ENGINEERING ABSTRACTS

Section 3. Shipbuilding and Marine Engineering

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Dutch 18½-knot Passenger-Cargo Liner

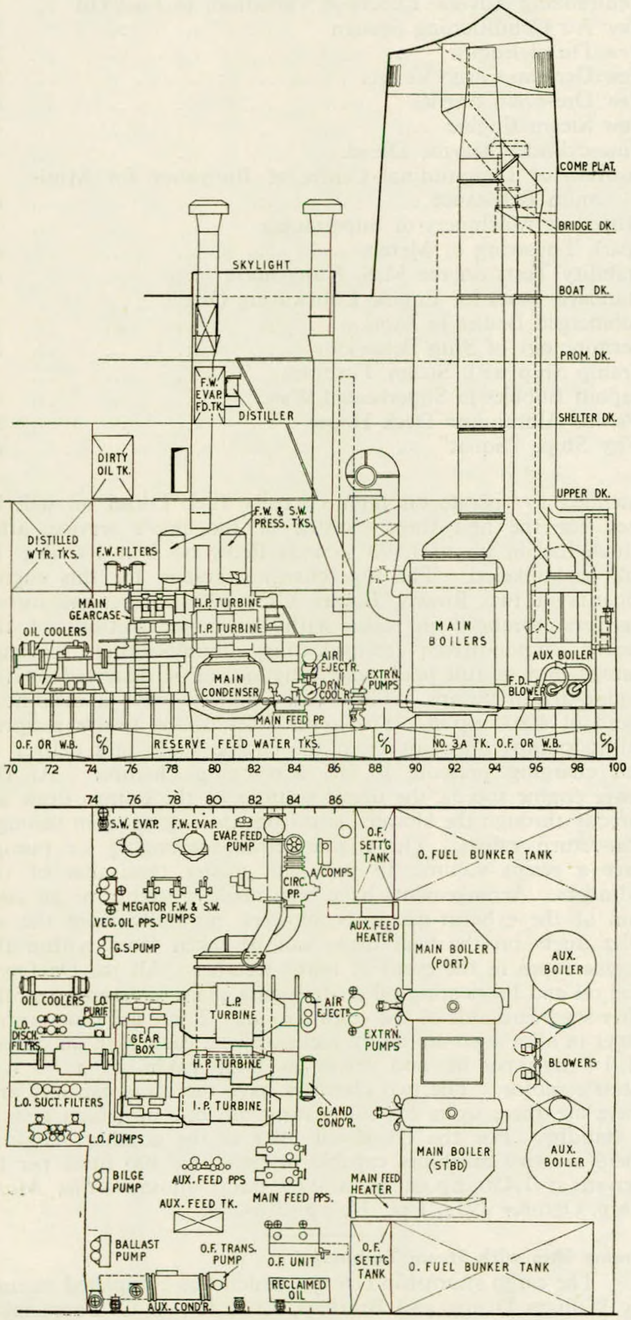
Anticipation of the effect of the St. Lawrence Seaway fosters greater interest in the design and performance of new ships intended for service to Canadian ports, and much scope is given for speculation in the new passenger-cargo liner *Prins Willem Van Oranje.* This highly attractive vessel, of 7,200 tons d.w.c., has accommodation for 60 passengers, and her service speed of 18½ knots is obtained by a 9,600-b.h.p. Werkspoor supercharged two-stroke engine. It has 12 cylinders, each with a bore of 680 mm. and a piston stroke of 1,250 mm., the mechanical efficiency being 88 per cent and the brake mean effective pressure 6·4 kg. per sq. cm., or 91lb. per sq. in. The engine is of the full crosshead design and, being designed to operate on boiler oil, has a division plate interposed between the scavenging air receiver and the crankcase to prevent contamination of the crankcase oil by combustion deposits. The main characteristics of the ship are as follows:—

Length overall	140·790 m. (462ft. 0in.) approximately
Length b.p.	131·065 m. (430ft. 0in.) approximately
Breadth	18·900 m. (62ft. 0in.) approximately
Depth to upper deck	11·125 m. (36ft. 6in.) approximately
Depth to upper 'tween deck	8·840 m. (29ft. 0in.) approximately
Deadweight capacity	7,200 tons of 1,016 kilos.
Corresponding draught	7·874 m. (25ft. 10in.)
Gross register	7,328 tons
Machinery	9,600 b.h.p.
Speed	19 knots

Although the Werkspoor-Lugt main engine is, as stated, intended to operate on high viscosity fuel, Diesel oil will be used for the first three months of the ship's service, after which fuel of up to 3,500 seconds Redwood I at 100 deg. F. will be bunkered. The supercharging system for this engine consists of two Brown, Boveri VTR 630 exhaust-gas turbochargers, arranged in series with the upper sections of the twelve engine-driven reciprocating double-acting scavenge pumps, but at full load practically no air is delivered by the engine-driven pumps. These turbo-blowers have a maximum speed of 6,800 r.p.m. The lower sides of the twelve reciprocating pumps draw air from the engine room and deliver at full charging pressure to the scavenging chamber. At the lower engine speeds, the upper sections of the pumps draw air directly through the blowers, also from the engine room through non-return valves. The engine-driven scavenging air pumps have a swept volume 30 per cent greater than that of the cylinders. Arrangements have been made whereby the air suction of the exhaust-gas turbo-blowers, normally from the air inlet ducts on the boat deck, can be taken from within the engine room in the event of heavy weather. All the De Laval fuel oil and lubricating oil centrifuging plant is arranged at the after starboard side of the engine room. There are nine centrifuges in all. Two are heavy fuel purifiers, each with an output of 1·8 tons per hr. and driven at 1,420 r.p.m. by a 7·7 h.p. electric motor. The two clarifiers have the same capacity and there are three units for lubricating oil, one of which acts as a standby. For the Diesel oil used in the auxiliary engines, there are two machines capable of handling 400 litres per hr. driven at 1,450 r.p.m. by a 0·65 b.h.p. motor.—*The Motor Ship, October 1953, Vol. 34; pp. 276-279.*

Tramp Ship with Steam Turbines

The cargo steamship *George,* which was completed recently by William Denny and Brothers, Ltd., for A. G. Pappadakis, is unusual in being a vessel intended primarily for tramping service propelled by steam turbine machinery. She is also of interest on account of her size, her deadweight being 12,140 tons. This large size explains to some extent the choice of machinery as, in order to obtain a speed of 14 knots, 5,500 shaft horsepower is required, and this power is beyond the normal range of steam reciprocating machinery. It is understood that the choice between steam turbines and Diesel propulsion was influenced by the earlier delivery that could be offered in the case of steam turbines, for which the builders had drawings and patterns already prepared and available for use. Although the *George* appears to be the first tramp ship with steam turbines to be built outside Japan since the war, she will be followed by a number of similar vessels. A sister ship for the same owners was launched recently by William Denny, where a third vessel is also under construction for Greek owners, while at least one ship of this sort is on order on the North-east Coast. Before the war there can have been very few vessels coming into this category, though two examples which might be mentioned are the ill-fated *Hopestar,* which had steam turbines developing 2,000 s.h.p., and the Ropner vessel *Clearpool,* built in 1935 with a 2,200 s.h.p. installation. In both cases these were Parsons turbines. The steam turbine installation develops 5,500 s.h.p. at full load. The turbine set comprises one h.p. ahead, one i.p. ahead and h.p. astern and one l.p. ahead and astern turbine, each driving its own pinion through a flexible coupling. The speed of all three turbines is 1,950 r.p.m. at full power. The blading of the turbines is of the reaction type, end-tightened in the h.p. and i.p. turbines. A two-stage impulse wheel is incorporated in the forward end of the h.p. ahead turbine. The gearing is of the single reduction type with double helical involute teeth of normal form. The ratio is such as to give a propeller speed of 97 r.p.m. The thrust and tunnel blocks are of Michell type. Superheated steam for the main machinery, and the turbo-driven auxiliaries, i.e. two main feed pumps of Weir's make, one main circulating pump of Drysdale's make and two forced draught fans of Howden's make, is supplied by two watertube boilers of Foster Wheeler type at a working pressure of 450lb. per sq. in. at superheater outlet. Superheaters and economizers are fitted to the boilers, the superheated steam temperature being 750 deg. F. The boilers are arranged to burn oil fuel with an open stokehold forced draught system. The other steam-driven auxiliaries are supplied with steam by two Cochran-type boilers at a working pressure of 150lb. per sq. in. These also supply steam for winches and other harbour duties, ship's heating, etc. Auxiliary boilers are arranged to burn oil fuel. The pressure oil burning unit for the main boilers and the low air pressure unit for the auxiliary boilers are supplied by the Wallsend Slipway and Engineering Co., Ltd.—*The Shipping World, 30th September 1953, Vol. 129; pp. 272-273.*



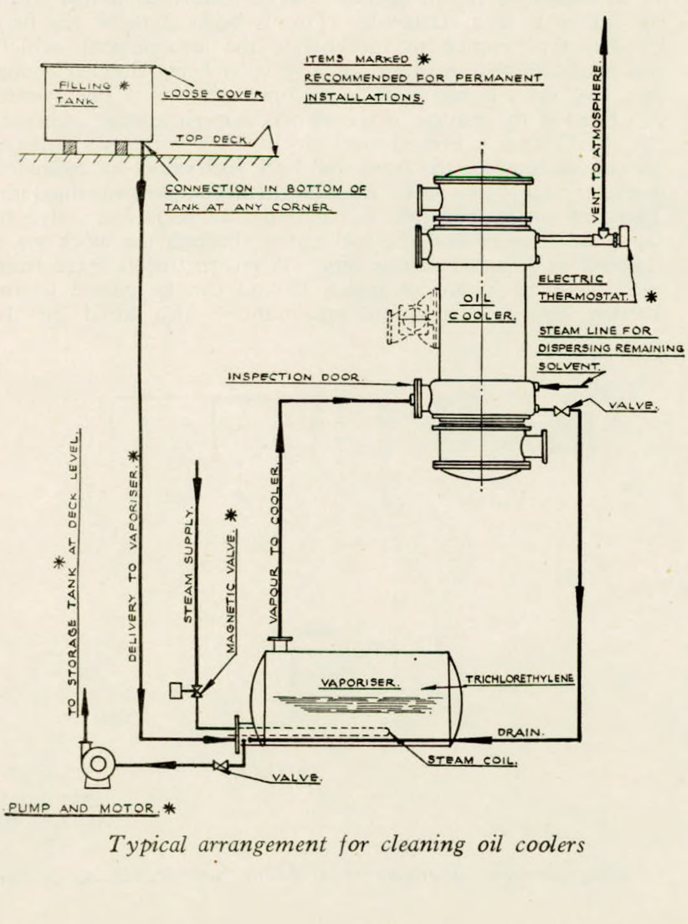
Engine room arrangement of the George

Standard Tests for Engine Lubricating Oils

The test procedures briefly described here and which are used at the Thornton Research Centre have been devised to assess some specific properties of lubricating oils. The properties that have been selected are ring-sticking, piston cleanliness, oil-ring blocking, and corrosion by seawater. Others which might also be included but which are not dealt with here are bearing-corrosion, wear, sludging, piston-ring groove packing, cylinder-bore lacquering, etc. The *Fowler Engine Ring-sticking Test* evaluates H.D. oils with respect to sticking of the piston rings when temperatures in the ring zone are high. The engine is run under constant conditions of moderately high speed and load until ring-sticking occurs. The test is then terminated, and the running time taken as the criterion of the performance of the oil. The performance of the oil is assessed by comparison with that of a straight mineral low-reference oil and an additive-type high-reference oil of MIL-O-2104 standard. The low-reference oil normally gives a ring-sticking time of two hours, and the high-reference oil eleven hours. The performance of the reference oils serves not only as a criterion for the ring-sticking times of other oils, but also as a check on the repeatability of results. In this test ring-sticking can still be obtained with high-ash oils, and a duration of 70 hours has in fact been obtained before ring-sticking occurred. *Gardner Engine Piston-cleanliness Test:* In this test the criterion of oil performance is the quantity of lacquer and carbon deposits formed on the piston skirt under medium-speed, medium-output conditions. At the end of test the piston is given a cleanliness merit rating, based on the quantity and nature of deposit. A straight mineral oil gives a rating of about 4·0, while an oil of MIL-O-2104 standard rates at about 9·0. The test is not, therefore, severe enough for rating oils above this standard. *Crossley Engine Ring-blocking Test:* This test evaluates lubricating oil performance with respect to blocking of the piston scraper rings by carbon deposits. The engine is run under cycling conditions simulating automotive operation, and the test duration is made reasonably short by using a low-cetane high-sulphur distillate fuel which has been found to be especially prone to form deposit in the rings. With a straight mineral oil, blocking is of the order of 60 per cent in the top scraper ring and 30 per cent in the bottom scraper; additive oils give markedly less blocking than this. *National Engine Seawater Corrosion Test:* This evaluates the ability of the lubricating oil to resist corrosion and sludging when seawater is added to the oil, a problem likely to be encountered in marine practice, where seawater is used for cooling the cylinder jackets. The coolant may find its way into the oil storage tanks or engine crankcase, in which case it will be circulated with the oil and may cause severe corrosion of piston rings and skirts, cylinder liners, and other components, and in addition may be responsible for the precipitation of heavy sludge and salt deposits. In the National engine test, the engine is run continuously for thirty hours at medium speed and load, with synthetic seawater added to the lubricating oil at a constant rate throughout the test. The principal criteria of oil performance are corrosion of the piston and other engine components and the extent of sludging in the crankcase and external oil tank. With a straight mineral oil, corrosion is severe and fairly considerable quantities of engine sludge are formed. There are marked differences between various additive oils in effecting improvements in these respects. *Petter Engine LA.3 Piston Cleanliness Test:* This test evaluates the performance of H.D. oils with respect to their tendency to form piston lacquer and carbon deposits at high operating temperatures, under which conditions the fuel plays little part in such formation. The principal criteria of oil performance are the deposits on the piston skirt and undercrown, but deposits in the scraper ring are also taken into account. The piston is inspected and rated at intermediate periods during test, as well as at the end of test, so that the rate of formation of deposits may be studied.—*J. Atkinson and D. Golothan, Journal of The Institute of Petroleum, August 1953, Vol. 39; pp. 504-507.*

Cleaning Heat Exchangers

Although the method of cleaning heat exchanging plant *in situ* by trichlorethylene vapour is no recent development, the demands for greater economy in ship operation and the use of more viscous fuels are leading to increased interest in this system. Many ships, mainly of Continental ownership, have such cleaning plant installed in the engine rooms as permanent fixtures, but the general tendency is to call in concerns specializing in this work and which have portable equipment. The advantage is, of course, that the time and cost of extensive dismantling of condensers, fuel heaters, calorifiers, etc., is greatly reduced. The original patent to the late Mr. Sterry B. Freeman and the rights were subsequently taken over by the manufacturers of the solvent trichlorethylene, Imperial Chemical Industries, who entrusted the commercial development of the process to the Atlas Preservative Co., Ltd., Erith, Kent. A typical arrangement for cleaning an oil cooler is shown in the adjacent diagram; the process is, of course, adaptable and similar apparatus can be used for cleaning many items of engine room equipment. Some units, however, owing to their designs, are unsuited to this method of cleaning. The main requirements are that the solvent vapour can be fed into the bottom of the unit to be cleaned and have free passage to the top. There must be an unrestricted downward flow for condensed solvent and oil, without pocketing or trapping. The vaporizer is almost invariably steam heated and, because of the high input required, is placed lower than the oil cooler, a wide-bore connexion leading to the inspection door at the bottom of the cooler. Alternatively, it may lead to the oil inlet. From the cooler drain point a connexion runs to the bottom of the vaporizer. A clean solvent storage tank is usually placed on the deck in the open. The solvent may be used for cleaning several heat exchangers before being pumped to a drum on deck for reclamation. An electric thermostat should be fitted in a vented adaptor in the cooler air vent, and connected to a magnetic cut-off valve in the steam supply line to the vaporizer. In operation, solvent vapour is fed into a cooler, where it condenses and dissolves oil and grease, and returns through the drain to the vaporizer. This process continues until the entire cooler attains the vapour temperature and cleaning is complete, the vapour then reaching the top thermostat and cutting off the steam supply. The steam and drain valves are next closed and live steam passed through the cooler until no trichlorethylene can be detected in the vent pipe outlet. The cooler is then free from residual solvent. The entire operation should take less than eight hours.—*The Motor Ship, September 1953, Vol. 34; p. 249.*



Typical arrangement for cleaning oil coolers

Neutralizing Adverse Effects of Vanadium in Fuel Oil

The vanadium content of certain fuel oils is known to be the source of considerable trouble in both boiler and gas turbine plants. A recent invention aims at suppressing the attack of the metal by alkali vanadates by the addition to the combustion products, or to the fuel oil itself, of at least one of the following materials: calcium magnesium, barium, strontium, zirconium, or a compound of such materials, other than a silicate. The amount of material added should be at least one half of the vanadium content of the fuel. At high temperatures, these compounds react with the corrosive vanadium compounds to form less active compounds, the melting points of which are higher than that of the vanadium ash contained in the combustion products. The additive materials can be used in the form of inorganic or organic compounds. In general, the organic compounds have the advantage of decomposing at relatively low temperatures, thus releasing the active substance. Of the calcium compounds, the oxide, carbonate, silicide, and carbide are claimed to be suitable; so too are the organic stearate, butyrate, naphthenate, oleate, and oxalate salts. Among other compounds for which good results are claimed are magnesium oxide, magnesium carbonate, barium oxide, and strontium oxide. The various materials mentioned may be introluced into the combustion zone, in the form of a powder or dissolved in a liquid, separately from the fuel, or they may be mixed with the fuel at a point in advance of the combustion zone. Furthermore, a number of the additive materials, or substances containing them, can be introduced separately into the combustion zone.—*The Engineers' Digest, October 1953, Vol. 14; p. 367.*

Laminar Flow About Cylinders in Axial Motion

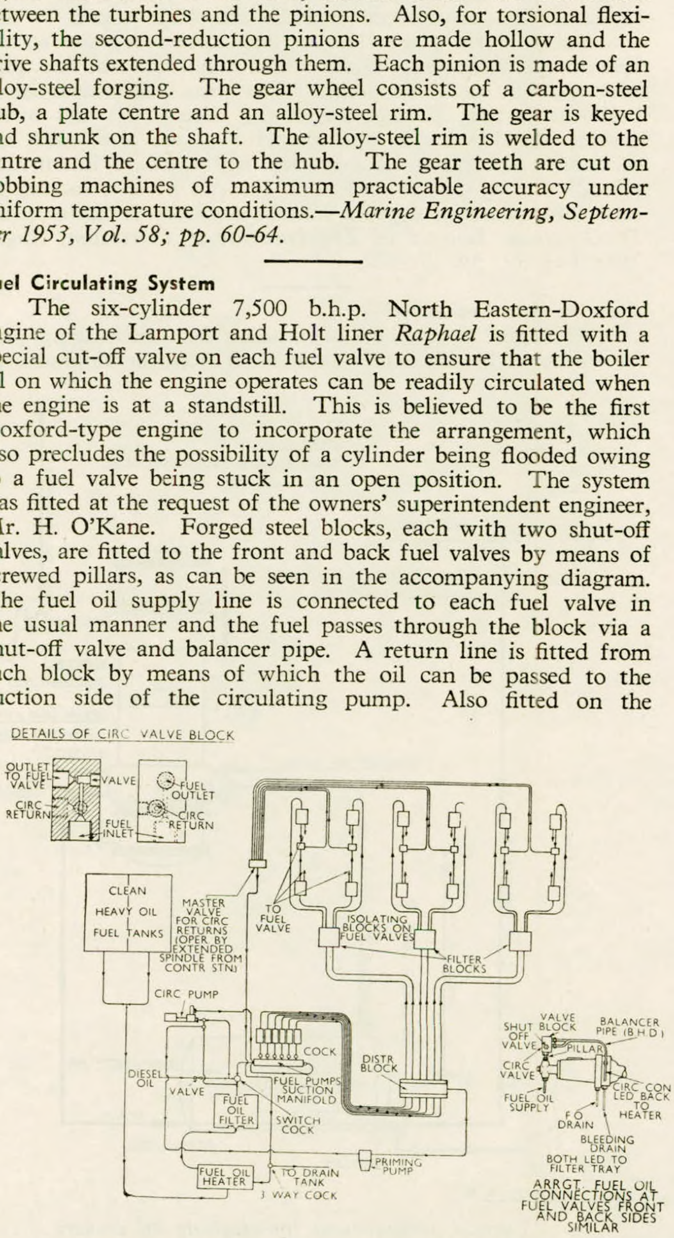
The effect of transverse curvature on laminar boundary layer characteristics has been investigated through the use of an approximate, linearized theory. In the case of a cylinder in steady, axial motion with no external pressure gradient, it is shown that the transverse curvature increases the wall shear stress. A general solution is obtained for the case of arbitrary unsteady motion of a cylinder initially at rest with no external pressure gradient. The particular case of an impulsive start is considered in detail; it is shown that for such motion, the flow field is divided into two distinct régimes, one independent of the time variable, the other independent of the axial spatial variable. The effect of external pressure gradients in the case of steady motion is investigated, and general solutions for a large class of external pressure gradients are obtained. The particular class of Falkner-Skan type pressure gradients are treated in detail. The results of this latter investigation show that the transverse curvature increases the wall shear stress for both favourable and adverse pressure gradients. The increase for favourable gradients is less than the increase for uniform flow; the converse is true for adverse gradients. So-called "similar" velocity profiles are shown to exist for a particular set of flows contained in the class of Falkner-Skan type external pressure gradients, and the ordinary differential equation defining them is derived.—*R. D. Cooper and M. P. Tulin, The David W. Taylor Model Basin, Report 838, April 1953.*

Propelling Machinery of Supertanker

The propelling machinery for the supertanker *Delaware Sun* and sister ships comprises a Westinghouse cross-compound turbine unit consisting of high-pressure and low-pressure turbine elements. Blading for the high-pressure unit is of the combination impulse and reaction type. Ahead blading for the low-pressure unit is all of the reaction type with an astern element of the impulse type. The turbine unit is connected to the propulsion shaft through an articulated, double-reduction gear. Steam conditions are 585lb. per sq. in. at the manœuvring valves with a total temperature of 825 deg. F. and a vacuum of 28·5 inches of mercury. The normal rating is 13,500 s.h.p. at a propeller speed of 100 r.p.m. The maximum rating is 14·850 s.h.p. at 103·2 r.p.m. The total shaft horsepower is equally divided between the high-pressure and low-pressure turbines, which run at 6,310 and 4,230 r.p.m. respectively, at normal power. For ahead power, high-pressure steam from the ahead manœuvring valve enters the high-pressure turbine at the forward end of the cylinder and flows toward the after end. The exhaust from the high-pressure turbine enters the low-pressure turbine in the after end of the cylinder and flows toward the forward end. The exhaust from the low-pressure turbine passes downward directly into the condenser. Part of the ahead steam is under control of three hand-operated nozzle-control valves, which are located in the high-pressure turbine cylinder. In operation, the nozzle-control valves are opened only as required to develop the desired power. In addition to normal operation, either turbine can be operated for ahead power independently of the other, by installing emergency piping. For astern power, high-pressure steam from the astern manœuvring valve enters the astern element in the exhaust end of the low-pressure turbine and flows through the astern blading, which is located in the exhaust end of the cylinder. The steam then passes downward through the main exhaust opening into the condenser. There are four nozzle groups located in the high-pressure turbine cylinder cover. The steam to all of them is controlled by the combined governor and manœuvring valve. Extraction openings are provided and arranged to supply steam at reduced pressure for feedwater heating and other purposes. The complete manœuvring valve system consists of two manually operated valves; one to control steam flow for ahead operation, the other for astern operation. Each is operated by a separate handwheel and is independent of the other. The two valves are contained in a single body and arranged so that only the ahead valve is controlled by the oil-operated overspeed governing piston. The Westinghouse reduction gear which transmits the power of the turbine to the propulsion shaft is of the two-pinion, articulated, double reduction type. Each high-speed pinion is connected to its driving turbine by means of flexible couplings of the internal-gear type. Each first reduction gear is connected to its second reduction pinion by means of a flexible shaft. The propeller thrust is carried, and the gear train is located axially, by a thrust bearing located in the gear housing at the forward end of the gear shaft. The two first-reduction-gear assemblies are mounted in integral extensions of the main gear housing, and the entire unit is supported on a foundation built up from the ship's double-bottom structure. The gear housing is fabricated from welded steel plates. The housing consists of a base, an intermediate section, a main gear cover, two first-reduction-gear covers and two pinion covers. In order to ensure that the pinions will have the necessary freedom to adjust themselves under load, flexible shafts are introduced between the turbines and the pinions. Also, for torsional flexibility, the second-reduction pinions are made hollow and the drive shafts extended through them. Each pinion is made of an alloy-steel forging. The gear wheel consists of a carbon-steel hub, a plate centre and an alloy-steel rim. The gear is keyed and shrunk on the shaft. The alloy-steel rim is welded to the centre and the centre to the hub. The gear teeth are cut on hobbing machines of maximum practicable accuracy under uniform temperature conditions.—*Marine Engineering, September 1953, Vol. 58; pp. 60-64.*

Fuel Circulating System

The six-cylinder 7,500 b.h.p. North Eastern-Doxford engine of the Lamport and Holt liner *Raphael* is fitted with a special cut-off valve on each fuel valve to ensure that the boiler oil on which the engine operates can be readily circulated when the engine is at a standstill. This is believed to be the first Doxford-type engine to incorporate the arrangement, which also precludes the possibility of a cylinder being flooded owing to a fuel valve being stuck in an open position. The system was fitted at the request of the owners' superintendent engineer, Mr. H. O'Kane. Forged steel blocks, each with two shut-off valves, are fitted to the front and back fuel valves by means of screwed pillars, as can be seen in the accompanying diagram. The fuel oil supply line is connected to each fuel valve in the usual manner and the fuel passes through the block via a shut-off valve and balancer pipe. A return line is fitted from each block by means of which the oil can be passed to the suction side of the circulating pump. Also fitted on the



Diagrammatic arrangement of heavy fuel circulating system circulating return line is a master shut-off valve which is operated from the starting platform. To heat and circulate the heavy oil prior to starting the engine, the shut-off valves are closed, and the circulating and master shut-off valve opened. By isolating the fuel valve spindle the danger is obviated of the circulating oil being passed into the cylinder if any fuel valve is slightly off its seat. Such an arrangement enables the heavy oil to be circulated during the stand-by periods and, if an unexpected order is given for an engine movement, the injector shut-off valves can be quickly opened and the master valve shut, leaving the return valves on the blocks to be closed at leisure. With experience showing that carbon formations on fuel valve nozzles are more likely to build up during the manœuvring periods when the fuel oil temperature tends to fall, it is felt that this system will avoid the temperature drop owing to the ease with which the fuel can be circulated without the need to open a large number of valves, besides eliminating the risk of oil being pumped into a cylinder.—*The Motor Ship, September 1953, Vol. 34; p. 261.*

Loud Speaker System for Tankers

The Galbraith loud speaker system, such as installed in the supertanker *Delaware Sun,* was especially developed for installation on supertankers and to overcome a condition that has caused the operators constant trouble and expense. In the process of docking any ship, communication between the bridge and docking stations forward and aft is essential. To communicate with the after station presents no great problem. A loud speaker connected to the pilot house or bridge by conducting cable serves the purpose, but for the forward station something else is required. When under full load conditions and when ploughing through heavy seas the bow of even a supertanker is frequently awash. Any electrical appliance or even electric cable installed adjacent to anchor winches or docking winch is subject to such drastic treatment that its survival as a properly functioning unit is more a matter of luck than of weatherproof construction. The installation of a powerful loud speaker on top of the pilot house with an outgoing voice range of up to 1,000 feet, not only will provide means for carrying audible orders to the forward docking station but to the docks for the handling of lines. This "top of the pilot house" loud speaker acts, as well, as a pick-up microphone, and carries back into the pilot house the response of the crew member carrying out orders of the docking officer. The officer in charge may be within the pilot house or on either end of the bridge. With his portable handset, consisting of a receiver and transmitter, with "press-to-talk" switch, he may talk to both forward and afterdocking stations at once or to either individually. No cable is run forward of the bridge and only a two-conductor cable is required to the after station for two-way talking.—*Marine Engineering, September 1953, Vol. 58; p. 68.*

Evaluation of Engine Lubricating Oils

In attempting to develop the principles underlying the evaluation of an engine lubricating oil, probably the most important concept which must first be accepted is that the oil is an integral material of construction of the engine, and for all purposes must be treated as such. As in the case of any materials of construction, the evaluation of the lubricating oil cannot be considered independently of the conditions under which it has to perform its duties; that is, it must be considered in relation to all factors such as engine design and engine operating conditions. There are then two separate performance patterns (a) that representing the requirements of any particular application (i.e. taking into account the combination of engine-design/operating-conditions); (b) that actually given by the oil being evaluated in order to determine suitability for this particular application. To be satisfactory, the performance pattern (b) must satisfy completely, that is, it must fall outside, or above, the performance pattern (a) representing the requirements. Bearing in mind these concepts, a logical approach to the subject of the evaluation of engine lubricating oils can be developed. First, the requirements in respect of lubricating-oil characteristics for the particular application in mind must be established. In other words, the required performance pattern (a) must be established, and for this it is necessary to have recourse to the field and adequate facilities to observe closely the requirements in respect of the lubricating oil for any given range of application. This is actually one of the functions of "uncontrolled field tests". The next stage is then to evaluate the proposed lubricating oil with respect to each of the pertinent specific properties and thence, by comparing this observed performance pattern with the required performance pattern, to make an assessment of the suitability of the oil for the particular application. Such an assessment makes two basic demands: (i) field experience on which to define the required performance pattern; (ii) ability to devise suitable methods for evaluating each specific property concerned. Simple though these two demands may appear, in practice they present frustratingly complex difficulties. The first difficulty is in identifying and characterizing many of the specific properties involved and in defining the operating conditions which are significant in any particular application. In fact, such difficulties are often virtually insuperable, and for this reason it is often necessary to resort to test engines rather than to glassware apparatus. The latter is always preferable on the score of cheapness, simplicity, and repeatability, but it does necessitate, first, a knowledge of the fundamental phenomena being observed and knowledge of all factors which may influence results. In the absence of such knowledge, the engine allows many of the complex factors encountered in practice (such as combustion, catalysis mechanical effects, etc.) to be introduced, and if the end effects are similar to those encountered in practice (nature of lacquer, deposits, used oil, etc.), there is some measure of confidence that the significant operating factors have been introduced. The test engine must be regarded, however, as a piece of test apparatus, serving the duel rôle of being the means of introducing various pertinent factors, as well as a means of measuring the end effect on the oil and providing a negotiable entity by means of which performance can be expressed; the test engine should not be regarded as a scaled-down version of the engine used in the field or as a means of simulating conditions in the field.—*N. Kendall and L. J. Richards, Journal of The Institute of Petroleum, August 1953, Vol. 39; pp. 502-504.*

Electronics in Construction and Navigation of Ships

Servo control or control by servomechanism is a term which occurs very frequently in current technical literature and one of the earliest servomechanisms was used on board ship—the steering engine. The use of a constant-delivery, electrically-driven hydraulic pump is general, but various systems of controlling the valve gear from the bridge have been used, among them magnetic amplifiers and selsyns. However, manual correction of course error is applied on the bridge by the helmsman. This is a routine task which can be performed very satisfactorily by the coupling of the gyro compass to the steering engine. This has been used with electronic and selsyn coupling units and shown to maintain a better course than a skilled helmsman. In a further development, control of helm can be carried out from a small portable unit located at any convenient point on the ship. While the automatic steering will maintain a fixed course and perform certain evolutions, such as a turn of given radius to a new course, it cannot take account of drift due to tide or wind. A possible development would be to combine the chart from the navigating system with the automatic steering. With this system, the required course would be plotted on the special chart. As the ship progressed, the actual course would be plotted and the helm automatically adjusted to correct for any divergence. The method employed might be a photo-electric scan of the chart with electronic amplifiers and compensating networks feeding the error signal to the steering engine. It does not require a great stretch of imagination to see crewless freighters making a lonely passage from port to port and only deviating from their charted course when their automatic radar gives warning of another ship with the right of way. Such an outcome is technically possible, but whether it is desirable either practically or economically in time of peace is quite another matter. Except for a few electrically propelled ships, there appears to have been no real attempt to control the main engines directly from the bridge. However, with steam turbines operating at high pressures and temperatures and with the gas turbine, it is necessary to keep a continuous and accurate watch on the operating conditions so that corrective action may be taken rapidly when conditions change. It is apparent that there will be great scope in this field for electronic instrumentation with servomechanism control so that eventually large power units may be controlled directly by a simple throttle lever located on the bridge or at any other convenient point. A recent article describes an electronic unit for controlling boiler-water level and monitoring the solids carry-over from the boiler. The unit is claimed to add greatly to the safety of high-temperature steam plant. By actuating motor-controlled valves from the unit output, another stage has been reached in the evolution of automatic power plant. As in all well designed control devices, this unit fails to safety.—*D. S. Fordon, European Shipbuilding, Vol. 2, No. 4, 1953; pp. 84-88.*

Liaaen Variable-pitch Propeller

A variable-pitch propeller and its operating mechanism are shown in Fig. 4. Any sea water entering the hub will, by centrifugal action, tend to accumulate near the periphery and is removed through the lubricating oil return passage (7). The mechanism (3) for turning the propeller blades (2) is operated by a rod (4) carried in the propeller shaft (6), a clearance space being provided for the passage of oil. The shaft rotates in liners (20, 21), which are located at each end of the casing (22). The return passage (7) communicates with grooves (8, 9) in the liners, and passes into an annular chamber (13) between the end

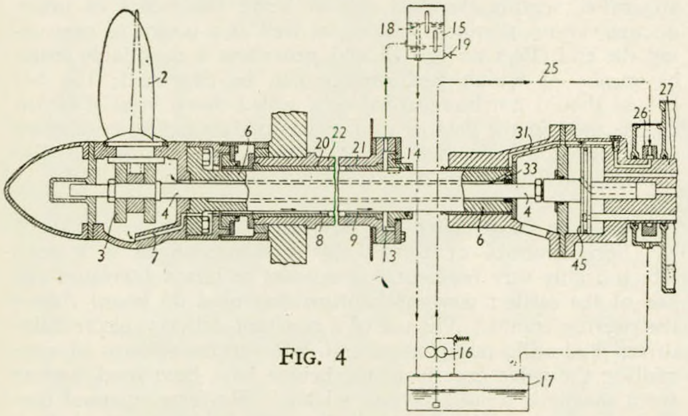


FIG. 4

of the casing (12) and a gland ring (14) on the shaft. A filter chamber (15) for the oil is connected to a reservoir (17), from which the oil is drawn by a pump (16). The chamber (15) serves to separate any sea water which may be present in the oil, the outlet (18) being above the level of the water extraction valve (19). The oil from the pump (16) is discharged through a pipe (25) to an inlet ring (26) on the intermediate shaft (27), which is enlarged to form a cylinder for the servo-motor plunger (45). This operates the rod (4). Oil from the passage (31) passes through ports (33) to the space between the rod (4) and the shaft.—*Patent No. 692,403. N. J. Liaaen, Alesund. The Motor Ship, September 1953, Vol. 34; p. 269.*

Dehumidifier System for Cargo Protection

A new dehumidifier system for cargo protection has been developed by Bethlehem Steel Company, Shipbuilding Division, in conjunction with Surface Combustion Corporation of Toledo, Ohio. Disclosure of the new system followed conclusion of successful tests of the new method on the *Yorkmar,* of the Calmar Steamship Corporation, on a recent voyage from Philadelphia to the West Coast, via the Panama Canal. Termed the "Bethlehem pressure system of cargo hold dehumidification", it dehumidifies the air entering cargo holds and prevents "sweating" of the bulkheads and dripping of this precipitation on to the cargo. Condensation occurs under certain conditions when the temperature of the moist humid air inside the cargo hold is higher than the temperature of the sea water through which the vessel is passing. Sweating resulting from this condition often damages or impairs corrodable or moisture-sensitive cargo. However, if the air is sufficiently dehumidified so that the moisture therein does not precipitate out of the air, condensation is eliminated and atmospheric corrosion becomes negligible. On the *Yorkmar,* moisture is removed from the air by a 7,000-c.f.m. Kathabar dehumidifier manufactured by the Surface Combustion Corporation. Test of the system began early in April when the *Yorkmar* was loaded with 11,000 tons of finished steel in the form of pipe, sheets, strip, cold finished bar stock, structural shapes, rails and wire, much of which was rain drenched. With the system in constant operation, the vessel made the voyage to Long Beach, Cal. The vessel's cargo was inspected there by a delegation of marine surveyors and Bethlehem Pacific Coast Steel Corporation. The delegation found the cargo almost completely free of corrosion from any source. The new system has the advantages of low initial cost, simple operation, minimum maintenance expense and avoids the necessity for expensive instrumentation and trained air-conditioning operating personnel on board.—*Marine Engineering, September 1953, Vol. 58; p. 77.*

Diagnosis of Engine Deposit Problems

Many different types of material can be deposited in the oil system of an internal combustion engine. The different deposit types are usually called sludge, lacquer, carbon, etc., according to their appearance. These terms are unsatisfactory in that they give no indication of the origin or mechanism of formation of the deposit. The term "carbon" does imply that the constitution of the deposit is known, but in fact this is very misleading as the so-called "carbon" formed in engines (for example, on piston undercrowns) sometimes contain as little as 5 per cent *w* of elementary or free carbon and rarely contain more than 40 per cent *w*. A better understanding of these deposits is gained when it is realized that all of them are mixtures of constituents produced by any or all of the following mechanisms: (1) incomplete combustion of fuel; (2) incomplete combustion lubricant; (3) oxidative polymerization of fuel; (4) oxidative polymerization of lubricant. Deposits formed from (1) and (2) are soots and have a high (Ca 70 per cent) free carbon content. Those from (3) and (4) represent a variety of products varying over a wide range of molecular weights, solubilities, and physical properties, i.e. from sticky cylinder-bore lacquers to hard piston undercrown deposits. The transition from soft, readily soluble, to hard highly insoluble material is most probably a further polymerization, and usually results from prolonged baking of the deposit. Two-stroke Diesel engines are liable to heavy deposition in exhaust ports. These port deposits appear to be of two types, depending on whether or not the lubricant has played a major part in deposit formation. The two types, which are identical in appearance, are: (a) deposits consisting of about 70 per cent *w* of soot, almost certainly from the fuel, the remaining 30 per cent *w* being polymerization material also formed chiefly from the fuel; (b) deposits containing less than 40 per cent *w* of soot, the main precursor of the polymerized material being the lubricant. It is probable that deposit (b) is actually a mixture of deposit (a) and large amounts of lubricant polymers. If, in investigation of a field problem of exhaust-port blocking, the deposit is found to be of the first type (i.e. formed almost entirely from the fuel), then the fundamental solution of the problem is to improve combustion efficiency. The lubricating oil property most likely to reduce deposition is obviously detergency, but it may be that insufficient oil reaches the exhaust port for detergent additives to have any pronounced effect. The piston deposits which are most likely to call for the use of an H.D. lubricant are those formed in piston ring grooves. These again are of two types: (a) Deposits consisting chiefly of combustion products (soot) bound together by small quantities of polymeric material; (b) Deposits containing relatively little soot and consisting largely of polymers to which fuel and lubricant both contribute. Fuel usually plays the major part. The formation of both types of deposit can be considerably reduced by using a detergent oil. In crankcase deposits fuel polymers are uncommon and three types of deposit occur. These are: (a) those consisting chiefly of combustion products; (b) polymeric material resulting from oil oxidation; (c) cold sludge. This last deposit, the frequently occurring mayonnaise-type sludge, is in the main an emulsion of water and oil, and is rarely met within the field of H.D. lubricants. Type (a) is similar to the type (a) ring-groove deposit discussed earlier, and requires the same remedy. In seeking to develop oils to overcome the deposition of lacquer-like material formed from oil oxidation and polymerization (type (b)) it is worth while considering the probable steps by which such a deposit gets into the engine. First, the oil is oxidized to primary oxidation products. These products then polymerize to form the deposit, which is then laid down on engine parts. (These last two steps are probably not separable, as the polymerization step may perhaps occur only in the proximity of metal.) If this hypothesis is correct, then deposition could be prevented either by improved anti-oxidant characteristics to interfere with the first step, by the incorporation of some additive which would inhibit polymerization, or by preventing the material, once formed, from depositing on engine parts. Consequently, the problem is similar to that of exhaust-port deposits. However, the solution to the problem should be considerably easier, because crankcase deposits are formed at much lower temperatures.—*J. Hughes and C. R. Major, Journal of The Institute of Petroleum, August 1953, Vol. 39; pp. 513-518.*

Marine Refrigerating Installations

The principal uses of refrigeration aboard ships are: air conditioning, +25 to +30 deg. C. (77 to 86 deg. F.); refrigeration and transport of bananas, +12 deg. C. (53·6 deg. F.); refrigeration and transport of tropical and citrus, +6 deg. C. (43 deg. F.); transport of fruit and vegetables, 0 deg. C., +2 deg. C. (32, 35·6 deg. F.); refrigeration and transport of fresh meat, 0 deg. C., −2 deg. C. (32, 28·4 deg. F.); transport of frozen meats, −10, −15 deg. C. (14, 5 deg. F.); transport of frozen fish, −15, −20 deg. C. (5, −4 deg. F.); shipboard freezing of fish, −20, −30 deg. C. (−4, —22 deg. F.); transport of ship's stores (from +4 to −10 deg. C.: 39·2 to 14 deg. F.). Regarding the transport of goods one definite improvement in the evolution of technique was the introduction of ventilation particularly for the transport of frozen meats. Fluids used have been CO_2, ammonia and since 1945 Freon-12 (the latter has the advantages of ammonia without having any smell). Brine has been widely used with CO_2 and ammonia. However, direct expansion has also been used, especially after 1930, for banana freighters in which refrigerating power was 300,000 to 400,000 f/h (100 to 133 tons). Direct expansion for the transport of frozen meat at −10 deg. C. (14 deg. F.) is used in British practice. With Freon-12 arrangements are varied: direct expansion and automatism for ship's stores, brine for larger installations. Direct expansion is gaining ground; for installations of a capacity above 500,000 f/h (166 tons) brine requires equipment which is heavier (140 per cent), taking up more space (50 per cent), more expensive (40 per cent) and requires more power (20 per cent). The trend is now to use automatic regulators together with bypass manual regulators. In most cases compressors are driven by D.C. motors (110 and 220 volts); A.C. motors are now coming into use with a voltage of 380 volts. V-belt drive are the more often used. The present trend is to use multi-cylinder compressors, either V, W, VV or radial type; weight is gained (7 kg. per 1,000 f/h in 1951 as against 30 kg. in 1915) as well as space (in 1951 a W compressor 750 r.p.m. took up 0·45 sq. m. per 100,000 f/h; in 1915 a horizontal compressor revolving at 110 r.p.m. took up 5 sq. m. per 100,000 f/h). Devices to vary capacity are very important in marine installations, especially in banana freighters. Regulation is obtained by varying the speed of the compressors, by varying space to be cooled, by bypass during compression, derivation at compression, controlled opening of the valves. Condensers are studied to reduce space (advantages of Freon). Corrosion is also being checked. Brine tanks have been of the land type with agitating propellers; now, dome multitubular evaporators are used. With Freon-12 either multitubular evaporators or V type exchangers in cylindrical calenders are used, Freon evaporating inside the tubes. Cooling units have changed from the long serpentine type to the horizontal tubes, the vertical tubes, and finally the cross-grids. The trend is now to use finned tubes. The fans which used to be of the centrifugal type are now of the helicoidal type with variable and reversible speed. The ventilation of banana freighters has been studied at length; the present trend is to use vertical ventilation (60 volumes per hour) by dividing each room horizontally into two parts. Defrosting is still a delicate problem; it is done with hot gases or by trickling of water or a non-freeze liquid. The steamer *Foch* has air conditioned rooms (dining room, lounge) cold rooms (stores 230 cu. m.; freight, 576 cu. m.). Refrigerating equipment includes 4 Freon-12 compressors (360,000 effective f/h (120 tons) for air conditioning, 100,000 effective f/h (33 tons) for stores and freight). Cold is distributed by brine.—*L. Dumolin, Abstract in Bulletin de l'Institut International du Froid, Vol. 33, No. 4, 1953; pp. 790, 792.*

Nine-cylinder Marine Diesel

British Polar Engines, Ltd., of Glasgow, have recently added a nine-cylinder marine engine to their 340 mm. bore by 570 mm. stroke, M-series of engines at 300 r.p.m. The design incorporates a tandem double-acting opposed-piston scavenge air pump at the free end of the engine as with other engines of this type. The test bed fuel consumption is given 0·370lb. per b.h.p. per hour at full and three-quarter loads and 0·382lb. at half load. A close study of the engine and shafting characteristics has shown that, fitted amidships only the fifth and fourth order, first node, criticals require investigation as the ninth order major is below the normal slow running speed of the engine (70 r.p.m.). If the engine is situated aft, then the ninth order critical is inside the running speed range, but the speed range can be rendered free from disturbing critical speeds by a suitable selection of firing order, flywheel inertia and shafting diameter.—*Gas and Oil Power, October 1953, Vol. 48, p. 232.*

Marine Engineering as Applied to Small Vessels

This paper supplies design criteria for commercial and government vessels up to nominal small lengths in the order of 125 feet. It is quite obvious that the limitation on length is, to a degree, meaningless as the horsepower of an installation to a large degree is independent of its physical size. It possibly would be better to put limitations on horsepower, but the basic concept of the paper is to outline a type of construction and design in between large vessel practice and motor boat practice. Considering present-day cost of propulsion plant, the author believes that the only advantageous cooling system to put in a vessel is one in which fresh water is used in the engine. Not only fresh water but treated fresh water, in which soluble oils, chromates or other corrosion inhibitors under controlled conditions are used in the water system. The fresh water system circulates through the engine by means of the circulating pump, which should have capacities to maintain low temperature differentials through the engine. The engine designer has complete control of this aspect and the vessel designer has but little choice and must accept the standard pump furnished by the engine builder. The fresh water can be cooled by salt water taken from the sea by a number of methods, namely, heat exchangers, both tube and shell and flat plate types, keel coolers or skin coolers, internal and external. The skin cooling on steel vessels is quite simple if it is incorporated in the design of the vessel with the cooling channels used as part of the structural components of the vessel. It is normal to obtain heat transfer coefficients of about 55 B.Th.U. per sq. ft. per deg. F. when using steel internal or external structural members and using velocities of the fresh water through the system of 2 to 4 f.p.s. Higher velocities are not normally indicated as the normal pump supplied by the engine manufacturer has insufficient head to drive it at high velocities. The normal installation is a heat exchanger of the tube and shell or plate type. The plate type has advantages of space and weight although generally it is not reputed to have the same degree of ruggedness as the tube and shell type. For vessels of the class under discussion, the installation of a heat exchanger external from the engine and of the tube and shell type using ⅜-in. tubes is the normal installation and extremely satisfactory. A typical diagram of the cooling system which has been proved satisfactory over a number of years is indicated in Fig. 10. In a system such as

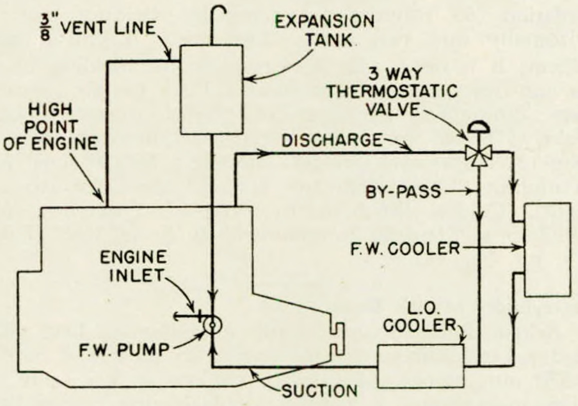


FIG. 10—*Cooling system diagram*

this, with external thermostatic valves, highly satisfactory operation of the cooling system is obtained. The so-called automotive thermostatic valves usually give a great deal of trouble and are not indicated for the usual marine installation. They can result in poor temperature control, early failure and other annoying difficulties. The modulating three-way thermostatic balanced valve is definitely indicated. The selection of the heat exchanger is usually left to the engine manufacturer. It is well to make a heat balance and a check selection based on the heat exchanger manufacturer's published information as to the capacity of the heat exchanger insofar as cooling of the engine is concerned. This avoids the possibility of not only too small a heat exchanger, but also the possibility of too large a heat exchanger, which is difficult to install, requiring more weight, as well as costing a great deal more. Many installations have heat exchangers entirely too large because everyone along the line has added a "margin". The heat balance for the engine is quite simple to make from percentage heat balances. Attention is called to the fact that the heat balance of a supercharged engine is considerably different from the heat balance of a naturally aspirated engine. This is obvious when given consideration due to the fact that the heat exchange surface in the engine has not changed. The temperatures have not appreciably changed and yet the power has been appreciably increased. Therefore, percentagewise there is far less heat rejected to the jackets in a turbo or supercharged engine than there is in a naturally aspirated engine. The preparation of a heat balance also is an excellent method for checking the results of trial trips by measuring temperature differentials across the heat exchangers when the vessel is first put in service. It will be noted in Fig. 10 that all the fresh water is indicated as going through the lubricating oil cooler. This is highly advantageous in order to stabilize the lubricating oil temperatures, not only cooling the oil but heating the oil on occasions. If the lubricating oil cooler is in the heat exchanger line before mixing is obtained, prior to returning to the engine, then the lubricating oil temperature falls off appreciably at low powers. This is a condition which can cause trouble in the maintenance of the engines and is definitely not recommended, although it is installed many times to maintain low lubricating oil temperatures.—*Paper by P. G. Tomalin, read at the Annual Meeting of the Society of Naval Architects and Marine Engineers, New York, 12th-13th November 1953.*

Liberty Ship Conversion Scheme

Plans for the re-engining of Liberty ships to give them greater speed have been drawn up on many occasions, and a number of ships of this type have in fact been fitted with Diesel engines which gave them an increase of speed of about two knots. This, of course, was to increase the economic efficiency of the ships, but the Americans are more concerned to increase the value in wartime of the vast numbers of Liberty ships which they still hold in their reserve fleets at various ports. Plans originating in the United States therefore normally postulate a considerable increase of speed, and the latest, which was put forward by Mr. Hugh Gallagher (president of the U.S. Propeller Club) to a Senate sub-committee, calls for an increase of speed to 18 knots. Naturally this could not be achieved without alterations to the existing hull form, but a scheme for conversion has been drawn up at the request of the committee by George G. Sharp, Inc., in which a satisfactory hull form could be achieved without altering the hull abaft the after end of No. 2 hatch. Forward of this, the lines would be fined considerably, and the waterline length increased by 34ft., reducing the block coefficient from 0·76 to 0·69. A sea speed of 18 knots could then be achieved with machinery developing 8,500 s.h.p. (the existing machinery develops 2,500 i.h.p.) on a draught of 26ft. 6in., which is a foot less than the present draught. On this basis, the existing 10,800-tons deadweight capacity is reduced to 9,415 tons. The cost of this conversion at present prices would be some 2¼ million dollars, or £800,000.—*The Shipping World, 18th November 1953, Vol. 129, p. 411.*

Geared Cargo Ship

On her return to London a short time ago, the *Otaki* had completed a maiden voyage of over 23,000 miles to New Zealand and back via the Panama Canal. She is the highest-powered geared Diesel-engined cargo ship yet constructed and the fourth of a series of geared oil-engined refrigerated ships built and building for the New Zealand Shipping Co., Ltd. Two, the *Cornwall* and *Surrey,* have Sulzer engines of a design that has been adopted for some time, but the third ship, the *Middlesex,* is equipped with two 10-cylinder engines of an entirely new Sulzer type. The *Otaki,* the fourth of the series, was built by John Brown and Co., Ltd., and has 12-cylinder Brown-Sulzer engines of the same design as in the *Middlesex,* the normal output being 10,300 b.h.p. in service, with a trial trip output of 12,000 b.h.p. They are arranged to operate on boiler oil. The new ship is larger than the previous vessels and the two in course of completion. These will have 10-cylinder engines. The main details of the *Otaki* are given in the following table.

Length, overall	525 feet
Length, b.p.	490 feet
Beam	70 feet
Depth	44ft. 6in.
Deadweight capacity	12.000 tons
Machinery (service)	10.300 b.h.p.
Service speed, loaded	16½ knots
Cargo capacity (refrigerated)	476.300 cu. ft.
Cargo capacity (general)	186,900 cu. ft.

She has six cargo holds, all designed to carry insulated cargo except the upper 'tween decks, which load general cargo. In holds Nos. 3 and 4 chilled meat can be carried in port and starboard compartments. The forward and after peaks accommodate water ballast or fuel oil and No. 1 double-bottom tank, fresh water or water ballast. Nos. 2, 3, 4 and 5 double-bottom tanks are arranged for oil fuel or water ballast, except the forward tank in No. 2 compartment, which is intended only for fresh water. Steel hatch covers are provided to all hatches. The two engines drive a single propeller through B.T.-H. magnetic couplings and gearing at a service speed of 117 r.p.m., the reduction being 2·23 to 1. The cylinders have a diameter of 580 mm. and a piston stroke of 760 mm., the mean effective pressure (brake) being about 4·5 kg. per sq. cm., or 64lb. per sq. in. This type of engine, which is specially designed to operate on boiler oil, is arranged so that the cylinder is separated from the lower part of the engine by a division plate containing a stuffing box. The fuel pumps are fitted at the level of the cylinder heads so as to reduce the length of the high-pressure piping between the pump and injection valve, and injection timing can be modified whilst the engine is running. This is of assistance in view of the likelihood of taking on different grades of oil at various bunkering stations. For each cylinder there is an oscillating exhaust valve which is actuated by eccentrics from the camshaft and closes the exhaust ports at the end of the scavenge period. In this way a certain degree of after-charge is obtained. This camshaft also actuates the eccentrics for the cylinder lubricators, the cam sleeve for the pilot starting air valves, the indicator eccentrics and the safety governor. It is chain-driven, the chain being tensioned in such a way that it does not affect the timing. When reversing the engine the exhaust valve drive passes through an angle of 70 degrees towards the new direction of rotation, and this is effected by means of a reversing coupling through which the camshaft is driven. Each piston is in two parts comprising the forged steel head, which is oil-cooled, and the cast-iron skirt. The cylinder liners are sealed at the top with cast steel covers and the jackets are fresh-water cooled. The bedplates and columns, and other parts of this engine, are fabricated. In normal service the scavenging air pressure is about 3·5lb. per sq. in. and the starting air pressure 425lb. per sq. in. The back pressure in the exhaust line is from 12 to 20 inches water gauge and the exhaust gas temperature at the normal output is about 650 deg. F. The scavenge pumps, one for each cylinder, form two blocks and each is driven from the crosshead by a rigid arm which drives the pump piston rod. There is a safety governor which cuts off the fuel if the engine runs at a higher speed than 270 r.p.m. One of the engines is arranged so that the exhaust gases may either pass through a Cochran composite boiler or direct to the silencer in the funnel, whilst the other engine exhausts to the silencer. In this boiler 3,750lb. of steam per hour may be raised at sea by gas alone at a pressure of 100lb. per sq. in. or 3,600lb. of steam by oil firing.—*The Motor Ship, November 1953, Vol. 34, pp. 342-348.*

Engines-aft Shaw Savill Liner to be Air-conditioned

The new 20,000 tons *Tourist-de-Luxe*-class Shaw Savill Liner now under construction at Belfast will have accommodation for 1,200 passengers. The machinery will be installed in the after part of the vessel in order to locate a much higher proportion of cabin accommodation than usual in the midship section, which is the least susceptible to motion in a seaway. All cabins, both inside and outside will be air-conditioned by an up-to-date system which will ensure maximum comfort irrespective of external atmospheric temperatures. In order to ensure uniformity, the ship is divided into zones in which average uniform temperatures and humidities can be readily maintained. Apart from the general zoning, ship's-side cabins are more susceptible to variations in temperature owing to the effect of sun and wind, and provision will be made to regulate conditions in these cabins separately from the inside rooms which are not normally subject to fluctuations caused by the weather. Passengers frequently desire to control the conditions in which they live and it will be possible for them to raise or lower the temperatures in the cabins by regulation of the intake of conditioned air. The air-conditioned accommodation is spread over five decks and the comprehensive system consists of 27 air-conditioning units placed in suitable positions. Thermotank, Ltd., are responsible for this important sub-contract. The capacity of the refrigerating plant involved will be about 9,000.000 B.Th.U. per hr. and is much greater than that in any other British passenger vessel in service or coming into service in the near future. The new vessel will not carry any cargo and strict sailing dates can therefore be maintained from all ports of call. Denny-Brown ship stabilizers will be incorporated in the hull and the service speed will be 20 knots.—*The Marine Engineer and Naval Architect, November 1953, Vol. 76, p. 451.*

Corrosion Hazards from D.C. Welding Currents

Fig. 3 illustrates a frequent source of corrosion from D.C. welding operations on two or more structures, using a common D.C. source. A large generator located on shore supplies a

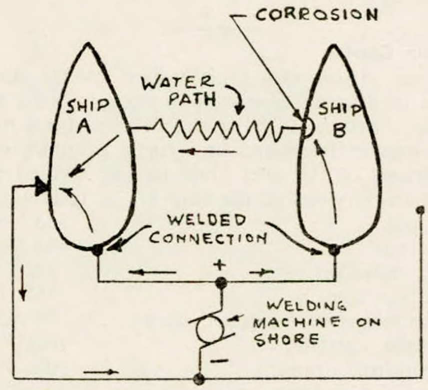


FIG. 3

D.C. bus for welding on ships. If a poor contact exists between the positive bus and Ship "B", the welding currents on Ship "B" will discharge from Ship "A", causing rapid destruction of the paint and severe pitting on Ship "A". In Fig. 4 the

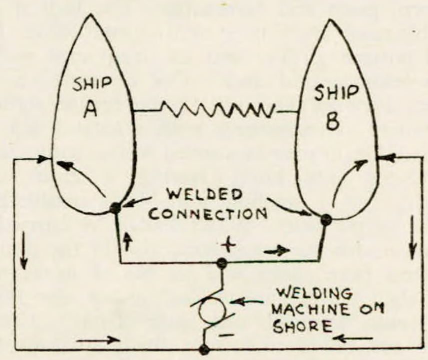


FIG. 4

positive grounding connexions are welded to the ship to ensure perfect connexions. This greatly reduces corrosion currents but does not completely eliminate them. Assume, for example, that welding operations are in progress on Ship "A" only, as illustrated in Fig. 5. The IR drop in the positive feeder from

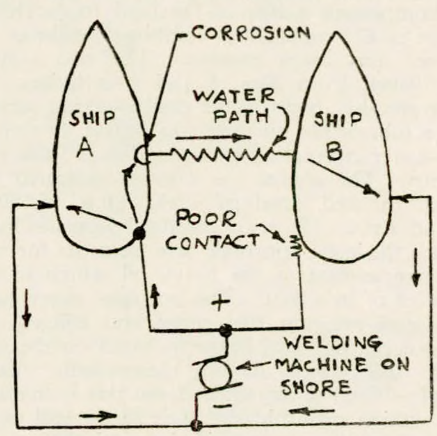


FIG. 5

the generator to Ship "A" causes current to discharge from Ship "B" and collect on Ship "A" due to the parallel path provided by the sea water. When it is considered that the resistance-to-earth of a ship in sea water may range from a few hundredths to only one thousandth of an ohm, it is evident that large D.C. bus conductors must be provided in situations of this type to mimimize corrosion from stray currents.—*H. W. Wahlquist, Corrosion, November 1953; Vol. 9; p. 1.*

New Ore-grain Carrier

The *River Afton* was built under special survey to the highest class of Lloyd's Register, to comply with the requirements for classification 100A1 with freeboard for a full scantling ship having tanker freeboard and for a draught of 30ft. 9in. from the bottom of the keel plate having welded flush butts. The main characteristics of the ship are as follows: —

Length o.a.	513 feet
Length b.p.	485 feet
Breadth, moulded	69ft. 3in.
Breadth, extreme	69ft. 6in.
Depth moulded to freeboard deck ...	39 feet
Deadweight capacity	16,550 tons
Corresponding draught	30ft. 9in.
Full loaded displacement	23,060 tons (approximately)
Gross register	11,557·7 tons
Block coefficient	0·7725 (approximately)

As with modern ore carriers, the machinery is arranged aft, and the ship has a straight raked stem of rolled steel plate, a cruiser stern, poop and forecastle. The hull is constructed on the combination framing system; longitudinal framing for the top and bottom girders and the transverse system for the sides and skeleton second deck. Ore is carried in the central compartments forward and aft of the bridge structure, there being seven such compartments with a total grain capacity of 343,944 cu. ft. Grain may be carried in the wing compartments as shown, these side holds having a grain capacity of 336,824 cu. ft. No. 1 ore hold and No. 1 double-bottom tank extend to the ship's side. Water ballast is carried in central compartments under the ore spaces, also in the double bottom, side spaces and peak tanks and in No. 4 grain holds. The butts of the shell plating are welded, as are also the bulkheads in the ore holds, and oil and water tanks. Care has been taken in the ore holds to ensure flush surfaces to facilitate ore-handling whilst the transverse bulkheads on frames 61, 97 and 138 in the ore holds are partial to enable bulldozers to work the double holds. The bulkheads stop at a height of about 10 feet from the tank tops. Strengthening has been provided in way of the panting area from forward of frame 156, and the shell plating is increased in thickness by 10 per cent. The main and auxiliary machinery has been supplied and installed by David Rowan and Co., Ltd., the propelling machinery comprising a Rowan-Doxford four-cylinder engine having a bore of 670 mm. and a combined stroke of 2,320 mm. for the upper and lower pistons. The two scavenging air pumps are driven from Nos. 1 and 2 cylinders. From the main engine are also driven three double-acting service pumps, these being a lubricating oil pump to deliver 45 tons per hr., a 270-ton sea-water circulating pump and a 200-ton jacket water cooling pump. The engine has a mean indicated pressure of 88lb. per sq. in. and develops 4,500 b.h.p. (5,150 i.h.p.) in service at 116 r.p.m. It is of standard Rowan-Doxford construction with the latest approved arrangements for maintaining the correct temperature of the heavy oil which is to be used, up to the point of injection. This includes steam heating coils around the high-pressure fuel pipes and efficient insulation. As is now general, one fuel heater is fitted on the suction side of the H.P. fuel pumps, having thermostatic steam control. An electrically-driven pump draws from this heater and delivers to the H.P. suction manifold and then to the fuel pump bodies, at the top of which are provided circulating outlet valves. Arrangements are also made for circulating oil through the fuel valve bodies prior to starting the engine. Provision has been made to enable the engine to run on either boiler or Diesel oil, and, although part of the acceptance trials were run on fuel of 1,500 sec. Redwood No. 1 at 100 deg. F.,Diesel oil only will be used for the first two or three months' service. For heavy oil purification there are two De Laval type 1900 purifiers, and two clarifiers, all with stainless-steel bowls and designed to purify three tons of 1,500 sec. Red. No. 1 fuel per hour. Sludge extracted by the Diesel oil and heavy fuel centrifuges is passed to a sludge tank with connexions through which the sludge flows by gravity from each machine.—*The Motor Ship, December 1953, Vol. 34, pp. 382-387.*

Committee for Sea Pollution

The International Chamber of Shipping has agreed that an international technical committee should be formed to examine the extent to which other countries may adopt the recent recommendation of the British Ministry of Transport for the prevention of pollution of the sea by oil. It is proposed to study the question of "prohibitive zones", the provision of shore facilities for receiving oil residues and other matters, in anticipation of the international conference of maritime governments which the British Government proposes to convene.—*The Motor Ship, December 1953, Vol. 34, p. 378.*

Stability Tests on the M.S. *Kungsholm*

A special premium has been paid by the Swedish-America Line to the N.V. Kon. Mij. De Schelde for the fulfilment of the special stability requirements for the new 22,000-ton passenger liner *Kungsholm.* It was stipulated that the ship had to have the same stability under all practical conditions of cargo, fuel, fresh water, etc. This was achieved by the provision of unusually large ballast tanks, in addition to the normal fresh water and fuel tanks. With the large-capacity evaporator plant and by the use of these tanks, it is possible to make ready compensation for all changes in cargo, quantities of fuel, water and other stores.—*The Motor Ship, December 1953, Vol. 34, p. 378.*

New Steam Engine

A new reciprocating steam engine has been developed recently by the Elsinore Shipbuilding and Engineering Company (Helsingor Skibsvaerft og Maskinbyggeri, A/S), of Elsinore, Denmark. The first three engines have been installed in three passenger vessels built by the company for Finnish owners, Finska Angfartygs A.B. A licence to construct the engine has been taken out by Scheeps Installatiebedrijf "Nederland", N.V., of Ijsselmonde, Holland, and three engines are being constructed by this firm for installation in ships building in Holland for the same owners. The first of these ships, the *Pallas,* has just been completed by Werf Jan Smit Czn. of Alblasserdam. The Elsinore engine is of quadruple expansion type, and as with the rather similar Götaverken engine, it employs an exhaust-driven turbo-compressor to raise the pressure (and the temperature) of the steam between stages. The two high pressure and the two low pressure cylinders are coupled in pairs on the Woolf principle, by which the steam passes directly from one cylinder to the other without any intermediate receiver, the outlet valve for the first cylinder of the pair acting equally as the inlet valve for the second. Cylindrical poppet valves are used in all cylinders. Between the two pairs the steam passes through the compressor. The turbo-compressor is mounted at the side of the engine on top of the condenser, where it is conveniently situated between the two pairs of cylinders. The exhaust turbine is of three-stage axial-flow type, while the compressor is centrifugal. The unit rotates at 6,000 r.p.m. The direction of rotation of the compressor is, of course, independent of that of the main engine, so that it can be used during manœuvring. Control of the compressor is effected by means of a servo motor mounted on the turbine, which can be controlled from the manœuvring platform. If the correct speed is exceeded, the turbine is automatically cut out by means of a bypass connecting the two pairs of cylinders. The engine itself is of heavy construction, to meet the requirements of navigation in ice: in the case of the crankshaft the strength is some 25 per cent above that specified by Lloyd's Register. The bedplate is of all-welded construction, and the guides are of locomotive type with water cooling. The diameters of the cylinders are 460 mm., 680 mm., 780 mm. and 1,350 mm. respectively, the stroke being 1,100 mm. Supplied with steam at 290lb. per sq. in. and 590 deg. F., the engine develops 3,400 i.h.p. at 113 r.p.m. Steam consumption is low. A figure of 7·25lb. of steam per i.h.p.-hour is quoted for the main engine, equivalent to 0·6lb. of oil per i.h.p.-hour with oil of 18,000 B.Th.U. per lb. for the main engine and auxiliaries. The engine has Klug's valve motion gear and Brown's reversing gear, the latter consisting of a steam-driven servo motor which makes the reversing frame follow the movements of a lever. The reversing of the engine from full ahead to full astern takes only a few seconds. Relatively late cut-offs are employed (60 per cent in the h.p. cylinder at normal output), giving the engine smooth running qualities and good manœuvring ability. The low pressure ejector is of three-stage type, and is operated by exhaust steam of only 28·5lb. per sq. in. pressure. This ejector is capable of maintaining a vacuum of 96 to 97 per cent in the condenser with a seawater temperature of 15 deg. C. The exhaust from the ejector is utilized for preheating the feed water, as is the remainder of the exhaust steam.—*World Shipbuilding, November 1953, Vol. 3, p. 150.*

New German Cargo Vessels

The Deutsche Dampfschiffahrtsgesellschaft Hansa recently placed in service the cargo motor vessel *Rolandseck,* which has the following particulars:—

Length o.a.	71·10 m.
Length b.p.	64·00 m.
Deadweight	1,303 tons
Propulsive power	1,380 b.h.p.

The ship is of the Maier-Form type and is largely welded. The propulsion plant consists in a seven-cylinder M.A.N. four-stroke cycle engine with pressure charging developing 1,380 b.h.p. when 60 per cent pressure charged. Fuel consumption is 166 grams per b.h.p. The auxiliary plant includes an hydraulic pump installation for operating the hydraulic cargo gear and the hydraulic anchor spill, etc. A 35-kW dynamo is driven by V-belt from the propeller shaft through a friction clutch. A second similar vessel is nearing completion.—*Hansa, 12th September 1953, Vol. 90; pp. 1544-1546.*

Large Screw Propeller for Tanker

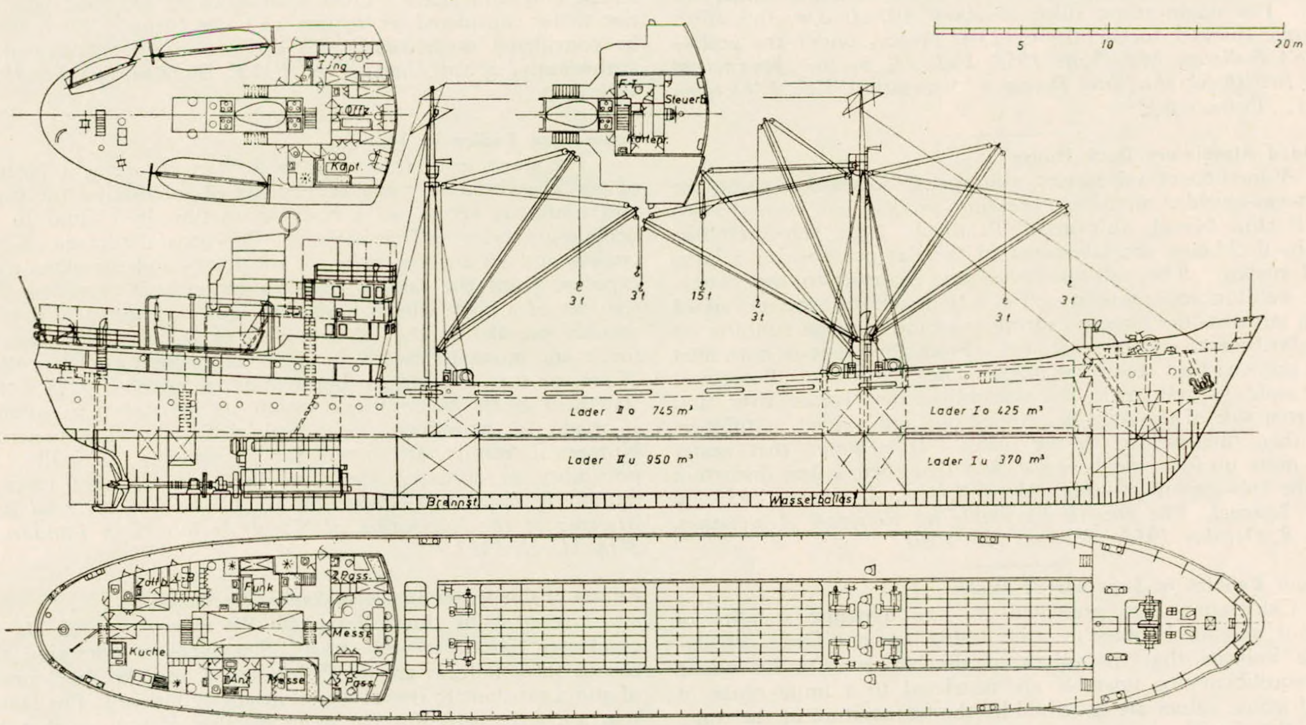
The largest cargo-ship propeller ever made in the United States is now being completed at Bethlehem Steel Company's Staten Island propeller plant. This five-bladed propeller has a diameter of 22ft. 6in., pitch 17ft. 4in., hub diameter 54in., and hub length 3ft. 10in. The finished weight will be about 70,000lb. The propeller is intended for a 45,000-ton tanker now under construction at one of Bethlehem's shipyards for the World Tankers Company. When completed, this single-screw vessel will be the largest cargo ship ever constructed in the United States. She will have an overall length of 737 feet, moulded breadth of 102 feet, moulded depth of 50 feet, and a service speed of about 16 knots.—*Naut. Gaz., September 1953, Vol. 148, p. 24. Journal, The British Shipbuilding Research Association, Vol. 8, November 1953; Abstract No. 8211.*

Modern Irish Fishing Boats

After a brief review of the Irish fishing boats of the late 19th century and of the early years of this century, the author goes on to discuss in detail the craft which have been built within the last fifteen years. These are mainly boats with the following principal characteristics:—

Length o.a.	50 feet
Beam, extreme	15·5 feet
Draught, forward	3·75 feet
Draught, aft	5·5 feet
Displacement	28·4 tons
Midship area	32·7 sq. ft.
Prismatic coefficient	0·652

These vessels are suitable for drift-net, long-line, and trawl fishing. The draught is limited because of shallow harbours. The propelling machinery usually consists of a 4-cylinder, 4-stroke Kelvin Diesel developing 88 h.p. at 750 r.p.m. A full speed of nine knots is attained. The internal arrangement is unusual in that the engine is placed forward, the fish-hold amidships, and the accommodation aft. This arrangement is shown to have several advantages, the main disadvantage being the long length of propeller shaft. It is stated that



m.v. Rolandseck

realignment of the shaft and engine is usually required after the first 12-18 months' service, after which no further trouble is experienced. Details of a similar boat used only for seine fishing are also given. These vessels are all fitted with echo sounders, and those more recently built with radio-telephones. A new larger type of boat is now being built for seine-net and trawling only. The chief characteristics are:—

Length o.a.	60 feet
Beam, extreme	18 feet
Draught, forward	5·25 feet
Draught, aft	7·5 feet
Displacement	57 tons
Midship area	55·8 sq. ft.
Prismatic coefficient	0·637

It has been possible to take advantage of larger draught in this design, and the boats should show a considerable improvement over the shallower boats. The machinery consists of either a 3-cylinder 2-stroke Alpha Diesel of 135 h.p. at 450 r.p.m. driving a two-bladed controllable-pitch propeller, or a 6-cylinder 4-stroke Gardner Diesel of 114 h.p. at 900 r.p.m. The latter will drive a solid 3-bladed propeller through a 3 to 1 reduction gear. Echo sounder and radio-telephone will be installed. Plans of these vessels and of the other described in the paper are given. The author ends with some notes on the methods of construction used in these Irish fishing boats.—*Paper by J. Tyrrell, read at the International Fishing Boat Congress, Paris, 12th-16th October 1953, and Miami, 16th-20th November 1953. Journal, The British Shipbuilding Research Association, Vol. 8, October 1953; Abstract No. 8092.*

Why Ships "Squat"

This article reviews investigations made at the U.S. Navy Model Basin and elsewhere into the increase of draught which is observed when ships steam in a narrow channel in shallow water. It is shown that the passage of the vessel causes a depression of the water level in the water immediately surrounding the ship; the ship as a result "squats" or floats at a level below the level of the undisturbed water. Relations between the "squat" and speed are shown, the rate of increase of squat becoming greater at higher speeds. In general, the squat for any given speed varies inversely as the depth of water under the keel. It is also shown that for a given power the speed of a vessel decreases with a decrease of depth under the keel. For ocean-going ships of about 10,000 d.w. this effect is quite marked for depths of 25ft. or less under the keel.—*Ship's Bulletin, May/June 1953, Vol. 33, p. 16. Journal of The British Shipbuilding Research Association, Vol. 8, October 1953; Abstract 8035.*

Welded Aluminium Deck House

A method of fabricating aluminium deckhouses using the inert-gas-shielded metal-arc welding process has been devised by Boston Naval Shipyard. Relatively large sub-assemblies of the deckhouse are fabricated in the flat position on a large steel platen. The sub-assemblies are clamped to the platen. The welding apparatus, which is a light-weight type, is carried on a straight-line oxygen cutting machine carriage running on standard tracks suitably mounted. From this position both fillet and grooved butt welds are made. The face side of all grooved butt welds is first made, the assembly is then turned over, and the root side of the joint is chipped out and welded. Stiffeners are then fillet-welded to the plates. It is stated that neater and more uniform welds result, and that there is less distortion of the sub-assembly.—*Bur. Ships. J. Vol. 2, August 1953; p. 35. Journal, The British Shipbuilding Research Association, Vol. 8, October 1953; Abstract No. 8060.*

Vapour Bubbles in Superheated Water

Calculations are presented for the dynamic stability of vapour and air bubbles in superheated water. These calculations indicate that the values of the bubble radii for which the equilibrium is unstable are restricted to a finite range of radii whose values are governed by the temperature of the water and the initial air content in the bubble. Two theoretical solutions for the rate of growth of these unstable bubbles are considered: (a) Solution of the equation of motion of the bubble radius with the assumption that there is no heat diffusion across the bubble wall, (b) solution which includes the effect of heat diffusion. The two solutions differ appreciably. These two solutions are then compared with the experimental data on the growth of the vapour bubbles in superheated water. This comparison shows agreement with the solution with the effect of heat diffusion included.—*Paper by P. Dergarabedian of U.S. Naval Ordnance Test Station, read at the 1953 A.S.M.E. Semi-Annual Meeting. Paper No. 53-SA-10.*

Terminology of Ship Behaviour

The Hull Structure Committee of the Society of Naval Architects and Marine Engineers (U.S.A.). in its study of vibrations of ships, has found that there is considerable confusion in the usage of the words slamming, pounding, whipping and panting. The committee has given considerable thought to definitions of these words and suggests the following definitions: Slamming is that phenomenon whereby a sharp transient loading, not unlike a shock load, occurs anywhere along the ship below the wave surface, but more predominantly under the forefoot, resulting in rapid changes in hydrodynamic pressure of extremely high magnitude acting on a localized area. Slamming appears to be associated with heaving and ship's forward motion rather than with pitching. Slams can occur when the forefoot is not out of water and also at various pitch angles such as bow up, bow down and even keel. Pounding is that phenomenon whereby the forward end of the ship, out of water, crashes into the sea, particularly during severe pitching and when bow is light. The effect is that of an impact load applied over a considerable area. Whipping is the reaction of the ship to a suddenly applied transient load where the ship as a whole vibrates so that a pronounced (whipping) motion occurs in parts of the ship remote from the point of applied load. The term "whipping" is usually used in connexion with vibratory motion when the loading is extremely severe, as in the case of an underwater explosion. Whipping action may also be caused by a slam, a heavy sea, etc. Panting is the flexing of plate panels and framing due to a variable load applied normal to the panel surface. The term "panting" is principally used in connexion with severe and varying hydrostatic loading at the bow and stern. These definitions as recorded here are not to be considered as definite and conclusive. They are to be considered as tentative and are not to be interpreted as authoritative at this time.—*S.N.A.M.E. Bulletin, October 1953; Vol. 8, p. 15.*

Submerged Bodies in Motion

The purpose of this paper is to state the present position of our knowledge of possible methods of calculating the forces and moments acting on a body in motion in a fluid in the neighbourhood of a free surface. The general problem is discussed, and an analysis given of the forces and moments to be expected from the various physical causes. It is shown that the case of a body with a horizontal axis of symmetry, moving steadily parallel to that axis, allows of calculation of all the forces and moments except the frictional resistance. The method of solving this problem mathematically is described with a clear statement of the assumptions which are made, but no attempt is made to reproduce the actual mathematical expressions. Numerical results are given for a spheroid. Finally, the possibility of applying the same methods to other cases of motion is discussed.—*Paper by W. C. S. Wigley, read at a Meeting of the Institution of Naval Architects in London on 27th March 1953.*

Failure of Condenser and Heat Exchanger Tubes

Up to about thirty years ago the expectation of life for condenser tubes was very short. The types of corrosion that caused these failures are briefly discussed and the development of alloys resistant to these types of attack is outlined. The factors that are responsible for the small number of failures that still occur are listed and discussed. Foremost amongst the reasons for the failure of tubes at the present time is the use of polluted cooling waters, which are particularly liable to affect aluminium brass tubes, though all materials are liable to suffer in greater or lesser degree. Partial obstructions are another important cause of trouble. The advantages and disadvantages of installing protector blocks in water boxes are discussed. Consideration is given to the possible effects of increasing nominal water speeds in condenser tubes above those commonly in use. The relative merits of aluminium brass and cupro-nickel are discussed from the point of view of heat transfer rates. Amongst steps that can be taken to minimize condenser-tube troubles, the importance is stressed of avoiding polluted waters in the early days of service of a ship.—*Paper by P. T. Gilbert, read at a meeting of the Institute of Marine Engineers, 10th November 1953.*

Gas Turbine Set for Oil Tanker *Auris*

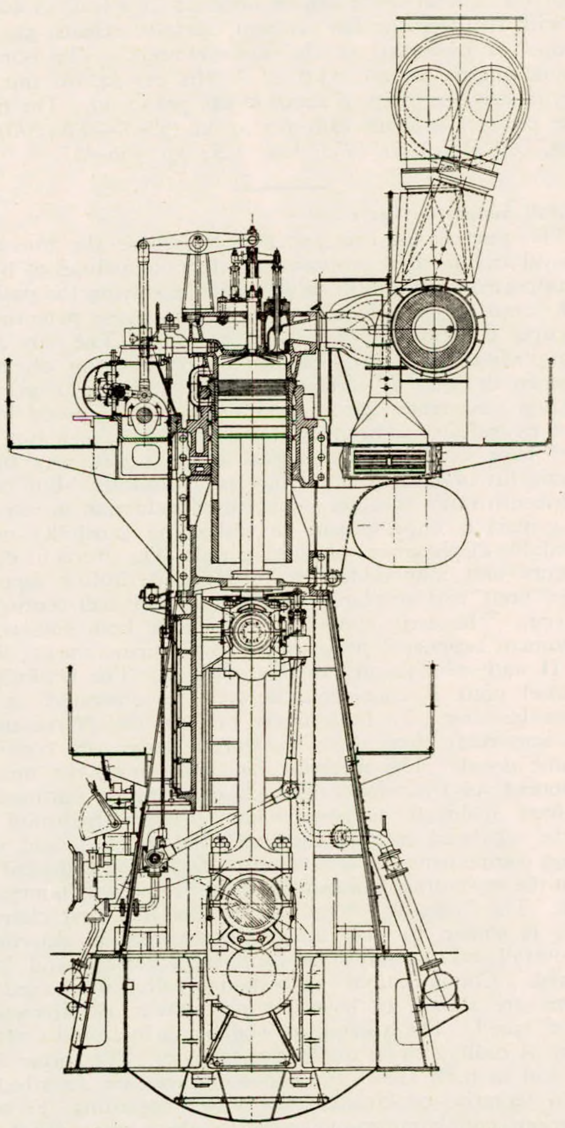
In order to obtain operating experience with a main propulsion gas turbine under service conditions at sea, the Diesel electric oil tanker *Auris* has been equipped with a gas-turbine set which replaces one of the four Diesel engines originally installed. A description is given of the set which operates on the simple open cycle with a heat exchanger and independent (low pressure) power turbine. The set thus has one compressor and two turbines in series, the high-pressure turbine driving the compressor and the low-pressure turbine driving the alternator which supplies power to the propeller motor. In the design the emphasis was laid on high component efficiencies which necessitated the use of multi-stage uncooled turbines. A moderately high initial gas-temperature was adopted in order to ensure adequate length of life. The experience during test operation, which covered a running time of 680 hours, is fully recorded, and complete performance figures for all conditions are given. The set was operated with some success on heavy (residual) fuel oil. The total running time on heavy fuel oil was 363 hours, which included a continuous run of 270 hours, mainly at full load.—*Paper by B. E. G. Forsling, read at a General Meeting of the Institution of Mechanical Engineers, 27th November 1953.*

Internal Stresses in Forgings

The standard equations for the thermal stresses in a cylinder during cooling are used to obtain heating and cooling rates for various sizes, corresponding to certain values of the stresses. The procedure adopted for dealing with the transformation in large forgings is described. Relaxation at high temperatures is considered, and estimates made of the residual stresses likely to be present after simple tempering and annealing treatments. Residual stress measurements on large ingots indicate such estimates are on the safe side. More complicated heat-treatment operations are then discussed. In forgings which are normalized but not tempered, and in hardened steel rolls, high residual stresses may be obtained, particularly in the case of water-quenched rolls where the stresses sometimes give rise to spontaneous brittle fracture. Relaxation at relatively low tempering temperatures was studied by stress measurements on quenched cylinders of CCr steel.—*Paper by C. Sykes, read at the 22nd Andrew Laing Lecture before the North-East Coast Institution of Engineers and Shipbuilders, 30th October 1953.*

New Diesel Engine

A new Stork Diesel engine was demonstrated for the first time in the workshops of the Koninklijke Machinefabriek Gebr. Stork and Co. N.V. at Hengelo. Designed for running on heavy fuel oil, the engine is a single-acting two-stroke Diesel with the uniflow scavenging system and equipped with exhaust gas turbo-blower superchargers. The engine, which was demonstrated on the test bed, is to be installed on completion in the vessel *Ouderkerk*, now building by the Netherlands Dock and Shipbuilding Co., for the Verenigde Nederlandsche Scheepvaart Mij. With supercharging the engine is rated at 8,500 b.h.p. at 115 r.p.m. and without supercharging develops an output of 7,200 b.h.p. at 115 r.p.m. Scavenge air is supplied by four turbo-blowers of Brown Boveri manufacture, type VTR500. As there is a considerable demand for engines with outputs up to about 10,000 b.h.p., suitable for operation with heavy fuel oil, and having exhaust gas temperatures enabling a reasonable steam production in an exhaust gas boiler; it was decided to develop a new engine type satisfying these requirements. The most suitable type for this purpose is the single-acting two-stroke engine with uniflow scavenging system, in which the scavenging air is admitted through ports at the bottom side of the cylinder, while the exhaust gases are discharged by means of exhaust valves at the top side of the cylinder. With this system the scavenging air does not change its direction and the exhaust gases, still present in the cylinder when scavenging starts, are almost completely and practically without any mixing driven out by the incoming column of scavenging air. Only a small quantity of excess air is required to ensure an air charge which is not contaminated by exhaust gas. It was found that supercharging on the exhaust pulse principle enabled a considerable saving of energy as compared with supercharging on the equal pressure principle. In order to make this saving of energy as high as possible, the exhaust valves must open quickly. This is obtained by fitting four small poppet-type exhaust valves instead of one large valve. As each of these four small valves has about half the diameter of one large valve, the lift required to give full passage to the gases is only half of one large valve. Tests carried out with a turbo-blower with dimensions that were based on the experience acquired, confirmed these results.



Sectional elevation of new type of Stork marine Diesel engine

As will be seen from the accompanying cross-sectional drawing, the shape of the combustion chamber is, in principle, identical to that of the normal Stork-Hesselman engine. By the application of four exhaust valves the central position of the fuel valve could be maintained. As with former Stork engines, the bedplate, columns and cylinder block are held together by means of throughgoing tie-rods. This allows manufacture of the bedplate and columns in either cast iron or welded steel. The new engine has eight cylinders with a bore of 750 mm. and a stroke of 1,500 mm. The engine without supercharging has a rating of 7,200 b.h.p. at 115 r.p.m. In the sister ship of the *Ouderkerk*, also to be built by N.D.S.M., a supercharged engine will be installed, having an output of 8,500 b.h.p. at 115 r.p.m. The engine with the piston type scavenging air pump has a weight of about 450 tons, the engine with turbo-blower weighs about 425 tons. So the weights per b.h.p. are 137½lb. per b.h.p. for the engine without supercharging, and 110lb. per b.h.p. for the supercharged engine. The fuel consumption at full load is about 0·34lb. per b.h.p. without supercharging, and about 5 per cent lower for the supercharged engine. When idling, the engine maintains running at about 18 r.p.m., which is considerably lower than required on board a vessel. The turbo-blowers are dimensioned in such a way that, when operating the engine at full load, the speed will only be 85 per cent of the allowable maximum. The engine can be manœuvred with one turbo-blower out of use. In this condition about 85 per cent of the normal speed can be obtained at a load in accordance with the propeller law without excessive exhaust gas temperatures or overspeed of the turbo-blowers. The normally aspirated engine has an m.i.p. of 75·3lb. per sq. in. and with supercharging the m.i.p. is about 89½lb. per sq. in. The turbo-blower pressure is about 19lb. per sq. in. (abs.).—*The Shipping World, 2nd December 1953, Vol. 129; pp. 466-467.*

Hydrofoil Supported Craft

The purpose of this paper is to arouse the interest of the naval architectural profession in the potentialities of hydrofoil supported craft and to enlist its aid in solving the problems which stand in the way of fully achieving these potentialities. The paper is essentially expository in nature. The only claims to originality lie in the manner of presentation of known fundamentals and in certain conclusions drawn from them regarding advantages gained and limitations imposed by the use of hydrofoils. The hydrofoil is described as a hull supported clear of the water surface while under way by the dynamic lift of underwater wings, or hydrofoils. For certain speed-length ratios it offers a substantial reduction in resistance and a marked improvement in seakeeping capability over a comparable displacement or planing craft. The efforts of various inventors and shipbuilders to produce hydrofoil supported surface craft and seaplanes during the past half-century are reviewed. The early sporadic efforts have been followed by government supported programmes in Germany during World War II and subsequently in this country. The feasibility of hydrofoil craft is considered as well demonstrated, at least for smaller sizes. To indicate the present state of the art, the most important elements of hydrofoil design are considered in some detail. The methods for determining the principal components of hydrofoil resistance, or drag, are outlined. In a fashion analogous to aerodynamic practice, hydrofoil drag may be separated into profile, parasitic, induced, and wave-making components. A characteristic feature of hydrofoil craft is that the wave drag coefficient decreases rapidly with increasing speed. The "take-off" speed, where the hull first clears the water, is shown to play an important role in determining the overall relationships between speed, drag, and power required. Configurations embodying fully submerged foil systems are shown to have "hump" power requirements at take-off speed. The question of stabilizing hydrofoil craft in a seaway is dealt with in qualitative fashion. The forces acting on a foil in both ahead and following seas are described and certain tentative conclusions are drawn regarding the ability of various configurations to negotiate these seas. While it is considered that a detailed consideration of the design and construction of hydrofoil craft lies beyond the scope of this paper, a brief discussion is given of certain points where hydrofoil craft depart from more conventional ship design practice. In evaluating the practicality of hydrofoil craft, comparisons are made of specific hydrofoil and conventional designs both in ranges where the hydrofoil shows a clear advantage and in ranges where the application of foils is obviously absurd. From this, a general study is made to determine where the proper field for hydrofoil applications lies. It is concluded that upper limits on size, together with lower limits on speed, fix the maximum size of hydrofoil craft, consistent with available powering, in the 1,500 to 3,500 ton range, and set the lower limit of Froude number based on overall length between 0·6 and 0·7. Within these bounds, the prospect is considered favourable for application of hydrofoils to high-speed passenger ferries, small premium cargo carriers, military patrol craft, and pleasure craft.—*Paper by T. M. Buermann, P. Leehey and J. J. Stilwell, read at the Annual Meeting of The Society of Naval Architects and Marine Engineers, New York, 12th-13th November 1953.*

Effective Breadth Concept

It has often been proposed to assess the effectiveness of deck and bottom plating as elements of the ship girder in accordance with effective breadth theory. If this is done, the effect of the system of framing upon the distribution of vertical loadings becomes significant. In the conventional longitudinal strength computation, it is usual to assume that the downward forces (weights) and upward forces (buoyancy) are distributed along the ship girder in accordance with certain assumptions. The longitudinal distribution of these forces, of course, determines the shape of the bending moment curve, but has no effect upon the computed section modulus, if the Navier hypothesis (simple beam theory) is used. The forces are assumed to be distributed rather than concentrated. In this situation, the system of framing is not a factor. If, however, the premise is accepted that only the ship sides behave according to Navier, while decks and bottoms function as flanges which exhibit shear lag, then the type of framing becomes important. If only transverse framing were present, then all loads, both weights and buoyancy, would be transmitted to the sides as distributed loads by the closely-spaced succession of frame rings. If, however, no transverse framing existed, then all such loads would be transmitted to the transverse bulkheads by whatever longitudinal framing existed (including keels, keelsons, deck longitudinals, etc.), which in turn would transmit these loads to the ship sides as loads concentrated at the transverse bulkheads. The effectiveness of decks and bottoms in acting as flanges will now be lessened sharply at the transverse bulkheads where these concentrated loads are applied, in accordance with effective breadth theory for concentrated loads; the section modulus decreases, and maximum longitudinal stresses increase. Longitudinal framing is present in some degree in practically all steel vessels; a vertical keel is a minimum example. The actual loading transmitted to the ship sides will therefore be a combination of distributed loading and concentrated loads at the bulkheads.—*Paper by H. A. Schade, read at the Annual Meeting of The Society of Naval Architects and Marine Engineers, New York, 12th-13th November 1953.*

Contribution to the Theory of Oil Whip

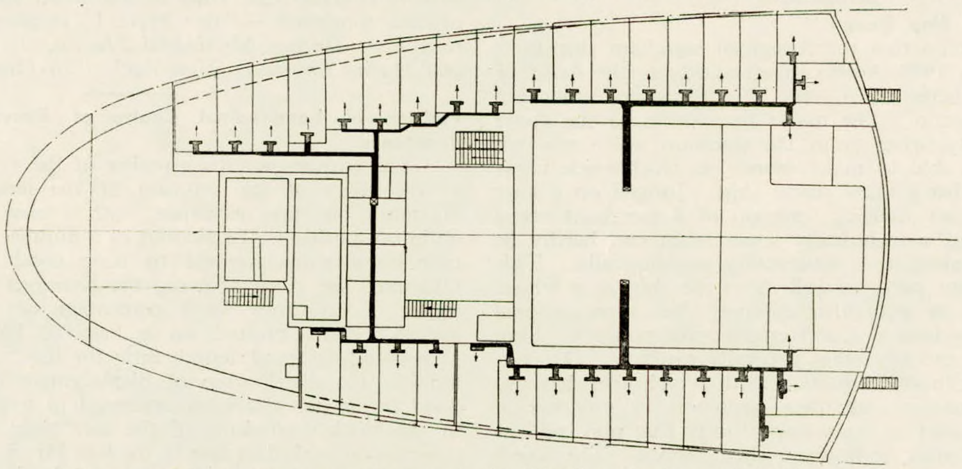
Shaft whipping due to oil action in journal bearings was first described by B. L. Newkirk and H. D. Taylor in 1925. They found that under certain conditions a rotor shaft mounted in sleeve bearings whipped when the rotor was running at any speed above double critical speed; this whipping originated in the bearings since it occurred only when the bearings were running full of oil and could be stopped by decreasing the amount of the oil. Newkirk gave a qualitative explanation of the phenomenon, based on the fact that on the average the velocity of the oil film is half the velocity of the shaft due to its rotation. Hence for rotational frequencies near double the critical frequency there will be a stimulus due to the oil motion of frequencies close to the critical; under this stimulus large displacements and shipping will occur. While the foregoing explanation furnishes a qualitative explanation of oil whip or oil whirl, it does not give a complete theory of it. In particular, this qualitative account of the cause of whipping does not explain why oil whip persists at speeds greater than double the critical speeds. A complete understanding of oil-whip phenomena requires the study of the behaviour of oil film under conditions of load and motion of the centre of the journal. It is shown that an explanation of oil-whip phenomena, at least for small eccentricities, can be obtained from the equations of hydrodynamic forces and the dynamical equations of motion provided that the hydrodynamic force expressions be modified to delete from the oil forces the hitherto included contributions from the regions of negative pressure. While the exact determination of the oil-flow and oil forces including this correction would be rather difficult to carry out at present, the approximate manner indicated will give a satisfactory account of oil-whip phenomena for small eccentricities; there is added a force component in the radial direction, linear in the radial displacement and with a proper "stiffness constant".—*H. Poritsky, Transactions A.S.M.E., August 1953; Vol. 75, pp. 1,153-1,161.*

New Air Conditioning System

Mechans, Ltd., of Glasgow, have recently obtained a licence to manufacture the Swedish "Indivent" system of air conditioning for ships. This system, introduced by the Swedish firm Flaktfabriken, gives each person on board the opportunity of setting the temperature and adjusting the ventilation in his own cabin. The system is based on a central pretreating of the air with its final condition controlled as desired by a manual control in each cabin. In the air conditioning centre the air is pretreated before it is introduced to the ducts. Generally a "Klimator" air conditioning apparatus is used, which is arranged in a special room with the necessary air intakes. If possible the room should be situated at the centre of the system. The apparatus consists of a filter section with dampers for fresh and used air, fan section and coils for heating and cooling. The air leaves the conditioning centre at the lowest supply temperature likely to be required at any part of the system. This temperature is usually maintained at about 59 deg. F. (15 deg. C.) but, to a certain extent, is governed by the temperature of the outside air. The automatic controls required in the conditioning centre being the only automatic controls in the system, control machinery is reduced to a minimum and is extremely simple. The air ducts, which have only to distribute the pretreated air, are on a single line system. The temperature loss is small compared with certain earlier types of installations with double line systems for distributing warm and cold air. A distribution temperature of 59 deg. F. (15 deg. C.) for the pretreated air can be maintained uniform to all supply positions and this temperature will not give rise to condensation. Because the drop in pressure at the supply point is relatively high it is possible to use a high air speed and small duct dimensions without difficulties in air distribution. For every supply position, i.e. as a rule in every cabin, an "Indivent" booster is connected to the duct system. This consists of an air heater and a dial control to adjust the temperature and amount of air supplied within reasonably desirable limits. The adjustment is made by a single hand control and gives an increase in the air supply when the temperature is reduced and *vice versa.* This means that an alteration of the temperature control also gives an alteration of the air movement and thus a quick change in the effective temperature which is comprised of temperature, relative humidity and air velocity.—*The Shipping World, 25th November 1953, Vol. 129; pp. 445-446.*

Mechanical Aspects of Seizing in Metal Wear

This paper presents the mechanical aspect of seizing in metal wear which accounts in a qualitative manner for the observed phenomena in accelerated mechanical abrasion. The mechanical analysis also explains the failure of externally lubricated alloys to equal the performance of self-lubricated alloys. According to the paper, the most widely accepted metal characteristic to reduce the tendency for galling is self-lubrication. Self-lubrication is exemplified by porous powder metals (oil-impregnated), gray cast irons and graphitic steels (graphite-lubricated), and leaded bronzes (lead-lubricated). The mechanical aspect of seizing provides an additional reason for the success of self-lubricated metals which is suggested to account for the failure of externally lubricated metals to equal self-lubricated metals in resistance to seizing. Thus, the facts that self-lubricated alloys are known to have a greater resistance to incipience of galling and to have a lower extent of damage from galling, are accounted for in the mechanical aspect of accelerated mechanical abrasion. On this basis, porous alloys without a lubricating agent would be expected to gall less than nonporous alloys, also unlubricated. The mechanical interpretation of accelerated mechanical abrasion would provide a principle upon which to design surfaces of "engineered roughness". It is evident that the roughness must be on a scale comparable with the asperity size, i.e., the microscopic scale. Many instances of the significance of such engineered roughness are known in practice. The importance of graphite-flake size, shape and distribution in gray cast-iron applications is well known. From the standpoint of self-lubrication, the dimensions of the "lubricant reservoir" would not be too critical. Yet, in practice, the graphite flake classification is controlled closely for such parts as piston rings, cylinders, and other wearing-surface applications. Investigations on the gall resistance of ductile iron as compared to flake gray iron revealed that for equal graphite contents and similar matrix structures, the gall resistance of the flake-graphite iron was superior to that of the ductile iron. The ductile iron was equal in gall resistance to gray iron in which the flake-graphite



Diagrammatic arrangement of part of the "Indivent" air conditioning system content is lower than the nodular-graphite content. These results correspond to predictions from the mechanical analysis of seizing. Porous chromium-plated surfaces are superior to the non-porous surfaces in resistance to galling. The effect of size, shape and distribution of porosity again was found to be more critical than would be predicted from lubrication theory, since channel porosity of the chromium plate is superior to round-pit-type porosity. For proper "wearing-in" of many metal parts, it is necessary to have good resistance to accelerated mechanical abrasion. Chemically pitted surfaces are known to be superior to "superfinished" surfaces in many such applications for wearing-in characterstics. In most industrial applications the mechanical effect is complementary to the lubrication-film effect of separating the surfaces, but somewhat decreases the importance accorded to the lubrication effect alone. There is ample evidence that empirical methods have determined in many cases the importance of microscale engineered roughness, which is predicted (qualitatively, at this time) from the mechanical analysis of accelerated mechanical association. In future surface engineering such interpretations undoubtedly will be tested for quantitative validity.—*Paper by H. Czyzewski, A.S.M.E. 1953 Semi-Annual Meeting; Paper No. 53-SA-26.*

Spark Throwing of Metals

"Non-sparking" tools or equipment of hardened copper have acquired a good reputation in the minds of those responsible for safety in places where explosion hazards exist. Possibly because hard steel is so often the alternative construction material and because hardened copper is relatively soft in comparison with steel, the impression may have been created that sparks are thrown off only from hard and comparatively brittle metal. It may therefore be surprising to know that sparks can be thrown from aluminium, magnesium, or brass when struck against concrete or steel. This is a curious phenomenon, especially where such a soft metal as magnesium is concerned. In the case of this metal, it has been suggested that the spark may be due to the presence of a chromate film, which is known to produce a sparkable coating on magnesium. This may have been so in certain cases but, in fact, a coating is apparently not necessary for sparking to occur and friction alone can produce sparks from magnesium. For example, if a very fine cut is taken on magnesium with a dull tool, the chip may ignite, or if a good tool idles against the work at the end of a cut, the fine particles torn off may burn. Sparks occur now and then when coatings are ground with abrasive wheels, and an abrasive cut-off disc will throw off a shower of dangerous sparks unless the work and the tool are flooded with a coolant. Thus, it is not necessary for a metal to be hard to throw off a spark. If the manner of impact is such that a small particle of soft metal is torn off and if the energy of impact is great enough to heat this particle to its ignition temperature in air, a spark will occur.—*The Engineers' Digest, November 1953; Vol. 14, p. 414.*

American Merchant Ship Design

The authors claim that the American merchant ship built in accordance with 1953 American standards is the finest of its class. It is a better ship than its forerunners according to almost any criterion. The major reservation to the above statement of quality arises from the question as to whether or not it is better able to make money in world-wide competition, for it is also a more costly ship. Judged on a long term basis, the money making function of a merchant vessel is so important that a technically sound ship can hardly be called successful unless it is satisfactory economically. This applies to constituent parts as well as to the ship as a whole. The ship designer, or any other engineer, has a professional responsibility to produce a sound engineering product. Ship designers, builders and operators generally must be: (1) conversant with the economic situation with which they have to deal; (2) cost conscious in their approach to engineering problems; and (3) alert to every opportunity that may present itself to reduce expenses or improve their service. The object of this paper is to call attention to these three approaches to the economic problems involved in current merchant ship design and construction, by giving some facts relevant to the first, by raising cost questions to encourage the second, and by pointing out a few of the possible fields for the third.—*Paper by H. H. Holly and J. A. Pennypacker, read at the Annual Meeting of The Society of Naval Architects and Marine Engineers, New York, 12th-13th November 1953.*

Aluminium Alloys in Ship Construction

Aluminium has become established as a shipbuilding material. The increasing number of applications both in this country and abroad demonstrate that under some circumstances the additional initial cost is justified. Aside from special conditions imposed by draft limitations, aluminium finds its greatest use on merchant ships in superstructures and deckhouses, where the saving of weight simplifies stability problems and it is of particular advantage to high-speed vessels. Other advantages accrue from the low modulus of elasticity which makes the material attractive for structures erected upon a steel hull, and decreased maintenance due to its resistance to corrosion. Relative high strength and availability of a reasonable range of plate lengths, widths and thicknesses, along with sufficient choice of shapes remove any size restrictions for incorporation in a design. Superstructures can be designed to be light in weight and yet meet the requirements of the classification societies and regulatory agencies, both as to strength and fire resistance. Fabrication and erection procedures follow the same general pattern that has been established for steel ships, although greater care in handling, fitting and forming is necessary. The limitation placed upon heating and prohibition of burning are handicaps. Rivets can be driven relatively fast and the welding of heat-treatable alloys is possible but is limited to secondary structure. The development of high-strength non-treatable alloys suitable for efficient welding has progressed beyond the experimental stage and they will probably displace much of the heat-treatable material now in general use. Being a comparatively new material, the application of structural aluminium to ships will necessarily be gradual. It will be many years before our knowledge of handling and forming it will have reached the status that now exists in respect to the older and more established shipbuilding materials. Unquestionably the lack of experience has retarded its use. When it is known what the problems are and how they can be economically solved, the barriers of apprehension and conjecture will have been removed and the way will be paved for greater acceptance. All engineering improvement and technical advancement has resulted in a tendency toward increase in initial costs. To pass judgement on any new development solely on the basis of first cost can lead to conclusions that might prove erroneous in the long run. Any increase in the use of aluminium will depend upon the economic gains that result, taking into account not only construction cost but also the relative economics of operation and maintenance in service. In the meantime it behoves the shipbuilder to assimilate all the experience possible, bearing in mind that what is new today may become standard practice tomorrow.—*Paper by C. V. Boykin and M. L. Sellers, read at the Annual Meeting of The Society of Naval Architects and Marine Engineers, New York, 12th-13th November 1953.*

Position of Longitudinal Centre of Buoyancy for Minimum Resistance

The author presents a review of the available data relating to the effect of the position of the longitudinal centre of buoyancy on ship resistance, with a view to determining its optimum position. Particulars of a number of single-screw and twin-screw ships reputed to have good lines are included. Diagrams are given showing the positions of the longitudinal centre of buoyancy, as a percentage of the length between perpendiculars, plotted on a base of block coefficient and corresponding speed:length ratio for the "suitable" speed. The longitudinal distribution of displacement in certain reputedly good forms has also been expressed in terms of the difference in the block coefficients of the fore body and after body for comparison with data due to the late Mr. S. B. Ralston.—*Paper by H. Bocler, submitted to the Institution of Engineers and Shipbuilders in Scotland for discussion in writing, October 1953.*