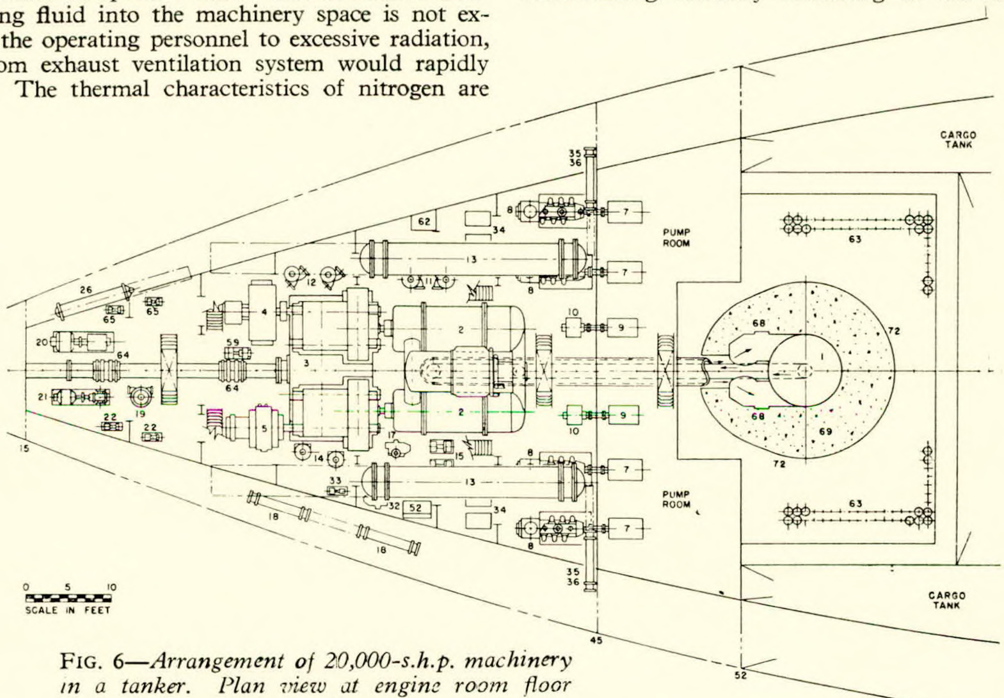


FIG. 6—*Arrangement of 20,000-s.h.p. machinery in a tanker. Plan view at engine room floor level*

(1) Nuclear reactor; (2) Propulsion gas turbines; (3) Main reduction gear; (4) Auxiliary motor; (5) Auxiliary generator; (6) Diesel generators; (7) Cargo oil pumps; (8) Cargo oil pump Diesel engines; (9) Stripping pumps; (10) Stripping pump motors; (11) Salt water circulating pumps; (12) Fresh water circulating pumps; (13) Heat exchangers; (14) Gear lub. oil pumps; (15) Turbine lub. oil pumps; (16) Turbine lub. oil pumps; (17) Lub. oil purifier; (18) Gear lub. oil coolers; (19) Fire pump; (20) Fire and Butterworth pump; (21) Bilge pump; (22) Sanitary pumps; (24) Sanitary compression tank; (26) Butterworth heater; (27) Main switchboard; (28) Main operating station; (29) Distilling plant; (31) Heating boiler; (32) Diesel oil purifier; (33) Transfer pump; (34) Lub. oil clarifiers; (35) Lub. oil coolers; (36) Fresh water coolers; (38) Air conditioning condensers; (39) Refrigeration compressors; (40) Refrigeration condensers; (41) Gas transfer compressors; (45) Lub. oil storage tank; (46) Lub. oil settling tank; (47) Lub. oil sump tank; (48) Lathe; (51) Workbench; (52) Purifier workbench; (55) Lub. oil gravity tanks; (56) Diesel oil service tank; (59) Salt water service pump; (62) Main circulating sea chest; (64) Bearings; (65) Diesel generator booster pumps; (68) Gas working fluid shut-off valves; (69) Concrete reactor shielding; (70) Air vents; (71) Top shield plug; (72) Metal shield; (73) Trunk

pressor, its rotor supported by a journal bearing at the inlet end. This rotor is coupled to the common shaft of the high pressure compressor and high pressure turbine unit, which is supported by two journal bearings. The thrust bearing is located at the low pressure compressor inlet end. The turbo-machinery of the compressor/turbine set is of axial flow design. It operates at a constant speed of 7,200 r.p.m. The low pressure compressor is of barrel type construction. Its radial joint, which is attached to the high pressure compressor/turbine housing, is subject to the discharge pressure of the low pressure compressor. The "wrapped around" intercooler arrangement results in a flow path of the gas leaving the low pressure compressor at the outside of the high pressure assembly. The nitrogen enters the low pressure axial-flow compressor from the two precoolers through a bifurcated inlet. After being compressed, the gas flows in a general radial direction to the outer face of a radial inflow plate pin intercooler surrounding the high pressure compressor set. The gas is then compressed in the high pressure compressor from where it is discharged into an annular space surrounding the hot end of the high pressure turbine. The compressed nitrogen is led to the recuperator through the outside space of a coaxial pipe. The efficiencies of the compressor/turbine set were chosen conservatively to be 86 per cent for the l.p. compressor, 85 per cent for the h.p. compressor and 89 per cent for the h.p. turbine. The pressure ratios for the low pressure and high pressure compressors were selected as 2·2 : 1 and 2·0 : 1 respectively. Fig. 6 shows a typical arrangement of machinery. It will be noted that the space in the engine room is more than ample for the equipment required and that the volume allocated to machinery is considerably less than for a steam power plant of similar power. Operation of the plant can be centralized from a control station on the auxiliary machinery flat level. A comparison of the space allocated to the machinery for this design compared to a recent design for a 20,000 shaft horse power tanker steam power plant is shown in the table, as well as a comparison with a pressurized water nuclear power plant of the same power.

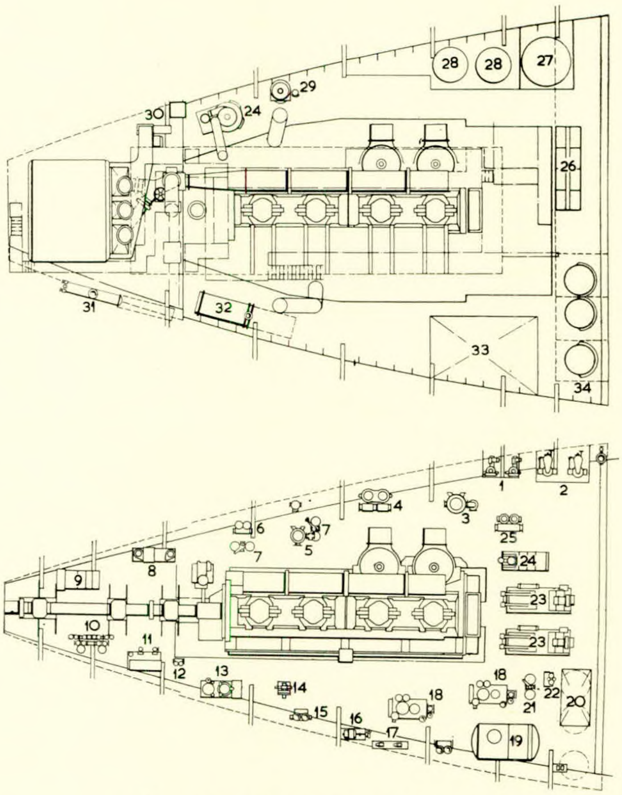
Type of plant	Gas turbine closed cycle nuclear	Steam turbine pressure water nuclear	Steam turbine oil-fired boilers
	Volume in cubic feet		
Main engines and auxiliaries	136,000	230,600	162,000
Engine room casing	2,500	37,400	57,700
Reactor and shielding	59,900	28,700*	—
Steam boilers and auxiliaries	—	—	104,500
Total spaces	198,400	296,700	324,200

*Excluding fuel oil and feed water shield tanks

This table indicates that the space required for the complete conventional steam plant is more than the other two designs, while the total volume required for the gas turbine nuclear plant is considerably less than either of the other two designs. In the case of the ship under consideration, this will probably result in a total machinery space volume less than the normal 13 per cent required to obtain maximum tonnage reductions. —*R. P. Giblon and G. H. Kurz, The Motor Ship, March 1957; Vol. 37, pp. 518-522.*

Large Raised Quarterdecker

In 1952 John Readhead and Sons, Ltd., South Shields, built for the Hudson Steamship Co., Ltd., the *Hudson Deep*, 7,800 tons deadweight, which was then the largest ship of raised quarterdeck type in service, previous vessels of this type not having exceeded about 4,600 tons d.w. Since then the same shipyard has built ships similar to the *Hudson Deep* for the Stag Line, Ltd., and Wm. France, Fenwick and Co., Ltd.,



Key to engine room arrangement

(1) Lubricating oil purifiers; (2) Heavy oil pump; (3) Jacket water cooler; (4) Ballast and standby salt water pump; (5) Lubricating oil cooler; (6) Standby lubricating oil pump; (7) Lubricating oil suction strainer; (8) Valve cooling pumps and tank; (9) Feed filter tank; (10) Boiler feed pumps; (11) Oil fuel unit; (12) Oil fuel unit suction strainer; (13) Auxiliary condenser air and circulating pump; (14) H.p. fuel oil priming pump; (15) General service pump; (16) Cooling water pump; (17) Feed water pumps; (18) Air compressors; (19) Oily water separator; (20) Recovered oil tank; (21) Oil fuel suction strainer; (22) Oil fuel transfer pump; (23) Diesel generators; (24) Turbo-generator; (25) Standby jacket water pump; (26) Switchboard; (27) Distilled water storage tank; (28) Clean lubricating oil tanks; (29) Evaporator; (30) Forced draught fan; (31) Feed water heater; (32) Auxiliary condenser; (33) Engineers' store; (34) Starting air receivers

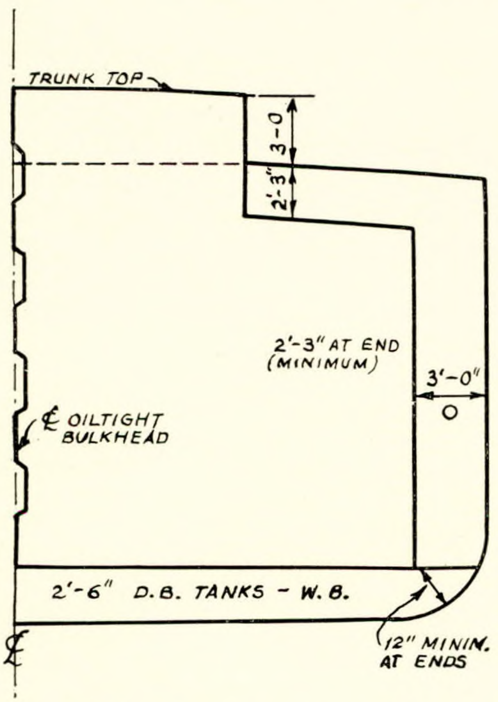
but the *Hudson Point*, which ran acceptance trials in January, carries the raised quarterdeck design a marked stage further by advancing the deadweight capacity to 10,560 tons. The ship has been designed specially for the carriage of bulk cargoes, for which purpose her five self-trimming holds, all arranged forward of the machinery, are equipped with MacGregor patent sliding steel hatches. The main propelling machinery consists of a four-cylinder opposed piston, airless, injection balanced oil engine of the diaphragm type manufactured by William Doxford and Sons, Ltd., and installed by the shipbuilders, who also supplied and installed the boilers and auxiliaries. The cylinders have a diameter of 670 mm. and a stroke of 2,320 mm., developing 4,400 b.h.p. at 112 r.p.m. The engine is fitted with two lever-driven scavenging air pumps. The following pumps are driven from the scavenging air pump levers attached to No. 1 centre connecting rod: one double-acting forced lubrication oil pump with a capacity of 45 tons per hour; one sea water cooling pump having a capacity of 270 tons per hour; and a jacket and piston distilled-water cooling pump with a capacity of 200 tons per hour. A Coastguard oily water separator with a capacity of 20 tons per hour is installed. There is one main boiler with three oil-fired furnaces arranged to accept exhaust gas in wing furnaces only, the working pressure being 120lb. per sq. in. The electrical installation comprises one 40-kW steam driven generator by Sunderland Forge and two National Diesel-driven 60-kW generators.—*The Shipping World, 20th February 1957; Vol. 136, pp. 221-223.*

Petroleum Chemical Tankers

Two tankers of an interesting design have been ordered for the carriage of petroleum chemicals in bulk. The vessels, which will be of about 1,700 tons d.w., have been designed to the order of Chemical Tankers (Bahamas), Ltd., a subsidiary of Gel Tankers, Ltd., London. The principal characteristics of the new vessels are as follows: —

Length b.p.	242ft.
Breadth moulded	40ft.
Depth moulded	19ft.
Draught, mean	14ft. 3in.
Draught, maximum	15ft. 6in.
Deadweight	1,700 tons
Service speed	13 knots
Cruising range	5,000 miles
Fuel consumption (about) ...	8 tons per day

The vessels have been designed to conform to Lloyd's Classification ✠ 100 A1 "carrying petroleum in bulk", and to



Midships section of tanker for carrying petroleum chemicals in bulk

Ministry of Transport regulations; also allowing for U.S. Coast Guard Regulations for Tank Vessels, Manchester Ship Canal regulations, Suez Canal regulations, and the St. Lawrence River Pilotage regulations. Each ship will have five compartments divided by cofferdams, double sides and double bottoms—to avoid contamination—to carry either a single, or up to five compartment cargoes entirely separate (one being a dry or drum compartment convertible into a tank). The cargo deadweight capacity will range from 1,000 tons at a specific gravity of 0·67 to 1,500 tons (at S.G. 1·00 or higher) to which should be added 200 tons for fuel oil and supercargo: i.e. 1,200 to 1,700 tons d.w. altogether, depending on the specific gravity of the cargo carried. The tanks are flush welded, polished and clear of all internal obstructions. No. 1 tank will be of stainless steel and the remaining tanks will be corrosion resistant, a feature which will facilitate the use of the latest chemical-cleaning methods. Each compartment will be equipped with bottom drainage, the pump suction connecting with the outlet in the well. No stripping pumps will be installed and there will be no wastage when discharging. The pipe connexions pass through the ballast tanks and cofferdam spaces, each compartment having its own pumps. Apart from a loading pipe running vertically from the hatch covers to within a few inches of the bottom of the tanks, in order to avoid foaming and excessive electrostatics during loading, there are no internal obstructions. A feature of the vessels is the inert gas generator which is installed on the upper deck abaft the pump room. This unit permanently impregnates the entire vessel forward of the engine room with a gas mixture consisting mainly of nitrogen, thereby eliminating any spontaneous combustion, and the effects of oxygen corrosion. This unit is a development of the original fire fighting equipment designated the Pyrene-E.D.-Hol system which was first installed in the cargo vessel *Oti.* The development was made by Gel Tankers, Ltd. Instead of the Polar Diesel engine originally planned and shown in the drawing, the vessels will each be propelled by an exhaust gas turbine driven by two GS.34 free piston gasifiers, and coupled to a variable pitch propeller through reduction gearing. The machinery, which will have an output of 2,000 b.h.p. at 250 r.p.m., will give the vessels a service speed of 13 knots over a cruising range of 5,000 miles with a fuel consumption of about 8 tons per day. Both turbine and variable pitch propeller will be controllable from the bridge as well as from the engine room. It is intended to operate the propelling machinery on heavy boiler fuel.—*The Shipping World, 26th December 1956; Vol. 135, pp. 561-562.*

Viscosimeter for Fuel Oil

A Dutch viscosity meter is now being marketed in Great Britain. A compact and relatively simple unit, this instrument was developed by the Vloeistofmeetapparatenfabriek, of Rotterdam, and has been used over a long period on one of the engines in the motor passenger liner *Wilhelm Ruys.* It is claimed to be very satisfactory in operation. The apparatus is fitted in the fuel supply line between the fuel heaters and the high pressure pump and consists of a fractional horse power electric motor driving, through gears, a precision pump. The pump extracts a small but constant volume of oil from the flow and forces it through a capillary tube. By measuring the pressure drop across the capillary a continuous reading of viscosity is obtained and this is clearly shown on a differential gauge. As the capillary tube, where the measurement is made, is immersed in, and is at the same temperature as the fuel to be measured, it is immaterial where the indicator gauge is situated. Thus the apparatus can be sited in the fuel line and the gauge placed in a central manœuvring control position without lessening in any way the accuracy of the readings due to loss of temperature. Furthermore, as the gauge is differential, fluctuations in pressure in the fuel line should have no effect on the accuracy of readings. The Viscotherm, as the instrument is named, is installed with bypass valves in the fuel system so that it can be taken out of operation and dismantled should it be necessary. Calibration is effected by flushing the instrument with a suitable solvent and filling it with oil at room temperature. Drain and filter plugs and a thermometer connexion are provided in the discharge end of the unit for this purpose. The gauge is calibrated for individual requirements. A red sector indicates that viscosity is too high and the operator then reads off against the pointer the number of degrees by which the temperature must be increased to correct the viscosity. He then makes such adjustment as may be necessary to the heater temperature. This instrument appears to have the following advantages: (1) It gives a continuous and direct reading. (2) It is suitable for remote presentation. (3) It could be used for automatic control. (4) Remedial action is clearly indicated. It would seem also that, apart from its use in engine fuel lines, there is an application for such an instrument for the measurement of cargo oil viscosity in tankers.—*The Motor Ship, January 1957; Vol. 37, p. 463.*

Novel Steel Hatch Cover

Recently a new type of steel hatch cover for ships has been made available under the name of Ermans Steel Rolling Hatch Cover. In order to satisfy all requirements of shipping companies as to the closing of hatches the inventor started from the following considerations: (a) Closing and opening time of hatches must be as short as possible; (b) Watertightness must be effected without having recourse to tarpaulins; (c) Stowed hatch covers must occupy a minimum amount of space so that decks around hatches are as free from obstacles as possible; (d) If coamings are deformed by blows from hoisted loads watertightness of the covers should not be jeopardized, nor the ease of handling them. The inventor developed a form of construction in which the covers are composed of hinged elements; each element has a section parallel to the ship's length and has a section modulus as large as possible, while at the same time being easily stowed. Fig. 2 shows the cover partly opened. Fig. 3 gives an impression of the cover completely stowed. The elements vary in breadth in accordance with the space they need when rolled around the stowing axis. Two consecutive elements always have the same breadth. The elements nearest the axis are, of course, the smallest. By this system elements are stowed inside each other so that storage room is as small as possible (see Fig. 3). Tightness is ensured by rubber strips reinforced by double linen inserts. The covers can be manipulated in three ways: (a) electrically; (b) by means of a winch; (c) by hand. (a) The electrical drive is fully automatic. The rolling winch needs a 4-5 h.p. motor with gearing. Opening or closing each require a time of about one minute. Handling this winch is so simple that it can be done eventually by unskilled hands. (b) If a wiredrum is mounted on the shaft of the rolling winch this can be driven by a cargo winch so that the cover is rolled up and stowed. To unroll the cover one can fix a wire to an eye on the last cover element and lead the wire via a block to the cargo hook. (c) The rolling winch can also be moved by hand and this needs a worm drive. This possibility is necessary to ensure against the chances of a failure of the electricity supply or damage to cargo winches. Because the covers glide freely over the hatch coamings these latter can be deformed transversely to a certain extent without hampering the efficient functioning of the covers. If hatches are of extraordinary length it is advisable to have a rolling winch at each end. The two parts of the cover are connected watertight in the closed condition. A cover in two parts gives the additional advantage that half of the hatch can be kept covered while loading or unloading goes on through the other half.—*International Shipbuilding Progress, January 1957; Vol. 4, pp. 40–42.*

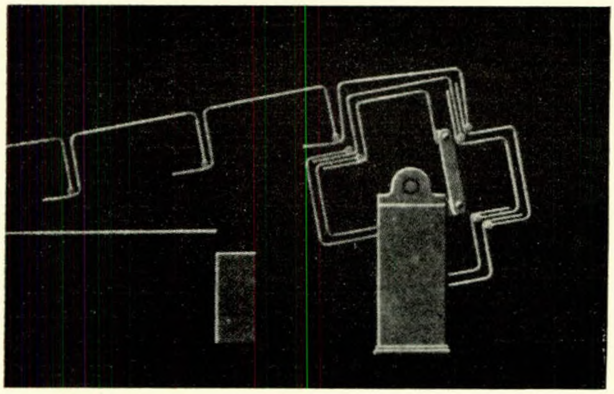


FIG. 2

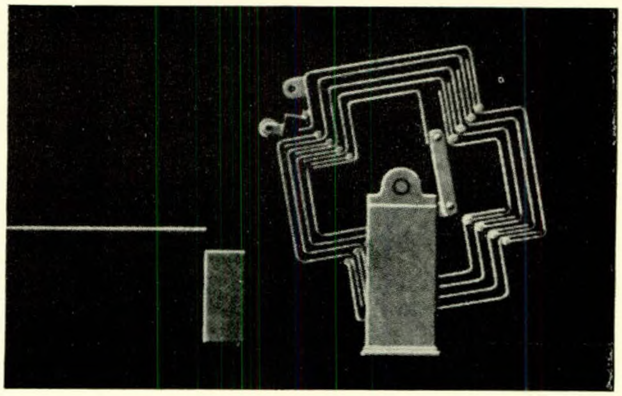


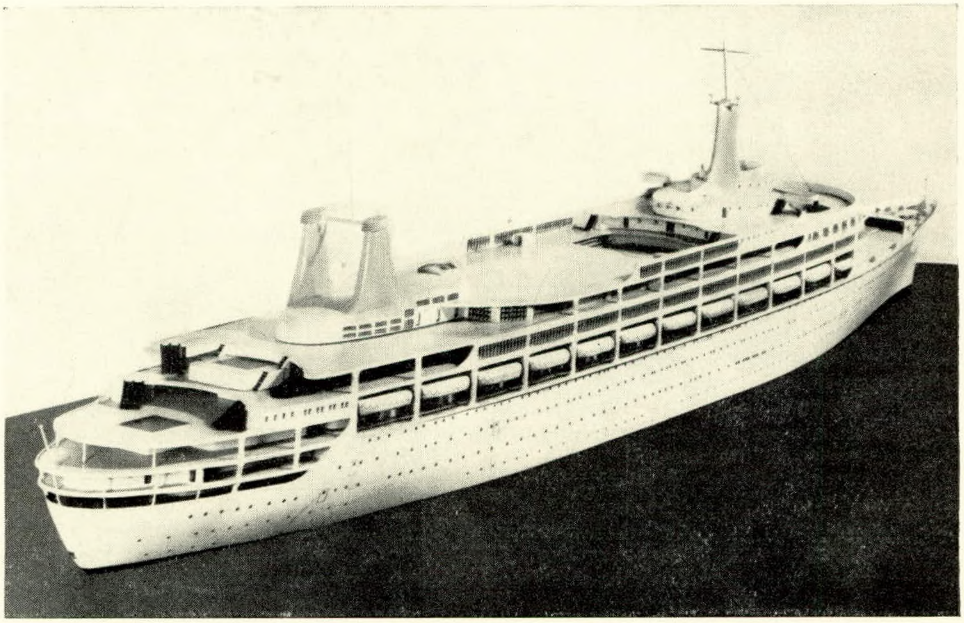
FIG. 3

Hydraulic Trawl Winches

For many years trawlers were propelled by steam reciprocating engines, and, consequently, were fitted with steam driven trawl winches. In recent years, however, there has been a tendency to replace the steam engine with Diesel prime movers, and one of the problems is to obtain a suitable alternative drive for the trawl winch. Although, in some smaller vessels, a direct drive from the main engine through shafting and V-belts, etc., has been used, the majority of the larger vessels have been fitted with an electric drive to the trawl winch. An electric drive, although fulfilling the main requirements for this particular purpose, is considered by many trawler owners not to give the same flexibility as the steam-driven winch, and the maintenance, servicing and repair of electric drives have also been somewhat of a problem. The hydraulic drive provides similar characteristics to those of the steam driven trawl winch, and, in particular, the winch is able to stall for long periods without detriment to either the hydraulic drive or mechanical portion of the winch. The combination of hydraulic transmission with a specially designed trawl winch provides a winch installation which is robust, easy to control, resistant to weather and ideally suited for direct drive from a Diesel auxiliary or main propulsion engine. VSG hydraulic variable-speed gear is used to transmit the power of the Diesel engine to the trawl winch, and to regulate the speed and pull of the winch. The variable speed gear consists of two units, a hydraulic pump and a hydraulic motor. It is usual to have the pump situated in the engine room, driven from the engine through a flexible coupling. The pump is connected by steel piping to the hydraulic motor driving the winch. The motor may be out on deck immediately behind the winch, or housed in the forward part of the superstructure, driving the winch through a short length of shaft. A safety device is fitted to the pump to prevent overload. This simple, reliable device consists of a hydraulic piston acting against a spring, and arranged to function as an overriding control which responds to the hydraulic pressure. The safety device comes into operation in the event of undue pull arising from the trawl. Should the trawl foul an underwater obstruction, the speed of the winch is automatically reduced, and if the pull becomes excessive, the winch will come to a standstill and stall at its maximum haulage effort.—*The Shipbuilder and Marine Engine-Builder, February 1957; Vol. 64, p. 127.*

Largest P. and O. Liner

The P. and O. Steam Navigation Company has released details of the new 45,000 tons gross passenger liner which is to enter service on the now extended U.K.-Australia-Vancouver route. A greatly increased speed of 27½ knots will enable the voyage time from the United Kingdom to Sydney to be reduced from four to three weeks. The new ship, which at



Twin side-by-side funnels, screened weather decks, and lifeboats at a low level will be features of the new ship. The hull will be all-welded

present is known only as No. 1,621, will have the following principal particulars:—

Length overall	814ft. 0in.
Length between perpendiculars	740ft. 0in.
Moulded breadth	102ft. 0in.
Loaded draught	31ft. 6in.
Cargo capacity	150,000 cu. ft.

The machinery will be installed aft and the design takes the fullest advantage of this layout. There will be thirteen decks and the all-light alloy superstructure will incorporate some 800 tons of this material. The lifeboats will be carried three decks below the uppermost deck. Two sets of Denny-Brown ship stabilizers will be provided. The ship is designed to accommodate 600 first class, 1,650 tourist class passengers and 960 crew. After an interval of over twenty-five years the P. and O. Company is returning to turbo-electric propulsion. It will be recalled that the *Viceroy of India,* built in 1928 and lost during the North African landings, was the first large turbo-electric passenger vessel. The *Strathnaver* and *Strathaird* of 1931 are still in service. The British Thomson-Houston Co., Ltd., will again be responsible for the machinery, which will consist of two single-cylinder main propulsion turbo-alternator sets taking steam at 700lb. per sq. in. and 950 deg. F. These will be of the latest power station design and will supply current to a pair of synchronous propulsion motors developing a total of 85,000 s.h.p. The arrangement will provide a considerable degree of flexibility as one turbo-alternator can be shut down while the ship is run on the remaining one. This can represent a considerable saving on the Australian coast where certain legs of the voyage are run at reduced speed in order to avoid arriving during the hours of darkness. The turbines are unirotational, and manœuvring is carried out on the propulsion motors. Auxiliary power will be provided by four 1,500 kW 440-volt, three-phase 60-cycle turbo-alternator sets.—*The Marine Engineer and Naval Architect, February 1957; Vol. 80, p. 43.*

Screw Pumps

A British manufacturer offers a range of positive displacement self-priming pumps of the screw type. These are said to be particularly suitable for pressure-oil-fired installations, cooling and lubricating Diesel engines, transfer duties in refineries and on ships, turbine lubricating, hydraulic presses, transferring viscous fluids, fats, oils, emulsions, and numerous other liquids. For normal pumping duties, internal bearings are fitted, and therefore the liquid being pumped must have reasonable lubricating properties. Certain sizes, however, are available with outside bearings and/or steam jacketing, and these are capable of dealing with fluids having no lubricating properties whatever. Sizes range from the smallest two-screw type with a capacity of 30 gal./hr., operating at pressures up to 1,500lb. per sq. in., to the largest three-screw type with a capacity of 66,000 gal./hr., usually arranged to operate at a maximum pressure of 300lb. per sq. in. The only moving parts are the intermeshing screws and these are of high quality steel, hardened and ground on their contacting surfaces. The screws are designed so that completely enclosed chambers are formed between the threads, and when they are rotated these chambers move in an axial direction without turbulence or pulsation. The screw spindles are supported on special bearings and an easily accessible gland is fitted to the driving end of the casing. Mechanical seals can be fitted to suit customers' individual requirements. When a relief valve is fitted, it is of the spring-loaded, mushroom-headed type, adjusted to operate at approximately 10 per cent above normal working pressure. —*Engineering and Boiler House Review, January 1957; Vol. 72, p. 21.*

Patent Specifications

Pitch Adjusting Gear for Variable Pitch Propeller

This invention relates to pitch adjusting gear for the blades of variable pitch propellers and the object of the invention is to provide a gear construction which is applicable even at the highest loads and numbers of revolutions of the propeller. In the drawing (1) is the propeller shaft which is made with a central bore and secured to the hub (3) through a flange (2). The hub (3) is open at the aft end and provided with a cap-like closure (4) and a suitably shaped end cap (5). The hub (3) is provided with openings for the accommodation of the individual propeller blades (6), each of which is provided with inner and outer mounting faces (7) and (8) respectively. The inner blade base part (9) forms a crank disc, and is connected to the outer part (10) by means of bolts (13). A sealing ring (14) is provided between the outer mounting face (8) and the blade base part (10) for the purpose of preventing water entering and oil from leaving the hub along the blade base shaft. The crank disc (9) is provided with two integral pins (15) and (16) respectively, situated approximately diametrically to the axis of rotation indicated at II-II in Fig. 1, of the blade. The propeller shaft (1) is formed with a bore, and a rod (17) and a tube (18) are mounted in the bore, coaxially to each other and to the shaft (1). The rod (17) extends through the hub (3) into the cap-like closure (4) and its end carries a crosshead (19) having a number of pins (20) corresponding to the number of blades of the propeller. The crosshead (19) is guided by means of ribs (21) provided in a guide member (22). Each pin (20) carries a crank arm (23), the opposite end of which is mounted on one of the pins (15) of the crank disc (9). Consequently an axial movement of the rod (17) will cause a crank-like movement of the pins (15) and thereby a rotation of the crank disc (9) together with the blades (6) about their respective axis of rotation.—*British Patent No. 765,647, issued to N. J. Liaaen. Complete specification published 9th January 1957.*

Fig. 1

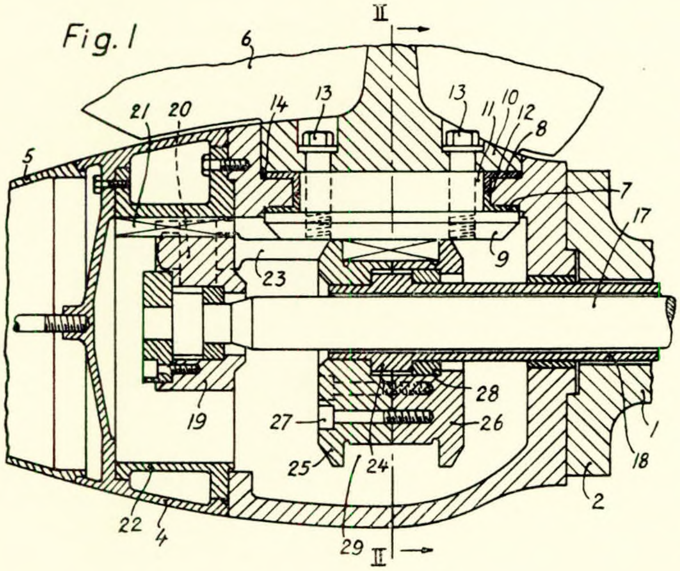


Fig. 3

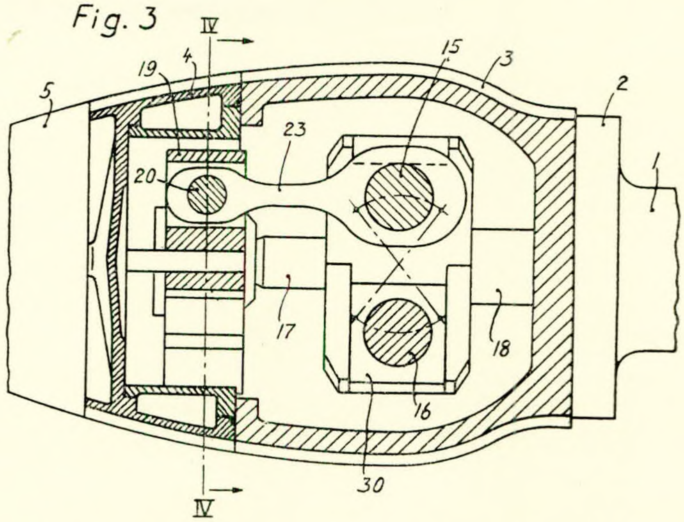
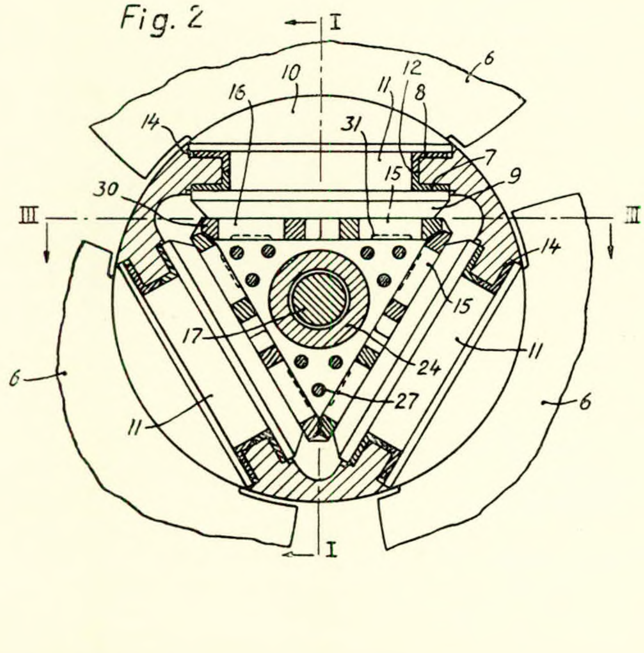


Fig. 2



Separation of Oil from Contaminant Liquids

This invention is primarily directed to the provision of methods and apparatus for avoiding the contamination of sea water with oil and for rendering usable again the oil residues from the cargo tanks of oil tankers. A further object is to provide apparatus for such purposes which may be readily built into oil tanker vessels or subsequently provided in oil tanker vessels already constructed. Referring to Figs. 1-4, the tanker (1) has storage tanks (2, 3, 4, 5, 6, 7, 8, and 9) which are separated by the cofferdams (10 and 11) against the fore and after parts of the ship. Two separation chambers (12, 13) are arranged over the storage tank (2), the cofferdam (10) being connected by an opening (14) with the chamber (12). A mixture of oil and water formed during the cleansing of the storage tanks is fed into the cofferdam (10), fills the cofferdam

Fig. 1

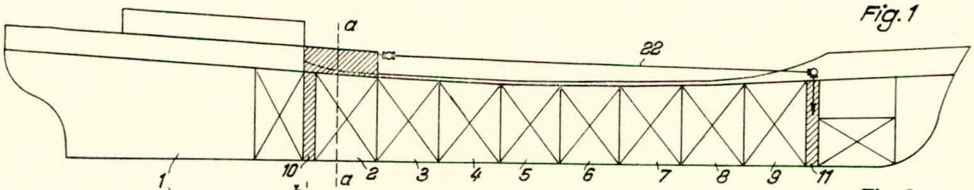


Fig. 2

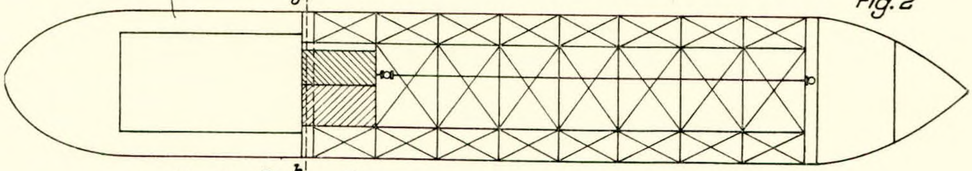


Fig. 4

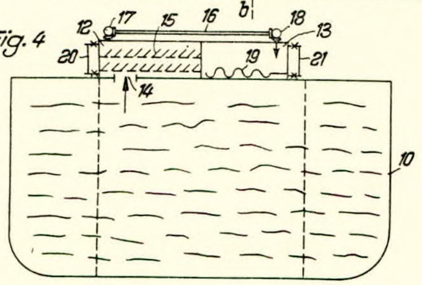
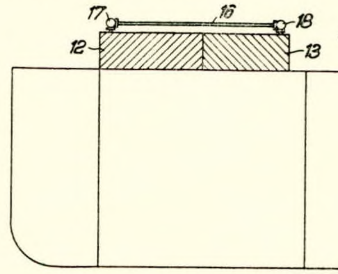


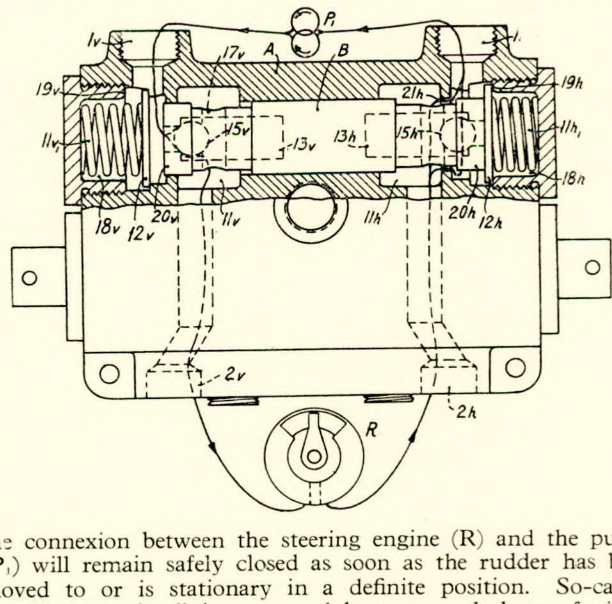
Fig. 3



and partly separates into two layers by reason of the difference in density of oil and water. The oil layer floating on the surface of the water passes into the chamber (12) through the opening (14), the chamber (12) being provided with apparatus for effecting or accelerating a mechanical separation of the oil from other liquids present. For further separation of the water or other liquid from the oil, the mixture passes through a delivery pipe (16), which is placed over the chambers into the chamber (13). The passage of the liquid through the tube (16) is controlled by an automatically rising float device (17, 18). Heating apparatus (19) is provided in the compartment (13). Indicating devices (20 and 21) are provided for showing the height of the oil in the chambers (12 and 13). Transfer of oil (15) into the fore cofferdam (11) takes place through the pipeline (22) which may for instance be an overflow pipe.—*British Patent No. 765,058, issued to Deutsche Werft A.G., Hamburg, Germany. Complete specification published 2nd January 1957.*

Steering Engine

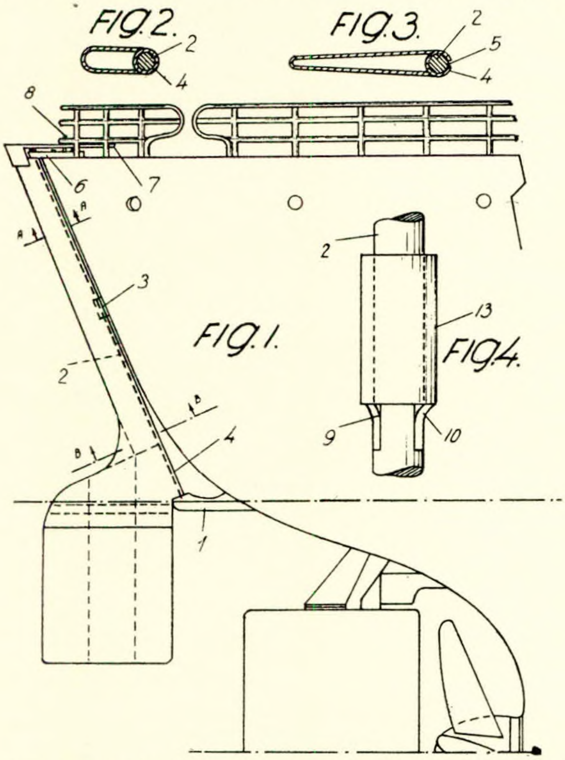
This invention relates to hydraulic control means for hydraulic or electro-hydraulic steering engines in ships wherein a reversible pump is used. Referring to Fig. 1, when the reversible pump (P_1) is started in one direction of rotation and pumps oil through the pipe conduits in the direction indicated by the arrows, the oil passes from the pressure side (to the left) of the pump and passes in at (1v) to the left space (18v) and presses against ball (15v) in the direction opposite to that of the spring action, so that the valve is opened. In contradistinction to the foregoing the oil on the suction side of the pump has a pressure lower than that on the pressure side, and as a consequence, the oil presses the whole slide B to the right, compressing thereby the right-hand spring (11h) until the ring (12h) contacts an annular surface (19h) in the housing. Slide B now occupies the position shown in Fig. 1 in which a head (20h) on the slide opens an annular aperture (21h) which places the annular space (11h) in direct communication with the socket (1h). The oil flow now passes from the pressure side of the pump (P_1) through socket (1v), space (18v), bore (13v), past the valve ball (15v) and through the annular space (11v) to the left-hand side of the steering engine (R) in order to act upon the rudder pin. The oil from the right-hand side of the steering engine is pressed through the annular space (11h), the annular aperture (21h) and the socket (1h) on to the suction side of the pump (P_1). The rudder is turned until the pump ceases to operate. The ball (15v) is then pressed against its seat by the spring (14v), and the slide B is moved by spring (11h) back to the central position. The connexion between (P_1) and the steering engine is now interrupted, and the rudder remains stationary in the position which it has attained. However, if the direction of rotation of the pump (P_1) is reversed, the slide B is of course pushed to the left by the oil, so that the ring (12v) contacts the annular surface (19v). The oil flows in a direction opposite to that mentioned above, and the rudder is also turned in the opposite direction. By this arrangement



the connexion between the steering engine (R) and the pump (P_1) will remain safely closed as soon as the rudder has been moved to or is stationary in a definite position. So-called "rudder creeping" is prevented because no leakage of oil in the reversible pump (P_1) can take place, the above-described communication being interrupted automatically as soon as the pump stops. For the same reason a strike from the rudder cannot be transmitted further than to the slide B. If the rudder "strikes" in the same direction as the oil is pumped, the pressure from the opposite side will disappear at the same moment as the strike on the oil takes place. The slide B will then rapidly go back to its central position and interrupt the connexion. If the rudder " strikes " in the opposite direction the valve will close.—*British Patent No. 768,274 issued to J. K. Tenfjord. Complete specification published 13th February 1957.*

Jury Rudder

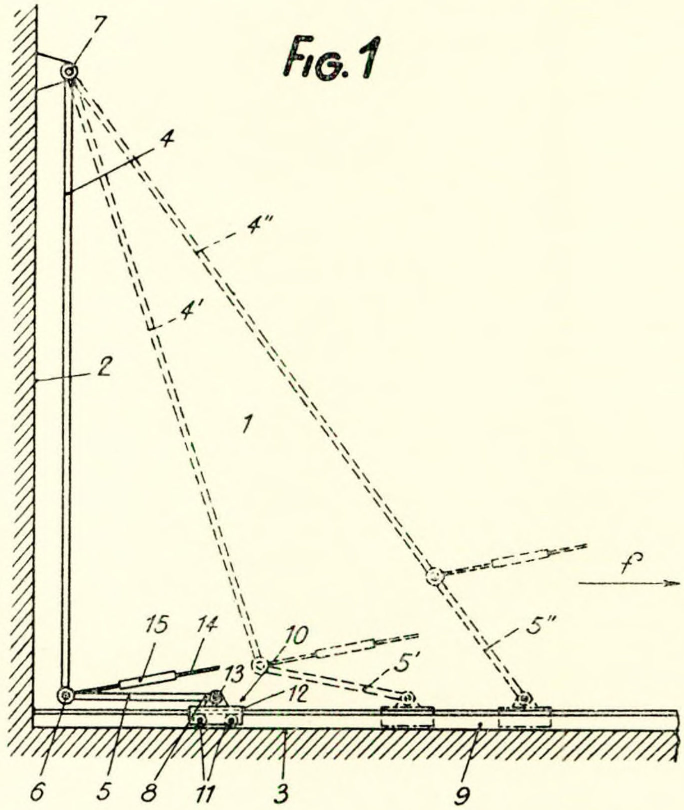
This invention relates to an improved jury rudder for ships of the kind equipped with eye bolts or a tubular member adapted to be fitted on a stern bolt at the aft end of the ship. Referring to Figs. 1-4, the stern of the ship is provided at its lower part with an extending bracket (1), into which a stern bolt (2) is fitted. The bolt (2) extends upwards to above the



deck line, where it is detachably connected by an ordinary locking device. In order to brace the bolt, one or more additional bearings or locating members (3) may be arranged between the bracket (1) and the top of the bolt (2). On the back of the jury rudder there is fixed a tubular member (4) which may be an integral part of the rudder. The rudder may be manufactured of aluminium, one of the plastics or like material in order to reduce the weight of the rudder. The bearing member (3) necessitates that the tubular member (4) should have a slot in (5) in the lower part (Fig. 3). The slot (5) allows the tubular member to pass the member (3) when the rudder is lowered on to the bolt (2). The upper part of the tubular member (4), however, is unslotted. When the rudder has to be applied the locking device (6) is removed to free the top of the bolt (2) and thus enable the member (4) to be engaged with the bolt top and the rudder, then lowered down to about the bracket (1), whereafter the top of the bolt (2) is fixed to the stern by the locking device (5). The latter consists, for example, of a plate slidable on a guide pin (8). A lock nut can then be secured to the pin (8). Instead of the tubular member (4), the rudder may be secured with eye bolts. The member (3) is not necessary for a proper use of the rudder which may be mounted in place by the use of the bracket (1) and the detachable means provided at the top. A tiller (7) may be connected to steering tackle arranged on deck. —*British Patent No. 769,153 issued to O. Brown. Complete specification published 27th February 1957.*

Handling Cargo in Ship Holds

This invention claims to provide a device which allows to gather the bulk material at the vicinity of the central portion of a ship hold. The device is of the type comprising a partition or wall assembly consisting of one or more hingedly interconnected elements in the vicinity of the lateral walls of the hold in which the goods are loosely stored, and means to displace this movable partition towards the central portion of the hold, while giving a variable inclination to the component



elements of this assembly. The device therefore aims at overcoming the difficulty experienced at the end of the unloading operation to reach for the goods located in the remotest lateral places and particularly in corners hardly accessible to the grab bucket or other means employed for unloading the ship.—*British Patent No. 765,644, issued to P. Legrand, Paris, France. Complete specification published 9th January 1957.*

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Marine Engineering and Shipbuilding Abstracts

Volume XX, No. 6, June 1957

A New Fitting for Hydraulic Pipelines

A new patented steel fitting of high quality and remarkable strength has recently been introduced. The new unit is not a true compression fitting and does not rely on pure friction or compression for its strength of joint. The compression principle is used to make the joint in the first place, but any change in the compression forces, due to expansion or contraction, fatigue or vibration, will not affect the efficiency of the joint. The fittings are satisfactory for a working pressure of 10,000lb. per sq. in., under the correct conditions and with correct sizes of tube. The fittings are made from steel bar and forgings to a suitable British Standard Specification, this being a manganese toughened quality steel, not of the free cutting varieties. The gripping ring, or cone, is made from a similar steel, and is suitably heat treated, in modern plant and under carefully controlled conditions, to ensure satisfaction in use. To make a joint, the tube is cut off square and the burrs removed. The tube is then pushed into the fitting as far as the stop face, and held there while the coupling nut is tightened. After tightening by hand, as near as possible to one full turn of the coupling nut should be given, to ensure maximum joint strength. The smooth radial nose of the cone is pushed against the bevelled seating of the fitting. This causes the end of the cone to curl slightly and to close down on to the tube. Further movement of the cone forces the heat treated edges of the cone into the surface of the tube, thus cutting the surface like a lathe tool. Continued movement forces the edges of the cone deeper into the tube and, in addition, the sliding action on the tube raises a substantial burr in front of the cone. This burr is one of the main factors in the strength of the joint. The cone acts as a cutting tool, and will continue to cut so long as there is "chip" clearance. In this case, however, the "chip" clearance is limited to the amount of space between the nose of the cone and the bevel of the fitting. Once this space is filled, the cutting action ceases and the tube is locked in position.—*The Shipbuilder and Marine Engine-Builder, January 1957; Vol. 64, pp. 60-61.*

Welded Aluminium Deep Tank Hatch Cover

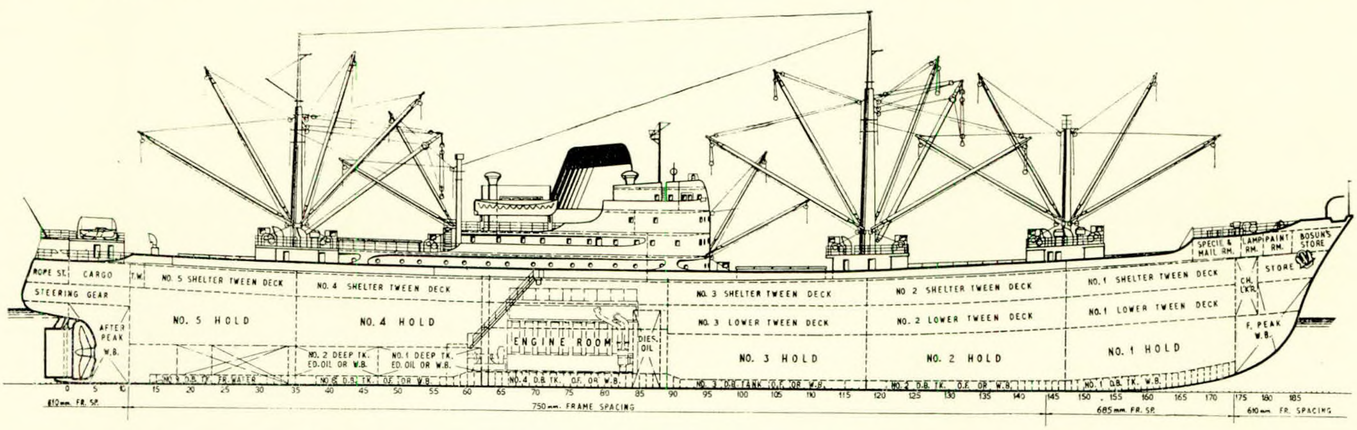
An interesting feature of the motorship *Kepwick Hall*—an open shelterdeck cargo motorship of 9,480 tons deadweight, recently completed by William Doxford and Sons (Shipbuilders), Ltd., for the West Hartlepool Steam Navigation Co., Ltd.—is the welded aluminium deep tank hatch cover on the second deck. This cover spans an opening of about 22ft. by 15ft. 9in., and its weight is just under 2 tons. The use of aluminium has greatly simplified the lifting and handling of the cover, which, if constructed of steel, would have weighed more than 5 tons. The alloys used were NP5/6 (Noral B54S) for the plate and NE6 (Noral A56S) for the extruded sections. These are now almost standard alloys for welded structures to the requirements of Lloyd's Register of Shipping. The top plating of the cover is 0·44in. thick, and the fore and aft stiffeners, spaced 24in. apart, consist of 8-in. bulb plates from the range of special aluminium welding sections standardized in British Standard No. 2614:1955. Three fabricated girders, 14-in. deep with 7-in. face plates, run transversely and are slotted in way of the stiffeners. The cover is secured to the steel coaming by galvanized bolts, and a gasket of natural rubber is fitted between the faying surfaces. The whole of the welding has been carried out by the inert-gas metal-arc process, and as a structure of this type is ideally suited for automatic welding, a Sigma, type SWM3, controlled-arc set, mounted on a travelling carriage, was used for the greater part of the work. The increasing use of automatic welding for this type of construction, coupled with the well-known inherent advantages of aluminium, is resulting in worth while reductions in fabricating, labour and maintenance costs.—*The Shipbuilder and Marine Engine-Builder, March 1957; Vol. 64, p. 167.*

German-built Cargo Liner for British Owners

The cargo liner *Arabic,* first of the three ships ordered in Germany by the Shaw Savill Line, has completed successful trials from the yard of her builders, Bremer Vulkan. The remaining two ships, *Afric* and *Aramaic,* are at present fitting out and are due to be delivered early this year. They are general cargo carriers, without refrigerated holds, and are of 9,240 tons deadweight. Their speed is 17½ knots. In their general design these ships are orthodox in conception. They are arranged with propelling machinery amidships, having three holds forward and two aft. The propelling machinery consists (in the German-built ships) of a turbocharged M.A.N. Diesel engine, direct-coupled to the propeller. The ships are open shelterdeckers, and have two 'tweendecks in the forward holds and one in the after holds. The principal dimensions of the *Arabic* are as follows: —

Length o.a.	475ft.
Length b.p.	440ft.
Breadth moulded	64ft.
Depth to shelter deck	39ft.
Contract draught	27ft. 1¼in.

The gross tonnage of the *Arabic* is 6,497, and the net tonnage 3,349. The cubic capacity of the holds (including the cargo deep tanks) is 596,825 cu. ft. grain, and 533,650 cu. ft. bale.



The capacity of the cargo tanks is 29,795 cu. ft. The weather-deck hatch covers are of the MacGregor single-pull type. The propelling machinery of the *Arabic* and her sister ships consists of a Bremer Vulcan-M.A.N. turbocharged two-stroke Diesel engine of type K9Z 78/140C. This engine is similar to that fitted in the Blue Star vessels built by Bremer Vulkan except for the number of cylinders, the *Arabic* engine having nine cylinders as against the ten of the *Canberra Star* class. The nine-cylinder engine delivers 8,600 b.h.p. at 109 r.p.m. The turbocharging system used in the Bremer Vulkan-M.A.N. engine is a somewhat complex one, involving the use of exhaust driven turbochargers, underpiston charging and a Roots blower. It is one of two alternative systems that can be used with the two-stroke M.A.N. engine, one of which is a pulse system and the other a constant pressure system. This one is the constant pressure system. In it, air is supplied by Brown Boveri turbochargers (three of them in the case of the nine-cylinder engine) to the under side of the inlet manifold, which latter is divided into two halves by a horizontal division. Scavenge air to the pistons is supplied from the upper or secondary part of the manifold, and the connexion between the two is made by the underpiston charging circuits, acting in parallel with the Roots blower. Although apparently complex, this system works well, and it has the advantage that various combinations of its members can be used in the event of any one of them breaking down. It is thus possible to dispense with the electrically-driven emergency blower often fitted with simpler systems. The engine is arranged to burn boiler oil. Shaw Savill are prepared to take oil of up to 1,500 seconds viscosity, although in practice oil of 700 to 800 seconds is often burnt. Titan purifiers are fitted, those for the fuel oil being of the NS 70 self-cleaning type. After passing through the turbochargers, the exhaust gases go to three La Mont exhaust boilers mounted in the funnel, which supply the ship's steam requirements in conjunction with two Cochran oil-fired boilers.—*The Shipping World, 16th January 1957; Vol. 136, pp. 112-114.*

New Italian Shipyard

Work was commenced recently on the construction of a large shipyard to build ships of 65,000 tons d.w.c. and ultimately up to 100,000 tons deadweight at La Spezia, Italy. Already orders have been placed by British owners for a number of tankers. The new shipbuilding concern is the Cantieri Navali San Benedetto, which has taken over an area at La Spezia amounting to some 500,000 sq. m. On this are being built five building berths suitable for ships up to 65,000 tons d.w., two fitting-out quays and a large fitting-out dock to take two 65,000-ton ships and two of 35,000 tons. In addition, a dry dock is to be built capable of accommodating tankers of about 100,000 tons d.w. and this may be ultimately used for the construction of vessels of that size. The following orders have been placed or berths reserved: —

Two tankers of 19,250 tons d.w.c. for Dashwood and Partners, Ltd., London, each to have a Sulzer turbocharged engine. The first is due for completion in 1958 and the second in April 1959.

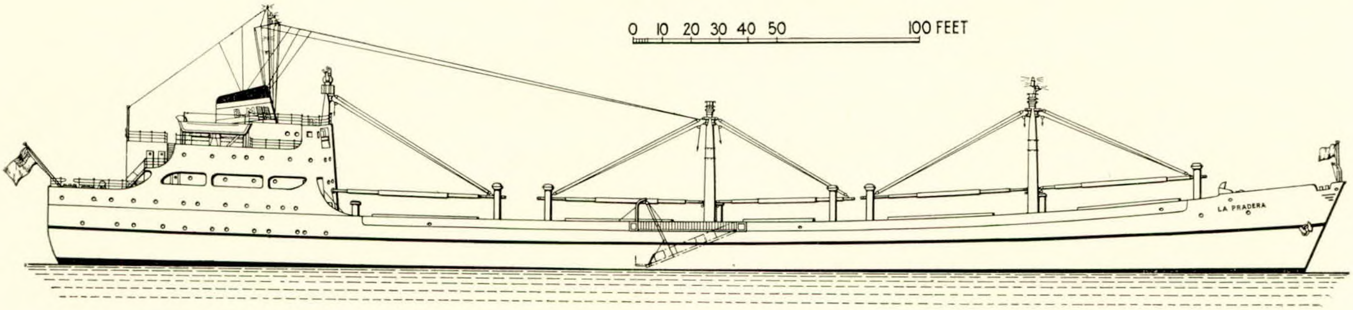
Two 19,500-ton turbine driven tankers for the Athel Line, Ltd.

Two tankers of 40,000 tons d.w.c. for clients of James Burness and Sons, Ltd. These will be 17-knot turbine-propelled ships and are due for completion in January and June 1959 respectively.

The turbine machinery is being supplied by Hawthorn Leslie (Engineers), Ltd., Brown Boveri and Westinghouse, U.S.A.—*The Motor Ship, January 1957; Vol. 37, p. 443.*

Bridge and/or Machinery Aft

By placing machinery aft there are two notable advantages as far as cargo space is concerned. Cargo can now be carried in the "best" part of the ship which is amidships and there is a slight but important gain in cargo cubic capacity. The amidships portion of a vessel is the "box" part where cargo stowage and handling is much more convenient, and, therefore, less time consuming and costly than the narrow wedge shaped holds at the ends. In a cargo vessel with an engine room three-quarters aft there is little, if any, gain in *useful* cargo space because there still remains a cargo hold abaft the engine room. This "odd man out" cargo hold is an extremely awkward shape in a fine-lined cargo vessel which is not made



Waterline profile of the cargo motorship La Pradera

any better by the necessity for the continued use of a shaft tunnel. The gain in cargo cubic capacity may vary slightly depending upon the size of ship, but for the optimum general cargo vessel of between 10,000 and 11,000 tons d.w. the gain in cubic space may be about 1½ per cent to 3 per cent. Higher figures than this have occasionally been put forward and for a design with an engine room three-quarters aft, which may be expected to have the smallest gain of all, a figure of from 4 per cent to 5 per cent was claimed. Such high increases would be extremely difficult to justify on a strict *pro rata* basis of comparison, but if the actual length of the engine room was decreased by the employment of modern light weight and compact machinery such as multi-engined high-speed Diesel installation with gearing, then the increase in cargo cubic space may be as high as 15 per cent. But that is quite another matter. The most obvious weight (not to mention cost) saving is that due to the elimination of a considerable length of propeller line shafting and with it, of course, the shaft tunnel. It has already been noted that structural efficiency can be increased by a more advantageous longitudinal distribution of material and this brings with it a small decrease in hull steel weight. These factors have a beneficial influence on the ship's deadweight/displacement ratio, which means that not only has the ship gained in cargo cubic space but also in her ability to use it. A fairly high stowage factor is important when modern packaged goods tend to take up more space per ton. The idea of placing all accommodation and bridge aft seems to be increasing in popularity in ships with machinery aft, especially on the Continent and with owners building ships in Japan. Two examples of this idea may be of comparative interest. The first is a group of seven similar dry cargo steamships being built for Greek interests by Ishikawajima Heavy Industries Company, of Tokyo. All are steam turbine, machinery aft ships with double-reduction gearing and Foster Wheeler D type boilers. All accommodation and the navigating bridge is arranged aft of the axis cargo holds. The second example is of a group of dry cargo motorships being built here and on the Continent for Buries Markes, Ltd., London. The first ship of this group, *La Pradera,* is now in service. She is of the open shelter deck type and is built on the combined longitudinal and transverse system of framing. Hatch coamings run continuously fore and aft, so that they are fully integral with the main deck girders. —*J. A. Hind, Shipbuilding and Shipping Record, January 24th 1957; Vol. 89, pp. 111-113.*

Corrosion Engineering Problems in High Purity Water

One of the important corrosion problems associated with materials in contact with high purity, high temperature water is crevice corrosion, about which there is very little information in the published literature. This form of corrosion has been a problem of major concern in the design considerations and the choice of materials for water cooled reactor components. Under the conditions of testing with which the author is concerned (i.e. in high purity water at temperatures up to 650 deg. F.), the mechanism of attack has been fairly well established as an oxygen concentration cell. Crevice corrosion is undesirable because it produces relatively large amounts of corrosion product at the mouth of a crevice which, in applications involving moving parts with small clearances, may cause excessive wear or complete seizure. Figs. 2 and 3 illustrate the effect of temperature, oxygen content and clearance on crevice corrosion. Although most of the work was done on the more corrosion resistant materials, such as the 18·8 type stainless steels, the information applies to most of the materials studies. Fig. 2 shows that this form of crevice corrosion is dependent on oxygen and that the bad effects of oxygen can be tolerated if clearances are maintained in excess of 5 mils. Fig. 3 illustrates the effect of temperature on crevice corrosion. At the lower temperature there is essentially no

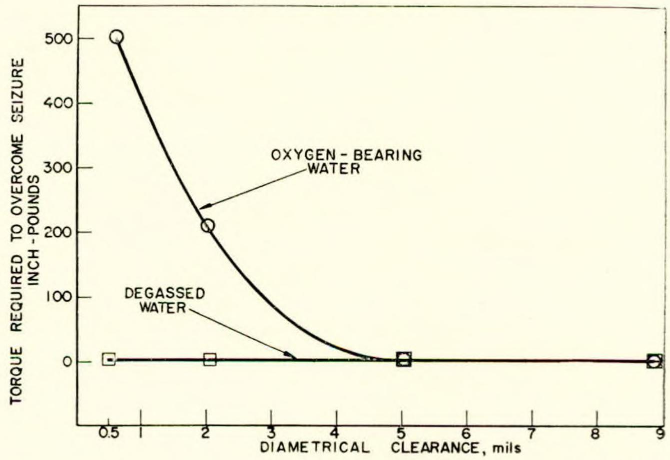


FIG. 2—*Effect of oxygen and clearance on crevice corrosion at 500 deg. F.*

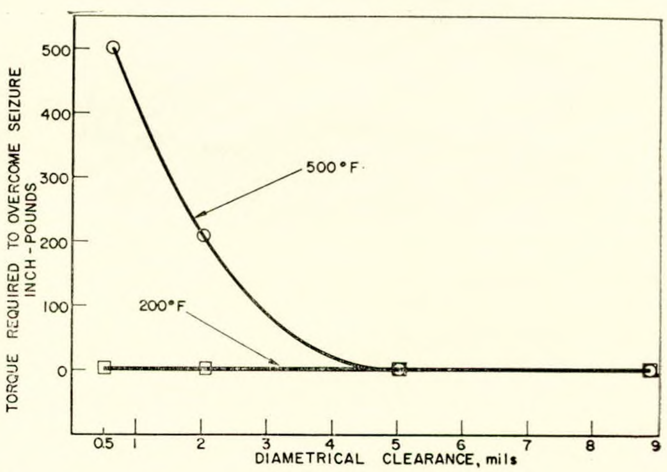


FIG. 3—*Effect of temperature on crevice corrosion in oxygen-bearing water*

problem of crevice corrosion, especially with the more corrosion resistant materials. Normally, the extent of contact corrosion found at 200 deg. F. amounts to nothing more than a discoloration of the metal without any measurable build. It can be seen that it is essential during the design stage to consider such items as materials, clearance in moving parts, extent of movement, temperature and oxygen content of the water. Permanent crevices formed by such items as threaded joints and flanged or socket weld joints are of no practical concern, since the amount of metal lost is exceedingly small and the corrosion build-up formed at the mouth of the crevice does not produce any harmful effects. Although the information presented on crevice corrosion was based on studies conducted in high purity water, there are indications that much of the data are applicable to general industrial water problems, especially in boiler plant systems. Generally speaking, dissimilar metal contact does not show evidence of preferential accelerated corrosion in high purity water. This has been found to be true even with carbon steel welded to stainless steel where there was a considerable difference in the nobility of the metals. The explanation for the lack of occurrence of galvanic corrosion is thought to be due to the high electrical resistivity of the water which serves to interrupt any galvanic currents which might normally exist at a lower resistivity level.—*D. J. De Paul, Corrosion, January 1957; Vol. 13, pp. 75t-80t.*

Lapping Machine for Smaller Valves

A new automatic lapping machine has been standardized by Hopkinsons, Ltd., for their full-bore parallel-slide valves of up to 4-in. bore. In common with the machine for larger sizes of valves (up to 12-in. bore), this equipment enables valve seats to be lapped without taking the valve out of the pipe line; similarly, lid studs in the valve body need not be removed during the lapping operation. The machine, which weighs only 8¾lb., comprises a pneumatically-driven motor operating

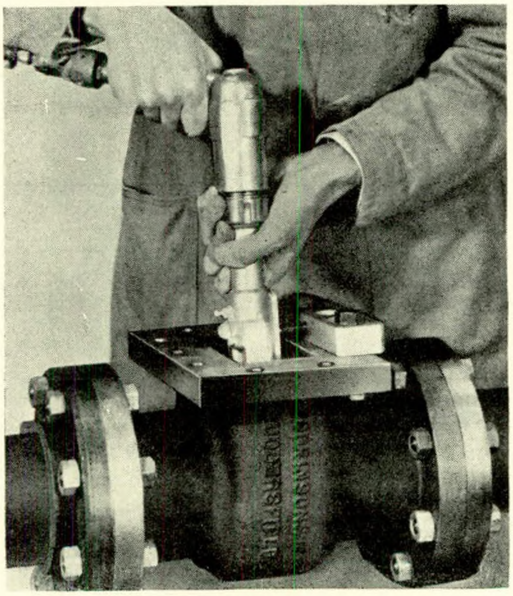


FIG. 8—*The Hopkinsons' automatic lapping machine in use on a 4-in. parallel-slide valve*

at approximately 2,500 r.p.m., reducing to a suitable speed at right angles to the main shaft, where a driving bit readily engages with a corresponding slot in a lapping plate. A range of plates is available to suit various sizes of valve. The machine operates at 80 to 100lb. per sq. in. pressure, the supply of air being controlled by a trigger in the hand grip. The air supply can be taken in any form of flexible pipe to the desired location. An automatic push grip coupling is supplied with the machine, one part being attached to the handle, the other available for fixing to the flexible pipe. Breaking the union automatically cuts off the air supply. Mechanical loading of the lapping plate is impracticable in the smaller range of valves serviced by this machine which is, therefore, supplied with an adjustable fulcrum plate; this can be bolted to the lid flange of the valve. The fulcrum plate carries a centre screw which fits into a groove in the machine and acts as a pivot by which pressure is easily applied.—*Engineering and Boiler House Review, March 1957; Vol. 72, p. 95.*

Wire Patterns for Bent Sections of Ships' Piping

In an article by J. Szarejko in "Budownictwo Okretowe" it is stated that when forming bent sections of ships' pipelines, it is usual to use wire patterns which serve as guides for the correct curvature of the piping. The patterns made for one ship may be used for subsequent ships of the same series, but this involves the inconvenience of storing the patterns, often for long periods, during which they may become deformed and therefore useless; also, storage space for the patterns is required. To overcome these disadvantages, some shipyards in the Soviet Union have adopted a method by which the wire patterns made for the prototype ship are photographed and, after a set of negatives has been obtained, discarded. Then, when a new set of patterns is needed, the negatives are projected on to a screen and the bends in the wire are made to coincide with the projected image. For this work a light-proof photographing and projecting room is needed, housing a combined plate camera and projector, a rigid metal screen with clamps for holding the wire, and good lighting equipment for illuminating the screen. A darkroom for processing the plates is also necessary. In photographing the prototype patterns and in reproducing the patterns from the projected images, a standard procedure must be followed strictly in order to avoid mistakes, particularly when the wire is to be bent in several planes. A standard method of indexing the negatives must also be adopted. The main advantage of the method is that the negatives can be preserved in a small space, for years if necessary, and can easily be sent out from one shipyard to another. This is especially useful when a series of ships is to be resumed after a lapse of time. The disadvantage is that it needs skilled labour both to record and reproduce the patterns. In general, the system is advantageous in shipyards building large numbers of a particular design of ship, and in which the facilities for storing a large number of wire patterns are restricted. The article gives a detailed description of the photographic and wire-forming procedure, and some comparisons with other methods of making the patterns are made.—*Journal, The British Shipbuilding Research Association, January 1957; Vol. 12; Abstract No. 12,454.*

Atomic Propulsion of Merchant Ships

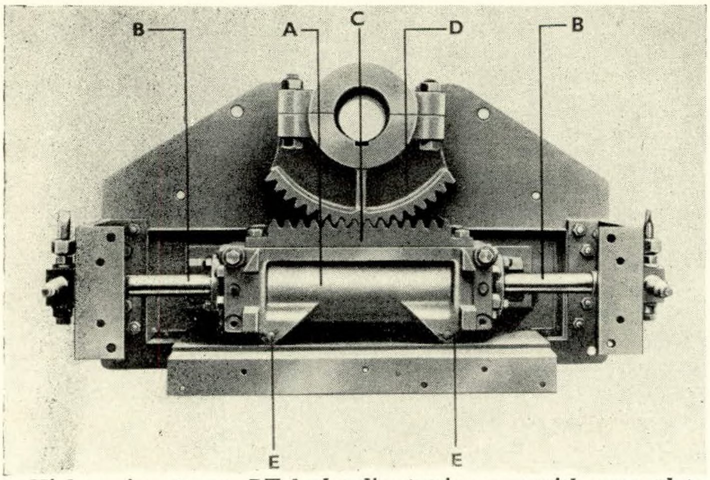
In a paper read at the Denver Conference of the Uranium and Atomic Industry, J. J. McMullen discusses the economic possibilities of applying nuclear power to merchant ships and finds that large oil or ore carriers operating on long trade routes are the ships in which nuclear propulsion has the best chance of becoming economically competitive. The present trend in shipbuilding towards very large tankers favours the development of nuclear propulsion plants. Today, the only nuclear plant that has reached a sufficient stage of development to be contemplated for merchant ship use without considerable research is the pressurized water reactor, the type successfully used on the *Nautilus;* but the present rate of progress in reactor technology suggests that more advanced and economically more suitable types will be developed. A long range appraisal of the potentialities of the nuclear systems that are being considered for merchant ship propulsion suggests the following order of preference: Closed cycle gas turbine with gas cooled reactor; organic moderated reactor; pressurized water reactor; liquid metal cooled reactor; liquid metal fuel reactor; boiling water reactor; aqueous homogeneous reactor.—*Journal, The British Shipbuilding Research Association, January 1957; Vol. 12; Abstract No. 12,442.*

Trials of Complete Marine Propulsion Turbine Installations

Trials of this nature, which may last up to three years and in the fully comprehensive sense have been confined so far to warship machinery, are described in detail. Mention is made of trials of merchant ship machinery rather more restricted in scope. The value of the comprehensive trials is emphasized, together with the planning of the programmes, operation, and some points of instrumentation. Referring to automatic control devices, the author points out that the question of automatic control is rapidly assuming large proportions, and completely new techniques are required to obtain useful test results from the various components. The necessary equipment and experience for such tests are being steadily accumulated and trials are rapidly becoming more scientific and workmanlike, but mistakes and failures are still much too frequent. Moreover, the controllers themeselves often give a disappointing performance, due sometimes to bad mechanical design, but also very frequently to incorrect specification, i.e. to the controller being of basically the wrong type for the job, although it may be perfectly satisfactory in itself. These latter cases are the result of ignorance of the dynamic characteristics of the plant to be controlled, and in this field further study is urgently required. After a few dismal failures one is apt to despair of satisfactory operation ever being achieved, and indeed in the present state of the art it is certainly a mistake to festoon a ship's installation with too many automatics. The advantages to be gained from really reliable, good performance automatic control of many systems and machines, however, are so great that a major effort to achieve this is warranted. Certain hitherto neglected characteristics of steam plant and auxiliaries will have to be investigated for future designs as a matter of routine, and here the steam plant designer must play his part. Without this information, the control system designer cannot hope to make full use of the many highly developed devices which are already his stock-in-trade and which, correctly applied, are capable of very high performance.—*Paper by H. E. C. Hims, read at a meeting of the North-East Coast Institution of Engineers and Shipbuilders on 11th January 1957.*

Vickers-Armstrongs Hand-hydraulic Steering Gears

Two simple forms of hydraulic steering gear for use in small craft such as yachts, tugs, trawlers and coasters have been introduced by Vickers-Armstrongs (Engineers), Ltd. The type "PT" (Plate Type) is suitable for rudder torques up to five tons ft. and consists of a moving cylinder (A) carrying a forged steel rack (C) and operating over fixed rams (B). The machine cut teeth engage with similar teeth on a quadrant (D) fitted to the rudder stock. Guide rollers (E) are provided on the cylinder to keep frictional losses to a minimum. The quadrant is supplied with a split boss arranged for bolting and keying to the stock and this not only facilitates the installation of the gear but also enables the quadrant to be easily disengaged should the stock require to be withdrawn for examination or repair. The power oil for the operation of the rudder stock unit is provided by the bridge telemotor which consists of a rotary-type pump operated by a teak handwheel. Connexion between these two units is by means of small bore copper piping and the telemotors are arranged so that the direction of rudder movement corresponds to the direction of rotation of the handwheel. A large, two-ram fixed cylinder type suitable for rudder torques up to eight tons ft. is also available. This is of Rapson-slide pattern employing a forked tiller instead of the tiller and trunnion arrangement of the standard opposed ram steering gear, enabling a considerable saving in space and weight to be achieved.—*The Marine Engineer and Naval Architect, January 1957; Vol. 80, p. 34.*



Vickers-Armstrongs PT hydraulic steering gear with cover plate lifted to show moving cylinder and fixed rams

Free Piston Gas Generators on Board Ships

The small weight, flexibility of installation of the gas generators, and the small space occupied by the turbo-reduction gear unit makes it possible in all cases to locate the engine room right aft in order to free completely the centre of the vessel, which is thus available for the cargo or equipment. For very specialized vessels (mail boats, ferries) the low height of the engine compartment renders it possible to provide particularly well adapted installations. As to the internal arrangement of the compartment, this is left to the free choice of the shipbuilder and shipowner. It depends principally on the type of grouping of the gas generators which it is desired to adopt and consequently on the number of turbines to be used. The water and oil circuits are similar to those for a slow speed Diesel. The compressed air circuit, on the other hand, is appreciably smaller. As with large Diesel engines, it is more rational to drive the auxiliaries by means of electric motors which consume a small percentage of the total normally provided and afford great flexibility of operation. Application to an oil tanker of 20,000 tons and 8,000 h.p. or to a cargo vessel of 7,000 tons and 7,000 h.p. yields the following arrangement: For reasons of good distribution and operation two groups of pumps are used, each supplying four gas generators and each consuming 80 kW. The installation of the engine compartment is set out in Fig. 5: eight free-piston gas generators supply a single gas manifold. From this manifold, two supply lines feed a turbine with divided admission, which is connected

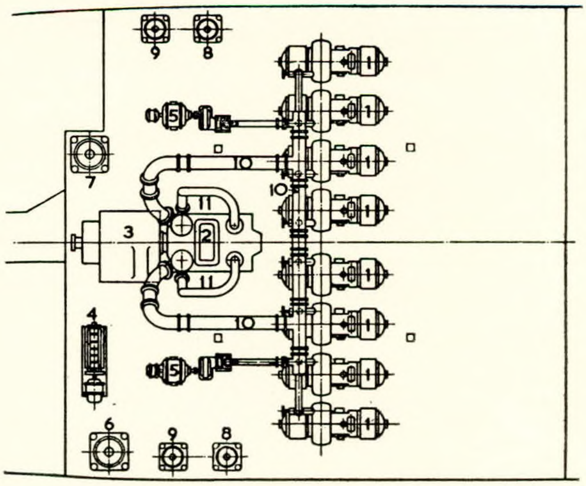


FIG. 5—*Free-piston gas generator and turbine propulsion equipment; power 8,000 h.p.*

(1) Free-piston gas generators; (2) Gas turbine; (3) Reduction gear; (4) Electrical generating set of 100 kW; (5) Turbo dynamo of 300 kW; (6) Turbine and reduction gear oil pump; (7) Turbine and reduction gear standby oil pump; (8) Duplex pump, fresh water and sea water; (9) Gas generator oil pump; (10) Gas manifold; (11) Gas pipe for astern operation

to the propeller shaft through a reduction gear. The same manifold supplies the intakes of 300-kW turbo-dynamos, two in the case of the cargo vessel and three for the tanker specified. With this installation the gas generators provide for propulsion, and for electricity generation from a single turbo-dynamo set, when at sea. In harbour, when the main turbine is not running, it is sufficient to leave one gas generator running in order to provide the power necessary for loading and unloading the cargo carried. To provide for the initial starting a 100-kW Diesel electric generating set can be provided which provides the current for one of the two groups of pumps, in addition to ship lighting in the case of prolonged immobilization. One problem still remains at present, namely the unification of the lubricating systems of the gas generators and of the turbo-reduction gear unit. In the case of the existing arrangements which employ the same quality of oil under the same temperature conditions, it would be of great advantage to be able to combine the two systems in order to reduce the number of pumps and oil coolers by half. One factor against this type of unification is the possible eventual contamination ‹ the oil by the piston cooling system. In actual fact measurement of the solid impurity content of the gas-generator piston cooling oil after 3,000 hours of operation on the coaster *Cantenac* shows that there is no danger of deterioration of the reduction gear or of the turbine by this oil. Unification on these lines could provide a saving in weight of the order of 8 to 10 tons in the case of a propulsion unit of 6,000 to 8,000 h.p.—*Paper by M. A. Augustin-Normand and M. Barthalon, read at a Meeting of The Institution of Naval Architects on 6th December 1956.*

German-built Vessel with British Engine

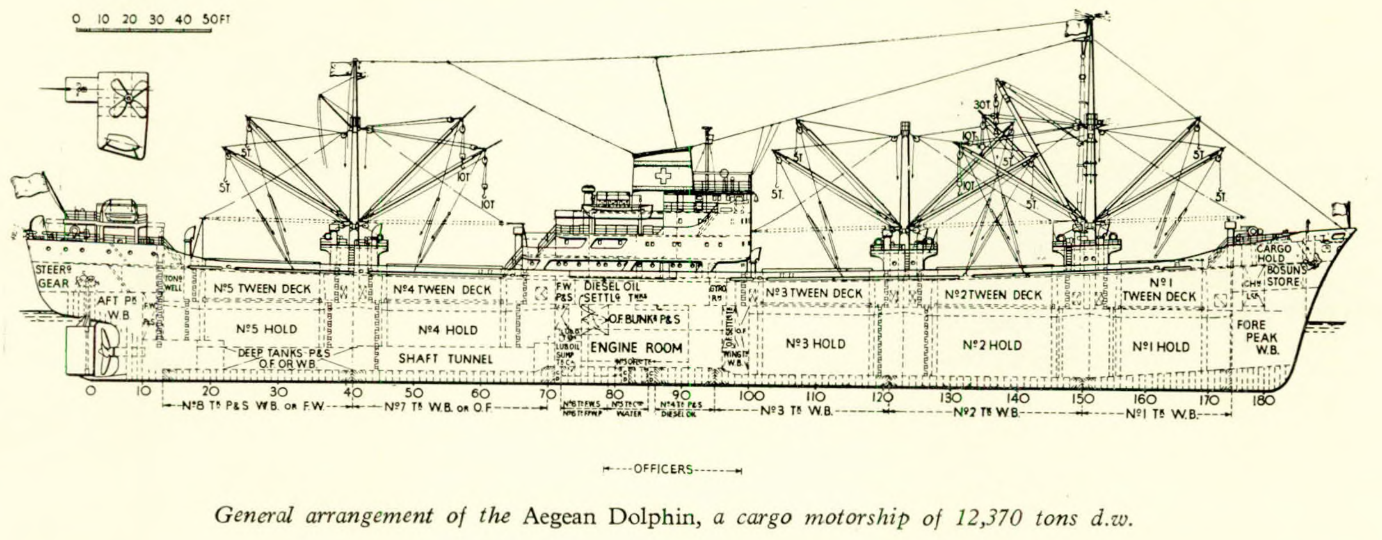
Towards the end of last year, trials were carried out at the mouth of the Elbe of the dry cargo motorship *Aegean Dolphin,* built to the order of the Pacific Trading Corporation Panama (managers, Coulouthros Limited, London) by Howaldtswerke, Hamburg, A.G. The vessel is designed for general cargo trading, and the equipment and accommodation are of a high standard. The principal particulars are:—

Length overall	490ft. 0in.
Length b.p.	450ft. 0in.
Breadth moulded...	62ft. 10in.
Depth moulded to first deck ...	38ft. 10in.
Depth moulded to second deck...	29ft. 10in.
Draught	26ft. 0in.
Gross tonnage	8,836 tons
Deadweight tonnage	12,730 tons
Cargo capacity	664,620 cu. ft
Speed on trials	17·15 knots

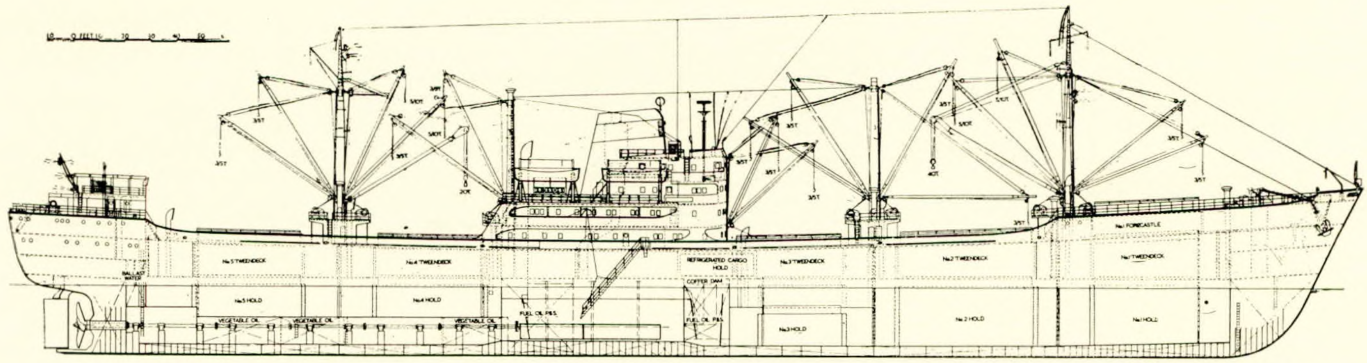
The vessel is divided into eight compartments by seven transverse bulkheads. All transverse bulkheads and longitudinal bulkheads below the second deck are of corrugated construction with the exception of the peak bulkheads. A double bottom extends from the after peak to the collision bulkhead. A considerable amount of welding was carried out in the construction of the ship. Butts and longitudinal seams are welded, while the sheer strake, bilge strake and keel strake are riveted. The freeing ports in way of the open deck consist of long narrow openings without flaps. The propelling machinery consists of a Clark-Sulzer two-stroke single acting type 8SD72 Diesel engine developing 5,600 b.h.p. at 125 r.p.m. The engine, which is designed for burning heavy fuel, has eight cylinders with a bore of 720 mm. and a piston stroke of 1,250 mm. Of the crosshead type and fitted with a lever-driven scavenging pump for each cylinder, the engine has a cast iron bedplate, the columns and cylinders being of the same material. Bearings are of steel, lined with white metal, while the cylinder covers are of cast steel. Cooling of the cylinders and fuel injection valves is by fresh water and the pistons by lubricating oil. In order to counteract the effect of sulphur when the engine is burning heavy fuel, drainage from the upper three piston scraper rings is led to a sludge tank. On shop trials, with the engine developing 5,732 b.h.p. at 124 r.p.m. and a m.e.p. of 90·9lb. per sq. in., the fuel consumption was 0·364lb./b.h.p. Figures for the sea trials are not yet available. A Michell thrust bearing is fitted and the shafting drives a 4-bladed Novastone propeller.—*Shipbuilding and Shipping Record, 17th January 1956; Vol. 89, pp. 75-78.*

German-built Cargo Liners for India

The Indian Government has the intention of ensuring that a substantial mercantile tonnage shall be built up under Indian ownership, and the Scindia Steam Navigation Co., Ltd., Bombay, in implementing this policy, has a programme of construction of ten large, fast cargo motor vessels. The *Jaladhan,* which ran trials recently, is the first of a series of six similar vessels, three ordered in March 1955, and the other three in June 1955. Four of another class have already been delivered and four more were recently ordered for completion in 1959. All of these were placed with the Lübecker Flender-Werke. The *Jaladhan* will be employed in the Scindia Company's liner service between Indian ports and Great Britain, Benelux and Germany. It is to be noted that, whilst the ship was constructed in Germany and the machinery built in that country, a substantial amount of British equipment is included. The vessel is propelled by a nine-cylinder two-stroke non-supercharged M.A.N. engine of 8,100 b.h.p. at 115 r.p.m. The cylinder diameter is 780 mm. and the piston stroke 1,400



General arrangement of the Aegean Dolphin, *a cargo motorship of 12,370 tons d.w.*



mm. The contract speed was 17·2 knots and on the trial trip, with the ship light, over 18 knots was attained. The engine runs on boiler oil. The following are the main characteristics:—

Gross register, tons	7,300
Net register, tons	4,050
Length overall	155·26 m. (509ft. 4⅝in.)
Length b.p.	143·50 m. (470ft. 10in.)
Beam, moulded	19·35 m. (63ft. 5⅞in.)
Depth to lower deck	8·88 m. (29ft. 1⅝in.)
Depth to upper deck	12·08 m. (39ft. 7⅝in.)
Draught	7·83 m. (25ft. 8⅜in.)
Corresponding d.w., tons ...	9,500
Machinery, b.h.p.	8,100
Contract speed, knots	17·2

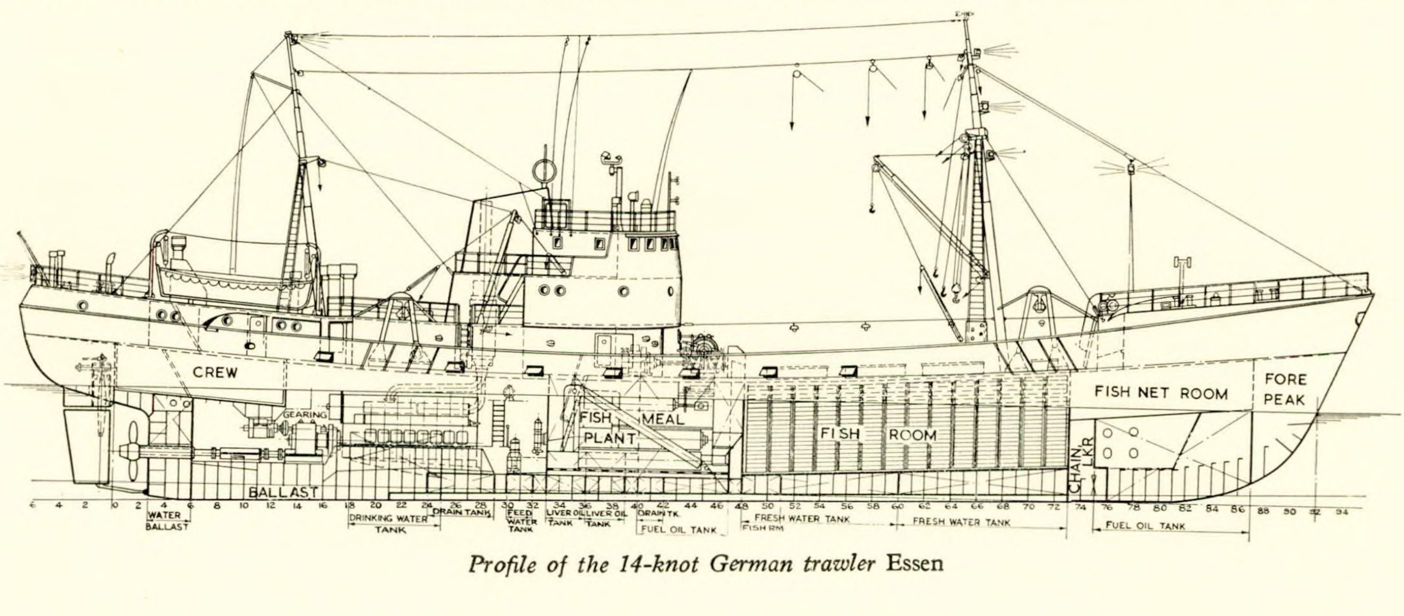
Three seven-cylinder M.A.N. 440-b.h.p. engines drive 275-kW Siemens-Schuckert generators at 375 r.p.m. and provide current throughout the ship. A 65-b.h.p. M.W.M. Diesel-engined 35-kW generator is installed for harbour purposes. Compressed air is supplied from two Hatlapa compressors with an output of 300 cu. m. per min. and each is driven by a 68·5-kW motor at 1,000 r.p.m.—*The Motor Ship, January 1957; Vol. 37, pp. 426-429.*

German Motor Trawler

Following the construction of six steam trawlers during the past two years, Rickmers Werft, Bremerhaven, received an order from the Nordsee Deutsche Hochseefischerei A.G. for two similar vessels but with Diesel engines. The first of these ships, the *Essen,* has recently been delivered and the second, named *Saarbrucken,* is under construction. Leading dimensions of the two new trawlers are as follows:—

Length overall	203ft. 8⅞in.
Length b.p.	181ft. 1¼in.
Breadth, moulded	31ft. 6in.
Depth	17ft. 0½in.
Draught (less keel)	14ft. 5¼in.
Gross register, tons ...	725
Speed, knots	14

The vessels are arranged with a fish hold capacity of 16,245 cu. ft., fish-meal space 45 tons, liver oil tanks 29 tons and fuel oil bunkers 165 tons, which is sufficient for forty days at sea at full speed. In the liver-boiling room are two boilers operating in conjunction with a vacuum system. The fish meal plant, arranged amidships, is capable of handling up to 15 tons of raw fish per day. Light alloy sheets have been used throughout the fish hold. A two-stroke, eight cylinder Henschel Diesel engine, type 8.H.2235, is installed, developing 1,050 b.h.p. at 300 r.p.m., the shaft speed being reduced by gearing to 200 r.p.m. at the Escher Wyss variable-pitch propeller, controlled from both the bridge and the engine room. Driven from the main reduction gearing is a 350-h.p. electric generator which may supply power to the ship's system (110 kVA) or the trawl winch (240 kVA) or, if required, can be used to supplement the main engine by operating as a motor. In the event of a breakdown it is possible for the main engine to be disconnected from the gearbox and for the electric motor to drive the propeller at reduced speed. This is a completely novel arrangement and presumably the dynamo, used as a motor, will be employed for propulsion when trawling. The



Profile of the 14-knot German trawler Essen auxiliary plant includes a Maybach Diesel engine driving a 350-kVA alternator and a Deutz unit coupled to an 80-kVA alternator. The whole electrical installation, with the exception of the trawl winch motor, which has Ward Leonard control, is served by 400/230-volt three-phase current.—*The Motor Ship, March 1957; Vol. 37, p. 523.*

Elimination of Plate Deformations

The elimination of ship structure deformations becomes increasingly important in the construction of steel hulls as welding and sub-assembly prefabrication come into greater use. Deformations are deflexions (bendings) in the plates and

Heat is applied in spots with a welding torch; the flames are applied parallel to the beams, at 45 degrees, and finally at an inclination that gives the proper results.

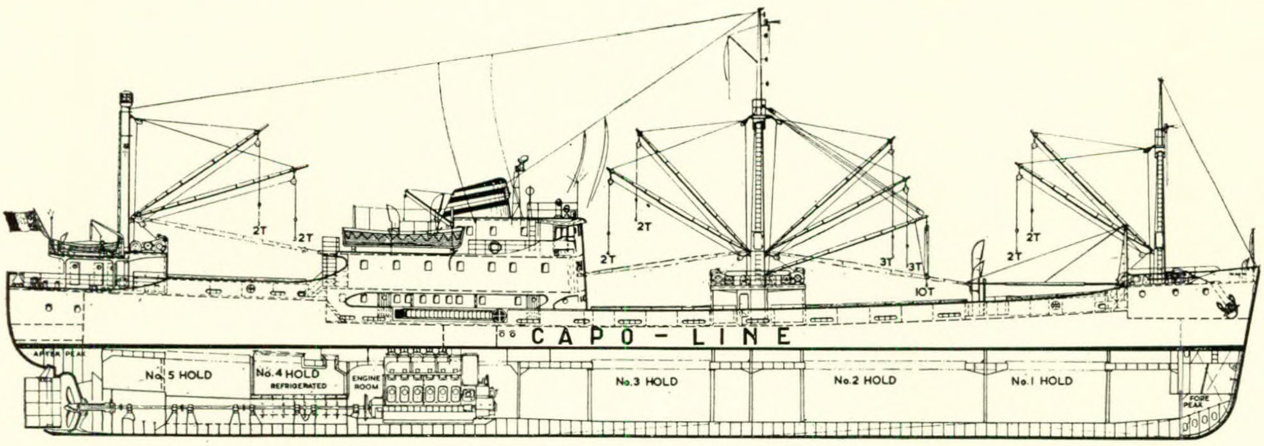
in the structural shapes of the decks or bulkheads. They are caused by shrinkage and by shrinking stresses that result from welding and from the settling of the ship on the building slip and in the water after launching. Deformations on the plates sometimes reach a deflexion of more than 20 millimeters. Circular and triangular spot heating is the system generally used in shipbuilding yards to eliminate deformations. The heated areas are then reheated and hammered so that the length of the plate fibres is reduced after the plate cools. However, spot heating causes considerable tension stresses. Although the hammering reduces local residual welding stresses, it can induce dangerous stresses in immediately adjacent areas because it does not allow for gradual expansion of the areas subject to stresses. In addition, hammering may work harden the material to a degree that may cause local failure. The Ansaldo shipyard of Genoa, Italy, has adapted and improved a method used in Sweden. The first trials of the new method at the shipyard were negative because it was necessary to find the most suitable welding torch, the correct heating temperature, and the exact areas to be heated. However, when these problems were worked out, deformations were treated successfully. The new method is now in the production stage. Hot spots are applied as illustrated in the accompanying sketch, in stretches of 80 millimeters with a welding torch having either one or two flames. Heat is applied first in a direction parallel to the position of the beams, then at 45 degrees. Finally, it is applied at an inclination that produces the desired results. The temperature of the hot spot does not go beyond 600 deg. C. The operation must be carried out simultaneously on the various areas to be straightened; results will not be evident until the heat treated plate has cooled. The new method reduces residual stresses because the temperature required is only 600 deg. C. instead of 800-850 deg. C. It eliminates hammering and reduces labour.—*Bureau of Ships Journal, December 1956; Vol. 5, pp. 39-40.*

Italian Vessel for Great Lakes Service

The m.s. *Capo Faro,* recently completed at Trieste by the Cantiere Navale Guiliano San Giusto, and delivered to the Capo Line of Palermo, is the first Italian-built vessel specifically ordered and constructed for service on the Great Lakes and has the following characteristics: —

Length o.a.	78·80 m. (258ft. 4¾in.)
Length b.p.	75·00 m. (246ft. 0in.)
Breadth	12·00 m. (39ft. 4in.)
Maximum draught	5·50 m. (18ft. 0½in.)
Corresponding d.w.c., tons ...	2,660
Great Lakes draught	14ft. 3in.
Corresponding d.w.c., tons ...	1,760
Gross register, tons	2,181
Net register, tons	1,294

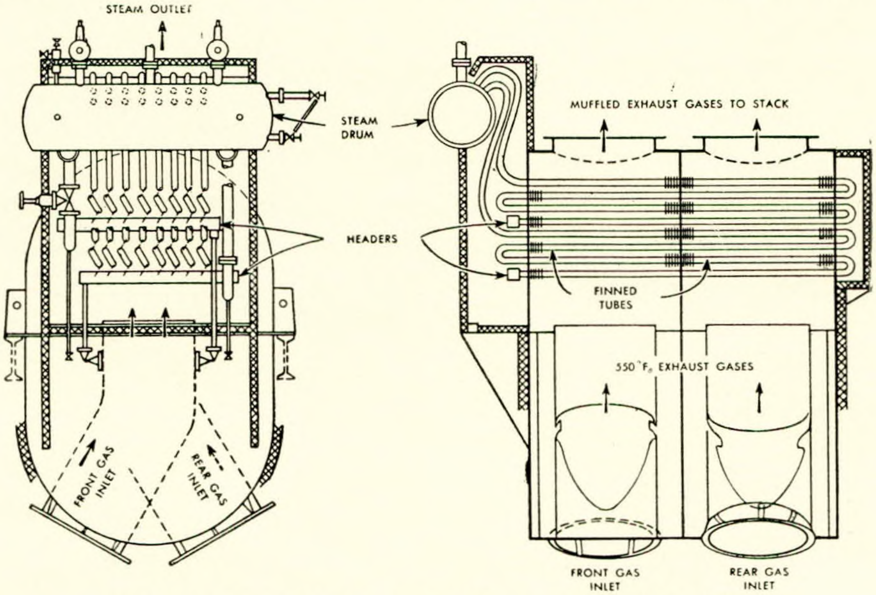
Built under the supervision of the American Bureau of Shipping, the *Capo Faro* has a cargo capacity of 137,089 cu. ft. (grain) and 130,100 cu. ft. (bale), the refrigerated spaces being 5,225 cu. ft. The Fiat main engine has an output at 250



r.p.m. of 1,800 b.h.p., this ensuring a loaded service speed of 13 knots.—*The Motor Ship, February 1957; Vol. 37, p. 489.*

Foster Wheeler Steam Generator/Exhaust Silencer

A compact waste heat boiler designed to deliver 3,000lb. of steam per hr. at 100lb. per sq. in., when supplied with exhaust gases at 550 deg. F. from two Diesel engines, has been installed in the converted Liberty ship *Thomas Nelson.* The drawing illustrates the compact, two-unit construction with separate exhaust gas sections for each engine. Silencing takes place both in the lower inlet stacks and in the acoustically baffled boiler section. Exhaust gases from each main engine pass over the boiler heating surfaces, generating steam which is collected in the drum. When steam demand falls below boiler output, the excess steam is automatically diverted to an F-W atmospheric condenser for return to the system. The boiler may also be operated at reduced capacity when only one engine is in service. If no steam is required, the boiler can be operated dry without damage to the tubes.—*The Marine Engineer and Naval Architect, February 1957; Vol. 80, p. 70.*



Sections through the Foster-Wheeler waste heat boiler/silencer

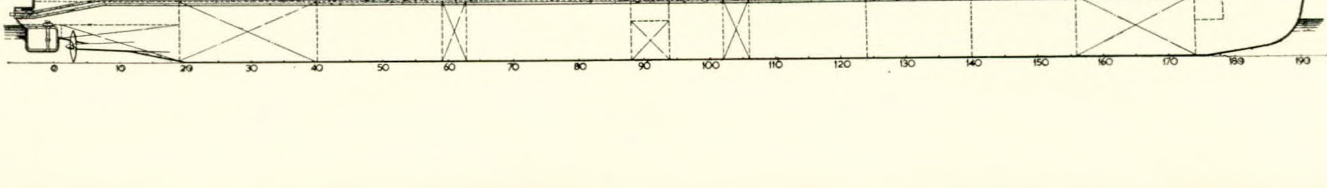
New Train Ferry

On completion of successful sea trials a new twin-screw train ferry has recently entered the Harwich-Zeebrugge service of the British Railways. This vessel, the *Essex Ferry,* has been built at the Clydebank shipyard of John Brown and Co. (Clydebank), Ltd., and was launched on 25th October 1956. A feature of the *Essex Ferry* is the electrically-operated container handling gear which permits easy handling of the containers from rail wagons on to the deck. In addition to the fore peak and aft peak, two trimming tanks are fitted, one forward and one aft. These have been arranged to give flexibility in increasing or decreasing trim so that the vessel may be quickly brought to the condition required for unloading rolling stock. The propelling machinery of the new ferry consists of two Brown-Sulzer Diesel engines, the same type of machinery as the *Norfolk Ferry* and the *Suffolk Ferry.* The propelling machinery in the *Essex Ferry* consists of two Brown-Sulzer type TS48, six-cylinder Diesel engines, each developing 1,600 h.p. at 200 r.p.m., and having a bore of 480 mm. and stroke of 700 mm. The engines are of the same output as those installed in the *Norfolk Ferry* and *Suffolk Ferry.* On the direct run from Zeebrugge to Harwich an average speed of 13 knots at about 190 r.p.m. was attained. Electricity for power and lighting at 220 volts d.c. is supplied by three Allen four-cylinder Diesel driven generators of 125-kW output. —*The Shipping World, 13th February 1957; Vol. 136, pp. 202-204.*

Cargo Vessel with Machinery and Bridge Aft

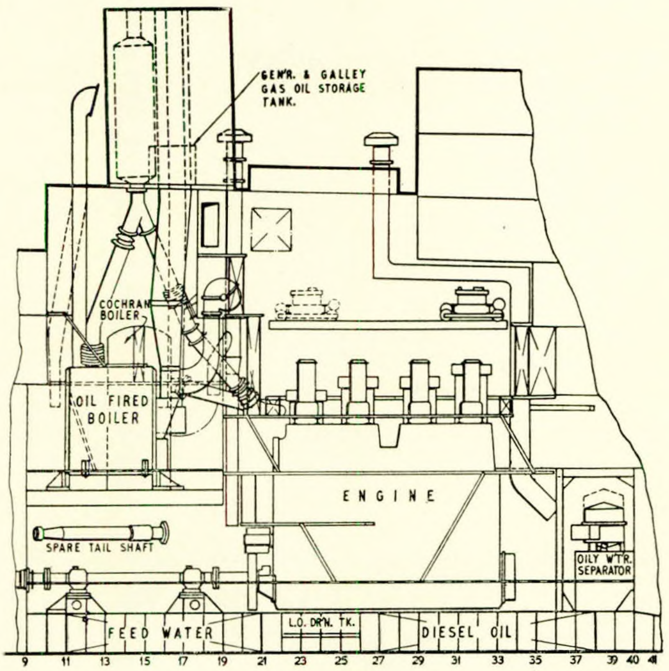
A cargo vessel with her bridge, accommodation and machinery all located aft has recently completed her sea trials. This vessel, the *Riseley,* 11,230 tons d.w., has been built by Swan, Hunter and Wigham Richardson, Ltd., for the Thomasson Shipping Co., Ltd., London. The *Riseley* is probably the largest British-built cargo vessel in service that has been built with her bridge as well as the machinery aft. The principal particulars of the *Riseley* are as follows:—

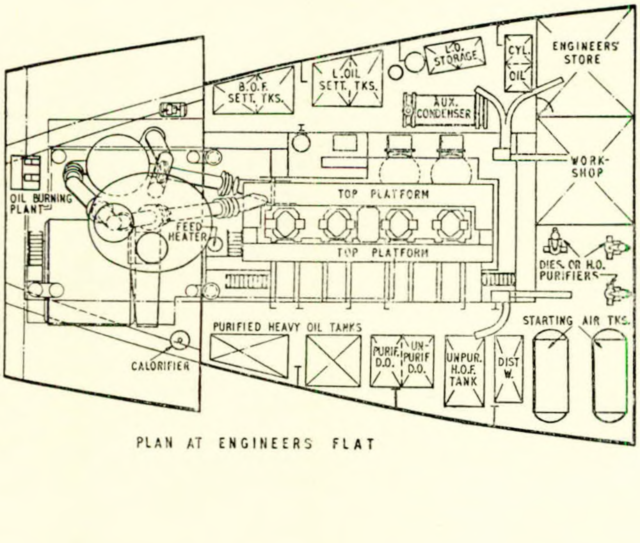
Length o.a.	470ft. 9in.
Length b.p....	440ft.
Breadth moulded	61ft.



Depth moulded to shelter deck ...	39ft. 1in.
Draught mean	26ft. 3in.
Deadweight, tons	11,230
Gross, tons	6,500
Net, tons	3,600
Machinery output, b.h.p.	3,300
Service speed, knots	12
Cargo capacity:	
(bale), cu. ft.	604,250
(grain), cu. ft....	643,450

The propelling machinery, which has been constructed at the Neptune Works, consists of a Swan Hunter-Doxford oil engine designed to develop 3,300 b.h.p. in service at about 108 r.p.m., and to operate on boiler fuel. The four cylinders are 600-mm. diameter and the combined stroke is 2,320 mm. The engine is of the latest diaphragm chamber type, designed to prevent contamination of the crankcase by the products of combustion. Scavenge air is supplied by two scavenge pumps driven by levers from Nos. 1 and 2 cylinders. The levers on No. 1 cylinder also drive lubricating oil, jacket water and sea water pumps: in fact, all the services necessary for the running of the main engine at sea. The main engine exhaust is passed through a Cochran composite boiler which provides the steam required for the remaining engine room auxiliaries and ship's services. For port and standby duties a multitubular boiler has been provided. All the standby pumps and other engine room auxiliaries are steam driven and include two 45-kW generators. For harbour duty a Ruston and Hornsby 4-cylinder type 4YEZ Diesel-driven generator has been installed.—*The Shipping World, 23rd January 1957; Vol. 136, pp. 135-138; p. 141.*





PLAN AT ENGINEERS FLAT

Highly Pressure Charged German Diesel Engine

The M.A.N. engines of the K6V45/66 m.H.A. type have a cylinder diameter of 450 mm., a stroke of 660 mm. and a service speed for ship propulsion of 250 r.p.m. In six-cylinder form and with a mean effective pressure of 227lb. per sq. in., the output is 2,800 s.h.p. The engine is built as a crosshead machine and, while it is naturally taller and heavier than one of trunk piston type, the separation of the crankcase from the cylinder liners enables heavy fuel to be burned. To date a total of eighteen engines of the type are in service. All the ships have two main engines coupled to a single propeller shaft through couplings and gears. The first vessel to enter service with the M.A.N. highly pressure-charged engine was the *Lichtenfels,* of the Hansa Line. Seven similar ships followed so that the Hansa Line has sixteen of these engines in service. The North German Lloyd has two ships, the *Torstein* and *Tannstein,* each with one of these engines and one trunk piston single-acting two-stroke of the G7Z52/70 type. The Hansa Line engines run at 250 r.p.m., whereas those in the NDL ships run at 220 r.p.m. in order to match the running speed of the older two-stroke type. By August 1956, these sixteen engines had run a total of 49,000 hours and two more engines were due to enter service in September 1956. The *Lichtenfels* has, naturally, the engines with the longest running hours, her port one having run approximately 6,350 hours and the starboard, due to having run 2,000 hours on test at Augsburg, has run about 8,350 hours. The high mean effective pressure of 227lb. per sq. in. results in a relatively small piston area compared with equivalent conventional engines. If the customary starting air pressures are employed, the starting torque of the highly pressure charged engine is naturally lower than that for a two-stroke engine of the same output and also lower than for a four-stroke engine having lower working pressures and correspondingly larger piston area. If only the engine itself had to be accelerated there would be no problem. However, since the engine is encumbered with the load of clutch, gear, shafting and propeller, which is high in the case of geared installations, and dimensioned according to output and not to piston diameter, the capacity for acceleration is less. The starting air pressure has therefore been raised from 440lb. to 590lb. per sq. in. which can be accommodated in air receivers of normal design. The engines of the *Lichtenfels* were changed over to heavy fuel after the starboard one had run for 1,150 hr. and the port one for 2,400 hr. on the ship. To date five of the Hansa Line ships (ten engines) are regularly running on this grade of fuel, the quality of which is not constant. As is well known, cylinder liner wear rises sharply with heavy oil, and this will be dealt with separately. Cylinder liner wear is combated, *inter alia,* by the lubricating oil. In the case of two-stroke engines, in conjunction with heavy oil, an emulsion lubricating oil has stood the test well, so that after a determined period this was introduced on board the *Lichtenfels* for cylinder lubrication. Since the engines are crosshead engines, the use of a separate lubricating oil for cylinder lubrication is quite possible. Whereas with heavy oil operation, in conjunction with the usual H.D. oils as cylinder lubricants, no appreciable deposits could be detected in the exhaust turbine, trouble of this nature arose when, on the fourth trip, emulsion lubricants were introduced. The back pressure gradually increased, the efficiency of the turbine decreased, and the output of the engines had to be reduced. After opening the turbine on return to Hamburg, it was found that the first stage nozzles of the gas turbine were severely choked with deposits. Analysis showed these deposits, taken from two points, to be as follows:

Na_2O	17·5%	(19·5)
NiO	2·3%	(2·5)
CuO	0·5%	(0·2)
SiO_2	0·4%	(0·3)
V_2O_5	11·4%	(15·3)
CaO	2·1%	(3·1)
C	1·7%	(3·0)
H_2O (combined)	12·7%	(2·7)
Fe_2O_3	4·3%	(2·2)
MgO	1·9%	(2·6)
SO_3	45·2%	(48·6)

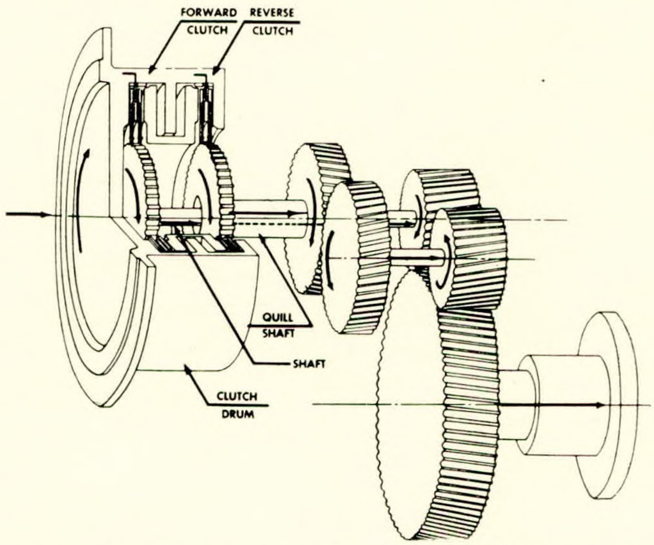
These residues are largely derived from the heavy oil and partly from the lubricating oil and it appears that the combination of the various constituents has led to still greater residue formation. Although only a part of this deposit is water soluble the whole coating could be washed off fairly easily.—*The Marine Engineer and Naval Architect, February 1957; pp. 45-49.*

Trawler *Aberdeen City*

In order that the normal design of trawler hull may be compared with the Hydroconic type, Aberdeen Motor Trawlers, Ltd., have on order two vessels with similar dimensions. The *Aberdeen City* was recently launched by Alexander Hall and Co., Ltd., Aberdeen, and the *Aberdeen Enterprise* has now taken the water at the Gateshead yard of T. Mitchison, Ltd. Traditional methods of construction have been used with the former, while the latter craft will be of Hydroconic design. The main dimensions of the trawlers are: length b.p., 113ft., breadth, moulded, 25ft., and depth, moulded, 12ft. 6in. The *Aberdeen City* has a soft nosed stem, flared bow, top gallant forecastle and cruiser stern. A total capacity of 6,700 cu. ft. is available in the fishroom, the sides of which are lined with Onazote. A 115-h.p. electric motor drives the trawl winch; the windlass and the steering gear are hand operated, the latter having hand hydraulic equipment. Right aft is a single lifeboat stowed under a Schat one-arm davit which permits launching on either side of the vessel. Accommodation is provided for a crew of sixteen men, with single cabins for the master, mate and chief engineer. All the quarters are heated throughout by radiators supplied by steam from an automatic oil-fired boiler in the engine room. A six-cylinder 760-b.h.p. (250 r.p.m.) Mirrlees Diesel engine is installed. Forward is the 95-kW winch generator, driven by a three-cylinder 183-b.h.p. Mirrlees engine. An auxiliary generator set is fitted, coupled to a 36-b.h.p. Russell Newbery engine. It is hoped to compare the performance of the two trawlers under hard weather conditions and over a number of voyages.—*The Motor Ship, April 1957; Vol. 38, p. 31.*

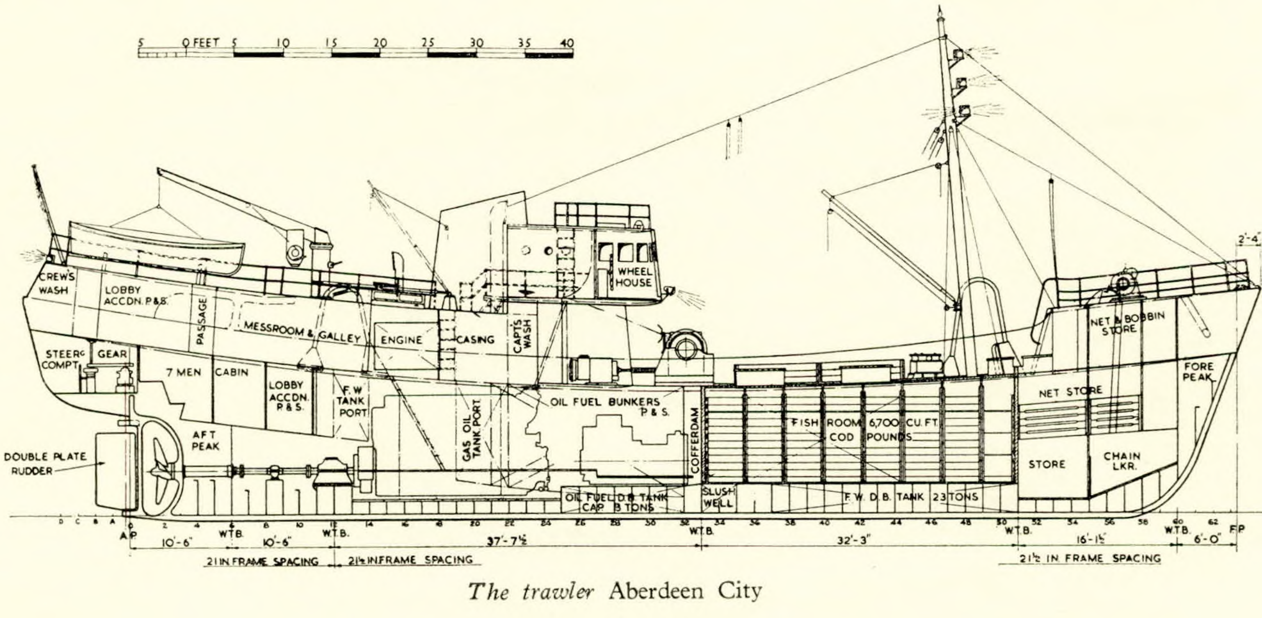
Hydraulically Operated Reverse-reduction Gear

A new line of hydraulically actuated reverse-reduction gears has been developed in the United States. The design of these gears has been developed around disc-type clutches and a simplified gear train, resulting in a light and compact unit. Anti-engine and engine rotations of the propeller shaft are obtained by two synchronized clutch units in a common clutch drum. These clutches consist of alternate sintered bronze and



Schematic of power train through hardened and ground straddle-mounted gearing

hardened-steel discs, oil cooled. When one of the clutches is engaged, the clearance of the discs in the other clutch is doubled to eliminate drag in the idling clutch. This means not only that both clutches in neutral position have equal clearance, but also that one clutch must disengage before the other clutch can be engaged. A very important feature of these transmissions is that they may be operated indefinitely



The trawler Aberdeen City

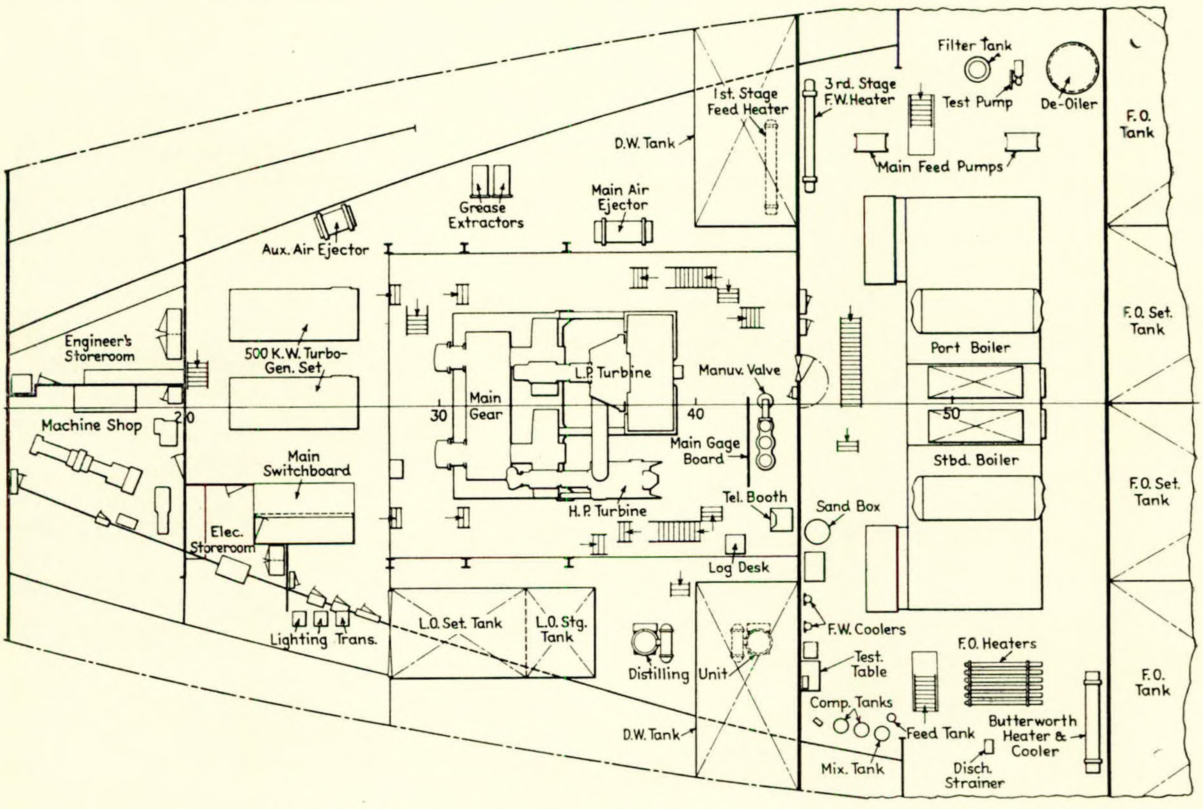
in either direction at full power and full speed. Therefore they may be used in twin installations with engines rotating in the same direction, yet having the propellers rotate in opposite directions. In other words, these gears have equal life regardless of the direction in which they operate. The pump is driven by a shaft through the centre of the gear. This shaft is connected directly to the flywheel and drives the pump through a gear on its after end. Hydraulic pressure and lubricating oil is supplied to the entire unit by this positive displacement type pump. High pressure oil is pumped through a regulating valve which controls the pressure of the oil delivered to the clutches. This pressure is maintained at approximately 125lb. per sq. in. and the bypass oil is used for lubricating the gear teeth, bearings and clutches, at approximately 5lb. per sq. in. The high pressure oil for actuating the clutches is delivered to a rotary seal insert in the pump adapter directly under the selector valve, through passages drilled in the adapter. The rotary seal insert directs the oil to the proper passage in the pump drive shaft, which in turn delivers oil to the desired clutch. An important feature of the design is that the rotary seal insert may be removed and replaced without opening the reverse gear housing. Part of the oil bypassed from the regulating valve is fed into a tube which sprays oil directly on the gear teeth. Lubricating oil also is fed into the shaft of the gear which distributes it to the bearings and clutches. The lubricating system operates whenever the engine is running. Therefore it allows indefinite idling of the clutches without generating heat.—*Marine Engineering/Log, January 1957; Vol. 62, pp. 94, 96.*

U.S. Navy T-5 Tankers

In the design of the U.S.N.S. *Maumee,* which is the first of four new T-5 type tankers, for a speed of 18 knots and 32-ft. keel draft with an appropriate deadweight/displacement ratio, a length of 590ft. evolved, with a block coefficient of 0·73. To provide a minimum length-depth ratio of 14 for longitudinal strength, as required by the regulatory bodies, a depth of 42ft. was established, and a check for cargo tank cubic space made. The cargo-cubic figure is 190,000 bbl; the vessel is able to carry four different grades of petroleum in any one cargo, including aviation gas, jet fuel, Diesel oil and bunker fuel. From a commercial point of view, the requirement for permanent bunkers to provide a steaming radius of 18,000 nautical miles at 18 knots was somewhat unusual, as it required cubic space for 4,300 tons of bunker-C fuel, making for large bunker tanks. The principal characteristics of the tanker are as follows:—

Length, overall	620ft.
Length, b.p.	590ft.
Breadth, moulded	83ft. 6in.
Depth, moulded	42ft.
Design draft, keel	32ft.
Cargo-oil capacity, 100 full bbl ...	190,300
Fuel capacity, full, tons	2,479
Reserve fuel, forward, full, tons...	1,930
Gross tonnage (Est.)	16,500
Shaft horsepower, normal ...	18,600
R.P.M. normal	101
Trial speed (80 per cent normal s.h.p.), knots	18

The cargo tank space is divided transversely into two wing tanks and a centre tank by two longitudinal oiltight bulkheads. Longitudinally there are nine subdivisions by transverse oiltight bulkheads, making a total of twenty-seven cargo oil compartments. Generally, the tanks are 36ft. long and the overall length of the cargo oil space is 342ft. The main propelling machinery, built by Westinghouse Electric Corporation, consists of cross-compound steam turbines and double-reduction gears driving a single-screw shaft. The machinery is designed to develop 18,600 normal s.h.p. and 20,460 maximum continuous at about 104 r.p.m. with normal steam condition of 585lb. per sq. in. gauge, 845 deg. F. total temperature at the throttle and vacuum at 28·5-in.Hg for normal power and 28·3in. at maximum power. There are three extraction points: one from the h.p. turbine, one from the exhaust of the h.p. turbine, and



Upper level of engine room shows ample working area one from the l.p. turbine. Steam pressure at these points gives 125, 62, and 12lb. per sq. in. abs. respectively at normal power. The turbine drives the first reduction gear through flexible couplings. The low speed pinions are driven by quill shafts from the high speed gears. Gear case and wheels are of built-up welded construction. The main thrust bearing is a Kingsbury eight-shoe self-aligning type located aft of the gear case. Power is transmitted through the thrust bearing and line shafting to a five-blade manganese bronze propeller. Each vessel has two boilers of Combustion Engineering Company make, which are air encased, two-drum type, each with an economizer, superheater, internal desuperheater supplying auxiliary steam and steam air heater. Each boiler is designed for a normal evaporation of 70,550lb.s/hr. with superheater outlet pressure of 600lb. per sq. in. gauge. Boiler water heating surface is 5,740 sq. ft., water walls 540 sq. ft., superheater 1,400 sq. ft., and economizer 4,210 sq. ft., steam air heater 2,940 sq. ft., furnace volume 1,340 cu. ft.—*F. L. Pavlik and D. Mylrea, Marine Engineering/Log, February 1957; Vol. 62, pp. 59-68.*

Austenitic Chromium-manganese Stainless Steel Weld Metal

This study covers the ductility of welded alloys in the area C 0·10-0·90, Mn 7-18, Cr 0-21, Ni 0-4; the effect of other alloy additions and combinations of additions on the strength and ductility of a 16 Cr, 1 Ni, 16 Mn analysis; and a comparison of the mechanical properties of several of the more promising alloys with conventional chromium-nickel types. The promising chromium-manganese alloys combine very high strength with reasonable ductility, and high crack resistance in a restrained joint with a completely non-magnetic structure. The results showed conclusively that properties of the weld metal could be altered and controlled over broad ranges of properties by variations in chemical composition of the deposit. Considerable additional investigation will be required and numerous questions must be answered before the full commercial and technological potentialities of this initial work can be realized. Despite the magnitude of these unanswered questions, the unusual combination of properties of these materials as they appear today suggest that chromium-manganese stainless steel weld metals may open the door to a new field of high strength fabrication. Fields of potential application for specific alloy combinations of this type include: 1. High-pressure, high-temperature pipe welding. The crack resistance of the completely austenitic alloy would be of value. The lack of ferrite as the source for subsequent formation of sigma phase offers interesting potentialities. However, high chromium-manganese alloys of this type might develop other objectionable phases at high temperature, or may develop sigma phase at a different rate than the conventional chromium-nickel alloys. This question requires further investigation. 2. For welding dissimilar metals: High strength and crack resistance of the weld deposit seem ideally suited for the high stress, hard-to-weld application. 3. For the welding of armour plate: Chromium-manganese stainless steel weld metals might provide a low nickel alternative to the nickel chromium steels. 4. For non-magnetic applications: Weld deposits of this type have proved stronger and more crack resistant than other completely non-magnetic materials frequently employed.—*W. T. Delong and H. F. Reid, The Welding Journal, January 1957; Vol. 36, pp. 41-s - 48-s.*

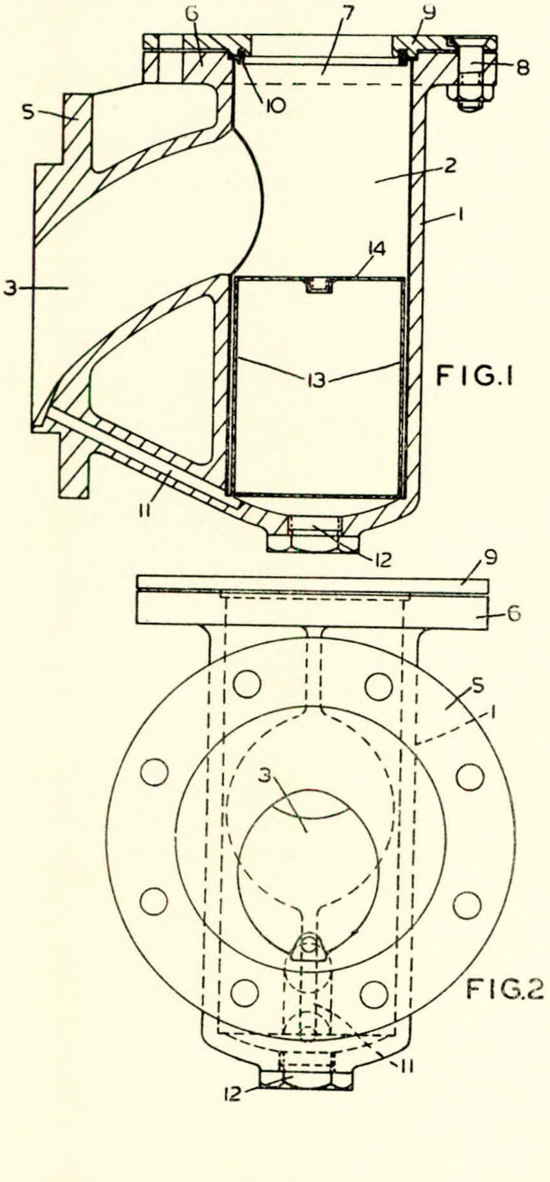
Welding in Japanese Shipbuilding

In a paper presented at the Annual Meeting of the International Institute of Welding, 1956, M. Yoshiki and H. Kihara state that, at the present time, Japanese shipyards have orders from abroad for more than 2,000,000 tons, in addition to domestic orders. With the increasing application of welding, block construction has been adopted and the size of the block has steadily increased in conformity with the size of the ship. The production procedure for the welding of ship hulls is organized in four stages, namely, fabrication, sub-assembly, assembly, and erection. The welding division comprises a welding and gas-cutting section and a research section. The research section sets out the working procedures for the type of ship under construction. Amongst the section's other duties is the collection of data, e.g., weight and gas-cutting length, covering all the four stages. The most important item in the control of man-hours is the efficiency of welders, whose man-hours amount to almost 40-50 per cent in the erection stage. It is reported that careful control of erection procedure at the Harima Shipyard reduced the man-hours per ton of erection weight by 60 per cent. In most Japanese shipyards, one or two large-sized flame planers (mostly domestic made) are being used. Non-marking cutting is largely practised, which results in a considerable saving of time. A new type of cutting machine called the Unigraph is used by the Harima Shipyard. A drawing of scale 1/10 is projected on a screen and magnified 10 times. The outline thus projected is traced by an operator, and the plate is cut to this projected line, with the cutting speed suited to the plate thickness. This machine is mostly used for double-bottom floor plates as well as for the forward and aft construction of the hull. A few shipyards are endeavouring to import Schichau-Monopol automatic flame-cutting machines. Another method of interest is the " Linear Heating Method ", patented by the Ishikawajima Shipyard, for the bending of steel plates. The plate is heated linearly by a gas-welding torch and is rapidly cooled by water immediately following the torch. By this method, man-hours for plate bending are reduced by more than 70 per cent. The principal work carried out in the sub-assembly stage is the welding of stiffening members to the steel plates prepared in the fabrication stage. To prevent any distortion caused by welding heat, elastic pre-springing is adopted in many shipyards with beehive-type surface plates. In the Kawasaki Dockyard, the use of clamping girders has reduced the labour per assembly steel weight by more than 40 per cent. At the Ishikawajima Shipyard, " cubical " welding is used in the assembly stage. For instance, three blocks of deck, side shell, and bulkhead are assembled on the ground and rotated 90 degrees to avoid vertical and overhead welding. Cutting at site for erection butts is mostly done by automatic gas cutting. Manual cutting is only used for very short line cutting. For cutting curved surfaces, a special automatic cutting machine, with a flexible rail guide bolted down along the curvature, is used. Manual arc welding in Japan is mostly done by a.c. The ilmenite type of electrode is extensively used, and has been successfully applied to all welding positions. Every shipyard is equipped with dry storage accommodation for electrodes as well as with portable dry storage boxes for low hydrogen electrodes. At present, only four to five months elapse from the laying to the launching of a 30,000 to 40,000-ton tanker.—*Journal, The British Shipbuilding Research Association, January 1957; Vol. 12, Abstract No. 12,423.*

Patent Specifications

Outboard Exhaust Valve

The invention relates to a valve construction which can be applied on ships as an outboard exhaust valve if desired and also used as a non-return valve and as a tonnage valve. In Figs. 1 and 2 the valve housing is indicated by 1. It is provided with a cylindrical chamber (2), and with a laterally connected outboard outlet conduit (3), which is preferably bent in a downward direction and which is surrounded by a flange (5), which is directly fixed on to the plating of the ship. The top flange (6) of the valve housing surrounding the inlet conduit (7) is connected to an intermediate flange (9) by means of countersunk bolts (8), the lower side of the intermediate flange being provided with an annular seat (10), e.g., a rubber ring. A conduit (11) preferably having a small cross section connects the outlet (3) with the lower part of the cylindrical chamber (2). During the movement of the valve, the throttling action in this narrow conduit will damp the movement of the valve, so that striking of the valve on the seat (10) is prevented. In the bottom of the chamber (2) a draining and inspection opening is provided, which can be closed by a screw plug (12). A closed cylindrical valve (13) is axially slidable in the cylindrical chamber (2), the valve having a buoyancy such that it will rise when the outer or sea water enters the chamber (2), so that it will first close the lateral outlet (3) and then will seal with its head surface (14) against the ring shaped seat (10). Preferably the cylindrical valve (13) is filled with a material having a small specific weight, e.g., foam rubber or the like, so that it cannot sink when damaged. By the application of a mechanism (not shown), e.g., a locking rod, passing through the inspection opening in the bottom of the cylindrical chamber and pressing the valve (13) against the seat (10), the valve can be used as a tonnage valve.—*British Patent No. 769,155 issued to N. V. Bronswerk. Complete specification published 27th February 1957.*



Remote Control of Ship with Variable Pitch Propeller

The invention provides for the remote control of a ship having propellers with adjustable pitch blades and a compression ignition motor for driving the propellers. The control arrangement comprises two control handles which can be set independently of each other, one of which serves to set the pitch angle of the propeller blades and the other to control the supply of fuel to the engine and a telethermometer which, on the basis of the exhaust gas of the motor, indicates the actual power output of the motor. The control handles are so located adjacent to each other that they can be moved simultaneously as a unit in order to adjust the pitch of the propeller blades and the supply of fuel to the engine to an approximately balanced state during quick navigation and,

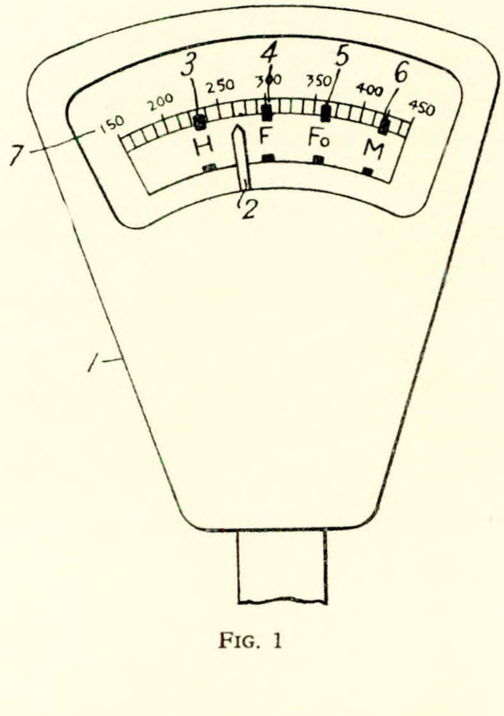
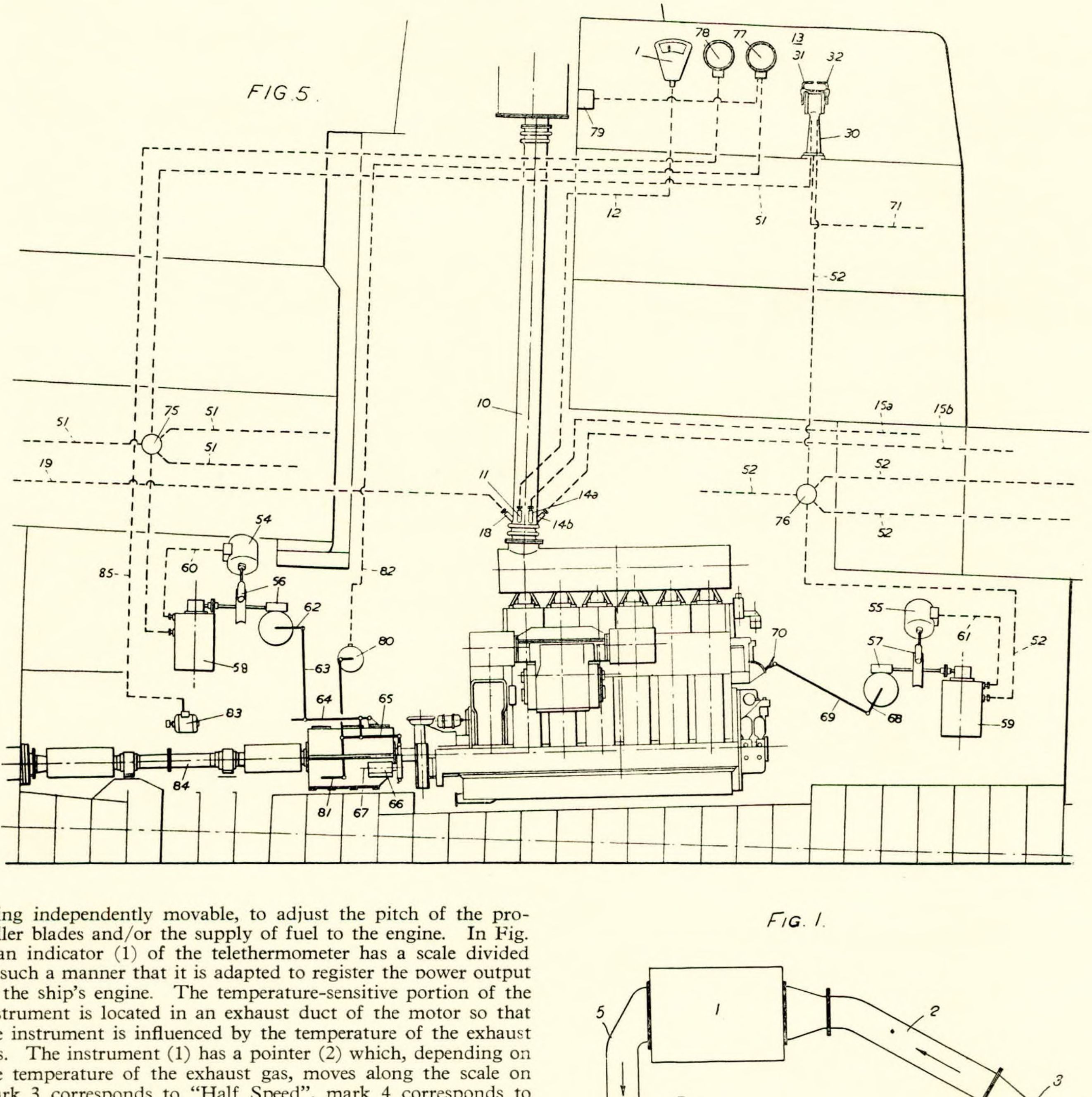


FIG. 1

FIG. 5.

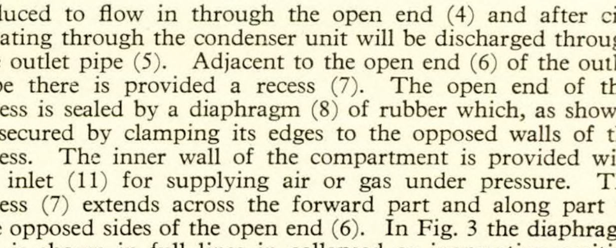


being independently movable, to adjust the pitch of the propeller blades and/or the supply of fuel to the engine. In Fig. 1 an indicator (1) of the telethermometer has a scale divided in such a manner that it is adapted to register the power output of the ship's engine. The temperature-sensitive portion of the instrument is located in an exhaust duct of the motor so that the instrument is influenced by the temperature of the exhaust gas. The instrument (1) has a pointer (2) which, depending on the temperature of the exhaust gas, moves along the scale on mark 3 corresponds to "Half Speed", mark 4 corresponds to "Full Speed", mark 5 corresponds to "Forced Speed" and mark 6 corresponds to the maximum power output. As shown in Fig. 5, such an instrument may be mounted on the bridge (13). The remote control arrangement comprises, in addition to the indicator instrument, navigation control means by which it is possible from the navigation station in use directly to adjust the rate of revolution of the Diesel engine and the pitch of the propeller blades.—*British Patent No. 768,541 issued to Aalborg Vaerft A/S. Complete specification published 20th February 1957.*

Scoop for Condenser Cooling System

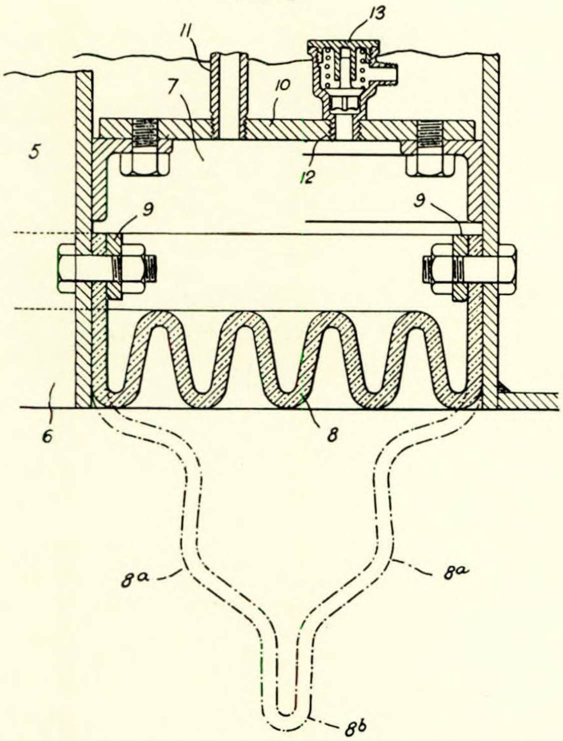
As shown in Figs. 1 and 3, a water cooling system of a condenser (1) includes a conduit (2) connected with a scoop (3) having an open end (4) on the outer surface of the hull. On the outlet side of the system there is an outlet pipe (5) having an open end also on the outer surface of the hull. In operation when the ship is moving ahead, sea water will be

FIG. 1.



induced to flow in through the open end (4) and after circulating through the condenser unit will be discharged through the outlet pipe (5). Adjacent to the open end (6) of the outlet pipe there is provided a recess (7). The open end of this recess is sealed by a diaphragm (8) of rubber which, as shown, is secured by clamping its edges to the opposed walls of the recess. The inner wall of the compartment is provided with an inlet (11) for supplying air or gas under pressure. The recess (7) extends across the forward part and along part of the opposed sides of the open end (6). In Fig. 3 the diaphragm (8) is shown in full lines in collapsed or inoperative position and it will be noted that in this position it assumes a corrugated condition extending across the open end of the compartment, the outer convolutions of the corrugations lying flush with the open end. Accordingly no part of the diaphragm

FIG. 3.



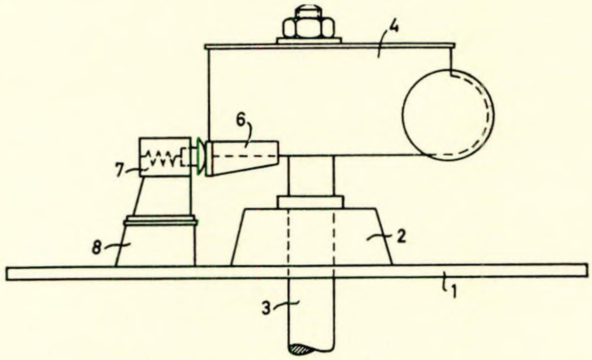


Fig. 1

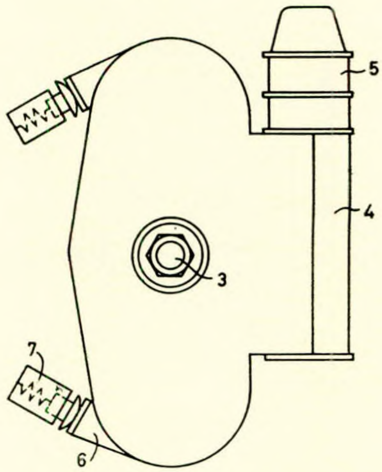


Fig. 2

extends outwardly of the hull to offer resistance to the flow of sea water past it. When the diaphragm is in this position the sea water circulating through the cooling system will be discharged through the open end (6) against the full back pressure of the sea water flowing past the open end. While the rate of flow of sea water through the system under these conditions may be adequate for normal cooling, if the temperature of the sea water is relatively high an increased rate of flow will be necessary. In this case gas or liquid under pressure is supplied to the compartment (7) through the inlet (11) and the increase in pressure in the compartment will cause the diaphragm (8) to flex outwardly into the position shown in dot-dash lines in Fig. 3 and form a projection at the forward and side parts of the open end (6) of the outlet pipe (5) and serve to divert sea water flowing past the ship's hull and thereby create a partial vacuum in the region of the open end. The pressure against which the water circulating in the cooling system has to be discharged is correspondingly reduced with the result that the rate of flow through the system will be increased.—*British Patent No. 770,781 issued to Yarrow and Co., Ltd., and Andre Rubber Co., Ltd. Complete specification published 27th March, 1957.*

Rudder Installation

Quadrant rudder installations have a number of disadvantages. Firstly, they take up a relatively large amount of space, which is undesirable. Another disadvantage is that precautions must be taken in order to allow for movements of the rudder stock along its axis. In Figs. 1 and 2, resting upon a bedplate (1) is a bearing (2) in which a rudder stock (3) is mounted. The latter has mounted on it a transmission means (4) to which is secured a driving motor (5) which is constructed as a normal flange-mounted electric motor. The driving motor (5) transmits its torque via the transmission means (4) to the rudder stock (3) and thus to the rudder (not shown). The transmission means (4) is provided with stops (6) for the substantial prevention of rotation of the transmission means. However, the stops bear against spring buffers (7) which permit some rotation of the transmission means so as to absorb any momentum of the stock. The spring buffers (7) bear against the bedplate (1) through the medium of supports (8).—*British Patent No. 769,034 issued to Siemens-Schuckertwerke A.G. Complete specification published 27th February 1957.*

Marine Engineering and Shipbuilding Abstracts

Volume XX, No. 7, July 1957

Oil Loss Past Pistons

The mechanism of oil loss past pistons was investigated in a laboratory test rig in which a piston was both reciprocated and subjected to gas pressures. The investigation brought to light the following facts and it is believed that this is the first time that they have been demonstrated: (1) Oil control is a two-way and not a one-way problem. The only reason why the rate of oil consumption is reasonably small in most engines is that, although a large amount of oil flows towards the combustion chamber past the rings during part of the engine cycle, nearly all of it returns to the sump during the remainder of the cycle. (2) The various routes permit surprisingly large rates of flow in both directions past rings. This applies even with good rings. (3) The best ring arrangements are down-passing as a whole. If oil is added to the combustion chamber, such an arrangement will pass the oil to the sump at a rate up to many thousand times larger than the normal rate of oil loss with the same arrangement. However, if just the lower portion of a ring arrangement is down-passing and the upper portion is not, it is still possible for the oil consumption to be satisfactory, provided that the lower portion, besides being down-passing, only allows a very small amount of oil to leak upward and reach the upper portion during any part of the cycle. The results are always bad if the lower portion is up-passing. Even if the remainder of the arrangement would normally be down-passing, an oil pressure is built up below this portion which forces the oil upward. (4) A taper-faced ring (fitted the normal side uppermost) is inherently down-passing. A stepped ring, from which the inner upper or the lower outer corner has been removed, is inherently down-passing unless the shape of the ring periphery has been unduly changed through wear, and unless the pressure drop across the ring is sufficient to remove the twist which gives the ring its down-passing characteristic. A truly square-faced ring has a symmetrical characteristic but, when combined with a scraper ring and its associated drain holes, it forms a down-passing arrangement. The performance of a square-faced ring is particularly sensitive to a minute twist, so it is essential that such a ring should not be twisted adversely by an inclination of the side wall of its groove. (5) With a down-passing ring arrangement, the rate of oil loss is absolutely zero if the oil drops, thrown into the combustion chamber, are allowed to run back on to the cylinder wall instead of being caught and led away. (6) The lands of a piston can have a beneficial effect, the top land usually having the most effect. If some oil has passed the rings, it may be thrown into the combustion chamber and be lost; alternatively an effective top land may retain this oil until it can be returned to the sump by a down-passing ring arrangement. (7) If the top edge of the piston can touch the cylinder wall, it may scatter any oil on the wall. It is particularly liable to scatter the trail of oil left behind by the ring gap, therefore the rotation of the top ring can affect the oil loss by altering the position of this trail of oil relative to that part of the top edge of the piston which is pressed against the wall by the side thrust of the connecting rod. The scattering can be avoided by tapering the top portion of the top land of the piston; note that the inclination of the tapered portion must be small. (8) Oil travels along the same routes as, and in company with, the gas leakage. In the test rig, the ring gap is a major route. It must be emphasized that the oil consumption in an engine will not necessarily be greater because of the existence of any particular route permitting upward flow past a ring; the route may even be beneficial. For example, the effect of oil leaking downward through the ring gap when the cylinder pressure is positive and high, may more than balance the corresponding effect of upward leakage when the cylinder pressure is negative during another part of the cycle. (9) Oil loss can be caused by oil issuing on to the cylinder wall from the clearance space between the upper side of the ring and the corresponding side of the groove. The closer the ring is to the top of its travel when the oil issues in this fashion, the less likely the ring is to be able to return the oil to the sump. The rate of oil loss will therefore depend on the timing of the axial movement of the ring, and this is a critical function of crankshaft speed and cylinder pressure. The rate of oil loss also depends on the size of the ring side-clearance and on the size of the back-clearance, but the exact manner in which these sizes change the rate of oil loss is not yet clear. The effect of both is complex and acts in a different direction under different operating conditions. (10) On its downward stroke, a square faced top ring may leave an appreciable amount of oil on the cylinder wall. On the next upward stroke, it is desirable that this oil should return past the ring by the various routes. However, these routes have sharp entrances, so some of the oil may be scattered and lost before it has another chance of entering the return routes.—*Paper by P. de K. Dykes, read at a meeting of The Institution of Mechanical Engineers on 25th January 1957.*

Dutch-built Dredger for Chile

The *Ingeniero Ruben Davila,* a self-propelled bucket dredge with shore delivery installation, has recently been delivered by I.H.C. Holland to the Direccion de Puertos, Ministerio de Obras Publicas, in Santiago de Chile. The *Davila* is a seagoing vessel with a rather normal ship's hull, with the exception of the ladder well in the bow. Her main dimensions are 190ft. 4in. x 36ft. 9in. x 14ft. 5in. (58 x 11·20 x 4·40 m.), the draught is 9½ft., and she has been built to Lloyd's Register of Shipping 100 A1. The engine installation apart from auxiliaries and accessories consists of two steam engines, both of the totally enclosed Brouwer type, built and installed by I.H.C. Holland. One engine of 1,000 h.p. serves for propulsion, the other has 250 h.p. and drives the bucket chain. Steam is raised in two marine Scotch boilers also made in I.H.C. Holland engineering works. They are suitable for oil- as well as coal-firing and work with forced draught. Electricity for power and lighting is supplied by a steam driven generator whilst a Diesel generator set is fitted for port and emergency use when steam is not available. All deck equipment is steam driven. The dredging installation consists of a bucket chain of 500-litre capacity and drop chutes in the usual way for loading barges alongside the dredger. Half of the number of buckets are provided with teeth, all have chrome nickel steel edges, for work in hard and heavy loam. The normal dredging depth is 12 metres, the maximum is 14 metres. There is also a dredge pump for delivering the spoil —then being dumped and diluted with water in a special hopper—ashore over a distance of about 700 yards. The installation has a calculated dredging capacity of 450 cubic metres per hour. The propulsion engine serves for driving the sand pump.—*Holland Shipbuilding, December 1956; Vol. 5, p. 20b.*

Reversible Propellers for Ships

It was far more difficult for the Diesel engine to oust the steam engine as propulsion unit in tugboats, trawlers, lighters, etc., than in ships such as coasters, which operate under entirely different conditions. The steam engine is much better suited to withstand overloading, thus providing a means to apply a variable torque to the propeller in accordance with the requirements of speed, tractive force and water flow. In the case of the former types of craft referred to above the constantly varying torque required for the propeller must be furnished by the Diesel engine, which at variable speed and optimum fuel injection develops a practically constant torque. The application of a variable pitch propeller makes it possible to operate the Diesel engine more efficiently as a propulsion unit. There are three categories of craft which come into consideration for propulsion by reversible propellers, viz.: 1. *Trawlers.* These must be capable of proceeding to the fishing grounds at high speed, towing the trawl with great tractive force and returning to their home port at high speed to ensure that the catch keeps fresh and can be sold at the auction at the most opportune moment. 2. *Tugboats.* It is essential that tugboats should be capable of developing great tractive force when towing. Further, they must be able to take up the slack in the hawsers gently and keep them taut at all times. Yet they must be able to develop a high speed when not towing so that especially at sea they can render assistance quickly. 3. *Ships which require a lot of manœuvring.* These include icebreakers and whalers, destroyers, channel steamers, river craft, ferries, etc. The reversible propeller offers many advantages for all these ships because most of the time they operate under constantly varying conditions of speed and propulsion power. Moreover, both with these vessels and other ships, the application of a reversible propeller can provide ample compensation for:—varying sea conditions, wind current and depth ;—fouling of the ship's bottom;—changes in the block coefficient of the ship owing to variations in draught. Apart from being able to adapt a propeller couple varying between wide limits to the maximum torque of the engine, the reversible propeller has a number of other advantages over the fixed propeller. The direction and the speed of the ship can be controlled from the bridge without the services of the engine room staff being required. Consequently, the time required for changing from "full speed ahead" to "full speed astern" (controlled from the bridge) can be reduced to from 8 to 12 seconds, which is very important when trying to avoid a collision. The reversing takes place gradually without shocks. Considerable forces are actually involved, but these build up slowly. When changing from "full speed ahead" to "full speed astern", the procedure while adjusting the propeller is as follows: During the first few seconds the ship is slowed down by the water, then the propeller assists in the slowing down process and after four or five seconds the propeller will "take", as by then the propeller blades have moved through the zero position. The Diesel engine can be run unchangingly at a constant speed, while the position of the propeller can be adapted to any prevailing conditions. The engine need not be reversible, because the reversing action is effected by changing the position of the propeller blades. Hence, manœuvring with the engine is no longer required, while the constant changing of engine speed is avoided. This reduces engine wear and thus gives the engine a longer life. Part of the fairly high investment for the reversible propeller is made good by a cheaper engine (not reversible). As shafting and propeller are given such dimensions that the normal speed of the engine is free of severe torsional vibrations and as the engine is to be run constantly at the normal speed over the whole of the propulsion power range, possible torsional critical vibrations speeds below normal r.p.m. of the engine are avoided.—*T. J. Hiemstra, VMF Review, December 1956; Vol. 1, pp. 113-116.*

New Vessel for Arctic Operation

The large fleet of Polar ships owned by the J. Lauritzen Lines was recently increased when the company took delivery of the m.s. *Thora Dan,* a 4,600-ton deadweight vessel built by Stülcken Werft, Western Germany. In common with the other Lauritzen Polar ships the strengthened hull of the *Thora Dan* is painted red and it is intended that she will operate in the ice-ridden waters off East Greenland, Canada and Finland. Service equipment includes an ice cutter, ice protection fins near the propeller and complete navigational equipment in the crow's nest which allows the vessel to be manœuvred from both there and the bridge. The ship's length is 98·4 m. (323ft.), the breadth 15·8 m. (51ft. 10in.) and the depth 7·32 m. (24ft.). Propulsion is by a Burmeister and Wain turbocharged Diesel engine having seven cylinders with a bore of 500 mm. and a stroke of 1,500 mm. It develops 4,500 i.h.p., at which power the ship's speed is approximately 14½ knots.—*The Motor Ship, February 1957; Vol. 37, p. 480.*

A Study of Piston Ring Lubrication

This paper describes an experimental study of the lubrication between a one-ring piston assembly and a cylinder. Instantaneous friction forces of the piston assembly were recorded by means of a special apparatus with a stationary piston and a reciprocating liner. Tests were carried out to determine the effects of viscosity, speed, and pressure on the friction, from which the oil film thickness under the ring was calculated. A suggested lubrication theory of piston rings is given, based on a balance of forces acting radially on the ring, leading to a theoretical formula for calculating the ring film thickness, which is in reasonably good agreement with the experimental results. Suggestions are made for the design of rings to give improved friction and heat transfer and to reduce cylinder wear, especially in the initial running-in period. It is suggested that the adoption of a parabolic profile in the first instance could provide a means for improving piston ring lubrication, reducing the amount of boundary lubrication and thereby cylinder wear, especially during the running-in period. Ring width and radius of curvature can be determined for known values of viscosity, velocity, and pressure to provide optimum running conditions. Since it is known that a large proportion of the heat transfer from the piston passes through the top rings, it would seem that the introduction of wide rings, as suggested by the lubrication theory, would greatly improve piston cooling, and it can be shown that ring friction would not increase linearly with b (calculations for the ring in the experimental apparatus revealed that if its axial width had been doubled, its friction would have increased only by about 20 per cent). With more efficient top rings it may be found that the number of rings per piston assembly could be diminished, thereby reducing total piston friction.—*Paper by S. Eilon and O. A. Saunders, read at a meeting of The Institution of Mechanical Engineers on 25th January 1957.*

Fundamentals of Ship Propulsion

The various propelling mechanisms may be divided into three principal groups: (a) The so-called jet propeller which imparts to the water flowing from ahead an impulse directed aft. A propeller whose construction is based on this principle is the Hotchkiss internal-cone propeller (Fig. 4). This propeller has the advantage of having practically no parts projecting from the ship's hull and the disadvantage of low efficiency. (b) Propellers which derive their thrust in the direction of the ship's course principally from the resisting forces on their moving parts. Among these propellers are the paddle wheels rotating around a horizontal shaft. Two types of paddle wheels are used in practice, viz., those having fixed and those having movable blades. The former type has the advantage of simplicity, solid construction, light weight and low cost of maintenance. Its main disadvantage is that for high efficiency the wheels require a large diameter and therefore must necessarily have a low number of revolutions. This results in general in having to use heavy, low speed engines. The diameter depends on the angle between the blades and their resultant velocity in respect to the water at their entrance into and exit out of the water. Attempts to improve the design of paddle wheels have led to a design with adjustable blades.

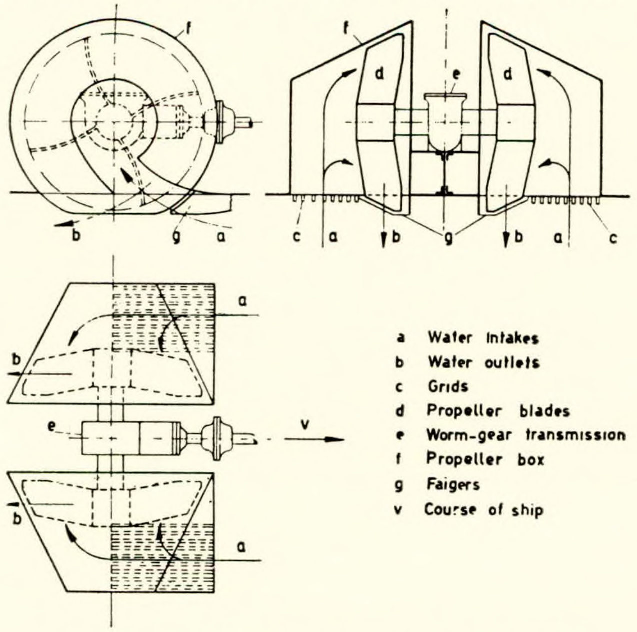


FIG. 4—*The Hotchkiss internal-cone propeller*

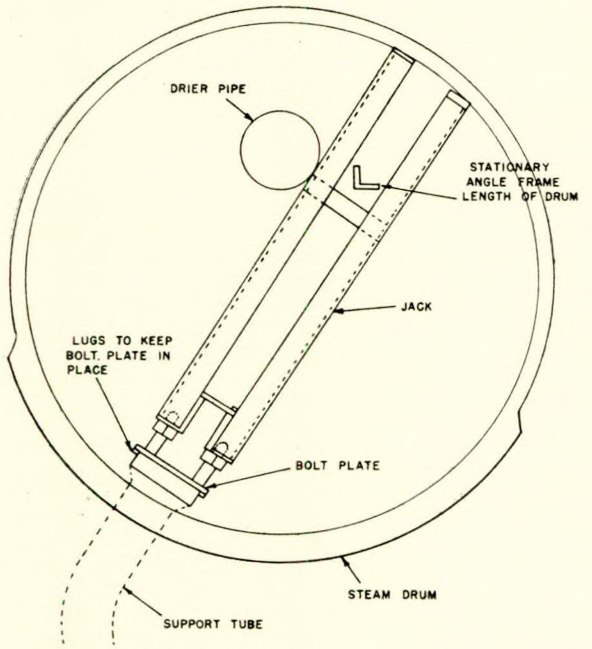
This permits the angles of entrance and exit to be reduced by means of a link mechanism. The entrance and exit of the blades can now take place gradually and the inevitable energy losses are reduced to a minimum. The revolutions per minute of this type of wheel can be increased above those of wheels having fixed blades. Disadvantages of the adjustable blade paddle wheel are its heavier weight and greater vulnerability. The efficiency of wheels with fixed blades may range from 50 to 60 per cent if the ship is without tow; that of wheels with adjustable blades is considerably higher and sometimes exceeds that of the screw propeller. (c) Propellers which derive their thrust in the direction of the ship's run principally from the lifting forces on their moving parts. To this group belongs the most important type of propellers, viz., the screw propeller. The screw propeller is usually constructed as a pushing screw and fitted as low as possible in way of the stern. In the case of seagoing vessels the screw should have a diameter such that when the ship is in her fully loaded condition the propeller is sufficiently submerged so that the phenomena of air drawing and racing of the propeller during pitching is avoided as much as possible. A rough rule of thumb which may be used in the preliminary design stage of single-screw ships is to set the screw diameter equal to 0·7 the draught. In certain types of river and canal vessels it may be that because of design requirements the screw diameter is too large for the propeller to be fitted entirely below the water level. In such cases the propeller is enclosed in a tunnel. This tunnel is constructed by extending the aft part of the hull so as to bulge over the entire screw or by fitting a counter plate in way of the propeller. The choice of the number of screws for a ship depends on many factors, such as the power, dimensions and type of engines and the ship's draught. Ships are generally built as single-screw or as twin-screw vessels, but triple- or quadruple-screws are sometimes used. Ferry boats are often propelled by a tractor screw at the bow and a pushing screw at the stern. When the ship's course is reversed, the screws change functions. The fore and after parts of such a ship are generally designed symmetrically. Ships which must have a combination of satisfactory towing performance and high free running speed (seagoing tugs, trawlers, etc.) are often fitted with adjustable-blade propellers. A conventional propeller fitted in a nozzle is very useful for two boats of restricted draught. The tandem propeller consisting of two propellers mounted on a single shaft with the propellers rotating either in the same or in opposite directions is a type which is only employed for very special purposes. Finally the Voith-Schneider propeller is a propelling mechanism of an entirely different construction designed in accordance with the aerofoil principle (6). In contrast to the screw propeller, the Voith-Schneider propeller must be classed as a non-stationary propeller. It is the only propeller used in engineering whose design has successfully incorporated a system of propulsion adapted from the natural life of birds and fishes based on the non-stationary aerofoil principle.—*J. D. Van Manen, International Shipbuilding Progress, February 1957; Vol. 4, pp. 107-124.*

Investigation of Slamming Pressures

This report presents the results of a theoretical investigation to determine the pressure distribution over the bottom of a slamming ship at zero forward speed when the impact with the still water surface results from a simple rotation in the plane of symmetry. The pressure mappings presented resulted from a study on five members of TMB Series 60, differing in block coefficient. A systematic series was used in order that some information concerning pressure dependence on hull shape might be obtained. The pressures were derived on the assumption that the vessels were all subject to the same rigid body velocities and accelerations. These pressures are presented in the form of plots showing the pressure distribution at various times after impact over the transverse sections of the forward 25 per cent of the ships' lengths. Included also are plots showing the maximum or peak pressures developed, along with their time and space variation. The analysis indicates that the magnitude of the maximum pressures decreases and the region experiencing the highest pressures shifts aft with decreasing block coefficient (or increasing fineness). In particular, for a block coefficient of 0·80, the highest pressures were found to occur at approximately 10 per cent of the ship's length aft of the forward perpendicular; for a block coefficient of 0·70, the region of highest pressures occurred at 22·5 per cent of the ship's length. The magnitude of the maximum pressures experienced a 10 per cent decrease for this change in block coefficient.—*Margaret D. Bledsoe, David W. Taylor Model Basin, Report 1043, December 1956.*

Jack for Hydrostatic Boiler Tests

An improved jack for making hydrostatic tests on individual boiler tubes has been devised at the Pearl Harbour Naval Shipyard. Formerly, a 10-ton screw jack was used to hold the cap plug down tight against the support tube opening when a hydrostatic test was made. The screw jack had to be set several times before pressure could be applied, because of its offset block base. The offset was caused by an angle frame and dry pipe standing in the line of the screw jack. Two men were needed to set the jack. To make the test, a pressure of 750lb. per sq. in. was applied to the support tubes. The operation was dangerous, because the screw jack sometimes slipped and became cocked from the pressure. The screw jack often jerked out, causing the jack and blocks to scatter with force. The new jack is much safer. In addition, it reduces testing time from 3 man-days to 2 man-hours for each boiler. The jack is made of ½ by ½in. angle iron; it backs straight against the boiler and passes the dry pipe and angle frame. It is tightened by two bolts with nuts welded to a ½-in. plate that is placed on the cap plug. To set it, only one man is required, and it needs to be set only once.—*Bureau of Ships Journal, March 1957; Vol. 5, p. 42.*



Hydrostatic tests on boiler tubes are made with this improved jack that backs straight against the boiler and passes the dry pipe and angle frame

Tests with Models of Fast Cargo Vessels

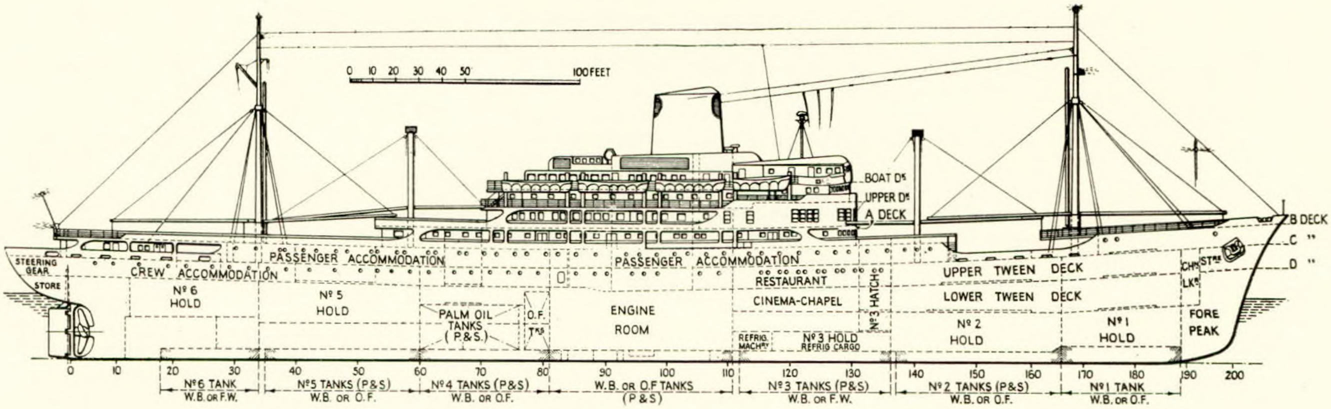
During recent years, some systematic experiments with models of fast cargo ships have been carried out at the Swedish State Shipbuilding Experimental Tank. These investigations were concerned in the first instance with the effects on the resistance and propulsion qualities of variations in the proportions and in the longitudinal position of the centre of buoyancy. The purpose of this series of experiments was to study the respective influences of variations in the length, the longitudinal position of the centre of buoyancy and the breadth-draught ratio. Resistance tests were carried out with all ten models, but self-propulsion tests were confined to those models in the L.C.B. variation group. The paper describes the results of resistance and self-propulsion tests on models with block coefficients of 0·525. This investigation, which was begun on the initiative of H. F. Nordström, may be regarded as a continuation of the earlier experiments on models with block coefficients of 0·625 and 0·575. In order that the experimental results could be related to those obtained previously, the principal dimensions and shape of sections, etc., for the models in question were chosen so as to conform as far as possible with the models described in the aforementioned publications. The low block coefficient 0·525, in combination with the normal cargo ship proportions, results in a very extreme form. These models may therefore perhaps be regarded as an extreme and rather unrealistic outer limit for an investigation which also includes block coefficients of 0·625 and 0·575. But these forms and their experimental results could be of practical significance in any development towards cargo ships with higher speeds than those prevalent today. Ten models in all have been tested. As in the case of the earlier investigations the main object was to study the separate influences of length, breadth-draught ratio and longitudinal position of centre of buoyancy on the resistance qualities. The influence of the L.C.B. position on the propulsive qualities was also investigated. Some secondary investigations were carried out side by side with the main experiments. These were concerned with the effects of adding a bulbous bow on one model and of varying the midship section coefficient of another. —*H. Edstrand and H. Lindgren, Publication of the Swedish State Shipbuilding Experimental Tank, No. 38, 1956.*

Effect of Fouling of Ship and Propeller upon Propulsive Performance

A series of systematic full scale and model tests has been carried out with the object of studying the influence of fouling on ship performance. The report describes these tests in three main parts, which discuss the effect of fouling of the hull only (i.e., with the propeller clean), the effect of fouling of the propeller only, and the effect of fouling of both propeller and hull. Each part gives detailed results of both the full scale and model tests, and discusses them at some length. One appendix to the report describes some wind tunnel tests carried out on the model of the ship, and a second appendix gives some particulars of the type of marine growths found on the ship. A small training vessel named the *Yaroi Maru* was used for this investigation. The instruments used on board the vessel included two types of speed meters, pitot tubes for measuring the wake, a spring-type tensionmeter attached to the end of the towing rope, a Michell thrustmeter, and an optical torsionmeter. In the full scale resistance tests, the ship was towed with the propeller removed, by a tug with a towline about 460ft. long, and in the self-propulsion tests investigating the effect of a fouled propeller and a clean hull, the propeller was artificially roughened by cementing to it rubber sheets with knurled or slotted surfaces. Comparative model tests were also made in a towing tank. In general, the tests show that the wake speed, resistance, propeller thrust, delivered horsepower, and r.p.m. are considerably increased by fouling. On the other hand, the thrust deduction coefficient is only slightly influenced. Roughening the propeller leads to a considerable decrease in propeller efficiency.—*Shipbuilding Research Association of Japan; Report No. 11, 1956. Journal, The British Shipbuilding Research Association, December 1956; Vol. 11, Abstract No. 12,260.*

Turbine-driven Vessels for Belgian Owners

Among the routes covered by the Compagnie Maritime Belge is a weekly service between Antwerp and Matadi, in the Belgian Congo, via Lobito (Angola). In order to maintain this service seven ships are required and until recently the fleet



General arrangement of the single screw, turbine driven, passenger and cargo liner Jadotville

consisted of the five vessels *Albertville, Elizabethville, Leopoldville, Baudouinville,* and *Charlesville,* augmented by two older vessels of another type. These two ships were transferred from the South American trade after the war and although suitable in many respects could not offer the standard of comfort of the other ships. Their speed of about 14 knots was also a disadvantage and did not permit a call at Lobito. The Compagnie Maritime Belge, in considering the replacement of these older ships, decided to base their ideas on the *Ville* class vessels but to introduce certain improvements such as more passenger accommodation and greater power while maintaining the same service speed of 16½ knots. In order to determine the best of several hull designs, models were run in the Wageningen experiment tank. Two vessels of the new type were ordered, the first from the Chantiers de l'Atlantique, Penhoët-Loire, St. Nazaire, and the second from the Chantier Naval Cockerill-Ougrée, Hoboken, Belgium. The first ship, the *Jadotville,* ran trials in June of last year and is now in service, while the second, a new *Baudouinville,* will join the Compagnie Maritime Belge fleet this year. Six cargo holds are provided, three forward and three aft of the machinery spaces. Nos. 1 and 2 holds are arranged with upper and lower 'tween decks, while No. 3 hold comprises three insulated chambers for refrigerated cargo, together with a compartment housing the refrigerating machinery. Cargo handling is effected by eighteen cargo derricks ranging from 5 to 60 tons in capacity. For the operation of these derricks sixteen electric winches are provided. Navigational appliances are particularly comprehensive and include a complete Sperry gyro compass installation with repeaters, radar, radio telegraphy, and echo sounder, electric log, radio direction finding apparatus, a Ganson stability indicator and a Götaverken Stalodicator. An Oertz rudder is fitted, controlled by steering gear of the electro-hydraulic type. In considering the type of propelling machinery to be installed in the *Jadotville,* the owners examined the possibilities of direct-driven Diesels, geared Diesels, and steam turbines. The decision was in favour of the latter type as being the most advantageous, so a departure has been made from the Diesel propelling machinery of the previous *Ville* class ships. The propelling machinery consists of a single set of steam turbines comprising h.p. and l.p. units, capable of developing a total of 9,500 s.h.p. at 100 r.p.m. in normal service and a maximum of 12,500 s.h.p. at 110 r.p.m. The h.p. turbine is of the combined impulse/reaction type and incorporates the h.p. astern turbine in the same casing. The l.p. turbine, of the reaction type, incorporates the l.p. astern turbine in the same casing. Power is transmitted to the 4-bladed propeller through double-reduction gearing. For the measurement of shaft horse power, a Siemens-Ford torsion meter is fitted to the shaft. Steam at a working pressure of 610lb. per sq. in. and a final temperature of 800 deg. F. is supplied by three oil-fired watertube boilers fitted with superheaters, economizers and air preheaters.—*Shipbuilding and Shipping Record, 31st January 1957; Vol. 89, pp. 139-144.*

Snap-through Action in Thin Cylindrical Shells

In the study of cylindrical shells under external pressure, serious discrepancies have been noted between theoretical and observed buckling loads. Although unconfirmed, this disparity has been attributed to initial departures from circularity in models, residual stresses caused by cold rolling and welding, the possibility that the boundary conditions for shell elements in models do not correspond with those assumed in the buckling theories, and the influence of reduced tangent modulus when the buckling stress approaches the yield stress of the material. However, these factors may not fully account for the observed differences. Because a similar disparity for spherical shells under external pressure and cylindrical shells under axial loading had been resolved by a "large-deflexion" analysis which indicated a snap-through mode of instability, the presence and possible significance of a snap-through type of elastic buckling should be investigated for cylindrical shells under external pressure. Some exploratory model tests have now been conducted, and the results show that snap-through buckling occurred for these models. These results, which are presented in this report, are also evaluated in terms of post-buckling equilibrium configurations for cylindrical shells which exist at loads less than the critical buckling pressures given by a "small-deflexion" analysis.—*A. F. Kirstein and E. Wenk, Jr., David W. Taylor Model Basin, 1956; Report 1062.*

Dutch-built Gas Tanker

The motor vessel *Marian P. Billups,* a gas tanker built by Bijker's Contracting Company Ltd., shipbuilders at Gorinchem, for the Marine Caribbean Lines, Inc., Monrovia, Liberia (Agents: Marine Transport Lines, Inc., New York, U.S.A.), has been delivered to her owners. The *Marian P. Billups* is one of the few ships in the world which is designed specially for the carriage of liquefied petroleum gases (butane, propane) and anhydrous ammonia. The ship has been built as a single-deck vessel with a double bottom. The cargo bottles are placed vertically in the two holds on continuous girders welded to the tank top. The supports are welded to the bottles and are of special design approved by A.B.S. and the U.S. Coast Guard. The leading characteristics of the ship are as follows:

Length overall	285ft. 8⅜in.
Length b.p.	262ft. 10⅜in.
Breadth moulded	39ft. 4in.
Depth moulded	22ft. 1⅜in.
Draught	15ft. 1in.
Gross tonnage (American) ...	1,928·46
Deadweight all told	1,360 tons
Capacity of cargo tanks (100 per cent)	2,030 cu.m.

Main propulsion machinery:
One single-acting four-stroke Werkspoor Diesel engine arranged for turbocharging and having an output of 1,700 h.p. at 275 r.p.m.

Trial speed (loaded)	13·03 knots

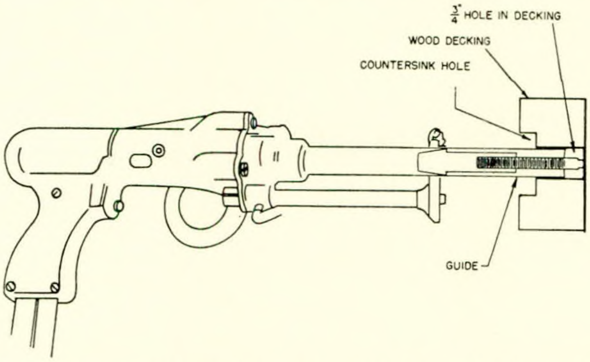
The cargo section comprises nineteen cylindrical bottles each with an outside diameter of 13ft. 9in. and a height of 28ft. 6in. They are constructed of special steel having a minimum tensile strength of 80,000lb. per sq. in. (58 kg./sq. cm.). The design pressure is 250lb. per sq. in. and the test pressure 430lb. per sq. in. After completion the bottles were entirely annealed and the welds were 100 per cent X-rayed. Each of the containers has a capacity of approximately 107 cu.m. and a weight of about 24 tons. All the connexions for liquid, vapour, safety valves, gauging devices, thermometers, and manometers are made on top of the bottles, which extend about 10ft. above the deck so that no gases can leak into the holds. Each of the gas containers is provided with two safety valves placed on a changeover valve which enables the changing over from one safety valve to another in case of failure. All safety valves are connected to a control system which indicates any safety valves which may be leaking. The indicators of this system are placed in the central control station. The cargo is handled by two "Demag" piston type compressors and two 3-stage K.S.B. centrifugal pumps. Each of the compressors has a maximum capacity of 147 cu. ft./min. and the capacity of each of the pumps is 880 U.S. gals. per min. The pumps and the compressors are placed in the pump room which is located in the 'tweendeck aft. They are driven by electric motors placed in a separate space. The driving shafts pass the separation bulkheads through gastight glands. The piping system is made so that two different products (ammonia and propane) can be carried at the same time, each product being handled by one pump and one compressor whereby the pipelines for each product are completely separated. With one pump and one compressor a complete cargo can be discharged in about twelve hours. The cross-over outlets (four at each side) are 6in. for liquid and 4in. for vapour. Two intermediate containers, each serving one pump, are placed on the maindeck above the pump room. With these intermediate tanks almost any possible loading and unloading procedure wanted by the shore receiving plant can be put into effect. Since the cargo is kept liquid under pressure, it is impossible to have direct suction of the pumps to the cargo bottles and, therefore, the liquid is heaved out of the bottles to the intermediate tanks. This is carried out by the compressors taking suction from the intermediate tanks and pumping to the cargo bottles, which causes a differential pressure necessary to heave the liquid. The pumps placed under the intermediate tanks take suction from these tanks. This procedure ensures a continuous flow of liquid without vapour bubbles to the pumps. This is also highly important for the exact working of the two quantity metres which are installed. The intermediate tanks, which in this way act as some kind of accumulators, each are provided with two liquid level control devices, one being pneumatically operated with distant reading indicator and the other being of a special flat glass type. Another feature of the cargo handling installation is the pneumatic operation of all the valves of the compressor manifolds, the valves of the quantity metering installation and of all the valves at the cross-over outlets. A total of thirty-six valves are operated pneumatically from the central control station. The valves are operated individually by means of compressed air supplied by an air bottle in the engine room and can be closed simultaneously in case of emergency. Moreover, the valves can also be operated by hand.—*Holland Shipbuilding, December 1956; Vol. 5, pp. 28-30; p. 32.*

Scavenging of Two-stroke Engines

The work described is an extension of the investigation into air flow in a naturally aspirated two-stroke engine described in an earlier paper. In that paper the scavenging process of an unblown, opposed piston engine with simple parallel pipe exhaust and inlet systems was analysed, and the results were compared with experimental observations. Only a brief reference was made to the very marked increase in air flow resulting from incorporation of a diffuser in the exhaust system. The present paper describes first a theoretical investigation of wave effects in the exhaust system using: (1) The small wave theory; (2) The method of characteristics; (3) An approximate analytical treatment designed to give better accuracy than (1), but to avoid the extremely laborious computations associated with (2); and, secondly, an experimental programme comprising: (1) Air-flow tests with various diffuser arrangements and over a wide range of speeds, and (2) Tests in which indicator diagrams were taken at salient points of the exhaust system. The general conclusions are: (1) The incorporation of a diffuser in the exhaust system leads to an increase in air flow of up to 85 per cent under favourable conditions. (2) The very complex phenomena occurring in the exhaust system can be dealt with satisfactorily by the approximate analytical treatment with great saving in time over the method of characteristics and very much better accuracy than can be achieved with the small-wave theory.—*Paper by F. J. Wallace and G. Boxer, submitted to The Institution of Mechanical Engineers for written discussion, 1957.*

Adapter for Shooting Studs into Deck Holes

An adapter for use in shooting studs into the steel deck through the holes in wood decking has been suggested. Formerly, studs were often shot off centre, causing damage to the screw nut on the stud when the stud was forced into correct alignment. The adapter consists of a piece of pipe the same diameter as the hole. It is welded over the chuck which holds



Pipe fits against the sides of the hole to guide the stud into the centre

the stud and it has a ½-in. tap hole at the top for attachment to the stud gun. Since the pipe fits snugly against the sides of the hole, it serves as a guide to centre the stud in the hole. —*Bureau of Ships Journal, March 1957; Vol. 5, p. 45.*

New Stork Marine Engine

Fig. 4 shows the new engine with uniflow scavenging, by scavenging ports in the liner and four poppet exhaust valves in the cylinder cover, which is built in three sizes to cover as turbocharged engine the field from 2,500 to 15,000 b.h.p. The maximum size has a bore of 750 mm. and a stroke of 1,600 mm., with a continuous rating of 1,200 b.h.p. per cylinder at 115 r.p.m., at a b.m.e.p. of 96lb. per sq. in. The performance of this engine with the exceptionally low specific fuel consumption of 0·33lb. per b.h.p. per hr. and only a slight rise beyond full load, indicates the ability to take higher loads without materially overloading the engine. On account of the new design, with up to now only three years' experience, Stork still prefers to limit the rating at the appropriate figures of 96 per sq. in. b.m.e.p. and 1,207ft. per min. mean piston speed until further accumulation of experience proves that a higher continuous load can be safely admitted. The fuel consumption with heavy fuel is normally about 0·02lb. per b.h.p. per hr. higher, mainly on account of the lower calorific value of this type of fuel. The low fuel consumption is explained by the high mechanical efficiency of 90·5 per cent as the specific indicated consumption has the normal value of 0·31b. per i.h.p. per hr. This high mechanical efficiency is mainly a consequence of the application of turbo-

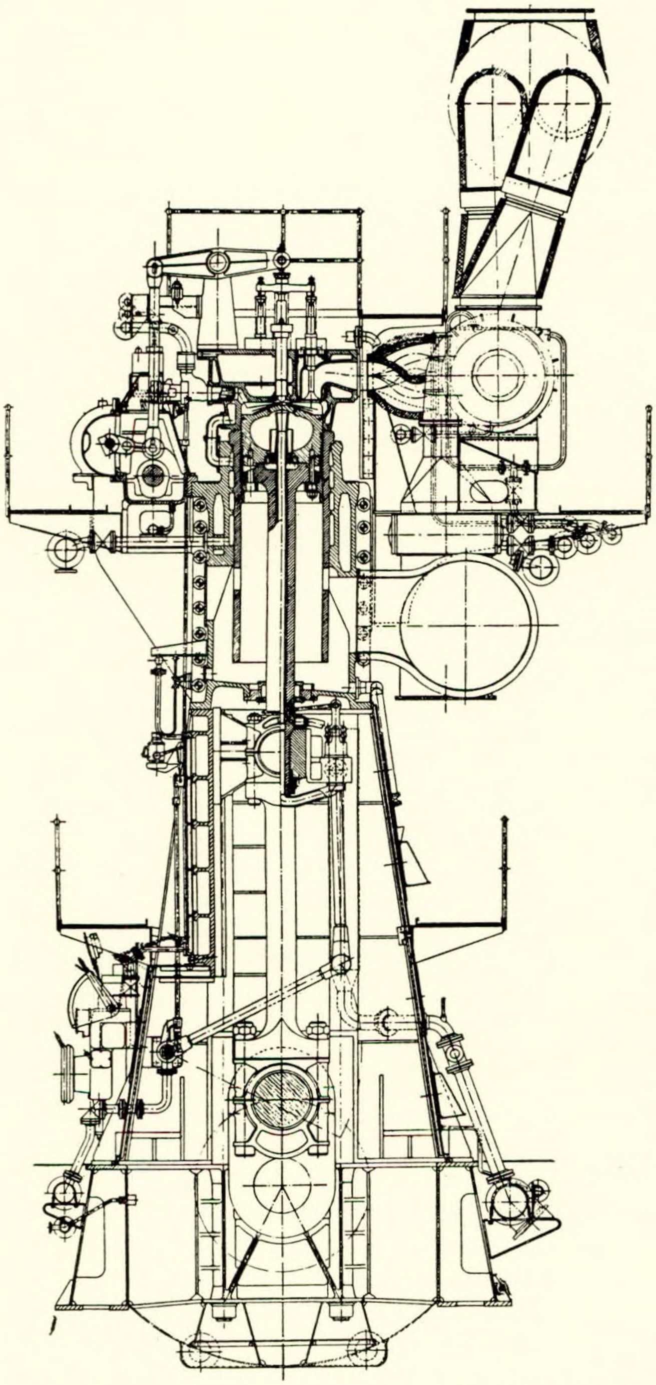


FIG. 4—*Cross section of engine with uniflow scavenging and turbocharging, type HOTLo 75/160, 750-mm. bore and 1,600-mm. stroke*

charging, which makes the engine-driven scavenging air pump, with an energy consumption of about 5 per cent of the indicated horsepower, completely superfluous. The engine is turbocharged on the pulse system which takes advantage of the kinetic energy in the exhaust gases as well as their static energy. In order to keep this energy on a high level and to transmit it with minimum losses to the exhaust turbines, special care has been taken to streamline the exhaust ducts in the cylinder cover and to situate the turbochargers as near to the covers as possible. Moreover, the use of one turbocharger per adjacent pair of cylinders has substantially assisted to the execution of this aim in perfection. And there are the four exhaust valves in the cylinder cover which enable a very rapid opening of the exhaust area to create an exhaust pulse with maximum energy. All this together has resulted in an engine that starts very easily and is able to run at every load and speed with completely clear exhaust and without any need for auxiliary scavenging air supply. The big 750-mm. bore engine has the very low minimum speed of 18 r.p.m., with the turbochargers running at 1,000 r.p.m., and tests discovered that at starting the turbochargers respond immediately to the first blast of starting air, after being released by the exhaust valves, reaching a speed of 2,000 r.p.m. These turbochargers of the BBC VTR 500 type are built for a maximum speed of 9,700 r.p.m. and thus run with a very safe limit, even at the maximum load of about 1,300 b.h.p. per cylinder, when their speed is only 8,000 r.p.m. By the use of one blower for every two cylinders there will be at least three blowers on the (six-cylinder) engine and this number permits the engine to run safely at about 85 per cent speed if ever one blower should fail. The Stork turbocharged two-stroke engine is therefore reliable in all circumstances without any emergency auxiliary air supply. As there are two completely separated exhaust ducts in every cylinder, each combining the gases of two exhaust valves, it is very easy to connect a turbocharger of the two gas-entry type with 1½ cylinders. This possibility allows a very satisfactory application of the multiple turbocharger principle in any engine with an odd number of cylinders, by connecting some turbochargers with two cylinders and some with 1½ cylinders. As a rule the number of turbochargers will then be half the cylinder number plus one half; thus, four turbochargers on a seven-cylinder engine, etc.—*Paper by A. Hootsen, read at a Meeting of the North East Coast Institution of Engineers and Shipbuilders on 11th March 1957.*

Progress Towards Nuclear Submarine

An announcement by Vickers Nuclear Engineering, Ltd., reveals the making of "good progress" in the development of a set of nuclear machinery suitable for use in a submarine. This prototype set is to be installed in a mock-up of a hull structure on land, where it will be used for trials purposes and also for the training of personnel. Of the three firms joined in Vickers Nuclear Engineering, Rolls-Royce is responsible for the design of the reactor and associated equipment, Foster Wheeler for the steam generators and Vickers-Armstrongs for the turbines, condensers and other auxiliaries. But designs of any sort are agreed upon by all three firms before being submitted to the Admiralty. The machinery, it is announced, is to be based on the pressurized water reactor, as used in the American submarine *Nautilus*. This is without doubt the safest starting point for any nuclear propulsion development programme. The *Nautilus* herself has amply proved the reliability of the pressurized water system, while the value of this reliability has been emphasized by the difficulties which the Americans have been experiencing with the sister submarine *Sea Wolf* and her more advanced liquid sodium propulsion system. The pressurized water reactor, however, is one of the least efficient systems from the point of view of fuel economy and the shipping industry will be hoping that any developments that may be incorporated in the design will be aimed at reducing the fuel consumption, which, though infinitesimally low in weight, is still high in cost.—*The Shipping World, 6th March 1957; Vol. 136, p. 255.*

Oil Disposal Raft

The Naval Repair Facility, San Diego, has recently built an oil disposal raft designed for cleanliness, speed, and economy in service provided to forces afloat. It is used for the disposal of water contaminated with oil, grease, or similar materials. The water is pumped from machinery spaces, bilges, ballast tanks, and cargo tanks containing ballast oil and water. The contaminated liquid is pumped into the oil disposal raft where the oil separates from the water and remains floating. Since the water is of heavier density, it flows out the bottom through several large openings provided for that purpose. On one end of the raft is a pipe connexion to which the discharge hose is joined. The lower end of the pipe is turned up to deflect the flow of liquid away from the bottom openings. In this way the contaminated liquid cannot be forced out through the openings before the oil or contamination has had time to separate from the water. The total capacity of the oil disposal raft is approximately 32,000 gals. Because of the turbulence set up by discharging fluid into the unit, and also when moving it, a 24-in. clearance must be left between the lower level of the oil and the bottom of the raft where the drain openings are located. Therefore, the effective capacity is reduced to approximately 27,000 gals. of oil. To avoid excessive turbulence and a resulting flow of contaminated oil and material into the open water, the rate of discharge is kept at a minimum. A watch is posted to check the area immediately around the raft for the appearance of any oil. Traces of oil observed when the raft is partly filled indicate that the rate of discharge is too great. The amount of oil remaining in the raft is measured by sounding. Either a sounding tape or rod graduated in inches and coated with water finding paste is used to show the total depth of oil. An oil reprocessing contractor removes the oil and purchases it. An important use for reclaimed oil is in the forge shop oil burners.—*Bureau of Ships Journal, January 1957; Vol. 5, p. 38.*

New Method of Shaft Aligning

A new procedure for aligning submarine shafting has been suggested by the Charleston Naval Shipyard. The method improves bearing fit so that there is less shaft noise. The shafting is supported in a long bed shaft lathe at the point of the after end of the after strut and the forward end of the stern tube. Experiments are made to determine the exact amount of sag at the aft end of the after stern tube. Then the shafting is installed in the vessel. Two jacks support the shafting at 4- and 8-o'clock positions at the forward end of the forward stern tube and the after end of the after strut. The shafting, lying in its normal sag, is aligned with the motor coupling. The alignment reading should be on the face 0·000 inches port, starboard, top, and bottom. The after coupling should be 0·015 inch higher than the motor coupling. Then the sag is checked at the after end of the after stern tube. The shafting is lifted the exact amount of the sag as shown in the lathe experiments. An indicator is used on each side of the shaft to prevent any side deflexion. A correction check is made at the end-board couplings. Any deflexion from the original settings is compensated to bring the shafting back to the original coupling setting. When the shaft is in the correct aligned position, readings are taken between the land and the journal on the bottom port and starboard sides forward and aft end of each bearing. The bearings are bored to an accurate crown dimension according to the readings. All bearing clearance is thrown into the upper half so that the lower crown reading is maintained. Sound data and return reports from vessels on which the new procedure has been used have shown improved sound readings on the shafts.—*Bureau of Ships Journal, January 1957; Vol. 5, p. 40.*

Paddle Wheel Experiments

A working link between model and ship has been established. This in itself represents a considerable advance on all previous paddle wheel investigations in which any ship/model correlation was ignored. It is perhaps unfortunate that the base parameter includes speed, r.p.m. and size and, if any of these are unknown, a second approximation may become necessary. Indeed the whole process of designing an optimum wheel for a given set of conditions is inevitably laborious, requiring considerable interpolations and a number of cross curves. The results in this paper confirm the evidence accumulated throughout the investigation that the behaviour and performance of the paddle wheel is a highly complex problem which cannot be readily reduced to any simple terms. Many of these complexities remain unexplored and are likely to remain unexplained unless much further experimental and theoretical work is undertaken. The terms of reference, however, for this limited research programme were to evolve a practical system whereby paddle wheels might be designed, similar to that available for screw propellers and with the expectation of a reasonable degree of accuracy. It is considered that this requirement has on the whole been fulfilled even if it has not been possible to avoid some degree of empiricism in the ship/model correlation.—*Paper by H. Volpich and I. C. Bridge, read at a meeting of the Institution of Engineers and Shipbuilders in Scotland on 26th February 1957.*

Deformable Packing Blocks in End Launchings

When preparing a ship for launching, the strength of the hull must be carefully considered, and if necessary it must be specially reinforced to enable it to withstand the launching stresses. This is particularly important in ships with the machinery placed aft, since with this weight distribution the way-end reactions may be especially high. The present article describes a method of reducing the peak launching stresses by introducing an appreciable amount of deformability into the sliding ways. This is done by placing special deformable flat blocks between the packing timbers. Rubber or similar materials, soft woods, or wax compounds may be used for these blocks. To determine the suitability of a particular material, the rate of deformation of the blocks for various rates of load increase can be determined in the laboratory by means of an hydraulic press; graphs showing this are given for blocks 5-cm. (about 2in.) thick made of paraffin-wax-vaseline compound. The graphs show clearly the behaviour of the material under load. To test the efficacy of the deformable blocks in practice, they were used in the launch of a 10,000-ton tanker (launching weight about 4,000 tons). Recording strain gauges, previously calibrated, were placed in sections 8 and 9 of the sliding ways, which were just amidships; section 8 contained deformable blocks about 40 mm. (1·6in.) thick of paraffin-wax-vaseline compound placed under each packing timber of the top packing layer, and section 9 was without such blocks. From the strain recordings taken during launching, the hull stress could be determined. The results show that the deformable blocks appreciably reduce the peak stresses; it is believed that this reduction may in some cases be sufficient to dispense with launching reinforcements altogether. Another consequence of the peak-stress reduction is the possibility of shortening the immersed parts of the standing ways. Plastically-deformable blocks seem to be preferable to elastic clocks. When it is known that the temperature of the blocks will not rise much above the ambient temperature, paraffin-wax-vaseline blocks are suitable; otherwise, blocks of soft wood (fir) are recommended. The deformable blocks are also useful in promoting a more even stress distribution when dry docking a ship.—*A. G. Arkhangorodsky and A. B. Litvin, Sudostroenie, 1956; Vol. 22, No. 7, p. 1. Journal, The British Shipbuilding Research Association, February 1957; Vol 12, Abstract No. 12,527.*

A Method of Analysing Voyage Data

This paper describes a method of analysing service performance data devised at the British Shipbuilding Research Association, by means of which it is possible to estimate the effect of weather and deterioration of the hull and propeller surface on the shaft horse power necessary to maintain a given speed. A method of multiple-regression analysis is used to obtain the separate effects of adverse weather, and of fouling. It is shown how a suitable weather scale has been evolved by which it is possible to estimate the performance of a vessel in unrestricted waters knowing only the wind strength and direction. A method estimating propeller efficiency in service is also described, and it is shown how this can be used to estimate the efficiency under most conditions likely to be encountered in service. By extending the analysis it is possible to estimate the thrust necessary to maintain a given speed under any conditions of weather or of fouling. In the present work, however, the thrust analysis has been restricted to comparisons between the thrust for a given speed on trial and the thrust for the same speed at the start of the maiden voyage and immediately after each subsequent dry docking. A numerical example is included and the results obtained from the analysis of voyage records of eight vessels are summarized. —*Paper by R. E. Clements, read at a meeting of the North East Coast Institution of Engineers and Shipbuilders on 25th January 1957.*

Arc Welding Injector

There would be a considerable advantage in being able to strike a welding arc without allowing the electrode to touch the workpiece. It is now believed the effect of an arc striking on the parent metal away from the joint is to produce a highly stressed point which may give trouble, and that any such marks should be chipped out and welded up. The Metrovick H.F. injector superimposes a high voltage on the output of the welding transformer or generator at a frequency high enough to remove the danger of electric shock. This allows the arc to be struck without coming into contact with the workpiece. It also provides an open circuit voltage high enough to maintain a stable arc under difficult combinations of electrode, shielding atmosphere, and composition of parent metal. The equipment, contained in a welded steel box, consists of a high voltage transformer with a secondary of 4,000 volts, a capacitor, a double spark gap with tungsten points, and a water-cooled air cored transformer to inject the high frequency voltage into the welding circuit, while isolating the power frequency high voltage from it. The secondary winding of the air cored transformer carries the full welding current and is suitable for 400 amperes continuous duty, or 450 amperes intermittent duty. The power consumption is 100 watts.—*Shipbuilding and Shipping Record, 14th February 1957; Vol. 89, p. 215.*

Cathodic Protection by Trailing Platinum-clad Anode

Current distribution in cathodic protection systems is dependent on several factors, the more important of which are driving potential, anode and cathode geometry, spacing between anode and cathode and the conductivity of the electrolyte. When designing cathodic protection systems for the underbody of active vessels, the corrosion engineer is plagued with the problem of achieving uniform current densities with anodes mounted directly on the hull. The consistent high conductivity of open sea water is favourable to good distribution of current. The corrosion engineer has wide control over the geometry of the anode and the driving potential of the system, but he is seriously limited with respect to the spacing between the anode and the cathode. In general, hull mounted cathodic protection systems require multiple anode assemblies fitted to the ship while in drydock. The system cannot be adequately inspected or maintained during service which is inconvenient and costly, if not operable. The difficulties of installation and maintenance of a cathodic protection system on a ship's hull as well as the variations in performance can be overcome by using a trailing anode. By removing the anodes from the hull proper and consolidating them into one light weight, streamlined, high capacity anode, one can trail such an anode a sufficient distance from the hull to give remote resistance to ground. This produces essentially uniform current paths to all parts of the hull. Edison in 1890 had tried to design such an anode but failed due to the lack of proper materials. Today with the knowledge of platinum cladding and special cable sealing techniques, a simple high capacity trailing inert anode is feasible, and practical. The trailing platinum anode has demonstrated its ability to protect cathodically a small ship with comparative simplicity in port and at sea. Uniform polarization of the hull was achieved with a current output of five amperes with the ship at rest and 12 amperes when underway at nine knots. Platinum anodes remain clean in service and do not build up resistant films on their surface as evidenced by the linear relationship between current output and driving voltage. The ease of trailing the platinum anode service conditions and its high current handling capacity make this type of system easily adaptable to larger ships. The trailing platinum-clad anode should find application on naval and commercial ships. It may not be well suited for high speed warships operating and manœuvring in convoy stations but it should lend itself to naval auxiliaries and merchantmen. The upper limit of current output for a trailing platinum-clad anode is dictated only by its dimensions and the driving voltage available. It is believed to be the least expensive and most versatile cathodic protection system known for an active ship and its installation and maintenance is relatively simple to achieve. Further tests are planned at sea with larger and faster ships to determine the anode's effectiveness in polarizing greater hull areas and to determine the anode trailing characteristics at higher speeds. Should these tests prove successful, plans will be drawn up to have the anode trail from an access pipe fitted through the hull and be retrieved automatically.—*H. S. Preiser and F. E. Cook, Corrosion, February 1957; Vol. 13, pp. 125-t – 131-t.*

Refractory Nozzle Blades for High Temperature Gas Turbines

The operating temperatures of gas turbines are dependent on the properties of the blade material. Since the advent of the jet engine, immense strides have been made in the improvement of metallic alloys, and long-life gas turbines can now be designed to run at temperatures up to about 1,400 deg. F., but this development is probably nearing its limit. In order to achieve a fuel economy comparable with that of a marine Diesel engine the operating temperature must be raised to something of the order of 2,200 deg. F., and this will clearly require cooling of the blades or alternatively the use of non-metallic materials. The existence of high centrifugal stresses in rotor blades facilitates cooling by the thermosiphon principle and at the same time militates against the use of non-metallic materials, which tend to be relatively weak in tension. The stator blades, however, offer a promising field for the application of non-metallic materials, and the present paper describes developments which have been carried out over the last eight years. Numerous materials were tested in the laboratory under conditions designed to simulate the stresses and thermal shocks that were encountered in service. Various methods of manufacturing the required blade shapes were also investigated, and the shapes themselves were modified to some extent so as to meet the special requirements of the materials. Cascades of blades were tested in a high velocity gas stream at temperatures up to 2,200 deg. F. Several materials were found to have reasonably good thermal shock resistance, and creep strength at high temperature appeared to be the most serious limitation. At least one material has emerged which seems likely to fulfil the necessary requirements. A single-stage turbine embodying refractory stator blades in conjunction with a liquid-cooled rotor is now in course of development.—*Paper by T. H. Blakeley and R. F. Darling, read at a meeting of the North East Coast Institution of Engineers and Shipbuilders on 8th February 1957.*

Combined Steering Console

The Brown combined steering console for bridge mounting is designed to meet the requirements for (1) telemotor hand steering, (2) electric hand steering, and (3) automatic steering. It is used in conjunction with Hastie Donkin, Brown and Mactaggart-Scott steering engines in this country and in one or two cases abroad with Siemens/Atlas and Asea engines. The console has already been fitted in six ships, the first being the *Hoegh Grace* (Brown/Hastie) built at Kiel, and the latest, the *Port Launceston* (Brown/Mactaggart-Scott) at Belfast. A vertical partition divides the console into two parts, the front part housing the bridge telemotor pump unit, and the after part, nearest to the large and small hand wheels, the apparatus associated with the automatic and hand electric steering systems, the cable connexions, etc. On top of the unit is the Brown multiple steering repeater with an expanded scale giving degree readings a quarter of an inch wide. Indicator dials show the operating pressure in each line. On the port side of the console there is a lever with three definite locating positions to provide for (1) hydraulic, (2) hand electric, and (3) automatic steering. In the first position, steering is effected by the movement of the large wheel and the rudder position obtained is indicated by a large mechanically-operated pointer. There is no connexion with the gyro compass but the ship's head direction may be read from the steering repeater. In this position of the main control lever, the bypass valve is fully closed. In the second position, for hand electric steering, the bypass valve is open and electrical connexion is made to the appropriate units of the Brown system. Steering is by the small wheel when the appropriate connexions are made via the hand-driven impulse transmitter, the breaker switch and the aft power units which allow for the latter moving the operating valve of the steering engine. The rudder movement and position are shown by an electric helm indicator. In the third position, the bypass valve is open and the necessary electrical connexions are made to the Brown gyro compass repeater. Steering is entirely automatic from the datum provided by the gyro compass. If the rudder angle control has been set to zero, it will be necessary to revert to the appropriate setting and also to adjust the "yaw control" to suit weather conditions. Both settings are quickly accessible through the starboard door of the console. If there is a failure of the repeater system, a very rare event, an alarm is given by a red light on the front panel and a bell inside the console. The red light persists until the fault is cleared and the relay reset by pressing the receiver button; in the meantime manual steering is used. Items of equipment used in conjunction with the Brown combined steering console and normally fitted in the steering compartment are the telemotor receiver, the charging tank and pump and the Brown after-power unit.—*Shipbuilding and Shipping Record, 31st January 1957; Vol. 89, pp. 148-149.*

Machinery of the *Statendam*

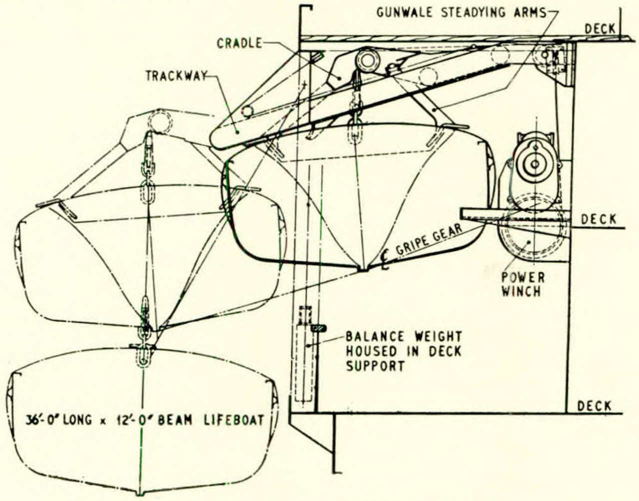
The new Holland-America steamship, which sailed on her maiden voyage from Rotterdam on 6th February, is indicative of the new thought prevailing in the passenger trade. The new ship was built in the Wilton-Fijenoord construction dry dock at Schiedam. The principal particulars are:—

Length overall	642ft. 3in.
Length between perpendiculars	578ft. 2½in.
Moulded breadth	78ft. 9in.
Service speed	19 knots
First class passengers	84
Tourist class passengers ...	871

The *Statendam* is of importance from an engineering viewpoint because she is the first passenger ship and also the first twin-screw vessel to have Pametrada-type turbines built by a foreign licensee. The main machinery is controlled from two desks, each of which carries only those instruments essential for manœuvring. These desks are arranged on the cross platform forward of and above each turbine set. The Wilton-Fijenoord Pametrada turbines develop a total of 22,000 s.h.p. at 120 r.p.m. and are of the Association's normal design with an impulse h.p. turbine and overhung astern element, cross-compounded with a single-flow impulse-reaction l.p. turbine. Speed reduction is by Maag double-reduction single-helical articulated gears with hardened and ground teeth in cast iron casings, the first to be installed in a twin-screw or passenger ship. The sound level is exceptionally low, a result one has come to expect from these famous Swiss gear makers. There are clear gangways across the engine room at the control platform and at the after end, also fore-and-aft gangways on the centre line and in the wings. The absence of the usual forest of large diameter steam pipes rising from the turbines enables a clear view of the entire compartment to be obtained from almost any point at platform level. A high and even level of illumination is maintained by fluorescent light fittings. The wings of the engine room at slightly above platform level are occupied by four 700 kW-Werkspoor-Smit turbo-alternator sets. The mechanical portion of these machines has been built under B.T.H. licence. The main switchboard extends across the after end of the engine room and incorporates a central desk type alternator control section. The auxiliaries concerned with the main machinery, Weir closed feed system and turbo-alternators, are arranged at tank-top level. The boiler room contains two Wilton-Fijenoord - Foster Wheeler two-furnace controlled superheat boilers which deliver steam at 625lb. per sq. in. at 850 deg. F. The saturated and superheated furnaces have four and five Todd burners respectively. There is a Bailey boiler control panel in the centre of the firing aisle but instead of full automatic control, the Bailey pneumatic servomotors for operating the control dampers are adjusted by hand control. The forced and induced draught fans are of Stork design and Weldex bled steam air heaters are employed. The working alleyway is on C-deck and thus extends through the engine room along the port side, where it has a number of side scuttles.—*The Marine Engineer and Naval Architect, March 1957; Vol. 80, p. 79.*

New Davit and Glass Fibre Lifeboat

Among the many interesting innovations that are likely to be seen in the Orient liner *Oriana,* 40,000 tons, when she is completed, are a davit of new design and a glass fibre lifeboat of a size larger than has previously been made in this material. The new davit has been designed by Welin-MacLachlan Davits, Ltd., in conjunction with Mr. C. F. Morris, naval architect to the Orient Line. Trackways are connected to the underside of the deck above the boat deck and the winches are mounted halfway up the deckhouse side, thus giving full headroom under them on the embarkation deck. The boats are suspended from cradles on the trackways. The cradles for lifeboat launching travel outboards, impelled by forces derived from track inclination and counter weights, thus meeting the Ministry of Transport regulations. The lifeboats are stowed



New davit and glass fibre lifeboat

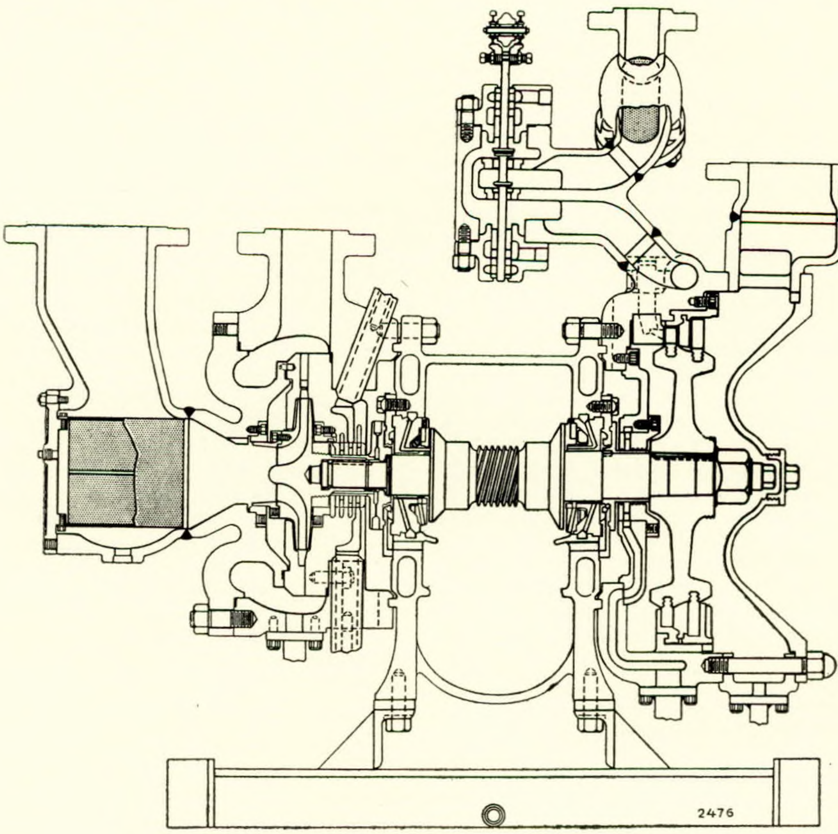
with a slight angle of heel in order to preserve the maximum view from the windows in cabins on the deck in between the boat deck and the promenade deck. The lifeboat cradles are in light alloy and it is of interest to note that the new davit complete with counter weights is 15 per cent lighter than the standard trackway type of overhead gravity davit. This new davit can launch a boat against an adverse list of 25 degrees and a 10-deg. trim, and this has been confirmed in tests carried out by the Ministry of Transport. The three upper deckhouses and the bridge superstructure of the *Oriana* will be of light alloy construction, and due to the saving in top weight which is being made in using this material, the improved stability will enable an extra deck of passengers to be carried. The ship's lifeboats will be carried on the highest strength deck in the new davits fixed to the deck head above, allowing space below the boats with a top clearance of about 7ft. for a promenade deck. This boat deck will consist of a bay occupying the height of two decks below the main promenade deck, and running the requisite length of the superstructure so as to accommodate the required number of boats. The lifeboat is moulded in one piece and, apart from the wooden rubbers, bilge rails, bulkhead cappings and gunwale—sheathed with glass fibre—practically all structural woodwork has been dispensed with. The bilges are filled completely with Onazote covered with glass fibre, giving a clean level floor and forming a strong structural sandwich. The fresh water and fuel tanks are of moulded reinforced resin, as are the seats and the hundred or more watertight compartments.—*The Shipping World, 20th March 1957; Vol. 136, pp. 313-314.*

Pacific Turbo-feed Pumps

A new range of single-stage high pressure pumps for boiler feed service is announced by Pacific Pumps, Inc. These are self-contained, compact units capable of continuous operation at speeds from 5,000 to 10,000 r.p.m. The design and construction conform to the rules of the American Bureau of Shipping. The outputs extend to 840,000lb. per hr. against pressures of up to 1,800lb. per sq. in. The steam turbine is made in two sizes, both of which can be furnished with either a single or a double row bucket wheel. Both are suitable for operation with steam conditions up to 1,500lb. per sq. in. and 950 deg. F. Two sizes of single-stage, single-inlet diffuser type pump have been developed. One size, having a 3-in. discharge flange, is suitable for capacities to 33,000lb. per hr. and discharge pressures to 1,500lb. per sq. in. The other size has a 4-in. discharge flange and is suitable for the maximum capacities already mentioned. The pump frame is made up of three parts, the pump case, bearing case and the turbine case, all of cylindrical form. The liberal diameter and short span between the bearings and the dynamically balanced bucket wheel and impeller ensure stiffness and freedom from shaft whip. The turbine end of the main shaft is sealed with a labyrinth gland for atmospheric or back pressure exhaust and by steam sealing for exhaust pressure below atmospheric. The main shaft bearings, the Kingsbury thrust bearings, the governor drive worm and gear are pressure-lubricated through a single manifold system. The nozzle ring is of rectangular section, made in segments cut from a circular forging of chrome alloy steel, each being bolted to the case with outside tap bolts. The steam nozzles are formed by accurately drilled and reamed passages in each segment. The number of openings and their throat diameters are arranged to furnish the necessary steam flow. The expansion ratio is determined by the inlet and exhaust steam pressures. The turbine bucket wheel is machined from an alloy steel forging finished all over, forced on the shaft on a taper fit and locked in position with a nut and ear-type lock washer. The drive is through three keys spaced 120 degrees apart. The buckets are machined from turbine quality chrome alloy steel bar with integral roots and shrouds.—*The Marine Engineer and Naval Architect, March 1957; Vol. 80, pp. 107-108.*

Corrosion of Alloys in Superheater

A series of heat-resistant alloy test racks containing some 30-odd test specimens was installed in the gas inlet of the second bank of the superheater in a Public Service Electric and Gas Company boiler. This boiler had been burning bunker C fuel oil for the three-year period before the test and used bunker C during the entire test period. Gas temperatures varied from 1,560 deg. F. at full load to 1,100 deg. F. at minimum load. The loading on the boiler during the investigation was considered typical of steam boiler operation. Scale taken from the build-up on the test specimens analysed as high as 36 per cent vanadium pentoxide and 32 per cent sulphuric anhydride. An analysis of the ash constituents present in the oil-fired boiler during the tests also showed high vanadium, sulphur, sodium, and calcium contents. All corrosion rates were measured by metallographic examination and were expressed as inches penetration for the duration of the test and extrapolated to inches per year (IPY). Corrosion results were obtained on both cast and wrought alloys as well as some alloys with various coatings. An extensive metallographic investigation was conducted on all alloys exposed to determine the type of attack each material underwent during the exposure period. Some alloys had a corrosion rate as much as ten times greater than others.—*D. W. McDowell, Jr., R. J. Raudebaugh, and W. E. Somers. Transactions A.S.M.E., February 1957; Vol. 79, pp. 319-328.*



Section through a Pacific turbo-feed pump. The oil pump is driven by the worm in the centre of the case

Mechanical Infinitely Variable Speed Drives

Innumerable applications for which variable-speed drives are desirable, or even essential, have stimulated the invention of an enormous crop of devices intended to provide infinitely variable speed regulation of equipment. Of these, some have been impracticable, many have been inefficient, and others have been prohibitively complicated and costly. Nevertheless, there still remains a number of well-defined types, produced by a comparatively large number of manufacturers all over the world, which, by virtue of their efficiency and inherently sound design, have satisfactorily withstood the demands of practical requirements. It should be understood, however, that the selection of the most suitable type of variable-speed gear for a given application is not necessarily a simple matter. Indeed, numerous aspects must be considered if the best compromise between such factors as initial cost, efficiency, speed range, power and torque characteristics, life expectancy, maintenance, and environmental conditions is to be achieved. Besides elaborating these factors, the purpose of this article is to describe and illustrate the major types of currently available mechanical infinitely variable speed drives throughout the world and to give some indications of the applications and particular advantages of each type, with performance data and the design features of individual makes. By definition, infinitely variable speed drives are mechanisms which convert a constant input speed into output speeds which are steplessly variable within a certain range. This range may cover a definite ratio of maximum to minimum output speed, e.g., 4 : 1, or it may reach from a maximum output speed right down to zero, in some cases in both directions without changing the direction of rotation of the input shaft. By their nature, then, these drives do not include stepped methods of speed adjustment providing certain predetermined fixed speed ratios, e.g., gearboxes and stepped pulleys, these constituting a class of transmission of their own.—*P. Cahn-Speyer, Engineers' Digest, February 1957; Vol. 18, pp. 41-65.*

Port Communications

In a paper on The Future of Marine Electronics, the author states that at the present time a ship approaching an average port has no means of communicating with the port authority or, say, the pilotage service except through such public correspondence channels as may exist. The coast stations providing these facilities are often some considerable distance from the port, and they have to serve all manner of passing traffic as well as that entering. On the whole, quality is poor by modern standards and delays may be long; Distress Working by other vessels may greatly aggravate delays in the very circumstances in which they need to be avoided. On the other hand, port authorities are becoming alive to the need to improve their facilities and methods of operation. There is no doubt that v.h.f. radio telephone is well suited to port communications. It is already being put into service in several ports throughout the world. There are, however, quite a number of problems which require international discussion and agreement before shipowners and radio manufacturers can get down to the business of making and buying suitable ship equipment for international use. The number of channels and whether single or double frequency should be used, or both, cannot be estimated until some essential principles of systems planning are laid down. Final arrangements must be held up until an essential minimum of "rules" are agreed. A v.h.f. system is planned for Southampton and another for Rotterdam.—*F. J. Wylie, The Journal of The Institute of Navigation, January 1957; Vol. 10, pp. 50-57.*

Fluidized Plastic Coatings for Corrosion Resistance

In the process known as fluidization coating, the parts to be coated are heated above the melting range of the plastic and then immersed for a few seconds in a bed of specially prepared plastic powder, maintained in a bubbling or "fluidized" condition by compressed air or inert gas. On removal, residual heat melts and levels the adhering powder particles to a smooth, non-porous resin coat, forming an adherent, tough, and corrosion- and electrically-resistant cladding. Plastics used at present include polyethylene, nylon, some polyethers, and certain fluorothenes. Conceivably, however, any thermoplastic resin may be used for fluidization coating, providing that it exhibits reasonable stability against degradation and oxidation above its melting range. Advantages of the process include the fact that (1) components, ready for use, can be produced in a short time; (2) solvents are eliminated, thereby improving coating porosity, compared with dispersion or solution types of coatings; (3) thick coatings can be applied in a single dip, without sagging, running, or crazing, while on heavy metal parts, coatings up to 100 mils can be applied in a few dips without cure or cool-down between immersions; (4) coatings are uniform in thickness, since the scrubbing, turbulent action of the fluidizer reaches the most recessed areas of the part; (5) there is no material loss, compared with spraying, when coating open-meshed or small objects; and (6) portable fluidizers can be used where on-the-spot protection must be applied. The absence of solvent or plasticizers develops maximum corrosion resistance in polyethlenes or fluorothenes. Plasticizer blends are not required, as in plastisol application, removing the problem of age brittlement caused by leaching or migration of plasticizers.—*J. A. Neumann and F. J. Bockhoff, Product Engineering, January 1957; Vol. 28, No. 1, pp. 140-143.*

Electrode Coatings

Electrode coatings have gone through a natural and systematic evolution, yet surprisingly little fundamental information is available concerning the precise effect of each of the various coating ingredients on electrode characteristics and weld quality. The analogy between welding and steel making practice has been heavily drawn upon in the course of development of present day coating, even though it is recognized that many important differences do exist. Modern coatings contain numerous ingredients; some promote fluxing and slag formation, others enhance electrical characteristics of the arc, still others provide binding. The study reported herein is a continuation of a research programme started two years ago at the University of California in which an attempt is being made to illuminate more clearly the influence of each of the individual coating ingredients on the quality of the final weld deposit. It is the immediate goal of this programme to isolate and to study the effect of each of the ingredients on important influencing factors such as surface tension of the molten metal, bead shape and quality and arc characteristics. The programme to date has been limited in scope since it is impossible at the present time to define clearly all of the variables involved or to investigate the many possible welding conditions. However, an attempt has been made to determine the effects of a number of the important compounds currently used in the manufacture of commercial welding electrodes.—*T. H. Hazlett, The Welding Journal, January 1957; Vol. 36, pp. 18-s – 22-s.*

Modern Steam Turbine Feed Systems

The relative efficiency of feed cycles for steam turbine installations under a given set of conditions can be obtained by comparing their cycle ratios, obtained by dividing the overall fuel rate by the fuel rate based on turbine non-bleed steam rate. This cycle ratio covers the magnitude of the auxiliary loads, the method of auxiliary drive, and the amount of regenerative heating employed while it is possible to analyse this ratio into feed heating gain and net auxiliary consumption if desired to compare the cycles more closely. When choosing a feed cycle it is also necessary to examine the first cost and to balance any additional cost and the charges thereon, together with additional maintenance, against the annual saving in fuel expenditure. This consideration tends to lessen the difference between the cycle ratios of the various circuits and in certain cases the saving in charges more than offsets the extra fuel bill. In particular, the cost of the turbo-generators is one of the principal items and the considerable saving from the use of the back-pressure type together with the good efficiency make this type economically attractive. The recent tendency has been to swing from complex four-heater cycles, to relatively cheap and simple cycles with only one heater. The resulting increase in the sum of fixed charges and fuel costs is small but minor improvements to these circuits could be made and give an appreciable saving for a small increase in outlay and with very little additional complication. There is still room for improvement in the steam consumption of auxiliary machinery and any such gain lessens the need for the complications of mixed pressure turbines and electro-feeders or geared pump drives. Equally good feed circuits can be devised with gas/air or steam/air heaters and the choice rests principally on the question of maintenance. With care in the choice of circuit it is possible to increase the feed temperature with steam/air heaters without loss from the overall economic aspect and with advantage to maintenance. Where the electric load is high it may be found difficult to fit in back-pressure turbo-alternators, in which case straight three- to four-heater circuits with gas/air heaters are suitable, or with two feed heaters and steam/air—preferably two-stage. For the lowest powers a single-cylinder main turbine with a gear-driven generator makes a simple and efficient installation of low cost.—*Paper by A. D. Bonny, read at a meeting of the North East Coast Institution of Engineers and Shipbuilders on 22nd February 1957.*

Brittle Fracture Initiation

A running crack will propagate at nominal stresses of 10,000lb. per sq. in. in steel plate. Nevertheless, structures of such steel operate at higher nominal stress. Furthermore, laboratory tests generally fail to initiate brittle fracture at nominal stresses below yield except by extreme cooling or impact loading. A strong barrier to the static initiation of brittle fracture thus exists. The object of the present investigation is to study the conditions under which this barrier is lowered. Until static laboratory tests reproduce fractures at the temperatures and nominal stresses encountered in service, brittle fracture will remain essentially unexplained. Moderate success has been achieved so far. Welded and unwelded notched steel plates with various prestrains were pulled at various temperatures. Transversely, prestrained plates with punched notches fractured consistently below yield under static loading. The fractures were as brittle as those found in service in the region of propagation and, far more important, also at the point of initiation.—*C. Mylonas, D. C. Drucker and L. Isberg, The Welding Journal, January 1957; Vol. 36, pp. 9-s – 17-s.*

Plastic Piping for Ships

One of the main problems associated with steel piping, particularly on board ship, is that of corrosion. With this problem in mind corrosion-resistant piping of a strong and light thermoplastic has been developed. This piping, known as Durapipe, is made in light, normal and heavy sizes of standard gauge up to six inches in diameter. One important feature of this new piping is its ability to withstand temperatures of up to 400 deg. F., which makes it suitable for low temperature steam lines. Its corrosive resistant qualities make it ideally suitable for deck lines carrying hot or cold water or, as previously mentioned, low temperature steam; and the light weight of this thermoplastic makes it extremely simple to handle. Other applications for which Durapipe could be used are ventilation ducting, fire lines, bilge lines, sanitary pipes, guard rails and probably many other uses which have not yet even been considered. The pipes are normally fabricated from one of three selected materials, depending upon the application and the material which it is desired to convey. One of the great advantages of Durapipe is that it can be easily installed. The material weighs about one-eighth of its equivalent in metal, and it can be joined so simply that a paint brush can be used and pipe threads dispensed with. This jointing process is extremely simple and no special training or equipment is required, since the solvent welding compound that is used is applied with a brush to the end of the pipe and the inside of the fitting. The pipe is then inserted, turned once or twice, or driven home in the case of the larger pipes, and the connexion is complete. For normal and heavy wall piping, however, threads may be used if required. As well as piping, the manufacturers are producing a range of sockets, tees, elbows, flanges, adaptors and unions. Development work on other fittings is also being carried out. Durapipe is now undergoing several tests on board various ships and it is certain that this material will solve many problems which have hitherto been of no small significance to the shipowner.—*The Shipping World, 20th March 1957; Vol. 136, p. 301.*

Hydrogen Peroxide Submarine

The submarine *Explorer* is in some respects the British equivalent of the American *Nautilus*. Both are capable of sustained high speeds under water without the necessity to draw air for combustion from the atmosphere. The difference in performance comes on the endurance side: the *Nautilus* is able to maintain her top speed for a virtually indefinite period, but the *Explorer* is limited by her fuel capacity, and her endurance is by comparison short. The hydrogen peroxide machinery for the *Explorer* and her sister *Excalibur* (both built by Vickers-Armstrongs, Ltd., Barrow) has been developed by the Admiralty from the German work done in this field during the war. The material is used in a highly concentrated form known as H.T.P. (high test peroxide), and in this form it is extremely dangerous unless correctly handled. However, the Admiralty's ideas have been proved in practice by the accident-free record of the *Explorer* during the year that she has been in commission. The hydrogen peroxide machinery operating cycle is fairly simple. H.T.P. is first decomposed in a catalyst chamber, where it turns into steam and oxygen and produces a great deal of heat. These pass to the combustion chamber, into which are injected Diesel fuel and further water, the latter to cool the combustion chamber and bring the outgoing gases to a suitable temperature for the turbine. These gases consist of carbon dioxide and steam, and after passing through the turbine the latter is condensed, and the former pumped out to dissolve in the sea. The fact that the Admiralty is preparing to build a nuclear submarine shows the H.T.P. propulsion has its limitations.—*The Shipping World, 27th March 1957; Vol. 136, p. 322.*

Compiled and published by the Institute of Marine Engineers

Patent Specifications

Joints for Hatch Cover

This invention relates to the cross joints between the separate sections of metal hatch covers and is particularly directed to the comparatively deep insulated covers employed in connexion with insulated holds. An object of the invention is to produce an air lock at the joint so as to seal the hatch effectively at the cross joint and reduce loss of refrigerating temperature by heat exchange between the external atmosphere and the air in the hold. A further object is to avoid wear and tear on the resilient packing usually employed in joints of this description. In Figs. 1 and 2 each of the cover sections (indicated generally by (a) and (b)) consists of a top metal

Fig.1.

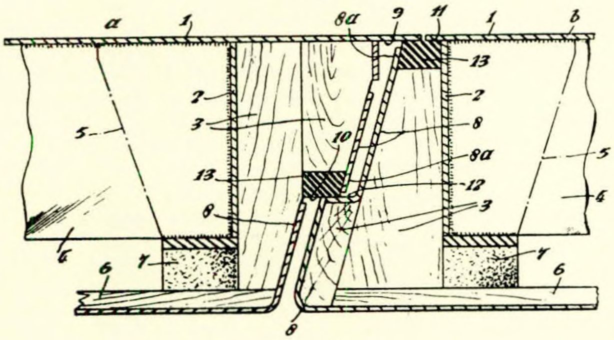


Fig.2.

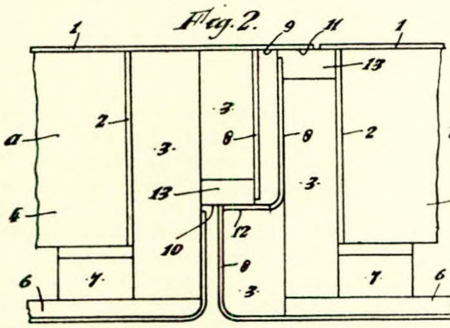
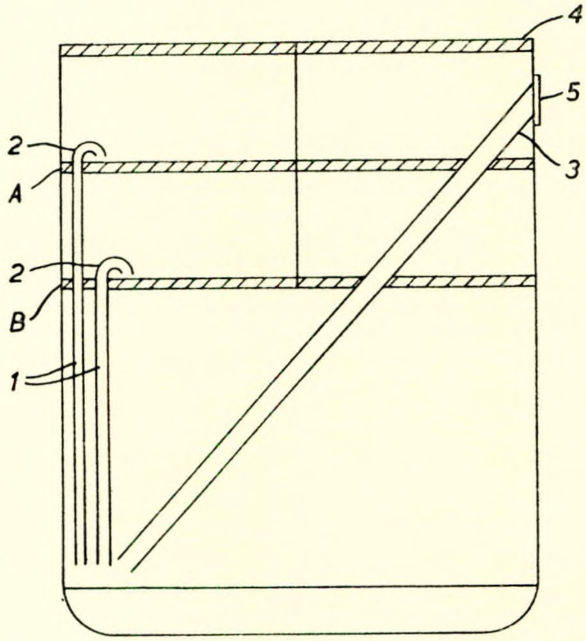


plate (1) with a welded right angular web (2) with wooden elements (3) secured in the angle between web and top to constitute the abutting ends of the sections. The numeral (4) refers to a central longitudinal girder, (5) are brackets spaced apart at a requisite distance across the sections, (6) are wooden plates, (7) are cork fillings, and (8) represent galvanized stiffening plates which form facings for the wooden parts. This construction makes up cover sections suitable for refrigerated hatches. The section (a) is hinged at its end remote from the abutting end illustrated so that it can be swung upwards clear of the section (b). The cover section (a) has two steps (9 and 10) facing downwardly, and the section (b) has two steps (11 and 12) facing upwardly and when the sections are closed together the steps 9 and 11 interfit and the steps 10 and 12 interfit. The pair of interfitting steps, 9, 11, are located at the upper surface of the sections and the pair, 10, 12, intermediately of the depth of the abutting ends. The metal stiffening plates (8) overlap the edge of their respective wooden elements (3) at 8a to form a channel in which is housed a rubber sealing strip (13), the strip on the section (b) at the upper location facing upwardly and that on the section (a) at the intermediate location facing downwardly. *—British Patent No. 771,169 issued to R. MacGregor and J. MacGregor. Complete specification published 27th March 1957.*

Drainage System for Ships

It is well known that if too great a volume of water collects in the upper deck regions of a ship there is a tendency for the ship to become top heavy and list or even capsize. Such circumstances have arisen where water introduced in fire fighting operations into a ship lying in dock has failed to drain away through the normal drainage channels and has eventually caused the ship to heel over and rest on its side on the dock bottom. It will also be apparent that in conditions



of violent storm a vessel might ship considerable quantities of water and if the normal surface water drainage arrangements are inadequate to the unusual circumstances or become blocked with floatage then the water will tend to accumulate and give rise to conditions of list or capsize. Referring to the drawing, the drainage ducts (1) descend directly from the upper decks (A and B) to the lowermost parts of the vessel, for example, over the bilge hatches and at the inlet of each duct on the upper deck level is provided a U-shaped return bend (2). Where the vessel is divided into separate port and starboard watertight sections at its lower levels, for instance, in the vicinity of the tunnel through which the propeller shaft extends, the drainage ducts (1) descend diagonally across the width of the vessel so that the water drained from the upper starboard section will be led to a lower port section and *vice versa,* thereby effecting an additional correcting influence on any tendency to list. The removal ducts (3) are connected from an inlet in the lowermost parts of the vessel on the port or starboard side diagonally across the width of the vessel to an outlet located just below the level of the main deck (4), access to the outlet being obtained through a watertight door or port in the outer plating of the vessel, the door being arranged to be opened from the outside. The removal ducts (3) are of appropriate cross section to permit the passage of suction hoses and are free from internal obstructions. The ducts (3) may either be used directly for the pumping out of water from the lower regions of the ship or indirectly as conduits for the lowering of an engine suction pipe down to the level of accumulated water. These ducts may also serve for the introduction of air into a sunken ship to restore buoyancy during salvage operations or for the introduction of non-combustible gas or foam in fire fighting operations. There is thus provided a water drainage system for ships which is less susceptible to blockage than conventional systems, which reduces any tendency to listing due to accumulations of water in the upper regions of a ship and which provide simple means for removing surplus water from inside the ship.—*British Patent No. 771,544, issued to E. Lakeman Trant. Complete specification published 3rd April 1957.*

Dead Light Fitting

This invention has for its object the provision of a dead light construction which enhances substantially its capacity for resisting pressures tending to distort it, and which incorporates means for supporting effectively the glass of a window or scuttle so that it will resist greater pressures than hitherto possible. According to the invention, a dead light is shaped so as to present a convex surface towards its outboard side, and it is provided on the outboard side with a pad or pads of rubber which, when the dead light is in closed position, will bear on the inner surface of the glass contained in the light frame to which the dead light is secured. The supporting and strengthening influence of the dead light may be applied to the whole exposed area of the glass, or a part of it, and, in one form of embodiment, the glass support includes a horizontal strip of rubber extending diametrically across the full width of the dead light. In conjunction with this a concentric ring of rubber may be fitted. The supports are located in recesses provided in webs formed on the outboard face of the dead light. The latter is strengthened by the provision of radial webs. Referring to the dead light (14) in Figs. 1 and 2, this is of a generally convex construction. Formed on the outboard side of the dead light is raised a circular web (14c), and in intersection with it are arranged two webs (14d), one being horizontal and the other vertical. The webs are recessed to receive a circular rubber pad (21) and diametrical pads (22 and 23). The circular pad (21) and the diametrical pads (22 and 23) are arranged, when the dead light is closed, to bear on the inner side of the glass (24) of the glass holder (13) to support it against wave impact.—*British Patent No. 773,241, issued to T. Utley. Complete specification published 24th April 1957.*

FIG. 1.

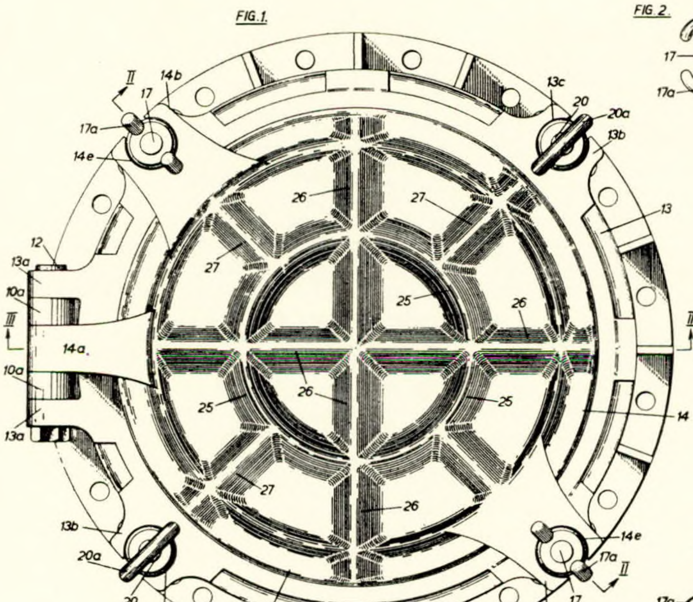
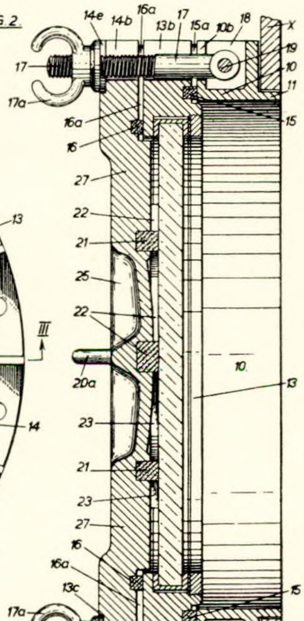


FIG 2.

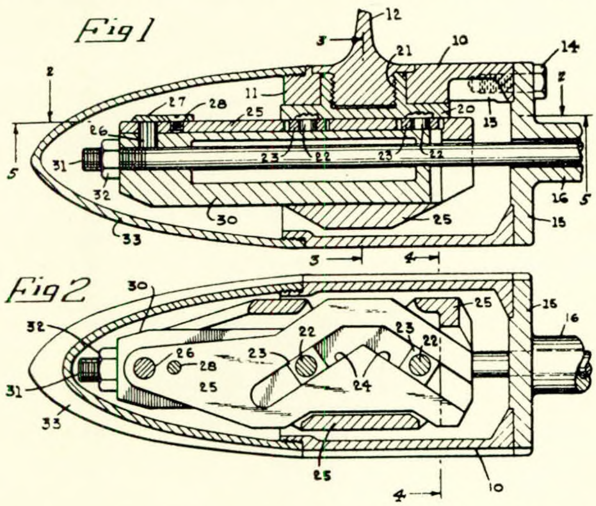


Ships' Holds with Feeders

It is known that the conveyance of certain bulk goods, and particularly grain, means that a certain number of conditions must be respected, and compels the shipowner scrupulously to comply with them. In particular, provision has to be made of means to prevent the unexpected shifting of the grain owing to the rolling and pitching of the ship which might risk its stability. Regulations now in force have led shipbuilders to provide over the centre of the hold a kind of compensating compartment, generally called a "feeder", with a capacity of about 2½ per cent to 8 per cent of the volume of the hold. These feeder bins or species of chimneys must be full and in free communication with the internal space of the hold in order to compensate for the vacuums that may possibly occur owing to the grain shifting. Moreover, it is incumbent on shipowners to provide a longitudinal bulkhead or shifting board placed inside the hold and extending for about one-third of its height from the ceiling. These various regulations and obligations mean that vessels intended to carry grain are not suitable for conveying materials other than bulk goods, unless relatively extensive and costly transformation work has been carried out. It will easily be seen that a compensating compartment frequently having a capacity of 70 to 80 tons cannot be easily taken down when it is situated in the hatchway space, and very speedily re-erected when the ship, after having carried packages, is due to take on a cargo of grain. The purpose of this invention is to obviate these various disadvantages, and relates to a method of arranging holds that enables a very speedy changeover to be made from the conditions required for conveying packages and those necessary for carrying bulk goods and *vice versa*. The holds arranged according to the invention are characterized in that the compensating compartments or bins generally called "feeders" assume, for example, the shape of semi-bins or semi-feeders, and are placed one at each end of the holds in question in the longitudinal direction of the ship, the hatchway space being entirely free. According to another feature of this invention, the volume of each one of the semi-bins or semi-feeders is practically equal to half the total volume of the feeder. According to still another feature of the invention, the hold comprises, for about one-third of its height, measured from its ceiling, longitudinal portions or shifting boards, fitted in such manner that they can be retracted by slewing, for example.—*British Patent No. 772,585 issued to P. A. Mege. Complete specification published 17th April 1957.*

Variable Pitch Propeller

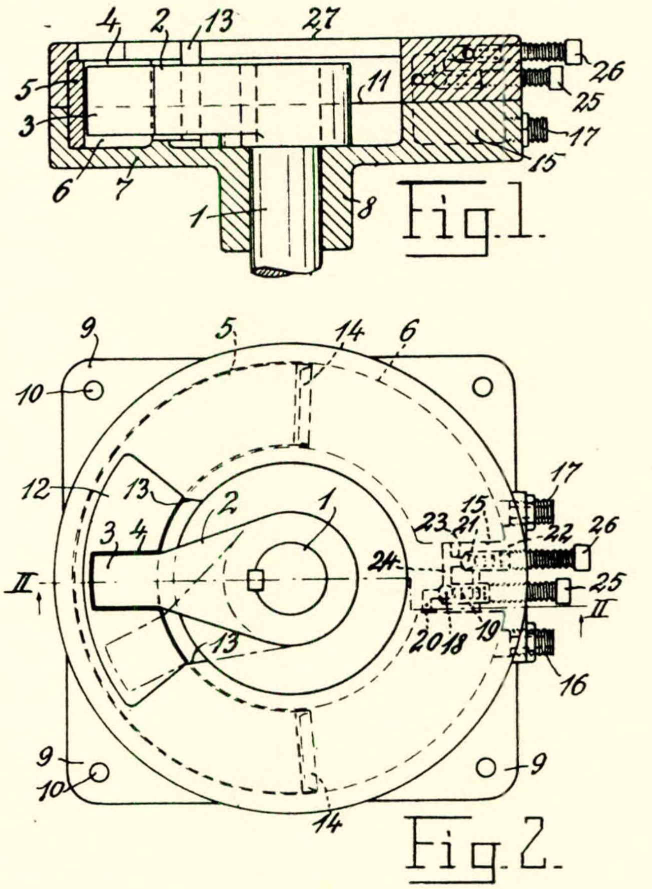
This invention relates to variable pitch propellers. Referring to Figs. 1 and 2, in operation axial movement of the control rod (31) and control member (30) will swing blades (12)



on their axes by the coaction of plates (25), grooves (24), sliding blocks (23) and pins (22). Since the torque applied to the blades by the axial movement of the control assembly is a couple, the loadings in the mechanism are half what they would be if the torque were a single turning moment. The use of sliding blocks (23) in combination with the location of pins (22) towards the ends of the longer diameter of each flange (20) provides a simple and sturdy mechanism of small hub diameter in which the pitch of the blades may be adjusted with minimum specific bearing loadings on the co-operating parts.—*British Patent No. 768,595, issued to A. Freroy. Complete specification published 20th February 1957.*

Hydraulic Steering Gear

This invention relates to a hydraulic steering apparatus for ships' rudders of the type in which the rudder head is connected with a piston arranged within an annular cylinder



whose walls are concentric with the rudder head. Referring to Figs. 1 and 2, to the rudder head (1) is keyed an arm (2), the so-called tiller, the exterior end (3) of which projects into a recess (4) in a semi-circular piston (5) which is arranged in the circular compartment of a cylinder (6). This cylinder is composed of two portions, a bottom portion (7) with a hub (8) for the rudder head (1) and a top portion (27). The cylinder is interrupted by a partition (15) containing valves. On one side of this partition a socket (16) is arranged and on the opposite side a socket (17). Through these sockets liquid under pressure, for instance oil, is introduced in such a manner that if oil is introduced at (16), it will press the semi-cylindrical piston (5) in clockwise direction together with the tiller (2, 3) for adjustment of the rudder. During this adjustment the oil from the opposite cylinder compartment will pass out through (17). The opposite movement takes place when the tiller is to be turned in the opposite direction. —*British Patent No. 773,880, issued to K. H. Tenfjord. Complete specification published 1st May 1957.*

Marine Engineering and Shipbuilding Abstracts

Volume XX, No. 8, August 1957

Large French Trawler

One of the largest trawlers yet to be built, the *Cap Fagnet III,* was launched recently from the At. et Ch. de la Seine Maritime, and this vessel, which has a fish hold capacity of 1,240 cu. m., is equipped with Diesel-electric propelling machinery of over 2,000 b.h.p. comprising three 750-b.h.p. Ruston Paxman engines each driving two generators, one for propulsion and one for operating the trawl winch or the auxiliaries. There are two 950-b.h.p. electric motors running at 850 r.p.m. driving a single propeller through reduction gearing. The three generators are all in operation when the ship is proceeding to the fishing grounds and two are sufficient when trawling. The economy thus effected should be considerable, as on an average fishing trip of 100-120 days' duration only sixteen days are spent on the passage. The length of the trawler is 245ft. 5in. overall, with a moulded breadth of 37ft. 9in., a maximum mean draught of 17ft. 7in. and a maximum draught astern of 19ft. 4in. The water ballast capacity is 230 tons and the cod liver oil tanks carry 90 tons. The bunkers total 535 tons.—*The Motor Ship, February 1957; Vol. 37, p. 473.*

Cooling Water for Diesel Engines

The U.S. Bureau of Ships recommends that all Diesel engines (above 500 revolutions per minute) that are cooled by fresh water shall operate within the following temperature ranges measured at or near the points of discharge of the lubricating oil and fresh water from the engine: —

	Preferred	*Minimum*	*Maximum*
Lubricating oil ...	165	140	180
Fresh water ...	160	140	170

Any operating temperature within this range is satisfactory. The preferred temperatures are mentioned only for guidance. Since the temperature gradient throughout the engine with these ranges is excellent, difficulties caused by incomplete combustion will be at a minimum. Keeping the engine jacket water temperature above the minimum specified temperature is very important since engines that are too cold will cause extremely high cylinder wall and piston ring wear. When the engines are operated at a temperature below the dew point of the water in the gases of combustion, the moisture will condense on the cylinder walls while the engine is running and will cause "cold corrosive" wear. Reports state that in some instances the cylinder and piston ring wear rates of an engine operating at 100 deg. F. may be ten times as great as that of the same engine operating under the same conditions at 160 deg. F. A number of reports have been received from ships operating in northern areas of high cylinder wear rates. In one instance, a wear rate of 0·042 of an inch in 1,500 hours of operation was recorded. The high wear rates are believed to be caused largely by low temperature operation. Lubricating oil temperatures should be maintained at 140 deg. F. or above. Low lubricating oil temperatures should not be used for the following reasons: (1) Water in the crankcase will condense, later forming sludge. (2) Oil does not properly wet the different moving surfaces, and does not provide adequate lubrication. (3) Cold oil is difficult to pump and an inadequate flow through the lubrication system may cause improper piston and engine cooling or starved bearings. When the lubricating oil and jacket water temperatures are thermostatically controlled, the preferred temperatures are impossible to maintain at all engine speeds. However, the preferred temperatures can be kept within the specified range if the thermostatic valves are set properly. If lubricating oil temperatures are controlled by a bulb in the oil line between the cooler and the engine inlet, the proper outlet temperature will be obtained with the valve set to keep the temperature at the bulb between 120 deg. F. and 140 deg. F. Where the jacket water temperature is controlled by a bulb located in the jacket water line leaving the engine, there should be no difficulty in keeping the temperature within the specified range under all load conditions. In the case of fresh water cooling of lubricating oil where both oil and water temperatures are controlled by a three-way thermostatic valve in the fresh water line, the lubricating oil temperature generally will be maintained at a higher level than that of the jacket water. Both temperatures should be kept within the desired range with very little difficulty if the thermostatic valve is properly set. The fresh water cooling temperatures for high-speed Diesel engines should not be confused with salt water cooling temperatures, which must be run below 130 deg. F. at the overboard discharge to prevent the deposition of solids from the salt water.—*B. I. Leefer, Bureau of Ships Journal, January 1957; Vol. 5, pp. 15-16.*

Larger Ships Built at Bruges

Larger ships are now being built at Bruges at the Chantiers Navals de Bruges, and recently the *Gertrud Bratt,* a vessel of 2,500 tons deadweight, was handed over to her owners, G. A. Bratt and Company, Gothenburg. The *Gertrud Bratt,* which has been constructed to the rules of Lloyd's Register of Shipping + 100 A.1 with ice reinforcement, has the following characteristics: —

Length overall	87·62 m.
Length b.p.	80·77 m.
Breadth	12·90 m.
Depth	7·70 m.
Draught	5·26 m.
Deadweight capacity ...	2,500 tons
Gross register	1,469·86 tons
Service speed	12 knots
Machinery	1,650 b.h.p. at 250 r.p.m.

The cargo handling equipment includes two 5 to 7·5-ton cranes with a radius of 13 m. and one of 3 to 5 tons with a radius of 17 m. The ship is equipped with radar and a gyro compass, also other navigating aids. The ten cabins for the crew are on the main deck, whilst the galley, the officers' messroom, that of the crew, also the recreation room, are situated on the shelter deck. The officers' smoke room is on the bridge deck. The vessel is propelled by a 1,650 b.h.p. Klockner Humboldt Deutz engine with eight cylinders, 420 mm. in diameter, the piston stroke being 660 mm. Current is supplied throughout the ship by two 150-kW and one 130-kW dynamos, driven by Deutz engines, and there is a 37-kW unit for port service. All the pumps and other accessory machinery in the engine room are electrically driven, and two Sharples centrifugal machines are installed. It is intended to build much larger ships at this yard.—*The Motor Ship, April 1957; Vol. 38, p. 36.*

First 15,000-b.h.p. Diesel-engined Supertanker Launched

The first of the numerous supertankers with 15,000-b.h.p. single-screw Diesel machinery was launched recently at the Innoshima yard of the Hitachi S.B. and E. Co., Ltd., for the Morita Kisen K.K. and named *Yuyo Maru No. 5.* It is to be equipped with the first B. and W.-type turbocharged engine of this power, with twelve cylinders, 760 mm. in diameter, and having a piston stroke of 1,600 mm. The vessel is 646ft. 2in. b.p. with a beam of 86ft. 7in. and a depth of 45ft. 11in., the deadweight being 33,500 tons. The launching was on 16th February and completion is anticipated in August. There are several supertankers of this size being built in Japan and an order has just been placed for a 32,800-ton 15-knot vessel to be equipped with a 13,000-b.h.p. Harima-Sulzer turbocharged engine. This ship is to be 629ft. 10in. in length with a beam of 87ft. and a depth of 45ft. 6in. It will be built for the Sanko Kisen K.K. by the Harima S.B. and E. Co. The keel is to be laid in December 1958; the launch will take place in April 1959, and the ship will be delivered at the end of June 1959.—*The Motor Ship, April 1957; Vol. 38, p. 40.*

Corrosion Protection During Fitting Out

Corrosion which arises during the "fitting-out" period may be of sufficient magnitude to demand instant attention at the acceptance docking, in the form of rivet and even plate renewals at the cost of the builder. Alternatively it may only amount to a roughening of the plates but this too is worthy of note when one bears in mind the increase in frictional resistances and the detrimental effect of this on speed, designed fuel consumption and ultimately operating costs. Bearing in mind the excellent results achieved in preventing corrosion of the hulls and in the cargo/ballast compartments of ships in service, through the use of cathodic protection, the extension of this method of combating corrosion to the fitting-out stage was, to a large extent, the logical development of a proved technique. However, like all new inventions and methods, when this was first employed some few years ago, certain practical difficulties had to be surmounted, in order that a number of shipyards could employ it as standard practice. The schemes must necessarily be of the simplest possible nature and involve the minimum of installation work. They must not impede the main work being carried out and must be capable of speedy installation and removal. Many of the large shipbuilding yards in this country are located on highly polluted rivers, the properties of which may vary considerably over the fitting-out period. Such variations, together with the effects of tidal water and the increase in the wetted surface of the hull as the fitting-out proceeds must be allowed for if full protection is to be provided throughout this period. Experience has shown that in the main these conditions can best be fulfilled by the use of sacrificial anode systems provided as part of a simple but essential survey and commissioning service. All the vessels must be considered individually, in accordance with normal cathodic protection practice. Important points are the area of the wetted surfaces, surface treatment, environment and, of course, the duration of the period of fitting-out. From this, a rough idea of the protective requirements can be gained in advance of launching. The next stage follows a few hours after launching, when an electrical survey of the vessel is conducted. This is mainly concerned with taking potential measurements between the hull and a silver/silver chloride (Ag/AgCl) reference electrode, using a potentiometer or high resistance voltmeter. These potential readings have a definite relationship to corrosion. For example, the potentials obtained between a marine Ag/AgCl reference electrode and the unprotected steelwork would normally be of the order of −0·6 of a volt, at which steel may corrode freely. Some considerable change, however, might be observed in the stern areas indicating the very common occurrence of electrolytic action between the hull and a propeller of dissimilar metal. A constantly varying potential at any point would usually disclose the presence of stray current, which frequently gives rise to corrosion and often originates from d.c. welding and crane equipment employed in the vicinity. The potential at which it is considered no corrosion will arise is generally recognized to be of the order of −0·78 of a volt, relevant to the Ag/AgCl reference electrode. The achievement of this potential, or a slightly better overall one, is the object of the protection scheme. Survey readings, properly interpreted, provide an indication of the amount of equipment which will be required to fulfil this purpose and its positioning, in order that even and effective current distribution is obtained. Within a short time of the survey, the protective anodes are suspended outboard from conducting cables terminating on small studs welded to the deck and, following the passage of a " settling down " period, representative potential readings are again taken to check the measure of protection. Any adjustments can be made following this check. Ballast resistances are often employed but the system is so simple that, in the event of the vessel moving to another fitting-out berth, the protective anodes can be lifted on board and re-positioned in the water in a matter of minutes. No attention is called for on the part of the shipyard, other than the following of advice in regard to the bonding of the vessel to the shore and the positioning and insulation of other vessels alongside. Periodic surveys during the fitting-out enable any subsequent adjustments to be made on account of the increasing draught.—*Corrosion Technology, January 1957; p. 24.*

Automatic Pilot for Ships

A Danish system of automatic steering, the Arkas automatic pilot, is being introduced to the British market by the

Decca Navigator Co., Ltd. This system, which has been developed by the Dansk Automatisk Ror-Kontrol A/S of Copenhagen, contrasts in a number of ways with the normal type of gyro pilot. In the first place it can be controlled either from the gyro compass or from a magnetic compass. This allows it to be used in ships where no gyro compass is fitted, and makes possible the continuance of automatic steering in gyro-fitted ships if the gyro compass should break down. Secondly, there is no unit of any considerable size on the bridge, so that it should be possible to fit the system without difficulty in vessels where bridge space is limited. The control system is electronic, and the main part of the equipment consists of a number of small units intended to be mounted on a convenient bulkhead. On the bridge itself there is the magnetic compass unit, which is certified as a steering compass and can be used as such in standard binnacles, a rudder indicator, and push-button controls for overriding the automatic control if necessary. The Arkas control is entirely separate from the ordinary bridge control of the steering gear, and thus offers a full alternative control. The connexion into the normal steering is made aft, the type of mechanism employed varying with the steering gear fitted in the ship. The system operates on 220-volts d.c., and so no motor alternator is needed, though a booster motor must be fitted in ships with 110 volts supply. A small amount of alternating current is in fact used in connexion with the drive from the magnetic compass (direct current would introduce compass errors), but this is supplied by an oscillator circuit. A course is set on the pilot by rotating either a bearing ring on the magnetic steering compass, or a circular graduated scale mounted above a gyro repeater. This gyro repeater is often mounted on the chart table, where it can also serve as the chartroom repeater. The magnetic compass operates on the Wheatstone Bridge principle: four electrodes positioned at the end of four lubber lines, together with two silver-plated segments on the compass card floating in an electrolyte, form the arms of the bridge. When working from the gyro compass a potentiometer is used, which is connected to the centre spindle of the gyro repeater. In either case two voltages are taken away to either side of a d.c. amplifier known as the sheering eliminator which forms the main unit of the system. If the ship is on course the two voltages applied to the amplifier are equal, and the whole system remains in balance. If the ship yaws, or if the course setting is altered, there is a difference between the two voltages applied to the amplifier and this is used to operate relays which initiate the necessary rudder movement. Re-setting is achieved by means of a unit known as the rudder translator, which is coupled to the rudder. As the rudder moves, this unit sends back electrical signals to the sheering eliminator which correct the original unbalance when the requisite amount of rudder angle has been applied. It is clear that by altering the signal fed back by the rudder translator the amount of rudder movement can be altered. In practice, matters are arranged so that when the ship is returning to its correct course the rudder reaches its amidships position before the ship is on course, and then continues on so that a little opposite rudder is applied before the rudder returns to the midships position. This checks the swing of the ship, and in service has proved most successful. As an example, it has been reported by the cargo liner *Manchester Spinner* that a 25 per cent reduction in helm movement was achieved by the use of the Arkas pilot, as compared with an experienced helmsman under similar conditions. Another control which is provided allows varying amounts of permanent weather helm to be applied.—*The Shipping World, 6th March 1957; Vol. 136, p. 265.*

Nuclear Propulsion of Merchant Ships

At the instigation of the Norwegian shipping industry, the two Norwegian Research Institutes, Institutt for Atomenergi at Kjeller and the Skipsteknisk Forskningsinstitutt, Trondheim, carried out a technical and economic study on merchant ship nuclear propulsion, and issued a report giving the conclusions which were reached. The object of the economic study was to determine the size and speed of the nuclear powered vessel which would give the maximum net return on invested capital and to compare this return with that of conventionally powered vessels. The study included only ships equipped with thermal heterogeneous boiling (heavy or light) water moderated and cooled reactors and no attempt was made to undertake similar studies for other reactor types. Moreover, the study was limited to oil tankers. The conclusion reached is that "the study seems to indicate that from a purely cost point of view nuclear powered tankers could, under the assumptions mentioned, operate with slight economic advantage when compared with conventional tankers". The report concludes with the comment that "the technical studies have revealed a number of special problems relating to nuclear propulsion, but none that appears insoluble". So far as hazards are concerned it is believed that "acceptable solutions can be found".—*The Motor Ship, April 1957; Vol. 38, pp. 18-19.*

Tramp with Turbocharged Engine

The m.s. *Apollon,* believed to be the highest-powered motor tramp ship yet built, has just been completed. Of about 11,050 tons d.w.c. as an open shelterdecker on a draught of around 27ft., and some 13,200 tons on a mean draught of 30ft., she is propelled by a 7,800-b.h.p. Sulzer Diesel engine. The *Apollon* is the first of three similar ships ordered from John Redhead and Sons, Ltd., South Shields, by the Athenian Shipping Co., S.A., and the Anax Shipping Co., S.A. (managers, The Faros Shipping Co., Ltd.). These vessels will all have a service speed of about 16 knots and will use heavy oil in the main engine. Completed as a closed shelterdecker and built under the special survey of Lloyd's Register for their highest class, the *Apollon* has the following principal characteristics:—

Length o.a. (approximately) ...	499ft. 3in.
Length b.p. on L.W.L. ...	465ft. 0in.
Breadth, moulded	63ft. 0in.
Breadth, extreme	63ft. 2½in.
Depth, moulded to 2nd deck...	31ft. 3in.
Height of shelter 'tweendeck ...	9ft. 0in.
Forecastle length (approximately)	46ft. 6in.

The main engine is the first turbocharged Sulzer engine to be installed in a tramp ship. In fact, it is only the second such engine yet built, although a great number of such units are now on order for all classes of ship and in powers up to 15,000 b.h.p. The engine in each case is a six-cylinder single-acting two-stroke unit with a bore of 760 mm. and a piston stroke of 1,500 mm. The full load output is 7,800 b.h.p., at which power the mechanical efficiency on test bed trials was 92 per cent, the combustion pressure 840lb. per sq. in. and the compression pressure 600lb. per sq. in. At 90 per cent full load the specific fuel consumption on fuel with a calorific value of 9,960 kcal/kg. was about 0·34lb. per b.h.p./hr. at which the scavenging air pressure was some 6lb. per sq. in., the exhaust gas temperature after the turbine 580 deg. F. and before, about 550 deg. F. It is intended that the engine will operate at about 90 per cent full load, i.e., at 7,050 b.h.p. at 115 r.p.m. with a mean effective pressure of 6.5 kg./cm^2, or 93lb. per sq. in. The exhaust gas after the turbine is further utilized in a Spanner boiler, 6ft. 6in. diameter with a height of 8ft. 6in. and having a steam generating capacity of 4,600 lb./hr. at 100lb. per sq. in. In addition, there is a Cochran oil fired boiler to produce 3,600 lb./hr. at the same pressure. —*The Motor Ship, April 1957; Vol. 38, pp. 16-17.*

Centreing Device for Drydocking Slips

In drydocking operations, the ship must be held over the centre line of the keel blocks so that the keel lands on the keel line pier blocks when the water is pumped down. After the ship is positioned in the flooded drydock, centreing lines are stretched across the dock at the bow and stern to guide the ship down. A float or plumb bob is commonly placed on the lines as a marker over the centre line of the dock. As the ship is lowered, the centreing line is let out at both ends to follow the drop in water level. Another method is to attach a buoyant marker from the bow or stern to a line running under a pulley fastened to a pier block on the dock floor. The other end of the line is passed up to the top of the dock. As the water recedes, the slack is taken up so that the line remains taut between the floating marker and the pulley while the marker stays on the surface. The marker forms an inverted plumb bob supported upward from the pulley; the force of gravity used in actual plumb bobs is replaced by the buoyancy of the marker. A device that combines these two methods has been invented at the Boston Naval Shipyard. It maintains the ship on the proper downward course as water is drawn from the dock so that the ship comes to rest in the centre of the keel line pier blocks. The device can easily be drawn out of the way as the ship is moved into place. It is not affected by turbulent weather conditions. The device can be adjusted quickly and accurately, and it will need no further adjustment. Fig. 1 shows the relative location of the centreing

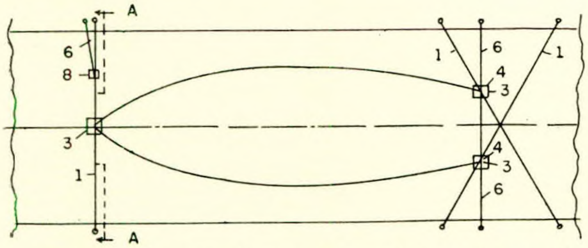


FIG. 1—*Relative location of ship and centreing device as viewed from above*

device when used with a ship that has a pointed bow and a squared stern. Fig. 2 is a port side view of the ship and centreing device. The ship is shown in position over the drydock floor centre line. The plumb bob line (1) is stretched across the drydock at the bow. The line has a plumb rod (2) hung on a ring fixed to it over the centre line. Similar plumb bob lines are stretched across the rear of the drydock; they have plumb rods adjacent to the corresponding corners at the

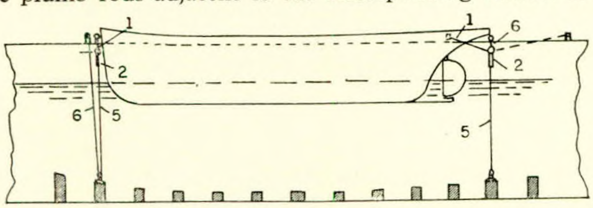


FIG. 2—*Port side view of ship and centreing device*

stern. (If the stern comes to a single point, only one line is used at the stern with one plumb rod at that point.) The pier block (3) on the centre line of the drydock floor has a mark on its top to designate the centre line. A pulley (4) is mounted so that its groove on the port side is tangential to the vertical plane of the centre line. The sight line (5) is passed around the pulley so that it runs to the starboard side of the pulley. It is fastened to a line (5) which is passed around a pulley (7) mounted on a block (8) at some distance from pier block (3). Then it passes up to the top of the drydock where it is fastened to keep the line taut. The other end of the sight line is connected to a buoy (9). To prevent the sight line from fouling while the ship is manœuvred into position, the buoy may be pulled down out of the way by pulling up the line (6). When the ship is in place, the line (6) is let out until the buoy comes to the surface. Then it is picked up by a worker in a punt, and the sight line is removed from the buoy and attached to the ring directly over the drydock centre line. The plumb bob line may be drawn up as desired so that the ring remains in the vertical plane of the centre line, and the line may also be drawn up as tight as desired and secured. The device is then ready for use not only in sighting any one point on the bow, but also for indicating any list that would bring the keel off centre even though one particular portion of the bow were kept centred during the ship lowering operation. In this way, the device keeps the keel properly aligned on the keel blocks even if the ship lists. The device is also more stable than previously used devices, and it needs no further adjustments once it is set up.—*Bureau of Ships Journal, April 1957; Vol. 5, pp. 38-40.*

National Physical Laboratory New Ship Hydrodynamics Laboratory

The paper begins with a brief outline of the development of experiment tank work, with particular reference to the development of Ship Division in the National Physical Laboratory. It then traces the development of the new Ship Hydrodynamics Laboratory which is under construction at Feltham. The history of this project is outlined, and various considerations leading to the final decision regarding the size of the tank and the speed of the carriage are described. Design considerations and general details of the construction of various items of the laboratory are given. These include the main waterway, the rails, the carriage, wave-makers, beaches, manœuvring and storage tanks, the building to house these facilities, and the associated workshops and offices. Reference is then made to the water tunnel with resorber which is also under construction on the same site, but housed in a separate building. The tunnel will also be used for cavitation studies

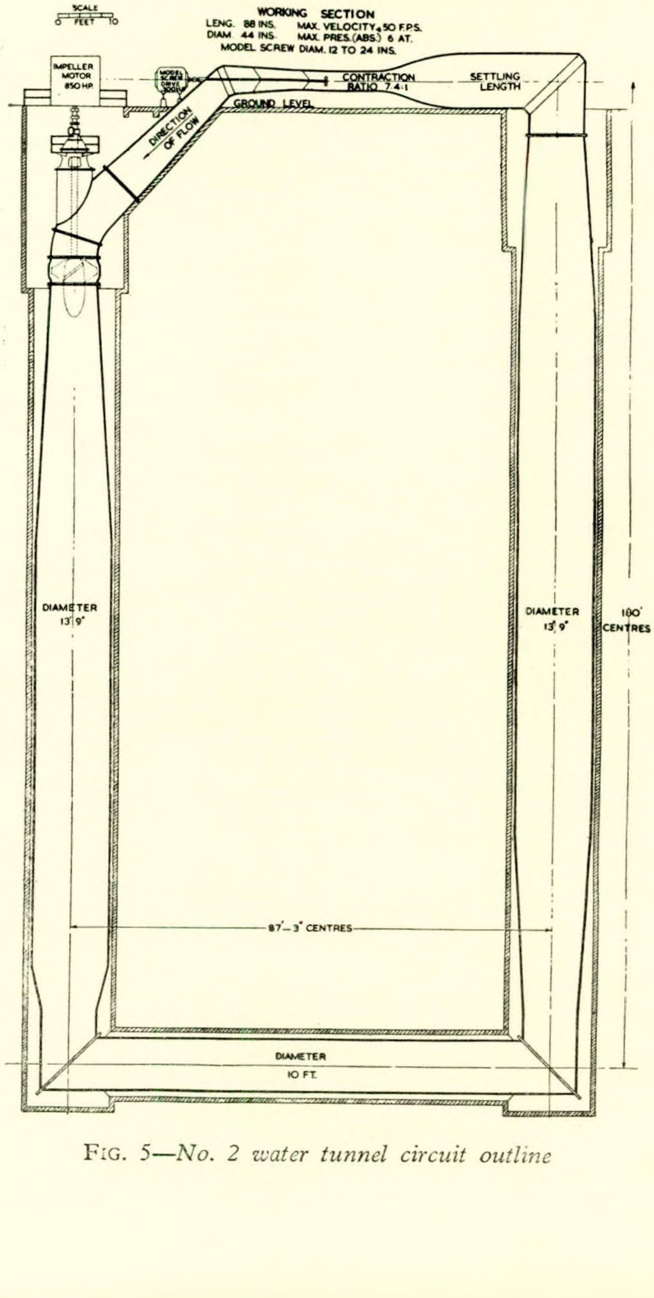


FIG. 5—*No. 2 water tunnel circuit outline*

on hydrofoils and other fixed bodies, and at least two additional test sections are contemplated, one of which will be a guided jet arrangement. Consideration was given to means of maintaining the water passing through the test section in a condition of controlled dissolved air content approaching saturation if necessary. The design of the tunnel includes a deep resorber circuit of large volume so that the water velocity through it is low. The general arrangement is shown in Fig. 5, from which it will be seen that the resorber circuit has a straightforward U-tube layout, the vertical limb being 180ft. deep from centre line of test section to the centre line of the bottom limb, and the horizontal limb 87ft. between centres of the vertical limbs. The impeller pump is mounted vertically as shown in Fig. 5 and this gives a pump head of 35ft. to the centre line of the test section. The pump is of axial flow type and has a diameter of 92in. It is provided with continuous speed and pitch controls to cover a range of flow rate from the maximum of 530 cu. secs. down to 50 cu. secs., the maximum power being 800 h.p., and the flow direction can be reversed by varying the pitch; an accuracy of 0·1 per cent of the set speed is specified down to 10 per cent of full speed. The model propeller shaft of high tensile steel is about 2·5-in. diameter and 23-ft. long. It is fitted in the downstream side of the circuit and carried in three bearings, two mounted on removable supports within the tunnel and the third in a boss at the 45-deg. bend. Arrangements have also been made to support a propeller shaft on the upstream side as for certain experiments this has advantages. The question of shaft whirling was given particular study and experiments were carried out in the Ship Division to determine the best positions for the bearings. The test screw is driven by an electric motor which is incorporated in the torque dynamometer. The normal maximum continuous rating is 300 h.p. at 20 revs. per sec., but a 25 per cent overload for a limited period is specified, as well as rotational speeds up to 55 revs. per sec. in both directions at lower torque. A short term control accuracy of the order of 0·1 per cent of set speed is specified over the main range of rotary speed. It is intended to test propellers ranging from 12-in. to 24-in. diameter at rotational speed from 15 revs. per sec. for large propellers to 55 revs. per sec. for small propellers. The corresponding torque range is about 2,000ft. lb. to 100ft. lb., and a maximum thrust of over 5,000lb. is allowed for. The speed of flow in the test section ranges from 10 to 50ft. per sec. and the pressure at the centre line can be varied from near zero to 6 atmospheres absolute. —*Paper by J. F. Allan, read at a meeting of the Institution of Naval Architects on 27th March 1957.*

German-built Cargo Vessel for Dutch Owners

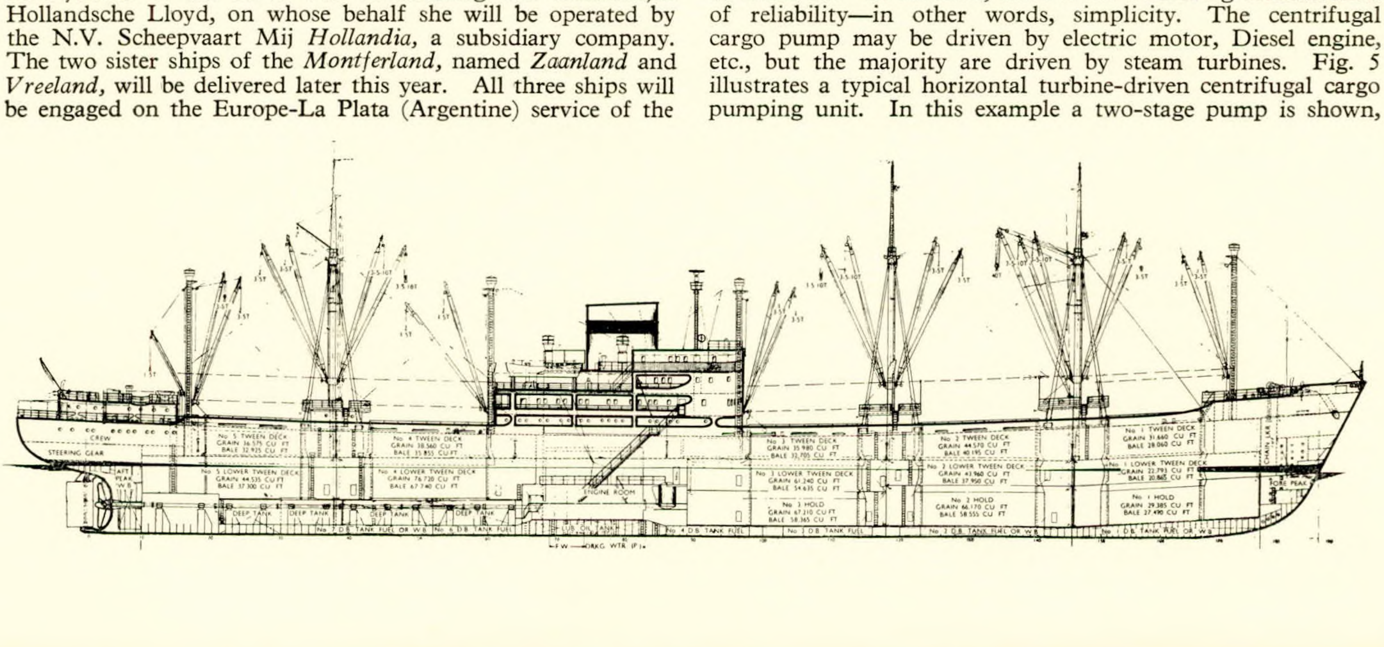
Blohm and Voss have recently delivered the m.s. *Montferland* to the N.V. Tot Voortzetting v.d. Koninklijke Hollandsche Lloyd, on whose behalf she will be operated by the N.V. Scheepvaart Mij *Hollandia,* a subsidiary company. The two sister ships of the *Montferland,* named *Zaanland* and *Vreeland,* will be delivered later this year. All three ships will be engaged on the Europe-La Plata (Argentine) service of the owners and have been built specifically for this purpose. The main particulars of the *Montferland* are:—

Length overall	513ft. 8¾in.
Length b.p.	470ft. 0in.
Beam, moulded	64ft. 6in.
Depth to shelterdeck ...	38ft. 6in.
Depth to freeboard deck ...	30ft. 0in.
Depth to 3rd deck	17ft. 6⅝in.
Deadweight capacity ...	10,350 tons
Corresponding draught (summer)	26ft. 1¼in.
Gross register	6,875 tons
Net register	3,891 tons

The *Montferland* has been built in accordance with Lloyd's Register class + 100 A1, her hull being almost entirely riveted. There are five cargo holds and three continuous decks, the total hold capacity being 590,200 cu. ft. (bale) and 656,290 cu. ft. (grain). These figures include a refrigerated cargo hold capacity of 7,700 cu. ft. Cargo handling equipment includes fourteen 5-ton derricks, six 10-ton derricks, a 1½-ton derrick and a 40-ton derrick located at No. 4 hold but designed to be erected at either the forward or the after hatches. The refrigerating plant situated in the upper 'tweendecks is capable of cooling down the relative cold spaces to −12 deg. C. Tanks are provided for the carriage of vegetable oil to a total of 26,315 cu. ft. All the hatches on the weather deck are provided with wooden covers. A 7,800-b.h.p. turbocharged seven-cylinder M.A.N. engine is installed. It has a cylinder bore of 780 mm., a stroke of 1,400 mm. and develops 7,800 b.h.p. at 115 r.p.m. At this power the contract speed is 16·75 knots and this was achieved on trials. Arrangements have been made for the engine to operate on heavy oil.—*The Motor Ship, March 1957; Vol. 37, pp. 514-515.*

Centrifugal Cargo Pumps

Centrifugal cargo pumps in general use today are of the non-self priming type, used in conjunction with positive displacement stripping pumps. The considerations which preclude the use of self-priming pumps are basically those of safety and reliability. For example, the extraction " air " pump would be handling explosive vapour, which could not be discharged freely to atmosphere, and would have to be pumped into the main cargo discharge line, against the full discharge pressure. To be effective, therefore, a piston or lobe type air pump would be necessary. There would then be the risk of explosion or fire if this type of pump were to overheat if running dry for any appreciable period. Also, automatic float-operated valves, as used in the conventional self-priming pump, would not be suitable when dealing with heavy oil and would especially be vulnerable to the grit and sludge content of crude oil. In addition, there is the overriding consideration of reliability—in other words, simplicity. The centrifugal cargo pump may be driven by electric motor, Diesel engine, etc., but the majority are driven by steam turbines. Fig. 5 illustrates a typical horizontal turbine-driven centrifugal cargo pumping unit. In this example a two-stage pump is shown,



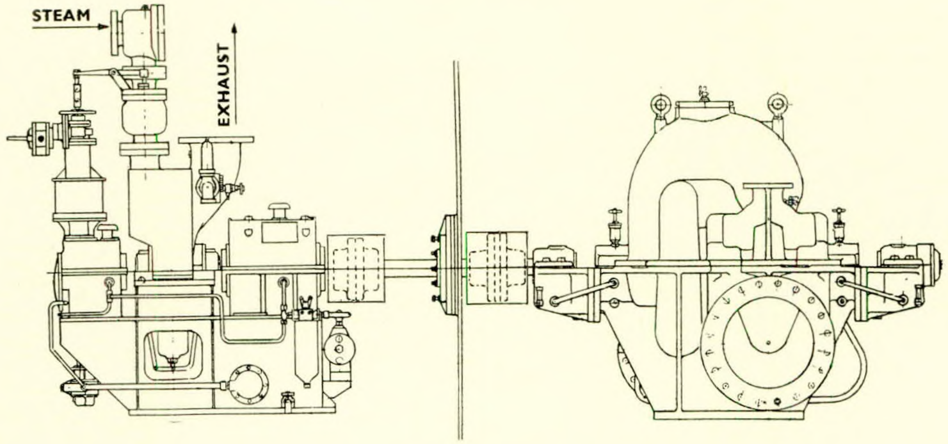
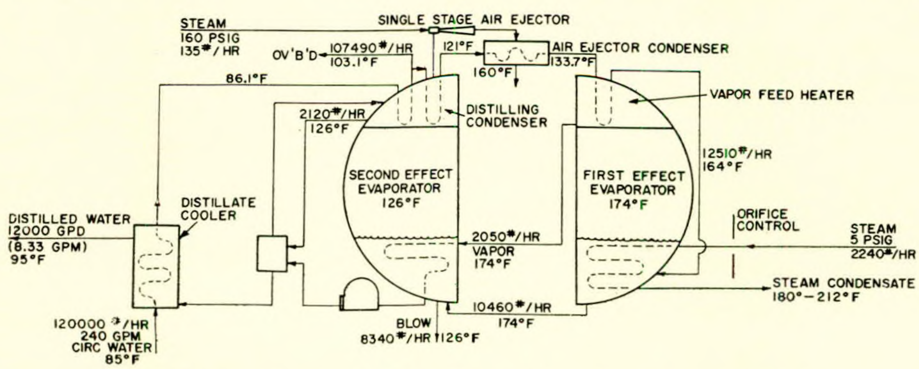


FIG. 5—*Horizontal geared turbine driven two-stage centrifugal pump*

although single stage pumps are also used. This particular pump has a double-entry first stage impeller for optimum suction performance, and single entry second stage impeller. The stuffing boxes can be arranged for mechanical seals, or for soft packing, and are designed so that both stuffing boxes are at half interstage pressure, to prevent ingress of air. Bearings are of the white metal lined journal type, with oil ring lubrication, and one ball thrust bearing. Stuffing boxes and bearing housings are water jacketed. Ball and roller bearings throughout are sometimes used but have been known to suffer from "Brinelling" due to the vibrations of the ship when under way. It is usual practice to locate the turbine in the engine room, to use to best advantage the steam pressures and temperatures available, and, equally important, to exhaust to the highest possible vacuum. The pump is therefore driven by a "jack" shaft passing through a gas-tight seal in the bulkhead. The turbine itself is of the single pressure stage velocity compounded type, with built-in reduction gearing. The bearings are the steel backed white metal type with forced lubrication throughout. A built-in oil filter, and an oil cooler, are standard fittings. The shaft glands are of the carbon segment type, with sealing steam provided by a reducing valve. An adjustable overspeed trip device is provided to shut down the turbine whenever the speed exceeds a predetermined figure. This is so arranged that the main steam stop valve has to be closed manually to re-set the trip mechanism before the turbine can be restarted. Also connected to the same mechanism is a low lubrication oil pressure trip, and high exhaust pressure trip. The usual drain valves, overload nozzle valve, casing relief valve, pressure gauges, etc., are fitted. Also included as a standard fitting is a hydraulic governor to keep speed variation to a minimum between full load and no load. This governor is arranged for hand control, and can be regulated down to 33 per cent full speed. The hand control regulator is arranged for remote positioning and can be erected on the upper deck. A mechanical tachometer is fitted to the turbine, also a remote reading electrical tachometer, to be located near the speed regulator. In some installations the hydraulic governor is omitted altogether, the speed of the turbine being regulated by a normal type steam stop valve operated from the upper deck through conventional extension linkage. Whichever type is used, some method of speed control is necessary, so that the pumping set may be slowed down when the level of oil in the cargo tank falls, to prevent the early formation of a vortex, which would result in the pump losing its suction. Thus the main cargo pumps can be made to deal with the maximum quantity of cargo, leaving less for the smaller capacity stripping pumps.—*P. J. Humphreys, Tanker Times, March 1957; Vol. 3, pp. 276-282.*

Flash-type Evaporators

Basically, there are three types of sea water distilling units, each of which has its own distinct advantages: 1, Submerged tube; 2, Flash type; 3, Vapour compression. For shipboard use vapour-compression distillation units have found their greatest application in Diesel driven ships. The submerged and flash types are installed usually in steam driven ships. Submerged-tube distilling plants have taken several forms depending on the particular manufacturer but, in general, the operating cycle is the same in each form and is well known to designers, operators, and shipbuilders. The flash-type distillation unit for shipboard use has entered the picture more recently and, thus far, has found its widest application in naval ships. Flash-type units have been furnished in capacities ranging from 8,000 gal. per day distillate output to 50,000 gal. per day distillate output, and in numbers of stages (effects) from 2 to 5. A distinctive feature of flash-type distilling units is that there are no evaporator bundles. Each stage of a flash-type unit will consist of a method for introducing heated sea water into the bottom of the shell, a method of achieving a



Typical heat balance—12,000 gal. per day, two-stage, flash-built unit pressure drop in the heated water to cause partial flashing to occur, a means for causing all the excess sea water to overflow automatically and proceed to the next stage, vapour separators, and a built-in vapour condenser. In the flash-type unit the ratio of feed to distillate may be 15 to 1, or greater. The output from a flash-type unit depends only upon the quantity of feed and the total flashing range. In the flash-plant cycle, the required amount of feed passes successively through the tubes of the stage condensers from the lowest to the highest temperature stage. It then passes through the tubes of an air-ejector condenser, and then through the tubes of a salt water heater. In passing through the tubes of the stage condensers the feed water is heated as it condenses the flashed vapour in each stage. As it passes through the salt water heater, the feed is further heated to approximately 170 deg. F. The heated sea water is then introduced into the first-stage compartment, in which the pressure is maintained at a value lower than that corresponding to the saturation pressure of water at 170 deg. F. A portion of the hot sea water will flash into vapour and will rise through the vapour separators to be condensed as it passes over the tubes of the stage condenser. All the excess feed will overflow automatically and pass directly to the next lower pressure stage, where an additional portion will flash into vapour. Several features of the flash-type unit should be noted; all sea water flow is constrained to the inside of tubes, there is no vapour piping, operation is completely automatic, and heat is not being added to the sea water at an evaporation surface. The flash-type plant is essentially a constant capacity device. Since scale formation is, in general, limited to the salt water heater, the capacity will remain constant as long as a constant supply of feed is pumped through the unit. The condensers are generally designed for a sea water velocity of 6ft. per second and the pumps selected accordingly. If a large overload capability is desired, the feed pump must be selected to pump additional feed in direct proportion to the overload capacity and with its discharge head increased in proportion to the velocity to the 1·8 power. The flash plant requires larger pumps than the submerged-tube unit, due to the greater quantities of feed and brine to be handled. The total b.h.p. required for flash plant pumps may be 2½ to 3 times that for a submerged-tube unit of the same capacity.—*W. A. Gardner, Marine Engineering/Log, April 1957; Vol. 62, pp. 78-80.*

Main Characteristics of Forced Recirculation Boilers

The forced recirculation boiler resembles the natural circulation boiler, and has a steam and water drum for the separation of the steam from the water in circulation. Fig. 1 is a schematic arrangement of the circuits. The main feature is the recirculating pump. In order to circulate through each tube element the amount of water corresponding to its heat absorption, controlled distribution of the water is provided. Controlled distribution by means of orifices or nozzles at the tube inlets was introduced by the La Mont system. Nearly all forced recirculation boilers work on this system, which, as will be mentioned later, has more recently been applied to forced-flow boilers. Due to the additional flow resistance at the tube inlet, stable flow is imposed independent of the heat absorption of the tube elements. This stable flow is established before the firing is brought into operation. It is, therefore, possible to raise steam in these boilers—and to a certain degree in forced-flow boilers also—as quickly as the firing permits. This is not the case in natural circulation boilers. The recirculating pump safeguards circulation at all loads, right down to minimum load, maintaining equal temperatures for all pressure parts, thus eliminating heat stresses. In the heated tube elements of large steam generators, a circulation factor of 3 to 4 is maintained, the circulation factor being the ratio of circulated-water weight to weight of steam generated. This results in a smaller steam and water drum than that required in natural circulation boilers, which have a higher circulation factor, and in which circulation is less steady, resulting in what is known as pulsations. Furthermore, there is a pressure difference available in forced recirculation boilers, by which good separation of the steam from the water is obtained in cyclone separators arranged inside or outside the drum. Another advantage of forced recirculation boilers is that control according to the water level in their drums is more simple than the regulation of forced flow boilers, in particular, if with large units several circuits in parallel have to be regulated separately. Manually operated regulation is still possible for forced recirculation boilers. The circulating pump can be used during starting-up for cooling the superheater and economizers; natural circulation boilers have recently been equipped with a special circulating pump for this purpose.—*Engineering and Boiler House Review, May 1957; Vol. 72, pp. 159-163.*

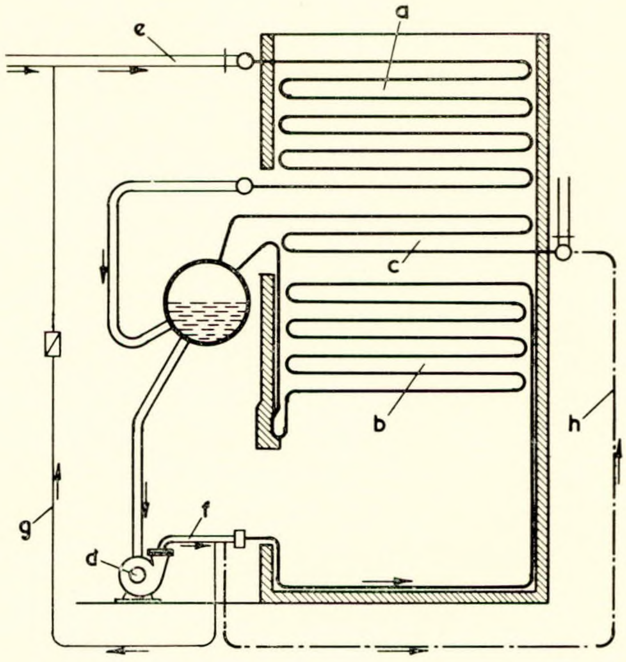
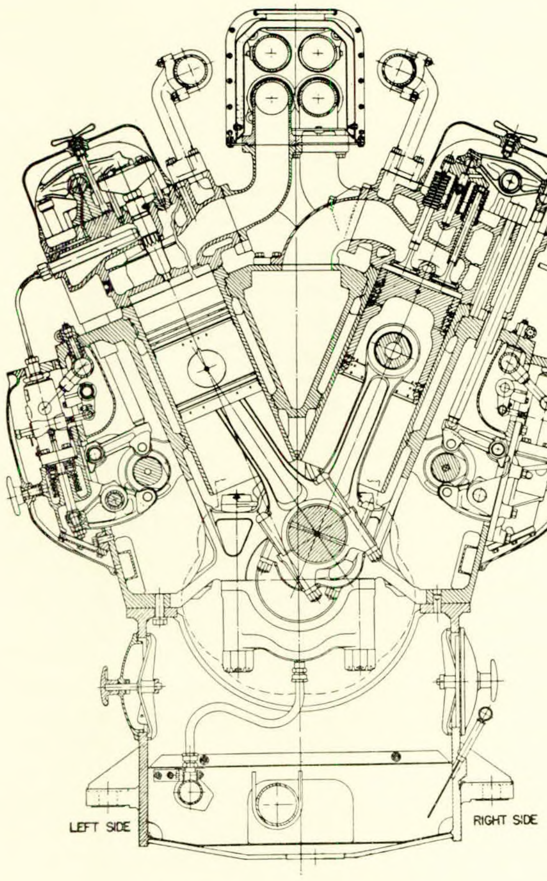


FIG. 1—*Diagrammatic arrangement of a La Mont type boiler*

a—Economiser. b—Evaporator. c—Superheater. d—Circulating pump. e—Feed inlet. f—Pump discharge. g—Connexion for cooling economizer when starting up or on part load. h—Connexion for cooling superheater, when starting up.

High-powered American Harbour Tugs

The first of two new craft which will rank as the most powerful harbour tugs in the United States was commissioned by the Curtis Bay Towing Company, Baltimore. The design provides for eventual operation with large ships, such as the ore carriers and supertankers now being built and for possible service on the high seas. The *Kings Point* and the *Fells Point* also fit the qualifications of a hawser tug for use in towing operations, by providing a large afterdeck with no obstructions. The all-welded hulls are of standard 105-ft. Equity design, with modifications such as a slightly higher wheelhouse and broader beam. The vessels were constructed for the Curtis Bay Company by the Equitable Equipment Company of New Orleans, La. They have a 27-ft. beam, a moulded depth of approximately 14ft. 8in., and a full-load draft of 12ft. 10in. They are each powered by a single ALCO-type 251 16-cylinder turbocharged Diesel engine rated at 2,100 b.h.p. at 1,000 r.p.m. for continuous towing duty. For intermittent operation, 2,400 h.p. is available. The engine has an overall length of 31ft. and weighs approximately 44,000lb., dry. ALCO introduced this engine in February 1956. It has a 9-in. bore, 10½-in. stroke and a fabricated bedplate and block. The air manifold is formed by the Vee



Section of ALCO 251 engine

within the block and a high efficiency turbocharger of ALCO's own design and manufacture is used. The fuel system on the new engine is totally enclosed and segregated, eliminating the danger of diluting the lubricating oil by fuel. The crankshaft bearing surfaces have no oil grooves, increasing the oil film thickness more than 100 per cent as compared with bearings with grooved surfaces. The forged aluminium pistons are equipped with Ni-Resist (nickel alloy) inserts in the top ring groove, and have replaceable ring carriers. The engine has a cradle bearing and a stub shaft to connect through a Falk Air-flex coupling to the Hindmarch-De Laval reverse reduction gear. Westinghouse Air Brake pneumatic controls are employed. The drive to the four-blade non-conventional design bronze propeller is provided from the ALCO 251 engine through the reverse reduction gear unit with an input speed of 1,000 r.p.m. and propeller shaft speed of 250 r.p.m. This reduction gear is one of the first American-made units to be installed, although several such gears manufactured by Modern Wheel Drive, Ltd., have previously been supplied for American river craft from Britain.—*The Marine Engineer and Naval Architect, April 1957; Vol. 80, p. 143.*

Norwegian Tanker of Unusual Design

Harland and Wolff, Ltd., recently completed at their Belfast shipyard the 36,000-ton tanker *Storfonn* which, after successful trials, was handed over to her owners, Messrs. Sigval Bergesen and Associated Companies, Stavanger, Norway. She is the third tanker which Harland and Wolff have built for these owners—the previous two being the 24,636-ton motorship *Dalfonn* in 1951, and the 29,650-ton *Solfonn* delivered last year—and is the largest tanker yet built in the United Kingdom for Norwegian account. The new tanker and the *Solfonn* have been built on a principle involving a new proportion between length and beam. Both tankers have built-in water ballast tanks above the wing tanks which, apart from providing increased structural strength, obviates the necessity for water ballast to be carried in cargo tanks and thereby eliminates a major cause of corrosion. A single screw steam turbine-driven ship, the *Storfonn's* principal particulars are as follows:—

Length overall	660ft. 0in.
Length b.p.	640ft. 0in.
Breadth moulded	86ft. 0in.
Depth moulded to trunk deck	51ft. 6in.
Deadweight	36,000 tons

Built under special survey of Det Norske Veritas to Class I.A.I. " Carrying Petroleum in Bulk " and to the requirements of the Norwegian Ship Control, the vessel is of trunk deck type with 'tweendeck water ballast tanks, propelling machinery aft, poop and bridge connected by fore and aft gangway, forecastle deck, promenade deck and boat deck aft, and lower bridge, upper bridge and navigating bridge deck amidships. She has a raked stem, a cruiser stern, stump fore mast, signal mast amidships and single funnel of distinctive design. The hull is divided by bulkheads into thirty cargo oil-carrying compartments and ten 'tweendeck ballast tanks. There is one main cargo pump room and a forehold pump room, two cofferdams, a gastight cargo hold forward and under this space four deep tanks for oil fuel or water ballast. Oil fuel is also carried in deep tanks forward of the engine room and in double bottom tanks under it, and water ballast in the forward and after peak tanks. A tank at the after end of the boiler room is arranged for the carriage of fresh water and boiler feed water, which is also carried in the double bottom. Modern equipment for efficient working is installed and includes steam driven cargo oil pumps, winches, windlass and capstans. The steering gear is of electric hydraulic type, controlled from the wheelhouse by telemotor and by mechanical control from the boat deck aft. Lifesaving appliances include four aluminium lifeboats, two of which are fitted with a motor, and the davits are of gravity type with wire falls and boat winches. Propelling machinery consists of a single shaft arrangement of compound, double reduction geared turbines, made by Harland and Wolff, Ltd., and is capable of developing 12,500 service s.h.p. ahead at a propeller speed of 112 r.p.m. Impulse blading is fitted for the h.p. ahead and the h.p. and l.p. astern turbines, while the l.p. ahead turbine is of the double flow reaction type. The h.p. astern turbine is arranged at the forward end of the h.p. ahead turbine and the l.p. astern turbine is housed in the l.p. ahead casing. Gears are of the double reduction articulated type and the gear case is of fabricated steel. The main condenser is slung underneath the l.p. turbine casing and is supported from the tank top by springs. The condenser is of the regenerative type and maintains a vacuum of 28½in. Hg. when working under service conditions. The propeller is of built-up design with four stainless steel blades and a cast steel boss. The thrust block is of the Michell type and is fitted in the engine room just abaft the gear case. There are two oil-fired Babcock and Wilcox integral furnace type watertube boilers made by Harland and Wolff, complete with superheaters and air heaters and the necessary fans for forced draught. A special feature is the automatic combustion control. The steam pressure is 450lb. per sq. in. gauge with a steam temperature of 750 degrees. The boilers are situated above the after end of the engine room, the boiler room being separated from the engine room casing by a screen bulkhead.—*Tanker Times, March 1957; Vol. 3, pp. 267-269.*

Hydrocone Construction Progress

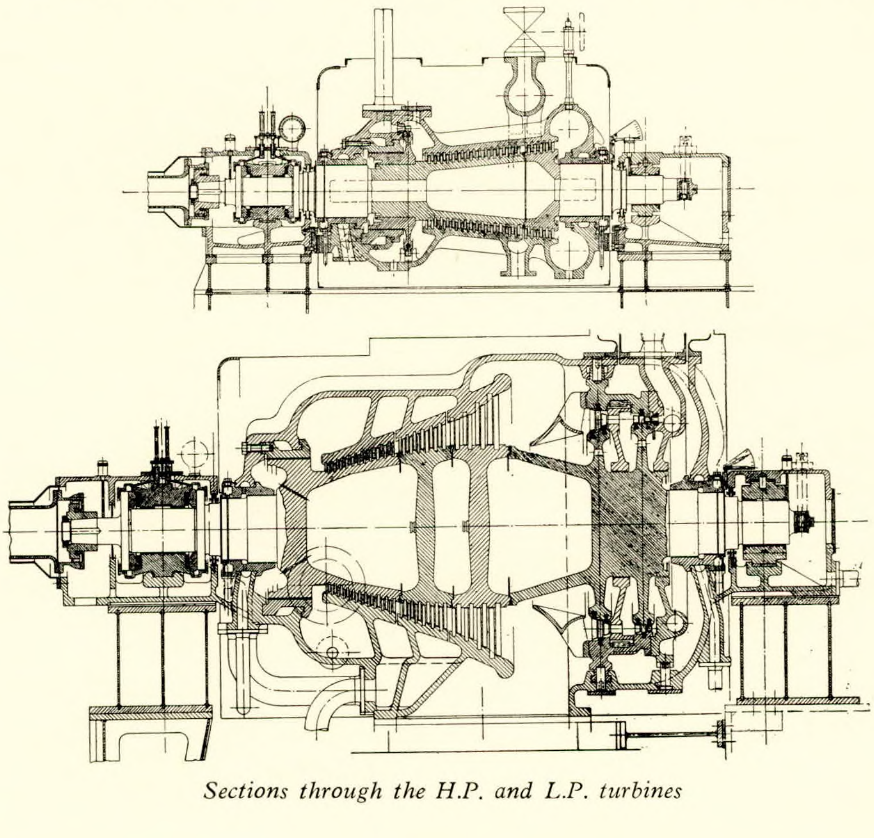
Originally, the Hydroconic method of construction was conceived as an attempt to produce comparatively efficient forms of fully developable type, anticipating that some slight sacrifice of efficiency might have to be accepted. The fully straight-line type of form, as exemplified by the *Tid* tug, was considered; but reference to the literature showed that this type of form was inherently extremely sensitive to the run and position of the chine-lines, especially forward. The efficient results achieved in, for instance, the case of the *Tid* tug have been stated to be the result of extensive tank testing of this particular form, and it was desired that efficient results should be obtainable, using the basic design principles for normal craft, without the necessity to run a tank test every time a vessel was constructed. The obvious way to do this was to eliminate the chine where necessary, and, in fact, the Hydroconic form—whatever the type and size of vessel in question—has, in all respects, a normal ship-shape bow, which nevertheless remains fully developable. The method was first applied to the 42-ft. tug *Gannet,* ordered by Luke Thomas and Co., Ltd., of Aden. The design approach to tugs was to obtain optimum operating conditions for the propeller. This was achieved by providing an easy flow to the propeller, by a run, giving a low thrust deduction, of wide flat sections over the screw to prevent air drawing, etc., and to accept the resultant shift forward of the L.C.B. by shaping the stern to prevent squatting at the higher speed/length ratios. The *Gannet* entered service in October 1953 and proved an immediate success. The *Jaycee* was the first Hydroconic tug to which the B.C.P. bulb rudder was fitted. This bulb rudder is an interesting hydrodynamic development, which achieves high steering efficiency and a constant degree of balance with respect to helm angle, and gives approximately 10 per cent increased thrust by extracting energy from the rotative component of the wake. It is, withal, extremely robust and economical to manufacture, and contributed to the very good results for the *Jaycee,* where a bollard pull of over 1·6 tons per 100 s.h.p. was achieved on carefully conducted trials. The *Guillemot* class, 95ft. b.p., is of a size eminently suitable for handling large tankers, etc., and two *Guillemot*-class tugs are on order for this purpose. The ship is of sufficient size to be suitable for ocean going work, if required, and is, in fact, a smaller edition of the *King Penguin* class, of which the *Sydney Cove* is the only example to date. The *Sydney Cove,* which is the largest Hydroconic tug so far built, operates at Sydney, N.S.W.—*The Shipbuilder and Marine Engine-Builder, February 1957; Vol. 64, pp. 119-121.*

Turbines for Tankers

A group of medium-sized turbine tankers has been delivered during the past year by the Bremen yard of the A.G. Weser. These ships were the *Atlantic Viscount, Viscountess, Marquess* and *Marchioness,* owned by Ocean Tankers, Ltd., one of the Livanos group of companies. All these ships are propelled by Brown Boveri main turbines of the same type, having design particulars as follows:—

Normal power	7,000 s.h.p.
Maximum power	7,600 s.h.p.
Maximum astern power ...	4,560 s.h.p.
Steam pressure at throttle ...	465lb. per sq. in.
Steam temperature at throttle	750 deg. F.
Shaft r.p.m., normal... ...	100
H.P. turbine r.p.m., normal...	5,360
L.P. turbine r.p.m., normal...	3,540

The general arrangement of the plant follows orthodox practice for cross-compound sets, but the owner required 60 per cent astern power and for this reason a special h.p. astern turbine has been arranged at the after end of the h.p. pinion shaft. The h.p. ahead turbine consists of a two-part welded rotor with shrunk-on and lip-welded impulse wheel carrying one single-row impulse stage and several reaction stages. The blade tips of the latter are backed off to reduce the effects of any slight contact with the casing. The gland packings and the labyrinth glands of the balance pistons are of usual B.B.C. type, designed to prevent bent shafts due to local heating of the rotor in the event of accidental rubbing. The h.p. astern turbine consists of a two-row Curtis wheel and is disposed at the after end of the h.p. ahead pinion. When running ahead, this wheel runs in a vacuum. The l.p. turbine is constructed in a similar manner to the h.p. ahead turbine. The l.p. astern blading in the forward part of the turbine consists of two double-row Curtis wheels arranged in series. Double-reduction, double-helical gearing with primary ratios of 8·13 to 1 for the h.p. turbine, 5·37 to 1 for the l.p. turbine and 6·59 to 1 for secondary trains is employed. The second-stage pinions are of three-bearing type. The motor driven turning gear is fitted at the rear end of the l.p. pinion shaft. The turbines are connected to their respective pinions by means of toothed couplings. The intermediate and main wheels are of cast iron with shrunk-on forged steel rims. The turbines are controlled by the B.B.C. single-handwheel hydraulic servo system. All ahead speed steps are obtained by



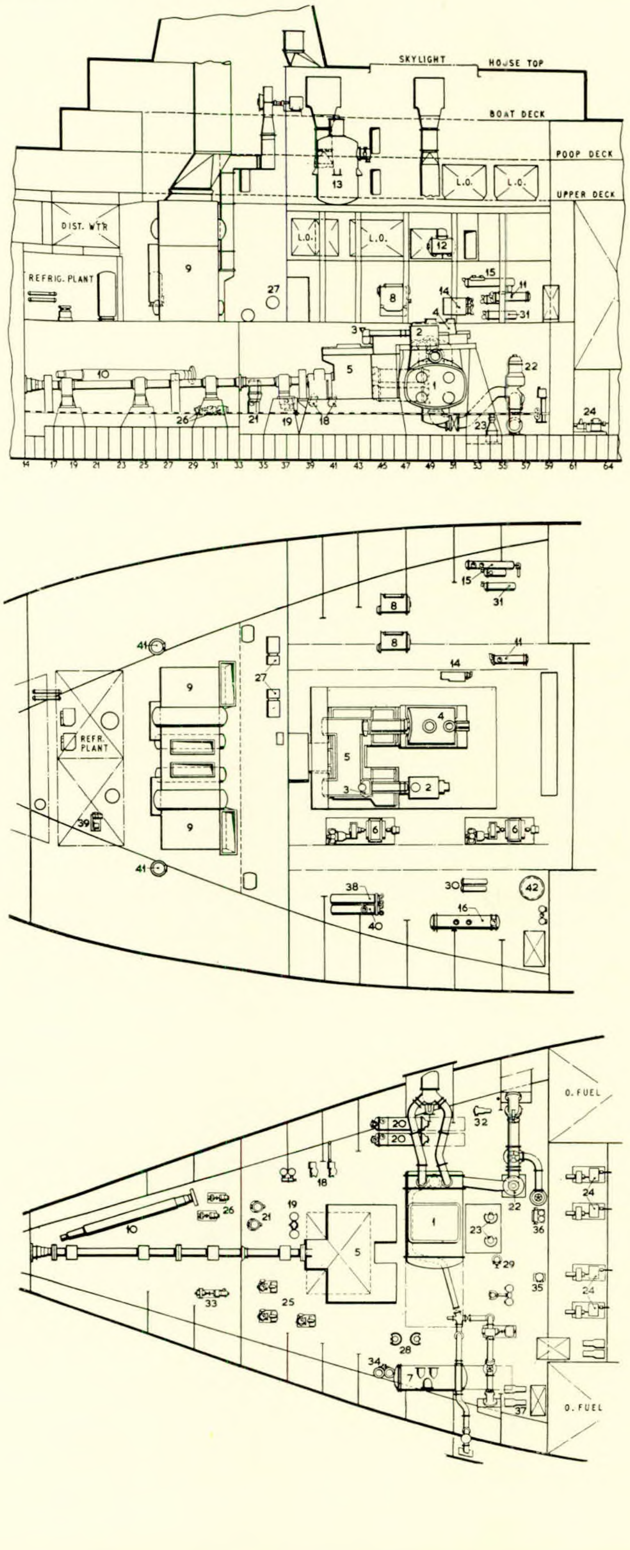
Sections through the H.P. and L.P. turbines turning the handwheel in a clockwise direction. Stopping and reversing manœuvres are carried out by turning the wheel in the opposite direction. The various valves are actuated by servo-oil pressure in correct sequence. Any number of governing devices can be connected to this hydraulic control system.—*The Marine Engineer and Naval Architect, May 1957; Vol. 80, pp. 172-174.*

Backing Tests of Geared Turbine Vessels

In order to ascertain the stresses in the propulsion shafts and reduction gears of geared turbine vessels when being manœuvred astern, backing tests were carried out on three tankers, three cargo ships, a training ship, and a ferry. In reversing, either from ahead to astern or *vice versa,* powerful torsional vibration was encountered at certain rates of revolution. The frequency was four times the product of the number of the propeller blades and the r.p.m., and the maximum amplitude of the vibration was in some cases 130 per cent of the ahead maximum continuous torque. The vibration in the astern from head case was more intense than in the ahead from astern case. It was found that the astern torque at half maximum continuous ahead r.p.m. was 60-80 per cent of the maximum continuous ahead torque; that the maximum astern torque was 70-80 per cent of the maximum continuous ahead torque; that a sudden rise in the steam pressure when running astern gives rise to an unexpectedly high peak of torque; and that the stresses in the gearing during the astern motion are considerably above the ahead designed value. The measured values are: root bending stress 1·4-1·8, shear stress 1·8-2·6, and rolling pressure 1·2-1·3 times the normal ahead values. These figures refer to mean torque; if torque fluctuation is taken into account the stresses will be greater.—*Shipbuilding Research Association of Japan, Report No. 9, 1956. Journal, The British Shipbuilding Research Association, December 1956; Vol. 11, Abstract No. 12,273.*

French-built Tanker for American Owners

The first of a series of four large oil tankers ordered from French shipyards has been delivered to her owners by the Chantiers de l'Atlantique (Penhoet-Loire), Saint Nazaire. This vessel, the *George F. Getty,* 53,085 tons d.w., is one of four sister ships, two of which were ordered from the Saint Nazaire shipyard, and two from the Ateliers et Chantiers de France, Dunkirk. A second vessel, the *Minnehoma,* was launched at Dunkirk early this year. The contract for these four vessels was signed by the Tidewater Associated Oil Company, Western Division, San Francisco, California. The *George F. Getty* has been delivered to an associate company, the Hemisphere Transportation Company, Wilmington, Delaware, and will operate under the Liberian flag. Three tankers of about 68,000 tons d.w. are also on order from the Dunkirk yard for the Hemisphere Transportation Company. The *George F. Getty* is a single-screw vessel powered by steam turbine machinery designed to give her a service speed of 16 knots. During sea trials, fully loaded, the vessel attained a speed of 17 knots at maximum horsepower. Oil cargo is handled by four high-output pumps and three stripping pumps. The principal particulars of the *George F. Getty* are as follows: —



KEY TO MACHINERY ARRANGEMENT

1. Main condenser
2. Ahead turbine
3. Astern turbine
4. L.P. turbine
5. Reduction gear
6. Turbo-generators
7. Aux. condenser
8. Evaporator
9. Boilers
10. Spare shaft
11. Condenser distiller
12. Make-up evaporator
13. Deaerator
14. Main ejector
15. L.P. heater
16. Condenser cooler
17. L.O. heater
18. L.O. purifiers
19. L.O. strainers
20. L.O. coolers
21. L.O. pumps
22. Main circ. pumps
23. Main condensate pump
24. Main cargo pumps
25. Main feed pumps
26. O.F. service pumps
27. O.F. heater
28. Aux. condensate pump
29. O.F. transfer pump
30. Aux. ejector
31. Drain cooler
32. Bilge pump
33. Turbo fire and Butterworth pump
34. F.W. strainer
35. G.S. and fire pump
36. W.B. and bilge pump
37. Contaminated drain pumps
38. Butterworth drain cooler
39. Control air compressor
40. Butterworth heater
41. S.W. pressure tank
42. De-oiler

Length o.a.	785ft. 10in.
Length b.p.	748ft.
Breadth moulded	102ft.
Depth moulded	53ft.
Draught (maximum)	39ft. 3in.
Deadweight	53,085 tons
Gross tonnage	33,705 tons
Net tonnage	21,396 tons
Service speed	16 knots
Output, normal	17,250 s.h.p.
Output, maximum	19,000 s.h.p.
Cargo capacity	461,375 barrels
Boiler fuel capacity	41,662 barrels

The main propulsion machinery in the *George F. Getty* consists of a high pressure ahead turbine of the impulse reaction type, and a low pressure ahead turbine of the reaction type. The astern h.p. turbine is of the impulse type and the astern l.p. turbine casing is of the reaction type. The turbines are of the C.E.M.-Parsons type and have been built by the Chantiers de l'Atlantique. The turbines have a service output of 17,250 s.h.p. at 101 r.p.m., and a maximum output of 19,000 s.h.p. at 104 r.p.m. Steam is supplied from two Foster Wheeler Marine type watertube boilers having a designed pressure of 700lb. per sq. in., and a capacity of 70,000 to 1,000,000lb. of steam per hour. The pressure at the superheater outlet is 600lb. per sq. in. and the temperature 840 deg. to 850 deg. F. Each boiler is fitted with four Pillard Todd Hex Press type burners. General Regulator combustion controls have been installed.—*The Shipping World, 10th April 1957; Vol. 136, pp. 371-373.*

Performance of m.s. *Dolius*

The *Dolius* is the second (the *Demodocus* being the first) of a series of similar ships ordered by Alfred Holt and Company, each of which is equipped with a Harland and Wolff six-cylinder engine with a diameter of 750 mm. and a combined stroke of 2,000 mm., the designed output being 8,000 b.h.p. or somewhat over. Of these engines the owners will have twenty-one in service by the end of 1958. The two ships are 487ft. overall, with a beam of 62ft. and a draught of 28ft. fully laden, and have a deadweight capacity of approximately 9,000 tons. Particulars are now available of the performance of the *Dolius*. The salient results and performance data of the ship are tabulated below. Taking the voyages as a whole, the following represent average readings:—

Exhaust gas temperature ...	830 deg. F.-840 deg. F.
Supercharging blower pressure	4·3-4·5lb. per sq. in.
Lubricating oil inlet temperature	114 deg. F.-115 deg. F.
Lubricating oil outlet temperature	116 deg. F.
Fresh water cooler outlet temperature	120 deg. F.-144 deg. F.
Fresh water inlet temperature	159 deg. F.-160 deg. F.
Fresh water outlet temperature	150 deg. F.-153 deg. F.

The Admiralty coefficient was 385·8 and the fuel coefficient 91,480. The fuel consumed at full speed for the complete voyages comprised 1,939·92 tons of boiler oil and 103·04 tons of Diesel oil. During stand-by periods and at reduced speed, 91·1 tons of boiler oil and 39·33 tons of Diesel oil were used, and, in port, 99·25 tons of Diesel oil. The propeller diameter is 18ft. 0·6in., the pitch being 16·43ft. and the surface 137 sq. ft.—*The Motor Ship, May 1957; Vol 38, p. 81.*

Safety Problem of Nuclear Propulsion

The first nuclear propelled merchant ship is due to sail on its maiden voyage within three years and, commercially, it would be virtually valueless unless the conditions on which a nuclear propelled ship may operate throughout the world are agreed internationally. There is not the least doubt that such an agreement will be one of the most difficult to negotiate with which the world shipping industry has yet been faced. The facile assumption, which appears to be made in some quarters, that because nuclear power production is comparatively simple on land, the nuclear propulsion of ships will be equally without complication, is not accepted by shipbuilders or shipowners. It is as well to make this point in view of the over-optimistic estimates which are being given as to the nearness of the widespread employment of nuclear propelled merchant ships on a commercial scale. The risk element is large and Dr. Georg Vedeler, director of Det Norske Veritas, went so far as to say that the danger of radioactive injury if there were an accident at sea was so great that it was desirable for efforts to be made to produce other artificial fuels before nuclear propulsion was employed in merchant ships. He thought that research should be carried out to find artificial fuels which were less dangerous. At the meeting of the United Nations Scientific Committee in Geneva dealing with the effects of atomic radiation, there was a wide measure of disagreement between the scientists whose reports were submitted and apparently the conclusions in some cases were quite contradictory. It is obvious that a mobile nuclear power plant trading over the seven seas, as a merchant ship must do, represents a much greater potential danger than any stationary plant, no matter where it is erected. In America a tentative approach to the question of ensuring the safety of a nuclear propelled ship has been made by a recent conference between the U.S. Government and the American Institute of Marine Underwriters, the outcome of which was the laconic comment that "considerable study will be required".—*The Motor Ship, May 1957; Vol. 38, p. 48.*

Automatic Control for Oil Fired Boilers

A simple automatic and metering control system for oil fired boilers has been developed by James Gordon and Co., Ltd. Known as the S.D. oil control, it is capable of handling quantities of fuel oil up to 6,000 lb. per hr., but equipment can also be supplied for dealing with larger quantities of oil if required. This system has been operated satisfactorily at sea in an oil tanker. By regulating the combustion air in relation to small variations in the pressure caused by changes in the demand for steam, the system will maintain steam pressure within narrow limits. It will also regulate the oil supply to the burners in the required relation to the air quantity, despite changes in the number of burners in use or changes in load. Two types of controller are used, one being a steam pressure controller and the other an air/oil ratio controller. The steam pressure controller (Fig. 1) embodies

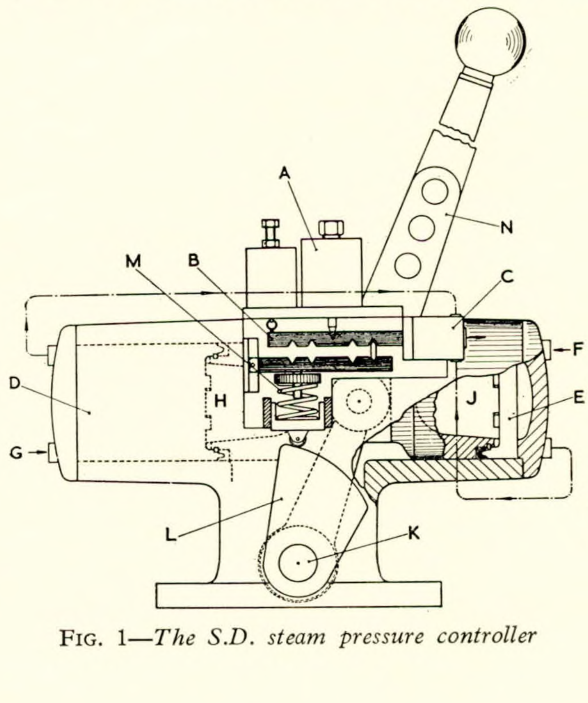


FIG. 1—*The S.D. steam pressure controller*

a metal bellows (A) in a chamber, by means of which pressure variations of a few ounces per sq. in. can be measured. The pressure on these bellows is transmitted via a suitable and adjustable lever system (B) to a pilot valve (C), which, while causing the pressure of a power medium to build up at one end (D) of the power cylinder, releases the pressure at the other end (E), or *vice versa*. The power medium, air or oil, is supplied to both these ends at F and G. The movement of the pistons (H and J) has to be transmitted to the lever of a damper in the air duct, and this is accomplished by first transforming the linear piston movement into a rotary movement of the spindle (K), on which is keyed a lever (N). This lever is then coupled, preferably by rigid means, to the damper lever. Rotation of the spindle (K) positions the cam (L) rigidly attached to it. In this way the force in the spring (M) is adjusted to the value required to balance the force exerted by the steam pressure on the bellows in the chamber (A). An increase of the steam pressure will be followed by an increase of pressure in the spring (M), whereas a drop in steam pressure has the opposite result. The cam (L) can be profiled, as required, to obtain an approximately linear relationship between piston position and corresponding air flow, thereby eliminating the unfavourable characteristic of a butterfly type air damper. In the air/oil ratio the controller diaphragm (A) measures the air flow to the burners, usually as the drop in pressure between burner windbox and combustion chamber. The force on the diaphragm is balanced through a lever system against the oil pressure as supplied to the burners. An oil valve receives oil from the pump through the inlet port, reduces the pressure as required and the oil then flows to the burners through the outlet port. The relationship between the air differential pressure and the oil pressure at the outlet, the air/oil ratio, can easily be set at the required point. Whereas for the steam pressure controller a small quantity of an operating medium—air, water or oil—at a maximum pressure of 100lb. per sq. in. is required, no operating medium is required for the air/oil ratio controller. Both controllers can easily be operated by hand in the event of failure of the supply of the operating medium.—*The Shipping World, 1st May 1957; Vol. 136, p. 432.*

Large Tanker Built in Holland

The 400th vessel launched from the Kockum yard since the company began shipbuilding operations in 1873 was also their largest ship. She is the 39,000 tons d.w. *World Splendour* for Merchant Tankers Corporation, one of the Niarchos group of companies. She is all-welded, except for the usual crack-arresters specified by the American Bureau of Shipping, and longitudinally framed. The twin longitudinal bulkheads are flat and the transverse ones are corrugated, all of them stiffened with transverse and longitudinal webs. The main dimensions are as follows: —

Length overall	700ft. 0in.
Length between perpendiculars	665ft. 0in.
Moulded breadth	97ft. 0in.
Moulded depth	49ft. 3in.
Summer draught	35ft. 6in.
Cargo capacity, about ...	2,002,000 cu. ft.
Bunker capacity, about ...	4,500 tons
Machinery	16,500 s.h.p.
Speed on loaded trials ...	17 knots
Range (at 16½ knots) ...	20,000 miles

A suspended spade-type rudder designed to reduce vibration by the omission of the traditional stern-post and tail piece is employed. The rudder stock is secured by three bearings, the upper one taking the whole weight (28·5 tons) and the others serving only as hinges. The cargo space is divided into 11×3 tanks. The main pump room, adjacent to the engine room, will have four 1,250-ton turbine-driven horizontal centrifugal pumps, and two 150-ton stripping pumps. A full oil cargo will be discharged in about 13 hours. The *World Splendour* will be propelled by a single-screw set of Kockum-Laval double reduction geared steam turbines, developing 16,500 s.h.p. at 103 r.p.m. and taking steam from two Foster Wheeler D-type watertube boilers at 600lb. per sq. in. and 865 deg. F. Two 900 kVA 450-volt 60 cycle turbo-alternators and one automatic-starting Diesel alternator of 150 kVA are to be installed.—*The Marine Engineer and Naval Architect, April 1957; Vol. 80, p. 126.*

Further Sea Trials on the *Lubumbashi*

The *Lubumbashi* trials of which the results have been published hitherto, were carried out on a newly built ship. It was decided by the Centre Belge de Recherches Navales to take advantage of the instrumentation of this ship, especially the torsionmeter, the thrustmeter and the pitometer log, to make further investigations on the behaviour of this vessel as she became older. Renewed measurements on the engine running on heavy fuel made it clear that heavy fuel is appropriate to a Diesel motor from the point of view of the economy of the ship. Records taken in varying weather conditions during two Atlantic winter voyages, with two different draughts, threw new light on the effect of weather on ship's speed. Great attention was paid during the first three years of the life of this vessel to the increase of frictional resistance. Information on the subject is given by speed, thrust and power data, as well as pitot traverses taken over the bottom in the centre line, fore, amidships, and aft. An attempt is made to correlate these data with the results of roughness measurements.—*Paper by G. Aertssen, read at a joint meeting of the Institution of Naval Architects and the Institute of Marine Engineers on 27th March 1957.*

Hydraulically Operated Hatch Covers

The hydraulic hatch cover is a tremendous boon to the shipowner and the mechanism and idea will continue to be developed for the ultimate benefit of both the shipper and shipowner. Hydraulically actuated covers are essential for any new ship design. With this type of cover single holds can be opened up completely in two minutes. Weather deck covers usually take longer—about three minutes. Dogging usually takes 1½ to two minutes. The weather deck hatches on the Mariner-type ships require 5·6 man hours to undog, while a similar ship with hydraulic hatch covers requires only 0·46 man hours by the crew. The opening, performed by the stevedore gang, requires 9·06 man hours on the Mariners to 0·46 man hours with hydraulic covers. The latter item does not include the time wasted by the full stevedoring gang waiting to go to work. For the crew to dog down the covers requires 6·3 man hours on the Mariners against 0·46 man hours on a vessel with hydraulic hatch covers. The figures on the opening and closing of hatch covers on a typical voyage plus handling of the booms represents $2,280 on a conventional ship. On a ship with hydraulic hatch covers and cranes this cost is only $245 per voyage.—*Marine Engineering/Log, April 1957; Vol. 62, pp. 81-82.*

Transverse Bow Propellers

A development of the last few years which has interesting possibilities is the use of a propeller, normally of Voith-Schneider type, operating in a transverse tunnel extending through the bows of a ship from one side to the other. In use, such a propeller produces a flow of water through the tunnel which emerges as a jet and imparts a sideways force to the bows, thus giving the shipmaster a degree of control over his ship which has not previously been available in ships with normal screw propulsion. The ability to control the motion of the bows in the ship in this way is valuable mainly when berthing or leaving a berth, and it is therefore not surprising that installations of this sort have so far been confined to ferries, a type of vessel that has to bring itself alongside without assistance at frequent intervals. The first example to attract attention was fitted in the Clyde-built Canadian ferry *Princess of Vancouver,* and more recently one has been installed in a Swedish ferry, the *Prinsessan Margaretha*. The recent tug strike at Southampton prompts the suggestion that a transverse bow propeller might prove useful in a large passenger liner. Under normal conditions it would give added safety in cross winds when berthing and might allow a smaller number of tugs to be employed. And when berthing under strike conditions without tugs, a manœuvre which has to be carried out at not infrequent intervals at New York, and less frequently elsewhere, it would be invaluable. On the technical side, there are various alternatives to the present arrangement which might be worth consideration. One is to replace the Voith-Schneider propeller by a straightforward propeller operating in the tunnel like a torpedo fan and driven by a submersible electric motor. The Pleuger active rudder would provide the right start for this installation, though the submersible motor is of squirrel cage type and would need alternating current. A further alternative is to dispense with the tunnel and employ a retractable unit protruding below the keel when in use. This might be either of Voith-Schneider or active rudder type, and would allow the propulsive force to be exerted in any direction, even to the extent of propelling the ship in the event of main machinery breakdown.—*The Shipping World, 10th April 1957; Vol. 136, p. 362.*

Seaworthiness Research

Fundamental to any study of the behaviour of a ship in a seaway is a knowledge of the dynamic characteristics of the sea itself. Although the static characteristics of a ship are completely definable, the dynamic characteristics which determine the response of a ship to the sea are to a great extent a matter of intelligent conjecture. To apply a logical technical basis to such conjecture, three basic phenomena must be completely defined: (1) The seaway; (2) The response of a statically defined ship to that seaway; (3) The response of the seaway to the statically defined ship. Definition of a seaway implies not only an instantaneous pictorial view of the sea surface but a knowledge of the change of that surface with time. In addition, the dynamic behaviour is influenced by gravitational forces, wind forces, and currents, the effect of all of which must be understood to define completely that complex phenomenon which is called a confused sea. The definition of the response of a ship to a seaway is equally complex. In addition to the motions of surge, heave, sidle, pitch, roll, and yaw, which are motions of the rigid body, there must be included bending, torsion, tension, and compression, which are the elastic phenomena existing in the complicated structure of a ship. As a ship moves through a dead calm sea it produces a wave train which is a pronounced disturbance of the water through which it moves. The disturbance is due only to forward translation. When such an effect is compounded by the motion in six degrees of freedom which a ship exhibits in a seaway, it is obvious that a much more pronounced effect will be evidenced on the surface of the sea. Detailed definition of this effect is also essential to the study of the behaviour of a ship at sea. An experimental investigation leading toward a complete definition of all of these phenomena for the case of a single representative ship is essential. The ship employed should be sized, proportioned, and powered to such a scale that experimental results can be immediately extrapolated to other designs of interest to the Navy and to the maritime industry in general. The suggested method for defining the seaway is centred around the most recent techniques of stereophotogrammetry. A helicopter equipped with stereocameras mounted at either end of a 50-ft. horizontal boom can be used. The helicopter can cruise with the ship over a selected area at an altitude of 200-400ft. taking photographs at two-second intervals. The cameras can be so oriented as to give full stereo coverage over a minimum of a 400-ft. square area. The helicopter can be instrumented to record accurately its flight altitude and attitude during all photographic periods. Co-ordinating time signals can be telemetered to the experimental ship throughout all measurements. A detailed study of this method demonstrates that the sea surface can be defined with an accuracy of plus or minus 0·10ft. relative to the helicopter. No difficulty in operating a helicopter in this manner, even under adverse weather conditions, is envisaged.—*R. Taggart, Journal of the American Society of Naval Engineers, February 1957; Vol. 69, pp. 51-57.*

Ocean Iron Ore Carriers

This paper deals with a number of points on the design of iron ore carriers. At the present time there are about fifty-seven ore carriers under construction or on order for United Kingdom account alone, and, as the requirements of the steel industry for imported ores grow, so will the demand for bulk carriers specially designed for the ore trade. Published figures of the different types of vessel under construction or on order in British shipyards show that ore carriers account for about 6 per cent of the total, compared with an output of tankers of just over 50 per cent and general cargo vessels just over 33 per cent. The British Iron and Steel Research on behalf of the British Iron and Steel Federation prepared in 1953 a very full report, the result of a wide and detailed investigation into all aspects of the transport of iron ore from abroad to United Kingdom ports. The sections in this report dealing directly with ore carriers have made a very thorough analysis of the ore trade requirements and as predesigning data this information is invaluable and especially so when the ships for home trade needs can be confined to approximately three classes, and the numbers in each class fairly large. The economics claimed for the mass production of standard bulk carriers are realized only when one shipyard undertakes the production of large numbers of the one type and all of the same dimensions. Whether a shipyard is prepared to follow this course is for the yard to decide and present conditions appear to give every opportunity in that direction. Also they encourage a more serious interest in the need for the co-ordination and standardization of plans, specifications and auxiliary equipments. The fewer in the team or group concerned with this work the easier it will be to achieve exact duplication and a better integration of all the problems concerned with cargo requirements, routes, limitations or dimensions, etc., and ultimately to the optimum design of carrier. It is equally important to consider in the same way the propelling machinery, its type and power. The optimum speed to meet economically the requirements of the selected route must be determined by the overall time table. To overpower the vessel for a speed beyond the needs of the service will lower the earning potential in addition to increasing the capital cost, and speed is not always an end in itself in a highly competitive enterprise.—*Paper by J. Lenaghan, read at a meeting of the Institution of Naval Architects on 26th March 1957.*

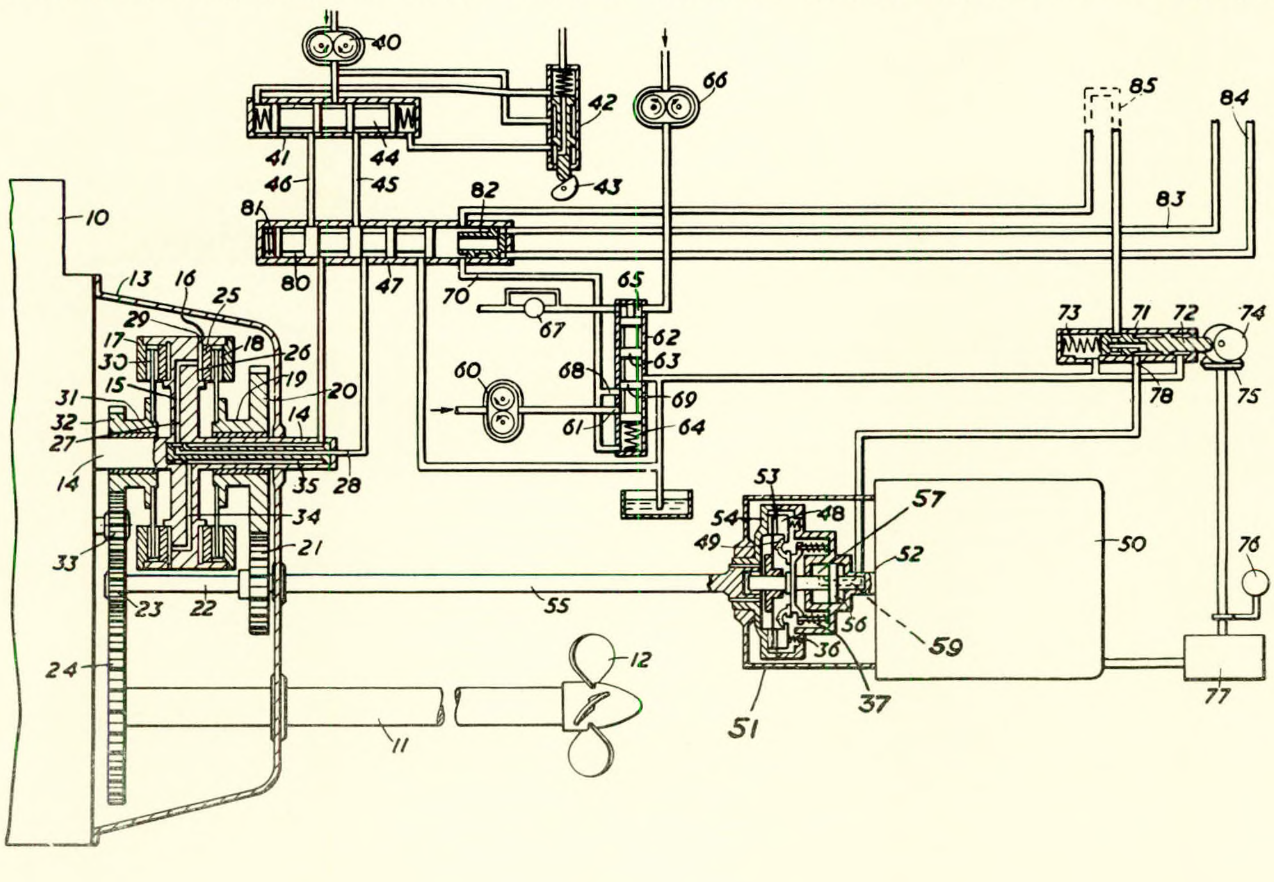
Patent Specifications

Rust-preventive Deck Covering

The decks of ships are much liable to rust formation and heretofore efforts have been made to prevent this by applying various paints. This, however, makes it necessary for the coating of paint to be regularly renewed, which involves high maintenance costs. It has been known to apply to decks of ships a covering of poured asphalt (a mixture of asphaltic bitumen and mineral constituents). A drawback of poured asphalt is the heavy weight per sq. m. at a thickness of 1½ to 2 cm. (40-50 kg.): against this there is the advantage that decks with these coverings may withstand heavy mechanical damage. Most ships have decks, such as for example promenade decks, which in general are not subjected to a heavy load. The following requirements are made of the coverings of these decks: (1) They must offer a complete protection against rust formation, brought about by air, sea water, etc. (2) The covering must be resistant to widely varying temperatures. (3) The covering must offer sufficient mechanical strength to be walked upon. (4) The covering must be of light weight. The object of this invention is to produce a rust-preventing protective covering on decks of ships which satisfies these requirements. The protective covering according to the invention consists of a preservative coating of blown asphaltic bitumen, a layer of felt impregnated with asphaltic bitumen and on this felt layer a top coating of blown asphaltic bitumen containing broken mineral constituents. The covering may be further reinforced by providing in it a layer of felt, asbestos felt, glass cloth, glass wool or the like. The rust-preventing protective covering, according to the invention, can be applied in the following manner. A deck, after rust has been removed from it, is provided with a primer coating. A material suitable for this purpose consists of a solution of blown asphaltic bitumen in solvent naphtha, benzene, mineral turpentine or some other organic solvent. To this primer coating a coating of a blown asphaltic bitumen of at most 2 to 3-mm. thickness is applied. To protect this coating from the mechanical effects there is applied to it a layer of felt impregnated with asphaltic bitumen and this layer is provided with a coating of asphaltic bitumen with a filler comprising crushed slate, rubble or other rock.—*British Patent No. 776,115, issued to Key and Kramper Asphalt Ruberoid N.V. Complete specification published 5th June 1957.*

Marine Propulsion Plant

This invention relates to marine propulsion plants comprising a main engine connected to the propeller through a transmission including a clutch and an auxiliary engine also capable of being connected to the propeller shaft through a clutch. The auxiliary engine might be an auxiliary internal combustion engine, and may be used for other purposes in



addition to, or alternative to, its use as an auxiliary propulsion engine. According to the invention the auxiliary engine clutch is hydraulically operated and the unit includes a source of hydraulic pressure controlled by or responsive to the speed of rotation of the auxiliary engine, to supply fluid to operate the auxiliary clutch. Thus connexion of the auxiliary motor to the propeller shaft when that motor is stationary is automatically prevented. Preferably the source of hydraulic pressure is a hydraulic pump which is driven by the auxiliary engine and is arranged to deliver pressure fluid to a pressure line and the power unit includes pressure responsive apparatus arranged to be responsive to the pressure in the auxiliary clutch pressure line, and so constructed and arranged that engagement of the auxiliary clutch can only be effected when the auxiliary engine, and hence the auxiliary engine pump, is rotating at or above a pre-determined speed.—*British Patent No. 774,189, issued to D. Napier and Son, Ltd. Complete specification published 8th May 1957.*

Ship's Form

The invention relates to a ship's form and has for its object the reduction of wave-making resistance of ships' forms, having a parallel middle body and V-shaped fore body sections, the connexion lines between the centres of area of half the sectional areas thus having the form of a slightly concavo-convex line. As is well known, the decisive factors for the size of the wave-making resistance are the pressure distribution on the wetted surface and the superposition or interference, respectively, of the wave systems created by the forward movement of the vessel. The object of the invention is to improve these given conditions by two principal methods, first by a particular given value of the tangent to the half-sectional area curve in the fore body, and secondly by an adequate shape of the transition from the fore body to the parallel middle body and from the latter to the stern. According to the present invention, the form of a ship's hull with V-shaped fore body and, if desired, after body with straight, convex concave heart-shaped or conical V-shaped frames and with obliquely raked stem and parallel middle body portion, in which the line joining the centres of area of the half-sectional areas has the form of a slightly concavo-convex curve, is characterized in that the entry tangent (t) of the half-sectional area curve ensuing from the above-mentioned shape of frame is given by a value in accordance with the following relationship: $t = kCp^{4 \cdot 5}$ in which the dimensional constant (k) varies between 4·5 and 5·5 and the longitudinal prismatic coefficient (Cp) between 0·55 and 0·82. While the ship's hull according to Figs. 2 and 2A has pronounced shoulders, and even in Fig. 2B non-uniform transitions at which flow separations and instabilities of stream line flow increasing the resistance occur, Figs. 3, 3A and 3B of the ship's form according to the invention exhibit a completely regular flow in the direction of the diagonals.—*British Patent No. 775,706, issued to E. R. F. Maier. Complete specification published 29th May 1957.*

Hatch Covers

The invention is a tent for protecting an open hatch of a ship, the tent having a roof adapted completely to mask the hatch opening from above and an end open or adapted to be opened and disposed outside the vertically projected area of the hatch opening and a slot in the roof. This slot extends

FIG. 1.

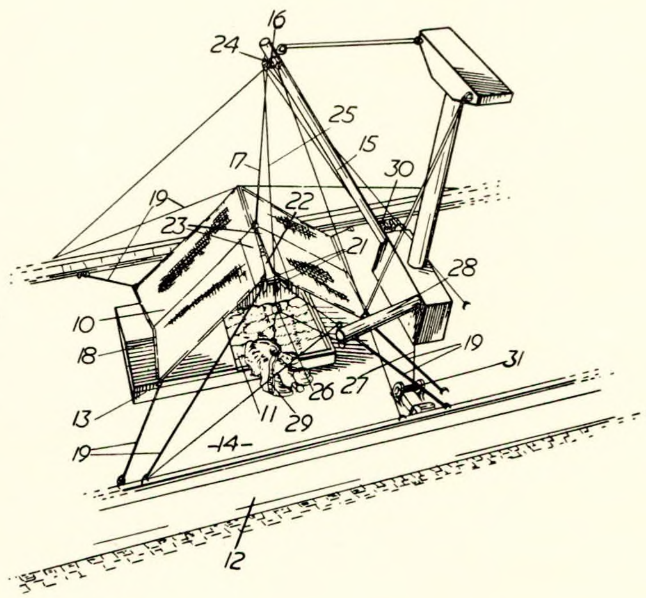


Fig. 2b.

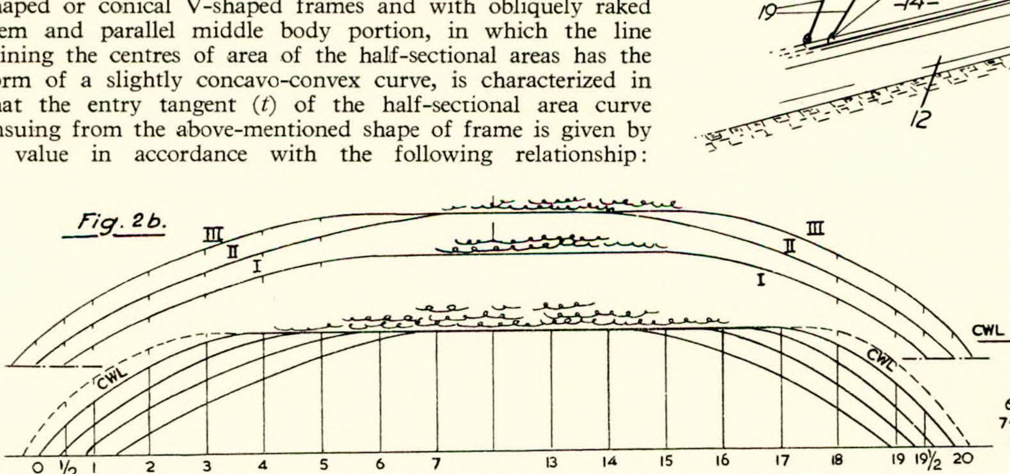


Fig. 2a.

Fig. 2.

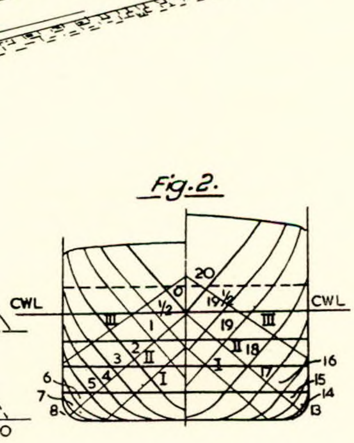


Fig. 3b.

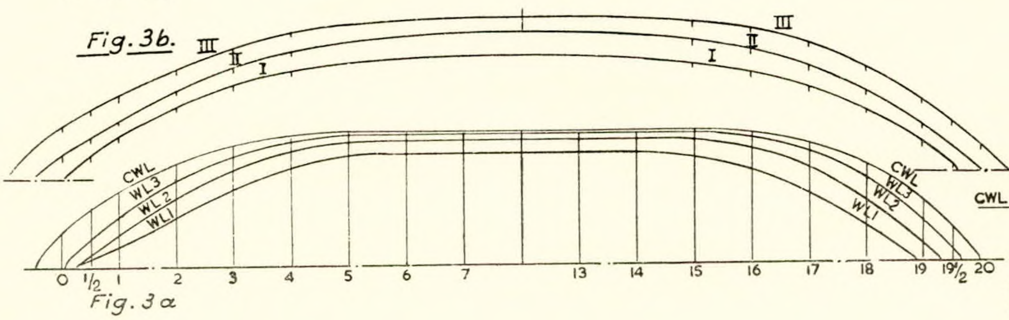
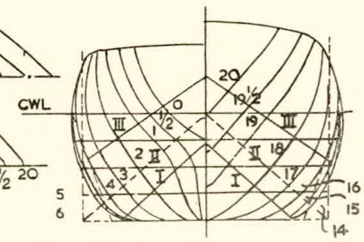


Fig. 3a

Fig. 3.

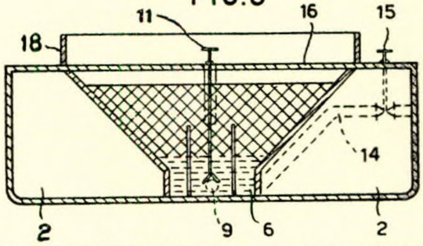


longitudinally from or adjacent the centre of the roof into the open end, whereby a fall or runner from a derrick can be moved along the slot to and from above deck and above hatch positions. This will make it possible to continue loading or unloading operations during weather detrimental to the cargo. The slot is closed with at least one rubber or other flexible flap which will not impede movement along the slot of the fall or runner. The tent (10) in Fig. 1 is supported in the required position by a ship's derrick (15) provided with a pulley (16) over which a fall or runner (17) passes, the free end of the runner (17) supporting the tent (10). The sides of the tent, which are provided with wire or otherwise reinforced edges (18) to prevent sagging of the tent (10), are stretched over the hatch opening (11) and retained in this position by guy ropes (19) attached to the deck (14) and its sides.—*British Patent No. 775,672, issued to A. Hood. Complete specification published 29th May 1957.*

Ship for Receiving, Separating and Transporting Mixtures of Water and Oil

The invention consists of a ship for receiving, separating and transporting mixtures of water and oil or other immiscible liquids, comprising a number of settling and separating tanks forming the hold of the ship. In the drawing (Figs. 1 and 2) a flat bottomed ship is represented with buoyancy tanks (1 and 2) formed in the ends and sides respectively of the hull and longitudinally extending separation tanks (3, 4 and 5). The bottom part of the separation tanks are provided with a relatively narrow gutter (6) of channel section (Fig. 3) in which settling water may concentrate, and this gutter leads

FIG.3



into the divergent sides of tanks. In this widening part oil can readily float upwards over a large surface. In the illustrated example three tanks are shown. To the initial tank (3), e.g. by means of a pump (7) with a discharge conduit (8), impurified water is supplied. The oil is indicated by cross hatching and the water by horizontal dotted lines. The tank (3) is connected by means of openings (9) in a partition (10), which openings may be controlled by a valve (11), with the tank (4) in which a separation takes place. This tank (4) communicates through adjustable openings (12) in a partition (13) with a tank (5) where a substantially complete separation can take place. Water is discharged from the lower part of the tank (5) by means of a siphon (14) with an adjustable valve (15). The siphon can be used when the buoyancy tanks (1 and 2) are employed and the impurified water level in the tanks is higher than the water level outside the ship. In the deck there are hatches (17).—*British Patent No. 775,951, issued to C. in't Veld and J. in't Veld. Complete specification published 29th May 1957.*

FIG.1

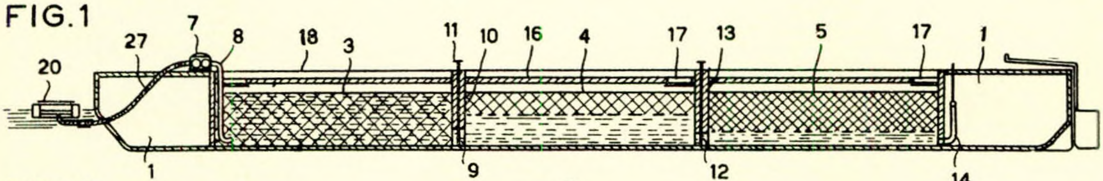
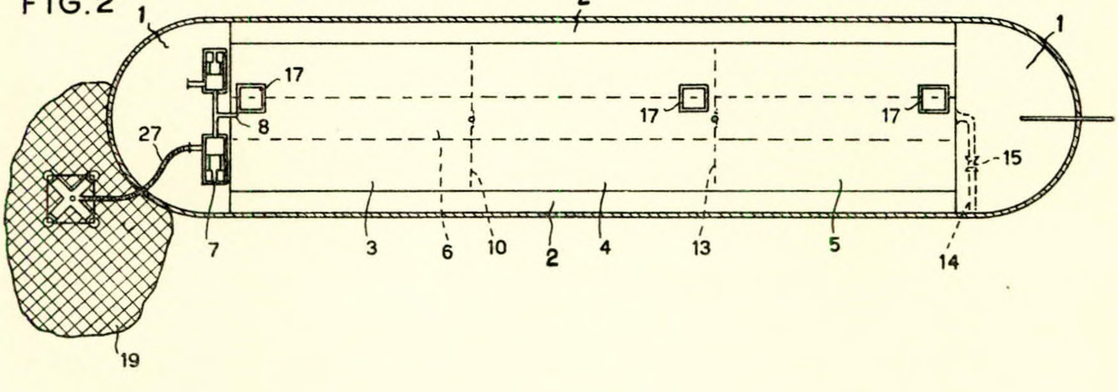


FIG.2



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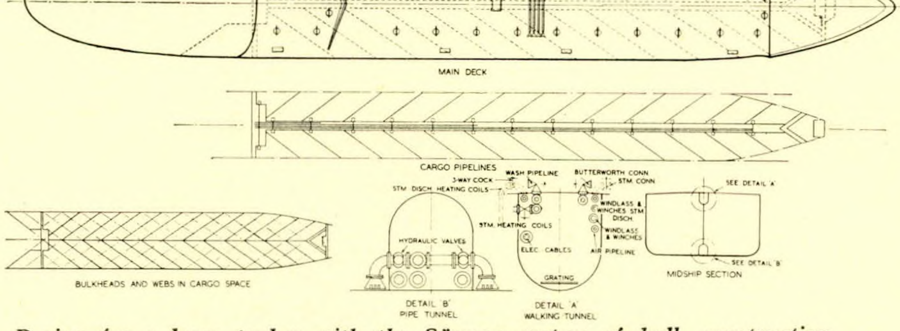
Marine Engineering and Shipbuilding Abstracts

Volume XX, No 9, September 1957

Large Tankers of Greater Strength

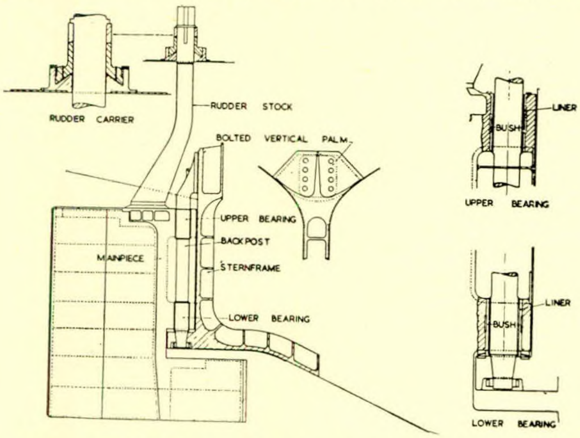
A proposal to increase the structural strength and seaworthiness of large tankers in the 45,000-ton class has been put forward by Mr. Rolf Sörman, head of Transoil-Rederierna, Gothenburg. Features of the new method of hull construction include the turning of the transverse bulkheads to an angle of 45 degrees with the centre line and the placing of the webs at a similar angle. In this way a reduction is claimed in the shearing forces induced in a tanker's hull when carrying cargoes of high specific gravity while at the same time certain tanks are empty. The longitudinal girders under the deck and on the hull bottom are arranged to form tunnels. The upper tunnel carries various pipelines and electric cables while the lower one is occupied by cargo pipelines having remotely controlled hydraulic valves. Full protection is thus obtained from the sea and weather. As will be seen from the plans there is no 'midship bridge structure, and, by shaping the front of the poop house in the form of a bow, it is anticipated that a sea taken on board will be deflected to the deck sides; a small breakwater is placed just forward of the deckhouse. Trimming of the ship slightly aft will ensure an easy run of cargo to the suctions. The dry cargo hold has been omitted



Design for a large tanker with the Sörman system of hull construction and the tanks immediately forward of the main deckhouse raised to improve the vessel's trim. Clearer decks and additional longitudinal strength are expected to result in better sea qualities under adverse weather conditions. Although the reduction in weight of a tanker constructed by this method is not marked, the increased strength provides a greater margin against corrosion. Lloyd's Register of Shipping and Norwegian Veritas have both approved this method of construction which is stated to be superior to that of the conventional tanker design. The general design is especially suitable for the large tankers where longitudinal strength is a major problem, particularly under certain conditions of loading and trim.—*The Motor Ship, May 1957; Vol. 38, p. 77.*

Improved Support for a Semi-balanced Rudder

Conventional practice when fitting a semi-balanced rudder to a twin-screw ship has been to use two-pintle support. Service reports sometimes indicate excessive wear of the pintles, due in all probability to the forces acting on the underslung portion of the rudder. It has also been noted that the wear of the horizontal bearings causes the vertical loads to be transferred to the rudder carrier and that oval wearing of the tapered hole in the lower gudgeon is caused by the lower pintle nut working loose and permitting the pintle to become slack. The technical staff at the Wallsend yard of Swan, Hunter



and Wigham Richardson, Ltd., have developed a backpost arrangement giving greater rigidity which, with increased bearing area and a slight widening of the hull at the deadwood, has reduced the risk of propeller-excited hull vibrations; there is also a reduction in the deflexion of the underslung area when subject to side loading. The bushes and liners are of stainless steel, the same material being used for the bearing rings which take the vertical loads at the lower stern frame gudgeon. Additional strength is provided by the adoption of a cast steel rudder main piece complete with gudgeons, the remainder of the assembly being of heavy steel plates. A ship fitted with the new design of rudder and backpost showed satisfactory steering qualities during trials and after nearly two years in service there have been no complaints, an encouraging indication of the success of the new arrangement.—*The Motor Ship, May 1957; Vol. 38, p. 84.*

Peen Plating

Peen plating is a newly developed method for plating hardware with such metals as zinc and brass, without the use of heat or electricity. The process comprises a combination of the technique of barrel finishing with the principle of cold welding. Plating is carried out in a modified tumbling barrel, containing the coating metal in the form of powder and also the impacting material. The operation is done under water, to which chemical substances are added to promote the cold welding of the powder to the basis metal. This paper describes equipment, materials and the process, with consideration of the scope of its application. Articles suitable for peen plating include tacks and nails, clamps and washers, small fasteners, hooks, springs, and small stampings of many kinds. —*Paper by G. H. Jenner and T. P. Hoar presented at the Spring 1957 Conference of the Institute of Metal Finishing. —Abstract, The Nickel Bulletin, May 1957; Vol. 30, p. 70.*

Aluminium Heating Coils

Since the first experimental installation was fitted in No. 5 centre tank of the tanker *British Gratitude* in October 1953, the Alacoil system of aluminium alloy heating coils has been fitted to or ordered for a total of 170 tankers. The original pipe clip used with the Alacoil system was made in heavy mild steel strip, to resist corrosion. It was bolted to a mild steel flat base, with four resin-asbestos pins to locate the tube and at the same time to insulate it from the steel structure of the ship. This method of holding down the pipe and ensuring insulation was entirely satisfactory from the operational aspect, but the total weight of the clipping system was roughly equal to the weight of the pipes in the tanks. The steel clip was therefore soon superseded by one made in resin-bonded glass fibre, which has excellent insulating properties. This clip has a tensile strength of 26,000lb. per sq. in. and a flexural strength of 32,000lb. per sq. in. with an impact strength of 30/40ft. lb./in. on an unnotched specimen. It stands up well to temperatures below 210/220 deg. C., and it is impervious to and unaffected by all types of cargo oil or sea water, cold or warm. A further modification to the clips used has recently been incorporated in the installation, a patented design of clip known as the Mark III being introduced. This clip incorporates a controlled leak-off resistance in the form of a semi-conducting plug on which the piping rests. It has been introduced to eliminate the slight possibility of static electricity building up on the heating coils and causing a spark. The resistance to earth of the system remains high enough to ensure that there is no possibility of electrolytic action on the coils, but is sufficiently low to allow any static electricity to leak away before a substantial voltage could be built up. Another item of equipment to which modifications have been made is the patented steam conditioning unit. The purpose of this unit is twofold: firstly to act as a separator, removing entrained moisture from the steam on its way to the system; and secondly to ensure the deposition of any copper or copper salt pickup from the deck steam lines. An ordinary steam trap and strainer would not serve this purpose. The original steam conditioning unit was fitted with sacrificial plates of zinc. The design of this item has been modified, and the cast steel box on deck, serving a series of two wing and one centre tanks, is now fitted with sacrificial plates made in aluminium alloy, with two specially designed plates in steel, which prevent erosion due to the impact of steam on the plates.—*The Shipping World, 5th June 1957; Vol. 136, pp. 546-547.*

Twenty-knot Lifeboats

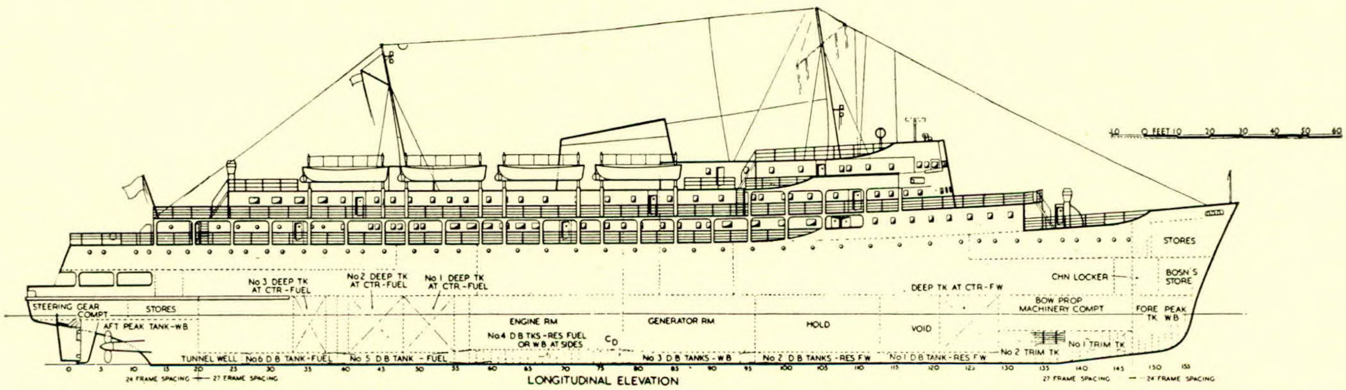
The first of a series of entirely new type of lifeboat, quite unlike anything built in this country, has been completed for the Deutsche Gesellschaft zur Rettung Schiffbrüchiger (corresponding to the R.N.L.I.) by the yard of Fr. Schweers, Bardenfleth. She is triple screw, has variable-pitch propellers, an outer hull of steel and an inner hull of aluminium alloy, the speed being 20 knots. The overall length is 76ft. 1¼in., and the normal draught is 4ft. 7⅛in., the breadth being 14ft. 4½in. She has been named *Theodor Heuss.* The central propeller is driven by a high-speed (1,500 r.p.m.) Daimler-Benz engine of 1,100 b.h.p. and the port and starboard shafts by 150-h.p. Daimler-Benz engines. The speed when operating the two wing engines is 11 knots and 20 knots is attained with the main engine in service. It is intended to build five similar vessels.—*The Motor Ship, April 1957; Vol. 38, p. 39.*

Australia-Tasmania Ferry

A vehicular passenger vessel is to be built by the State Dockyard, Newcastle, for the Commonwealth Government, in order to provide fast overnight travel between the mainland of Australia and Tasmania for both passengers and vehicles. In many respects this will be the most important and interesting ship yet constructed in an Australian shipyard. The design of the vessel and the machinery to be installed was developed after a very close inspection by executive officers of the Australian Shipbuilding Board of such vessels operating in other parts of the world. The main details of the vessels are:—

Length, overall	370ft. 0in.
Breadth, moulded	58ft. 0in.
Breadth on waterline	58ft. 0in.
Depth, moulded to vehicle deck	20ft. 0in.
Depth, moulded to upper deck	36ft. 3in.
Draught	15ft. 0in. (approx.)
'Tweendeck heights	
Vehicle to upper deck ...	15ft. 6in.
Upper to bridge deck ...	8ft. 0in.
Bridge to navigating bridge deck	8ft. 0in.
Navigating bridge to wheelhouse top	7ft. 6in.
Crew flat aft to upper deck	7ft. 6in.
Service speed	About 17¾ knots

The vessel will carry cargo in trailers, also motorcars up to a total of 550 tons in the vehicle 'tweendecks, up to 100 tons of general cargo in the hold and a maximum of 400 passengers. Denny Brown stabilizers will be fitted to provide as much comfort for the passengers as is possible on a crossing which



LONGITUDINAL ELEVATION

at times becomes very rough and a Voith-Schneider bow propeller will also be provided. The double bottom is to be arranged to carry oil fuel, fresh water and water ballast. The water ballast tanks are to act as trimming tanks, and deep tanks will be arranged to carry oil fuel and fresh water. Deep tanks are to be fitted with divisions and wash bulkheads. The vessel has twin rudders and a 500-h.p. Voith-Schneider bow propeller, which should give good manœuvring control and reduced speed in confined waters at the terminal ports. The twin-screw machinery installation comprises two M69TS Nohab Polar two-stroke engines, giving a combined output of 8,600 b.h.p. when operating at 230 r.p.m. A speed of 17¾ knots will be obtained in service at approximately 215 r.p.m. and 7,000 b.h.p. Each engine is fitted with three exhaust turboblowers; motor driven blowers are also provided for use when manœuvring. Fuel is admitted to the cylinders on the pilot injection principle, which is standard equipment on Nohab engines and the vessel will be operated on Class B Diesel fuel. The pistons will be oil cooled and the jackets fresh water cooled. Electric power at 240-volts d.c. will be supplied by four 435-kW Ruston Diesel-engined generators running at 428 r.p.m. Two of these will be used normally in service, the third when entering and leaving port, whilst the fourth will serve as a standby unit. The generators, which are located in a compartment forward of the engine room, will have their own fuel oil supply system and lubricating oil purifier. The Denny Brown stabilizers will also be installed in this space.—*The Motor Ship, May 1957; Vol. 38, pp. 86-87.*

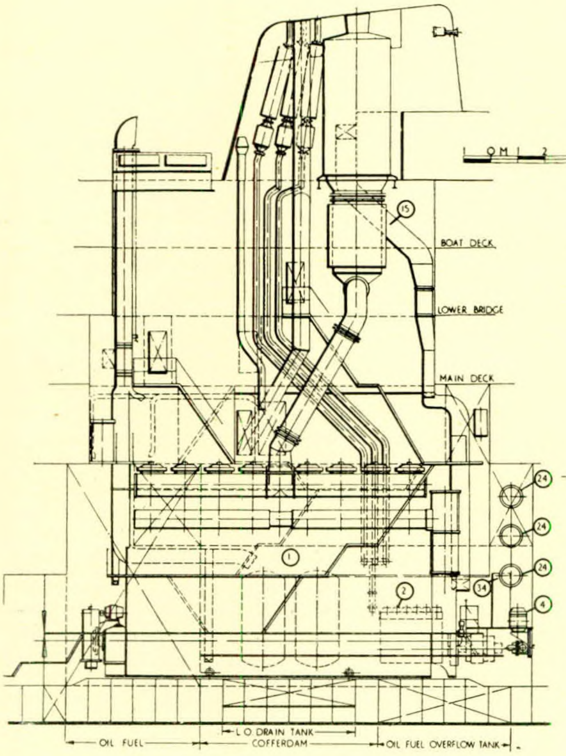
New Thames Tug

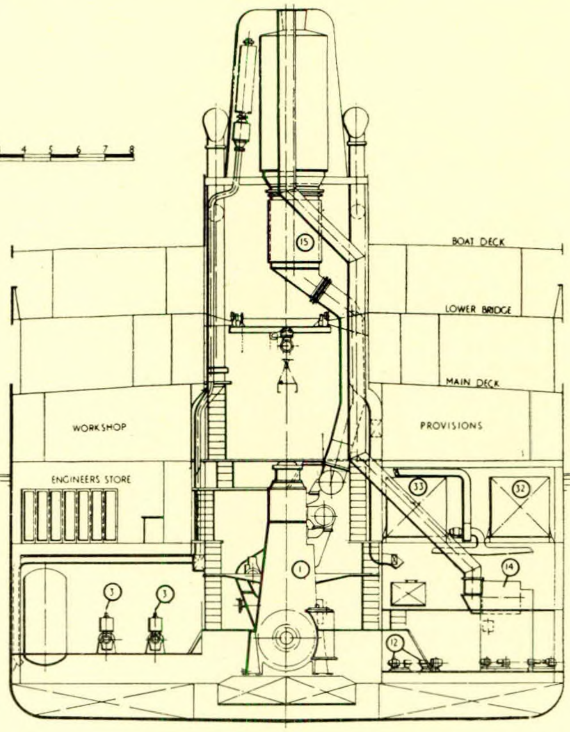
The motor tug *Arthur* which commenced continuous 24-hour service on the Thames recently incorporates a number of unusual developments. Built by T. Mitchinson, Ltd., Gateshead-on-Tyne, to the order of Samuel Williams and Sons, Ltd., she is of the Hydroconic design of Messrs. Burness, Corlett and Partners, which simplifies construction and reduces first cost. A patent bulb rudder is also fitted. The length b.p. is 66ft., the moulded breadth 18ft. 3in. and the moulded depth 9ft. 6in., the draught aft being 8ft. 6in. Being designed for the almost continuous handling of lighters, particular attention has been paid to the machinery installation, the main engines comprising two National R4AM6-type units with chrome-hardened liners and developing a total of 460 s.h.p. at 600 r.p.m.; they are connected to a twin-input single-output M.W.D. reverse-reduction gearbox by Fluidrive traction-type hydraulic couplings. The couplings are cooled by integral fans and trunking which serve also to ventilate the engine room. The 2·4:1 gearbox incorporates extended drives on the primary shafts for the hydraulic steering gear pump, air compressor and generators. Another interesting feature is an automatically operated brake fitted on the intermediate shaft. This can be applied manually if necessary but is normally engaged when the gear is moved into a neutral position. The machinery is arranged for remote control from the bridge and the equipment is of the pneumatically-assisted type. As the vessel is designed essentially for barge towing, the propeller has been designed on the basis of a 4-knot towing condition. Consequently the static bollard pull is reduced but nevertheless a steady observed pull of 7¼ tons was obtained on trials with the engines developing 460 s.h.p. The full-running speed on both engines was 9·47 knots and it may be noted that the same speed was attained on one engine only, this giving maximum economy when the vessel is running light. Again, the same speed can be maintained with both engines running at half-load.—*The Motor Ship, June 1957; Vol. 38, p. 134.*

Israel's Largest Cargo Ship

The *Har Carmel,* recently built by Deutsche Werft, Hamburg, and delivered to Cargo Ships "El Yam", Ltd., Tel Aviv (managing agents, Maritime Overseas Corporation, New York), is Israel's largest general cargo ship and the first of four similar ships for these owners, each to have a deadweight as an open shelterdecker of 12,750 tons (of 1,016 kg.) on a summer draught of 26ft. 8in., or 14,750 tons (closed) on 29ft. 6in. Now on a three-year coal charter, the *Har Carmel* has the following principal characteristics:—

	Metres	*Ft.*	*in.*
Length o.a. ... ... about	157·00	515	0
Length b.p.	144·78	475	0
Breadth, moulded	20·20	66	3¼
Depth, moulded to 1st deck	11·96	39	2⅞
Depth, moulded to 2nd deck	9·51	31	2⅝





Deadweight (open)	...	12,750 tons		
Corresponding draught	...	8·13	26	8
Deadweight (closed)	...	14,750 tons		
Corresponding draught	...	8·99	29	6
Service speed ...	...	14-14$\frac{1}{2}$ knots		

Designed expressly to carry heavy cargo, this is a single-screw vessel with a raked stem, cruiser stern, vertical sides in the 'midship section and with the hull subdivided by eight watertight bulkheads, extending from the double bottom to the first deck. In addition, there are two watertight bulkheads in the 'tweendecks and another at the end of the forecastle. The hull is largely of welded construction although the sheer strake is connected to the stringer plate by double-riveted angle bar; two bilge seams and the lower seam of the sheer strake are riveted. Longitudinal framing has been adopted for the double bottom and upper deck; elsewhere there are transverse frames and beams. The double bottom extends from the afterpeak to the collision bulkhead and is subdivided as shown for the carriage of fuel, fresh water and water ballast. Wooden ceilings are fitted on the tank tops, of double thickness below the hatches. There are six large hatchways serving the cargo spaces, each 10·66 m. by 8·5 m. (34ft. 11$\frac{3}{4}$in. by 27ft. 10$\frac{5}{8}$in.). Steel Simplex-type hatch covers by Deutsche Werft are provided on the upper deck and are moved along on rollers, by means of cargo winches, to stow vertically at the end of the respective hatch. For the hatchways on the second deck there are roller-type steel shifting beams and wooden covers. The main engine is a M.A.N. single-acting two-stroke unit, type K8Z 70/120, with eight cylinders each with a bore of 700 mm. and a piston stroke of 1,200 mm. Of the crosshead design, this has a maximum continuous output of 5,340 b.h.p. at 125 r.p.m., with a mean indicated pressure of 6·1 atmospheres. In accordance with general custom, this engine was installed in the ship without shop trials. This engine is of standard design, with about two-thirds of the scavenge air being supplied by the under pistons and the remainder by a double-acting tandem pump at the forward end of the engine. The scavenge pump consists of two cylinders having a diameter of 930 mm. and a piston stroke of 800 mm.; they are arranged one above the other and the pump is driven from the crankshaft. All necessary arrangements have been made to enable the engine to run on boiler oil, although for the first six months' service Diesel oil only will be used, with Shell Alexia emulsion-type oil for lubricating the cylinders. For accommodation and heavy oil heating at sea, there is a La Mont exhaust-gas boiler with a heating surface of about 150 m.^{2} and a steam generating capacity of $1\frac{1}{2}$ tons/hr. at a pressure of 7 atmospheres. This boiler acts also as a silencer and owing to the low exhaust gas temperature may, at any time, be kept without water. It is arranged to work in conjunction with an oil-fired boiler, one of the two electrically driven circulating pumps, drawing water from the drum of the oil-fired unit and discharging into the exhaust gas boiler where about one-third of the water is evaporated. The remaining water is led back to the drum of the oil-fired boiler which has a heating surface of about 37 m.^{2} and a steam capacity of $1\frac{1}{2}$ tons/hr. For this boiler also, the two feed pumps (one a standby) are centrifugal electrically driven units, each with a capacity of $2\frac{1}{2}$ m.^{3}/hr. at 10 atmospheres.—*The Motor Ship, March 1957; Vol. 37, pp. 500-503.*

Cavitation Erosion in Diesel Engines

Cavitation erosion or pitting of the water side of Diesel engine cylinder liners can be alleviated by such means as: (1) Adding a corrosion inhibitor to the water; (2) Reducing the vibratory stresses in the affected areas; (3) Making the liner out of a more erosion-resistant material; (4) Coating the liner surface with a more erosion-resistant material; (5) Increasing the water pressure. In many cases, the use of more than one of these methods in combination results in an even more worth while improvement in the liner condition. It appears that resonant vibration of the cylinder liner as excited by the impacts of piston slap and cylinder firing pulses is one of the important factors in causing cavitation erosion. Thus, any measures that can be taken to reduce it bring important dividends. In fact, it has been found that no material can resist pitting if the amplitude of vibration exceeds 0·003in. Two ways to reduce vibration of the cylinder liner are: (1) Reducing piston clearance; (2) Increasing liner wall thickness. Vibration tests have indicated, for example, that liner wall vibrations can be reduced from slightly over 0·006in. for a thin-wall (0·156in.) liner with a production piston to less than 0·001in. for a 2-piece heavy-wall (0·25in.) cylinder with a piston having 20 per cent less clearance. Use of a harder or more corrosion-resistant material for the liner also lessens cavitation erosion.—*SAE Journal, March 1957; Vol. 65, pp. 68-70.*

Bending Moments in Bracketed Beams

The paper describes a method of calculating the maximum bending moments, stresses and deflexion in loaded units comprising panels of plating associated with bracketed stiffeners of the type used in shipbuilding. The rigidities of various types of end supports such as deck beams, longitudinals, etc., are examined in some detail, and a relationship between end fixation and relative rigidity is developed. The effect of the sectional inertia of a bracket on the beam to which it is attached is discussed, and the optimum size of bracket and effective unsupported span of a bracketed beam are deduced. Conditions such as unequal end fixation, pyramidal loading, and twisting and tilting are investigated, and a comparison made of the relative efficiency in fixation of welded and riveted brackets. Some practical examples from typical ship structural units are included, and individual calculations are compared with the published results from the B.S.R.A. Ship Structural Research Establishment at Glengarnock.—*Paper by J. McCallum, submitted to the Institution of Naval Architects for written discussion, 1957.*

Metallic Rectifiers for D.C. Power from A.C. Plants

The development of metallic rectifiers within the past few years has resulted in the more frequent application of rectifiers as d.c. power sources on ships with a.c. plants. The practice at present is to provide each d.c. auxiliary with its own d.c. power supply. Metallic rectifiers are now being used for most applications. However, in some instances, motor generators are preferred for such applications as arc welding, degaussing, and exciter systems. The rectifier is located close to the equipment for which it is to be used, and a power distribution panel or distribution switchboard nearby provides the a.c. power to the rectifier. This practice has obviated the need for a d.c. section of the ship service switchboard and has all but eliminated the d.c. distribution system. Studies indicate that the practice of providing each d.c. auxiliary with its own power supply, rather than using a central d.c. power source, has reduced the cost of the d.c. system by as much as 50 per cent and weight and volume by as much as 40 per cent. Although the total volume of all the rectifiers on a vessel exceeds the volume of motor generators of equivalent capacity, there is an overall space saving as a result of the elimination of the d.c. section of the ship service switchboard. Since the rectifiers are distributed throughout the vessel, most of the space saving is in the engine or generator spaces, where space is at a premium. Likewise, the weight reduction is largely because of the elimination of the d.c. section of the ship service switchboard. When rectifiers are used, the weight of the d.c. system is spread throughout the ship rather than being concentrated in one area. Metallic rectifiers now being used are, for the most part, of the selenium type. Basically, the selenium rectifier is a nickel-plated aluminium base plate coated with selenium over which a low temperature alloy has been sprayed. The aluminium base plate serves as the negative electrode, and the alloy as the positive. Current flows readily from the base plate to the alloy but meets high resistance in the opposite direction. This phenomenon results in effective rectification of an alternating input voltage and current. Since rectifiers have no moving parts and operate silently, maintenance is simple and inexpensive. What little repair or maintenance is necessary can be done by the ship's force. Also, the number of on-board repair parts is reduced considerably. Like motor generators and most other electric equipment, metallic rectifiers are thermally rated devices, a term which means that they generate heat. To operate them at their specified electrical ratings and to expect the desired reliability and low maintenance requires that adequate cooling be provided. The selenium rectifiers—now being extensively used on many ships that have a.c. power plants—have considerably reduced the size and weight of installations for providing d.c. power for necessary applications. However, research has disclosed that a silicon rectifier has even more promising characteristics. Application of these rectifiers is expected to bring an even greater reduction in size and weight.—*N. Horowitz, Bureau of Ships Journal, May 1957; Vol. 6, pp. 20-22.*

Scotch Boilers

A large percentage of failures in Scotch marine boilers on lake boats is caused by cracking of the corrosion fatigue type. In general, corrosion is the dominant factor with much of the corrosion likely to occur during earlier operation without adequate water treatment control. Many of the cracks are thick in relation to their length, probably indicating low frequency of stress application. These stresses, therefore, are probably associated with boiler start-up and shut-down practices. The cracking observed in Scotch marine boilers is frequently called water grooving. These cracks are of the corrosion fatigue type and could not be caused by caustic embrittlement. In raising steam, it would be advantageous to start off with a small coal fire for at least eight to ten hours with no attempt made actually to produce steam. The purpose of this fire is to slowly equalize the temperature gradients within the boiler, thereby reducing the effects of unequal expansion. The ideal way to bring a boiler up to heat would be to install a small pump of about 30 g.p.m. capacity which would take its suction from the bottom of the boiler discharging it into the top of the boiler. This would reduce the water temperature gradients. Because of the small amount of circulation when beginning to raise steam, the bottom of the boiler can be actually cold while the water in the top of the boiler may begin to produce steam. The next four hours should be spent in slowly bringing about larger fires to produce the steam. The overall period should never be less than twelve hours and fourteen would be better. In blowing off a boiler the most advantageous procedure would be merely to shut the steam stops, pull the fire and let the boiler cool naturally, opening up the air crack when there is about twenty-five pounds. However, this would probably take two days to accomplish. The more practical way, and the method that is probably followed by most people on the lakes, is to open the surface blow very slightly and continue the blowing-off of the pressure. When supplying the necessary feed water to the boiler to make up for that which is blown overboard, it should always go through the feed water heater so that there is no shock effect due to the admission of cold water.—*E. G. Johnson and W. E. Zimmie, Proceedings of the Merchant Marine Council, United States Coast Guard, April 1957; Vol. 14, pp. 54-57.*

Conditions for Unstable Rupturing of a Wide Plate

The catastrophic fractures which have occurred in large steel structures in recent years, and particularly in ships, have given rise to a very great deal of research, in which the problem has been approached from many angles. One of these approaches, which has received increasing attention in recent years, has become known as " Fracture Mechanics ", and has for its object the elucidation of the conditions for the initiation and propagation of fractures in terms of the mechanical concepts of force, energy, velocity, etc. At the present time, although the subject has been greatly advanced, and much experimental evidence has been accumulated, it is still under active discussion, and as yet there is not a wholly satisfactory theory for the brittle rupture of mild steel. The object of the present paper is to review the present position, and to consider how the available theories might be extended, at least qualitatively. A qualitative hypothesis, based on the assumption that unstable fracturing settles down to a steady state, is proposed, which shows general qualitative agreement with experience, explains some experimental observations, and reveals a correlation between the Tipper and Robertson tests. The hypothesis requires a great deal more development and experimental support to render it quantitative. It is shown that the hypothesis helps to understand the observed facts of ship fracturing.—*Paper by G. M. Boyd, read at a meeting of the Institution of Naval Architects on 28th March 1957.*

Effect of Superstructure on Longitudinal Strength of Ships

A theoretical analysis is presented of the behaviour, under any distribution of longitudinal bending moments, of ships with superstructures. The stress distribution in the superstructure is complicated by various non-linear effects, not usually considered in conventional strength calculations, such as shear lag in the superstructure, deformation or "slip" in the connexion between superstructure and upper deck, and differential curvature between hull and superstructure due to flexibility of the upper deck. These factors influence the contribution of the superstructure to the longitudinal strength of the ship, and have therefore to be considered in calculating the "efficiency" of the superstructure. Following the general theoretical analysis, a practical method for estimating superstructure efficiency is given. Numerical results are then obtained, which show how superstructure efficiency varies with the ratio of its length to transverse dimensions, with the flexibility of the upper deck, and with the distribution of bending moments applied to the ship. The theory and results concern single-deck superstructures; but the method may evidently be extended to multi-deck superstructures.—*Paper by J. B. Caldwell, read at a meeting of the Institution of Naval Architects on 28th March 1957.*

Welded Bracketed Connexions in Aluminium Alloy Structural Members

The paper explains the theory underlying the behaviour and influence of bracketed connexions as applied to structural problems, and in particular, to ship structures. It also reviews some existing work concerning bracketed structures, their effectiveness in providing rigid joints and the estimation of stresses in such triangular plates. An account of the experimental work describes a series of twenty-six tests performed on a specially designed testing frame. The tests were made on a graded group of structural specimens in aluminium alloy embracing three common structural sections and having different sizes and thicknesses of brackets. The results are analysed, and explanations are advanced to account for some of the anomalies met with in reconciling theory with experiment. In general, the method used for testing bracketed members in structural problems is shown to be satisfactory for such light alloy members and presumably for members in other metals. Such bracketed connexions appear to provide adequate joint rigidity regardless of bracket size, although the corresponding bracket stresses may be very high in small brackets. Some tests show that an effective connexion may also be provided, without the use of brackets, by butt welding adjacent members at their joints. No reliable method of predicting bracket stresses has been evolved, although some semi-empirical suggestions are made.—*Paper by B. P. Opie, submitted to the Institution of Naval Architects for written discussion, 1957.*

Performance of a Series of 16-in. Model Propellers

Thirty propellers made to the same design, but having blade area ratios from 0·5 to 1·1 and face pitch ratios from 0·6 to 2·0, have each been tested at six cavitation numbers over a wide range of slip. The results are presented as thrust coefficients, torque coefficients and efficiencies to a base of advance coefficient. Representative photographs illustrate the growth of cavitation with increasing slip and with decreasing pressure. This illustration is generalized by a plotting of the extent of cavitation applicable to the whole series.—*Paper by R. W. L. Gawn and L. C. Burrill, read at a meeting of the Institution of Naval Architects on 27th March 1957.*

Interaction Between Ship's Hull and a Long Superstructure

Owing to transverse distortion of the hull, the deflexion of a ship's superstructure is not, in general, the same as that of the hull sides; the stress in the superstructure therefore differs from that calculated by the simple theory of bending on the assumption that the hull and the superstructure act integrally. This paper gives a method for calculating deflexions and stresses in the hull and superstructure taking account of the differential deflexion. Changes in cross section of the hull or superstructure and in the transverse stiffness of the supporting deck present no obstacle. Only those concepts familiar in the simple theory of bending are used, and shear lag effects are ignored, so that the method is applicable only to long superstructures. The work described was part of an investigation undertaken for the British Shipbuilding Research Association. Differential equations governing the deflexion of the hull and superstructure are formulated, and the relaxation technique is used for their solution. To illustrate the basis of the relaxation technique as applied to the solution of beam problems, the method is applied first to a propped cantilever which can be easily analysed by other methods. A further example illustrates the application of the method to a hull-superstructure combination which has also been studied experimentally. The relaxation technique is presented as a numerical routine and demands no specialized mathematical knowledge. Although the method is one of successive approximations, the calculations should be no more lengthy than some others customarily undertaken in the design office, and the method is sufficiently flexible for an inspired anticipation of the solution to be put to good advantage. In certain simple cases the governing equations can be readily integrated, and such exact solutions enable the accuracy of a relaxation solution to be estimated. The solution of the problem, by this or any other method, entails an assessment of the transverse stiffness (foundation modulus) of the supporting deck, and some suggestions are put forward for estimating this property. Fortunately the behaviour of the superstructure can be described with fair accuracy even if this estimate is only approximate, and where several bulkheads are present it may be reasonable to assume that the foundation modulus is zero.—*Paper by J. C. Chapman, read at a meeting of the Institution of Naval Architects on 28th March 1957.*

Hull Frequency Data and the Influence of Deck Houses

The paper is concerned with the presentation of ship vibration data in a form aimed at enabling quick estimates of frequencies to be made at the design stage. The subject matter represents one aspect of a comprehensive programme of work on ship vibration being carried out by the British Shipbuilding Research Association. Previously published methods have been adapted for this purpose, but in the light of further experimental results obtained by the authors, it has been possible to extend these methods and suggest certain modifications intended to improve the accuracy of the estimates. Detailed consideration is given to the flexural rigidity of ships having deck houses. On the basis of a recently published method for estimating the distribution of longitudinal stresses in deck houses, a series of calculations was carried out for a wide range of ships in order to derive graphs from which inertia correction factors can be readily obtained. Examples are given illustrating the derivation and use of these factors.—*Paper by A. J. Johnson and P. W. Ayling, read at a meeting of the North East Coast Institution of Engineers and Shipbuilders on 1st April 1957.*

Stresses in Deck Houses and Superstructures

The investigation described was undertaken with the object of devising methods of estimating the contribution of deck houses and superstructures to the flexural rigidity of ships. The work represents one aspect of a comprehensive programme of research being carried out by the British Shipbuilding Research Association. Earlier published work on the subject is briefly reviewed. While some of these publications provide valuable information on the mechanics of the problem, it is concluded that most of the solutions proposed have very limited practical application. The problem is discussed in a general way, the various factors influencing the deck house and superstructure characteristics being enunciated. It is shown that simple bending theory is inadequate to deal with the problem and recourse must be made to plane-stress theory in order to provide a foundation for the treatment of the subject. General solutions are obtained in an idealized form of the problem, and by the application of approximate corrections the solutions can be adapted to predict the structural behaviour of any particular type of deck house structure within the limits of the analysis. Owing to the complex nature of the problem it would appear improbable that an exact solution can be obtained. The proposed solutions are of such a nature that the operations could be carried out quite simply in any technical design or drawing office. While results predicted by these methods have given promising comparisons with values measured on ships, further empirical data must be obtained from tests on ships before methods of the type developed in this paper can be applied with confidence to a wide range of ship designs. Several worked examples are given in order to illustrate the sequence of the calculations and to show the general effect of introducing expansion joints into a long deck house. The adaptation of the proposed solutions to the case of the light alloy or composite steel-light-alloy deck house is discussed briefly.—*Paper by A. J. Johnson, read at a meeting of the Institution of Naval Architects on 28th March 1957.*

Dutch Cargo Motor Liner

The cargo motor liner *Alcor,* built by the Werf De Noord N.V., Alblasserdam, has now entered the service of her owners, the N.V. Van Nievelt, Goudriaan and Co.'s Stoomvaart Maatschappij, Rotterdam. The vessel is the first constructed by Werf De Noord for these owners; she is also the first of a series of five ships to be built by "De Noord" for Van Nievelt, Goudriaan and Co. Three of the five vessels still on order are to be identical to the *Alcor,* whereas two other vessels will have the machinery space and living accommodation about one hold-length further aft. Two further vessels will be built by Boele's Scheepswerven N.V., Bolnes, and one by Werf Gusto, Schiedam. The four last-named vessels will have deck cranes instead of derricks. Of the open shelterdeck type with long forecastle and poop, the *Alcor* has the following leading characteristics:—

Length overall	496ft. (151·18 m.)
Length b.p.	455ft. (138·68 m.)
Breadth moulded ...	63ft. (19·20 m.)
Deadweight, about ...	9,200 tons
Gross tonnage	6,734 R.T.
Main propelling machinery:	
One Stork Diesel engine of 7,200 s.h.p. at ...	115 r.p.m.
Service speed	16 knots

The main propulsion machinery consists of a supercharged direct reversible, single-acting, two-stroke, six-cylinder Stork Diesel engine, of the HOTLO type. Each of the cylinders has a diameter of 750 mm. and a stroke of 1,600 mm. The cylinder liners are Porus-Krome hardened. The engine has an output of 7,200 s.h.p. at 115 r.p.m. The pistons of the engine are oil cooled, the cylinder liners, covers and fuel valves are fresh water cooled. The main Diesel engine drives a four-bladed bronze Lips propeller with a diameter of 5,240 mm. and a weight of 13,080 kg.—*Holland Shipbuilding, March 1956; Vol. 6, pp. 28-34; p. 43.*

Diesel Engine Lubricants

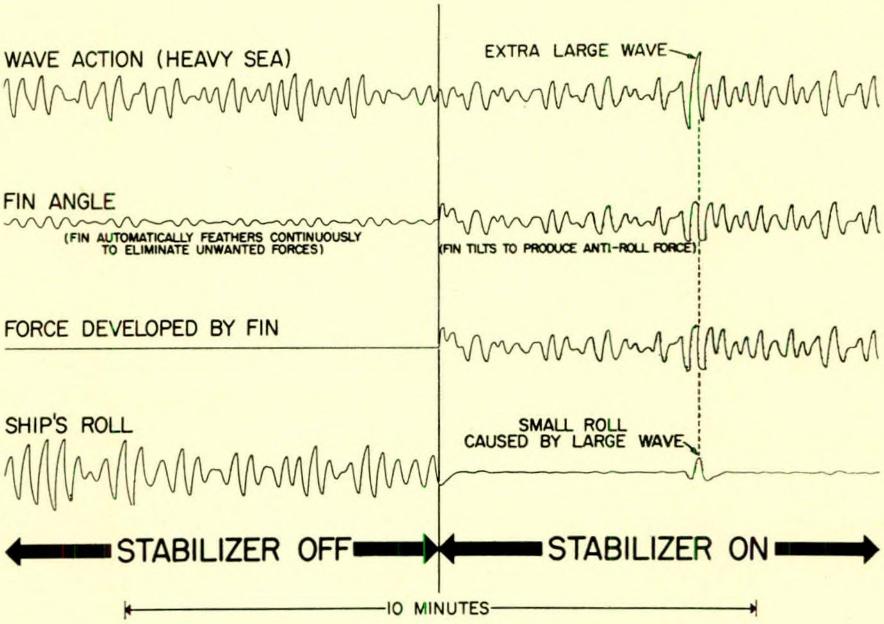
The main functions of heavy duty, lubricating oil additives are to control engine fouling, bearing corrosion, and wear of liners and piston rings. Alkalinity is desirable for the control of wear and is one of the major requirements for the avoidance of piston fouling when conventional organo-metallic additives are used. In a well-balanced oil, sufficient dispersive power and oxidation stability may be incorporated to ensure adequate piston cleanliness and freedom from bearing corrosion provided that the alkalinity level is satisfactory. The alkalinity level falls during service, and for satisfactory performance with certain types of additive in common use it must be kept above a minimum value. For these additives equations are given which enable the variation of the alkalinity level with time in given circumstances to be predicted approximately. The most satisfactory arrangement is to use an oil containing sufficient alkalinity so that the concentration never falls below the critical value. The oil change period is then determined by other considerations, for example, contamination with abrasives. If an oil of lower alkalinity concentration is used, then the equations developed permit an approximate estimate of the oil change period, determined solely from the aspect of additive effectiveness. The application of these results to engines with separate cylinder lubrication is discussed. Thus oils, and, where appropriate, oil change periods, may be selected on a rational basis instead of by trial and error.—*Paper by A. Dyson, L. J. Richards and K. R. Williams, presented at a meeting of the Institution of Mechanical Engineers on 17th May 1957.*

Effect of Resistance and Propulsion of Variations in LCB Positions

This paper describes, and presents the results of, experiments carried out to examine the effect of variation in the longitudinal position of the centre of buoyancy on the resistance and propulsion characteristics of a series of 0·70 block coefficient forms. The experiments were carried out for the British Shipbuilding Research Association by the Ship Division, National Physical Laboratory. The resistance experiments were carried out on five models covering a range of LCB from 1 per cent L aft to 2 per cent L forward of amidships BP. Complementary self-propulsion tests were carried out over a range of LCB from amidships to 2 per cent L forward of amidships. Three propellers were used on each model tested to examine the effect of propeller diameter. The experiments show that up to a speed-length ratio of 0·70 the LCB position is not critical as regards resistance, but above this speed LCB amidships appears to be the optimum for this series. The effect of variation of LCB on the components of propulsive efficiency is small, the effect of variation of propeller diameter and speed being more marked.—*Paper by R. E. Blackwell and G. J. Goodrich, read at a meeting of the Institution of Naval Architects on 27th March 1957.*

Liner with Gyrofins

The Matson liner *Mariposa* of 20,000 tons, which is equipped with Sperry gyrofins, accomplished the round trip to Australia and New Zealand in six weeks, averaging slightly under 20 knots. She was at sea for thirty-six days, and was stabilized for twenty-eight. Considering periods of stabilization, the first night out was characteristic. She met moderate to heavy seas a few hours after departure, and, for the better part of the night, water was taken over the starboard bow. Rolling and pitching of a liner on her maiden voyage obviously is not inducive to passenger comfort, so her speed was reduced a knot or so to ease pitching to a comfortable level. Application of the gyrofins made rolling imperceptible. A few hours ahead of the *Mariposa,* also bound for Honolulu, was the *Hawaiian Planter,* a C-3, which in form and displacement is somewhat similar to the Mariner hull from which the *Mariposa* evolved. The *Planter's* skipper claimed a 25-deg. average roll throughout this weather. The *Mariposa's* movement never exceeded 3 degrees. The gyrofin system develops righting forces by lift generated by two hydrofoil-fin surfaces positioned in a streamline. The fins are tilted by hydraulic piston and crank, and have an outreach of 14ft., and a chord width of 7ft. They are retractable, are swung back into the hull by a system of rams and cross head. A required degree of lift, and speed of its application, is computed by sensing instrumentation which combines measurements of the instantaneous angle of roll, the speed or rate at which it is rolling, and the changes in rolling rate, or acceleration. For about half the stabilized time, lifts required to counter rolling were in the range of 30 to 40 tons per fin. Lighter weather required less than this amount, while for three days in the Tasman Sea, the system operated to its full extent, producing its limit of 70 tons for about 30 per cent of this time. During this period, the ship encountered short, high seas and heavy swells on the starboard bow, and recorded pitch and heave motions indicated a stem movement of about 28ft. The gyrofin system restrained the vessel's rolling motion to an average roll of 4 degrees, measured vertical to out. Weather conditions



Typical recordings taken from extensive computer tests show effectiveness of stabilizer against heavy sea. Top line represents wave action of a rough random sea. Bottom line shows (left) rolling action of ship (about 15 deg.) that normally occurs in such a sea when the stabilizer is turned off; and (right) how this heavy roll is immediately reduced to about 1 deg. when the stabilizer is turned on. The stabilizer can maintain this steadiness except for an occasional extra large wave (upper right) that may result in a small, 5-deg. roll.

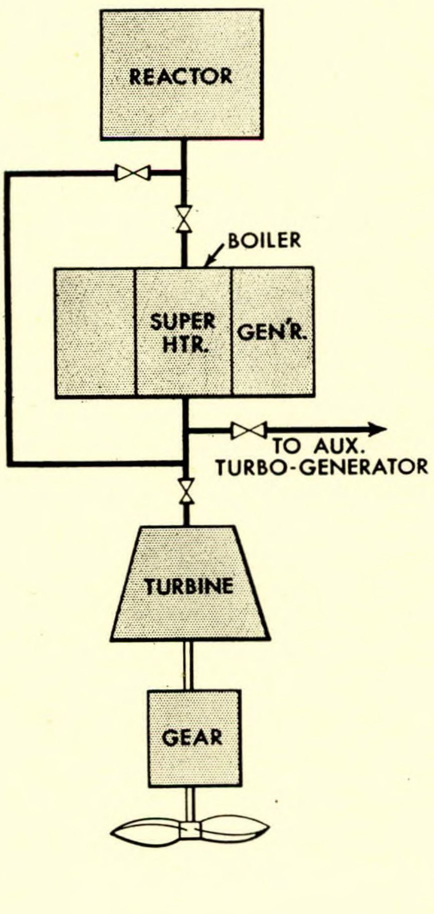
included squalls, with wind velocities to 50 knots. Ship speed was 17 knots. While this particular sea condition cannot be considered an optimum for measurement of stabilization efficiency (i.e. stabilization against beam sea and synchronous rolling) it did prove the success of the most important of the gyrofin design concepts. This is the concept of "lift control", wherein the fin is positioned in the stream until the required lift is developed as measured by strain gauge deflexion of the fin and its shaft when under load. Model tests indicated that a lift of 70 tons would be realized with the fin positioned to a 16-deg. angle of attack against a horizontal streamline. Sea trial investigation generally confirmed this. Relative vertical motion of the water from waves, pitching, or heaving, causing the streamlines to shift from the ideal, and producing greater or lesser lifts than anticipated, gives rise to the concept of false angle of attack. Tracking of this streamline shift by the fin is therefore necessary for it to produce exactly its ordered force. For the noted heavy sea condition, both fins were observed to be running between their positional limits of ±23 degrees relative to the base line of the ship, and can be considered to have been tracking and cancelling false angles of about 6 to 8 deg. Without this capacity, it can be deduced that lift losses would have amounted to 30 per cent of the system capacity. Likewise, false angles of this magnitude could develop equivalent overloads on the fin and structure, producing additional undesirable fatigue loadings. Closely allied with the problem of overloading is that of the generation of flow separation and its resulting noise. Again, model tests indicated rapid deterioration of lift capacity at angles of attack in excess of 18 degrees. It is at this point that the effect of hydrodynamic separation begins to be felt in earnest. Inasmuch as the gyrofin control system includes the ability to measure the instantaneous false angle and to compensate for it, exceptionally bad separation angles of attack were not attained. Separation noise, discernible as low rumbling against the hull, was noted but momentarily and only during heaviest weather. On occasion, observation of the fins under water did show a separation trail from the fin tip, and from the inboard quarter span of the fin flap. The most obvious measurement of successful stabilization is, of course, the comfort of the resulting ride. This, however, has its subtleties; in particular, to try to determine whether comfort depends more on the angle of inclination or on the stiffness or roll acceleration. The control system provides for a balance between these two sensations by varying the proportion of roll-rate signal obtained from a rate gyroscope, as against the signals produced by a linear accelerometer, and a rotary accelerometer. The first of these is used to measure roll angle for extremely low rolling frequencies. The inclusion of an instrument to measure rotary accelerations proved useful in anticipating the erratic roll movement of the ship due to yawing seas, and in producing smooth stabilization in seas encountered forward of the bow. Throughout the trip the balance of heel angle *versus* accelerations was comfortably maintained for all changes in sea conditions.—*R. C. Strasser, Marine Engineering/Log, May 1957; Vol. 62, pp. 72-73, 120, 122.*

Russian Atomic Icebreaker

After a brief discussion of the types of machinery used in icebreakers, some details are given of the Russian nuclear-propelled icebreaker now under construction. The propulsion plant for this ship will be turbo-electric, the steam for the turbines being raised by heat liberated by nuclear fission. The turbo-generators, working at 1,200 volts d.c., will supply three propulsion motors driving three stern propellers; the centre motor will absorb 50 per cent of the total propulsive power, and the other two a quarter of the total power each. The centre motor, which is to have a continuous rating of 20,000 h.p., will be the largest d.c. marine motor in existence; it will weigh about 180 tons, will be about 24ft. long, and will have a casing diameter of about 14·5ft. The turbo-generators will be housed in two separate compartments amidships, and the motors will be aft. The general electric-supply system of the icebreaker will be very extensive; it will include more than 500 electric motors of various sizes and about 200 miles of cable. Much use will be made of fluorescent lighting. The crew will be accommodated in comfortable one-berth and two-berth cabins, and particular attention has been paid to provide good living and recreational facilities. Well-equipped surgeries are provided. Very comprehensive navigational and communication equipment will be fitted, and two helicopters will be carried, for which there will be a hangar and a fuelling station on deck. Special care has been exercised to achieve the most rational methods of hull assembly. There are 176 main prefabricated sections, the largest of which weigh 65-70 tons. A new brand of high-tensile steel is used for all heavily-stressed hull components. The shell-plate thickness in the middle body is 32 mm. (1·26in.), and in the bows and stern it is gradually increased to 52 mm. (2·05in.). Optical marking-off was used for the preparation of hull parts. The block-assembly method will be used for the superstructures, each block being assembled on a base plate, and fitted out as fully as possible on shore. The erection of the blocks on deck will be done after launching.—*Sudostroenie, January 1957; Vol. 23, p. 11. Journal, The British Shipbuilding Research Association, April 1957; Vol. 12, Abstract No. 12,799.*

Nuclear-powered Merchant Ship

The nuclear powered merchant ship sponsored by President Eisenhower and authorized by Congress is now being designed as a passenger cargo vessel. According to published announcements, it is understood that the reactor will be of the pressurized water type, will produce steam at 335lb. per sq. in. gauge, dry and saturated, and will be rated at 60,000 kW (thermal). It will deliver steam sufficient to develop 22,000 s.h.p. in the propelling turbine, and operate the auxiliaries. It is assumed that the designers will depend upon Diesel generators for use in port, as it is understood to be against good reactor practice to operate at fractional loads for long periods of time. From the viewpoints of first cost, operating cost and ability to make port in an emergency, however, its power plant possibly should be more than just a pressurized water reactor to generate dry and saturated steam. Accordingly, it is proposed that the A.E.C. and power plant reactor designers give



serious consideration to a power plant consisting of the following components: One pressurized water reactor, 45,000 kW (thermal rating), generating steam at 335lb. per sq. in. gauge, dry and saturated; one superheat boiler, with separately fired generating section, to raise reactor steam temperature to 825 deg. F., and to generate 40,000lb. of steam per hour at 350lb. per sq. in. gauge, for port use—or at 150 per cent of rating 60,000lb. of steam for emergency use; two auxiliary turbine generators of sufficient capacity to take care of total auxiliary load at sea or in port and one emergency Diesel generator. Such a combination reactor, boiler and auxiliary steam turbine generators would have roughly the following performance: *At Sea.* Steam from reactor 335lb. per sq. in. gauge, dry and saturated. Steam from superheat boiler, 325lb. per sq. in. gauge, 825 deg. F. Steam to turbine throttle, 315lb. per sq. in. gauge, 815 deg. F. Total steam generated by reactor would be approximately 200,000lb. per hr. *In Port.* Steam from generating section of boiler, superheated, 40,000lb. per hr. at 350lb. per sq. in. gauge, 825 deg. F. *At Sea—Reactor Shut Down.* Steam from generating section of boiler at 150 per cent rating, superheated, 60,000lb. per hr. At 350lb. per sq. in. gauge, 825 deg. F., 40,000lb. per hr. would be available for auxiliary purposes, and 20,000lb. per hr. for emergency propulsion. This 20,000lb. of steam per hr. should develop approximately 1,500 s.h.p. or more, giving a vessel speed of over nine knots. Such a proposed power plant would have the following advantages when compared with a reactor steam generator only for sea, and Diesel generators for port use: 1. Reactor would be smaller, weigh less and cost less. 2. No port use Diesel generators would be required. 3. In case of casualty to the reactor the ship could still make port, using steam from the generating section of the superheat boiler. 4. Vessel could manœuvre in and out of port on boiler steam only, cutting in the reactor for major power demand on voyage. 5. Steam from the boiler could be used for warming up the reactor, also for smothering or fire fighting in port. 6. Main and auxiliary condensers would be smaller, weigh less, cost less. 7. Steam piping would be smaller, weigh less.—*Marine Engineering/Log, May 1957; Vol. 62, p. 104, pp. 130-131.*

Propellers in the Wake of an Axisymmetric Body

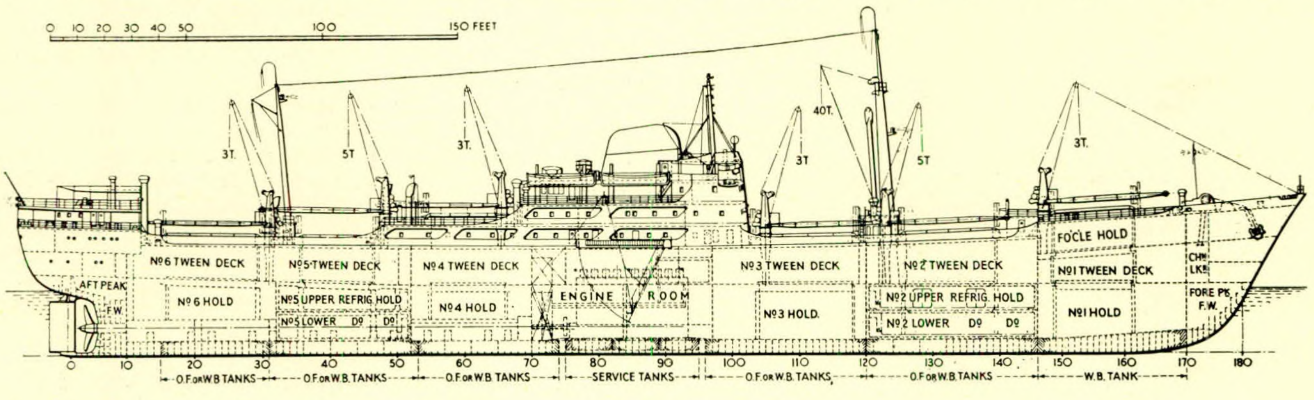
The operation of propellers in the viscous wake of a body is an old problem which has been considered in a number of papers. The problem, however, has never been satisfactorily resolved, and has usually been improperly understood, possibly through lack of basic scientific data. The problem has to be approached in its entirety. For example, the original conception of thrust deduction as the thrust developed by a propeller in a free stream minus the towing drag of the body is obviously wrong when it is considered that a propeller designed to operate most efficiently in a free stream will operate less efficiently in the wake of the body. In addition, it will be seen that all effects in a propeller body system are interactive and that if the propeller induces a drag on the body, the presence of the body will similarly affect the performance of the propeller, even though it has been originally designed to operate in the wake of the body. Secondly, it seems certain that no realistic or accurate design theory can be constructed unless fairly elaborate mathematical methods based on fundamental hydrodynamic data are used. It would seem therefore that little of practical value can be attained by making vitiating simplifying assumptions. Also it is hard to see how it is possible to construct a design theory from the results of a series of tests. The number of parameters involved would make such a task impossible. This paper therefore was written with the intention of developing an accurate design theory of wake-adapted propellers and of propeller-body combinations in general. In it the use of the new hydrodynamic facilities and of the PEARL high-speed computer at the Admiralty Research Laboratory, Teddington, is envisaged, the theory to be tested and modified in the light of experimental verification of its predictions. The kind of body used in this paper is a body of revolution co-axial with the propeller axis of rotation. This configuration would apply to underwater missiles and possibly to submarines. No account is taken of the effect of the surface of the sea, but the theory has been developed for a contra-rotating propeller of which the case of a single screw is a simplification.—*Paper by R. Hickling, submitted to The Institution of Naval Architects for written discussion, 1957.*

German-built Cargo Motorship for Swedish Owners

A recent delivery from the yard of H. C. Stülcken Sohn, Hamburg, was the cargo motorship *Elgaren* built to the order of Rederiaktiebolaget Transatlantic, Gothenburg. The vessel which is shown in the general arrangement is a high class cargo ship with accommodation for twelve passengers and embodies a number of features not usually included in vessels of this type. The principal particulars are:—

Length overall	469ft. 9in.
Length b.p.	429ft. 9in.
Breadth	63ft. 0in.
Depth to shelter deck	38ft. 0in.
Draught	23ft. 7½in.
Gross tonnage	5,737 tons
Deadweight tonnage	7,750 tons
Speed	17 knots

Of the open shelter deck type, the vessel was built to the requirements of Lloyd's Register, and is strengthened for navigation in ice. Six cargo holds and 'tween decks are arranged, three forward and three aft of the machinery space. One of the novel features of the *Elgaren* is the provision of twelve deck cranes for cargo handling. These cranes of M.A.N./Siemens manufacture include eight of 3-tons and four of 5-tons capacity. There is also a 40-ton derrick arranged at No. 3 hatch operated by two 8-ton electric winches of Siemens/Schärffe make. The same firm also supplied the windlass, designed for the handling of $2\frac{7}{16}$-in. anchor cable. MacGregor steel covers are fitted to all hatches on the weather decks. The propelling machinery consists of a nine-cylinder, single-acting, supercharged, two-stroke M.A.N. Diesel engine, type K9Z 70/120C, developing 7,500 b.h.p. at 125 r.p.m. Electric current at 220 volts d.c. is supplied by Siemens generators.—*Shipbuilding and Shipping Record, 18th April 1957; Vol. 89, p. 503.*



Cargo ship Elgaren

Electric Windlasses on Modern Ships

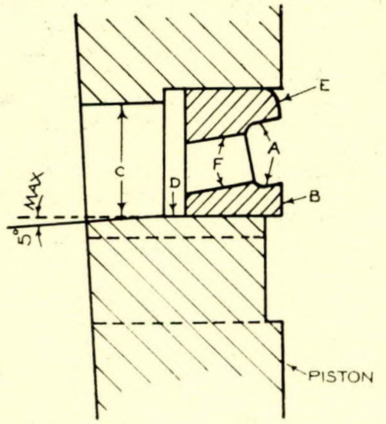
The author first reviews the historical development of capstans and windlasses. Apart from certain special applications, steam drive has now been superseded by electric motors. Modern windlass drives incorporate electromagnetic contacts and relays, which render them semi-automatic; the amount of manual switching required is thereby greatly reduced. Whatever the nature of the mains supply, or the type of motor employed, suitable contactor arrangements permit an approach to the most desirable speed-torque characteristic. The shape of this characteristic is described with reference to the operations of heaving short, breaking out, lifting, etc., and to the mechanical construction of the windlass. Besides the gypsies, the latter usually incorporates two auxiliary drums for the warps; it must be possible to drive these at relatively high speed and low torque. A windlass drive must be reversible, and operate reliably over a wide range of loads and speeds. It must be able to stall safely if a certain torque is exceeded, and to re-start automatically when the torque is subsequently reduced. In France, some progress has been made towards standardization of small and medium-sized windlasses for chain sizes up to 73 mm. (about 3in.) and nominal (breaking-out) powers up to about 100 h.p. Some details are given concerning recommendations of the Comité d'Organisation de la Construction Navale, and the range of standard contactor switchgears manufactured by La Télémécanique Electrique for d.c. windlass drives. There are detailed descriptions of the circuits and control gear used with several types of d.c. or a.c. windlass drives. Vibrating relays aid in maintaining constant power at varying speed. With d.c. supply, shunt motors appear to be preferable to series motors; they are lightly compounded at the higher powers. With a.c. mains, the motor most frequently employed is the asynchronous type with wound rotor and two sets of poles, pole-changing taking place in both rotor and stator. Rheostatic control is possible with each set of poles. Such motors have better characteristics than the squirrel-cage type but require more complex switchgear. The performance is always inferior to that of a similarly-rated d.c. motor and problems arise in providing a high speed for the warping drums. For large high-power windlasses, Ward-Leonard control is by far the best solution; it can be used on both d.c. and a.c. ships. In the C.G.T. liners *Flandre* and *Antilles,* seventeen deck auxiliaries are fed from five converter sets in two groups, one forward and one aft. Four of these sets have two generators each, the second generator being switched in when it is desired to operate a windlass. With such an arrangement it is not possible to operate all the auxiliaries simultaneously but this is never necessary.—*P. Sizaire, Publication of La Télémécanique Electrique, 1956. Journal, The British Shipbuilding Research Association, April 1957; Vol. 12, Abstract No. 12,813.*

Diphenyl as a Thermodynamic Fluid

This article presents the results of a new investigation into the use of diphenyl as the working fluid in a heat engine. This information should prove to be of interest for two basic reasons. First, experience with the Organic Moderated Reactor Experiment may lead to consideration of a boiling-diphenyl reactor, similar in principle to the boiling water reactors. Second, diphenyl may prove to be advantageous as an intermediate fluid for liquid-metal-cooled reactors. Diphenyl will not react violently with liquid metals such as bismuth, sodium, or NaK, nor does it react with water or steam. Diphenyl has certain limitations as a thermodynamic fluid. Thermal decomposition will limit the temperature of diphenyl cycles to somewhere in the neighbourhood of 750-800 deg. F. Radiation damage may be more of a problem where a thermodynamic cycle is involved than where diphenyl is used simply as a coolant because of changes in vapour pressure. In addition, diphenyl is inflammable. Thermodynamically, diphenyl resembles steam in that both the liquid and vapour phases are encountered in the cycle, and in that most commonly pressures used are below the critical pressure. A substantial portion of the heat absorbed in both steam and diphenyl cycles is absorbed during boiling. A diphenyl turbine should be somewhat more efficient than a steam turbine since its volume flow is quite high and there is no moisture problem to contend with in the last few stages. On the other hand, its efficiency may not be so good as a gas turbine's, because the volume flow rate varies over a wide range from the first to last stages. Heat transfer in a diphenyl regenerator should be better than that in an air regenerator since the fluid in the cold side is a liquid. One of the major limitations of diphenyl is its low vapour pressure, which requires extraordinarily low condenser pressure to permit reasonable condenser temperature. In this study it was found that 1lb. per sq. in. absolute was about as low as one could reasonably expect to use. At this pressure, the turbine-exhaust volume flow is about the same as that for a steam turbine. Using appreciably lower condenser pressure would require a larger turbine. High-temperature fluid, and steam as the low-temperature fluid, were investigated. It was hoped that this approach would lead to increased thermal efficiency. In this type of cycle, the diphenyl loop can be regarded as an intermediate loop with one added component—the turbine. By using this arrangement, a thermodynamic advantage can be gained from the existence of the intermediate loop. The diphenyl portions of the binary cycles are similar to the cycles with 85 per cent regeneration, except that the exhaust pressure has been raised to 5lb. per sq. in. absolute. The steam cycle uses saturated steam at 200lb. per sq. in. absolute, with a condenser vacuum of 2in. of mercury, absolute. An alternative arrangement, eliminating regeneration in the diphenyl loop and using superheated steam, was rejected because it would have involved a gas to gas heat exchanger. A series of calculations using an initial diphenyl pressure of 200 lb. per sq. in. absolute showed that varying diphenyl condenser pressure and steam boiler pressure did not influence the cycle efficiency or the steam flow appreciably. Use of a diphenyl-steam binary cycle will yield excellent thermodynamic efficiency. It may be a logical choice where the reactor plant requires the use of an intermediate loop.—*D. C. Purdy, Nucleonics, April 1957; Vol. 15, pp. 109-112.*

Piston Ring

The operation of the "Inertia-Flow" oil-control ring differs substantially from that of the orthodox slotted scraper ring which gives adequate oil control under normal operating conditions. Modern design tendencies towards higher rotational speeds, involving shorter connecting rods and pistons with a reduced overall length, often promote conditions whereby oil fling is outside the capacity of the orthodox type of slotted scraper ring; under such conditions the ring grooves and drainage holes are overloaded with oil which is actually returned to the liner via the upper face of the ring, and is eventually lost to the combustion chamber. Whilst the "Inertia-Flow" ring provides an efficient two-way scraping action, it takes advantage of the influence inertia has on the oil droplets on the *upward* stroke and so obtains the maximum scavenging action in that direction, instead of on the downward stroke as in the case of the orthodox type of slotted ring. On the upward stroke oil scraped from the cylinder wall is forced rapidly through the drain holes in the ring to the escape holes drilled in the piston, adjacent to the lower wall of the groove. From the section illustration it will be seen that the annular groove (*A*) in the ring is machined at an angle to promote inward drainage; this not only ensures a rapid clearing of the oil from the groove but also gives the upper edge of the lower land (*B*) extremely efficient scraping properties in an upward direction. The oil draining holes (*C*) leading from the piston ring grooves are also inclined in the same direction and are flush with the lower edge of the groove (*D*) to eliminate the formation of a reservoir where oil might accumulate. The upper edge of the top land (*E*) is radiused and chamfered to allow the oil to pass freely over the periphery and so avoid any upward scraping action. Unlike some special rings, the "Inertia-Flow" design does not rely on extremely high radial loadings which may be obtained by the fitment of expander springs or the adoption of narrow lands. It is well known that all rings carry a penalty in the shape of frictional drag or "parasitic horse power" which is roughly proportional to the radial pressure. Therefore, although increased oil control can be obtained by using rings with a high peripheral pressure, this may be achieved at the expense of mechanical efficiency, plus higher wear on both the ring and the cylinder.—*The Oil Engine and Gas Turbine, April 1957; Vol. 24, p. 445.*



Experiments on Sideways Launching

In the case of trawler forms the experiment technique developed gives satisfactory simulation of full-scale side launching behaviour. The model results now presented cover a wide range of launching conditions and hull forms. As further ship data becomes available it will be possible to extend further our knowledge of model and full-scale correlation, and it is the intention of the authors to present a further publication on this subject. The present work confirms the previous conclusion that bilge keels and similar projections from the hull are extremely effective in reducing angular rotation of a side-launched vessel towards the quay edge. The liability of damage in this condition is thereby reduced. The critical point of no return, when vessels capsize on water entry, has been accurately determined, and it is now possible to predetermine the conditions for a satisfactory side launch. The limiting stability necessary to avoid capsizing has been presented in diagrammatic form. The impact pressures acting on the hull of a vessel when side launched have been shown to have values up to 16lb. per sq. in.[2] for the usual range of launching conditions. It is recommended that in cases where the static drop is likely to exceed 25 per cent of the depth of water, some temporary internal strengthening of the hull be provided. The minimum clearances of the keel from the river bed have been determined over a wide range of launching conditions, and from these results the likelihood of impact damage due to striking the bed or banks of the waterway can be ascertained. The results indicate that vessels of normal form up to 450 tons displacement in the "as launched" condition can be safely side launched in 15·25ft. depth of water. There is no evidence from these results to suggest that absolute size of vessel affects its satisfactory behaviour when side-launched, provided that sufficient depth and width of waterway are available. This proviso, of course, is applicable to an ever greater extent in the case of end-on launching. A theoretical solution for the motion of a side-launched vessel has been developed on the assumption that the vessel behaves in a similar manner to that of a circular cylinder on water entry. The rise in water level produced by the entry of the hull in the water has been neglected. With these assumptions it has been established that the pressure distribution over the hull and the motion of the vessel up to the maximum roll angle are in good agreement with the model results.—*Paper by D. J. Doust and E. Macdonald, read at a Meeting of The Institution of Naval Architects on 6th June 1957.*

Electrical Control of Variable Pitch Propellers

Fig. 1 is a schematic diagram of an electrical remote-control system for variable pitch propellers. The control gear comprises a combined sequence- and time-control system. The variation in the pitch of the propeller is analogous to the movement of the control lever. When the position of this main control lever, located on the bridge, is varied, the regulating unit in a closed control loop comprising an electrical measuring bridge is also varied, with the result that a current in relation to the position of the lever and of definite magnitude and direction flows in the zero arm of the bridge, causing one of the two directional relays in this circuit to respond, which in turn energizes the appropriate contactor and starts the servo-motor. This servo-motor continues to run until the position of the measuring element simultaneously moved by it coincides with that of the regulating unit and until no current flows in the zero arm of the bridge, whereupon the directional relay and motor contactor trip, at the same time initiating a short-circuit braking of the armature to prevent overshooting. On account of the closed loop with sequence control, any movement of the positioning element not resulting from a direct command from the bridge is immediately and automatically cancelled. The setting accuracy of this sequence control is 98·5 per cent. A changeover to time control is effected through a selector switch on the bridge. Hereby, the variation in pitch of the propeller is dependent upon the length of time a contact signal is given by means of the toggle lever accommodated in one housing, together with the main control lever. This permits the servomotor

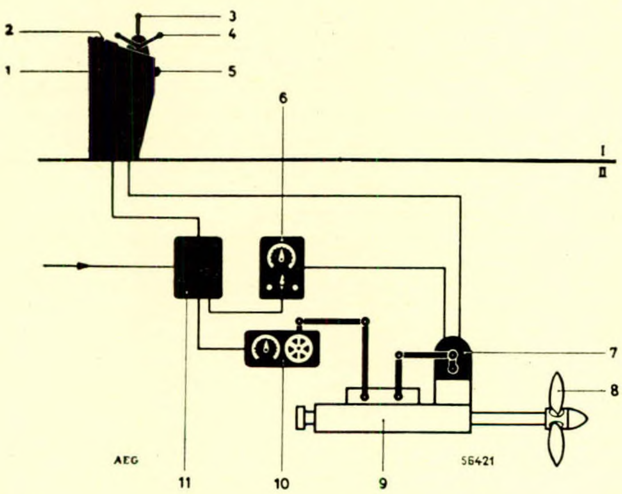


FIG. 1—*Schematic diagram of electrical remote-control system for variable-pitch propeller*

(I) Bridge; (II) Engine room and shaft tunnel; (1) Bridge desk; (2) Electrical pitch indicator; (3) Main control lever; (4) Reserve control, toggle lever; (5) Control selector switch; (6) Electrical pitch indicator with reserve time-control and control-position selector switch; (7) Remote electrical pitch transmitter; (8) Variable pitch propeller; (9) Hydraulic control gear; (10) Electrical regulating gear with mechanical pitch indicator and emergency control; (11) Contact amplifier

to be operated directly without any intermediate relays and contactors. On the electric position indicator the blade pitch adjustment can be continuously checked and, upon the desired angle of pitch being attained, the lever released, whereby the servomotor is immediately brought to a standstill.—*H. Hinterthuer, AEG Progress 1957; No. 1, pp. 25-27.*

Passenger Liner *Gripsholm*

Delivery of Scandinavia's largest passenger liner took place after successful trials off Rapallo, east of Genoa. The new vessel, the *Gripsholm,* 23,500 tons gross, built by Ansaldo S.A., Genoa, for the Swedish American Line, is similar in many respects to the *Kungsholm,* built at the de Schelde yard, Flushing, in 1953. The principal particulars of the *Gripsholm* and, for the sake of comparison, the *Kungsholm,* are as follows:—

	Gripsholm	*Kungsholm*
Length o.a., ft.	631	600
Length, b.p.	550ft. 2½in.	530ft.
Breadth moulded	81ft. 8¾in.	77ft. 1in.
Draught, loaded	27ft. 1in.	26ft. 5¾in.
Gross tonnage, tons ...	23,500	22,000
Service speed, knots ...	19	19
Passenger accommodation:—		
First class	150	176
Tourist class	612	626
Interchangeable ...	80	—
Dining room seats cruising	450	426
Cargo capacity (bale), cu. ft.	76,850	120,000
Refrigerated capacity, cu. ft.	5,650	10,500
Machinery output:—		
Main engines, s.h.p. ...	16,200	14,000
Generators, kW ...	3,500	3,300

As in the case of the *Kungsholm,* welding has been used fairly extensively. The shell plating is welded at the butts and in three longitudinal strakes, but is riveted at the frames and seams. All the decks are welded. Both welding and riveting have been used in the construction of the bottom of the ship. One of the main differences between the two ships is that the *Gripsholm* is fitted with Denny-Brown stabilizers: the *Kungsholm* was fitted with a special anti-rolling tank in which free water could run from one side of the ship to the other in order to counteract rolling. The main propelling machinery in the *Gripsholm* consists of two nine-cylinder Gotaverken two-stroke, single-acting Diesel engines of 760-mm. bore and 1,500-mm. stroke. The total normal output of the two engines is 16,200 s.h.p. at 112 r.p.m., but this can be increased to 18,600 s.h.p. at 117 r.p.m. if required. Contrary to Gotaverken's normal practice the frames and columns of these engines have been cast instead of being of welded construction. The engines have been designed to run on Diesel oil as well as on heavy oil with a viscosity of up to 3,000 sec. Redwood No 1 at 100 deg. F. Purifiers for the heavy oil for the main engines as well as for the auxiliary engines are installed in the shaft tunnel. The auxiliary engine room is arranged forward of the main engine room and contains the electrical plant and two auxiliary boilers. The electricity supply is 3-phase, 60-cycles alternating current, with insulated neutral, and is as follows: 440 volts supply for the larger electric motors in the deck and engine room departments, 220/127 volts for smaller electric motors and 24 volts for bells and other low-voltage installations. Direct current for the winch motors is obtained through converters. The power is supplied by five 700-kW alternators, each driven by a six-cylinder, four-stroke Ansaldo Diesel engine. These engines have also been designed to run on heavy fuel oil. There is also a 150-kW emergency Diesel-driven alternator installed in the forward funnel. The Diesel engine is a four-cylinder, four-stroke engine made by Ansaldo. The two boilers in the auxiliary engine room are equipped with automatically-controlled oil firing, and have a maximum working pressure of about 114lb. per sq. in., and a capacity of about 8,820lb. of steam per hour each. In addition there are two waste heat boilers installed in the engine casing, each with a capacity of about 6,615lb. of steam per hour at a maximum working pressure of 147lb. per sq. in. To meet the intensive demand for fresh water, two low pressure, two-stage evaporators have been installed in addition to the supply in the ship's tanks. Each of these evaporators has a capacity of 175 tons per 24 hours.—*The Shipping World, 5th June 1957; Vol. 136, pp. 541-544.*

Model Tests for Saunders ABC Ships

This report contains the results of a complete model testing programme on a unique design of a 510-ft. passenger cargo vessel. This design embodies two alternative hull forms, using single-screw propulsion. The forebody is the same for each form and features a large ram-bulbous bow. The afterbody of one has a transom stern with a centre line skeg; the second has an arch or tunnel stern with double skegs. The model test programme included resistance tests with and without appendages, self-propulsion tests, wake survey, lines of flow and wave-profile test, flow studies in the circulating water channel, change-of-level readings, and determination of bilge keel trace. The results of these tests are presented in graphical form. The hull form with the transom stern gave excellent performance. It has several distinct advantages and can compete successfully with the normal cruiser-stern type merchant ship. The performance of the arch or tunnel stern vessel was satisfactory for its type, but did not equal the efficiency of the transom stern. This form may be useful for special types of services.—*E. R. Meyer, David Taylor Model Basin; Report No. 1006, February 1957.*

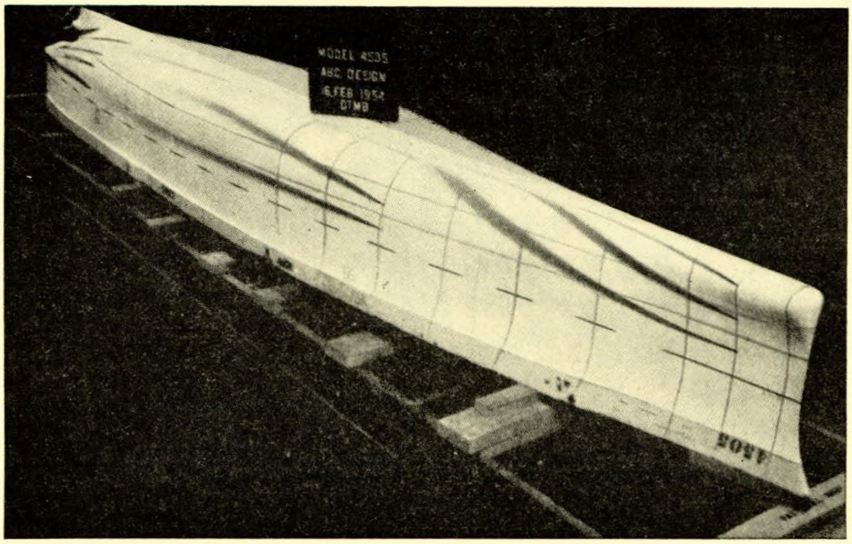


FIG. 21—*Lines of flow on forebody, Model 4505*

Structural Steels for Warships

Scientific and technical developments are revolutionizing weapons, equipment, and the warship hulls which carry them into service. Of the many problems which these changes are bringing to warship design, two particular aspects, namely, developments in hull structural design and structural steels, are dealt with in this lecture. It describes some of the special problems in the hull strength of surface warships and submarines—particularly those due to the dynamic loading set up by underwater explosions—which led to the inauguration of the Naval Construction Research Establishment and the provision of special facilities, some of which are unique. Reference is made to the theoretical and experimental research work which is being undertaken to obtain a better appreciation of the way various design features of warship hulls resist the static and dynamic forces imposed upon them, and special emphasis is given to the post-war research on steel materials, welding, and the phenomenon of brittle fracture. These researches are being undertaken with the principal objects of avoiding brittle fracture in normal structural steels and of developing high-yield steels for special applications in warship construction. The difficulties that have been met in the development of these steels are discussed, including the problems of welding them under shipyard conditions. The discussion also brings out the need for more modern plant in our steelworks. This research work is directed not only towards the building of warship hulls that are capable of withstanding the forces of nature and enemy attack, but also towards achieving these aims with minimum hull weight, in order that the largest military loads may be carried.—*Paper by Sir Victor Shepheard, K.C.B., R.C.N.C., read as the 25th Andrew Laing lecture before the North East Coast Institution of Engineers and Shipbuilders on 22nd March 1957.*

Patent Specifications

Arrangement for Shipping Freight

This invention relates to water-borne freight movements and in particular to a method for storing cargo aboard an otherwise single-purpose vessel; it also relates to an apparatus which allows the pay load of such vessels to be greatly augmented. The invention provides a structure well above the main or weather deck of a tank ship (Figs. 1 and 2) and this structure is adapted for the expeditious loading and securing on it of a number of containers. These containers are arranged in rows extending the length of the vessel. The usual amidship house or bridge structure is placed on the forecastle deck and as far forward as may be consistent with good ship design practices. The containers are of sturdy construction capable of withstanding the forces to be encountered on the open seas and are adapted to receive the freight to be transported. In a specific, and particularly advantageous, embodiment of the invention each container may be of a construction adapted to receive a complete truck-trailer fully loaded with weight.—*British Patent No. 775,527, issued to M. P. McLean. Complete specification published 22nd May 1957.*

FIG. 1

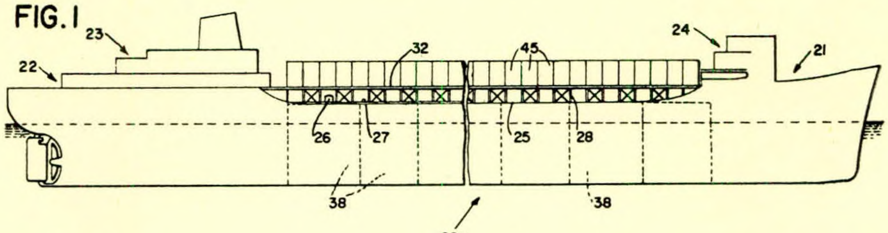
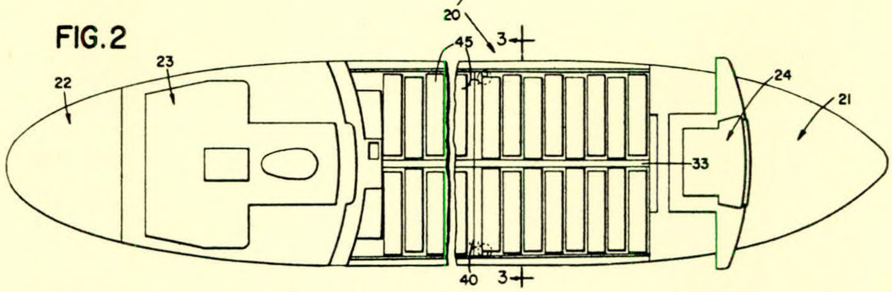


FIG. 2



Oil Separator Craft

The oil separator craft to which this invention refers has been developed in order to salvage substantial quantities of fuel oil which are found in oil tank cleaning slops, oily ballast

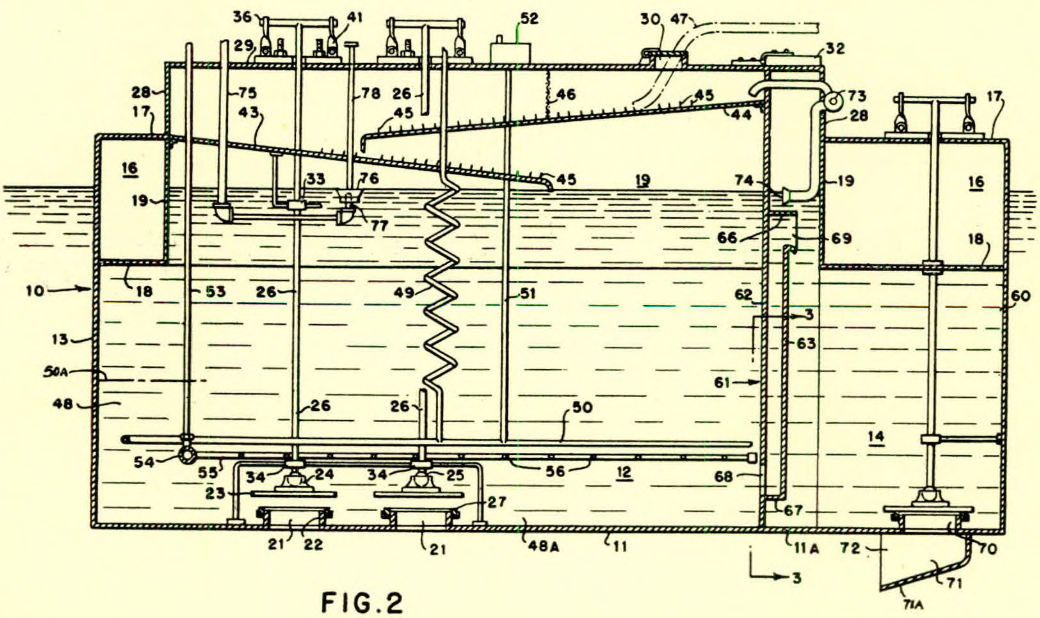


FIG. 2

water, or water contaminated oil cargoes. The oil separator craft is seen to have a normally submerged tank portion (Fig. 2) for the reception of oil and water to be separated. The tank portion also has one or more openings in its bottom, normally open, and means for closing these openings. The craft has buoyancy compartments which maintain it afloat. The improvement comprises a deck covering the tank, an inclined baffle plate in the tank below and adjacent to the deck. There is a screen above and in contact with the baffle plate. The deck also has a manhole above the baffle plate and screen and provides access for discharge of oil and water on to the baffle plate above the screen. Also, the manhole provides access to the material collected by the screen.—*British Patent No. 776,422, issued to A. Y. Lanphier. Complete specification published 5th June 1957.*

Ships' Hull Construction

This invention relates to ships' hull construction and has for its object the provision of a ship's hull which will reduce skin friction, as concerns a comparable ship's hull. The proposed hull (shown in Figs. 4 and 5) has a generally flat bottom beneath which are formed two air tunnels formed by three downwardly extending projections, two of which are extensions of the sides of the hull. Preferably air is forced in the bottom of the hull at the bow, which air is directed into the tunnels U_1 and U_2 in a direction substantially parallel with the bottom of the hull and towards the stern. Means are to be provided at the stern of the ship to allow for exit of the air from the tunnels in such a manner that turbulence in the water around the stern of the hull is prevented. Preferably also the supply of air under pressure to each tunnel is separately controlled, whereby a layer of air in the two tunnels along the bottom of the hull can be maintained and, if necessary, the air in one tunnel can be at a pressure different from that in the other

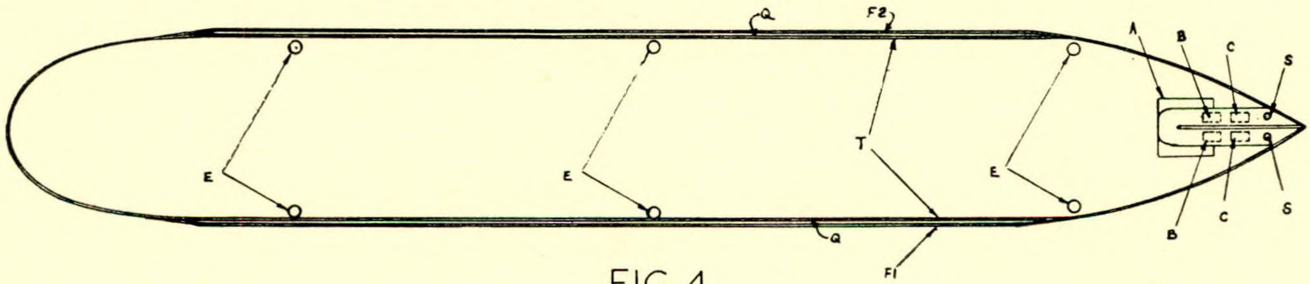


FIG 4

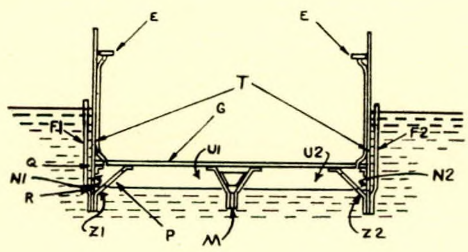
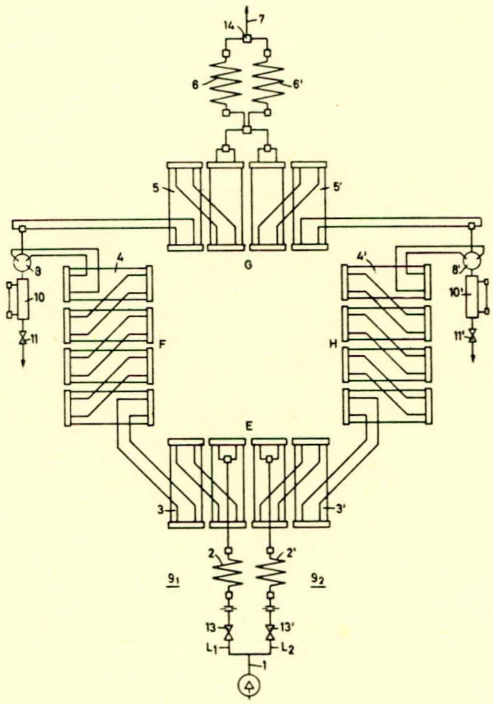


FIG 5

tunnel, thereby providing means for counteracting the rolling of the ship. The means for supplying air under pressure may consist of a turboblower (A) or the like, with a damper control, mounted within the ship adjacent the bow.—*British Patent No. 776,000 issued to K. F. Williams. Complete specification published 29th May 1957.*

Removing Salt from Once-through Boiler

Owing to the deposition of salt in forced-flow boilers,



it has been usual to arrange for the final stage of evaporation to take place in a region in which the flue gas temperature is relatively low. However, now that forced flow boilers can be supplied with water substantially free from salts, it is possible to arrange for the final evaporation to take place in the combustion chamber. The advantage of this arrangement is that the region of relatively cool flue gases which formerly contained the final evaporation section of the boiler can now accommodate a feedwater heater of appropriate size so that the heating surface of the boiler can be reduced and the boiler efficiency increased. This invention is concerned with the problem of salt removal in the case of large forced flow once-through boilers in which the boiler tubes are subdivided into two parts which operate in parallel, and which can be regulated separately. Referring to Fig. 6, the boiler comprises two heating surface systems (2 to 6) and (2^1 to 6^1) which operate in parallel and are independently controllable. Feedwater is supplied to these systems through a common pipe (1) by means of a common feed pump, the water entering the systems (2 to 6) and (2^1 to 6^1) through respective pipes (L_1 and L_2). The heating surface includes feedwater preheaters (2 and 2^1) and radiation evaporator heating surfaces (E, F and H). The last heating sets of the surface E are designated by (3 and 3^1) and the last heating sets of the surfaces F and H are denoted by (4 and 4^1) respectively. In the example it is assumed that (4 and 4^1) lie within the presuperheating range of the radiation part. The salt separating apparatus, consisting of the separators (8 and 8^1), and the separator containers (10 and 10^1) are arranged after the last heating surface sets (4 and 4^1) which are associated with the radiation presuperheating zone. Whilst the separated salt-containing feedwater leaves the separator apparatus by way of valves (11 and 11^1), the already partially superheated steam enters inlet sets (5 and 5^1) of the radiation superheater (G). From here on, it is supplied after further superheating in the heating surfaces (6 and 6^1), which are situated in the convection section, through a header (14) and pipe (7) to the turbine. If the final evaporator zone of the left-hand part of the boiler is to be flushed, the flow regulating valve (13) is opened further and a relatively large quantity of water enters this part of the boiler. After the last heating set (4), the steam is no longer dry saturated or slightly superheated but is wet so that the whole of the tubing of the heating surface (F) is flushed. At the same time, the supply of heat to the right-hand portion of the boiler can be increased in order to establish the original working temperature again at the boiler outlet by mixing the two streams of steam.—*British Patent No. 770,091, issued to Siemens-Schuckertwerke A.G. Complete specification published 13th March 1957. Engineering and Boiler House Review, June 1957; Vol. 72, p. 215.*

Marine Engineering and Shipbuilding Abstracts

Volume XX, No. 10, October 1957

* Patent Specifications

Japanese-built Indian Cargo Vessel

Recently completed by the Hitachi Shipbuilding and Engineering Co., Ltd., at their Sakurajima Shipyard, Osaka, for the Great Eastern Shipping Co., Ltd. (mgrs. A. H. Bhiwandiwalla and Co. (Bombay), Ltd.), the motor cargo vessel *Jag Laxmi* is the largest dry cargo motor ship in the Indian registry. She is the ninth vessel in the fleet of her owners which was founded in 1949 with the acquisition of a Fort-type steamer. This fleet now comprises five motor ships and four steamers, and two motor cargo vessels are on order. The *Jag Laxmi* is designed for service both as a tramp and as a cargo liner. The vessel has been built under the supervision of Lloyd's Register to comply with the class ✠ 100 A1 and + LMC. She has two continuous decks, being designed as a shelterdecker with a forecastle and having a raked stem and cruiser stern. There is a continuous double bottom. The following are the main characteristics:—

Length o.a.	489ft. 8in.
Length b.p.	452ft. 9in.
Breadth, moulded	61ft. 8in.
Depth to shelterdeck... ...	38ft. 11in.
Depth to main deck... ...	30ft. 2in.

According to modern practice, the *Jag Laxmi* can be used both as a closed or an open shelterdecker. As a closed shelterdecker the particulars are:—

Deadweight capacity ...	12,970 tons
Corresponding draught ...	29ft. $2\frac{11}{32}$in.
Gross register	8,798 tons
Net register	6,230 tons
Full load displacement ...	17,175 tons

Used as an open shelterdecker the main particulars are:—

Deadweight capacity ...	11,015 tons
Corresponding draught ...	26ft. $3\frac{15}{32}$in.
Gross register	6,472 tons
Net register	4,180 tons
Light weight	4,205 tons

Propelled by a Hitachi-B. and W. engine, the ship attained a trial speed of 17·18 knots when loaded, about 20 per cent as a closed shelterdecker and at continuous full output. The speed, fully loaded, and at three-quarter output, is 14·25 knots as an open shelterdecker and 14 knots (closed), while the speed as a fully loaded closed shelterdecker is about 15 knots at normal service output. The *Jag Laxmi* is propelled by a six-cylinder two-stroke single-acting Hitachi-B. and W. turbocharged engine with cylinders having a bore of 620 mm. and a stroke of 1,400 mm. Running at present on Diesel oil, the engine is to operate on heavy oil after six months' service. Two BBC exhaust-gas turbochargers have been fitted, running at about 6,200 r.p.m. at three-quarters output of the main engine. The engine has a service output of 4,950 b.h.p. at 131 r.p.m., while the continuous full output is rated at 5,400 b.h.p. at 135 r.p.m., with an overload of 5,940 b.h.p. at 140 r.p.m. The fuel consumption is given as 19·74 metric tons per day at a service output of 4,950 b.h.p. and a lower calorific value of 10,500 Kcal/kg. During the first voyage from North China to Stettin the ship had a mean speed of about 14 knots with the engine running at 123 r.p.m. and developing 4,300 b.h.p. On this voyage the fuel consumption of the main engine was about 18·2 metric tons per day. The main and auxiliary engines have fresh water cooling for the jackets, the exhaust valves and the cylinder heads. The engine is the first Japanese-built unit of this cylinder diameter and is of welded construction. It has a maximum pressure of 55 kg./cm.^{2}, and it is planned by the builders to increase this to 65 kg./cm.^{2} in future units. It is the first Japanese-built B. and W. engine to have two camshafts in order to facilitate starting and reversing operations. The upper camshaft operates the exhaust valves, while the lower drives the fuel pumps, and each camshaft has one cam for each valve or pump respectively. The engine drives a four-bladed Aerofoil propeller made of manganese bronze, and there is a spare propeller of cast iron. The three main generators develop 200 kW each and are driven by five-cylinder four-stroke 300 b.h.p. Hitachi-B. and W. engines developing 300 b.h.p. There is also a Daihatsu-Hitachi emergency generator developing 25 kW at 900 r.p.m., and placed above the main generators.

An Innoshima auxiliary boiler has been installed aft of the main engine. It is used primarily for heating the vegetable oil tanks and for domestic purposes. There is also an exhaust gas boiler in the funnel.—*The Motor Ship, June 1957; Vol. 37, pp. 124-125.*

Small Spanish Fruit Carriers

The Astilleros de Sevilla are building a series of small 12-knot motorships for different Spanish owners. The first to be completed was the *Astene Primero* (the name is derived from **Ast**illeros de la **E**mpresa **N**acional **E**lcano), an open shelterdecker with principal characteristics as follows:—

Length overall	222ft. 0in.
Length between perpendiculars	201ft. 6in.
Moulded breadth	36ft. 10in.
Depth to shelterdeck... ...	20ft. 5in.
Depth to main deck... ...	12ft. 11in.
Summer draught	12ft. 9½in.
Deadweight capacity... ...	1,100 tons
Grain capacity	83,200 cu. ft.
Bale capacity	78,100 cu. ft.
Gross tonnage	688 tons

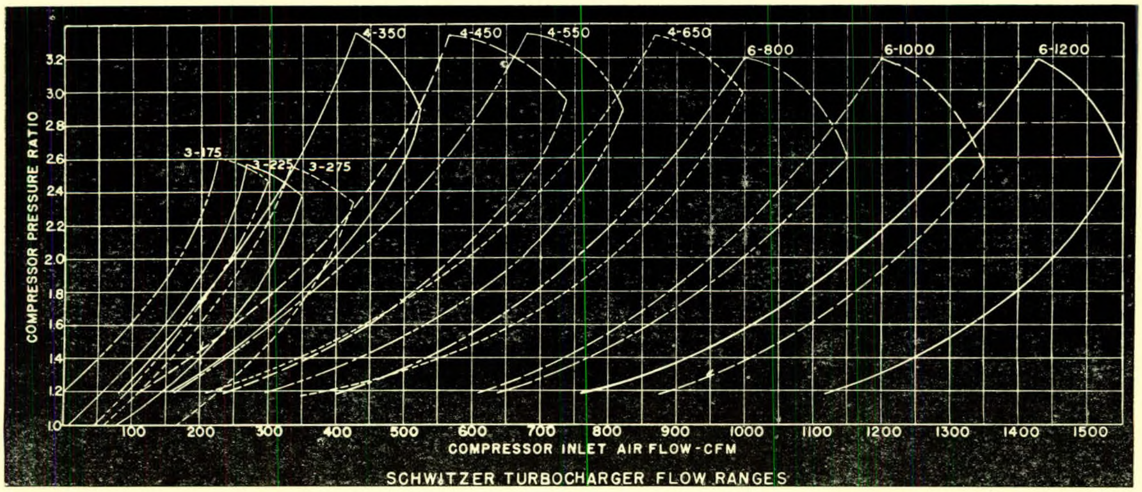
The propelling machinery consists of an eight-cylinder M487 Naval-Polar engine built in Spain under licence from Nydqvist and Holm and developing 1,280 b.h.p. at 250 r.p.m. During trials a speed of 13·02 knots was obtained in half-loaded conditions. Two 50-kW generators are driven by K52E Naval-Polar 90 b.h.p. at 500 r.p.m. Diesel engines. All accommodation, bridge and machinery is aft. There are two holds closed by Elcano-MacGregor steel hatch covers. A bipod mast carries four derricks. All deck machinery is by Elcano BDT and comprises four 1½/3-ton derricks, windlass, capstan and steering gear.—*The Marine Engineer and Naval Architect, May 1957; Vol. 80, p. 167.*

New American Turbocharger

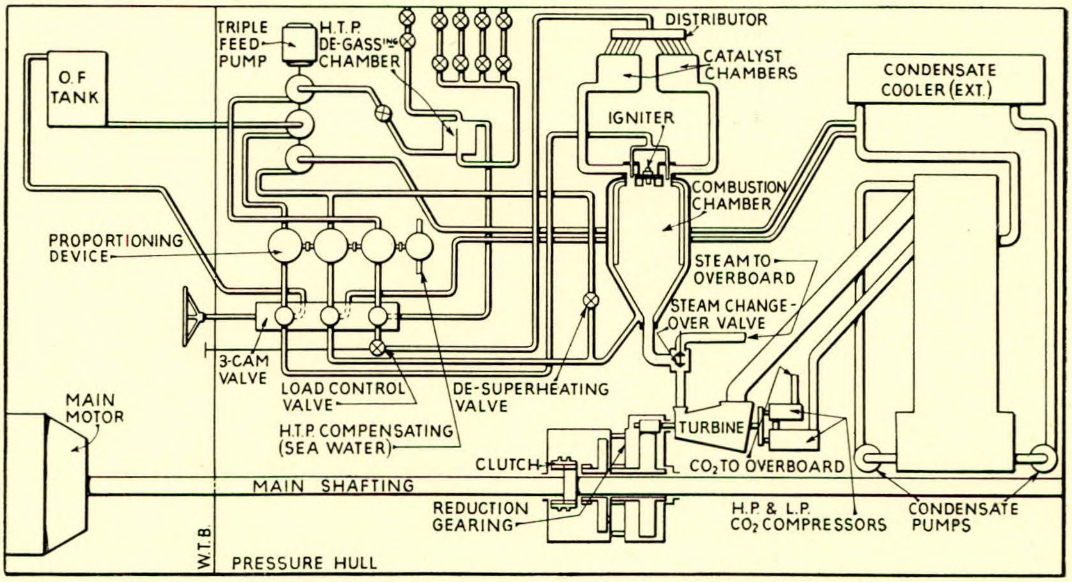
The Schwitzer turbocharger can be adapted to a wide variety of engines without change in basic design. Interchangeable turbine nozzle rings are available to match the operating conditions of each specific engine application. The divided turbine housings and nozzles are designed so that the turbine can be operated either as a blow-down turbine or as a steady flow turbine. Ordinarily, better results can be obtained by operating on the blow-down principle. This involves dividing the exhaust manifold so that there is no overlap of the exhaust processes of the cylinders discharging into any one branch of the manifold. The divided manifold and turbine housing will make better use of the exhaust gas energy. Lower average exhaust manifold pressures and better cylinder scavenging will result. Increased efficiencies have been developed and turbine operating speeds have been increased beyond those believed practical only a short time ago. Also, significant results are being obtained with the use of after coolers which reduce the temperature of air leaving the turbocharger and permit increased performance over a wider range. The use of the after cooler is encouraged wherever possible. The turbine housing is a "Ni-Resist" casting capable of withstanding high temperatures without scaling or stress failures. The bearing housing is a high quality grey cast iron and the compressor housing and cover are heat treated aluminium alloy castings. The present recommended exhaust gas temperature limit is 1,500 deg. F. for intermittent operation. The bearings are of the sleeve type and are lubricated by engine oil. A double piston ring type oil seal is used at each end of the shaft. It is pressurized by an air bleed from the compressor effectively to seal off leakage. Axial shaft location is fixed at the compressor end. The compressor and turbine wheels are designed for thrust equilibrium so that their thrusts between them is balanced out. The design configuration of the cobalt base alloy turbine wheel combined with aluminium compressor wheel, result in low inertia and low stress. Only a fraction of a second is required for the turbo to reach full speed when the load is suddenly applied to the engine.—*D. P. Robison, Diesel Progress, April 1957; pp. 26-27.*

An Experimental High-speed Submarine

H.M.S. *Explorer,* the first British submarine to use high-test peroxide in conjunction with her propulsion machinery, has an overall length of 225ft. 6½in. and a breadth of 15ft. 8in. Most of her superstructure fittings are retractable and she is well streamlined. By means of her special experimental machinery, the vessel is able to develop full power when completely submerged and independent of atmospheric oxygen. For lower speeds the conventional means of propulsion, consisting of Diesel engines for surface passages and electric motors supplied by batteries when submerged, are provided. It is believed that speeds of over 20 knots can be attained in service. H.M.S. *Explorer* has two shafts normally driven by conventional submarine motors but to each of which a turbine, deriving power from Diesel fuel burnt in decomposed H.T.P., may be clutched to provide high speeds under water. Each turbine installation operates on a direct cycle, so called because heat transfer takes place between hot gas and water which passes a mixture of steam and carbon dioxide to the turbine. Reference to the diagrammatic sketch will show the operation of the plant. The triple pump supplies H.T.P., special fuel oil and feed water through a proportioning device to a control valve known as the "three cam" valve. The first movement of this valve allows H.T.P. to be fed through a regulating or load valve to the catalyst chamber where violent decomposition of the H.T.P. occurs and steam-oxygen mixture passes to the combustion chamber. The second and third movements of



SCHWITZER TURBOCHARGER FLOW RANGES



Diagrammatic arrangement of high test peroxide plant in H.M.S. Explorer

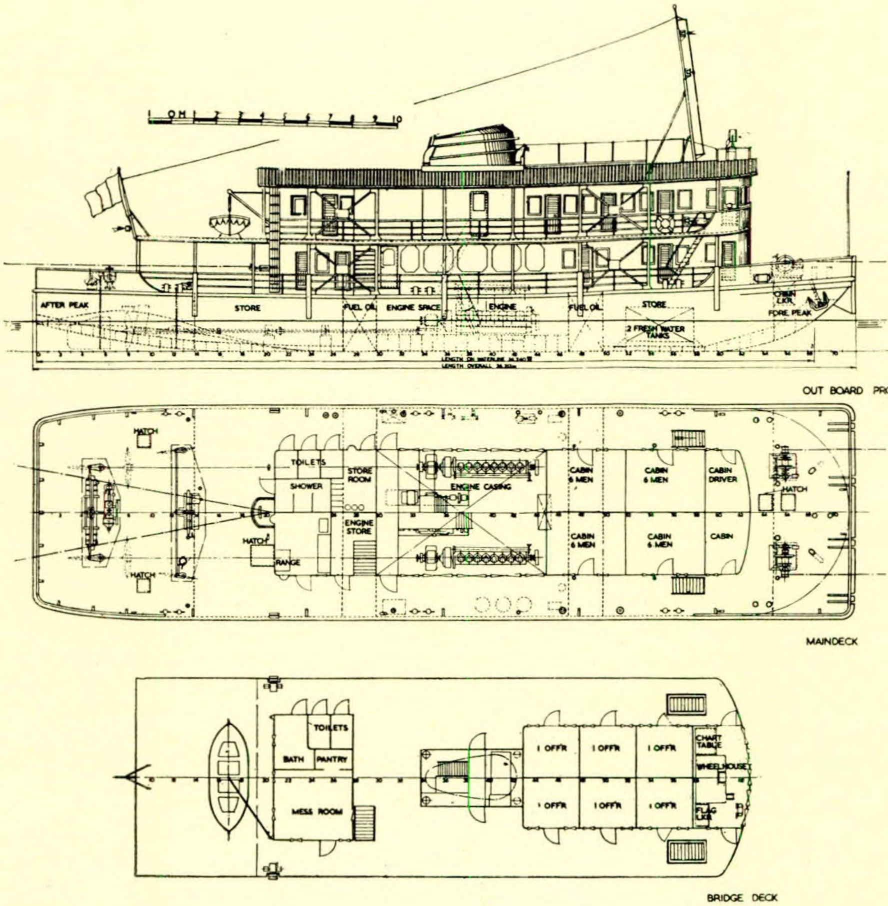
the valve passes water and then fuel to the combustion chamber. The water serves the important purpose of cooling the combustion chamber and reduces the gas outlet temperature to that required at the turbine inlet. From the combustion chamber the steam-oxygen mixture passes to the turbine through a changeover valve. During starting this valve discharges the products of the combustion chamber overboard to avoid a build-up of neat oxygen in the condenser and also serves to separate combustion chamber and turbine controls. The condenser design is such that the steam is condensed and the carbon dioxide separated, the former being returned to a feed tank and the gas discharged overboard by a Lysholm compressor. The triple pump passes the three working fluids H.T.P., special Diesel fuel and water which feed to the combustion chamber. The pump is driven by a 150 h.p. motor. Coupled to the motor shaft is a gearwheel which engages around its periphery three pinions turning at approximately 20,000 r.p.m. Each pinion drives an impeller. The triple pump is fitted with its own lubricating oil pump, cooler and filter. A proportioning device consisting of four meters for the alteration in weight due to expenditure of the H.T.P. water and fuel at any load controlled by the load valve sited in the H.T.P. line to the catalyst chamber is provided. Three of the meters pass H.T.P. water and fuel. The fourth meter passes water from sea into the submarine to allow automatically for the alteration in weight due to expenditure of the H.T.P and fuel. After the installation has been prepared for running, and the circulating water, lubricating oil, condensate and triple pumps are running, the plant is started by operating the three-cam valve. During the standby period this valve is in the "off" position and H.T.P. fuel and water pass back to the triple suction through a bypass in the three-cam valve. Two catalyst chambers per plant decompose the H.T.P. to steam and oxygen, which is passed to the combustion chamber. In the combustion chamber an igniter is energized by a switch operating a handle that also admits primary oil to the igniter sprays. The igniter is switched on after the first movement of the three-cam valve. Once ignition has been established the three-cam valve is moved through its final two positions and the productions of the combustion chamber are then available for admission to the turbine by operating the changeover valve. An impulse cycle was chosen for the turbine because of the large clearances necessary in order to meet the operational requirement for quick starting. Gases leaving the combustion chamber contain 14 per cent by weight of CO_2, the remainder being steam. The admission pressure at full load is 526lb. per in.^{2} The turbine drives on to the main propeller shaft through double reduction gears and is connected while the propeller shaft is rotating by a clutch which is engaged when the turbine has been run up to synchronous speed with the propeller shaft.—*Shipbuilding and Shipping Record, 4th April 1957; Vol. 89, pp. 433–434.*

Gas Turbine for the *Ormara*

The British Thomson-Houston Co., Ltd., are to design and manufacture the gas turbine and gearing which will be installed as replacement machinery in the British India cargo steamship *Ormara.* This ship is to be re-engined by Alexander Stephen and Sons, Ltd., Linthouse, with a free-piston gas generator/gas turbine installation. The gas turbine will be a 4,000 s.h.p. reversing unit comprising ahead and astern turbines with reduction gearbox. The 4,000 s.h.p., 6,800 r.p.m. six-stage reaction ahead turbine will be flexibly coupled to the after end of 6,800/110 r.p.m. locked train double-reduction gears. The high-speed pinion, which will be of single-helical type, partially to balance the ahead turbine thrust, will drive two high-speed wheels and the two double helical low-speed pinions will drive the main gearwheel, connected to the propeller shaft. A single-stage impulse astern turbine, capable of about 60 per cent ahead power will be flexibly coupled to the forward end of the ahead turbine. The four GS 34 free-piston gasifiers are to be manufactured by Stephens who hold a non-exclusive sub-licence from Alan Muntz and Co. for marine and industrial purposes. The power gas will be supplied to two oil-operated reversing valves mounted at the two inlets on the ahead turbines. These valves will control the gas flow to the ahead and astern turbines when manœuvring but at all other times power will be controlled by regulation of fuel to the gasifiers. The gas turbines for the *Ormara* will be the first B.T.H. units for operation with free-piston gasifiers. The ahead turbine being separate, a uni-directional unit can be offered for installations using a controllable-pitch propeller for reversing.—*The Marine Engineer and Naval Architect, May 1957; Vol. 80, p. 164.*

Belgian-built Pusher Tugs with British Machinery

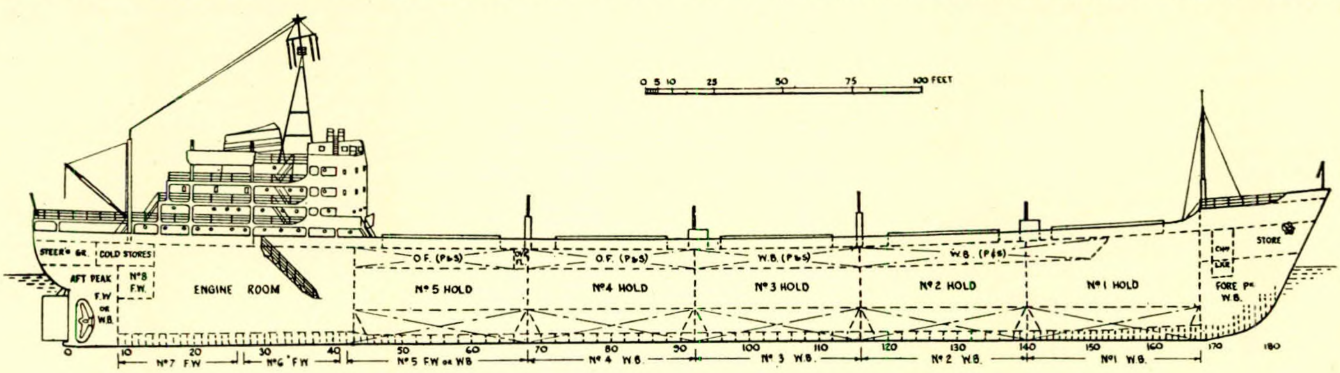
Three twin-screw tunnel-stern pusher-type tugs of unusual design recently completed trials on the River Scheldt, at Antwerp. They were built by three member firms of SYNABEL (the export syndicate of Belgian shipyards), namely, the Fulton Marine Engineering, S.A., Willebroek-Antwerp, Ch. Naval St. Pierre, S.A., Hemixem-Antwerp, and Ch. Naval de Rupelmonde, S.A., Rupelmonde. Each tug is designed to



push four 500-ton oil barges on the River Irrawaddy between Mandalay and Rangoon, i.e. from the recently discovered oil-fields to the Burma Oil Company's installations, which is a distance of approximately 500 miles. Of a design resulting from model tests at the Hamburgische Schiffbau-Versuchs-Anstalt to determine the suitability of the hull form, propellers and steering, and flanking rudders, each tug has an overall length of 118ft., a length L.W.L. of 112ft., and a moulded breadth of 30ft., the depth being 8ft. and the draught (level keel) 4ft. 3in. To have a free running speed of 12 knots, or 7 knots when pushing 2,000 tons, each vessel is of the raised-propeller design with a spoon bow, pusher knees and, as stated, a tunnel stern. For propulsion, two National R4AM8 eight-cylinder single-acting four-stroke Diesel engines of the normally aspirated design are installed, each developing 400 b.h.p. at 600 r.p.m. and driving through hydraulic reverse-reduction 2:1 gearboxes. The manganese-bronze propellers are 1·7 m. in diameter. Most of the auxiliary machinery is of British manufacture, including the engine driven Mono bilge pumps and Reavell air compressors, the two MacLaren-engined 20-kW Brush generators and the independent electrically driven Hamworthy air compressor.—*The Motor Ship, March 1957; Vol. 37, p. 533.*

Australian-built Motor Vessel

The first of a group of four motor bulk cargo vessels ordered by the Australian Shipbuilding Board is now in service. This vessel, the *Lake Barrine,* has been handed over by the builders, Evans Deakin and Co., Ltd., Brisbane, Australia, where the other three vessels, *Lake Boga, Lake Colac* and *Lake Sorrell* are also under construction. The *Lake Barrine* is owned and operated by the Australian Shipping Board, and is primarily designed for the ore and coal trade. She is a flush-deck vessel with all accommodation and the



Motor cargo vessel Lake Barrine

machinery aft. Each of the five self-trimming holds is fitted with MacGregor steel hatch covers which are operated by electric capstans. The principal particulars are as follows:—

Length o.a.	467ft. 6in.
Length b.p.	440ft.
Breadth moulded	57ft.
Depth moulded to upper deck	34ft.
Loaded draught	25ft. 6¼in.
Deadweight	10,437 tons
Cargo capacity (self-trimmed) ...	460,453 cu. ft.
Gross tonnage	7,215 tons
Service speed	12 knots
Horsepower	3,300 b.h.p.

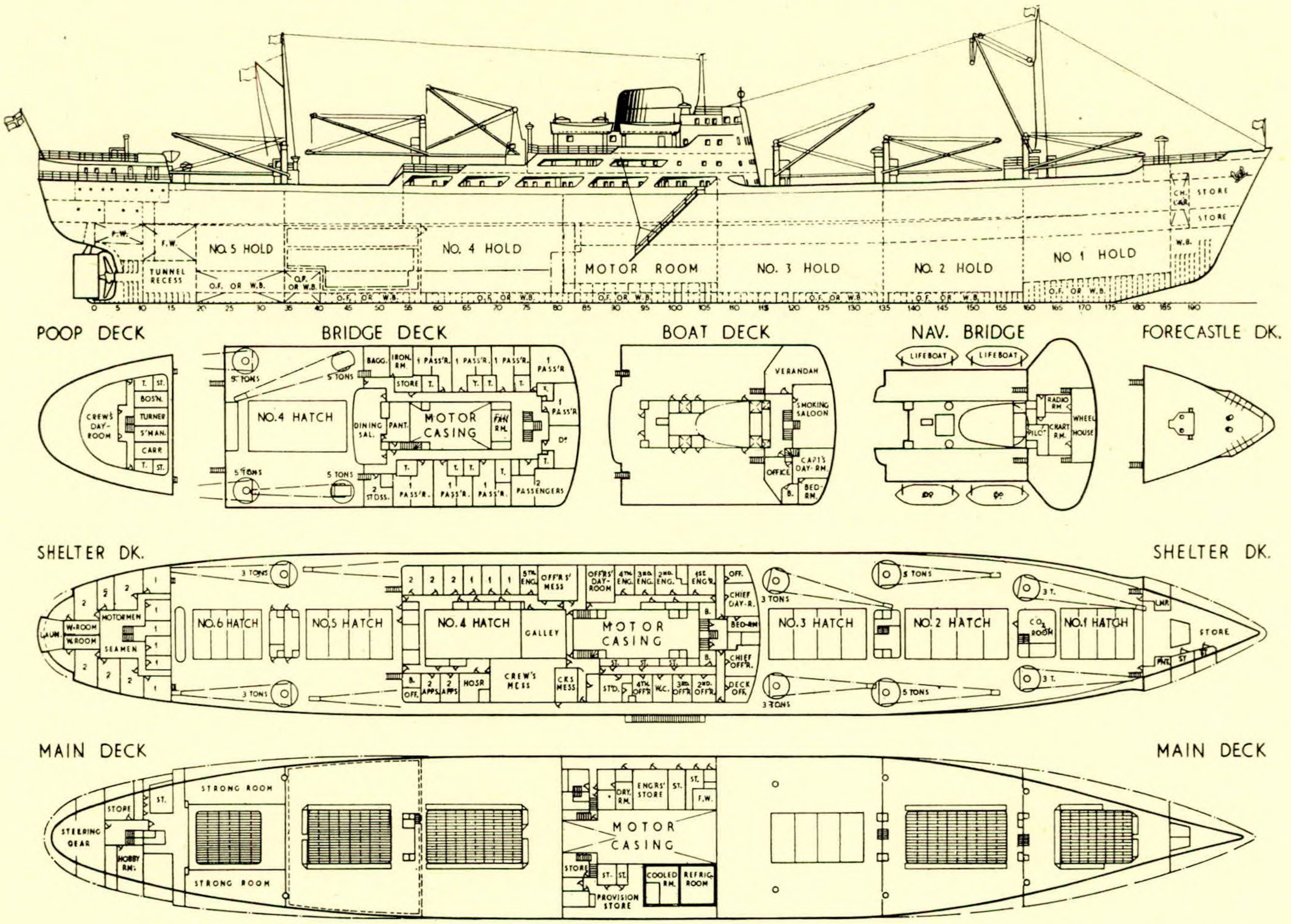
The vessel does not carry any cargo handling equipment as she will trade only between ports where bulk loading and discharge grab facilities are available. No timber ceiling is fitted in any of the holds, the tanktop plating and hopper plating being increased in thickness as a protective measure against grab damage. The *Lake Barrine* is powered by a four-cylinder Doxford opposed-piston oil engine developing 3,300 b.h.p. at 108 r.p.m. and giving a normal service speed of 12 knots in a fully loaded condition. These engines are built in Australia under licence by the Commonwealth Marine Engine Works, Melbourne. The engine is fresh water cooled, with steam heating to the jacket and piston cooling water to facilitate starting. The steam heating is supplied by a Spanner "Swirlyflo" vertical exhaust gas boiler which is fitted in the main engine exhaust. The boiler has alternative oil firing equipment. The steam heating is used for oil fuel, cooling water, domestic fresh water and accommodation services. The domestic fresh water is alternatively electrically heated. The auxiliary machinery includes three 150-kW generators powered by Polar Diesel engines developing 230 b.h.p. at 600 r.p.m., three electrically-driven Reavell air compressors, and a petrol/paraffin-driven emergency compressor. All pumps are electrically driven.—*The Shipping World, 12th June, 1957; Vol. 136, pp. 559-561.*

Production of Nickel-aluminium Bronze Propellers by the CO_2 Process

The article describes the technique used in the Brooklyn Foundry of Columbian Bronze Corporation, Freeport, where the CO_2 process has been in large-scale use for about a year. The benefits which accrue from the use of this process include much increased fidelity of the castings to pattern dimensions, saving on moulding time, and all-over economy in production costs. The main output of the foundry is ships' propellers in manganese bronze and nickel-aluminium bronze (copper 80, aluminium 10·5, nickel 5, iron 4, manganese 0·5, per cent). Some aluminium castings are also made.—*R. H. Herrmann: "Making Ship Propellers with CO_2 Molds and Cores".—Foundry, Feb. 1957; Vol. 85, pp. 104-9.—Abstract, The Nickel Bulletin, May 1957; Vol. 30, p. 74.*

Belgian-built Swedish Cargo Liner

A fast cargo motor liner has recently been handed over to her owners, Rederi Transatlantic A/B, Gothenburg. This



General arrangement of the Kirribilli, *10,560 tons d.w., hull built by Jos. Boel and Fils, S.A., Tamise, Belgium, and engined and completed by Götaverken, for Rederi Transatlantic A/B, Gothenburg* vessel, the *Kirribilli,* 10,560 tons d.w., was completed by Götaverken after the hull had been towed from the Belgian shipyard of Jos. Boel and Fils, S.A., Tamise, where it had been built. The principal particulars of the *Kirribilli* are as follows:—

Length o.a.	511ft. 1in.
Length b.p.	470ft.
Breadth moulded	68ft.
Depth moulded to shelterdeck ...	44ft.
Depth moulded to main deck ...	31ft. 6in.
Draught	27ft. 7⅝in.
Deadweight	10,560 tons
Gross	7,630 tons
Net	4,353 tons
Cargo capacity:	
General cargo	581,500 cu. ft.
Refrigerated cargo	49,100 cu. ft.
Vegetable oil	38,500 cu. ft.

The main engine is a ten-cylinder direct reversible, two-stroke Diesel engine of Götaverken's own design, developing 10,800 i.h.p. or 8,800 b.h.p. at 110 r.p.m. The cylinder diameter is 760 mm. and the stroke is 1,500 mm. In keeping with modern practice the main engine is arranged for running on heavy oil. The engine gives the vessel a speed of 17½ knots fully loaded. Electric current is supplied at 220-volts d.c. by four auxiliary Diesel generators running at 350 r.p.m. The total generated capacity is 810 kW. Among the auxiliary equipment is an oily water separator with a capacity of 80 tons per hour.—*The Shipping World, 6th February 1957; Vol. 136, pp. 178-179.*

Electromagnetic Slip Couplings

Fig. 7 illustrates the main and auxiliary engine layout on the trawler *Euros* built at H. C. Stülcken and Son's shipyard for the Cranzer Fischdampfer A.G. A 1,000-h.p. Diesel engine and a second Diesel engine of 500 h.p. are connected to the shaft reduction gearing through electromagnetic slip couplings. The generator for the 200-h.p. trawl winch is located between the auxiliary Diesel and its appropriate coupling, the generator shaft being dimensioned so as to transmit the entire power of the engine to the coupling. The couplings are provided with sheet steel covers having an air outlet at the top through which cooling air drawn from the engine room is expelled into the engine room ventilating shaft. At continuous full load the maximum measured temperature rise of the cooling air was 9 deg. C. All AEG couplings supplied are self-ventilating and require no separate equipment. It is advantageous, however, to arrange the engine room ventilation so that sufficient cooling air be conveyed as close as possible to the electromagnetic couplings.—*H. Borrs, AEG Progress, 1957; No. 1, pp. 19-22.*

Superchargers and Supercharging

Problem of Pressure Charging.—It is shown from curves illustrating the effect of pressure ratio on charge density for various compressor adiabatic efficiencies that the efficiency of the pressure charging system has a great influence on engine output. The Roots blower has been the accepted component for pressure charging for many years, but because of its low efficiency attractive engine outputs have not been obtainable. It is also very noisy and not easy to drive, so its use has been unpopular. To eliminate the shortcomings of the Roots machine the "Bicera" compressor was designed, and the operation of this novel form of pressure charger is described. Test results are given which prove the performance to be superior to that of the Roots type, and when fitted to an engine showed that a pressure charged, four-cylinder engine can give the output of a six-cylinder, naturally aspirated engine, and can be as economical in fuel consumption. (*D. W. Tryhorn.*)

Supercharging of High-speed Diesel Engines of Mechanically Driven Compressors.—The most useful application of the mechanically driven supercharger is in the field of the smaller Diesel engine which operates over a wide range of speed, and especially where the duty calls for a high torque to be developed at low speed. Road vehicles and earth-moving plant commonly require these characteristics. In the interest of fuel economy a high adiabatic efficiency is of first importance in a supercharger so used. It is also desirable to be able to control the rate of air delivery from the supercharger to match the load on the engine; methods are discussed and test figures from the Fell locomotive propulsion engines are quoted. Where a simple fixed ratio drive to the supercharger is used a good volumetric efficiency at low speed and at the working pressure ratio is important in order that the boost at low engine speed shall be high. The Ricardo wobble-plate compressor designed to meet these conditions is described and performance data are given. Overall performance figures of an engine when supercharged by this compressor are also included and the methods of applying the engine in vehicle service are discussed. (*B. W. Millington.*)

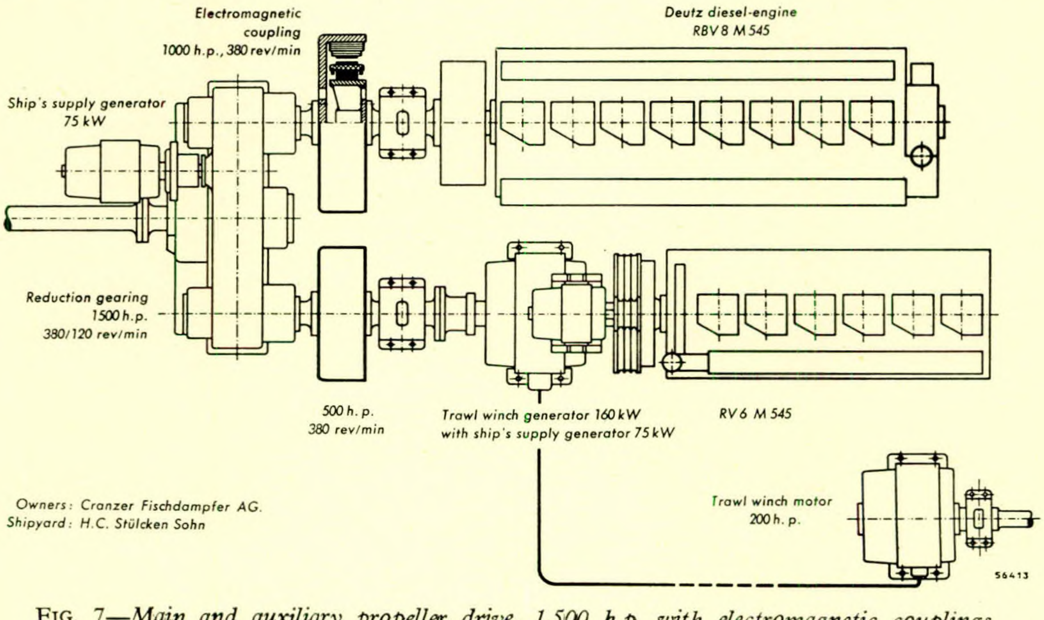


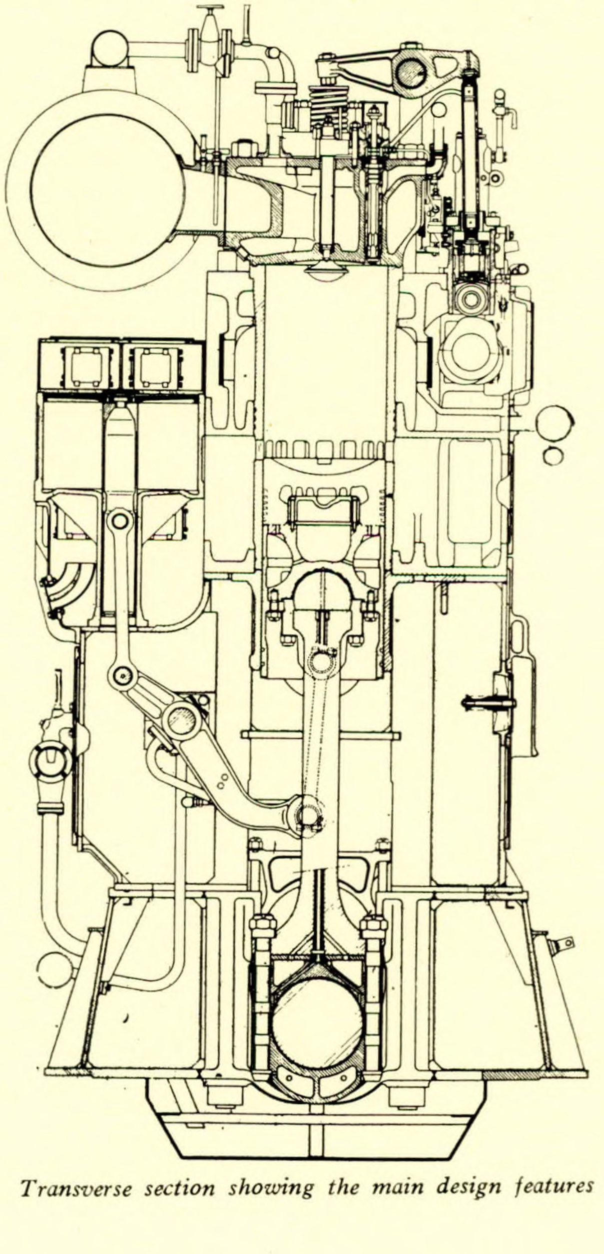
FIG. 7—*Main and auxiliary propeller drive, 1,500 h.p. with electromagnetic couplings and trawl winch plant on the trawler* Euros

Present Position and Future Prospects of Turbocharging of High-speed Diesel Engines.—Consideration is given to the basic principles of supercharging and the advantages and/or drawbacks of using surplus exhaust energy to drive the supercharger. A review is given of the application of the exhaust turbocharger to high-speed engines of the road vehicle type. It is shown that there are now in existence units which can be mounted without adding appreciably to the overall bulk and weight of the engine, and which enable the latter to develop brake mean effective pressures of up to 140lb. per sq. in. without undue smoke. Torque and acceleration characteristics are considered and limitations *vis-a-vis* positive displacement blowers are briefly surveyed. Lessons to be learnt from large engine practice are then reviewed, particularly from the standpoint of charge cooling. The Miller and Bone systems are briefly described. The trend of development towards higher pressure ratio turbochargers is then considered, together with the possibilities of incorporating in the engine design improved pistons and fuel injection systems. The paper concludes with a brief reference to the present position and practice of turbocharging two-cycle engines. (*C. H. Bradbury.*)

Design and Development of Small Radial-flow Turbochargers.—Turbochargers are already well established for Diesel engines of moderate and high outputs, and there is no reason why this should not also apply to Diesel engines of 100 h.p. and upwards. In the former case axial-flow turbines are used but these suffer from serious disadvantages in the smaller sizes, so that radial-flow turbochargers are more likely to be adopted. The range of applications on the smaller Diesel engine is very extensive compared with its larger counterpart, and, in particular, automotive engines will have a serious influence on the design and development of the small turbocharger. Owing to the rapidly and widely varying load conditions, the component efficiency must be high and the rotating assembly as light as possible to give rapid response times. This involves intensive development of turbine and compressor, and the use of shaft speeds of 40,000-50,000 r.p.m., which until recently would have been regarded as impracticable for service conditions where a life of several thousand hours must be accepted. The automotive field is also highly competitive, and production quantities are large. Consequently, the design of the turbocharger must be such that it can be made economically and in large numbers, and at the same time be as efficient as possible over a wide range of operational conditions. Although economic conditions may fix the minimum size of the turbocharger, the limit does not yet appear to have been reached on an ærodynamic basis, and reasonably high component efficiencies are readily attainable with rotors of 4 inches diameter or less. The onset of exhaust smoke in some automotive service applications will not only dictate the matching conditions for the turbocharger, but may also necessitate some further development to prevent it being a limitation on the power output from a given installation. (*C. A. Judson and E. Kellet.*) —*Papers presented at a Meeting of The Institution of Mechanical Engineers on 9th April 1957.*

Uniflow Scavenged Two-stroke Engine

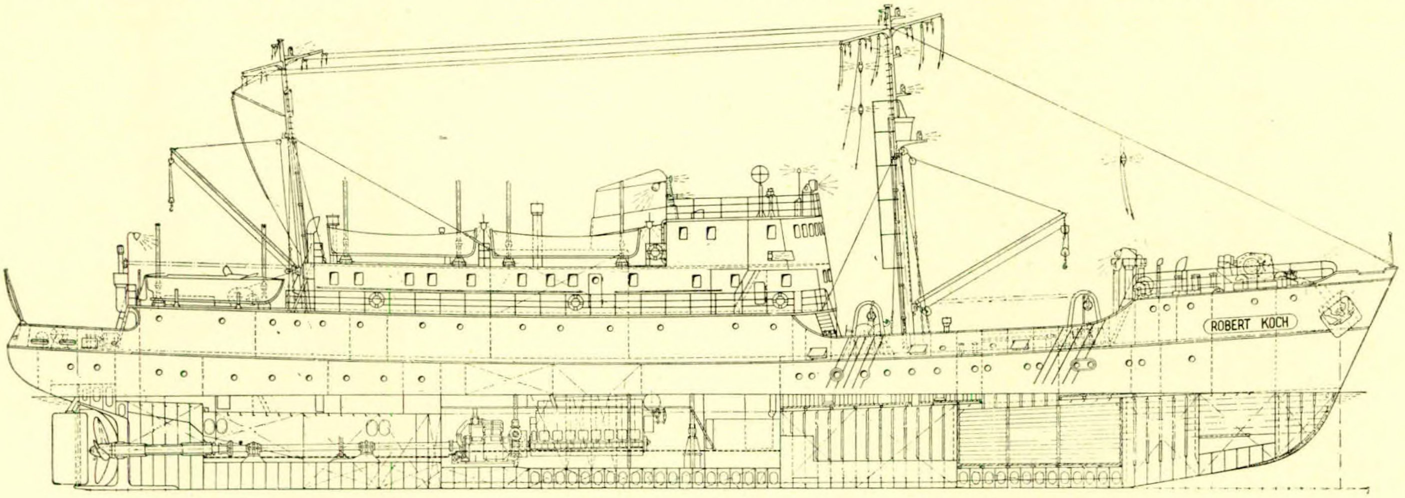
Werkspoor N.V. of Amsterdam have recently introduced a new trunk piston two-stroke engine to supplement their T.M. four-stroke range. The new unit has a bore and stroke of 450 mm. and 700 mm. respectively and a rated output in normal form of 300 b.h.p. per cylinder at 250 r.p.m. (b.m.e.p. 71lb. per sq. in., piston speed 1,150ft. per min.). It has been designed for direct coupling to the shaft or for multi-engine operation in six- and eight-cylinder models. The bedplate is fabricated from mild steel and incorporates cast steel main bearings. The crankcase is also fabricated and it supports a cast iron cylinder block, all these components being tied by steel through-bolts which are tightened hydraulically in three stages during assembly. The crankshaft is carried well within the bedplate with a centre line below the joint face. The thrust block and seat for the turning gear are integral with the bedplate. Air for scavenging is provided by lateral reciprocating pumps with trunk guides driven from each crosshead by swinging links and levers. The air enters through Venturi-type inlet silencers. The valve housings can be removed easily for inspection. A piston of composite design with light alloy, oil-cooled crown, cast steel, white-metal-lined gudgeon pin bearing and cast iron skirt is employed. The gudgeon pin is connected to the connecting rod by four bolts and is itself secured by two caps. Top end lubrication can be quite a problem in two-stroke engines since the load is nearly always directed downwards and the rocking movement of the pin with respect to the bearing is relatively small. For this reason a construction with a large bearing surface was chosen. The piston bears on the gudgeon pin for the full width as can be seen from the drawing. The pin is chromium-plated and subsequently "super-finished". Separate lubricating oil and circulating water pumps are employed. The oil for piston cooling and lubrication of the running gear is delivered to the reciprocating assembly from the busrail alongside the bedplate to one of the trunnion boxes for each scavenging pump rocking lever. From this point the oil is conveyed through the lever and swinging link to the piston head where it is



Transverse section showing the main design features constrained to take a spiral path in close contact with the crown. A lead is taken from this oil supply to the gudgeon pin bearing and also through a drilling in the connecting rod to the crankpin. There are thus no stress-raising oilways in the crankshaft. Piston cooling oil returns through the other swinging link and trunnion box whence it emerges through a sight glass with thermometer at the rear of the engine.—*Gas and Oil Power, June 1957; Vol. 52, pp. 149-152.*

Mother Ship for Fishing Fleet

Delivery was taken some time ago of an unusual type of vessel designed to serve with the fishing fleet of the East German Republic (Deutsche Demokratische Republik). She is named *Robert Koch* and her purpose is to carry out repairs in vessels engaged in fishing, to provide hospital accommodation for men in the fleet who may become ill, and to supply fuel or other necessary provisions to the fishing vessels. She is 196ft. in length b.p. with a beam of 31ft. 9in., the net register being 1,108·5 tons. For propulsion there are two four-stroke 1,170-b.h.p. engines with eight cylinders, the speed being 315 r.p.m. The drive is taken through reduction gearing to a single propeller shaft. The designed speed is 13·6 knots. The fuel oil capacity is sufficient for 22 days' service.—*The Motor Ship, June 1957; Vol. 38, p. 128.*



Sectioned profile of an East-German-built mother ship for fishing vessels

Rotary Shaft Seals

The paper describes an investigation to establish the mechanism by which sealing is achieved in the conventional synthetic rubber oil seal when used on a rotating shaft. The experiments demonstrate quite clearly that when the seal is functioning normally there is a coherent film of oil between the sealing lip and the rotating shaft. The most important single variable in the operation of the oil seal is the contact pressure which in turn depends on the applied load. At low loads there is copious leakage and this leakage decreases gradually as the applied load is made greater. There is no sudden change either in friction or in sealing which would indicate the elimination of the oil film as the sealing point is reached. Two methods were used for destroying the effect of surface tension at the oil-air interface of the sealing point. In the first method pressure was applied to the oil and increased gradually until leakage suddenly started; in the second method the air at the interface was replaced by fluid and this also gave rise to leakage. The mechanism of sealing, therefore, is that the surface tension of the oil forms a meniscus which prevents oil flow through the gap between the seal and the shaft.—*Paper by E. T. Jagger, read at a Meeting of The Institution of Mechanical Engineers on 27th March 1957.*

New Flame Cutting Machines in Shipbuilding

Flame cutting in shipbuilding is still in the evolutionary state, and is not yet fully developed. It is thus essential for shipyard engineers and manufacturers of flame cutting equipment to co-operate closely, so as to produce the best design for each application. The machines referred to in this article are large and relatively heavy stationary types, used for making straight cuts to prepare the edges for welding and for producing curved cuts. The first principle of construction is weight—adequate weight to afford stable and accurate operation for a long time, even under the most adverse conditions. The driving mechanism of the flame cutting machine has to be designed with the severe working conditions prevailing in shipyards in mind. In theory, a very light driving motor would be adequate; but, in practice, it has been found that considerable power is necessary to drive the main portal. A flame cutting machine, and especially the guide rails, can be accidentally struck by heavy objects; and, as the accuracy of the guide rails determines the accurate performance of the machine, the carriage and the guide rails must be robustly designed and so arranged that they can withstand hard knocks. One reason for a really stable machine design is the high degree of accuracy demanded by the shipbuilding engineers. Prefabricated construction requires accurate preparation of all material, because adjustments on the slipway would cancel all the advantages obtained by prefabrication. Due to the increasing activity in the shipbuilding yards, automatic welding is becoming increasingly popular, and submerged arc weld-

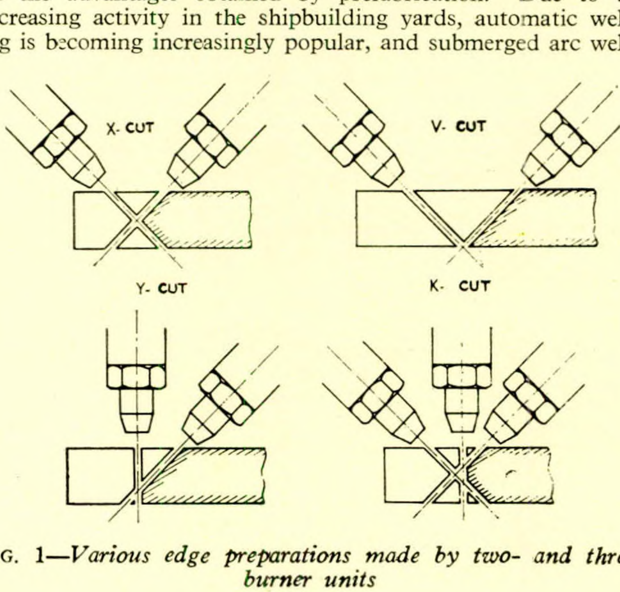


FIG. 1—*Various edge preparations made by two- and three-burner units*

ing requires an especially high degree of accuracy on the prepared plate edges. Even with very long plates, the gap between two butting edges must not exceed $\frac{1}{32}$in. The tolerance for any deviation from a straight line is thus fixed. The shape of the plate edge, prepared for welding, depends on the thickness of the job. On thin plates, simple vertical cuts are generally made; and on greater thicknesses of plate, simple or bevel cuts with a root face are necessary. For submerged arc welding, this edge preparation is often used. Various edge preparations made by two- and three-burner units are illustrated in Fig. 1.—*R. Bechtle, The Shipbuilder and Marine Engine-Builder, May 1957; Vol. 64, pp. 334-337.*

Ship-plating Subjected to Loads

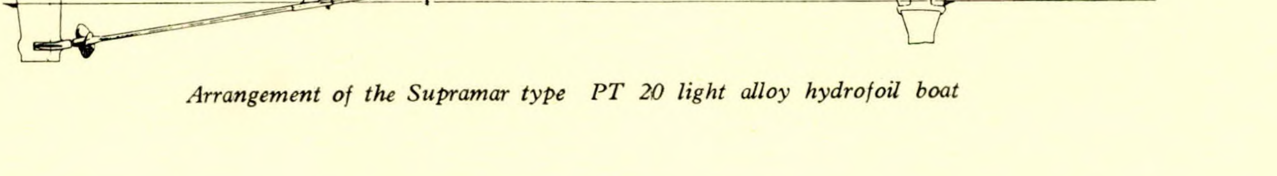
Although the problem of the rectangular plate subjected to loads both normal to and in the plane of the plate is one of prime importance to the ship designer, it is only recently that theoretical solutions for the problem have been advanced. The solutions, which apply in the elastic range, lack experimental verification. In this paper, experimental work is described which was undertaken for the British Shipbuilding Research Association in an endeavour to remedy this deficiency. Results of stress and deflexion obtained from full-scale tests on four mild steel plates, two square and two rectangular in shape, are presented and compared with solutions given by Timoshenko, Bleich, and Chang and Conway. It is shown that the approximate method of solution put forward by Bleich, which is of general application, is sufficiently accurate for use in the design of ship plating. Families of design curves, incorporating the theoretical solutions and the main experimental results, are presented. The scope of these curves, the various conditions which restrict their application, and their use in design work are discussed.—*Paper by A. G. Young, submitted to The Institution of Naval Architects for written discussion, 1957.*

Hydrofoil Boat

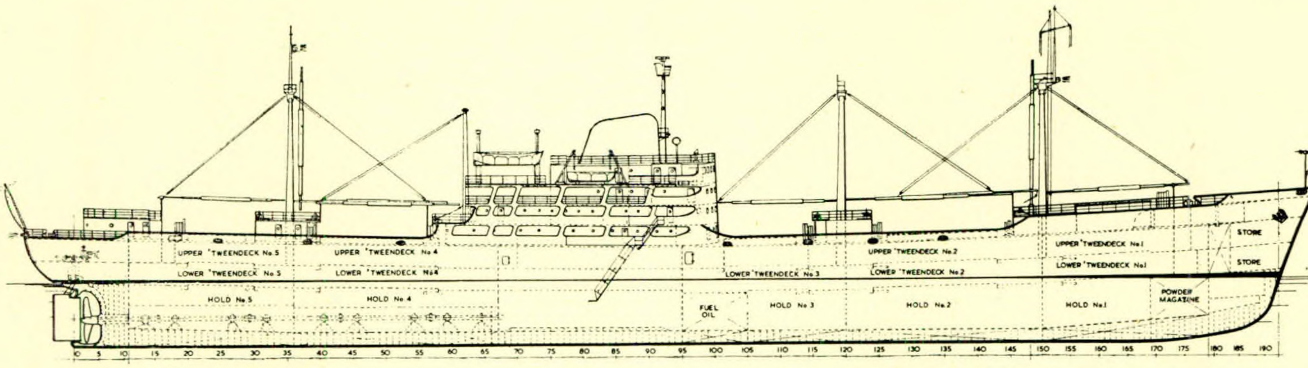
The latest hydrofoil boat to enter commercial service is the *Freccia del Sole,* built in Messina, Sicily. Powered by a Mercedes-Benz type MB 820 Db Diesel engine, and fitted with special transmission by Zahnradfabrik Friedrichshafen, the vessel is built on the Schertel-Sachsenberg hydrofoil system developed by Supramar, Ltd., Switzerland. The principal particulars of the Supramar type PT 20 hydrofoil boat are as follows: —

Length overall	20·7 m.
Maximum breadth	4·8 m.
Width over hydrofoils	5·9 m.
Draught immersed	2·6 m.
Cruising draught	1·2 m.
Number of seats	70
Payload capacity	7 tons

With a Diesel engine developing 1,350 h.p., a maximum speed of 80 kilometres per hour is attained (50 m.p.h.), while a cruising speed of 70 k.p.h. is achieved with a Diesel engine developing 1,100 h.p., for an approximate fuel consumption of 188 kilograms per hour. Like all recent Supramar hydrofoil boats, the *Freccia del Sole* is built of light alloys throughout, the light weight of the material being an obvious attraction for a speedboat of this kind. Contrary to the conventional type of ship (i.e. vessels which displace water) hydrofoil boats obtain their support, when travelling through the water, by means of the lifting power exerted by the immersed surfaces of so-called hydrofoils. These foils or streamlined planes are attached to the undersurface of the hull. As a result of the forces acting on them, when the boat moves forward through the water, similar to the effect of the airstream on the wings of an aeroplane, the hull is gradually lifted out of the water as the speed of the craft increases, until eventually the whole hull is lifted clear of the water. The water resistance is thereby reduced by about 50 per cent and at the same time the braking action of waves upon the hull is almost entirely eliminated. Resulting from trials, extending over more than twenty years, with many different types of hydrofoil boats, the most favourable designs for the shape of hydrofoils and their relation to boats hulls have been evolved in order to achieve maximum speed, seaworthiness and manœuvrability. At the same time the best methods were discovered for the form and arrangement of propellers, propeller shafts, rudders and steering gear. The stability of hydrofoil boats is much greater than that of any ordinary type of surface vessel. They are easy to handle and contrary to the action of an ordinary craft, which heels outwards when turning, hydrofoil boats heel, or bank, inwards. When travelling at slow speed the hydrofoils have a dampening effect in rough water. As speed increases the hull rises gradually and quite smoothly out of the water and thereafter travels freely through the air, with only the hydrofoils still immersed, and at a determined height above the water surface. Compared with other types of speedboats of equal size, the following advantages are claimed for hydrofoil boats: (1) Only half the engine power is required to attain the same speed. Fuel consumption is reduced by 50 per cent and consequently cruising range is nearly doubled. (2) The economy on fuel, as well as the possibility to cover an increased daily distance as a result of high speed, are allowing a reduction of operating cost of about 30 per cent. (3) Their motion in rough water is much less and they are able to maintain their maximum speed under conditions which would force other type of fast craft to slow down considerably. (4) Even at maximum speed bow waves and wash created by the craft are so small that narrow rivers, canals and congested waters can be travelled through at high speed without any danger to embankments, moored craft or vessels being towed. (5) Hydrofoil boats show high acceleration and remarkable stopping power. (6) Without influencing the cruising qualities of the craft when travelling in the emerged state over the water surface, the length/beam ratio of hydrofoil boats can be altered within wide limits so that the boats can be designed for various purposes. (7) The motion in rough water is negligible and too small to cause sea sickness. —*The Shipping World, 5th December 1956; Vol. 135, pp. 500-501.*



Arrangement of the Supramar type PT 20 light alloy hydrofoil boat



19-knot French Cargo Ships

For replacement of tonnage engaged on their various services, mainly from France to Mediterranean ports and to the East, the Cie. des Messageries Maritimes, Paris, prepared a programme for the construction of five 9,300-ton ships and the contract was placed with the Ch. Navals de la Ciotat. These vessels are designed for a speed of 19·5 knots on trials and 18 knots in service. The following are the main characteristics of these vessels:—

Length overall ...	156·560 m.—513·3ft.
Length on waterline ...	148·480 m.—487ft.
Length b.p.	146·000 m.—478·8ft.
Moulded breadth ...	19·700 m.—63·6ft.
Displacement loaded ...	15,366 tons
Corresponding draught	7·918 m.—24·97ft.
Deadweight capacity ...	9,300 tons
Gross register (approximately)	7,300 tons
Net register (approximately)	4,000 tons
Volume of hold and 'tween decks ...	15,000 cu. m.
Volume of refrigerated chambers	300 cu. m.
Volume of vegetable oil	600 cu. m.

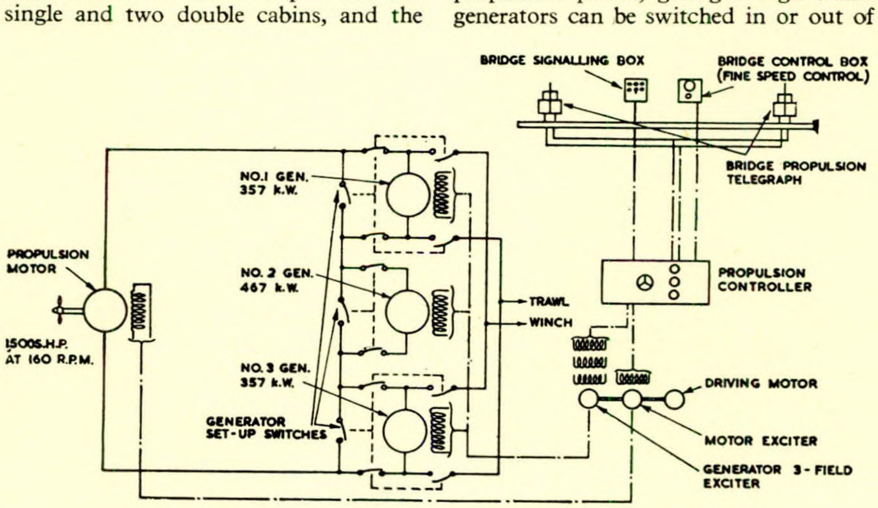
The ships will each be equipped with a ten-cylinder Sulzer turbocharged engine having cylinders 760 mm. in diameter with a piston stroke of 1,550 mm. In service the output will be 13,450 b.h.p. at 119 r.p.m. The radius of action at 18 knots will be 12,000 miles. There are three holds forward of the machinery compartment and two aft, and all of the holds and 'tweendecks will be ventilated on the "Cargocaire" system. The cargo handling machinery will include four 10-ton and ten 5-ton winches. Accommodation is provided for five passengers in one single and two double cabins, and the crew totals fifty-nine. In addition to these ships three more of identical dimensions and general design have been ordered, two from the Chantiers and Ateliers de Provence, at Port de Bouc, and one with the Soc. Cockerill-Ougrée, Hoboken, Belgium. In each of these ships a Burmeister and Wain-type engine of 13,700 b.h.p. is to be installed. It is an eleven-cylinder turbocharged design with cylinders 740 mm. diameter and a stroke of 1,600 mm. The engines for the Provence ships will be constructed by Schneider Creusot in France and that for the Cockerill vessel by Cockerill-Ougrée at Seraing. —*The Motor Ship, June 1957; Vol. 37, pp. 122-123.*

Diesel Electric Trawler

One of the most important advances in the propulsion of British fishing vessels took place recently when the Diesel-electric trawler *Portia* entered commercial service. This vessel is noteworthy in many respects, not only because she is Britain's largest and fastest trawler but for the new standards of seaworthiness and crew comfort which have been incorporated in her design. *Portia* has been built to Lloyd's Register class and has dimensions as follows:—

Length overall	210ft. 6in.
Length between perpendiculars ...	190ft. 0in.
Moulded breadth	34ft. 6in.
Moulded depth	17ft. 10in.
Gross tonnage, tons	882
Fish hold capacity, cu. ft. ...	17,500
Bunkers (Diesel oil), tons ...	227

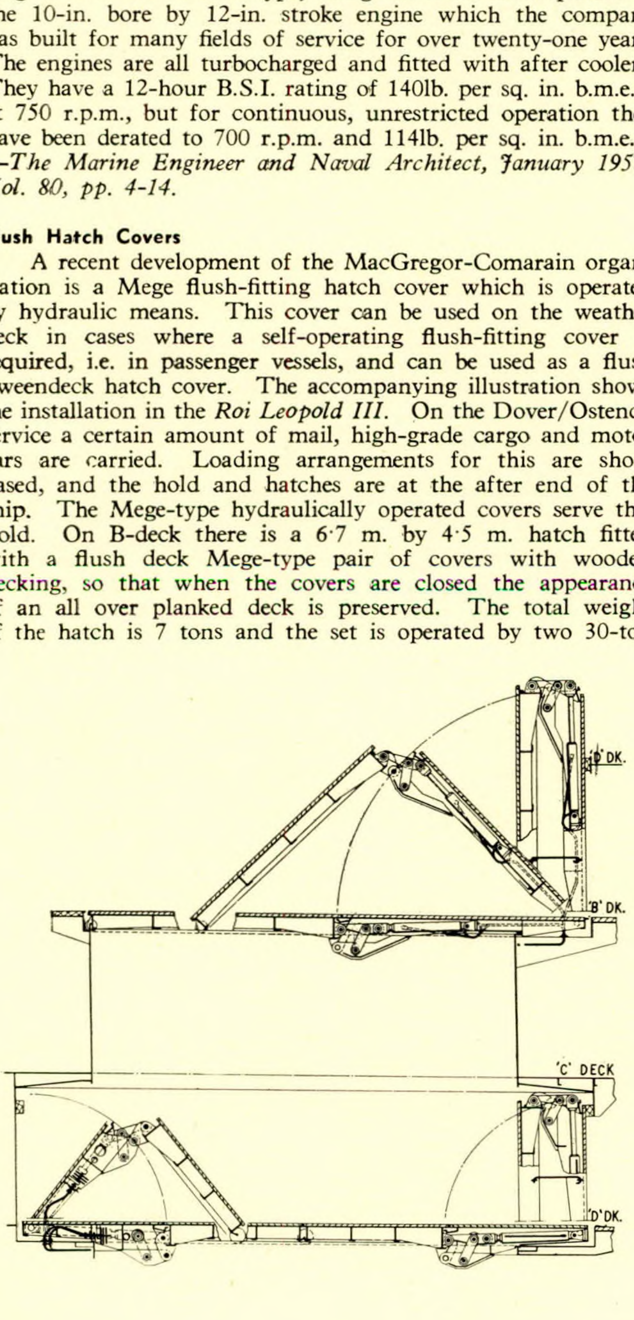
The propulsion machinery operates on the controlled voltage principle, using a modified form of Ward-Leonard control. The main generators and propulsion motor are arranged in a series circuit and one, two or three generators can provide propulsion power, giving a high standard of reliability. The generators can be switched in or out of the propulsion circuit,



Simplified main circuit diagram showing control system whilst the vessel is under way, without interrupting propulsion power, and the engines cannot be overloaded in any condition of operation or combination of generators, propulsion motor or trawl winch. The inherent characteristics of the system limit the power demand from the engines to their designed service rating. The single-armature 850-volt d.c. 1,400-amp propulsion motor delivers 1,500 s.h.p. at 160 r.p.m. It is a totally-enclosed forced-ventilated eight-pole machine with class-B insulation designed for unrestricted operation in any climate. The propulsion motor can be controlled either directly from one of the bridge telegraphs or from the engine room in response to telegraphed orders, as may be required. For normal manœuvring the six fixed speed steps of the bridge telegraph/controllers give adequate propeller speed control but, when fishing, a finer adjustment is required. This is provided by a Vernier control fitted to the bridge propeller speed indicator box whereby any speed required, between the speed steps on the telegraph, can be selected. Since this adjustment cannot exceed the chosen speed step on the telegraph, it cannot be mal-operated. The propulsion generator engines are of English Electric SRKM type, being the latest development of the 10-in. bore by 12-in. stroke engine which the company has built for many fields of service for over twenty-one years. The engines are all turbocharged and fitted with after coolers. They have a 12-hour B.S.I. rating of 140lb. per sq. in. b.m.e.p. at 750 r.p.m., but for continuous, unrestricted operation they have been derated to 700 r.p.m. and 114lb. per sq. in. b.m.e.p. —*The Marine Engineer and Naval Architect, January 1957; Vol. 80, pp. 4-14.*

Flush Hatch Covers

A recent development of the MacGregor-Comarain organization is a Mege flush-fitting hatch cover which is operated by hydraulic means. This cover can be used on the weather deck in cases where a self-operating flush-fitting cover is required, i.e. in passenger vessels, and can be used as a flush 'tweendeck hatch cover. The accompanying illustration shows the installation in the *Roi Leopold III*. On the Dover/Ostende service a certain amount of mail, high-grade cargo and motor cars are carried. Loading arrangements for this are shore based, and the hold and hatches are at the after end of the ship. The Mege-type hydraulically operated covers serve this hold. On B-deck there is a 6·7 m. by 4·5 m. hatch fitted with a flush deck Mege-type pair of covers with wooden decking, so that when the covers are closed the appearance of an all over planked deck is preserved. The total weight of the hatch is 7 tons and the set is operated by two 30-ton



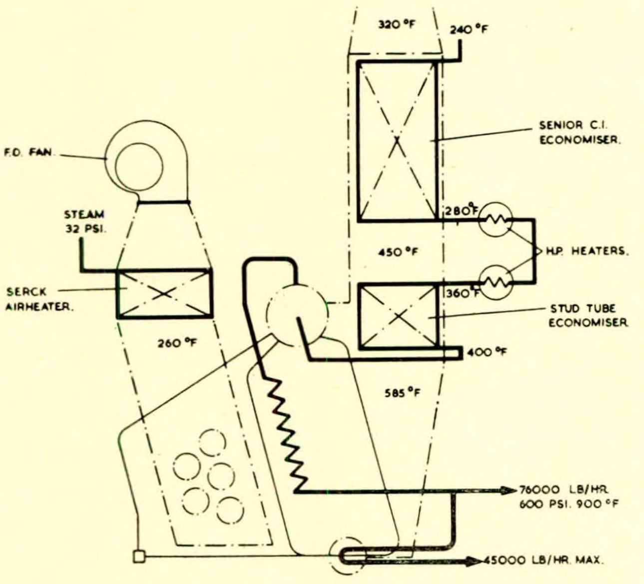
jacks placed in the central hinge; opening takes 63 seconds asd closing only 37 seconds. On D-deck, i.e. the 'tweendecks, there is a hatch of the same dimensions as on B-deck, fitted with two pairs of Mege covers, one opening forward and one aft. They are also planked. In this case exhaustive tests and practical operation have demonstrated that opening takes 30 seconds and closing 17 seconds. Opening and closing of each pair of covers is carried out by an hydraulic distributor placed near each hatch so that the operator can see the covers while opening and closing. The hydraulic installation is fed by a unit housed on D-deck.—*The Shipping World, 8th May 1957; Vol. 136, p. 452.*

Large British Tanker

The first of the large oil tankers of 38,000 tons d.w. on order for Shell Tankers, Ltd., has been delivered. This vessel, the *Zaphon*, built at the Wallsend shipyard of Swan, Hunter and Wigham Richardson, Ltd., is also the largest oil tanker to be built on the North East coast, and the largest to sail under the British flag. The *Zaphon* is powered by a set of geared turbines constructed by the Wallsend Slipway and Engineering Co., Ltd. During her sea trials a maximum speed of 17·79 knots was attained. The principal particulars of the *Zaphon* are as follows:—

Length o.a.	700ft.
Length b.p.	675ft.
Breadth moulded	89ft.
Depth moulded...	49ft.
Draught	36ft. 2½in.
Deadweight	38,390 tons
Gross tonnage	24,802 tons
Service speed	16½ knots
Machinery output	16,500 s.h.p.
Block coefficient	0·793
Cargo tank capacity	38,211 tons

The *Zaphon* is a single-screw vessel with geared steam turbine machinery aft. She has been built on the combined longitudinal and transverse framing system and has a raked stem of welded plate type, cruiser stern, poop, bridge houses amidships and forecastle. There is no sheer on the upper deck at the centre from the after perpendicular to the forward end of the forward cofferdam. The *Zaphon* is propelled by a single screw actuated by a geared turbine propulsion unit consisting of one ahead h.p. turbine of the all impulse type and one ahead l.p. turbine of the single flow impulse-reaction type. The gearing is of the double-reduction articulated type with flexible couplings at the turbines, the drive from the h.p. and l.p. primary gear wheels being transmitted to the secondary pinions through quill shafts and flexible couplings. The secondary pinions have been made hollow and the quill shafts pass through them to give maximum flexibility. The turbines work in conjunction with two watertube boilers having a working pressure of 600lb. per sq. in. gauge and a steam temperature of 850 deg. F. The propelling installation has been designed to develop a total service power of 15,000 s.h.p. when the propeller is rotating at about 105 r.p.m. and a maximum continuous power of about 16,500 s.h.p. at about 108 r.p.m. For astern working, one h.p. astern turbine of the impulse type is arranged at the forward end of the h.p. ahead turbine and one l.p. astern impulse turbine incorporated in the l.p. ahead turbine casing. When operating astern, the unit is designed to develop 65 per cent of the maximum ahead power, i.e. 10,700 s.h.p. The materials used in the construction of boiler and turbine components, etc., are suitable for continuous duty at 900 deg. F. The h.p. astern turbine consists of one impulse wheel carried on an integral extension of the h.p. rotor. The astern turbine casing is an entirely separate molybdenum-vanadium steel casting attached to the forward end of the h.p. ahead casing and fitted with vertical and horizontal keys to ensure correct alignment while maintaining freedom for expansion. The h.p. ahead turbine casing is a molybdenum-vanadium steel casting. The after end of the casing is supported by palms resting on a fabricated steel



Economizer and air heater design temperatures in B. and W. selective superheat boiler

pedestal attached to the h.p. turbine support girder. The l.p. ahead turbine is of the single-flow mixed-impulse and reaction type with five impulse stages followed by seven stages of reaction blading. The cylinder blades are carried by a cast steel casing fitted inside the fabricated outer casing. The steam inlet pipe has a flexible connexion between the inner and outer casings to allow for expansion. The l.p. astern turbine consists of one two-row and one single-row impulse wheel housed in a separate cast steel casing inside the l.p. fabricated casing. The steam inlet pipe has a flexible connexion between inner and outer casings to allow for expansion. Each turbine is connected to its primary pinion and each quill shaft to its secondary pinion through a flexible coupling consisting of a forged steel sleeve with small-pitch, machine-cut teeth at each end engaging with similar teeth cut in claw pieces secured to the rotor and pinion shafts. The main gearing is of the double-reduction articulated type. The pinions, both primary and secondary, are of forged alloy steel, the secondary pinions being made hollow to accommodate the quill shafts. The gear wheels are of the built-up type with rims and spindles of forged steel. The gear case is of fabricated steel construction designed so that all bearings are supported in the main structure. Covers are arranged to facilitate inspection and overhauling and sight holes are provided for sighting the ahead oil sprayers. Steam is obtained from two Babcock and Wilcox marine integral furnace controlled-superheat boilers, each fitted with a superheater and stud tube economizer followed by a Senior cast iron gilled tube economizer. A desuperheating element of the internal surface type is fitted in the water drum of each boiler. Each element is capable of desuperheating 45,000lb. of steam per hour. A Bailey automatic control system for fuel and air supply has been provided. The main control board is also of Bailey's make. The principal particulars of each boiler are as follows: —

Evaporation, normal ...	76,000lb. per hr.
Evaporation, maximum ...	87,000lb. per hr.
Steam pressure at superheater outlet	600lb. per sq. in.
Steam temperature at superheater outlet	850 and 900 deg. F.
Feed temperature	240 deg. F. and 360 deg. F. entering the economizers
Efficiency for normal evaporation ...	87·7 per cent
Number of oil burners	4
Total boiler tube heating surface	16,350 sq. ft. (8,175 sq. ft. each boiler)
Total superheater heating surface	3,800 sq. ft. (1,900 sq. ft. each boiler)
Total economizer heating surface, Babcock stud tube	5,840 sq. ft. (2,920 sq. ft. each boiler)
Total economizer heating surface, Senior economizer	13,824 sq. ft. (6,912 sq. ft. each boiler)

The superheater is designed to give a final steam temperature of 900 deg. F. Suitable means are provided for reducing this temperature and controlling it at 850 deg. F.—*The Shipping World, 29th May 1957; Vol. 136, pp. 524-528.*

Application of Gas Dynamic Theories to Exhaust Systems of Internal Combustion Engines

The primary purpose of an engine exhaust pipe is to convey the products of combustion from the cylinder to atmosphere. In so doing it must cause the minimum resistance to gas flow and in multi-cylinder engines causes the minimum interference to the charging of the cylinders. In an unsupercharged four-stroke cycle engine with no valve overlap the design of an exhaust pipe is not critical to interference effects, and, provided reasonable pipe sizes are used, the resistance effects are negligible. In the case of the two-stroke cycle engine, it was appreciated at an early stage in the development of these engines that the exhaust system had a critical effect on the charging and gas exchange process and hence on the overall performance of the engine. This arises from the cycle of events during the exhaust and recharging phases when the exhaust pipe and induction manifold are both open to the cylinder at the same time. In the earliest designs these effects were minimized by using exhaust pipes of large diameter and in some cases by fitting separate exhaust chambers to each cylinder; the former practice is still extant in many slow speed engines. At a later stage in the development of the internal combustion engine, attention was drawn to the pressure phenomena occurring in the exhaust pipe. Both theoretical and practical investigations showed that although the pressure phenomena were somewhat complex an approximate rational solution to the problem could be obtained by the use of the linearized join theory. In this theory, the pressure pulses propagated in to the exhaust pipe when the exhaust pipes are first opened are considered to act in a manner similar to sound waves. By the use of this analytical approach certain features in the design of exhausts system could be considered in the design stage. For many years this technique was the theoretical method available. With the advance in theoretical studies of gas flow during the war years, new techniques were made available by mathematicians to solve the complex equations of the type associated with gas flow in exhaust pipes. This paper is concerned with such techniques and how they may be applied to study certain physical features in the design of exhaust pipe systems.—*Paper by R. S. Benson, read at a Meeting of the Liverpool Engineering Society on 27th February 1957.*

Liquid Methane Carriers

In recent years, various proposals have been made for the carriage of natural gas from the Middle East to the United Kingdom and considerable thought has been given of course to the design of ships able to do this through the Suez Canal. So far no such ship has been ordered and experts seem to suggest that the conversion of existing ships to carry liquefied gas is not a practical proposition, though attention is drawn to the experience gained in solving the problems involved by the operation of barges built to carry liquefied natural gas up to the Mississippi to Chicago. The design of a methane carrier would have to be approached very much from first principles, although clearly there would be a great deal of data upon which to draw from the design, construction and operation of existing liquid petroleum gas and chemical carriers, whether self-propelled or of barge type. Problems of design and operation of specialized ships for the ocean transport of liquid methane were examined recently in a paper entitled "Liquid Methane", presented to the Institution of Gas Engineers by the chief engineer and development engineer respectively of the North Thames Gas Board. Three types are proposed. Firstly, ships of double hull construction with independent, insulated methane tanks arranged in groups within the holds. The tanks can be of vertical cylindrical type; of double shell construction with insulating materials between; or they can have a single outer shell with internal insulation only, capable of resisting liquid penetration. A horizontal disposition of tanks is also acceptable. The second type has the hull integral with large insulated liquid methane tanks filling practically the entire cross section of the hull, the tanks being either of cylindrical form with axes parallel to the bottom of the ship or long and spherical. The third type has large insulated cold boxes filling the entire space of each hull, vapour-tight, with steel walls insulated internally with balsa wood, fibre glass or other suitable insulating material. Inside these cold boxes are clusters of tanks of special material, for example stainless steel or aluminium, to contain the liquid methane. These tanks can be either of vertical cylindrical type or rectangular, the latter giving less broken stowage, discharge connexions and a boil-off vapour line. The authors appear to think that the third is the best solution since less space is wasted in insulation. It is also safer. Leakage or spillage of liquid gas from the internal metal tanks can be contained within the balsa-lined hold without contacting or damaging the hull structure. Indicating devices give warning of any leakage from the internal metal containers.—*A. C. Hardy, The Journal of Commerce, Shipbuilding and Engineering Edition, 11th April 1957; p. 5.*

Marine Steam Turbine Feed Systems

Depending on the auxiliary loads, the method of auxiliary drive, and the amount of regenerative heating employed, a range of ten per cent in fuel rate is found on analysing a number of tanker designs of similar power. In a specific design with given loads, alteration to the auxiliary drive and feed heating can alter the fuel rate by 6 to 7 per cent. The methods of auxiliary drive are examined and similarly the use of regenerative heating is evaluated. These are combined to give a "Cycle Ratio" which affords a useful criterion of the efficiency of a system. The cycle ratios of a number of feed circuits are compared and their relative first costs are briefly examined. While the more complex cycles are generally more efficient than the simpler ones, their cost is higher and this tends to lessen the saving from their adoption. Fairly simple cycles can be evolved that are both cheap and efficient, while in other cases the quest for simplicity may lead to too great an increase in fuel rate and hence an appreciable increase in cost taken over the life of the ship.—*Paper by A. D. Bonny, read at a Meeting of the North East Coast Institution of Engineers and Shipbuilders on 22nd February 1957.*

Vibration of Ships

Some investigations into the flexural vibrations of ships' hulls have been carried out. The author points out that such higher modes of hull vibration as are resonant with the propeller blade force excited through hydrodynamic action is considered as shearing vibration. The vibration profiles of the higher modes of hull vibration for certain assumed distributions of shear rigidity and of mass are calculated based on shearing vibration. Also, the natural frequencies of the fundamental mode to the eight-node mode of the vertical vibration of a cargo vessel, for example, are calculated based on shearing vibration, taking the corrections for bending moment and rotatory inertia of the hull section into account. To obtain a simple method for the estimation of the natural frequencies of hull vibration including the higher modes in the initial design stage an empirical formula based on shearing vibration is presented. The empirical factors of the formula are determined, based on data which have been supplied to the author by the leading shipbuilders in Japan.—*Toyoji Kumai, International Shipbuilding Progress, April 1957; Vol. 4, pp. 220-228.*

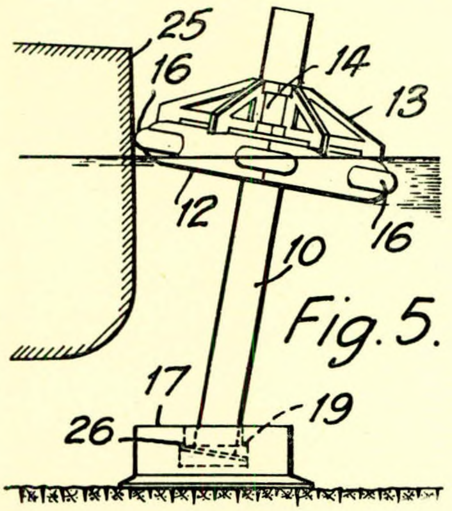
Ocean-going Motor Tug

The single-screw ocean-going tug *Clyde,* built by J. and K. Smit's Scheepswerven N.V., Inderdjik, for L. Smit and Co.'s Internationale Sleepdienst, Rotterdam, has entered the service of her owners. The ship is fitted with two large rudders so as to obtain the best possible steering qualities. Special protection has been provided to avoid fouling of the propeller by the hawser. The steering gear is of Stork-Jaffa electric-hydraulic design with tandem rams working in two cylinders. The rams are connected by tie rods to the rudder stocks of both rudders. The electrically-driven pump set is capable of turning the rudder from one hard-over position to the other within 20 seconds. The pumps are telemotor-controlled from the wheelhouse, the flying bridge aft and the wheelhouse top. In case of failure of the electric power supply, the operation of the steering engine is automatically switched over to hand steering. In addition, the telemotor unit placed on the flying bridge aft may be changed over to an emergency system of hand-hydraulic steering. The towing winch designed and manufactured by J. and K. Smit is fitted with two drums, each having a hydraulically operated spooling gear. As the drums can be used independently it is possible to handle two tows at the same time. The spooling gears are arranged to lay uniformly on the drums the turns of steel towing wires. A new feature is that the spooling gear is driven by threaded shaft and nut construction of patented design. The brakes of the drums can be operated in the towing winch compartment as well as from the flying bridge aft. A 50-in. warping drum is fitted on the main shaft of the towing winch outside the deckhouse. It is intended for the handling of the 22-in. manila hawsers and other towing gear. The main propelling machinery consists of two four-stroke, single-acting direct reversible turbocharged Smit-M.A.N. Diesel engines, type R.B. 666. The engines have a total output of 4,000 h.p. The diameter of each of the cylinders is 450 mm. and the stroke 660 mm. The cylinder liners are chrome-hardened in accordance with the "Porus Krome" process and the pistons are of a specialized type of light alloy obviating any piston cooling arrangement. The two engines, which are built by J. and K. Smit, drive a single propeller through a Lohmann and Stolterfoht reduction gear of the 2 ES 125 type. The gear case is arranged for a total output of 4,000 h.p. and is fitted with a built-in thrust bearing for a thrust of 40,000 kg. Lohmann reverse gears, type HF 12, are fitted between the two propelling engines and the gear case. These gears are arranged for a maximum output of 2,000 h.p. Reversal of the turning direction of the propeller is achieved by Airflex couplings. These are pneumatically operated and clutched separately so that the direction of the propeller is reversed. In this manner the machinery can be reversed within 6 seconds.—*Holland Shipbuilding, April 1957; Vol. 6, pp. 28-32.*

Patent Specifications

Dolphin

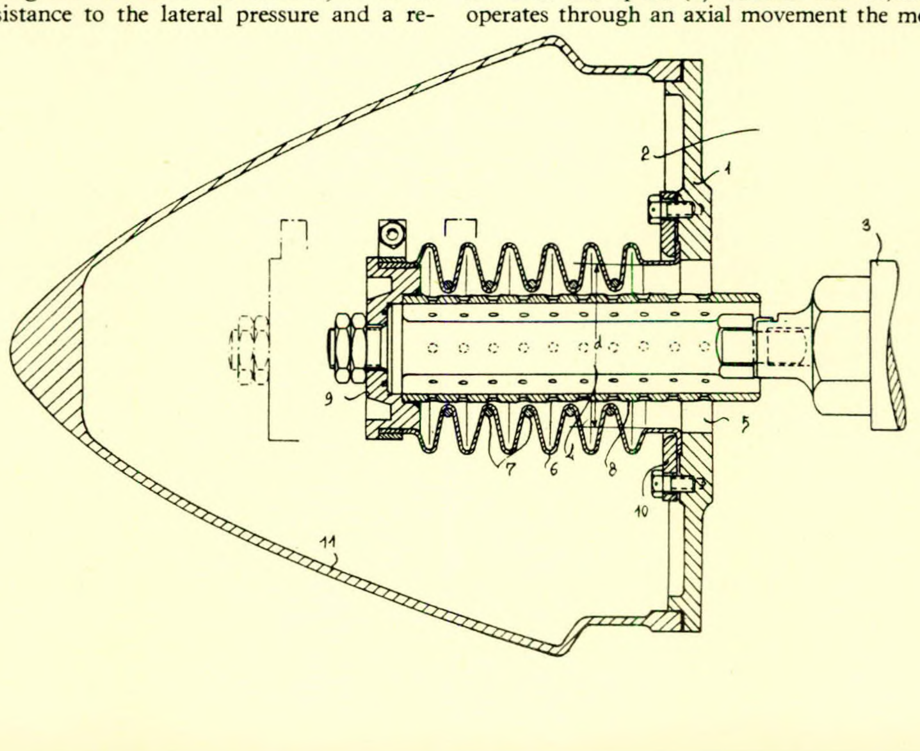
This invention relates to dolphins or bumper buoys employed for receiving and absorbing impact from a ship's side during its berthing, mooring or other movements, as in going alongside a wall or jetty or in passing a pier supporting a bridge. According to the invention, the dolphin comprises a pontoon, a column extending down through the pontoon, and an anchorage member to which the base of the column is connected for tilting movement from the vertical, the pontoon being free to rise and fall in relation to the anchored column



when the latter is vertical but being provided with means for engaging the column under lateral pressure due, for example, to impact from a ship whereby the pontoon becomes partly submerged by the tilting movement of the column, so as to provide a resilient resistance to the lateral pressure and a restoring force to bring the column back to the vertical. When a vessel bears against the floating pontoon (12) of the anchored dolphin, as shown in Fig. 5, the side (25) of the vessel will impact against the pontoon, engaging with one or more of the fenders (16) and forcing the hub (14) laterally against the column (10). The pontoon and the column will then rotate together about a point (26), where the flange of the column maintains contact with the annular surface of the ring (19) within the clump. The upper part of the column will therefore become partly submerged, and the pontoon (12) will also be partly submerged because its vertical or sliding movement relative to the column is arrested by the frictional grip of the hub (14) upon the surface of the column. The energy of the ship's impact is therefore absorbed by the increased buoyancy of the column and the pontoon provides a resilient resistance and a restoring force in opposition to the momentum of the vessel.—*British Patent No. 778,450, issued to R. Pavry. Complete specification published 10th July 1957.*

Propeller with Adjustable Blades

This invention relates to a ship's propeller with adjustable blades, the adjustment being effected by means of a rod moving in a bore in the propeller shaft. The propeller boss and the bore are filled with oil and the propeller is provided with a device for compensating the change of volume of the free space in the boss owing to the reciprocating movement of the adjusting rod and for sealing the mechanism in the boss in relation to the outboard water. The adjusting rod or its extension passes through an opening in the back wall of the boss chamber and is sealed in relation to the latter by means of a concentric bellow, the area of the operative cross section of the bellows being equal to the area of the cross section of the adjusting rod where the latter enters the boss chamber in front. In the drawing the back wall of the boss, i.e. the back wall of the boss chamber (2), filled with oil, is designated by 1. Into the space (2) extends the adjusting rod (3), which operates through an axial movement the mechanism for adjust-



ing the propeller blades. The end (4) of the adjusting rod passes through the opening (5) in the wall (1). It is surrounded by a bellows (6), which may be made of flexible material, such as canvas-reinforced rubber, but possibly also of metal.—*British Patent No. 777,908, issued to N.V. Koninklijke Maatschappij De Schelde. Complete specification published 26th June 1957.*

Preventing Electric Currents in a Ship's Hull

This invention relates to ships, and in particular to mine sweepers, having a metallic framework constituted by light

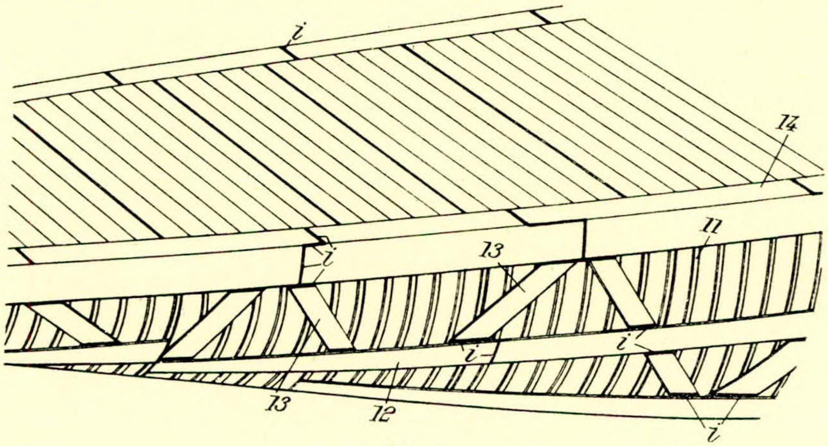


Fig. 1

metal girders and plates. This assemblage of girders and plates constitutes a network of electrical circuits which are highly conductive because the cross sectional areas of the elementary girders and plates are very large and the light alloys have very low resistivities. When such a ship is moving under the effect of waves, the conductive network thus constituted undergoes oscillatory displacements of substantial amplitude in the earth's magnetic field and due to this relative displacement currents are induced in the electrical circuits formed by the girders and plates of the network. These currents have high intensities and create about the ship a variable field of a period corresponding to that of the rolling movement of the ship, and of an amplitude the higher as the ship is rolling more.

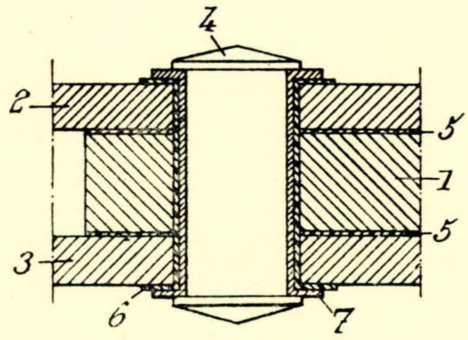


Fig. 2

The existence of such a field is highly objectionable because it makes the ship detectable. In particular, if the ship is a magnetic minesweeper, this field constitutes a serious danger even when the rolling amplitudes of the ship remain within acceptable limits. The object of this invention is to provide a ship which is free from this drawback. The framework shown in Fig. 1 includes frames (11) assembled with strakes (12), diagonals (13) and planksheers (14). These framework units are constituted by the assemblage of juxtaposed elements such as girders and plates made of light alloy. According to the invention, at least some of the lines or surfaces of the assemblage of consecutive plates or girders are provided with electric insulating means (such as *i* in order to prevent electric connexion between two adjacent girders or plates. Fig. 2 shows an example of such insulation.—*British Patent No. 777,834, issued to F. Amiot. Complete specification published 25th June 1957.*

Gravity Davit

Referring to Figs. 1 and 2, the davit comprises a fixed bracket (10) and a movable stand (11), the bracket (10) being mounted, for example on a ship's deck. The stand (11) is pivotally attached to the bracket (10) as indicated at (12). The davit arm is indicated (at 13) and this is pivotally mounted (at 14) on the stand. A rest or support (15) for the

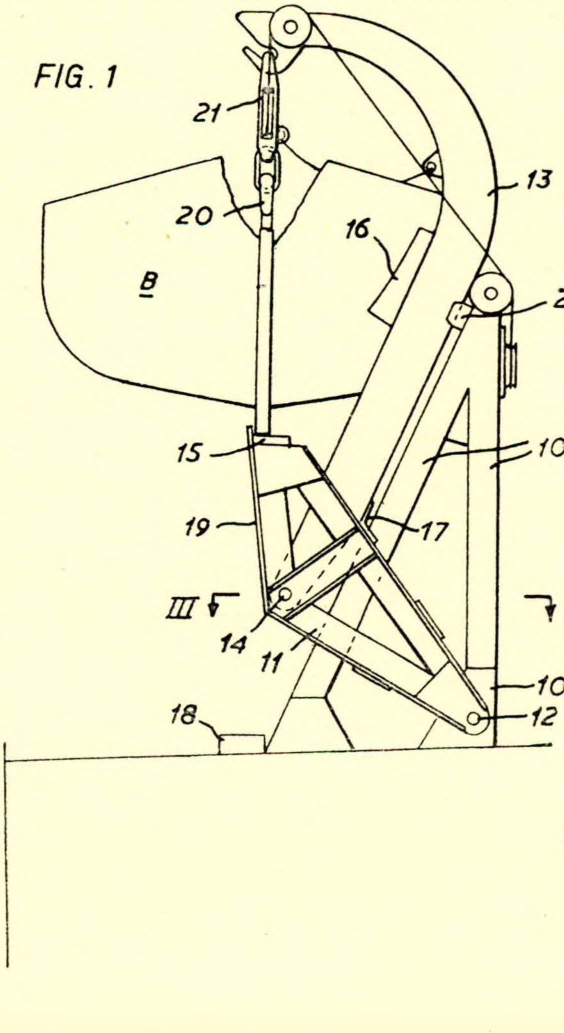
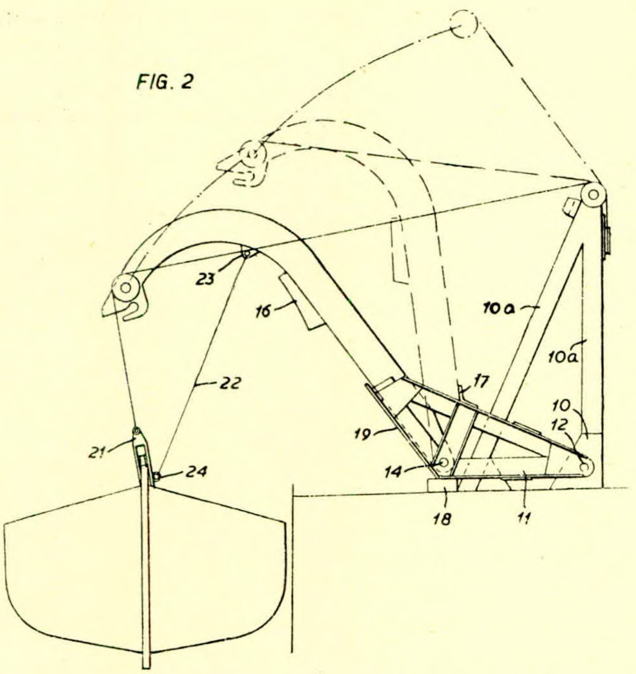


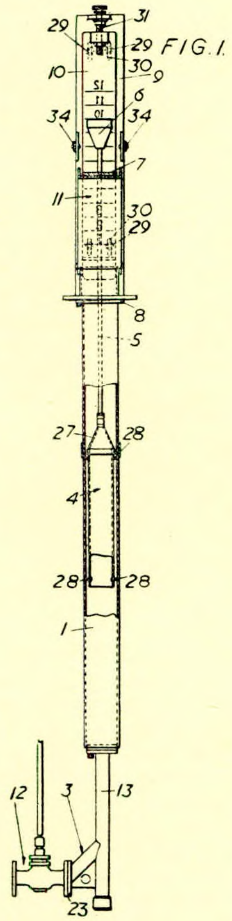
FIG. 2



keel of the boat (B) is provided on the free end of the movable stand (11). A breasting block (16) is provided on the davit arm for the side of the boat. In its fully inboard position the arm (13) rests against a supporting bracket or inturning stop (17) on the stand (11). Outboard movement of the stand (11) is halted by a stop or block (18), on, for example, the ship's deck whilst outboard movement of the arm (13) is halted at the fully outboard position by a stop plate or the like (19) on the foremost end of the movable stand. Numeral 20 indicates the hook on the boat and it will be noted that the span or tackle (21) connecting this hook to the horn on the davit arm head is very short. Consequently even when the tackle (21) leaves the horn (Fig. 2) and a part of the fall rope is paid out to bring the boat to the embarkation position, the span or length of tackle between the horn and the hook is still comparatively short and thereby excessive uncontrolled swing of the boat is avoided.—*British Patent No. 778,030, issued to Marepa Trust, Ltd. Complete specification published 3rd July 1957.*

Liquid Level Indicator

This invention relates in particular to an apparatus for determining the loading of ships with respect to their Plimsoll lines. Since water conditions in harbours are rarely absolutely calm it is frequently difficult to assess accurately the state of loading of the ship by reference to the position of the Plimsoll line with respect to the disturbed water level, with the result that ships are often underloaded or considerable time is taken in checking this position, thus slowing down loading. It is an object of the present invention to provide an apparatus suitable for use *inter alia* in connexion with ships which will give an accurate reading of the mean water level, thus enabling the state of loading of ships to be accurately determined. The apparatus shown in Fig. 1 is put into communication with the water through a sea cock, the water entering the chamber



(1) through the tube (2), the float thus being rendered buoyant, being supported on a cushion of trapped air. Within a short period of time the pointer will achieve a steady position whence a reading of the mean level may be obtained by reference to the scale (10). It will be appreciated that the Plimsoll circle on a ship is 5 inches radius and if an appropriately selected apparatus according to the invention is put in communication with the sea as the apparent water level approaches the lower edge of the Plimsoll circle, the pointer will be giving a steady reading of the mean outside water level with reference to the Plimsoll line before the ship is overloaded so that the final loading up to the Plimsoll line can be continuously observed.—*British Patent No. 778,740, issued to W. S. Adams. Complete specification published 10th July 1957.*

Marine Engineering and Shipbuilding Abstracts

Volume XX, No 11, November 1957

* Patent Specifications

Nuclear Propulsion for Polish Ships

Poland is the largest shipbuilder of the Soviet bloc next to the U.S.S.R. itself; by 1960 it will be launching 200,000 tons a year. Biggest ships built so far are 10,000-ton motor freighters for Far Eastern service, but a series of 18,000-ton tankers is planned. Poland appears to be the likely country to specialize in nuclear propulsion for merchant ships—which in the Soviet is already held to be more economic than conventional power. The Nuclear Research Institute jointly with the Maritime Institute and the Central Bureau of Ship Construction are already studying the economics of the matter.—*Nucleonics, April 1957; Vol. 15, p. 26.*

Largest Motor Tanker

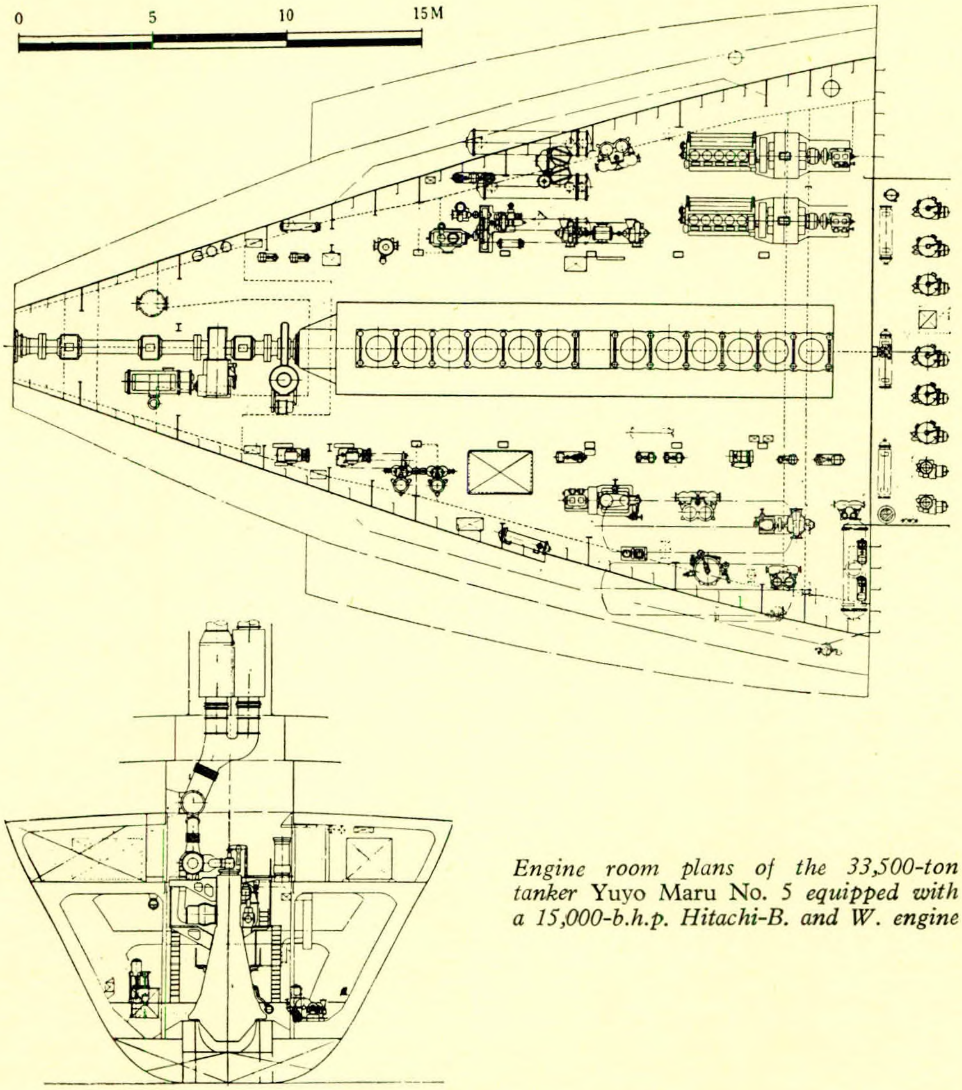
The *Berge Sigval,* completed by Eriksbergs Mek. Verkstads A/B, is the largest Diesel engined oil-carrying ship yet completed. She has been built for Sig. Bergensen d.y. and Co., Stavanger, and has been chartered to the British Tanker Company for four years. She sailed directly after her trial trip to Bandar Mashur via the Suez Canal. The keel was laid on 12th February, the vessel was launched on 14th May and she was completed in 119 working days. The particulars are:—

Deadweight capacity ...	35,750 tons
Length overall	681ft. 10in.
Breadth, moulded ...	87ft.
Depth, moulded	47ft. 6in.
Draught, fully loaded ...	35ft. 9in.
Machinery	12,500 b.h.p.
Contract speed	16 knots
Cargo capacity	1,669,000 cu. ft.

The new tanker is all-welded and has been built on the longitudinal and transverse bulkhead corrugated system developed by the shipyard. Guardion cathodic protection is provided for the hull. There are 11 centre and 22 side tanks. In the pump room between tanks 2 and 3 are two vertical compound 750-ton pumps and in the aft pump room two electrically driven centrifugal pumps of the same capacity, the driving motors being placed in the engine room. The winches and anchor winches are steam driven and the steering gear is of the electric hydraulic type. Nautical equipment includes radar, a gyro-compass with auto-pilot, echo sounding apparatus and Sal log, and there are four aluminium alloy lifeboats, two of which are propelled by Diesel engines. The propelling engine is a standard Eriksberg-B. and W. two-stroke unit with 10 cylinders (740-mm. bore with a piston stroke of 1,600 mm.) developing the rated power at 115 r.p.m.—*The Motor Ship, August 1957; Vol. 38, p. 200.*

15,000-b.h.p. Engine on Test

Official trials were carried out recently of a 15,000-b.h.p. twelve-cylinder Hitachi-B. and W. Diesel engine built for installation in the Morita Steamship Company's 33,500-ton tanker *Yuyo Maru No. 5.* The results of the trials are given in the accompanying table. The engine has cylinders 740 mm. in diameter and the piston stroke is 1,600 mm. The rated output is developed at 115 r.p.m., corresponding to a brake mean pressure of 101lb. per sq. in. During the test bed trials there was a notable absence of vibration. At full load the fuel consumption was 159 gr. per b.h.p. hr. and at 11,370 b.h.p.—75 per cent load—the figure was 155·8 gr. per b.h.p. hr. at 105 r.p.m. At 50 per cent load (7,590 b.h.p.) the consumption was 161 gr. per b.h.p. hr. Even when the engine was developing about 25 per cent output (3,760 b.h.p.) the figure was no more than 172 gr. per b.h.p. hr. (0·385lb. per b.h.p. hr.). The mechanical efficiency was 89·5 per cent at full load, falling to 77·1 per cent at quarter load. The speed of the turboblowers ranged from 6,900 r.p.m. at 15,000 b.h.p. to 2,950 r.p.m. at 25 per cent load. There are four Rateau exhaust gas turbochargers and, after the main tests were concluded, trials were carried out with alternately three and two turbochargers in service, thus cutting off three and six cylinders



Engine room plans of the 33,500-ton tanker Yuyo Maru No. 5 *equipped with a 15,000-b.h.p. Hitachi-B. and W. engine*

respectively. Under these conditions the engine ran at 95 r.p.m. and 70 r.p.m. respectively, the corresponding outputs being 8,450 b.h.p., and 3,400 b.h.p. It is anticipated that the daily fuel consumption for all purposes will be forty-nine tons with the ship operating at 15¾ knots. Test results of 15,000-b.h.p. Hitachi-B. and W. engine:—

Load percentage	100
R.P.M.	115
Indicated mean pressure, kg./cm.² ...	7·94
Maximum pressure, kg./cm.²	53·4
I.H.P.	16,800
B.H.P.	15,000
Mechanical efficiency, per cent	89·5
Pressure kg./cm.²:	
Cooling oil	1·75
Lubricating oil	1·65
Scavenging air	0·34
Temperature, deg. C.:	
Exhaust gas, cylinder outlet ...	365
Cooling oil, engine inlet	32
Cooling oil, engine outlet	44
Cooling water, engine inlet	59
Cooling water, engine outlet ...	63
Scavenging air (after-air cooler) ...	35
Sea water	19
Test room	22

Fuel consumption:	
Gr./i.h.p./hr.	142·3
Gr./b.h.p./hr.	159·0
R.P.M. of turbocharger	6,900

The arrangement of the machinery is shown in the accompanying plan. There are two Diesel-driven generators and a dynamo is driven off the propeller shaft.—*The Motor Ship, July 1957; Vol. 38, p. 147.*

Application of Thermostats to I.C. Engine Cooling System

The history of the thermostat is shown and the way in which effective engine-coolant temperature control has been obtained through the years. The main reasons for the use of thermostats are: (1) Lower engine wear. (2) Improved thermal efficiency. The basic design requirements for the ideal engine thermostat are then enumerated. The paper discusses how near the modern thermostat comes to these requirements. The poppet type and butterfly type valve are used for thermostats, the advantages and disadvantages of each type being discussed. The main problem is dirt, which becomes trapped in the valve seating and increases the warm-up time of the cooling system. The design features of the bellows type and wax type thermostat, and the fillings used in the bellows type, are discussed. The damping of vibration periods by hydraulic and friction methods are shown. The wax type thermostat has gained popularity because of its insensitivity to pressure when the coolant system is pressurized. The heat transfer of the wax pellet is slower than the vapour in the bellows type, and the reasons for this are given. Thermostat installations are discussed and illustrated.—*Paper by S. H. Blazey, read at a Meeting of The Institution of Mechanical Engineers on 12th February 1957.*

Container Ship Design

The container ship is designed to carry large and light objects of standardized size, generally in the form of closed containers with perishable or fragile contents. The nature of the cargo presents the designer with an unusually interesting but also difficult problem, and the successful container ship design depends on a very careful appreciation of all the basic factors involved. One must consider the container before the ship which is to carry it, and immediately a difficult problem is presented. International standardization of containers is at an early stage and if a ship is to carry a certain type and range of containers with a high degree of stowing efficiency, it may be quite inflexible on other routes involving other types of container. For example, the standard B class railway container in the United Kingdom stows to a first degree of approximation within a rectangular space, 17ft. (5·2 m.) long with a cross-section of 8ft. (2·44 m.) square. Fundamentally, there are two methods of stowing containers which may be considered as being longer than they are wide. These containers may be stowed with their major axis athwartships, or with their major axis longitudinally. The method of stowing and the size of containers in a given category of ships will decide the dimensions of the vessel. For example, if one is to consider containers of 8ft. (2·4 m.) width and 8ft. (2·4 m.) height stowed with their major axis longitudinally, then the beam of the ship must be related to a multiple of 8ft., i.e. it must be 24ft. (7·3 m.) plus stowing allowances, or 32ft. (9·75 m.) plus stowing allowances, giving a breadth of approximately 29ft. (8·8 m.) in one case and 37ft. (11·25 m.) in the other. If these types of containers are to be stowed five abreast, the beam will then be approximately 45ft. (14 m.). An example of a hypothetical vessel is given. The requirement has been postulated that the ship should carry about 110 B-class containers at 15 knots service speed, and it should be emphasized that this ship is not related in any way to ships either built or building in Britain, and is purely an example. Hypothetical ship design:—

Principal dimensions:

Length, b.p. ...	...	300ft. (91 m.)
Breadth moulded	...	46ft. (14 m.)
Depth moulded	...	16ft. (4·9 m.) to second deck
Depth moulded	...	25ft. 6in. (7·75 m.) to upper deck
Loaded draft ...	...	14ft. (4·3 m.)
Service speed ...	...	15 knots

This vessel is designed to carry approximately 110 containers, and is of full scantling type with a continuous aluminium alloy trunk on the upper deck. The vessel is propelled by Pescara gas turbine machinery, situated at the after end, coupled to a single screw, oil fuel being carried in an engine room double bottom tank. The height of the double bottom is 5ft. 6in. (1·68 m.), this depth being necessary to provide adequate water ballast capacity to maintain positive stability in all conditions of loading. In way of the hold the double bottom is divided longitudinally into five compartments and transversely into three compartments, this being necessary to adjust the stability and seakindliness of the vessel in any condition. A deep tank forward and a large forepeak have been arranged in order to give the vessel reasonable trim in the ballast condition. The tank top is all welded and perfectly flush so that, if necessary, roller gear for moving the containers out to the wings of the ship may be used, or, alternatively, fork-lift trucks. The second deck is arranged with flush type MacGregor hatch covers, which stow at each end of the holds in a recessed well, as shown. The upper deck in way of the trunk is fitted with roller type beams on to which the containers may be placed and secured in position. When not in use, the roller beams are stowed in a recessed well at the ends of the cargo holds. Ordinary type MacGregor aluminium hatch covers are arranged on the trunk top and stow at each end of the hatch as shown. Each hatch is served by one travelling crane, which is capable of movement fore and aft along a guide on the trunk side.—*E. C. B. Corlett, The Shipping World, 10th July 1957; Vol. 137, pp. 31-34.*

Air Conditioning System

The cargo motorship *Queensville,* recently completed by Lithgows, Ltd., Port Glasgow, for A. F. Klaveness and Co., A/S., Lysaker, Oslo, is the first ship to be fitted with the Colvin-Smith Hi-Pres air conditioning system. Incorporated in the Hi-Pres plant are a conditioning central unit, piping and cabinets. An air filter, inlet sound trap, preheater, cooler, high pressure centrifugal fan and outlet sound trap are contained in the central unit. Conditioned fresh air is piped from the central unit to the cabinets in the cabins and public spaces. The small area of the air pipes in the Hi-Pres system reduces the problems of installation to a minimum. Pipes ranging from 100 mm. to 45 mm. are employed, and standard elements are used throughout. The volume of air delivered by the fan is generally too large for one Hi-Pres pipe, and the central unit is often arranged to discharge into three or four pipes. The individual pipes are assembled with sleeves, which have a packing ring of non-inflammable rubber to ensure airtight connexions and sound insulation between the parts. Full-scale fire tests of the Hi-Pres system have been carried out at the Danish State Test Laboratory. The system only uses fresh air; and as the recirculation of air only takes place in each room, the amount of oxygen is not sufficient for a fire, which might occur in one room, to spread dangerously. A further advantage is that the small-diameter pipes restrict the passage of smoke and inflammable gases, so that the risk of fire spreading through the ventilating system is virtually eliminated.—*The Shipbuilder and Marine Engine-Builder, July 1957; Vol. 64, p. 457.*

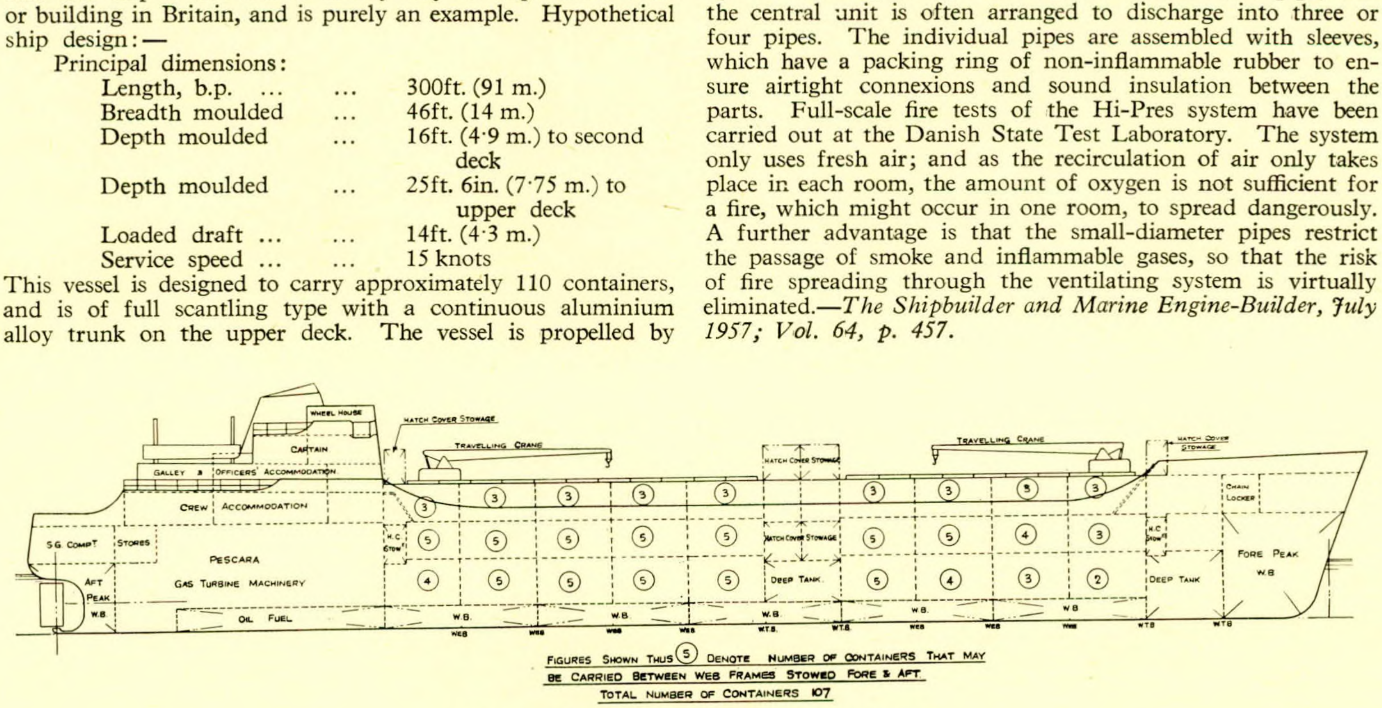
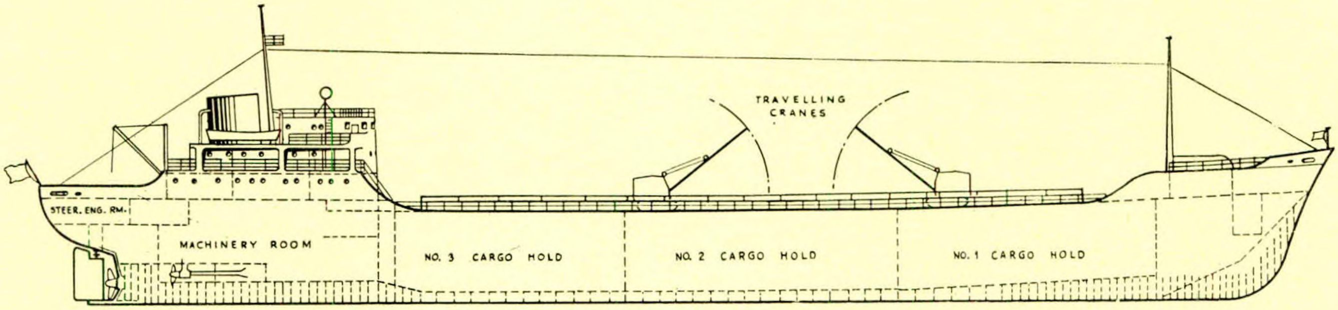


FIG. 2—*Hypothetical container ship design*



Japanese-built triple screw bulk carrier

Japanese-built Triple-screw Bulk Carrier

A triple-screw motor vessel built in Japan for Inagua Transports Incorporated has recently been delivered. This vessel, the *Cécile Erickson,* 5,500 tons d.w., embodies several features of interest which include cargo handling cranes which may be stowed away in hangars on the forecastle when not in use, and probably the longest continuous folding steel hatch covers on any ship yet built. The machinery, which consists of four Diesel engines driving three propeller shafts, is also an unusual arrangement for a cargo vessel. The *Cécile Erickson* has been built by the Osaka Shipbuilding Co., Ltd., as a bulk carrier, particularly for the carriage of salt from the Great Inagua Island to the East Coast of the United States of America. Her keel was laid on 18th June 1956, she was launched on 4th December of the same year and delivery took place about four months later. The principal particulars of the *Cécile Erickson* are as follows:—

Length o.a.	373ft.
Length b.p.	344ft. 6in.
Breadth moulded	50ft. 6in.
Depth moulded	25ft. 7in.
Draught	20ft. 3in.
Deadweight	5,500 tons
Gross tonnage	3,070 tons
Hold capacity	206,975 cu. ft.
Machinery output	2,000 b.h.p.
Service speed	12 knots

The *Cécile Erickson* has been built under the special survey of Lloyd's Register of Shipping and has longitudinal framing at the bottom and two longitudinal and five transverse water and/or oiltight bulkheads. This gives three holds, each consisting of a cargo space at the centre with wing tanks for ballast at either side. As will be noted from the drawing, the accommodation, bridge and machinery are all installed aft. When the vessel is secured for sea, the deck is completely clear, apart from the continuous hatch coaming. Cargo handling is effected by means of two Thew-Lorain crawler-type shovel cranes powered by Caterpillar Diesel engines. The cranes run along wooden tracks arranged on either side of the hold, and when it is impossible to work cargo from both sides two cranes can be moved to whichever side of the ship they are required. Two large storage hangars are arranged in the forecastle so that when the vessel puts to sea the cranes can be run in under cover, leaving the deck perfectly clear. So that the cranes may be run out of the hangars on to either side of the deck before the hatch cover is opened, the forward cargo hatch ramp is made so that it can be folded up out of the way. The three cargo holds have one continuous hatchway covered by a 190-ft. long MacGregor patent steel hatch. This unusually long hatch cover consists of twenty-nine sections, and when the hatch is open twenty of these sections are stowed forward and the remaining nine aft, leaving a completely clear hatch for loading or discharging. The *Cécile Erickson* is powered by a set of four standard Caterpillar turbocharged type D397 marine Diesel engines, each developing 500 h.p. at 1,225 r.p.m. Each engine has twelve cylinders having a diameter of 5¾in. and 8-in. stroke. The two inboard engines are connected to the centre propeller shaft through Lufkin compound reduction gearing. Each of the outboard engines drives a separate propeller through Lufkin gearing. The wing propellers are geared down to 280 r.p.m., and the centre propeller to 203 r.p.m. Provision has been made in the design of the engine room for the installation of two more engines if it is required. The engines are pneumatically controlled, the manœuvring station being located on the starboard side of the engine room. The main reason for installing three propellers is that the *Cécile Erickson* has to operate in seas containing many coral reefs, so that a very high degree of manœuvrability is necessary. Furthermore, in having four small Diesel engines there is the advantage of easier maintenance, and if one engine breaks down during service it can be repaired without stopping the ship.—*The Shipping World, 19th June 1957; Vol. 136, pp. 580-581.*

Casting of Alloy Pistons

The metallurgical/foundry aspect is equally important as the design side when considering pistons capable of catering for the operating conditions of the highly rated oil engines now coming into service and it can safely be stated that the actual machining of the piston is relatively unimportant, particularly from the visual aspect. The most important machining operation is the ring grooving which must be sufficiently accurate to avoid chatter that can cause heavy oil consumption, blowby and ultimate seizure. For large pistons, above 8-in. diameter, the material which has been most commonly used is Y-Alloy, having a 3·5 to 4·5 per cent copper and a 2·5 per cent nickel content, but unfortunately this material has a high coefficient of expansion which necessitates large cold clearances, an undesirable feature since the present trend with engine design is to reduce the overall length of pistons, and a length/diameter ratio of 1·3 : 1, or even less is common. Such pistons tend to "toe and heel" when idling or running on light load, and this is often responsible for heavy groove wear and other undesirable troubles. The obvious answer to this problem is the use of low expansion silicon alloys, and engine tests and service experience with pistons manufactured from this material have given highly satisfactory results. Some designers are opposed to the use of silicon alloys for large diameter pistons, and the objection is usually based on the slightly lower hot strength quoted in text books for this class of material, by comparison with high copper content Y-Alloys. The figures given often refer to chill-cast test bars, but the improved mechanical properties obtained with die-casting methods, with the application of accelerated chilling, together with the effect of heat treatment, are not taken into account. Experience has shown that when using high silicon content alloys for large diameter pistons which normally have heavy sections on the crown, ring belt and above the gudgeon pin bosses, the semi-chilling method (i.e. sand coring with a metal die-body) is undesirable and, with these alloys, it is essential to use a chilled die-body and core. Laboratory investigations carried out on the properties of chilled core, die-cast pistons and sand core, die-cast pistons leave no doubt that the chilled core is the better method, as the structure of the alloy is improved substantially with an appreciable increase in the tensile strength. Test pieces taken from the crown, ring belt and boss strut sections show that in every case, both in the "as-cast" and heat treated conditions, higher values are obtained from pistons cast by the chilled core method. Fatigue tests show that such castings have an appreciably higher fatigue strength and greater uniformity between the various parts of a piston. The most favoured form of cooling is water to the die-body and air to the core, but low pressure steam is under consideration for this application as it is thought that this medium will maintain a more uniform temperature throughout the die and core. Accelerated chilling relative to the heavy sections prevents feeding by the lighter sections which are the first to chill, thus producing a coarse irregular grain structure due to shrinkage at a point where the heavy crown sections merge into the ring belt and the boss struts into the skirt. Both are vital points where mechanical failure of a piston is likely to commence. Experiments have shown that silicon alloys have a greater resistance to wear than most Y-alloys, and this is all the more significant with hyper-eutectic alloys, which are undoubtedly the next step in piston development. Such alloys have a 20 to 25 per cent silicon content and very high Brinell figures can be obtained with heat treatment, 150-160 being common. This material also has a very low coefficient of expansion, enabling a substantial reduction to be made on cold clearances which are often comparable with those of a cast-iron piston; this is a most important step forward.—*Gas and Oil Power, May 1957; Vol. 52, pp. 122-123.*

British Polar Turbocharged Engine

The first turbocharged five-cylinder British Polar Diesel engine to be built at the Govan works has now completed test bed trials and has been installed in the trawler *Samarian.* The new engine is a standard Polar M45M type, direct-reversing, loop-scavenged, two-stroke unit having a bore of 340 mm. and a stroke of 570 mm. The normally-aspirated version of this engine operates at a b.m.e.p. of 78 to 80lb. per sq. in. over the range of 220 to 300 r.p.m., giving outputs of 700 b.h.p. to 935 b.h.p. The turbocharged engine is rated at 100lb. per sq. in. b.e.p., which over the same speed range gives an output of 880 b.h.p. to 1,170 b.h.p. This is equivalent to 25 per cent supercharge. The engines for the *Samarian,* and her sister ship *Rhodesian,* have been rated at 960 b.h.p. at 260 r.p.m., which is equivalent to 15 per cent supercharge. The *Samarian* has been built by Cochrane and Sons, Ltd., Selby, for the Onward Steam Fishing Co., Ltd., an associate of Sir Thomas Robinson and Son (Grimsby), Ltd. The *Rhodesian* is being built at Selby for the same owners. During trials on the river Humber a mean speed of 11½ knots was obtained over the measured mile with the engine running at 260 r.p.m. The mean of the exhaust temperatures was about 470 deg. F., from which it was apparent that at the service rating, and in free running condition, a considerable reserve margin of power was available. In order to use up some of this margin the speed was increased to 275 r.p.m., which improved the performance of the vessel by about ¼ knot, the exhaust temperatures rising to 550 deg. F. It was estimated that at this rating about 10 per cent boost was in use, and bearing in mind the performance of the engine on the test bed, it will be appreciated that the unit was running well below its full capacity. Reports of the trials state that the noise level throughout the full range of engine speed and loading was very low, and that the exhaust was remarkably clean, indicating perfect combustion and consequently a high degree of efficiency. The engine is turbocharged using the constant pressure system shown diagrammatically in Fig. 1. Air is induced by a Napier TS300 turbocharger (1) and delivered to the engine driven scavenging air pump (3) through a Serck air cooler (2). In the scavenge pump the air is further compressed and delivered through a Serck aftercooler (4) to the engine inlet manifold (5). After combustion the exhaust gases are led from a common exhaust manifold (6) to the twin-entry casing of the turboblower.—*The Shipping World, 24th July 1957; Vol. 137, pp. 79-80.*

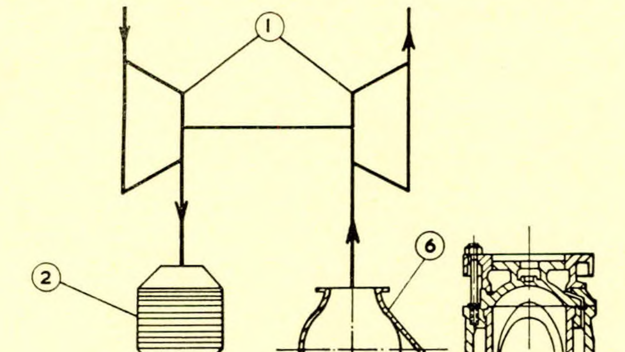


FIG. 1—*Diagram showing turbocharging system*

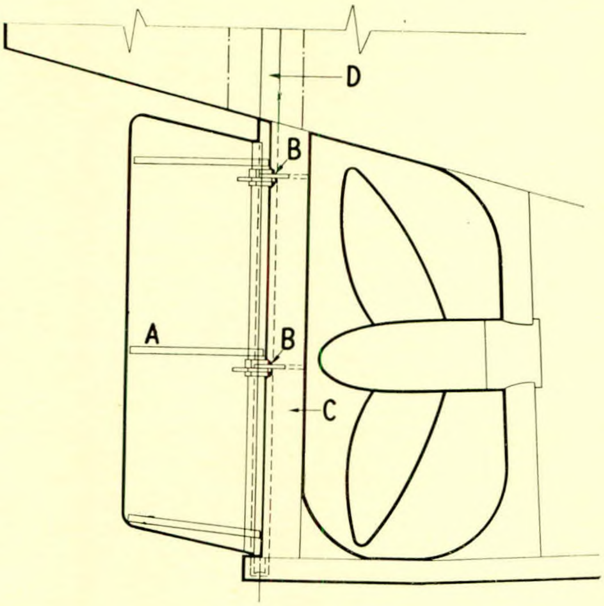
1 Exhaust turbocharger
2 Intercooler
3 Engine scavenge pump
4 Aftercooler
5 Inlet manifold
6 Exhaust manifold

Hydroconic Diesel Tug

A hydroconic tug has recently been delivered to Samuel Williams and Sons, Ltd., for use on the river Thames. This vessel, the *Arthur,* has been built by T. Mitchison, Ltd., Gateshead, to the instructions of Seawork, Ltd., London. The *Arthur* is used for barge towing and the propeller has been designed on the basis of a 4-knot towing condition. This means that the optimum towing performance is obtained when the tug and its tow proceed at four knots. In consequence, the static bollard pull is not nearly so high as could have been attained with a bollard propeller; nevertheless, on trials the vessel gave a bollard pull of not less than 7¼ tons, which, for 460 s.h.p. gives the excellent ratio of 63½ h.p. per ton, this being achieved at 250 r.p.m. on the propeller absorbing the full load of both engines. The free-running speed of the tug is 9·47 knots and this can be achieved with either one engine running at full speed, or both engines at half load. One feature of interest is the bulb rudder with which this vessel is fitted, and which is a patent of Burness, Corlett and Partners, Ltd. The initial bollard pull, taken before the bulb was fitted, gave a result of 6·85 tons, from which it will be seen that the thrust augmentation obtained from the bulb is about 6 per cent. This percentage is, of course, not the optimum and is due to the characteristics of the propeller. Fitting a bulb to the rudder has also improved the steering. The principal particulars of the *Arthur* are as follows:—

Length o.a.	72ft. 3in.
Length b.p.	66ft.
Breadth moulded	18ft. 3in.
Depth moulded	9ft. 6in.
Draught aft	8ft. 6in.
Engine output	460 s.h.p.

The hull is of hydroconic construction, and this has enabled the shell plate forward to be ¾in. thick without the considerable expense necessary to mould material of this thickness, and



The B.C.P. bulb rudder

A. Plate rudder
B. Bending stop
C. Plate bulb
D. Rudder stock

combined with frame spacing of 14in. has provided great strength. This construction has also enabled engine seatings of enormous strength and rigidity to be provided as well as allowing an extremely free run of water to the propeller. The deck arrangement has been designed to obtain the maximum amount of all-round visibility from the wheelhouse and for this reason the orthodox funnel has been omitted, the main engine exhausts being led up small trunks built into the aft corners of the wheelhouse structure. This gives an excellent view of the deck aft over the sloped machinery casing. The rudder consists of a fixed fabricated plate bulb approximately $\frac{3}{8}$in. thick which is faired into the rudder stock. To the stock is welded a single plate rudder stiffened by horizontal arms. Two or more bearings which act as bending stops are attached to the bulb. This means that the stock need only be of sufficient diameter to withstand the maximum rudder torque. The rudder has three main characteristics: namely, any degree of balance can be obtained; the steady torque is independent of the angle of helm at a given speed; an approximate 10 per cent increase in thrust is produced by the absorption of the radial component in the wake. In addition to being efficient the rudder is entirely fabricated and, being of single plate construction, is cheap to manufacture. In order to obtain a satisfactory bollard pull with the limited draught, it was found that a propeller shaft speed of 250 r.p.m. was the maximum, so that geared machinery was necessary. It was decided to install two type R4AM6 National Diesel engines, each rated at 250 h.p. at 600 r.p.m. Each engine is fresh water cooled, having an independent heat exchanger and oil cooler, centrifugal pumps being used for both fresh water and salt water circuits. The owners make a practice of installing a large oil cooler in the fresh water circuit in order to avoid possible contamination of the engine lubricating oil with salt water. Part of the flywheel weight is carried on an outboard Cooper roller bearing from which the drive is taken to a traction type hydraulic coupling. It was felt that the expense and complication of scoop tube-type couplings conventionally used were not justified in this instance. An interesting feature of the installation is that the after end of the coupling is in the form of a centrifugal fan, and is enclosed in a casing with trunking leading outside the engine room. These fans not only cool the couplings but also provide ventilation for the engine room. The output from the two engines is led into a 2·4:1 ratio reverse/reduction gearbox supplied by Modern Wheel Drive, Ltd. It is an incidental advantage that either engine can be isolated from the propeller shaft by a changeover cock on the gearbox which is separate from the main gearbox control. The control of the two engines and the gearbox is normally from the wheelhouse by means of a new type of mechanism. Due to the load from the spring type governors on the engines direct manual control from the bridge could not be adopted. The arrangement installed comprises an apparently orthodox Chadburns telegraph system from the bridge to the engine room to which has been added a Telektron air operated follow-up mechanism.—*The Shipping World, 17th July 1957; Vol. 137, pp. 60-61.*

Magnetic Flux Gas-shielded Arc Welding

During the last twenty-five years, many advances have been made in the art of welding mild steel. New processes, such as submerged arc and inert gas welding, have been developed and widely exploited by industry. The introduction of these processes has led to greater productivity and manufacturing efficiency and lower manufacturing costs. Although the submerged arc and inert gas processes have been successful and have been of great assistance, industry has searched for a manual welding process capable of depositing a continuously fed electrode at a rapid rate and yet be as versatile as covered electrodes. During the last five years, the Linde research and development laboratories have been engaged in an intensive programme to develop a versatile manual welding process. This work culminated with the development of a

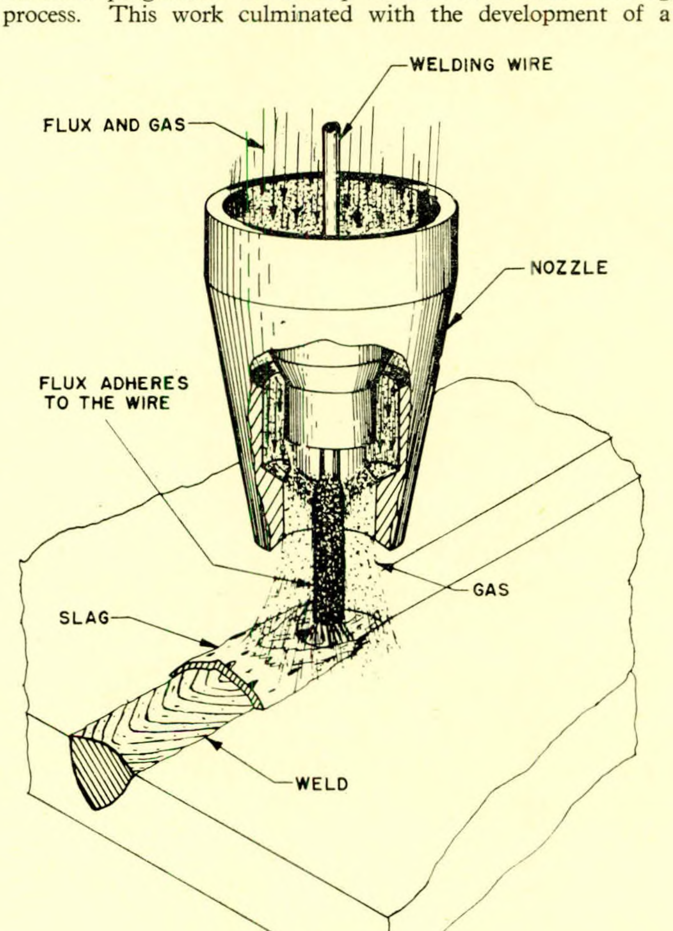
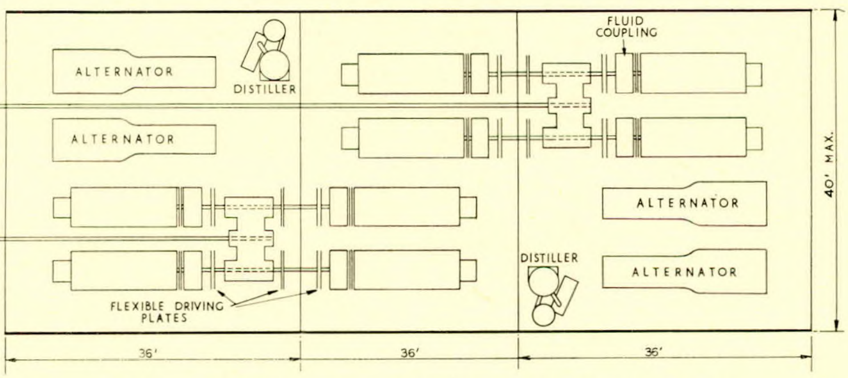


FIG. 1—*The flux is attracted to the wire by the magnetic field surrounding the wire when welding current flows*

magnetic flux gas-shielded arc welding process that produces high quality welds in mild steel in all positions at greater welding speeds and lower welding cost than covered electrodes. This paper briefly describes the outstanding features of this new arc welding process. The essential components of magnetic flux gas-shielded arc welding are: bare-steel welding wire, magnetic flux, carbon dioxide gas and welding current. During welding, the wire and a suspension of flux in gas are fed simultaneously to the torch. Here, the welding current flowing through the wire establishes a magnetic field that attracts the magnetizable flux to the wire (Fig. 1). As a result, the electrode or wire is "flux-coated". The flux stabilizes the arc, refines and protects the puddle and controls the weld contour and coalescence. Besides conveying the flux to the torch, the carbon dioxide provides supplementary shielding and in combination with the flux provides desirable arc characteristics. Magnetic flux gas-shielded arc welding exhibits the following advantages: 1. Welding speeds are approximately 50-100 per cent greater than those obtained with covered electrodes. 2. Welding costs are approximately 35-75 per cent less than those of covered-electrode welding. 3. Weld quality and appearance are excellent. 4. Penetration is excellent, uniform and readily controlled. 5. There is very little tendency toward undercutting. 6. Sizeable gaps can be bridged; and weld soundness is virtually unaffected by base metal composition and surface condition.—*N. Davis and R. T. Telford, The Welding Journal, May 1957; Vol. 36, pp. 475-480.*

Multiple Diesel Frigate

A multiple Diesel machinery installation, with four or eight engines geared to one or two shafts, is one of the types of main machinery considered by shipowners from time to time. The Royal Navy has already adopted this type of machinery installation for two classes of frigate. The accompanying sketch shows the layout of the propelling machinery spaces in H.M.S. *Salisbury,* first of the new Diesel engined frigates. There are eight main and four auxiliary engines, arranged four to an engine room in such a way that any two adjacent compartments can be flooded without total loss of either propulsive or auxiliary power. The main engines are 16-cylinder Vee A.S.R.I. engines, and the alternators are driven by six-cylinder in-line versions of the same engine. All machinery is controlled from a separate control compartment. —*The Shipping World, 26th June 1957; Vol. 136, p. 595, p. 616.*



New Russian Steam Engine Design

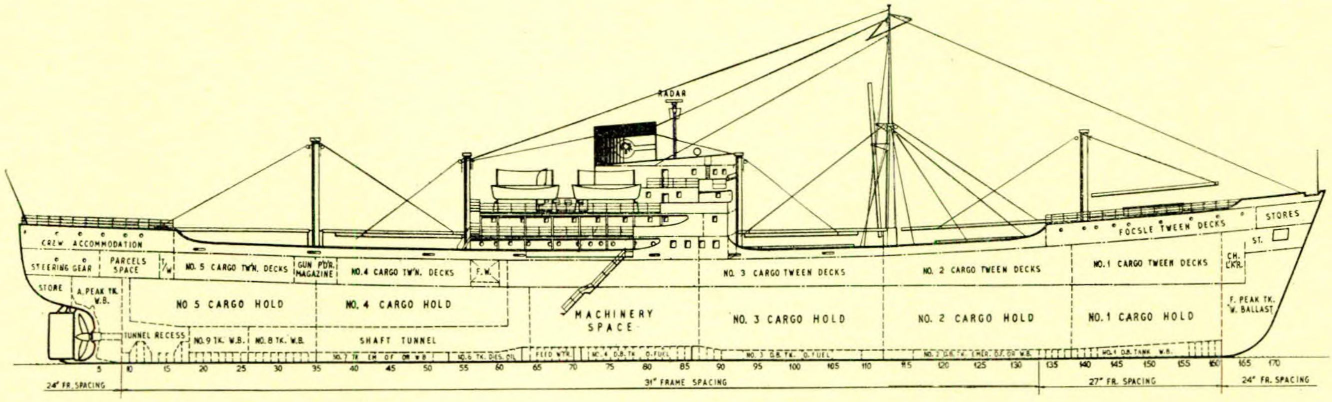
While Soviet marine engineers continue to work on Diesel engine designs for the merchant fleet, counterpart engineers working for the river fleet have problems with steam engine design. River vessels propelled by steam engines vary in their installations from 200 to 1,200 i.h.p. with models of many different designs. Many of these older installations have become uneconomical to operate and many others are simply obsolete. A new small marine steam engine to be used for main propulsion was turned over to the "Teplokhod" Works for construction and testing. After a series of tests were concluded last year the engine went into serial production in the Teplokhod Works. The new engine is reported to differ from previously built engines in that it is more economical, has a wide range of power rating and revolutions, and is of modern construction. The MP-10 is designed to operate normally at a boiler pressure of 16-18 kg. per sq. cm. and temperatures between 360 deg. and 380 deg. C. can also be used. The indicated horse power rating of the MP-10 can be varied from 200 to 450 and, at the higher pressure and temperatures, it is reported as indicating 600 horse power. Engine revolutions, depending on the steam supply, vary between 200 and 400 per minute. The new engine can be used for main propulsion in vessels rated variously from 200 to 1,200 h.p., the latter size based on a twin engine installation. It may be installed directly coupled to the shaft, or a reduction gear may be used if required. In addition, its use is not restricted to transport and towing ships. The MP-10 may also be installed in vessels of the so-called "technical" fleet (dredges, etc.) or it may be used as an auxiliary steam engine for driving electrical generators on large seagoing ships, according to the designers. In design the engine is of vertical construction, with slide-valve steam distribution, reversible, double expansion, duplex, double acting, closed type with a system of lubricating oil circulation. Condensate is recovered from a surface condenser, from which air is excluded by an ejector installation. The engine can be manufactured for both left- and right-hand rotation. The two blocks of cylinders each contain an h.p. and l.p. cylinder. Engine dimensions are given as follows:

H.P. cylinder, diameter, mm. ...	175
L.P. cylinder, diameter, mm. ...	380
Stroke, mm.	340
H.P. valve, diameter, mm.	130
L.P. valve, diameter, mm.	180

Factory tests were made with a KV-5 boiler providing steam for the 18 kg./cm.^{2}. Results were reported as good, with the trial model developing from 200 to 330 horse power at 200 r.p.m., depending on steam flow. At 400 r.p.m. and maximum steam flow the engine indicated 550 horse power.—*B. M. Kassell, Journal of the American Society of Naval Engineers, May 1957; Vol. 69, pp. 309-318.*

Cargo Liner for West African Service

A single-screw motor cargo liner has recently been completed by Swan, Hunter and Wigham Richardson, Ltd., for the Palm Line, Ltd. This vessel, the *Elmina Palm,* 7,900 tons d.w., built at the Neptune Works, Wallsend-on-Tyne, is the third of the present series of ships building for the Palm Line, Ltd., but differs from her predecessors in that considerable use has been made of aluminium in her superstructure. By careful design it has been made possible to replace a weight of 87 tons of steel in the deckhouses and funnel with 32 tons of aluminium; which represents a saving in weight of 63 per cent. Aluminium has been used because the increase in deadweight gained will be of considerable value when the vessel is operating up West African rivers with severe draught restrictions at the mouth. It is also of advantage to have less top



weight as the *Elmina Palm* will be carrying loads of timber, the heaviest of which is frequently placed at the top of the holds. The principal particulars of the *Elmina Palm* are as follows: —

Length o.a.	454ft. 11in.
Length b.p.	425ft.
Breadth moulded	60ft. 3in.
Depth moulded	35ft. 4½in.
Draught	23ft. 4½in.
Deadweight	7,900 tons
Gross tonnage	5,700 tons
Net tonnage	2,850 tons
Machinery output	4,500 b.h.p.
Service speed	13½ knots
Cargo capacity—Grain ...	530,775 cu. ft.
Bale ...	495,450 cu. ft.

The *Elmina Palm* has been designed as an open shelterdeck vessel with freeboard and with a tonnage opening aft, and with scantlings increased on the shell and shelterdeck. She has been built with a raked stem, cruiser stern, poop and long forecastle, and is divided into five cargo holds, three forward of the machinery space and two aft. The holds are free of pillars so that heavy logs, etc., may be carried. The propelling machinery, which has been built at the Neptune Engine Works, consists of a Swan, Hunter-Doxford oil engine having four cylinders 670-mm. diameter and with a combined stroke of 2,320 mm. The engine has been designed to develop 4,500 b.h.p. at about 118 r.p.m., and is arranged to burn boiler fuel; a Sharples installation of three separators being fitted for the clarification and purification of the oil. The engine is of the latest diaphragm type with oil cooled lower pistons.—*The Shipping World, 8th May 1957; Vol. 136, pp. 455-457.*

Safety of Nuclear Merchant Ships

The central question of nuclear ship safety concerns unforeseen accidents, such as ship collisions and sinkings, with the accompanying possible release of accumulated radioactive materials. The mobility of the ship and the medium in which it moves make this safety question different from the problem for land-based reactors. The first consideration requires that a nuclear ship be designed in the anticipation that it may quite likely suffer an accident. With the present trend toward supertankers and giant bulk-cargo carriers of up to 85,000 tons deadweight, it seems conservative to estimate from historical data that there may be about seven accidents per year of minor to major degree involving ships of over 20,000 tons deadweight (the size of proposed nuclear vessels). Perhaps about three or four of these will occur in or near ports, or in contained bodies of water such as the Mediterranean, San Francisco Bay, etc. When nuclear ship programmes are in full stride, say about 1970, they may suffer one or two of these accidents each year near populated areas. Thus, although land-based reactors may be designed with the understanding that major catastrophes (earthquakes or floods) are quite unlikely, ship reactor designs must consider the fact that earthquake-proportion damage is a real possibility. The fact that the failure of a nautical reactor will occur over a body of water further complicates the hazard picture. Depending on the circumstances, the water could either suppress or accelerate the release of fission products to the atmosphere. In addition, the amount of radioactivity present in a ship-size reactor is large enough to make contamination of the water body itself a consideration. A typical merchant ship reactor providing a shaft power output of about 20,000 h.p. will have a heat output of the order of 60 MW. After a long period of operation, the accumulation of fission-product activity within the reactor would reach a saturation value of ~ 6 curies per watt of power or a total activity of $\sim 3{\cdot}6 \times 10^8$ curies. The table below shows the contamination level after ten days in various bodies of water with the addition of this much activity, assuming complete mixing. One sees that the contamination exceeds the maximum permissible concentration of activity for drinking water for many of the water bodies considered. The assumption of complete mixing, however, is unrealistic except for very small water volumes. Actually the fission products will be disseminated from the point of release by diffusion and marine currents at a moderate rate. A rough idea of the distribution of contamination in a sizeable body of water can be obtained by solving the diffusion equation for a point source in an infinite medium. If the dispersed fission products are assumed to behave like an ionic solution, a representative value can be assigned for the diffusion coefficient and the activity concentration can be calculated as a function of distance from the release point for various times. The figure shows the results of such a calculation for the 60-MW reactor previously described. The specific activity is found to be lower than the tolerance level of 10^{-10} curies/litre for distances greater than a mile. Thus, as far as hazard to inhabited areas is concerned, water contamination will be important only when the ship is very close. When the ship is at sea, the most urgent considerations will be exposure of the ship's crew and passengers and salvage operations. When the ship is in port, the shore-line contamination must also be evaluated and will involve contamination of the water as well as the usual atmospheric pollution. Compared with a land-based nuclear plant accident, the shore-line accident therefore presents the additional complications of salvage difficulties and transport of radioactivity by water. Fission products could be set free by an explosive release of nuclear energy. Fortunately, a well designed reactor is generally a very sluggish machine in this respect; the maximum energy release from the nuclear explosion of such a reactor might be of the order of magnitude of a mild chemical explosion. An explosion of this sort might happen if the reactor is allowed to run away. To prevent such a run-away from being too serious, the reactor must be designed with strongly negative temperature and pressure coefficients of reactivity, and must provide fully dependable safety mechanisms for shut-down. Fission products could also be released by mechanical rupture of primary vessels and piping, with subsequent loss of coolant and mechanical, thermal, or chemical expulsion of hazardous materials. From a safety viewpoint, it is evident that nuclear ship propulsion systems using low pressure coolant, even at high temperatures, are especially desirable. But this consideration must in practice be balanced against the improved thermodynamic-cycle efficiencies and reduced operating costs of high pressure systems. Although maintenance, fuel changes, and salvage procedures also have a bearing, safety will probably have an important influence on the location of a reactor aboard ship. Historical data are not yet available on the probability and extent of accidents to engine room spaces. Such data should be collected and reviewed with regard to the prospect of placing reactors in the normal locations aboard ship, or in unconventional positions.

CONTAMINATION OF VARIOUS BODIES OF WATER*

Body of Water	Volume (10^{17} litres)	Contamination (10^{-10} curies/litre)
Caribbean Sea	9·573	0·25
San Francisco Bay	0·000076	22,400
Mediterranean plus Black Sea	4·238	0·55
Baltic Sea	0·023	102
Red Sea	0·215	11
Persian Gulf	0·006	392
Atlantic Ocean	323	0·0073
Pacific Ocean...	707	0·0033
Maximum permissible concentration for drinking water		1·0

* For complete mixing of the saturated fission products from a 60-MW reactor ten days after shutdown.

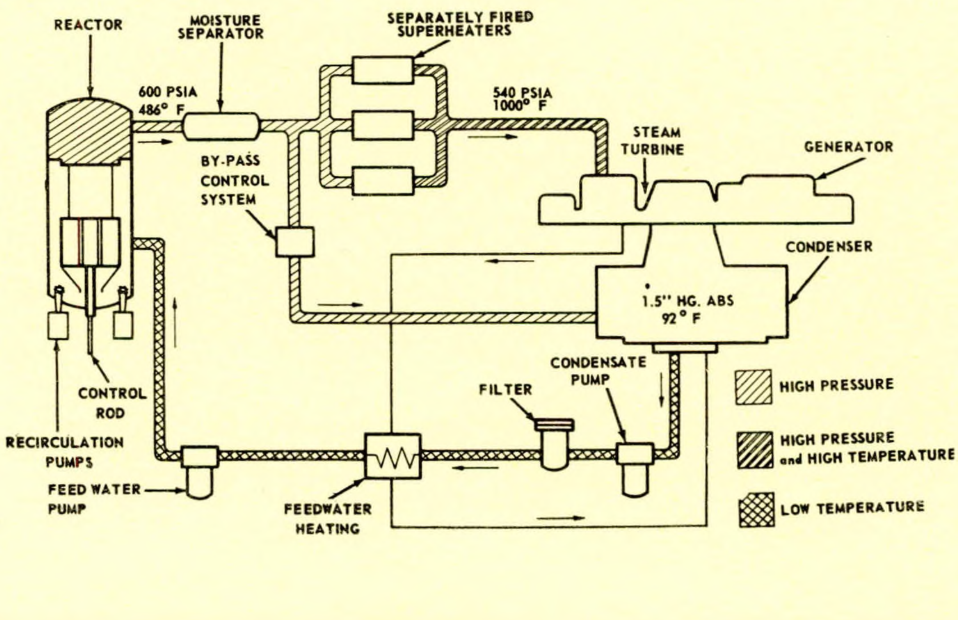
—*R. A. Fayram and H. J. Schneider, Nucleonics, May 1957; Vol. 15, pp. 109-111.*

Factors Affecting Tensile Properties of Steel Weld Metal

The tensile properties of carbon-steel weld metal compare with cast steel in some respects and with wrought steel in others. For a given chemical composition, weld metal is usually stronger and less ductile than cast or wrought steel, and its yield-tensile ratio is significantly higher. At high temperatures, weld metal experiences a ductility trough in the interval 900 to 1,100 deg. C.; this effect is peculiar to a cast steel. At less than 900 deg. C., and to temperatures below room temperature, the tensile properties of weld metal show a trend comparable to that of a wrought steel. There is good evidence to indicate that weld-metal tensile properties are essentially independent of the welding process for welds of similar composition. The effect of alloying elements on the hardenability of weld metal are much the same as in wrought steel. Additions of alloying elements have been correlated with the degree of fissure formation. Of the common elements, Mo seems to be the most desirable and Mn the least desirable for retarding fissure formation; slight amounts of Cr and V are beneficial, while Ni has no apparent effect. Carbon also plays a role in fissure formation. There is also evidence to suggest that alloying elements affect the solubility of nitrogen and distribution of nitrides in weld metal. Nitrogen, when present in amounts greater than 0·02 per cent, embrittles weld metal. The distribution of nitrogen whether present as needles or massive nitrides also affects the strength-ductility characteristics; nitride needles appear to be more significant in decreasing ductility. It is also believed that nitrogen as well as hydrogen contributes to fissure formation in welds, but the influence of nitrogen is less pronounced. Contrary to past belief, there is evolution of nitrogen upon solidification and cooling. Oxygen also detracts from weld metal ductility, but is generally not present in significant quantity unless bare rod welding in an air atmosphere is practised. It has been stated that oxygen causes embrittlement by the solution and subsequent precipitation of compounds. Oxide inclusions are also undesirable. Hydrogen is also known for its embrittling action, which is most evident following rapid cooling of the weld metal. The exact nature of hydrogen embrittlement is not clear, and it is possible that several mechanisms may be at play, simultaneously or separately. The embrittlement caused by hydrogen may not be revealed in the conventional yield and ultimate strengths; its effects are more evident in bending or in measurements of tensile ductility. The low-hydrogen electrode, inert-gas-shielded metallic arc, and submerged arc methods produce deposits that are comparatively low in hydrogen. The effect of residual stresses resulting from solidification shrinkage and expansion and contraction is not effectively revealed in all-weld-metal tensile specimens. This indicates that the process of thermal adjustment does not use up the inherent ductility of the weld metal itself. Residual stresses can cause localized plastic flow at below-yield-point stresses; but after the initial adjustment the material behaves normally. Bend ductility also appears to be independent of residual stresses. Post-weld treatment at 1,200 deg. F. will almost completely eliminate gross residual stresses in weld metal.—*C. M. Wayman and R. D. Stout, The Welding Journal, May 1957; Vol. 36, pp. 252-s-262-s.*

Controlled-recirculation Boiling Reactor

The Allis-Chalmers Manufacturing Company of Milwaukee is to build a 60,000-kW nuclear power plant which will employ an advanced cycle. The 5ft. by 5ft. cylindrical core will produce 133 MW of heat in the form of steam at 600lb. per sq. in. at 488 deg. F. This will be separately superheated with conventional fuel to 1,000 deg. F. at 540lb. per sq. in. This modification of the boiling water reactor has an unusually high energy output per unit of volume because of the large quantity of water continuously being recirculated. Although the core is only 1ft. larger in each direction than that of the Argonne Experimental Boiling Water Reactor (EBWR) at the National Laboratory, and the reactor vessel



2ft. greater in ID, and 2ft. less in height, the electric power produced is twelve times as great.—*The Marine Engineer and Naval Architect, June 1957; Vol. 80, p. 206.*

Stress and Deflexion of Expansion Bellows

In the design of pipelines for high temperature service, allowance has to be made for the thermal expansion of the heated pipe. One particular aspect of this problem is the use of bellows expansion joints which, particularly on board ship, can transmit the pipe movements through bulkheads whilst providing a watertight flexible anchorage. An approximate theory has been developed for the compression of a bellows, the convolutions of which have a cross section formed by circular arcs subtending any semi-angle α. Numerical results have been calculated for the two cases $\alpha = \pi/_2$ and $\alpha = 3\pi/_4$. Experiments have been carried out on six bellows, four corrugated-pipe type $\alpha = \pi/_2$, one S-type $\alpha = 3\pi/_4$, and one flat-plate type which was not analysed theoretically. Deflexion and resistance strain gauge readings were taken on each bellows, and reasonable agreement was found between the theoretical and experimental results for the five bellows in which these could be compared. It has been found that for certain design conditions optimum relationships exist between bore, wall thickness, and radius of convolution for the maximum flexibility. Stress intensification and flexibility factors have been calculated to cover a range of bellows proportions.—*Paper by C. E. Turner and H. Ford, submitted to The Institution of Mechanical Engineers for written discussion, 1957.*

Stresses in Pipe Bends

Much theoretical and experimental work has been published on the flexure of pipe bends. Examination of the results to date show considerable differences between certain measured results and the predicted behaviour. The theoretical works relate mainly to long radius bends and all make many simplifying assumptions. This paper presents a more detailed theoretical study, for bending in the plane of the pipe bend. Examination of both typical and extreme cases and a comparison with the experimental evidence shows which of the various theoretical simplifications can be accepted for a given pipe bend. It is concluded that although complete stress distribution calculated by previous theories may be seriously in error over particular ranges of the variables, the peak stresses and flexibilities are, by a combination of circumstances, unlikely to be more than 5-10 per cent in error if certain factors introduced in the more recent of these theories are used. Manufacturing and other variations between nominally similar pipe bends, however, seem to preclude the possibility of predicting the behaviour of a particular bend to closer than ± 20 per cent. —*Paper by C. E. Turner and H. Ford, submitted to The Institution of Mechanical Engineers for written discussion, 1957.*

Cavitation Inception

It is now generally recognized that the onset of cavitation requires the presence of nuclei in the water. The nature and size of the nuclei determines the critical pressure for cavitation. Accordingly, certain scale effects in cavitation inception are associated with the nuclei. Two fundamentally distinct forms of cavitation exist: gaseous cavitation involving the slow formation of permanent air bubbles by diffusion of dissolved air out of water and vaporous cavitation involving the sudden and intermittent growth and collapse of cavities containing primarily water vapour. Vaporous cavitation is the more important type. If the nuclei are simply free air bubbles, the critical pressure for either gaseous or vaporous cavitation can be calculated in straightforward fashion. For vaporous cavitation, the critical pressure is always below the vapour pressure of the water, and is negative for bubbles with radii smaller than about 10^{-3} cm. Accordingly, it would seem desirable that cavitation inception measurements be performed with water containing bubbles whose sizes range from 10^{-3} to 10^{-2} cm., in order to avoid negative inception pressures and associated scale effects. If the water is undersaturated with air, or if the test is conducted in a water runnel equipped with a resorber, free bubbles will dissolve and so cannot persist in the water. The nuclei in these cases are believed to be some form of stabilized air cavities. Two forms of stabilized cavities have been suggested: bubbles surrounded by mono-molecular skins of organic impurities, or air trapped in microscopic cracks on suspended solid particles. For either form of stabilized cavity, theory indicates that the inception pressure depends on both the dissolved-air content of the water and the maximum pressure to which the water is subjected. Accordingly, the maximum pressure in the resorber may be a significant parameter in cavitation tests. Some cavitation inception measurements performed with ultrasonic apparatus seem to verify the theoretical predictions for the behaviour of stabilized air nuclei. However, the measurements were made with undisturbed water, and it is not known if the nuclei in flowing water behave the same way. In fact, the behaviour of nuclei is only partially understood, so that only tentative conclusions may be drawn at this time.—*M. Strasberg, David Taylor Model Basin, Research and Development Report 1078, May 1957.*

Marine Machinery Breakdowns

The object of this paper, which is of a descriptive and practical nature, is to endeavour to convey some idea of the problems which can confront surveyors and superintendents when serious defects, necessitating the delay of a vessel, have developed at sea or have been brought to light at a survey. The most difficult cases are nearly always those on ships which, for one reason or another, must, if possible, remain in service for a further limited period, before any major repair is effected. In such cases, especially in small foreign ports with limited machining and repair facilities, the task of effecting efficient temporary repairs is made much more difficult. Serious delays through troubles of this nature, requiring spare parts for rectification, have in recent years been greatly reduced by the advent of air transport. It is not uncommon nowadays for a spare tail shaft to be flown out to a vessel in the Far East in a matter of a few days.—*Paper by J. H. Milton, read at a meeting of the North East Coast Institution of Engineers and Shipbuilders on 12th April 1957.*

Spheroidal Graphite Cast Iron for Marine Applications

The writers make an introductory reference to the increasing ascendancy in recent years of the large Diesel engine (mainly manufactured from cast iron and mild steel) over the geared steam turbine as a means of propulsion in the medium-power range. The continued improvement of the Diesel engine in recent years has required more exacting specifications for the materials used in its construction. This has led in this instance to the adoption of nodular (spheroidal graphite) cast iron for Diesel engine pistons. The optimum properties of piston materials are given, and on this basis high duty cast iron, spheroidal graphite cast iron and 13 per cent chromium steel are compared. Failures in cast iron pistons are caused by a process of creep and residual stress. A full account is given of experiences with, and improvements effected in, the use of spheroidal graphite irons for Diesel pistons. An analysis of service performance and the causes of such failures as have occurred are given. The effect of piston operating temperatures is studied and details are given of future developments in spheroidal graphite iron pistons. Further applications of this material in marine engineering are discussed: these include Diesel engine liners, ship side valves, and Diesel engine bedplates.—*A. G. Arnold and B. Todd, B.C.I.R.A. Journal, April 1957; Vol. 6, No. 11, pp. 588-599.*

Nuclear Powered Merchant Ship

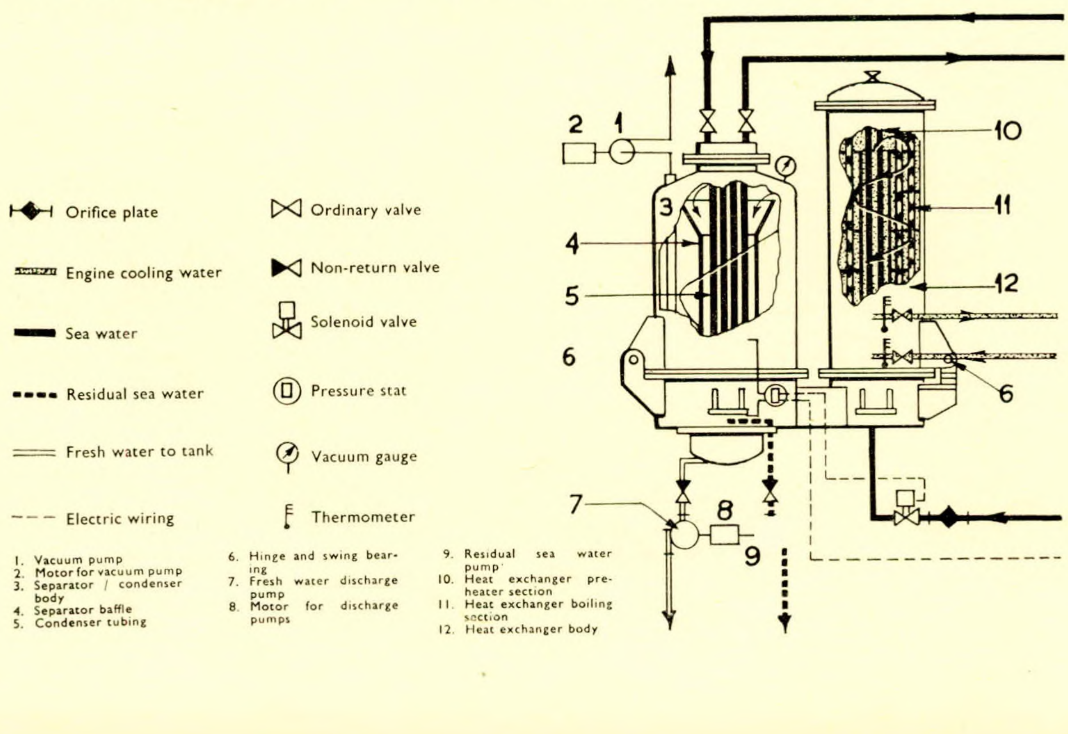
It is expected that nuclear ships will be able to achieve higher sustained speeds over longer runs than conventionally powered ships, factors which will add to gross income. Since refuelling will not be required, moreover, nuclear ships will require less time for turn-around in port, further adding to their earnings efficiency. Factors such as these are believed to have a total effect more than offsetting what might at first be relatively higher fuel costs for nuclear-propelled ships as compared with ships using conventional fuels. It is not considered, moreover, that higher fuel costs need be the rule for long. Much progress is being made in the development of reactors for land-based central station power plants. A large part of this technology is transferable to maritime applications. Also, it is to be expected that in the next several years a number of new concepts of specific application to maritime propulsion will have been proven out. Additional cost cutting factors which may be expected to take effect in the years just ahead include the elimination of security costs as reactors are built more and more on an unclassified basis, and the saving which may result from bringing to bear on reactor work the cost cutting incentives of private industry. Also marine plants can probably be built in existing shipyards already well equipped with facilities and personnel. Another question which might arise is why there is need for a separate maritime nuclear propulsion programme in view of the fact that so much work has already been done on nuclear propulsion for naval vessels. There is no doubt that much of the information accumulated in the naval reactors programme will be exceedingly useful. It is also true, however, that the needs and major emphases of a merchant ship programme differ considerably from those which govern a naval programme. For example, the adaptation of nuclear energy to naval vessels is a matter of national defence and possibly national survival itself. In the areas of merchant shipping the acceptance of nuclear power will be determined by economics. Thus, it is not necessary for commercial ships to have the manœuvrability or ability to meet military emergencies which are required in naval vessels. On the other hand, the maritime programme will stress cost reduction to a far greater degree than would be prudent in military programmes—*R. P. Godwin, Proceedings of the Merchant Marine Council, U.S. Coast Guard, 1957; Vol. 14, No. 3, pp. 35-37.*

Fresh Water Generator

A new type of fresh water generator which distils sea water under vacuum is being manufactured. This generator operates on the principle of using the cooling water from the main engine as a heating medium. Fresh water from the main engine cooling system enters the heat exchanger, usually at a temperature of about 150 deg. F., and is distributed over the tubes by a system of baffle plates. The heat exchanger consists of a preheating section and a boiling section. The sea water, usually drawn from the fresh water cooler of the main engine, enters the heat exchanger at 86 to 89 deg. F., and is first passed through the preheater section in which its temperature is raised to about 114 to 123 deg. F. From the distributor bowl the preheated salt water enters the boiling section where vaporization takes place. The resultant mixture of sea water and vapour flows from the bottom of the heat exchanger into the special combined separator/condenser unit in which the residual salt water remains at the bottom for continuous removal. The vapour is forced upwards to the top of the separator and then downwards into the condenser, from the bottom of which fresh water is continuously led out to a storage tank. A high vacuum is maintained throughout the system ensuring a boiling temperature not higher than 104 to 113 deg. F. The sea water passes through the system at high velocity and in such quantity that only a comparatively small part is evaporated, and thus the salt concentration is only slightly increased. As a result the risk of precipitation and salt formation is practically eliminated. Protection against overflow of the separator is provided by means of a simple pressurestat and magnetic valve cutting off the sea water supply in the event of a possible pump failure or interruption of the power supply. The largest standard Nirex unit has a fresh water production of 30 tons per 24 hours, but larger units can be supplied to order.—*The Shipping World, 22nd May 1957; Vol. 136, p. 509.*

Flash Evaporator for Distillation of Sea Water

Flash evaporators of the type discussed here have several advantages. Low temperature waste-heat sources can be utilized. Sustained high efficiency of operation can be obtained since evaporation takes place on the water droplets rather than on tubular heat-transfer surfaces, which corrode and scale. Multiple-effect arrangements can be used with low temperature differences. The equipment is greatly reduced in weight and space requirements compared to conventional shell-and-tube evaporators. Initial cost is lower and maintenance time and expense are reduced to a bare minimum. The maximum amount of vapour evaporated in flashing saturated water from a high pressure to a lower pressure is given by the thermodynamic equilibrium states of the water before and after flashing. This information can be obtained from steam tables or charts (or tables of thermodynamic properties of salt-water solutions). The information in these tables is based on complete thermodynamic equilibrium conditions existing between the vapour and liquid phases in mutual contact. Since in actual processes such equilibrium conditions are not present, two methods are used to approach equilibrium—promoting turbulence, and increasing the effective surface area for maximum evaporation. In this connexion a common belief is that fine atomization



of a stream of water is most effective in presenting a large surface for evaporation. Sometimes overlooked is the fact that increasingly finer atomization approaches a diminishing rate of returns rapidly with an accompanying adverse effect on carry-over. The mechanism of flashing saturated or metastable water is similar to an "explosion" of a sheet of water. This is primarily due to the rapid increase in the volume of dissolved gases and preformed nuclei within the bulk of the water, caused by the decrease in pressure at the instant of flashing. The major portion of the vapour release takes place almost simultaneously with the disintegration of the liquid mass. Therefore, beyond a reasonable degree of break-up, any further atomization rapidly reaches a state of diminishing returns, because of the small temperature differential present between the fine liquid particles and the vapour surrounding them. In other words, droplet formation and vapour release essentially take place simultaneously and fine droplets are relatively low in average temperature at the very outset, thus contributing only a small fraction of the total vapour released by surface evaporation. Certain criteria must be established in the design of flash-evaporation equipment for sea water distillation. The temperature of the feed should be as low as possible to enable the utilization of low-grade energy sources available, and to minimize serious corrosion problems encountered at high temperatures. The temperature difference available for flashing should be as high as possible, from the vapour release standpoint, because the percentage of flash vapour is proportional to the temperature drop (being almost 1 per cent for each 10 deg. F. drop in temperature within the range of 100 deg. F. to 200 deg. F.). High fractions of vapour release per pound of water circulated result in smaller pumping power requirements for a given water generation capacity. Within these features conflicting interests are evident. The lower the temperature of the feed and also the greater the temperature drop in the flashing process, the greater the specific volume of the vapour becomes (nearly inversely proportional to the pressure level in the evaporator). This necessitates large piping and large evaporator volumes to keep the vapour velocities within a reasonable range and thus avoid large friction losses and excessive carryover. One problem studied was the determination of the effectiveness of flashing of sea water. A comparison between the measured evaporator sump temperatures and the saturation temperatures for pure water showed the difference between the two temperatures to be nearly 1·5 deg. F. For a shell concentration of 37,000 p.p.m. the true equilibrium temperature is very nearly 1 deg. F. higher than that for pure water in this range of temperatures. Hence, the unflashed portion of the water comes within 0·5 deg. F. of the equilibrium temperature, thus giving flashing efficiencies of 90 per cent and 97 per cent, corresponding to temperature drops due to flashing of 5 and 30 degrees respectively.—*R. L. Coit and Y. S. Touloukian, Westinghouse Engineer, 1957; Vol. 7, No. 2, pp. 58-60.*

Liquid Nitrogen Shrinking

During the fabrication of a capstan for a large tanker at the Dunston-on-Tyne factory of Emerson Walker, Ltd., it was found necessary to make the main shaft in two parts. The screwed extension, which has a square thread and which is used for raising or lowering the shaft to engage and disengage the clutch, had to be turned separately. It was decided to expand the shaft so that the two parts could be joined, but due to the varying thicknesses of the metal the expansion would have been uneven. After consultation with British Oxygen Gases, Ltd., the firm decided to employ liquid nitrogen to shrink the screwed extension so that it would fit into the main shaft. The 11¾-in. diameter shaft was 15ft. in length and weighed 2½ tons. The 8-in. diameter extension was turned to give 0·003-in. interference. After immersion in liquid oxygen for a period of 45 minutes a micrometer reading showed that the part had shrunk to 7·995 in., which allowed 0·005-in. clearance. The extension was then lifted by overhead crane and dropped into a 3-in. deep recess in the end of the shaft. Air vents had been cut in the extension, which dropped easily into position. It was later drilled and pinned in position. The firm has since used the same process to shrink another capstan shaft and a total of 90lb. of liquid nitrogen was used for the two operations.—*The Marine Engineer and Naval Architect, June 1957; Vol. 80, p. 221.*

German Built Passenger Liner for Israel

The twin-screw passenger and cargo liner *Theodor Herzl*, built by Deutsche Werft, A.G., of Hamburg, for the Zim Israel Navigation Co., Ltd., of Haifa, Israel, is intended for her owners' Mediterranean service between Israel and Southern Europe. The principal dimensions and other leading particulars are given in the accompanying table. Accommodation is provided for 550 passengers, and the special facilities include a synagogue, barber's shop, cinema, swimming pool and laundry. Part of the accommodation has Thermotank air conditioning, and there is a comprehensive Thermotank mechanical ventilating system. There are two cargo holds with a total capacity of 50,000 cu. ft., and, for serving these spaces, there are four 5-ton derricks. The hatchways are fitted with hydraulically-operated hatch covers, and to assist in the rapid handling of cargo and baggage there are three doors in the ship's sides. Principal dimensions and other leading particulars of the *Theodor Herzl* are:

Length overall	487ft. 10in.
Length b.p.	435ft. 0in.
Breadth	64ft. 6in.
Depth to second deck	27ft. 4¾in.
Draught	21ft. 5in.
Corresponding deadweight, tons ...	2,800
Gross tonnage	10,000
Number of decks	7
Passengers	550
Crew	180
Total complement	730
Cargo capacity, cu. ft.	50,000
S.H.P.	12,000
Trial speed, knots	19·5
Service speed, knots	18

Special arrangements are made for carrying fruit and vegetables in ventilated and chilled compartments. Steering is effected by means of a Hastie electro-hydraulic four-ram steering gear, and there is an installation of Deutsche Werft stabilizers. These stabilizers are extended and retracted by means of an electrically-operated screw jack and are actuated by gyro-controlled hydraulic rams; it is understood that these stabilizers are the first of their type to be fitted in any vessel. An inspection of the wheelhouse shows that the owners have taken every precaution to ensure that the *Theodor Herzl* will be safely and efficiently navigated in all climatic conditions. Prominent among the comprehensive range of equipment are a Sperry gyro compass and auto pilot, Decca 45 radar installation and Marconi radio equipment. The vessel is propelled by twin screws, the power for which is provided by two sets of A.E.G. steam turbines, each developing 6,000 s.h.p. The turbines are connected to the propeller shafts through double reduction gearing. Steam is generated in two Babcock and Wilcox boilers, each of which has a heating surface of 7,456 sq. ft. and a superheater surface of 1,086 sq. ft. The boilers are equipped with Iowa burners and are designed for a pressure of 670lb. per sq. in. and a pressure at the superheater outlet of 570lb. per sq. in.; they have a total evaporation of 60,550lb. per hour. Electrical power is provided by three 500-kW turbo-generators, which produce alternating current at a tension of 440 volts for the engine room and deck auxiliaries. For the galley equipment, current at a tension of 220 volts is used, while current at 110 volts is employed for lighting.—*The Shipbuilder and Marine Engine-Builder, July 1957; Vol. 64, pp. 443-444.*

Shipboard Combustion Controls

Before the selection of any automatic combustion control system can be made, the quantity being controlled must be given prime consideration. Invariably, a combustion control system is used to control boiler drum pressure, steam header pressure or superheat temperature. The selected system hinges on the quantity to be controlled and the disturbances that can be expected in the quantity to be controlled—the controlled variable. The measuring means, controlling means and feedback selected in the control system vary with the components that different manufacturers have to offer. A wide assortment of components is offered in the control fields, but most of these are designed to accomplish the same functions. The four general classifications of automatic combustion control systems are: (1) Series-fuel. (2) Series-air. (3) Parallel. (4) Steam flow-air flow. In a series-fuel system a master steam pressure controller regulates the firing rate of the ship's boilers by directly adjusting fuel flow to the boiler to maintain steam pressure set point. Fuel flow to the boiler is measured and the measuring device, a fuel flow transmitter, is used to regulate air flow to the boiler in proportion to the fuel flow to maintain combustion efficiency. A typical, simplified, positioning-type, series-fuel automatic combustion control system is shown in Fig. 7. A master steam pressure controller is used to sense fluctuations in steam header pressure. This controller transmits a signal that is proportional to the error in steam header pressure set point. This signal of error in steam header pressure is used to position the oil return control valve and

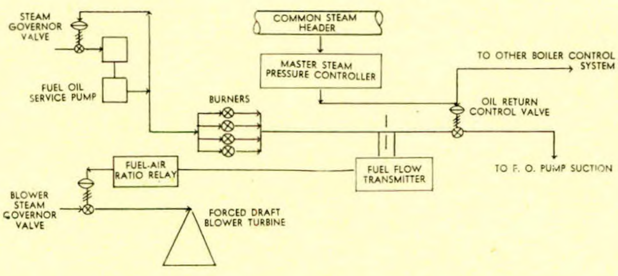


FIG. 7—*Series-fuel system*

vary the flow of oil to the boiler. A constant oil supply header pressure is maintained by a steam governor valve on the fuel oil service pump. The governor valve senses pressure in the oil supply header and regulates the flow of steam to the turbine of the fuel oil service pump to maintain a pump speed that ensures a constant oil supply header pressure. An orifice is installed in the oil return line from the burners. A fuel flow transmitter is connected across this orifice. The fuel flow transmitter sends a signal of oil flow to the steam turbine governor valve of the forced draft blower. This automatic signal regulates fan speed to supply the boiler registers with an air flow that is proportional to the fuel flow to the burners. The function of the fuel-air ratio relay that is installed between the fuel flow transmitter and the blower turbine governor valve is to vary the ratio between oil and air. Since fuel oil can be relatively easily metred, the series-fuel system has found widespread use aboard ships. One of the main advantages of the series-fuel system is that, since air flow is regulated by a signal of oil flow, the possibility of carrying a high quantity of excess air for combustion for long periods of time is eliminated. A series-air control system is one in which combustion air flow is regulated by a master steam pressure controller. Combustion air flow across the burner air registers is metred. The metred signal of air flow is used to control the flow of oil for combustion. Another system of automatic combustion control, which is suitable for use aboard naval ships, is the parallel system. In this system the master steam pressure controller simultaneously regulates the flow of oil and air to the boiler to maintain a combustion rate that will keep the steam pressure at its set point value. One of the major advantages of a parallel control system is that no metering equipment is absolutely required. Therefore, it is actually one of the simplest of all control circuits. In addition, the simplified design eliminates some of the components found in other systems with a resultant decrease in the initial cost of installation.—*G. J. Rascher, Journal of the American Society of Naval Engineers, May 1957; Vol. 69, pp. 331-339.*

Cunard Liner *Sylvania*

The new Cunard passenger liner *Sylvania* is the final ship in the company's new construction programme for its Canadian service. This programme has comprised four ships, the previous three vessels being the *Saxonia, Ivernia* and *Carinthia.* All four have been built at the Clydebank shipyard of John Brown and Co. (Clydebank), Ltd., in common with the larger part of the Cunard fleet. They have followed each other into service at yearly intervals, the *Saxonia* being completed in September 1954, the *Ivernia* in July 1955, the *Carinthia* in June 1956 and the *Sylvania* in June 1957. The *Sylvania* is identical in external appearance, hull dimensions and construction with her three sisters, while her steam turbine machinery is the same as that of the *Carinthia,* with three Foster Wheeler external superheater boilers in place of the four Yarrow boilers of the first two ships. The *Sylvania* has a gross tonnage of 22,000 tons. Her principal particulars are given below.

Length o.a. ...	608ft. 3in.
Length b.p. ...	570ft.
Breadth, moulded ...	80ft.
Depth, moulded ...	46ft. 3in.
Draught, loaded ...	28ft. 1in.
Passengers:	
First class ...	154
Tourist class ...	724
Total ...	878
Officers and crew ...	460
Cargo space, ordinary and insulated ...	300,000 cu. ft.
Speed ...	18 knots

Equipment fitted in the *Sylvania* includes Denny-Brown stabilizers. The radar set has been supplied by the British Thomson-Houston Co., Ltd., and consists of the latest B.T.H. type RMS-2 radar installed in the wheelhouse, with a slave display unit fitted in the chart room. Decca Navigator is fitted, and this, of course, can now be used for navigation in Canadian waters as well as United Kingdom waters. Like her sisters, the *Sylvania* has a twin-screw steam turbine machinery installation. The three Foster Wheeler external superheater boilers supply steam at 550lb. per sq. in. and 850 deg. F. The external superheaters are of "MeLeSco" type, while the bled steam air heaters are of "Weldex" type. Electrical power is provided by four turbogenerators, each of 750-kW capacity. The turbines for these are of the "pass-out" type, and each has its own condenser. Two 75-kW Diesel generators are fitted for emergency purposes, while alternating current for fluorescent lighting and other purposes is supplied by two 75-kVA motor alternator sets.—*The Shipping World, 3rd July 1957; Vol. 137, pp. 9-11.*

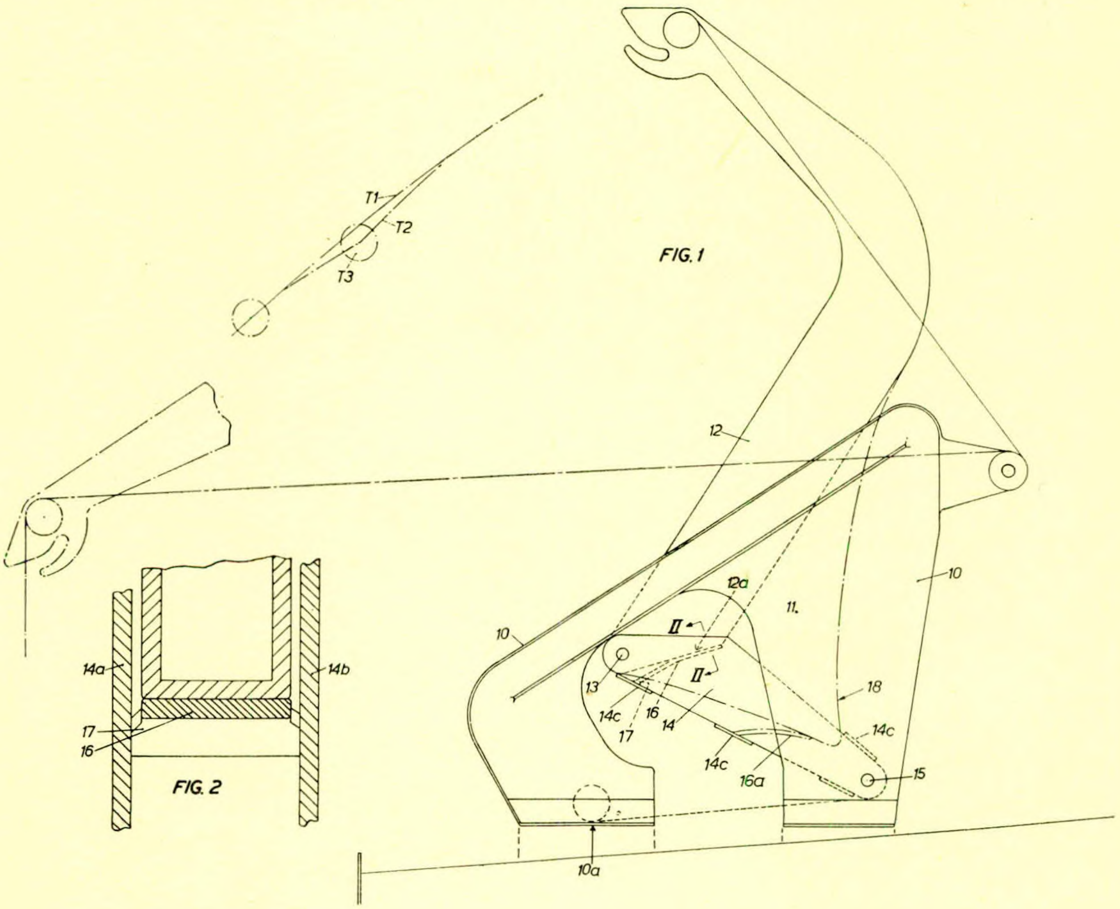
Patent Specifications

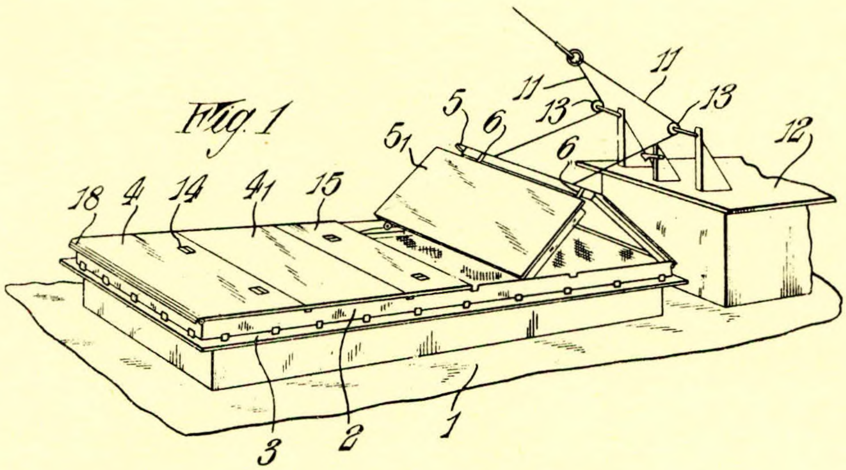
Ship's Davit

This invention is for improvement in ships' davits such as are used for launching a lifeboat over a ship's side. The invention is concerned with davits of the kind comprising a stand, a davit arm and a link pivotally connected to the stand and pivotally supporting the arm. Heretofore in such davits initial movement of the arm in the outboard direction takes place by pivoting the link about its pivotal support on the stand and when the link comes to rest, further outboard movement of the arm takes place by the arm pivoting about its pivotal support on the link. There are thus two separate pivot points about which, in effect, the arm pivots. One object of the invention is to provide a davit with means for assisting outboard movement of the arm. According to the invention, a resilient device is interposed between the davit arm and the link on which the arm is pivoted, the resilient device being stressed during inboard movement of the arm so as to assist in the outboard movement of the arm. Referring to Figs. 1 and 2, the davit arm is pivotally supported at 13, on a link (14) pivotally connected to the stand (11) at 15. The link (14) comprises two side plates (14a and 14b) connected together by bracing plates (14c). A very strong leaf spring (16) is secured at one of its ends as indicated at (17), between the side plates (14a and 14b). The free end of the spring (16) lies under the heel (12a) of the davit arm (12) and supports the arm in its stowed position shown in Fig. 1. During outboard movement of the davit the link (14) is halted when it comes up against the base of the stand as indicated at (10a). Further outboard movement of the arm then takes place by the arm pivoting about its pivot (13) on the link until it reaches the fully outboard position. Halting of the arm at this position may be effected by a stop. The spring (16) gives some impetus to the arm when it moves outboard. This is an advantage with a gravity davit. If the spring is made sufficiently strong it will tend to make the davit head follow the smooth part (T1) and not the path (T2) with the objectionable dip at T3.—*British Patent No. 778,416, issued to Marine and Allied Industries (C and L), Ltd. Complete specification published 10th July 1957.*

Hatchway Cover

The hatchway cover consists of two independent cover parts, each covering its own portion of the hatchway, and each being composed of two sections hinged to each other; one section of each part is also hinged to the front edge side and rear edge side respectively of the hatchway, while the free edge of the other section is guided by means of travelling wheels along the longitudinal side of the hatchway, so that each hatchway cover part can swing from a covering position upward in opposite directions and folded. In Fig. 1, numeral

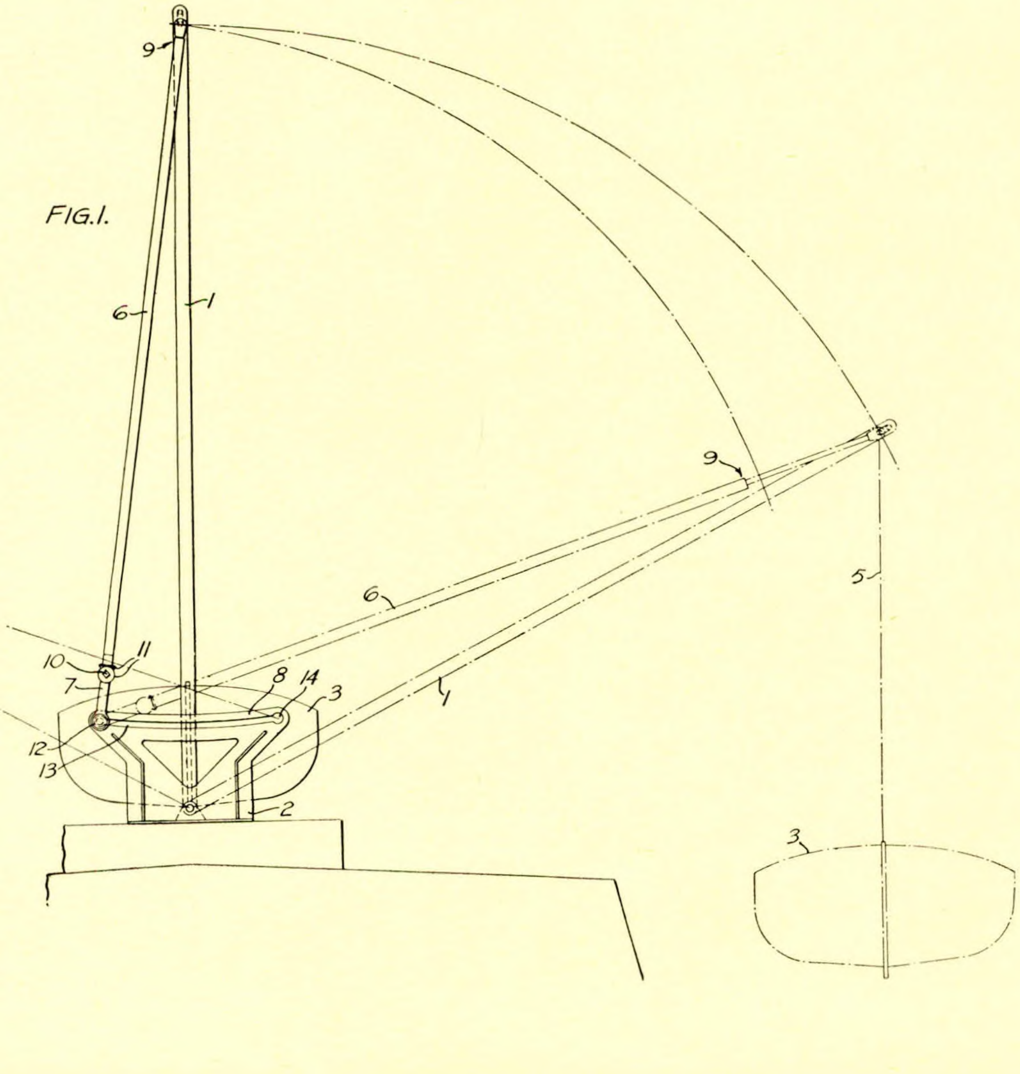




1 denotes the ship's deck, 2 the hatchway coaming, and 3 the flange around the coaming. The hatchway cover consists of four steel plate sections (4, 4_1 and 5, 5_1) which are connected to each other in pairs by hinges (6). Each section is provided underneath with ribs. Sections 4 and 5, which cover the end portions of the hatchway, are hinged to the shorter end parts of the coaming. The hatchway cover is opened by lifting the end sections (4 and 5) around the hinges connecting them to the hatchway coaming. Sections 4_1 and 5_1, hinged to sections 4 and 5 respectively, will thereby follow suit and the free edge of the former section will slide along the tracks (9), mounted longitudinally inside the hatchway coaming.—*British Patent No. 778,866, issued to A. E. Hansen Velle. Complete specification published 10th July 1957.*

Double-acting Luffing Davit

This invention relates to ships' boat davits and deals particularly with double-acting luffing davits adapted to launch a boat or other craft on either the port or starboard side of the ship. According to the invention, a double-acting luffing davit comprises an arm pivotally connected at its lower end to a davit frame and at its upper end to an extendable



member, which is capable of being secured to the davit frame in either of two positions, one on each side of the arm, thus supporting the arm. Each davit comprises an arm (1) pivotally mounted at its lower end in a davit frame (2), a ship's boat (3) being suspended between two such arms (Fig. 1). Each davit arm (1), at its upper end is fitted with a hook or loop from which the boat is suspended by suitable falls (5). Pivotally connected to the top of each arm (1) is an extendable member (6) which at its lower end is forked (as at 7) to straddle a curved guide (8) in the davit frame. The member (6) includes a telescopic screw device (9). Such a device may consist of two threaded rods one within the other, the outer rotating so that the inner moves linearly with respect to the outer. The device (9) is actuable for extension or shortening of its length by a handle or crank (not shown) detachably mounted on a keyed extension (10) carried by one of a pair of bevel gears (11), whereby the length of the member (6) may be extended or shortened. If it is to be desired to launch the boat, e.g. over the port side of the ship, as shown in Fig. 1, the extendable members (6) are moved through the slots (13) to the starboard side of the davit arms, and the forked ends (7) of the members locked in the enlarged portions (14) of the slotted guide (8) by the sleeve (15).—*British Patent No. 777,476, issued to H. A. Spruzen. Complete specification published 26th June 1957.*

High Temperature Pipe Joint

This invention relates to bolted flange joints or similar joints where, as in steam installations, tightness against leakage has to be secured by means of packing between the joint faces. In order to make joints at contact faces of parts under internal fluid pressure tight against leakage, it is customary to use a type of resilient or deformable packing in various forms, made of sheet material such as asbestos sheet, and at higher temperatures and pressures, even sheet metal. However, as operating temperatures become higher, the metal packing must offer adequate resistance to deformation or creep comparable to that of the parts forming the joints, as otherwise excessive creep of the packing material may result in the packing material being squeezed out or alternatively causing relaxation of the bolt loading. Consequently, for high temperature and high loading, packing is required that offers adequate resistance to creep, and at the same time has a surface which can adapt itself to the irregularities of the joint face so as to enable a tight joint to be made. Thus the joint material must possess two contradictory properties, namely, ability to offer high resistance to deformation as a whole, and secondly a limited but sufficient plasticity at high temperature over a surface layer to allow a joint face to deform and accommodate the irregularities of the abutting surface with which the joint has to be made. To this end the invention provides a joint packing member composed of an alloy steel having high temperature creep properties which is suitable for use in the manufacture of the abutting parts of the joint, the member being faced with a thin layer of nickel intimately bonded to it. Conveniently the member is coated with nickel by deposition,

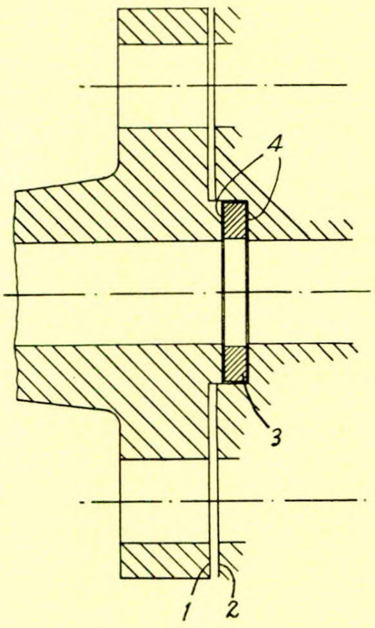


FIG. 1

for example by fusion or electrolytically. The composite member is preferably machined or otherwise surfaced ready for use, the final thickness of the nickel being for example, of the order of 0·02in. In a simple form of joint a packing ring or washer may be used which is made of molybdenum vanadium steel coated by electrodeposition with nickel to provide the contact joint. This electrodeposited metal is built up to the desired thickness and after deposition, the faces of the ring or washer are machined to present the desired contact face. Preferably the surfaces of the joint are plain, especially when the contact loading is high; however, it may sometimes be desirable to employ surfaces of other form; for example, the cross section of the ring or washer may be of lenticular form, the surfaces carrying the deposited nickel at which contact is made being convex. Fig. 1 illustrates a joint between a spigot (1) and a socket (2). In such a case the joint packing member (3) is normally located in the socket (2) and thus fully constrained in the outward radial direction. Such an arrangement, however, is not essential, as in some cases clearance may be left around the outer peripheral surface of the member (3). The member (3) of alloy steel has a thin layer of (4) nickel deposited on its jointing surfaces.—(*British Patent No. 773,159, issued to Metropolitan-Vickers Electrical Co., Ltd. Complete specification published 24th April 1957.*) *Engineering and Boiler House Review, August 1957, Vol. 72, p. 284.*

(Complete British Specifications can be obtained from the Patent Office, 25, Southampton Buildings, London, W.C.2. These extracts from British Patent Specifications are reproduced by permission of the Controller of H.M. Stationery Office. Price 3s. 6d. each both inland and abroad.)

Marine Engineering and Shipbuilding Abstracts

Volume XX, No. 12, December 1957

Magnetic Force Welding

Much time and energy has been expended developing the technique of resistance welding and in perfecting resistance-welding machines. One of the most outstanding accomplishments since the advent of electronic timers and contactors has been the development of a device which allows extremely accurate synchronization of the electrode force application. Due to mechanical limitations, programmed pneumatic systems are subject to variations which result from inertia, friction, air flow and erratic valving. Since the current magnitude and duration can be very accurately controlled, the most logical solution to the problem of controlling the electrode force is that of making it a dependent function of the welding current. Machines deriving their electrode force from an electro-magnet which is interposed in the secondary circuit of the transformer and energized by the welding current have been under development since 1946. The magnetic force welder is fundamentally a conventional direct acting ram-type spot welder which has undergone design changes to accommodate an electro-magnet for application of electrode pressure. Fig. 2 is a skeleton schematic of the electrical and physical arrangement used in present day single-phase equipment. The current flow is controlled by a conventional electronic timer and sequence control, and passes into the primary of the welding transformer which is of a special design, having components sized in such a manner as to meet the requirements of magnetic-force welding. The secondary current is directed to the split-iron-core magnet and makes one turn around the stator, after which it passes through a flexible laminated roll lead to the upper electrode holder. The return path is completed after the current passes through the weldments and lower electrode. A small air cylinder is provided to lift the electrode to the desired opening for work insertion and removal, and to provide clamping pressure. The ram is of the quill type being rigidly supported by linear action ball bushings which allow extremely fast follow-up with a minimum of mechanical friction. The centre portion of the ram is threaded to allow adjustment of the force collar which supports the magnet armature; the effect of this adjustment is discussed elsewhere in this report. The lower end of the ram is securely attached to the upper electrode holder which is usually of an ejector type, designed for number two Morse-taper tips. The lower knee is a high-conductivity copper casting which is provided with a tee-slotted platen for universal die mounting and supported by means of the face plate and an adjustable knee jack.—*E. J. Funk, The Welding Journal, June 1957; Vol. 36, pp. 576-582.*

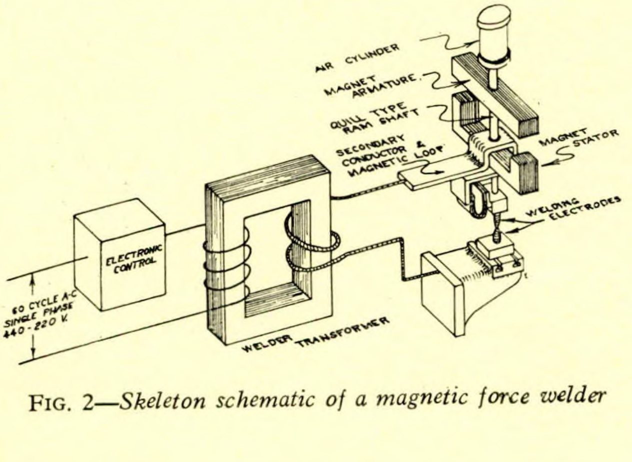


FIG. 2—*Skeleton schematic of a magnetic force welder*

A Packaged Power Reactor

The American army recently put into operation at Fort Belvoir a packaged power reactor, the A.P.P.R., a 2,035-kW pressurized water nuclear electric generating plant, built by A.L.C.O. Products Inc., under contract to the U.S. Atomic Energy Commission. The plant comprises two systems—a primary system consisting of the reactor vessel and core, control rods and drives, two coolant circulating pumps, piping, steam generator, pressurizer, water purification equipment and shielding, and a secondary system comprising the secondary side of the steam generator, the turbo-alternator set, surface

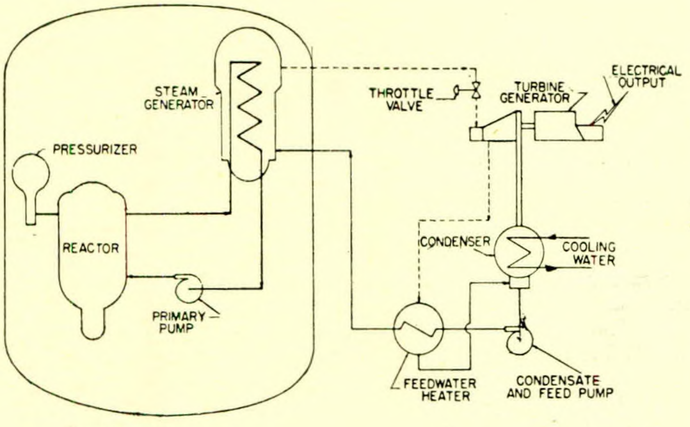


FIG. 3—*Diagrammatic arrangement of the A.P.P.R. pressurized-water nuclear electric generating plant*

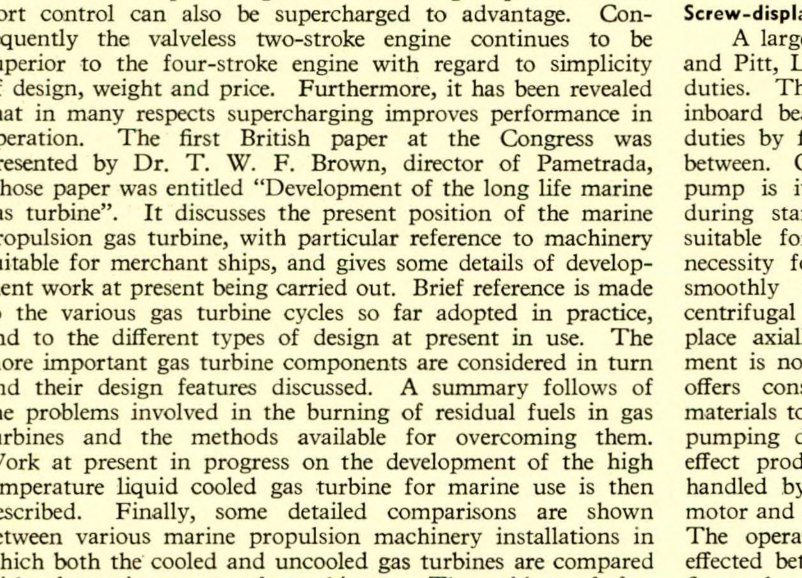
condenser and auxiliaries, feedwater heater, evaporator, pumps and storage tanks. The reactor core houses thirty-eight fuel elements containing fully-enriched uranium, together with seven control rods—all in a grid structure of a 7 by 7 design with the corners missing. Cooling water enters the reactor vessel at 431 deg. F., under a pressure of 1,200lb. per sq. in., to prevent boiling. While circulating upwards between the fuel plates at 4,000 gal./min. it is heated to 450 deg. F., and then passes on to the steam generator, where the heat from the high pressure water is transferred to the water in the secondary system, thus producing the steam to drive the turbine. The cycle is completed by the use of special Westinghouse "canned-rotor" pumps to return the condensate from the condenser to the reactor. The steam generator produces 34,700lb. per hr. at 200lb. per sq. in. and 407 deg. F. A superheater, contained in the same shell as the steam generating section, but separated from it by a baffle, raises the temperature to 432 deg. F. As can be seen from the diagram, Fig. 3, the reactor, pressurizer, steam generator and the primary pump are all contained in a vapour container. This is a steel vessel lined with 2ft. of concrete, and surrounded by an additional 3ft. of concrete from ground level to over half its height. The turbo-alternator used in conjunction with this unit is a 2,500-kW General Electric set, driving the generator through a reduction gear at a speed of 1,200 r.p.m. It delivers 60 c/s current at 4,160 V. The condenser is a Lummus two-pass divided water box, single tube-sheet surface unit. The feedwater heater, located between the condensate and feed pump and the steam generator, receives steam from a bleed point in the turbine. A Hagen three-element regulating valve is provided in the feedwater connexion to the steam generator.—*Engineering and Boiler House Review, August 1957; Vol. 72, p. 282.*

Fourth International Congress on Combustion Engines

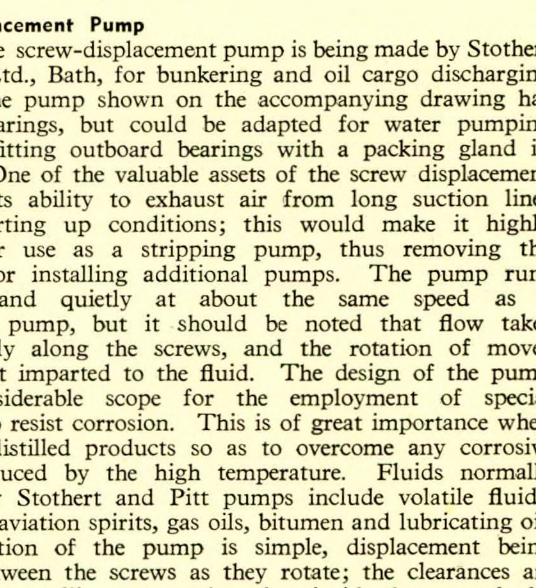
Dr. Alfred J. Buchi presented a paper entitled "Four cycle internal combustion engines with Buchi telescope valve system". It deals comprehensively with a new design of four-stroke internal combustion engine, specially built for pressure charging, preferably combined with scavenging. The latter is intended not only to sweep out the exhaust gases from, and cool, the combustion space, but also to promote swirl generation in this space to improve the fuel distribution and combustion. The engine has a combustion space which is provided with a cross section smaller than that of the cylinder bore. In the front end of the combustion chamber two co-axially arranged valves are provided, one arranged within the other, and one of them serving for admitting the charge and the other for discharging the exhaust gases. This system, the so-called "Buchi telescope or concentric valve system", by means of its concentric valves situated in the cylinder heads in conjunction with the form of the upper part of the piston closing the combustion chamber, relatively reduced in diameter, in its outer dead centre position gives the combustion chamber the shape of a relatively deep disc of a smaller diameter than the cylinder bore while the valves are closed. The insides of the cylinder head and of the piston top surrounding the combustion space are absolutely symmetrical to the cylinder axis and have a minimum of relatively plain and regular surfaces. For a sufficient and possible opening of both kinds of valves even during the scavenging period, it is not necessary to arrange cuts or the like in the piston head or in the cylinder liner for these valves, especially during their inlet and scavenging period when the piston is near or in its top dead centre position. Such cuts, etc., are however necessary in most other conventional internal combustion engines of the open combustion space type, or pre-combustion and other engines, at least when scavenging should and can be adopted. The fuel injector or injectors are positioned in relation to the axis of the combustion space in such a manner that the injected fuel arrives uniformly distributed at points where the required amount of combustion air is present in the combustion space. The air charge is preferably taken in by an inlet duct and a spiral-type continuation of this duct to the inlet valve seat. Due to the spiral-type form of this duct, the air charge is conveyed into the combustion space with a large tangential velocity component in the direction of the circumference of the combustion space and the cylinder liner. A vigorous swirl will therefore be created in the said space. The swirl movement tends to improve the fuel distribution, the scavenging action and the cooling of the walls of the combustion space, the cylinder liner and of the valves. Apertures in the inlet valve body and the outlet duct provide air charge and exhaust gas passages. Mr. G. Camner, chief of Diesel research at Nydqvist and Holm, presented a paper entitled "Turbocharging a two-stroke loop-scavenged Diesel engine, using a pulse system without scavenging pump". The results obtained show that the exhaust-gas blower can be employed for scavenging and super-charging two-stroke Diesel engines embodying the loop scavenging system. Three different forms of application may be envisaged: (1) Engine with normal scavenge pump with direct drive and with turbocharger coupled in series; (2) No scavenge pump, the turbocharger alone producing the scavenging and supercharging air; (3) A form of alternative (2) in which the air is supplied to the turboblower at a comparatively low pressure by a scavenging blower with separate drive. Comparison of the test results obtained with alternatives (1) and (2) shows that an engine with the equipment envisaged under alternative (1) would be able to develop a higher output and would have a lower exhaust gas temperature on account of the greater quantity of scavenging and supercharging air. Alternative (2) offers the advantages of lower fuel consumption, greater reliability in service and simpler maintenance owing to the smaller number of mechanical parts, while the noise level is also reduced by the elimination of the scavenge pump and its transmission. In addition, manufacturing costs per horse power would be lower than those for alternative (1). Alternative (3) involves a certain degree of complication by comparison with alternative (2) as a result of the introduction of the supplementary scavenging blower with continuous separate drive. This variant, however, has a number of advantages which should not be forgotten, namely: A considerable increase in output in return for the relatively low input of the auxiliary blower. The possibility of running the engine on the auxiliary blower alone in the event of damage to the turbocharging system. Loop-scavenged two-stroke engines were also the subject of a paper by Dr. Ing. M. Leiker of Klockner-Humboldt-Deutz. His paper was entitled "On the exhaust gas turbocharging of loop-scavenged engines and their use in marine and rail propulsion". It deals with the exhaust gas turbocharging of loop scavenged valveless two-stroke Diesel engines, a process which for a long period had appeared to be impracticable. The Deutz Diesel engines described have already rendered good service in practical operation without supercharging. They were adapted to the needs of supercharging, and the degree of supercharging was kept at a level which would guarantee complete reliability of operation. A brief explanation is given of several characteristics and nomenclature to facilitate better understanding of the contents. The appendix contains an explanation of certain nomenclature together with the simple formulæ used for calculating the characteristics for the benefit of those less familiar with the subject. The operating results obtained for engines when used as prime movers for ships and locomotives show that the loop scavenged two-stroke engine provided with port control can also be supercharged to advantage. Consequently the valveless two-stroke engine continues to be superior to the four-stroke engine with regard to simplicity of design, weight and price. Furthermore, it has been revealed that in many respects supercharging improves performance in operation. The first British paper at the Congress was presented by Dr. T. W. F. Brown, director of Pametrada, whose paper was entitled "Development of the long life marine gas turbine". It discusses the present position of the marine propulsion gas turbine, with particular reference to machinery suitable for merchant ships, and gives some details of development work at present being carried out. Brief reference is made to the various gas turbine cycles so far adopted in practice, and to the different types of design at present in use. The more important gas turbine components are considered in turn and their design features discussed. A summary follows of the problems involved in the burning of residual fuels in gas turbines and the methods available for overcoming them. Work at present in progress on the development of the high temperature liquid cooled gas turbine for marine use is then described. Finally, some detailed comparisons are shown between various marine propulsion machinery installations in which both the cooled and uncooled gas turbines are compared with alternative types of machinery. The subject of free piston engines was dealt with by M. R. Huber in his paper "Adaptation du debit de gaz des generateurs a piston libres a la loi debit-pression des turbines". The first part of the paper refers to the operation of the stabilizer of a free piston gasifier and describes its action on the compression pressure, the weight of gas delivered and the power output. The possibilities of output variation between important limits by varying the compression pressure are studied and other means of adjusting the gas delivery are mentioned. In the second part a description is given of the 6,000 kW electrical set and of the test results. —*Gas and Oil Power, July 1957; Vol. 52, pp. 155-157, p. 171.*

Screw-displacement Pump

A large screw-displacement pump is being made by Stothert and Pitt, Ltd., Bath, for bunkering and oil cargo discharging duties. The pump shown on the accompanying drawing has inboard bearings, but could be adapted for water pumping duties by fitting outboard bearings with a packing gland in between. One of the valuable assets of the screw displacement pump is its ability to exhaust air from long suction lines during starting up conditions; this would make it highly suitable for use as a stripping pump, thus removing the necessity for installing additional pumps. The pump runs smoothly and quietly at about the same speed as a centrifugal pump, but it should be noted that flow takes place axially along the screws, and the rotation of movement is not imparted to the fluid. The design of the pump offers considerable scope for the employment of special materials to resist corrosion. This is of great importance when pumping distilled products so as to overcome any corrosive effect produced by the high temperature. Fluids normally handled by Stothert and Pitt pumps include volatile fluids, motor and aviation spirits, gas oils, bitumen and lubricating oil. The operation of the pump is simple, displacement being effected between the screws as they rotate; the clearances are fine and no metallic contact takes place inside the pump body. Fluid is drawn through the suction branch and propelled along



SECTION A-B-C-D



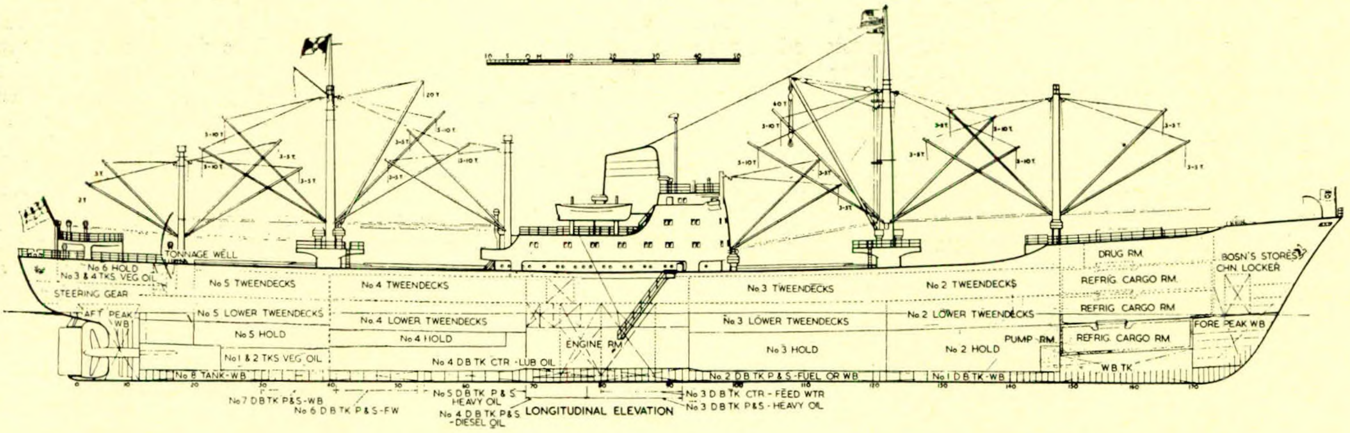
SECTION E-E

1 Pump body
2 Driving end cover
3 Gear end cover
4 Driving screw shaft
5 Driven screw shaft
6 Timing gear
7 Bearing nut
8 Gear end cover cap
9 Bearing drive end
10 Bearing gear end
11 Gland
12 Gland packing
13 Stuffing box and cap
14 Bearing nut
15 Adjusting washer
16 Lantern ring
17 Sleeve
18 Sleeve locknut
19 R.V. stuff. box cover
20 Body drain plug
21 R.V. packing
22 R.V. return spring
23 **Valve**
24 Valve seat
25 Pilot valve plunger
26 Cylinder
27 P.V. bonnet
28 Adjusting screw
29 Spring guide
30 Bearing anti-tip washer
31 Spring housing
32 R.V. gland
33 R.V. pillar
34 Piston end cover
35 Piston rings
36 Piston
37 Piston sleeve
38 P.V. spring
39 P.V. cap
40 Air cock
41 Pointer
42 Crosshead
43 Handwheel
44 Locking lever
45 Locknut
46 Pilot valve and handle
47 P.V. clamp
48 P.V. cyl. keep pin
49 P.V. body
50 Adj. screw cap
51 P.V. handwheel
52 Valve spindle
53 "O"-Rings
54 Dowty seal
55 Retaining cap
56 Eyebolt
57 Loose spigot
58 Filler plug

the pump body from both ends to the centre, where it is discharged through the outlet branch. The flow is practically pulseless and the screws are in hydraulic balance axially as the load is equally divided. The handwheel (43) shown on the accompanying drawing operates a combined relief valve and output regulating valve which is controlled by a pilot valve.—*The Shipping World, 7th August 1957; Vol. 137, p. 119.*

German 17½-knot Cargo Liner

The recently completed *Weimar,* built by Howaldtswerke Hamburg A.G., is the fourth of a series of six similar ships ordered by the Hamburg-American Line but the first to be delivered as a closed shelterdecker. The first three in the series are the *Tubingen, Gottingen,* and *Erlangen,* built in 1955-56. As open shelterdeckers these have a deadweight capacity of 8,360 tons on a draught of 25ft. 11in., the gross register being 6,341 tons. As a closed shelterdecker, the *Weimar* carries 9,425 tons on a draught of 27ft. 8in., and is of 8,408·32 gross tons and 4,994·18 tons net register. The ships remaining on order will be full-scantling vessels of increased deadweight capacity on a draught of about 29ft. 4in. Built to comply with the requirements of the highest class of Germanischer Lloyd, the



Weimar is mainly of welded construction and will be employed in the owners' cargo liner services to the west coast of South America. Her main particulars are:—

Length o.a.	510ft. 3in.
Length b.p.	461ft. 0in.
Breadth, moulded	60ft. 6in.
Depth to upper deck	38ft. 6½in.
Depth to second deck	29ft. 8½in.

The Howaldt-M.A.N. main engine is a turbocharged single-acting two-stroke unit with eight cylinders, the bore being 780 mm. and the piston stroke 1,400 mm. At 115 r.p.m., the output of the engine is 9,000 b.h.p. which gives a service speed of 17½ knots. Pressure charging is effected by two Brown Boveri blowers—one to every four cylinders. The engine is to operate on heavy oil, and three centrifuges for purifying this fuel are installed. All are of Westfalia manufacture as are two other purifiers for lubricating and Diesel oils. Three 220-kW Siemens d.c. generators supply the electrical services, each being driven at 375 r.p.m. by a M.A.N. five-cylinder four-stroke engine of 315 b.h.p.—*The Motor Ship, July 1957; Vol. 38, pp. 158-160.*

French Cargo Ship with Active Rudder

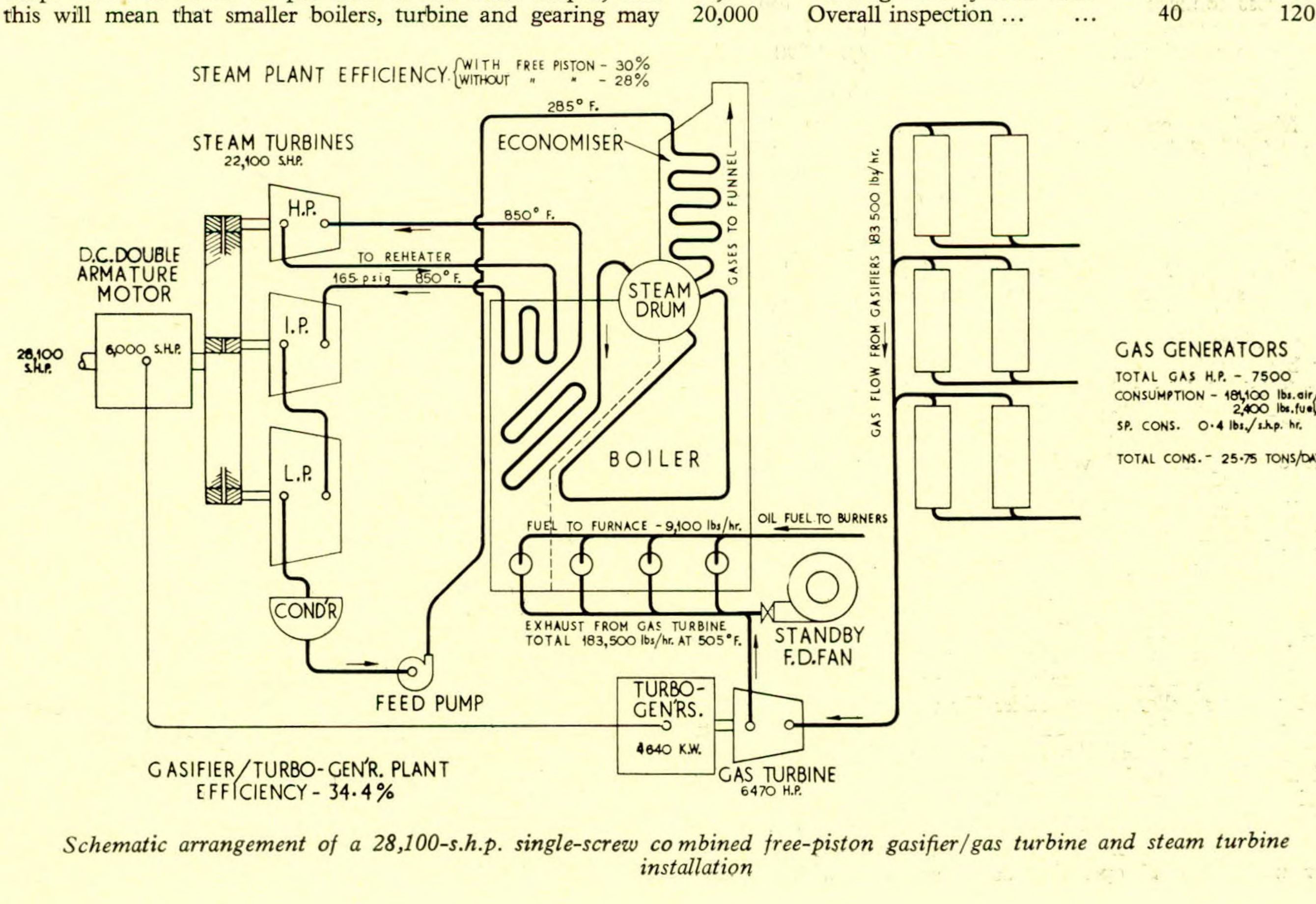
The 11,000-ton motor cargo ship *Ville de Rouen* which was recently completed by the Ch. de la Seine Maritime for the Nouvelle Compagnie Havraise Peninsulaire, is the "largest vessel to be equipped with an 'Active'" rudder and is of interest also from the employment of alternating current throughout. The demand for current is large since the "Active" rudder is operated by a 500 h.p. asynchronous motor. This type of rudder incorporates a small propeller which is driven by the motor. The vessel is 136·23 m. (b.p.) with a beam of 18·28 m., and a loaded draught of 11·506 m., the total capacity of the holds and 'tweendecks being 14,400 m^3. A 7,500-b.h.p. B. and W. engine built by Penhoët is installed. It has six cylinders and is of standard design, operating at 115 r.p.m. to develop its rated output. The speed of the ship is 16¼ knots. The engine is arranged to use boiler oil and an exhaust gas boiler is installed supplying steam for the auxiliary services on board. Three-phase alternating current distribution at constant voltage and 50 cycles per sec. is employed throughout. The motors are supplied at 380 volts, the heating circuits, the galley, etc., at 220 volts and lighting at 110 volts. There are three generating units, each driven by Diesel engines and having outputs of 400 kVA., 320 kVA. and 215 kVA. respectively at 400 volts, with a power factor of 0·75. The largest engine runs at 1,000 r.p.m. and the other two at 500 r.p.m. The 400-kVA alternator, which has a heavy flywheel, may be operated in different manners at constant speed, according to whether the ship is in port or at sea. In the first case, it is driven by the Diesel engine through a magnetic coupling, which is automatically operated when no current passes. In the second case, it is driven by a direct-coupled d.c. motor of 362 kW running at 1,000 r.p.m. The source of current for operating the d.c. motor in the second case is a d.c. generator of 380-kW 440 volts, driven from the propeller shaft. When the speed of the propelling engine falls below 80 r.p.m. this motor-generator equipment is automatically cut out and the Diesel engine immediately started, thus driving the 440-kVA alternator through the magnetic coupling. The heavy flywheel ensures the absolute continuity of the electricity supply without variation in voltage during this operation. The operation of the "Active" rudder motor which is submerged presents some problems, but these are overcome by supplying the motor with current through the intermediary of a converter which utilizes as an alternator the synchronous motor coupled to the spare blower. This converter group is supplied with continuous current from the direct current motor of the 400-kVA alternator set, which then functions as a generator. —*The Motor Ship, July 1957; Vol. 38, p. 169.*

Combined Steam and Gas Generator Machinery

An interesting proposal has been put forward by Lieut.-Cdr. A. F. Gillingham, R.N. (ret.), of the Free Piston Engine Company, Ltd., London, for the propulsion of a large oil tanker by a combined free-piston gasifier/gas turbine and steam turbine installation having a maximum output of 28,100 s.h.p. By this arrangement a proportion of the power required is supplied under normal full power conditions by a free-piston installation working in conjunction with a conventional high pressure steam turbine plant incorporating reheat, the latter supplying the major proportion of the power. The resulting combination reduces by half the existing difference in specific consumption between typical steam and Diesel installations as they are at present, while at the same time enabling powers to be achieved which are far beyond the reach of the large single-screw, direct-drive motor vessel. The free-piston gas turbine engine can be described as a modified steam turbine driven by hot gases instead of steam. The gasifier, which consists of a horizontal Diesel engine cylinder with opposed pistons operating on a two-stroke cycle, replaces the boiler. There is no crankshaft and the pistons are only lightly linked to keep them in phase. The products of combustion, together with the excess scavenge air, constitute the output of the machine in the form of hot gas under pressure for use in a turbine for the production of mechanical power. In the scheme put forward by Cdr. Gillingham, the steam turbine drives the propeller shaft direct through reduction gearing, while the free-piston gasifiers supply a gas turbine-driven electric generator which supplies power to an electric motor coupled to the propeller shaft abaft the gearbox. The proportional powers of the two separate plants have been so designed that all the furnace air requirements of the boiler can be satisfied by utilizing the excess air in the high temperature exhaust from the gas turbine, this feature being responsible for the large saving in fuel shown by the combined installation over a conventional high pressure boiler/turbine installation of comparable horsepower using furnace air at ambient temperature. The free-piston installation comprises the following components: Six GS.34-type gasifiers having a total output of 7,500 gas horsepower. A constant speed non-reversible gas turbine having an output of 6,470 h.p. direct coupled to an electric generator of 4,640-kW output, and an electric motor on the main propeller shaft having an output of 6,000 s.h.p. The steam turbine installation comprises a high-pressure boiler operating at a pressure of 850lb. per sq. in., with a superheat temperature of 850 deg. F., supplying steam to a three-stage geared turbine incorporating reheat between the h.p. and l.p. stages. The output of this turbine is 22,100 s.h.p. This is about 75 per cent of the total output, and this will mean that smaller boilers, turbine and gearing may be used. Manœuvring and astern power may be obtained from the electric propulsion motor alone or by combined electric motor and steam turbine. It is considered that when the oil firing installation is shut down, the heat from the gasifiers is not high enough to damage the reheater tubes which, under astern conditions, are starved of steam. An interlock system is provided to ensure that the sprayers supplying oil to the reheater section of the boiler are shut off during manœuvring and when going astern. The fuel used with a combined gasifier/turbo-electric and steam turbine plant is 123·25 tons per day, and a steam plant supplying all the power would consume 147·25 tons per day. The saving in fuel is therefore 24 tons per day. With fuel at 240s. per ton this means a saving of about £288 per day, so that on a 30-days' voyage as much as £8,640 would be saved, and assuming five voyages per year £43,200. In addition to the above saving on fuel costs, there will also be a saving of bunker fuel requirements to the extent that 916·5 tons less fuel will be needed for each single voyage. The space saved can be used to carry more cargo. The addition of the free-piston installation will entail very little additional maintenance measured in man hours. Most of the maintenance work is far more simple than that on a large Diesel engine, and does not require such a high degree of skilled labour. The gas turbine itself requires virtually no maintenance. The accompanying estimates of maintenance and wear figures for the gasifiers have been compiled from extensive running experience under both service and test conditions in France, the United States of America and at sea.

(a) INTERVALS BETWEEN INSPECTIONS AND MAINTENANCE MAN HOURS

Interval (hours)	*Parts*	*Time taken (hours)*	*No. of man hours*
2,500	Moving parts	4	12
5,000	Glands	8	32
10,000	Engine case	—	—
20,000	Exhaust end cylinder liner	20	60
20,000	Scavenge end cylinder liner	20	60
20,000	Overall inspection	40	120



Schematic arrangement of a 28,100-s.h.p. single-screw combined free-piston gasifier/gas turbine and steam turbine installation

(b) Wear

Parts	*Wear per 1,000 hours* *m.m.*	*in.*
Diesel cylinder liner, exhaust end ...	0·08	0·0032
Diesel cylinder liner, scavenge end ...	0·055	0·0027
Top piston ring, exhaust end	0·27	0·0107
Top piston ring, scavenge end... ...	0·16	0·0063
Remaining piston rings	0·08	0·0032
Compressor cylinder liners	0·05	0·0020
Compressor piston rings	0·08	0·0032

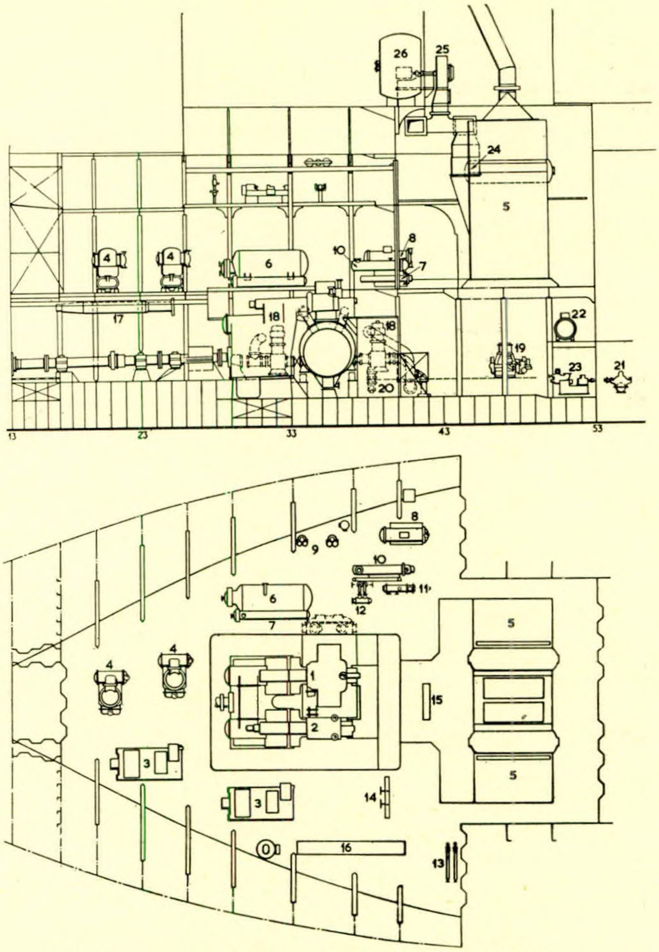
—The Shipping World, 10th April 1957; Vol. 136, pp. 367-369.

Swedish-built Tanker *World Splendour*

An oil tanker of 40,532 tons deadweight, the largest yet built in Scandinavia, was delivered to the Niarchos group on 3rd June. This vessel, the *World Splendour,* built by Kockums Mekaniska Verkstads AB, Malmo, Sweden, is the 400th vessel that has been built by these shipbuilders. The *World Splendour* is powered by a set of Kockum-Laval steam turbines developing 16,500 s.h.p. at 103 r.p.m., and designed to give a service speed of 17 knots. The pump room is one of the most spacious to be found in an oil tanker of comparable size and contains four cargo pumps of 1,250 tons capacity and two 150-ton stripping pumps. De Laval turbo-alternators of the back pressure type are fitted. The first system of this kind was installed in the *World Sincerity.* The principal particulars of the *World Splendour* are as follows:—

Length o.a.	699ft. 6in.
Length b.p.	665ft.
Breadth moulded	97ft. ¼in.
Depth moulded	49ft. 3in.
Draught, summer	36ft. 3¼in.
Deadweight	40,532 tons
Gross tonnage	26,307 tons
Net tonnage	16,345 tons
Cargo capacity	2,011,590 cu. ft.
Bunker capacity	4,495 tons
Service speed	17 knots
Machinery output	16,500 s.h.p.

The *World Splendour* has been built to comply with the rules of the American Bureau of Shipping, and also satisfies Ministry of Transport regulations as to her machinery and equipment. She has a raked stem and streamlined midship and poop deckhouses, the latter structure being reduced in breadth in order to provide additional longitudinal strength and a satisfactory accommodation arrangement. Apart from the A.B.S. crack arresters, the vessel is all-welded. A combined longitudinal and transverse framing system has been used in the construction of this vessel and the hull is subdivided into thirty-three cargo tanks, eleven centre and twenty-two wing. The longitudinal bulkheads are of flat plate construction, only the transverse bulkheads being corrugated. The cargo pumps, of which there are four, are driven by steam turbines, located in the engine room. They are of Swedish manufacture and each has a capacity of 1,250 tons of water per hour at 1,650 r.p.m. The two steam reciprocating stripping pumps have been supplied by Thom, Lamont and Co., Ltd., and each has a capacity of 150 tons per hour. The forward pump room contains a bilge pump of 50 tons per hour capacity and a fuel oil transfer pump of 75 tons per hour capacity. The use of a spade rudder allows a generous space round the propellers; and this, together with the adoption of a five-bladed propeller, has considerably reduced vibration. The rudder, which weighs 28½ tons, is suspended from three points, the top one taking the full weight. The propeller weighs 32 tons and has five blades integral with the boss. Electro-hydraulic steering gear equipment supplied by John Hastie and Co., Ltd. is fitted. Steam is supplied by two Foster Wheeler "D" type oil-fired boilers generating 26 tons of superheated steam per hour under normal operating conditions and with a maximum output of 36 tons of steam per hour. The steam conditions are 600lb. per sq. in. pressure and 865 deg. F. The main propelling turbine is an axial-flow cross-compound unit consisting of one high pressure and one low pressure turbine connected through flexible couplings to double reduction gearing. The l.p. turbine has impulse reaction blading. The stainless steel blading in both the h.p. and the l.p. turbine rotors is fixed to the discs by the usual de Laval side entry, bulb root method. The astern turbine is built into the forward end of the l.p. turbine, an arrangement which not only simplifies construction and design, but is also of great advantage in maintenance and servicing. It consists of one Curtis wheel and a single-stage wheel. A deflector plate is fitted in the exhaust between the ahead and the astern blading to prevent the steam from impinging on the opposing blades. A number of bleed points have been fitted to the turbines so that steam may be extracted for feed heating and similar purposes. The pipe connexions are so arranged that either one of the turbines may be operated in an emergency, with direct admission of steam and discharge to the condenser. Electricity for power and lighting is supplied by two de Laval turbo-alternators, each having an output of 900 kVA.



Machinery arrangement in the World Splendour

1 L.P. turbine	10 1st stage feed heater	19 Gen. serv. pumps
2 H.P. turbine	11 Gland condenser	20 Main extr. pumps
3 Turbo-alternators	12 Main air ejector	21 Cargo pump
4 Evaporators	13 Fuel oil heaters	22 Cargo pump condenser
5 Boilers	14 Main control	23 Cargo pump turbine
6 Steam to steam generator	15 Boiler control	24 Steam air heater
7 Drain cooler	16 Main switchboard	25 F.D. fans
8 Aux. condenser	17 Spare shaft	26 Deaerator
9 L.P. feed pumps	18 Main circ. pumps	

The four steam turbines driving the cargo pumps have been installed at the forward end of the engine room. They are of Swedish manufacture and drive the pumps through Renk reduction gearing. A gastight gland is fitted in the bulkhead through which the pump driving shaft passes.—*The Shipping World, 26th June 1957; Vol. 136, pp. 617-621; p. 625.*

Inflatable Life Rafts

Most rafts are supplied in a canvas valise. The commonest form of stowage at present in use, and one which can be made up by any shipwright, consists of a collapsing box in which the front and both ends are hinged to the base and held in position by a lip around the lid with the result that when the latter is raised the three sides fall away. The base should be raised by chocks 4 to 6in. high above the deck to allow air to circulate and reduce the risk of water entering. There should be a number of holes in the base, again for ventilation and to allow any water which should enter to drain away. Between the valise and the base of the box a grating should be fitted. The raft should not be too tight a fit in the stowage box so that there is room for air to circulate all round. The positioning of the rafts aboard a ship is a question which obviously varies considerably with each type of vessel. In cargo vessels where either two or four rafts are carried it is suggested that half the complement be carried somewhere in the bridge structure and the other half right aft. Stowage amidships is preferable to stowage on the ship's side as being less liable to damage, making it easier to launch either to port or starboard. Apart from being invaluable in the event of disaster to one's own ship, the inflatable life raft can be of great assistance in picking up a man overboard or rendering assistance to another vessel in distress when the weather is too rough to lower a boat. For this purpose a coil of 3in. grass rope should be available near the raft so that it may be secured to the end of the operating cord which becomes the raft's painter once it is inflated. In passenger vessels where a large number of rafts are carried, their positioning must be governed largely by the deck space available. Where the promenade deck is open to the sea a continuous collapsible box stowage against the inboard bulkhead is recommended as this can be used as seating accommodation. However, many passenger vessels may be permitted to carry the rafts in "nests" of 12 or 16 in lieu of some of the lifeboats; here the rafts are stowed on inclined launching ways made of hardwood rollers. Bearing in mind that one of the advantages of the inflatable life raft over the life boat is that it can be carried to and launched from any part of the ship, it is considered important that the "nests" are so fitted that the rafts can be dropped on deck should weather or other conditions render it undesirable to launch them abreast of the "nests" themselves. Inflatable life rafts can be boarded from the sea by means of the boarding ladders but chances of survival are increased if it is possible to get aboard dry shod. One way of achieving this is by jumping straight on to the canopy—it is better to remove one's boots before doing this and it should not be attempted from a greater height than about fifteen feet. If time and conditions permit, a simpler way is via a Jacob's ladder. In cargo vessels it may be recommended that one of these ladders with flat wooden rungs and hooks to fit the ship's rail be kept adjoining each raft stowage. In passenger vessels the method at present preferred appears to be 6ft. wide rope ladders with flat wooden rungs about 5in. deep.—*The Motor Ship, August 1957; Vol. 38, p. 211.*

Turbocharged Two-stroke Engine

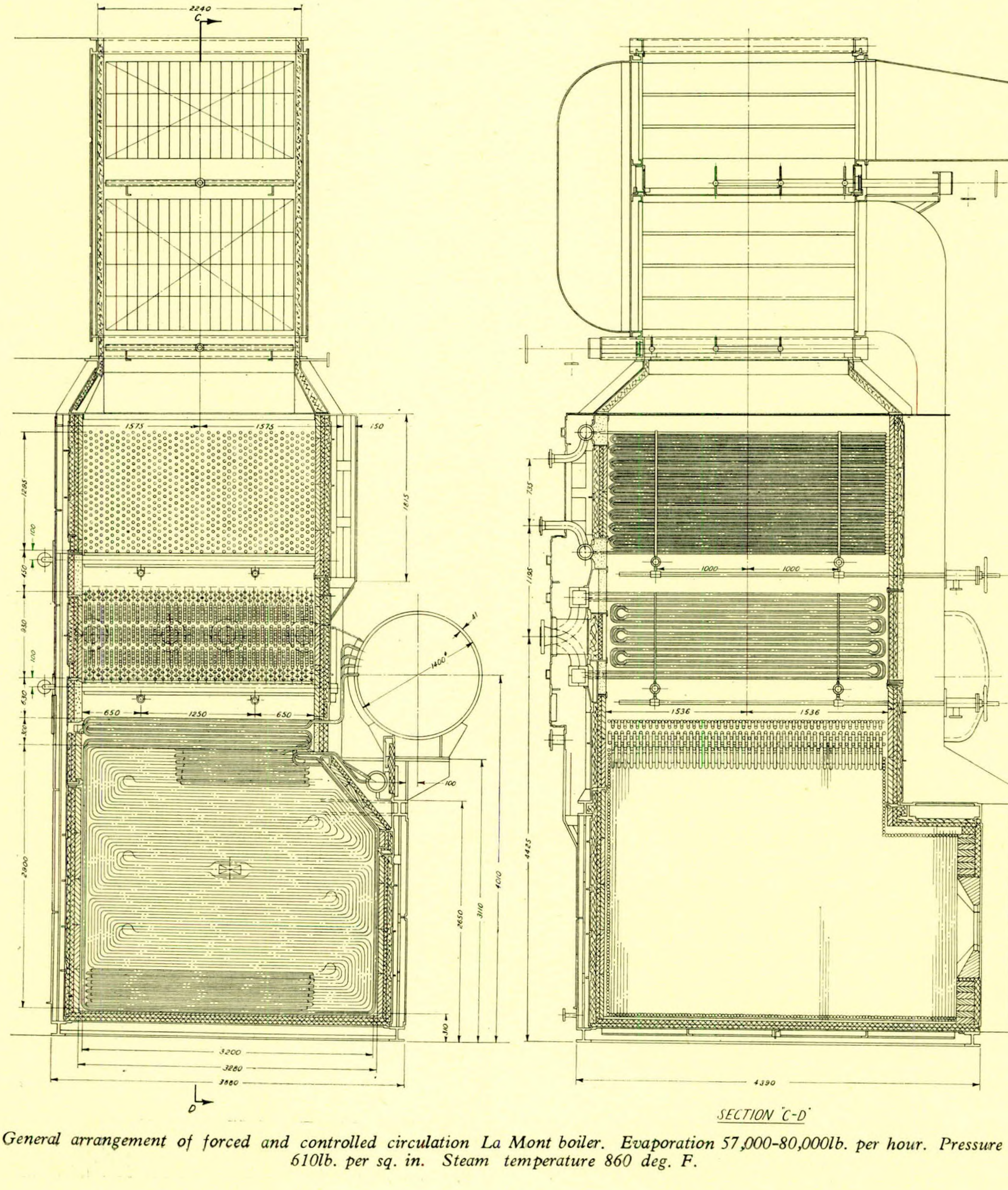
Preliminary test bed trials have been running since the end of last year on a new Diesel engine at Werkspoor N.V., Amsterdam. The engine, which has been designated type TES 456/70, is a trunk piston, two-stroke, reversible engine having six cylinders of 450 mm. bore and 700 mm. stroke. It will develop 1,800 b.h.p. at 250 r.p.m., and will also be produced as an eight-cylinder engine with an output of 2,400 b.h.p. The engine now undergoing trials is not turbocharged, but it is intended that trials shall be carried out shortly with a turbocharged engine. The new Werkspoor engine has been designed for uniflow scavenging, and has two exhaust valves in the cylinder head and scavenging ports in the cylinder. This type of construction has, it is claimed, among other advantages, a better scavenging effect than that obtained by other methods. The timing of exhaust valves and scavenging ports can be set to the optimum point, and the inlet ports in the liner are of simple design. The bedplate and frame are of welded construction. They have the same rigidity as if they were of cast construction but require less material, since steel has a higher modulus of elasticity than cast iron. The cylinder block is of cast iron. The camshaft, which is arranged along the side of the cylinder block, is driven from the crankshaft through gearing. It has double fuel valve and exhaust valve cams for ahead and astern running. The camshaft is moved axially by means of the reversing mechanism, which is mounted in line with the camshaft on the front of the engine above the control panel. This operation is effected by means of a compressed air system. The cylinder head contains two exhaust valves, the fuel injection valve and a starting air valve. The relief valve is mounted in the side of the head which is water cooled, the water entering the head through four outlets in the cylinder block, whence it flows to the cooling water outlet via the exhaust valves. The piston is of the built-up type and consists of three parts, crown, skirt and gudgeon pin bush. The piston crown is of light alloy to ensure adequate heat transfer to the cooling oil with only slight temperature stresses. Cast iron is used for the piston skirt. Since lubrication of the gudgeon pin is a fairly difficult problem, because in a two-stroke engine the load is largely directed downwards, the bearing has a large surface and the gudgeon pin has been chromium-plated. The pistons can be removed without taking off the cylinder head. Each of the cylinders has a piston scavenging pump which is driven from the main piston through two levers and connecting rods. The cooling water and lubricating oil pumps are separate from the engine and are electrically driven. There are no drilled holes in the crankshaft for the lubricating oil, and the engine has a dry sump. Manœuvring is effected from a handwheel control fitted at one end of the engine. When turbocharging tests are carried out the scavenging pumps will be replaced by turboblowers.—*The Shipping World, 17th July 1957; Vol. 137, p. 63.*

German Tanker with La Mont Boilers

The largest oil tanker so far built for a German owner has been delivered from the Deutsche Werft, Hamburg-Finkenwerder. This vessel, the *Caroline Oetker,* 32,000 tons d.w., has been built for Rudolf A. Oetker. The *Caroline Oetker* is powered by a set of steam turbines geared to a single screw, and has a steaming range of about 12,000 nautical miles. Her two La Mont boilers are believed to be the largest of their type to be installed in a European merchant vessel. The *Caroline Oetker* is a three-island vessel having one complete deck, closed forecastle with tonnage opening, bridge and poop. The principal particulars are as follows: —

Length o.a.	660ft.
Length b.p.	630ft.
Breadth moulded	87ft.
Depth moulded	45ft. 6in.
Draught, summer	34ft. 2in.
Deadweight	32,000 tons
Gross tonnage	21,000 tons
Speed at maximum s.h.p. ...	17 knots
Machinery output—Maximum	16,500 s.h.p.
Normal	15,000 s.h.p.
Cargo capacity (about) ...	1,530,000 cu. ft.

The hull is divided by oiltight and watertight bulkheads into the following compartments: Fore peak, chain locker and stores, dry cargo space forward deep tank, pump room and cofferdam, Nos. 1 to 10 cargo tanks, after cofferdam and main cargo pump room, fuel oil bunker tanks, engine and boiler room, with double bottom tanks, after peak and steering gear compartment. The cargo space is divided into thirty tanks having a total capacity of about 1,530,000 cu. ft., by means of two longitudinal bulkheads and nine transverse bulkheads. The main cargo pump room has been fitted aft between the wing longitudinal bulkheads, and is equipped with four steam turbo-driven cargo pumps each of 980 tons per hour capacity. The pumps are driven by AEG steam turbines through reduction gearing; the turbines are located in the engine room and drive the pumps through a gastight gland fitted in the bulkhead separating the two compartments. The turbine speed is 7,500 r.p.m. and the pump speed 1,750 r.p.m., at 550 h.p. The steam pressure is 544lb. per sq. in. at 617 deg. F. There are also two stripping pumps each of about 153 tons per hour capacity. The propelling machinery in the *Caroline Oetker* consists of a reversible two-cylinder AEG turbine set, with a single astern turbine in the l.p. casing, driving through double reduction gearing. The installation has been designed for a



General arrangement of forced and controlled circulation La Mont boiler. Evaporation 57,000-80,000lb. per hour. Pressure 610lb. per sq. in. Steam temperature 860 deg. F.

rated output of 15,000 s.h.p. at 110 r.p.m., the maximum power obtainable at 10 per cent overload being 16,500 s.h.p. at 113 r.p.m. The astern turbine is capable of developing 80 per cent of the normal forward s.h.p. at 50 per cent of the normal r.p.m. The h.p. turbine rotates at 5,500 r.p.m. and the l.p. turbine at 3,800 r.p.m. when the rated output is being developed. The steam conditions are 600 lb. per sq. in. and 840 deg. F. at the turbine inlet. To give economic operation at all powers, steam is admitted to the h.p. turbine through a total of five sets of nozzles. One of these is always open, and gives powers up to half the rated output, i.e., 7,500 s.h.p. To achieve higher powers, successive nozzle groups are opened by hand, the resultant powers obtainable being 10,000 s.h.p. with two groups, 12,500 with three, 15,000 with four and 16,500 with five. The principal particulars of each boiler are as follows:—

Steam pressure at superheater outlet	612lb. per sq. in.
Superheat temperature ...	860 deg. F.
Normal full power evaporation	58,000lb. per hr.
Maximum evaporation ...	80,000lb. per hr.
Feed water temperature to economizer	320 deg. F.
Air temperature	392 deg. F.

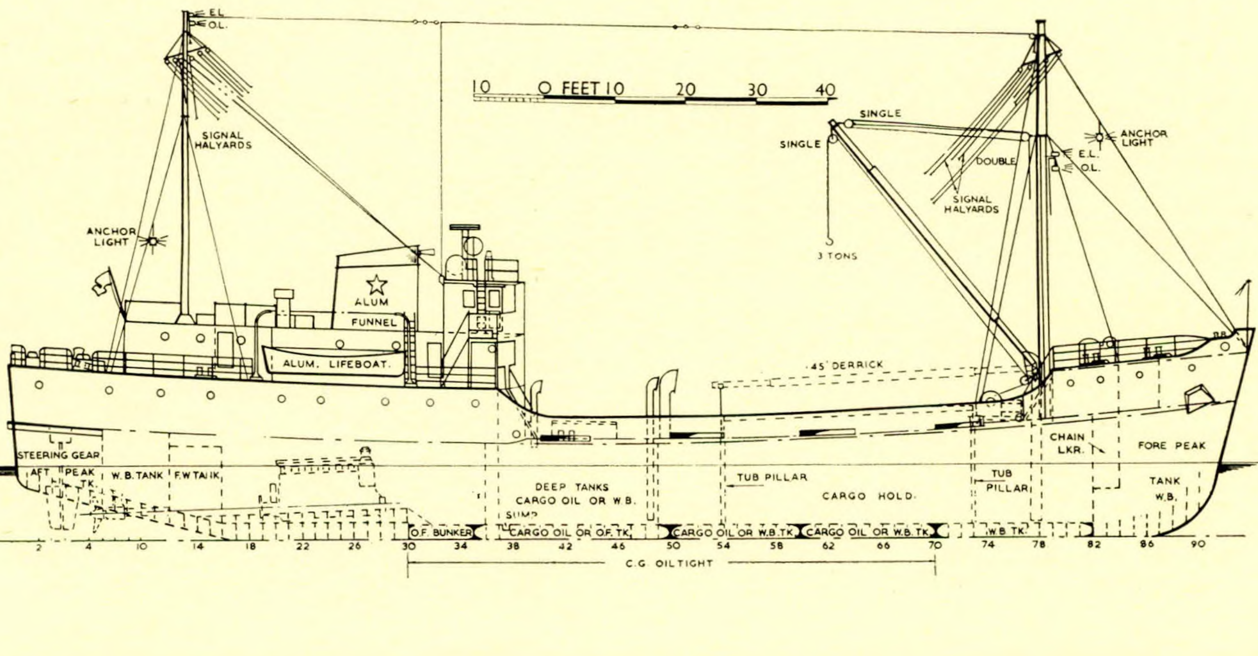
The combustion chamber is almost completely water-cooled by lining the floor, side walls, rear wall and the roof with close-pitched tubes of $1\frac{1}{4}$-in. diameter. Only the front wall, which contains the two spinning cup burners of the Saacke type, is of refractory. The combustion gases pass on their way to the funnel the evaporating tube bank, six rows deep, the two-stage superheater, the economizer, and finally the two-stage air heater which is made of cast-iron elements. Circulation is maintained by only one circulating pump per boiler having a capacity of 435,000lb. per hour at a differential pressure of 35lb. per sq. in.—*The Shipping World, 15th May 1957; Vol. 136, pp. 481-486.*

Vessel for South African Coastal Service

The m.s. *Zulu Coast,* launched from the Ardrossan Dockyard, Ltd., is a coastal vessel specially designed for service in South African waters. She will operate mainly in the trade between Cape Town and Port Nolloth, on which service she will maintain essential supplies to the fishing industry at that port, the diamond mines at Oranjemund and Alexander Bay, the copper mines at O'Okiep and Nabakeep, and the many general traders in Namaqualand. She is owned by Thesen's Steamship Co., Ltd., Cape Town, which is within the Coast Line organization, and is built for the carriage of general cargo and cargo oil, the latter being carried in deep tanks abaft the cargo hold and in Nos. 2 and 3 double bottom tanks. The oil cargo is discharged by a special 50-tons/hr. Borga pump. Of about 760 gross tons, the *Zulu Coast* has an overall length of 173ft. 8in., a moulded breadth of 36ft. and a moulded depth of 13ft. 6in. There are five watertight bulkheads and three hatches, the two aftermost having MacGregor hand-operated oil-tight covers. To facilitate rapid manœuvring, twin spade-type rudders are fitted, each with an area of about 23ft.^{2} The deck machinery, comprising one cargo winch, a windlass and a capstan, is of Vickers-Armstrong hydraulic type, while the electro-hydraulic steering gear by Brown Brothers has hand steering provided by hydraulically connecting the bridge steering pedestal direct to the main hydraulic cylinders; in the event of the motor being stopped, the switch-over from power to hand takes place automatically. The two main 480-b.h.p. engines are British Polar type M461 direct reversing single-acting two-stroke units, each directly coupled to an outward-turning propeller. The rated speed is 325 r.p.m. and in tropical conditions with a sea water temperature of 85 deg. F. the output at the same speed is not less than 400 b.h.p.—*The Motor Ship, August 1957; Vol. 38, pp. 214-215.*

Free-piston Engined Liberty Ship

Trials were carried out of the fourth Liberty ship, converted under the programme of the American Maritime Administration involving the installation of four different types of machinery to replace the 2,500 i.h.p. reciprocating steam engines. For the *William Patterson,* free-piston machinery was chosen and she is the first large cargo ship to have plant of this class, although in France about a score of vessels have been built, including coasters and minesweepers, with such equipment. The order was placed with the Cleveland Diesel Engine Division of General Motors Corporation on October 6th 1954. In principle the machinery comprises two major components, the free piston gas generator and the turbine. The generator is fundamentally a two-stroke opposed piston Diesel engine with the pistons rigidly connected to pistons of a compressor cylinder located at at each end of the centre housing. In the *William Patterson* there are six standard type gas generators supplying the combustion gases to two turbines, which drive a single propeller shaft through gearing. The bore of the power cylinders is 13·39in. and of the compressor cylinders 35·43in., the common stroke being 18in. Each of the turbines has two gas inlets, the gases from the



degree of viscous damping. Single-bank running can be carried out by disconnecting the drive of the bank to be isolated at the input coupling to the gearbox. This permits operation for extended periods on low load, and also enables half power to be developed in an emergency, should a fault develop in one bank. Twin-bank engines designed for single-bank operation are fitted with a duplex governor system, so that each bank is under the control of its own governor. Balanced governor response and equal sharing of load between banks are ensured by the control of the hydraulic feed to both governors by a single pilot valve. In addition, the two governor outputs contribute movement to a common floating linkage, the mean deflexion of which is used to control the fuel pump racks of both banks of cylinders. For generating sets, speed-increasing ratios may be used to give alternator speeds of 750 and 1,000 r.p.m. for 50 cycles supply, or 900 and 1,200 r.p.m. for 60 cycles supply. The maximum continuous power available is 1,320 b.h.p. with a crankshaft speed of 750 r.p.m. or 865 kW. For industrial units, the output is 930 kW. As a marine propulsion unit, the maximum continuous power is 1,190 b.h.p. with a crankshaft speed of 750 r.p.m. Reduction ratios of up to 3 : 1 are available, and the engine is capable of operating over a speed range of from full speed to one-third full speed. The reverse/reduction gearbox is built in to the engine unit itself and great compactness is therefore possible, there is no problem regarding gearbox alignment.—*The Shipbuilder and Marine Engine-Builder, October 1957; Vol. 64, p. 598.*

Floatless Liquid Level Control

A new, floatless, liquid level control is now available for a wide range of industrial applications, where it is necessary to control the level of liquid in boilers, tanks or other vessels. The level control has been designed for use in conjunction with electrically driven pumps and incorporates the Teddington liquid level switch. The unit is suitable for steam boiler applications, in which it can be used for feed water pump control and/or a low water alarm and cut out. The principle on which the design has been based is that heat is dissipated more rapidly through a liquid than through air or vapour. The liquid level control switch comprises two probes, each consisting of an Invar rod in a brass tube. One end of the Invar rod is left free, while the other end is connected to the brass tube. It is known that metals expand with a rise in temperature, and the rate of expansion varies with different metals. With Invar, the expansion is very small, while with brass expansion is considerable; use is made of this difference in rate of expansion in the operation of the control. On each end of the Invar rod there is a pivoted switch lever. One of the rods has a small, continuously-heated, electrical heater wound around it so that one probe is always hotter than the other. The other probe works in opposition, to compensate for changes in the ambient temperature. When the probes are immersed in liquid, the heat is dissipated very rapidly, and the difference in temperature between the two probes is not sufficient to operate the switch. When the liquid level falls, however, and the probes are no longer immersed, the heat is dissipated at a much slower rate, and this increases the difference in temperature between the two probes, causing the switch to close the pump circuit. The main body housing the switch probes is normally made in close-grained high-duty cast iron (semi-steel). For special applications, it can be supplied in gunmetal or aluminium. The switch head is completely sealed from the liquid under control.—*The Shipbuilder and Marine Engine-Builder, October 1957; Vol. 64, p. 601.*

Gears for the *Statendam*

The two main propulsion gears for the twin-screw passenger liner *Statendam,* a recent addition to the fleet of the Holland-America Line, were designed and built by the Maag Gear-Wheel Co., Ltd., of Zurich. The gears are of the double-reduction, articulated type. The gear cases are constructed of cast iron, and single-helical gears are installed in both reductions. The pinions of both the first and second reduction are of nickel chrome steel and have case-hardened tooth flanks, while the gear wheel rims are of nickel chrome molybdenum heat-treated steel. All the teeth were ground with highest precision on Maag machines, and the pinions are provided with special tooth corrections. The primary and secondary trains are connected by quill shafts, which are located by thrust collars. The main thrust bearing, which takes the propeller thrust, is built-in and placed at the forward end; all the bearings are easily accessible for inspection and maintenance. Remote temperature control makes it possible to concentrate all thermometers for each gear on one instrument panel, which is located on top of the gear case. This arrangement allows all bearing temperatures to be checked from the operator's platform. The main gear data are given in Tables I and II.

TABLE I

Condition	*Unit*	*H.P.*	*R.P.M.*
Service power ...	High pressure ...	5,600	4,464
	Low pressure ...	4,400	2,842
	Propellers ...	10,000	117
Full power ...	High pressure ...	5,900	4,578
	Low pressure ...	5,100	2,915
	Propellers ...	11,000	120
Reverse power ...	High pressure ...	1,800	3,930
	Low pressure ...	4,200	2,500
	Propellers ...	6,000	103

TABLE II

Item	*First Reduction* *High pressure*	*First Reduction* *Low pressure*	*Second Reduction* *High pressure*	*Second Reduction* *Low pressure*
Pitch circle diameter, in.	9·57/56·3	13·0/52	20·5/133	
Face width, in. ...	11·0	9·5	18·0	
Tooth loading, lb. per sq in. at 10,000 h.p....	1,500	1,500	2,510	1,950
K-factor at 10,000 h.p....	184	136	139	110

—*The Shipbuilder and Marine Engine-Builder, July 1957; Vol. 64, p. 459.*

Fluidrive for Shaft-driven Generators

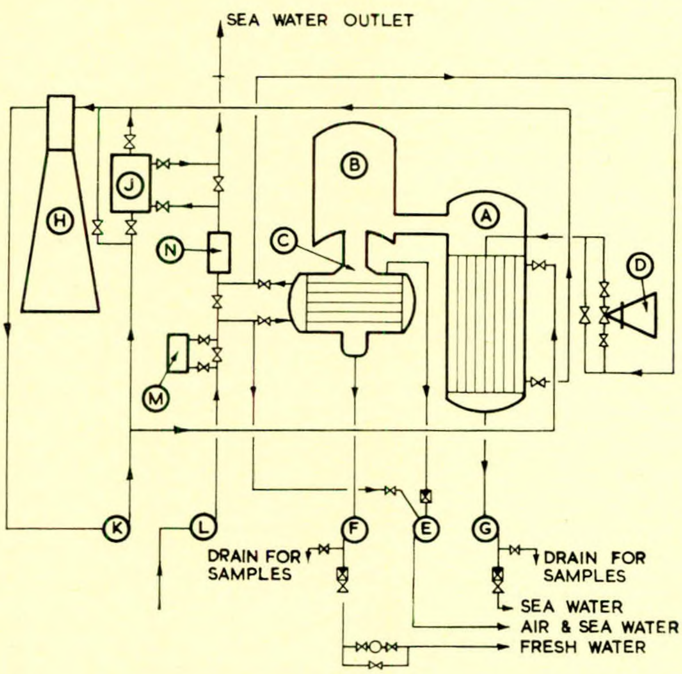
A recently completed 27,000 tons d.w. motor tanker powered by a six-cylinder 7,750 b.h.p. at 103 r.p.m. Doxford engine has a 150 kW 500 r.p.m. Sunderland Forge compound-wound dynamo driven from the shaft through Renold chains and a size 41 Vulcan-Sinclair scoop-control fluid coupling. The auxiliary generator is normally in operation when the engine is running at the service speed of 103 r.p.m., provision being made in the installation for automatic voltage control over a 10 per cent speed range. When manœuvring or running at reduced speed the scoop tube is withdrawn, thereby disconnecting entirely the drive to the generator. The fluid coupling permits the main engine to start, manœuvre and run up to full speed without driving the generator, for when the scoop is withdrawn the working circuit is empty. The coupling cannot transmit torsional vibrations, shocks or the cyclic speed variation of the engine to the generator, which can be accelerated to working speed very smoothly, the rate being determined by the engagement of the scoop tube. The essential auxiliaries served by the shaft-driven generator include the main engine jacket water pumps, valve cooling water pumps, sea water cooling pumps, engine forced lubrication pump, boiler forced draught fan, air compressor, ballast and transfer pumps, general service pump and purifiers and workshop motors. There is no prime mover maintenance with this arrangement and the power is generated with the much higher fuel economy of the larger engine.—*The Marine Engineer and Naval Architect, August 1957; Vol. 80, p. 306.*

Improved Lithium-based Multi-purpose Greases

The remarkable multi-purpose properties of lubricating greases based on lithium 12 hydroxy stearate have become well known to the industry over the past seven years. Greases of this type are marketed under the Shell Alvania brand. Recent work at the Shell Thornton Research Centre near Chester has led to improvements in the already outstanding properties of these greases, particularly with respect to the protection they afford against moisture corrosion in bearings, and in their working lives at high temperatures. The use of lithium 12 hydroxy stearate soap in a grease gives outstanding properties compared with conventional sodium, calcium or aluminium soap greases, and even compared with other lithium soap greases. These properties, which include great mechanical and high-temperature stability, high melting point, excellent resistance to water, long "shelf life" and resistance to drying out and hardening in service, enable such greases to be used for the large majority of industrial and marine grease applications, replacing numerous conventional types of grease. Further research and development work carried out at Thornton Research Centre has led to the inclusion in the new greases of a powerful corrosion inhibitor, which confers outstanding anti-corrosion properties even in the presence of large quantities of water. The oxidation stability of the greases, under both working and static storage conditions, has also been considerably improved, so that they can now be used at higher working temperatures than could the earlier Shell Alvania greases, or for considerably longer periods at the same working temperatures.—*The Shipbuilder and Marine Engine-Builder, October 1957; Vol. 64, p. 603.*

Fresh Water Generator

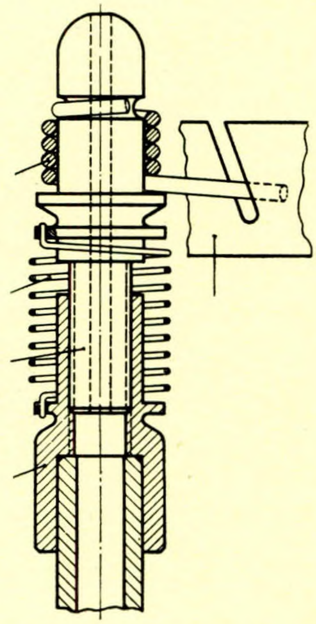
A means of producing about eight tons of fresh water daily per 1,000 b.h.p. of a Diesel engine without any additional fuel cost is claimed for the fresh water generator produced by Atlas Maskinfabrik of Copenhagen. This unit operates on a system different from that of other distilling plants. It is based on the utilization of the waste heat in the cooling water from the Diesel engine coupled with a high vacuum. The only cost involved is that to produce the power for the pumps, which amounts to about 7 kW per ton of fresh water produced. Assuming the temperature of the cooling water discharged from the Diesel engine at about 65 deg. C. (150 deg. F.), some of the water is led, in parallel with the normal cooler, to the heat exchanger of the fresh water generator, whence it is returned to the engine at a temperature of about 55 deg. C. (131 deg. F.). As the plant works under 93 per cent vacuum, these temperatures are sufficient to evaporate the desired quantity of sea water. The vapour from the heat exchanger is passed into a separator where any entrained water particles are drained off, after which it is condensed to fresh water in the condenser. A special pump maintains the vacuum and the condensate is transferred from the condenser to the fresh water tank by a special pump. Only a small part of the sea water fed in is, in fact, evaporated. About three-quarters of the sea water introduced is removed by an extraction pump, thus keeping salt concentration to the minimum. By this means and with the low evaporating temperature, the usual work of clearing salt deposits is said to be eliminated.—*The Motor Ship, September 1957; Vol. 38, p. 250.*



Schematic arrangement of fresh water generator: (A) Heat exchanger; (B) Separator; (C) Condenser; (D) Float valve; (E) Vacuum pump; (F) Condensate pump; (G) Bleeder pump; (H) Main Diesel engine; (J) Cooler for Diesel engine; (K) Cooling water pump for fresh water; (L) Cooling water pump for sea water; (M) Air cooler; (N) Lubricating oil cooler

Automatic Tappet Adjuster

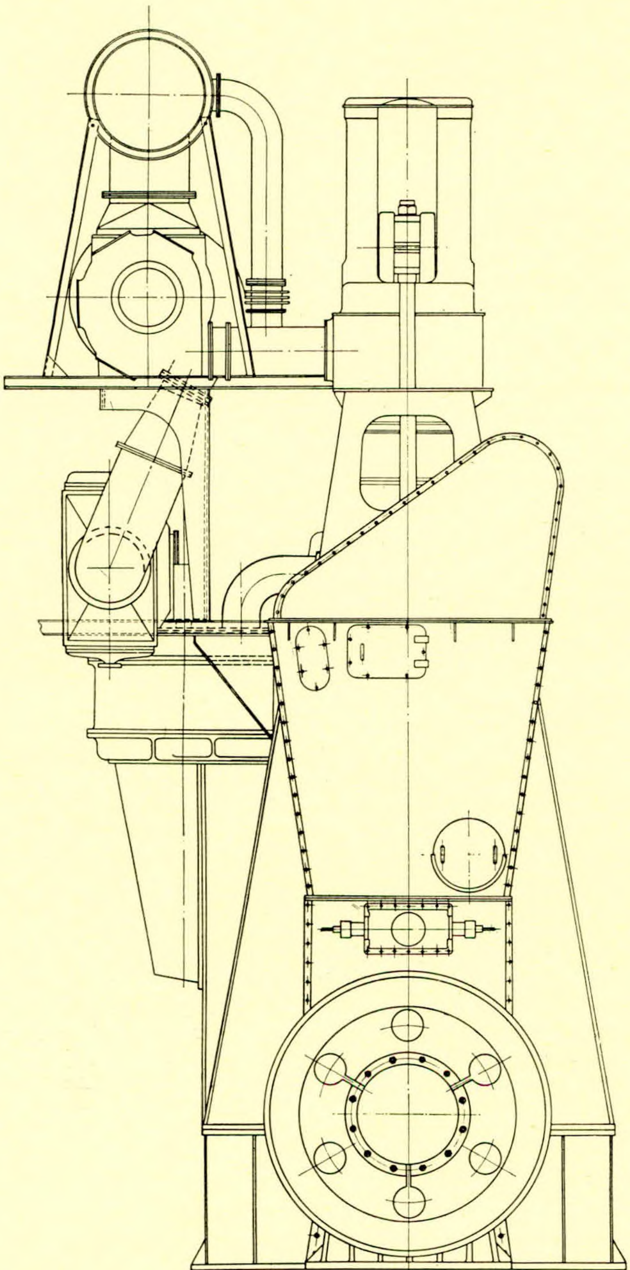
Teveswerke have developed a mechanical valve clearance adjuster for all types of internal combustion engine. The fitting is attached to the top of the push rod and consists basically of a threaded spindle and a nut, the two elements being connected by a torsion spring. Seated on the collar of the threaded part is a freewheeling spring, the lower end of which is extended to form a tail which rides in the slanting slot of a control plate. When the push rod rises to open the valve, the lower end of the spring is guided in the slot so that the freewheel spring opens up and slips freely on the cam of the threaded part without imparting any rotary motion to it. On the downward movement of the push rod the slot causes the freewheel spring to contract and to grip tightly on the collar of the threaded part, thus screwing it down slightly into the nut and decreasing the clearance. The followers are set on the base circle of the cam, the clearance is automatically taken up by the torsion spring until the valve motion becomes positive.—*Gas and Oil Power, August 1957; Vol. 52, p. 191.*



Teves auto tappet adjuster fitted to push rod end

Doxford Supercharged Engine

The first engine order from Wm. Doxford and Sons (Engineers), Ltd., as a pressure-charged unit is now running shop trials. It has six cylinders with a bore of 650 mm. and a combined piston stroke of 2,320 mm. (920 mm. upper end



End view of the 8,000-b.h.p. Doxford turbocharged engine

1,400 mm. lower), the service output being 8,000 b.h.p. at 115 r.p.m., with a mean indicated pressure of 110lb. per sq. in. The engine will be installed in an 18,000-ton d.w.c. tanker being built by Messrs. J. L. Thompson and Sons for Westfal-Larsen and Co., A/S., Bergen, who have ordered a similar turbocharged unit from the North Eastern Marine Engineering Co., Ltd., for a cargo ship; this engine will run shop trials at an early date. As stated, the service output is 8,000 b.h.p. at 115 r.p.m. with a m.i.p. of 110lb. per sq. in., but the maximum rating is 9,000 b.h.p. at the same number of revolutions and a m.i.p. of 125lb. per sq. in. The maximum combustion pressure is in the region of 55 atmospheres (780lb. per sq. in.). The three lever-driven scavenging air pumps have been retained, but these are of smaller dimensions than would be customary, the bore being 1,570 mm. and the stroke 475 mm. We are advised that the retention of these pumps has been considered necessary to ensure good combustion conditions at low powers and to facilitate starting and manœuvring, while they also constitute a safety factor which enables the engine to continue operation in the event of failure of the turboblowers. The displacement of these pumps is substantially less than that necessary for a non-pressure charged engine, hence the mechanical efficiency is increased and this contributes to the boost of 30 per cent. Essentially the engine is of Doxford's standard design with the diaphragm chamber and the lower pistons oil-cooled, distilled water cooling being used for the upper pistons. The exhaust and scavenging air ports have been modified to suit the designed timings. The pulse system is adopted for the two turboblowers, which operate in series with the three scavenging air pumps lever-driven from Nos. 1, 2 and 3 centre connecting rods. As stated, the turboblowers are Brown Boveri units, each with a single-entry turbine and integral blower with the shaft supported by a double ball bearing on the blower end and a roller bearing on the turbine end. At each end of the shaft is mounted a gear type pump arranged to draw from separate oil wells and discharge to the respective bearings. A Brown Boveri air cooler is fitted on each air discharge from the blowers—which draw from within the engine room—to the air receiver mounted at the back of the engine. The air coolers are circulated by sea water from the cooling main. Among the modifications which have had to be made to the engine are an alteration to the design and area of the cylinder ports, while the crankshaft side cranks have been advanced 8 degrees to give the exhaust ports sufficient lead over the air ports. The mean air pressure into the engine is about 7½lb. per sq. in., the average excess of pressure on the air side for scavenging and charging the cylinders being some 3lb. per sq. in. When the exhaust ports close the air remaining in the cylinder is at a pressure of about 6lb. per sq. in. The engine is, of course, arranged to operate on heavy oil and, according to specification, will use fuel of about 1,500 sec. Redwood No. I at 100 deg. F.; alternative connexions will provide for immediate switching over to Diesel oil.—*The Motor Ship, September 1957; Vol. 38, pp. 238-240.*

Patent Specifications

Evaporator for Loeffler Steam Generator

In the Loeffler system the furnace or other source of heat is used to superheat saturated steam, a major portion of the superheated, or primary steam, being used to generate saturated or secondary steam by direct injection in an evaporator, and the remainder delivered as the steam output. The saturated steam leaving the evaporator, consisting of a mixture of the desuperheated primary, and the generated secondary steam, is dried and circulated to the superheater by means of a compressor. In a system delivering steam at, for example, 2,000lb./sq. in. and 1,000 deg. F., it is necessary to recirculate about two-thirds of the superheated steam. The steam flow from the evaporator, therefore, is three times greater than the useful output of the system. For other pressures and temperatures the recirculation ratio may be even less favourable. This is a major disadvantage of the Loeffler system in that a heavy load is imposed on the steam drying equipment which must, therefore, be of a large size compared with that required on a conventional boiler of the same useful output. In an evaporator, in accordance with this invention, the load on the drying equipment is substantially reduced by the use of a surface desuperheater in which the major part of the saturated secondary steam is generated by desuperheating the whole or part of the primary steam. The whole or part of the primary steam, after desuperheating, is led directly to the steam outlet from the evaporator, thus bypassing the steam drying equipment, and the remainder of the primary steam is injected directly into the water. The steam drying equipment thus has to deal only with this mixture of the generated secondary steam and the primary steam injected into the water. In the evaporator shown in Fig. 5, superheated steam at about 1,000 deg. F. is passed through a nest of heat exchange tubes (12) arranged in the lower part of a horizontal evaporator drum (10) on a steam inlet header (15), leaving the tubes at about 650 deg. F. superheat and passing into an intermediate header (13). About nine-tenths of the desuperheated steam is then passed out of the evaporator by way of conduit (14) and recirculated with the dried, generated steam to the superheater of the Loeffler system. The remaining one-tenth of the desuperheated steam is directly injected into the water in the drum (10) through ejector nozzles (16) arranged to circulate the mixture of steam and water through centrifugal primary water separators ((18) and to ensure a flow of water over the indirect heat exchange tubes (12). The generated steam is then passed through chevron driers (20) and withdrawn as dry steam through the steam outlet (22). In the modified arrangement shown in Fig. 6 the steam to be injected into the water is withdrawn at the inlet header (15) and is desuperheated in a separate tube arrangement (24) before reaching the ejector nozzles (16). It is an advantage of these two arranagements that the temperature difference between the nozzle steam and saturation can be reduced to a few degrees which in turn will help to reduce nozzle corrosion. The energy for circulation may alternatively be obtained from the feedwater, or from a circulating pump. In the evaporator shown in Fig. 6, feedwater inlets (26) are arranged to assist the circulation. All or part of the energy required for circulation may be supplied by steam other than primary steam, for example, by steam taken from the discharge of the compressor of the Loeffler system. Circulation may be maintained by making use of density variations of the fluids within the evaporator. This effect is used in the evaporator shown in Fig. 7 which comprises two vessels (30 and 32) one above the other, interconnected by a riser (34) for steam and water and a bifurcated downcomer (36) for water. The lower vessel (30) contains two surface-type desuperheaters (12 and 24) through which superheated primary steam is passed from

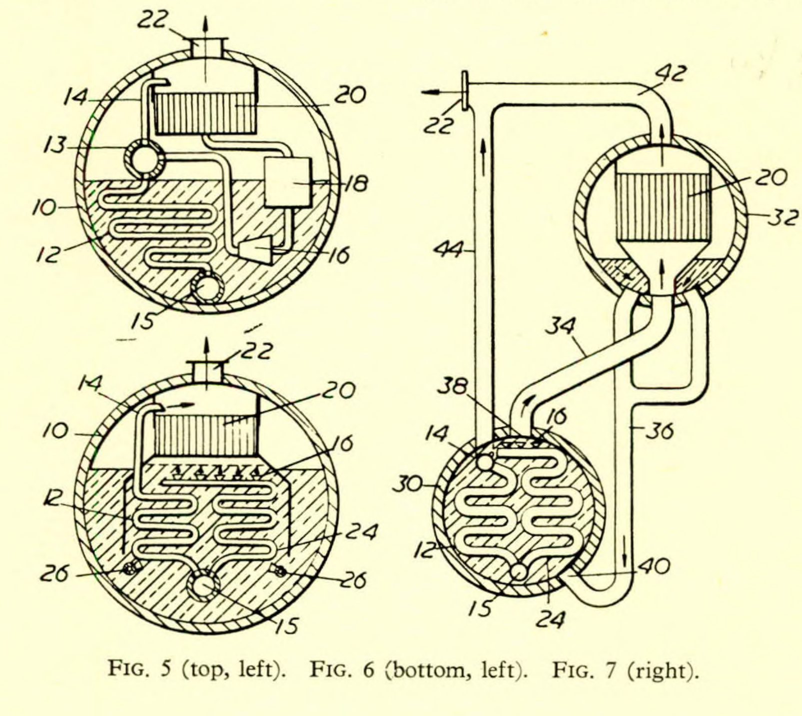
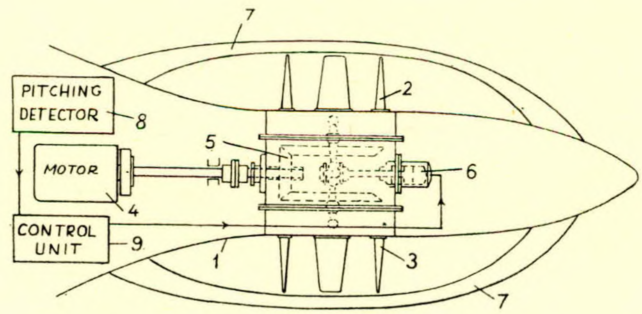


FIG. 5 (top, left). FIG. 6 (bottom, left). FIG. 7 (right).

the inlet header (15). Desuperheated steam from the desuperheater (12) is passed directly to the steam outlet, while that from the other (24) is injected through the nozzles (16) into the water in the lower vessel. The upper vessel (34) contains the steam and water separating equipment (20). The lower vessel has a steam and water outlet (38) at the top and a water inlet (40) at the bottom. The steam and water mixture in the riser (34) is less dense than the separated water in the downcomer (36) and rises spontaneously into the upper vessel (32). The dried steam leaves by a steam pipe (42) while the separated water falls back into the lower vessel. The cooled primary steam from the desuperheater tubes (12) leaves the vessel (30) by a conduit (44) and joins the dried secondary steam at the steam outlet (22).—(*British Patent No. 772,814, issued to Foster Wheeler, Ltd. Complete specification published 7th April 1957.*) *Engineering and Boiler House Review, Vol. 72, July 1957, pp. 252-253.*

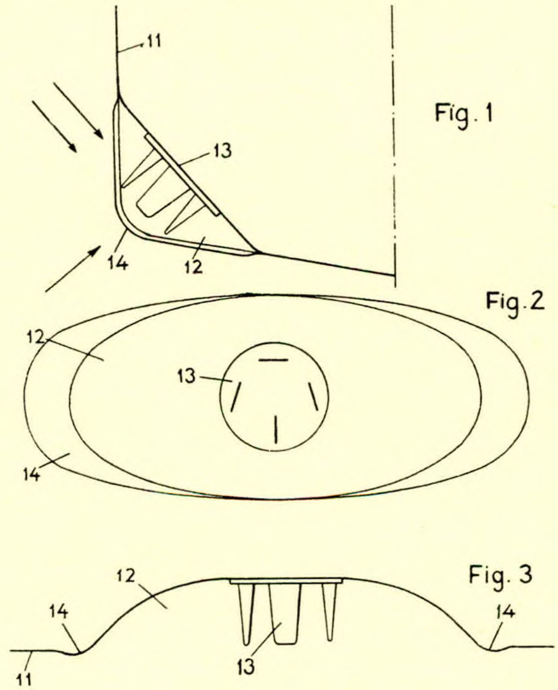
Propulsion System for Ships

It is known that advantages can be achieved in ship propulsion if the propelling means are located at the bow of the ship. There is the disadvantage, however, that due to pitching the bow of the vessel often moves completely out of the water, which not only produces inefficiency but also is liable to produce undue strain on the ship's motor. This disadvantage can be overcome by means of the invention according to which the ship is propelled by blade-wheel propellers mounted to rotate about the same or substantially horizontal axes at opposite sides of the hull and adjacent the bow of the ship. The ship carries means for detecting a pitching movement and means for automatically controlling the blades of the blade wheel propellers in accordance with the response of the detecting means in a manner to cause the blade wheel propellers also to exert a thrust in a direction which counteracts the pitching movement. Thus the anti-pitching control arrangement serves to keep the blade wheel propellers immersed below the surface of the water, thereby improving the efficiency of the propulsion system, whilst at the same time also reducing the pitching of the ship. As shown in the drawing, the bow of the ship is formed with a narrow, forwardly extending portion (1), from opposite sides of which project the blades of two blade wheel propellers (2 and 3). The propellers are rotated at a nearly constant speed by a common driving motor (4) which drives the two propellers through gearing (5). The servomotor (6) controls the inclination of the blades for determining the direction or directions of the thrust. The blades may be protected from damage by guards (7). This ship is also provided with a pitching detecting device (8), the output of which is fed into a control device (9) which automatically controls the operation of the servomotor (6) in a manner to cause the blades to produce also a thrust so as to counteract the pitching. The control (9) is also actuated manually; for example, remotely from the bridge, to control the operation of the servomotor (6) for driving the ship ahead or astern. The automatic part of the control produced by the anti-pitching detector is combined with that part produced manually.—*British Patent No. 780,837, issued to Research Interests, Ltd. Complete specification published 7th August 1957.*



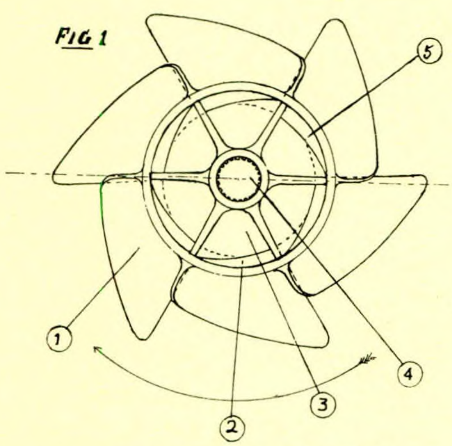
Propulsion System

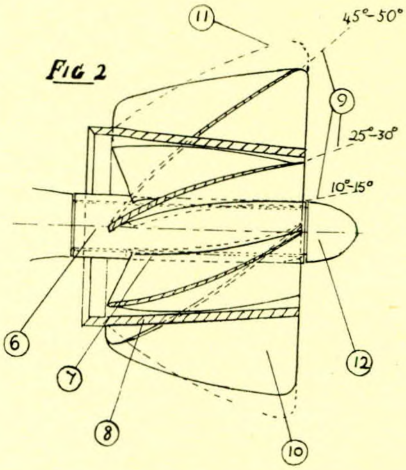
This invention relates to a propulsion system for ships which employs blade wheel propellers or cycloidal propulsion units, such as the Voith-Schneider propeller. Such propellers can be used simultaneously for the propulsion and steering of the ship. The direction of thrust of such propellers can be easily changed to any direction around the axis of rotation of the propeller. According to the invention, driving, steering and anti-rolling of a ship is achieved by means of at least one pair of blade wheel propellers arranged respectively at each side of the ship at the bilge and with their axes inclined both to the horizontal and the vertical and preferably in directions which pass through the longitudinal axis extending through the centre of the ship. The propellers are arranged in dimple-like recesses formed in the external surface of the hull of the ship. According to a feature of the invention at least four propellers are provided, arranged in pairs at each side of the ship, and preferably as far apart as possible in the longitudinal direction of the ship, whereby pitching, as well as rolling, can also be counteracted. As shown in the drawing, Figs. 1-3, the hull (11) is provided at the bilge with a dimple-like recess (12) within which the blades of a blade wheel propeller (13) are arranged. The leading end of the recess and preferably also its trailing end is formed with an outwardly bulging portion (14) to provide the appropriate hydrodynamic shape for the water to flow smoothly, and without vortex, into the recess and through the propeller blades.—*British Patent No. 780,836, issued to Research Interests, Ltd. Complete specification published 7th August 1957.*



Ship's Propeller

The propeller comprises a hollow tapered barrel in which are set around a centre boss six blades at an angle of 10 deg.-15 deg. to 25 deg.-30 deg. on the inside diameter of the tapered barrel. The outer set of six blades are set around the outside circumference of the tapered barrel at an angle of 25 deg.-30 deg. to 45 deg.-50 deg. at the rear edge of the blades and at the outside circumference of its circle of rotation. The whole is designed to give a greater pressurized rearward jet stream of water through combining an inner set of six blades,





nozzled by the tapered barrel, and an outer set of six blades, the roots of the outer blades following along the lines of the tips of the inside set of blades. Fig. 1 shows the external blading (1) on and around the outer circumference of the tapered barrel. Numeral 2 indicates the inside of the tapered barrel. The internal blades (3) are placed around the central boss, suitably splined to receive the shaft (4). Numeral 5 indicates the jet end of the tapered barrel. Fig. 2 shows a side sectional view of the internal blading arrangement, the

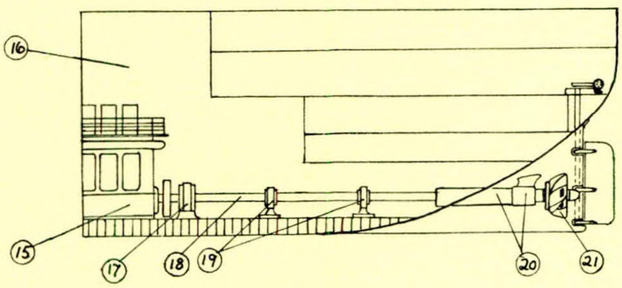


Fig. 4

internal blades (7) being placed at an angle of 10 deg.-15 deg. around the boss (6), and 25 deg.-30 deg. on the inside circumference of the tapered barrel (8). Numeral 10 indicates the outer blading and the full diameter of rotation is shown at 11. Fig. 4 shows the side elevation of the vessel with the Diesel engine (15) inside the hull (16). The magnetic coupling (17) drives the shaft (18) carried in the bearings (19). Numeral 20 indicates the shaft housing and 21 shows the propeller in position.—*British Patent No. 780,910, issued to E. Taylor and R. Shipp. Complete specification published 7th August 1957.*

Hatchway for Ship Hold

The object of the invention is to provide a ship hold arrangement comprising at least two superimposed decks, the hatchway borders of which are offset horizontally with respect to each other so that at least one deck shows alongside the border of its hatchway one or a number of areas constituting an uncovered advance with respect to the hatchways of the upper decks, this arrangement being characterized in that the said displacement of the decks is combined with a vertical raising of the coamings of the displaced decks. The advantage is that it thus becomes possible to load a deck without closing the relevant hatchway, the cargo being lowered directly upon the permanently uncovered zone. Consequently it is possible to load simultaneously a given deck and the adjacent upper or lower deck. Fig. 1 shows a half section of a two-deck ship, illustrating in thick lines the conventional deck arrangement and in dotted lines the new offset deck edge or coaming arrangement according to the invention. The ship has two superimposed decks (1 and 2). In a conventional arrangement the edges (3 and 4) consisting, for example, of coamings, are positioned in a common, substantially vertical plane. In the arrangement proposed the edges are offset horizontally by either

FIG. 1

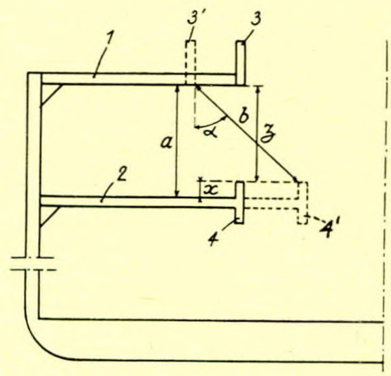
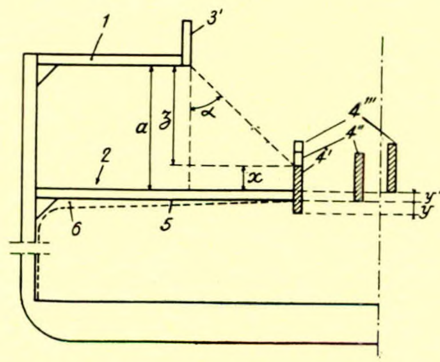


FIG. 2



increasing from 4 to 4′ the overhang of deck 2, or reducing from 3 to 3′ the overhang of deck 1, unless a combination of both is resorted to, as shown in Fig. 2 by way of example. The increase from 4 to 4′ is advantageous in that the fixed loading area of the 'tweendeck is increased proportionally.—*British Patent No. 781,012, issued to P. Legrand. Complete specification published 14th August 1957.*

Marine Engineering and Shipbuilding Abstracts

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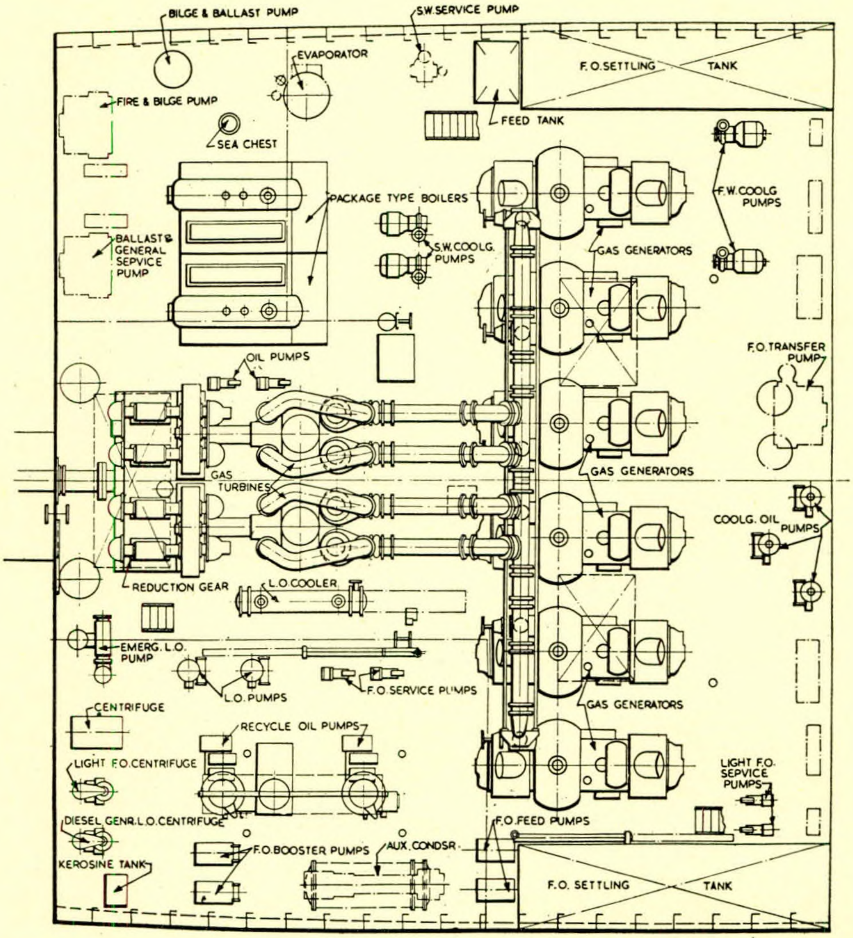
* Patent Specification



Engine room plans of the William Patterson

six generators being supplied to a common manifold, and four pipes are taken from this manifold to the turbine inlets. The turbine has six ahead and two astern stages, and while a gain in efficiency of 1 per cent would have been attainable by the employment of a single-stage astern turbine, the performance when running ahead would have been inferior to that with two rows of blades. It is claimed that the overall thermal efficiency will be 33 per cent for the complete propelling plant when operating on boiler oil. Each gasifier delivers 1,233 gas e.h.p. at a thermal efficiency of 41 per cent, the turbine efficiency being 85 per cent, the reduction gear losses 3 per cent and the piping losses 1½-2 per cent. It is hoped by the designers that as a result of the experiences gained in this vessel it will be possible to make further improvements which will lead to an overall thermal efficiency of 35 per cent, corresponding to a consumption of 180 gr., or 0·4lb. per b.h.p.-hr. with fuel at 10,000 cal./kg. The substantial loss in a normal gas turbine, in which the astern turbine is incorporated in the same casing as the ahead, is well known. In the turbines of the *William Patterson* (as, in fact, with previous free piston engine installations), a masking disc is provided, fitting round the main turbine shaft in the mid-section of the turbine. It is arranged, by means of a hydraulic control system, to move laterally along the shaft so that when ahead movement is desired it is adjacent to the astern wheel gas outlet ring. The flow of gas coming from the ahead wheel is therefore diverted upwards and out of the exhaust outlet and, conversely, with astern operation, the flow is arranged to come from the astern wheels upward to the exhaust. The purpose of this is to reduce the losses due to the rotation of the astern wheel to a figure, it is understood, of about 2 per cent.—*The Motor Ship, July 1957; Vol. 38, pp. 152-154.*

Twin-bank Engine

Blackstone and Co., Ltd., have introduced a new twin-bank series of Lister-Blackstone vertical Diesel engines for marine and industrial duties. The development of the twin-bank series has been based throughout on a complete interchangeability of main working parts with the EV/ER series of two to eight-cylinder in-line engines, which have a maximum power output of 660 b.h.p. at 750 r.p.m. This power range has now been more than doubled by the twin-bank range in twelve and sixteen-cylinder sizes, either turbocharged or normally aspirated, thus enabling high power engine performance to be obtained with the use of small, standard, cylinder components. The twin-bank engines are designated EV12 and EV16 for nominal crankshaft speeds up to 600 r.p.m., while crankshaft speeds over 600 r.p.m. are designated by ER12 or ER16 (cast iron pistons are used for the lower speeds, and aluminium silicon-alloy pistons for the higher speeds). In all cases, a suffix "S" indicates pressure charging. The two vertical banks of cylinders are bolted together back to back, and also to the common engine base, which carries both crankshafts in steel backed, whitemetal lined, bearing shells. The crankshafts are staggered slightly on their longitudinal axis to allow the normal size flywheels to be retained by overlapping. Each bank may be considered to be an individual engine unit, complete with its own governor, running gear and oil and water pumps, the common output drive being taken through a gearbox built in to the flywheel end of the engine. Each crankshaft is fitted with a hydraulic damper at the fore end and a flywheel of the largest practicable proportions at the drive end. An essential feature of the twin-bank drive is that, between each flywheel and input gear to the gearbox there is interposed a Blackstone nodal damper coupling of patent construction. This provides both great flexibility and a high