# ENGINEERING ABSTRACTS

Section 3. Shipbuilding and Marine Engineering

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## Welding of Alloy Steels

This paper deals with the measures to be taken in the welding of alloy steels. For low and medium alloy steels the cooling rates and other physical welding conditions can be determined by having reference to the S-curves of isothermal transformation. The choice of suitable electrodes is discussed with reference to molybdenum and chromium-molybdenum steels. The welding of stainless steel and the difficulties encountered with various types of stainless steels is discussed with particular reference to American investigations.—*H. G. Geerlings, De Ingenieur, Vol. 61, 21st January 1949, pp. MK 1-10.*

## Construction of Energy Charts for a Dissociated Gas

Estimation of the theoretical performance of an internal combustion engine is incomplete without consideration of dissociation, and it is the purpose of the article to develop the basic theory required for the construction of a simple chart for a single gas to extend the theory to a mixture of gases. The type of chart developed by the author for a single gas may be applied to the determination of the ideal cycle in an internal combustion engine. The limitation of such a chart and factors affecting its use are discussed. Firstly, one chart serves for one air-fuel ratio only, as it is practically impossible to indicate the effect of different air-fuel ratios on a single chart. Secondly, if it is considered that dissociation should be taken into account, distinction must be made between the pre- and post-combustion mixtures, and in this case a separate chart for the pre-combustion mixture must be drawn. Thirdly, the pre-combustion mixture is affected by the gases left in the cylinder after exhaust has taken place. The analysis and amount of the residual gases must be determined if the chart for the mixture is to take account of them. Fourthly, the mass of the residual gases is affected by variation in compression ratio, and so, strictly speaking, any one chart will be correct for one compression ratio only. Fifthly, it is assumed that the gases obey the gas laws, and that at all points the mixture is in thermodynamic equilibrium. If these five conditions are adhered to rigidly, any chart will have very little use as it will be possible to represent only one cycle on it. This difficulty may be overcome without appreciable error by assuming a mean value for the mass of the residual gases, with the result that the only limitation placed upon the use of the chart is the fuel-air ratio. Charts are given showing the properties of the pre-combustion mixture and the post-combustion mixture in the dissociated region respectively. It is shown how these charts can be used for the determination of the ideal attainable efficiency of an internal combustion engine operating on the constant-volume or Otto cycle.—*E. A. Bruges, Engineering, Vol. 167, 28th January 1949, pp. 73-75; 4th February 1949, pp. 97-98.*

**Combustion System for Burning Bunker C Oil in a Gas Turbine**

The combustion system for a 4,800 h.p. gas turbine is described and the results of tests made during its development are given. The liner temperatures, outlet-gas-temperature pattern, smoke and carbon formation, ignition, and operating range obtained with the combustion system are given, and factors which affect these features are discussed. The properties of the fuel used are outlined and the effect of these on the gas turbines is described. Comparisons are made of tests on the complete plant with tests of the combustion components. The General Electric Co. gas turbine division in Schenectady has designed and built a 4,800 h.p. gas turbine for locomotive service. This plant is arranged to burn Bunker C fuel oil or any other fuel oil at least as good. Combustion components have been tested in order to accumulate the information needed for the design of the combustion system. The unit itself has been tested for about 700 hours, during which time approximately 230,000 gallons of Bunker C have been burned. The tests made thus far on the components and on the unit show that the combustion-system is generally satisfactory. The tests also indicate that the combustion-system pressure drop and the liner temperature may profitably receive additional attention now. The remaining features are sufficiently well developed so that further operating experience is required to point the way to further improvement. The pressure drop at present is somewhat high, but this, it is felt, can be reduced with further development now in progress. More efficient liner-cooling arrangements are being designed and tested to produce cooler and longer-level liners. Attempts have been made in the testing to obtain Bunker C oils which would introduce the obstacles associated with the quantity and character of the ash. The tests show that these obstacles have been overcome on the oils tested. However, Bunker C fuel is a rather indefinite substance and the qualities of all of the ashes to be encountered in the fuel, as well as their effect on the turbine, will have to be found by operating experience. This experience may show that further improvements, such as the forced cooling of turbine parts, may be required if and when turbine-inlet temperatures are increased. Until operating experience shows some limit to be necessary, no limits have been specified concerning the quality of the fuel to be used on the unit except that it shall be commercial Bunker C.—*Paper by B. O. Buckland and D. C. Berkey, read at the 1948 Annual Meeting of the American Society of Mechanical Engineers, Paper No. 48-A-109.*

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**Sulzer Gas Turbine**

In certain instances it is necessary to by-pass a quantity of gas round the driving turbine of a gas turbine installation and introduce the gas to the turbine driving the auxiliaries, in order to diminish the useful output of the plant. The auxiliary as well as the main turbine must, therefore, be constructed of materials highly resistant to heat, and with the arrangement shown in Fig. 4 it is possible to avoid this disadvantage. The machine consists of a compressor (1) taking air through a pipe (2) and discharging it through a cooler (3). The compressed air is delivered partly as circuit air through a pipe (4) into a heat exchanger (5) and partly as combustion air through a pipe (6) into a second heat exchanger (7). The circuit air flows through a pipe (8) into a gas heater (40). The heated air then flows through a circuit turbine (42) and is discharged into the heat exchanger (5). Part of the heat still contained in the expanded circuit air is transmitted to the air passing from the pipe (4), and a further part of the residual heat is withdrawn from the circuit air in a cooler

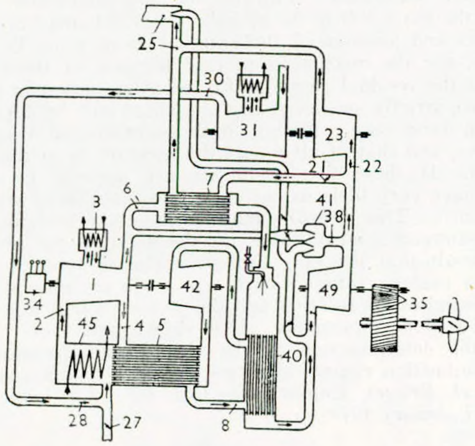


FIG. 4.

(45). After cooling, the circuit air again passes through a pipe (2) into the compressor (1) and the circuit is repeated. The turbine (42) is operated with pure air and drives the compressor (1), and a starting motor (34) is provided. The output of the power turbine (49) is transmitted to the propeller through gearing (35). For reducing the output of the turbine (49) there is a by-pass (38), through which part of the gas can be fed to the exhaust pipe (25) and the remainder passes into a mixing chamber (41). For increasing the output, the valves (22, 28) are opened and the valves (21, 27) are closed. The exhaust gas from the turbine (49) consequently flows through the turbine (23) into the exhaust pipe. This exhaust gas turbine drives a make-up air compressor (31), which delivers the air into the circuit through a pipe (30) and raises the pressure level. Thus the useful output of the plant can be increased to 5-10 times the normal output, and reasonable efficiencies can still be obtained.—*Brit. Pat. No. 604,156, issued to Sulzer Frères, Winterthur.—The Motor Ship, Vol. 29, February 1949, p. 466.*

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**Advantages of High Inlet Temperature for Gas Turbines**

The purpose of this paper, which is specifically concerned with gas turbines for aircraft, is to show the large gains in power and economy that can be realized by increasing the turbine-inlet temperature and the pressure ratio of the gas-turbine engine. The extent to which the various methods of cooling the turbine blades are effective in obtaining these high gas temperatures is shown. High-temperature protective coatings of extremely low thermal conductivity (thorium oxide) might be used to reduce the heat flowing into the blades. The effectiveness of the comparatively cool layer of gas in contact with the cooled blades in insulating the blade from the hot gases should be determined. Such an insulating layer may be effective enough to reduce the importance of the ceramic coating, particularly if the flow is laminar. Such a layer should be very effective in protecting the trailing edge of blades which cannot be easily cooled because of mechanical limitations on the size of the passage. A heat-resisting material of high conductivity is most desirable because the temperature is more uniform through the blade and only heat flowing through the coating is transferred to the cooling medium. A coolant with a high specific heat and boiling point is also desirable. The combination of a low-conductivity protective coating, heat-resisting material of good conductivity, and coolant liquid of high specific heat and suitable for high-temperature operation should result in a compact cooling system. Air cooling of the rim or roots of turbine blades should give an increase in effective gas temperature of 100 to 150 deg. F.; hollow-blade cooling should give 300 to 500 deg. F. with approximately 8 per cent dilution without inserts and the same effective cooling with less than 3 per cent dilution with inserts; film cooling or finned hollow blades of good heat-resisting alloys should give 1,200 to 1,500 deg. F. with 6 per cent dilution; and liquid cooling should permit safe operation at stoichiometric temperatures.—*Paper by O. W. Schey, read at the 1948 Annual Meeting of the American Society of Mechanical Engineers; Paper No. 48-A-105.*

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**Optimum Compression Ratios for Gas Turbine Cycles**

In gas turbine engineering the choice of the compression ratio may be based on various requirements. Thus, for example, the optimum compression ratio to be found in a given case may be that which yields minimum specific fuel consumption, or minimum weight of working fluid per horsepower developed; or, if the plant is to be fitted with a heat recuperator, the optimum compression ratio will be that which calls for the smallest or least expensive recuperator. The author investigates four typical gas turbine cycles consisting of (1) two adiabatics and two isobars, (2) adiabatic compression, isothermal expansion, and two isobars, (3) isothermal compression, adiabatic expansion, and two isobars, and (4) two isotherms and two isobars. In all cases the isotherms are assumed to be approximated by a number of para-isotherms, each comprising an adiabatic and an isobar. In all cycles the optimum compression ratio is found to increase with the number of para-isotherms employed. The optimum compression ratio is found to decrease with increasing recuperation, and in the case of complete recuperation it would be unity. With regard to No. 4 cycle, it is found that its overall efficiency is independent of the compression ratio, provided an infinitely large number of para-isotherms are employed. Concerning the relationship between compression ratio and weight of working fluid supplied to the turbine per kW. hour, the optimum pressure ratio increases in the order in which the cycles are listed, but is independent of the heat recuperation. —*P. Ferretti, Ricerca Scientifica, Vol. 18, October 1948, pp. 1243-1261.*

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**Corrosion of Turbine Journals**

Corrosion in turbine lubrication systems is seldom severe enough to require a drastic remedy. On occasion, however, water of a corrosive nature, such as sea-water, has been known to gain access and to cause extensive damage to journals, gears, etc. It has been found that corrosion-inhibiting turbine oils, effective in ordinary circumstances, cannot be relied upon to prevent corrosion by sea-water, and that owing to the persistent retention of salt by microscopic pits in the steel, sea-water corrosion, once established, is not easily arrested. For the prevention of damage in these circumstances the author proposes the use of a water-miscible inhibitor consisting of sodium nitrite solution by circulating it with the oil until the measures to eliminate sea-water from the oil system have been successfully applied, and any existing corrosion arrested. Practical trials have been carried out at sea in ships of the Royal Navy, proving that sodium nitrite gives satisfactory protection again corrosion in presence of a large amount of added sea-water, and that corrosion already in progress can be stopped.—*Paper by S. E. Bowrey, read at a meeting of The Institute of Marine Engineers, 8th February 1949.*

**Corrosion in Exhaust Pipes of Internal Combustion Engines**

A general description is given of the reactions which may occur between the products of combustion and the exhaust-pipe materials of internal-combustion engines, and an apparatus is described whereby this corrosion can be studied at various positions in the pipe, i.e., corresponding to different exhaust-gas temperatures. Results are given for pipes made of cast iron and copper acted upon by an exhaust gas of the composition, carbon dioxide 11·5, carbon monoxide 0·1, oxygen 2·6, water vapour 11·0, and nitrogen 74·8 per cent. At 400 deg. C., cast iron suffers rapid initial attack, but after one day the rate falls to approximately 0·06 mm. per year; at 180 deg. C. the rate of attack is so small as to be of no consequence, but at 70 deg. C. it is again considerable and probably attains a rate of 1·4 mm. per year. Like cast iron, copper is rapidly attacked at 400 deg. C., but, unlike cast iron, the scale formed gives no protective action; at 180 deg. C. the rate of attack is approximately 0·03 mm. per year, and at 70 deg. C. approximately 0·08 mm. per year.—*G. Schikorr, Metalloberfläche, Vol. 2, No. 4, 1948, pp. 73-79. (Journal, Inst. of Metals, Metallurgical Abstracts, Vol. 16, January 1949, p. 281.)*

**Moment of Inertia of Small Marine Propellers**

The moment of inertia of the propeller is important because it is one of the dominant factors controlling the actual critical speed of the installation. This, in itself, is obviously sufficient to make it a matter for careful attention; but there is another aspect which, at the present time, gives rise to greater concern, viz., the circumstance that the moments of inertia of propellers (at any rate of the smaller sizes) are found to be subject to wide variations. Propellers of the same material, and having the same diameter, pitch, surface area and number of blades, might logically be expected to have, within reasonable limits of tolerance, the same moment of inertia. In practice, however, this is far from being the case, and instances are known in which variations of more than 50 per cent in the value of the moment of inertia have been observed in "similar" propellers. Variations of such magnitude are a source of real trouble to designers and superintendent engineers. For the time being, the only measures whereby these difficulties may be overcome are for the designer and the superintendent engineer to insist that all propellers be "swung" to determine their actual moments of inertia, before the shafting system is designed; and to keep accurate records of the moments of inertia of the propellers fitted to individual ships, so that future replacement propellers may have the same value. There is a feeling in some quarters, however, that the problem should be tackled more drastically, and that the uncertainties which at present surround the question of the moment of inertia of small marine propellers might be eliminated by establishing a series of standard moments of inertia. It is suggested that manufacturers, users and classification societies should collaborate to determine suitable formulae as a basis for such standards. It may be argued that this suggestion is unacceptable, on the ground that the chief aim of propeller design is to achieve propulsive efficiency, and that manufacturers should not be restricted in this aim by having to produce propellers conforming to a moment of inertia specified in advance. The author, however, holds the view that it is both desirable and practicable to compile a list of standard moments of inertia, at any rate for small commercial propellers, without any serious restriction being imposed on propeller performance. He has determined experimentally the actual moments of inertia of a considerable number of service propellers which are given in separate lists for cast iron and bronze propellers respectively, together with the leading dimensions of diameter and other characteristics of these propellers. The tables also include a comparison of the actual moment of inertia with that calculated from a simple formula proposed by the author.—*J. Whitaker, The Shipbuilder and Marine Engine-Builder, Vol. 56, March 1949, pp. 163-165.*

**Anchor Trials**

During the past few years considerable technical research has been undertaken by the Admiralty on the design of anchors. The general method of testing on the present site at East Fleet is to pull the anchor by steel wire ropes through a series of blocks by a 6-ton four-wheel-drive lorry at a speed of 1·5 to 2ft. per min. A statimeter or force recorder is inserted in the system near the anchor. At given intervals this instrument measures the force necessary to drag the anchor and, also, the force it will just hold without dragging. Pulls up to 50 tons can be measured, which are sufficient for the largest anchors, and allowance is made for the anchors to drag as much as 100 feet. The new 5½-ton Admiralty Mooring Anchor 7 represents the first fruits of the research programme, and its design is based on experience gained on previous trials. The new design will hold more than eight times its own weight before dragging, whereas the old Admiralty moving anchor would withstand a pull of only twice its own weight. Other types of anchors tested include the new improved design A.M.11, the 10,000lb. U.S.N. Standard stockless anchor and the 750lb. Danforth anchor.—*Admiralty: Dept. of Chief of Naval Information, Bulletin No. 14. Journal, The British Shipbuilding Research Association, Vol. 4, February 1949; Abstract No. 2433, p. 90.*

**Comparison of Recent High-pressure Marine Installations—I**

This paper represents a study of the design and operation characteristics of outstanding recent high-pressure propulsion plants as compared to a standard installation operating at 450lb. per sq. in. g. and 750 deg. F. As standard performance that of the propulsion plant of the *Sea Fox* was chosen. The following high pressure installations were considered. (1) S.S. *Marore*, which is one of the eight vessels of the Venore class designed as ore carriers and operated by the Ore Steamship Corporation. The designed steam conditions are 1,450lb. per sq. in. gauge and 750 deg. F. at the superheater outlet, with a double steam reheat at 185 and 15lb. per sq. in. gauge respectively, both to about 565 deg. F. temperature. Normal power is 11,000 s.h.p. at 95 r.p.m. The main propulsion unit consists of high-pressure intermediate-pressure and low-pressure turbines, which drive the propeller shaft through double-reduction gearing. The steam reheaters located in the high-pressure and intermediate-pressure receiver pipes superheat the turbine steam to a total temperature of about 565 deg. F. The heating steam for both reheaters is obtained directly from the main boilers and the drains are pumped back to their respective boilers by motor-driven reheater drain pumps. (2) S.S. *Beaverglen* of the Canadian Pacific Railway Co., operating at 850lb. per sq. in. g. and 850 deg. F. with single gas reheat at 180lb. per sq. in. g. to 850 deg. F. This is a turbo-electric vessel, the designed output of the shaft motor being 9,000 s.h.p. at 108 r.p.m. (3) S.S. *Examiner* of the American Export Lines, operating at 1,200lb. per sq. in. 750 deg. F. with single gas reheat at 220lb. per sq. in. g. to 750 deg. F. Normal power is 8,000 s.h.p., the main shaft being driven through double-reduction gearing at 96 r.p.m. (4) S.S. *Admiral Benson*, which is one of eight vessels of the P-2 Passenger Cargo Class and was built in 1944. The steam conditions are 600lb. per sq. in. g. and 850 deg. F. without reheat. The normal power rating of this twin-screw turbo-electric vessel is 18,000 s.h.p. total at 120 r.p.m. Compared with that of the *Sea Fox*, the fuel rate of the *Marore* is 13·4 per cent less. In the case of the *Beaverglen* the reduction in oil rate in the main and auxiliary plant (excluding cargo refrigeration) of about 4 per cent as referred to the *Sea Fox* does not compare well with the decreases noted on the other reheat installations. A breakdown of the gains and the losses shows, however, that the efficiency of the main turbine closely approached the full improvement that can be expected from a single gas-reheat cycle. The improvement due to thermal efficiency of the unit is about 10 per cent referred to the *Sea Fox*. However, the *Beaverglen* is a turbo-electric installation, and the losses due to the electrical transmission and excitation almost cancel the entire gain due to reheat in the main unit. The plant of the *Examiner* lost some of its inherent gains in the main unit efficiency, since both initial and reheat designed temperatures were not realized on trials. In comparison to the other geared installations a larger number of bearings operating at higher speeds contributed a greater share of the losses. A net improvement in oil rate of about 8 per cent over the *Sea Fox* was obtained on trials at the same vacuum conditions. The plant of the *Admiral Benson* was designed primarily for passenger service, a larger portion of its auxiliary load normally is used for hotel services, and this was in operation during trials. The oil rate on trials without hotel load was about 4·5 per cent less than on the *Sea Fox* when corrected to the same vacuum conditions. The net gain of about 6·0 per cent in the main unit efficiency was reduced in part due to the additional transmission and excitation losses associated with a turbo-electric installation.—*W. I. H. Budd and O. Praznik, Marine Engineering and Shipping Review, Vol. 54, February 1949, pp. 58-64.*

**Comparison of Recent High-pressure Marine Installations—II**

An abstract of the first part of this paper is given immediately above. The second part gives a comparison of service results obtained with the S.S. *Examiner* and vessels of the Venore class. Voyage logs of the *Examiner* and of a low-pressure (450lb. per sq. in. gauge of 750 deg. F.) sister ship covering a recent period of one year's operation on the same route were analysed. A comparison of results on either fuel coefficient or oil consumed per shaft horse-power basis showed the *Examiner* to have no appreciable gain in economy over the sister ship. This would indicate an increase in oil consumption of about 10 per cent when compared with trials. It is not possible either to verify or to discredit the apparent changes in the total fuel rate by a consideration of the component parts since there are not sufficient data available for this purpose. However, an analysis of the logs does indicate that the cycle efficiency has been reduced due to difficulty encountered with the motor-driven feed pumps. The turbine-driven feed pump is usually in service at sea. Both overload nozzles are open in order to supply adequate feed for full-power operation and therefore more steam is supplied to the pump than normally would be expected for this power. This may indicate either loss of turbine efficiency, which seems unlikely, deterioration of the pump, or larger amounts of feed resulting from loss in general plant efficiency. The use of the less efficient steam-driven standby pump supplies so much exhaust to the second-stage feed heater that the first stage heater must be by-passed to avoid discharging this steam directly to the condenser. These difficulties can explain only about one-third of the apparent increase in oil rate. The remainder, if real, cannot be allocated from the existing data. Boiler operation has been satisfactory. Burners have carbonized rapidly when using bunker C bonded oil, but similar difficulties have been reported from other vessels when burning this type of oil. Operating personnel try to avoid popping the boiler safety valves, since they usually leak upon reseating. The high pressure turbine was opened for inspection in 1945 and blade damage was found to have occurred. The damaged blading was of the impulse type made of a special stainless steel. It was found that metallurgical deterioration of the blade material had occurred and all blades in the h.p. turbine were replaced with blading made from standard 13 per cent chromium stainless steel. In Venore class vessels two air heater fires have occurred with considerable damage to the air heater section and some damage to the economizer. A number of air heater tubes were removed to improve the action of the air-puff soot blowers and to increase stack temperature. Threaded clean-out plugs in the economizer bends were found to leak and seal welds were applied. Leakage of handholes in superheater and economizer headers was stopped by substituting key caps and seal welding. After eight months of successful operation the high pressure turbine of the Feltore became a casualty, eight hours after leaving port. It was found that several blades had rubbed and the rotor was bent. This damage is ascribed to the introduction of foreign material, the source and type of which is still unknown. The reheater head joints of both high and low pressure reheaters developed leaks and larger bolts were provided. The fifth, sixth and seventh vessels of the Venore class are supplied with a welded type head joint, but it is too early to draw conclusions regarding its success in service, although it is felt that this arrangement shows considerable promise. Tube leakage in the reheater was also experienced, and it was decided to seal weld tube ends to the tube sheet by a single-pass weld after expanding.—*W. I. Budd and O. Praznik, Marine Engineering and Shipping Review, Vol. 54, March 1949, pp. 57-61.*

**Flow Dynamic Considerations in Turbine Exhaust Design**

This article which is based upon original research carried out at the Central Boiler and Turbine Research Institute, Leningrad, contains a critical survey of tests conducted on models of turbine exhaust ends. The conclusion is drawn that the comparison of different casing designs should be based upon the use of a flow resistance coefficient in conjunction with a coefficient characterising the inequality of kinetic energy distribution in the exhaust steam passng to the condenser.—*L. N. Ilin and D. A. Mariupolskaya, Kotloturbostroenie, October 1948, pp. 22-26.*

**Sea Speed and Steering**

Demand for sea-kindliness has directed the author's attention to what really happens to an ocean-going ship at sea. Researches by yacht constructors have developed a "balanced form", that should have potentialities for better sea-speed and steering at sea. As the theory on which the "law of balance" is based has met with differences of opinion, an analysis of the details of the problem is submitted. The result of this analysis tends to prove the "law of balance" to be correct. Lessons learned from the analysis, which shorten the otherwise long and laborious work of altering lines, are submitted. Observations based on practical experience are mentioned, and the detrimental influence, that want of balance will be likely to produce at sea, is discussed. As balanced forms often require the lines which promise least resistance to the upright ship to be altered to a certain extent, research work by tank testing is suggested as the only means to settle—as far as possible—whether the balanced form should be adopted, in view of the actual conditions at sea, so as to arrive at an overall optimum form. Three models have been investigated, and their different peculiarities given. In conclusions and suggestions for further research are given some alterations in procedure and design, some probable improvements accruing from balanced forms, and subjects for further consideration. An appendix gives details for calculations.—*Paper by Capt. C. Blom, read at a meeting of the Institute of Naval Architects, 8th April 1949.*

**Crankcase Safety Device**

In a paper entitled "Some Current Types of Marine Diesel Engine" C. C. Pounder describes the latest patented form of a crankcase safety device consisting of aluminium door hinged at one side and held by a very lightly loaded spring sneck on the other. The door opens at a pressure of not more than ½lb. per sq. in. and, on the release of the pressure, closes again. This re-closing is important, because it has happened that the first explosion—inside the crankcase—has been mild, but, as soon as the external air has gained free access to the crankcase contents, a much more violent secondary explosion has ensued. On at least six occasions, within the author's experience, safety doors have functioned in the way described. All the engines were either marine auxiliaries or stationary engines. He knows one ship, of non-British make and ownership, where a main-engine crankcase explosion, with fatal results, occurred at the outset of the machinery's life. Safety doors were then fitted and thereafter these doors lifted periodically, but there were no explosions. Some years later, alterations were made to the engine parts which were the focal points of the trouble and hot spots never again appeared. A chemist may say that a hot spot is not an essential pre-requisite for an explosion. The author, to date, has seen nothing to support this dictum. The next thing is to ensure that the crankcase doors are well bolted to the framing. The common practice on shipboard—a practice which every engineer likes—is for the crankcase doors to be lightly held, and at as few points as possible. Fig. 22 shows typical fastenings. The doors are, in fact, nothing more than light, sheet steel diaphragms rendered reasonably oil-tight around their perimeters. Such doors can, however, be blown off, or "sucked off", by pressures of 2 or 3lb. per sq. in. In the author's opinion, all crankcase doors should be well bolted around their perimeters, the plate thickness being left as before, but the flange being stiffened to suit the bolting, and the plate being either dished or stiffened by cross-bars. For example, a door, say, 80 inch by 40 inch by 0·094 inch, held at points say 14 inch apart, the flange being stiffened by a 1·25-inch angle-bar, can be pulled off, or "sucked off" by a live pressure of appreciably less than 1 atmos. The same plate, if dished to a depth of 4·75 inch bounded by an angle-bar 2 inch by 2·5 inch pitch, the flange being 2·5 inch wide and 0·69 inch thick, will withstand 10 atmos. The importance of bolting-on the crankcase doors may be more apparent in installations comprising more than one engine. The tremendous amount of heat energy liberated, when a crankcase door blows off, may well be responsible for creating such a dynamic uprush of highly heated air to the casing and skylight that the ensuing nega-

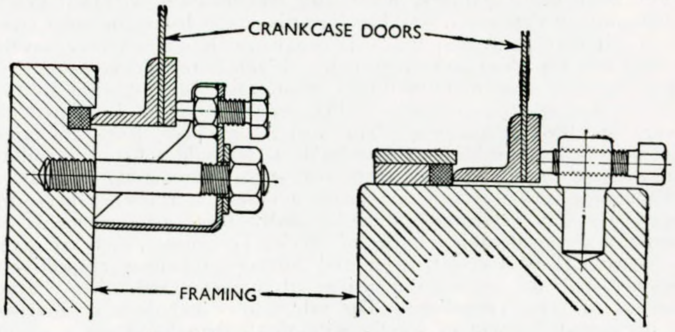



FIG. 22.—*Crankcase door fastening*

tive pressure-wave—the vacuum which succeeds the uprush—can "pull off" other crankcase doors, from the same or adjacent engines. Thus an incident of local significance can lead to something much more serious. Discussing materials of construction, the author points out that there are many things to be considered before a departure is made from mild steel, for important components. For example, in say, a mild steel rod of 10-inch diameter there are no locked-up stresses, but, in a similar rod of alloy steel—say, forged manganese molybdenum steel—the internal stresses may be twice the working stresses, according to metallurgical experts. Some phenomena experienced in service are only explicable on this basis.—*Paper by C. C. Pounder, read at an Extra General Meeting of the Institution of Mechanical Engineers, 18th March 1949.*

### New Machinery in the *Eurybates*

In 1924, the *Dolius*, the first Scott-Still-engined ship, was completed, and in 1928 the *Eurybates*, also built for the Blue Funnel Line, was placed in service, equipped with twin-screw 5,000 b.h.p. Scott-Still machinery, a development of the design of that installed in the first ship. The main principle in the Still engine is in the recovery of some of the heat usually lost in the circulating water, also in the exhaust gases. Steam is generated by this means, and was supplied in the first engines to the under side of the working pistons. In the machinery of the *Eurybates* it was delivered to two separate steam cylinders, with pistons driving the crankshaft. The temperature of the cylinder jacket water was maintained at 350 deg. F., and steam and water at this temperature were passed through a bank of cross tubes in the boiler and to a feed water heater. The internal-combustion engine section operated as a two-stroke unit, with port scavenging, the exhaust gases being discharged through ports. In order still further to improve the efficiency, the exhaust from the auxiliary generating sets could be connected to the main engine exhaust system. With this design it was hoped to achieve a high degree of economy in operation, but possibly the hopes were not entirely fulfilled. It was decided in view of the long period the machinery had been in service to modify it so that the engines should be converted to the normal Diesel type, so far as possible, without too substantial alterations. The work was carried out by the original builders, Scotts' Shipbuilding and Engineering Co., in conjunction with the engineers of Messrs. Alfred Holt and Co., and the reinstallation of the plant was effected at Harland and Wolff's Belfast yard. As originally designed, each of the engines had an output of 2,500 b.h.p. at 105 r.p.m., although this speed has not latterly been maintained. New propellers have been fitted and the speed will now be 112 r.p.m. There were five combustion cylinders with a diameter of 27 inch, and a piston stroke of 44 inch, whilst the two steam cylinders were 24 inch in diameter with a piston stroke of 45 inch. The whole of the steam side of the engines and the accessories involved have been removed. In place of the two steam cylinders, two vertical Doxford-type scavenge pumps are installed, supplying the scavenge air to the combustion cylinders. The original separate turbo-scavenging pumps have been removed, also the high-pressure and regenerator boilers. The balance weights on the crankshafts were repositioned in order to eliminate vibrations, and there is an entirely new design of cylinder head, starting air valves now being provided in the cylinder covers. These were not needed previously, as the engine started on steam. Moreover, a new type of fuel oil valve has been introduced.—*The Motor Ship, Vol. 29, March 1949, pp. 476-479.*

### Engine Indicator

Indicators come under two general classifications, the card-drawing type and the non-recording, maximum pressure type. The former consists of a stylus whose up and down movement, actuated by the pressures within the cylinder, is traced on a card wrapped around a rotating drum. This produces a continuous record of cylinder pressures over a given period of time. On the other hand, the non-recording type only shows the maximum cylinder pressure occurring during the time that the indicator is connected to the combustion chamber. Non-recording indicators fall into two categories, the spring balanced type and the trapped pressure type. The first contains a small piston which is exposed to the cylinder pressure after the instrument is attached to the engine. This piston is forced upwards against a spring whose tension is adjusted by a micrometer sleeve enclosing the entire unit. When the force of the spring on the piston balances th epressure on the piston, a direct pressure reading can be made from the micrometer scale. The second, known as the trapped pressure indicator, is attached to the cylinder head of an engine where the pressure of the cylinder gases forces open a check valve leading to a small chamber. Here a bourdon gauge reads the maximum pressure trapped within the chamber. The choice of an indicator for use on the high speed automotive and industrial Diesels is limited by the nature of such units. When considering the card-drawing type for this service, problems arise regarding the rapid rate of pressure change which is a limiting factor in indicators operating on this principle. Inertia forces, difficulty with reducing mechanism and drum speeds as well as the matter of accessibility on the smaller engines are also points to be considered. Consequently, it is generally impractical to use the card-drawing instrument on high-speed engines, and the maximum pressure type takes preference here. Although this limits the information that may be obtained, the measurement of maximum compression and firing pressures, provided by the maximum pressure type indicator, furnishes data that is valuable in balancing engines and in detecting operating troubles.—*Diesel Power and Diesel Transportation, Vol. 27, February 1949, pp. 66-70.*

### Engine Fuel Timing System

In Fig. 2 is illustrated a timing valve for a fuel injection system. The valve (1) is of the partly balanced type with a reduced diameter (3) making an annular space in conjunction with the fuel inlet port (4). The balancing stem (5), which is smaller than the plunger (2), has a conical valve (7) sealing the flow of fuel to the injection supply pipe (13). The valve timing is effected by the eccentric adjustment of the fulcrum (20) of the lever (19). The shaft (33) carries a segment (34) meshing with a worm (35). This worm is connected to the control

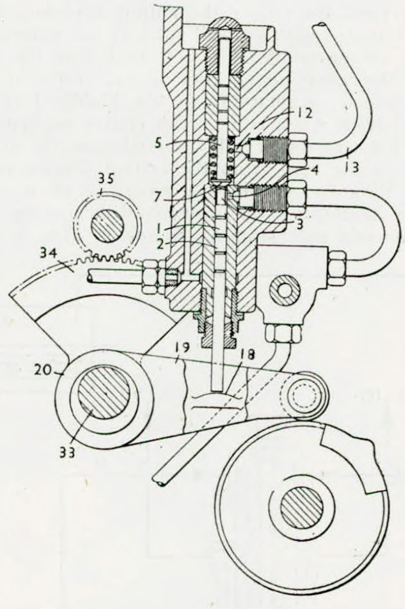



FIG. 2.

wheel. The shoulder (18) is curved to a contour giving the desired duration of valve opening in conjunction with the timing and in relation to the linear position of the main piston. A slight pumping action occurs in the pipe (13) as the control valve lifts before the injection valve opens, and before the ports (4, 12) connect with each other, thus giving a quick rise in pressure. Conversely, a reverse effect occurs when the control valve closes.—*(Patent) No. 608,951. Vickers-Armstrongs, Ltd., and E. Davies, London.) The Motor Ship, Vol. 29, March 1949, p. 505.*

### Diesel Cylinder Wear with Boiler Fuel

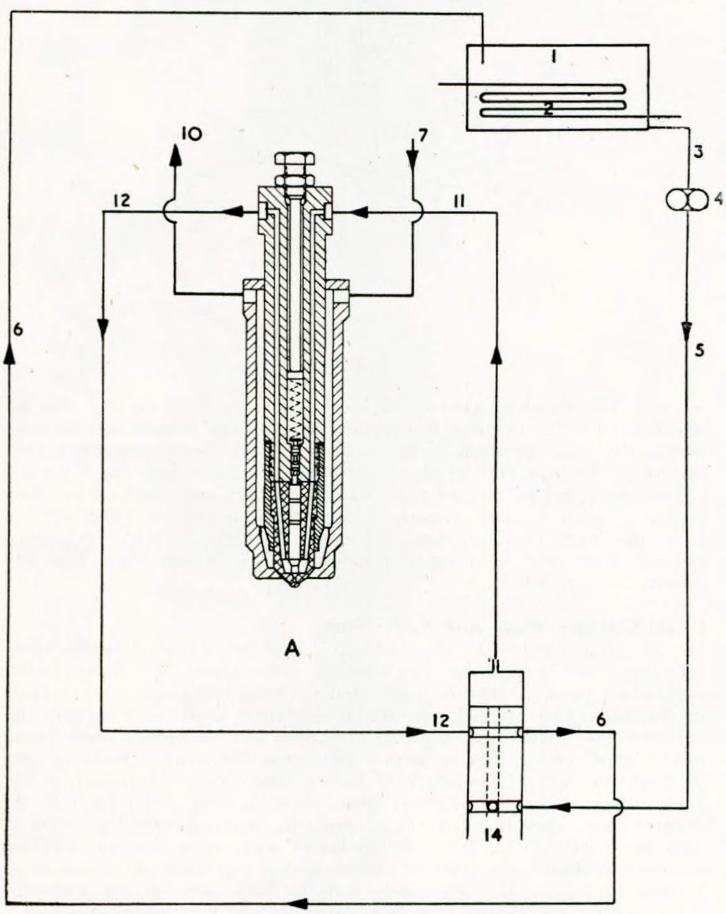
In a paper entitled "The Modern Trend in Tanker Construction", recently read before the Institute of Petroleum, Mr. John Lamb refers to a peculiar feature associated with the burning of boiler fuels in marine Diesel engines. With Diesel fuels, as is well known, the cylinder-liners wear in a tapered direction, the maximum wear being at the upper ends, whereas with boiler fuels the wear is more or less uniform throughout the length of the cylinder-liner. Moreover, it has been proved that the maximum wear when burning boiler fuels is not greater than when burning Diesel fuel, the actual figures being 0·0022 inch per 1,000 hr. When cylinder-liners wear in a tapered direction severe conditions are imposed on the piston-rings owing to the rings having to follow the constantly varying diameter of the cylinder. Also, when new piston-rings are fitted into a tapered cylinder-liner the gap or end clearance must be governed by that necessary when the rings are in the lower or least worn part of the cylinder, and this means that when the rings are in the upper or high-pressure part of the cylinder they are less effective as a gas seal. As the maximum wear rate is no greater when burning boiler fuels (and the present indications in the several Shell Group tankers operating on this fuel are that it is even less than when burning Diesel fuel), this peculiar feature when burning boiler fuel is not undesirable, since it might well result in the piston-rings giving longer service and in a reduction in the amount of gas leakage past the piston-rings. As to the cause of this unusual feature the following theory is put forward. With Diesel fuels, cylinder-liner wear is mainly due to abrasion, but when sulphur-containing fuels are used a proportion of the enlargement may be due to corrosion. In the form or sulphurous fumes, which is the condition in the upper and hottest part of the cylinder, no detrimental effect will result from its presence, but when the expanded gases reach the lower and cooler end of the cylinder the $SO_3$ gas forms sulphuric acid ($H_2SO_4$), which condenses on and corrodes the lower part of the cylinder wall. Most of this corrosion will take place when the engine is stopped or operating at comparatively low working temperatures, such as when manœuvring in and out of port.—*Lloyd's List and Shipping Gazette, No. 41,884, 10th March 1948, pp. 4-6.*

**Werkspoor-Lugt Two-stroke Engine**

The author reviews the considerations which led to the design and construction of this new type of supercharged two-stroke Diesel engine. In describing the engine the author reveals a number of important details, hitherto undescribed, such as the valve gear and some new fuel-pump arrangements. It is stated that the new engine is able to run on the same heavy fuel as the *Auricola,* and on other fuels, some of which originate from the Middle East. One of the problems to deal with when running an engine on heavy fuel is how to keep the fuel warm under all circumstances of the ship's speed. The circulating of the fuel from the heated service tank to the h.p. fuel pumps and back to the tank can easily be arranged on all types of pumps. The heating of the water-cooled injector holders in the case of slow-running and stopped engine is also a simple matter. The only part of the fuel system that causes further difficulties in this respect is the delivery pipe from the h.p. fuel pump to the injector, which often is lagged for this purpose. The well-known Pilgrim system provides a very nice solution to this problem, although it is only applicable to a certain type of pump. The new pneumatic fuel pump of the Werkspoor-Lugt engine incorporates some very suitable features for the application of a circulating system for the fuel between h.p. pump and injector. The system is shown in Fig. 29 (a) and explained as follows. The fuel flows from the service tank (1) (with heating coil (2)) to the l.p. circulating pump (4) and is delivered through the supply pipe (5) to the h.p. pump. From there it circulates through the delivery pipe (11) to the injector and from there again to the h.p. pump through the return pipe (12), from which it returns to the service tank. In Fig 29 (b) the passage through the h.p. fuel pump is shown more closely. The pump plunger (14) is traced in its bottom position where it stays during the major part of the cycle and where it always remains when the engine is stopped. The fuel enters at (15) and flows through the radial hole (16) and the centre hole (17) in the plunger to the pumping space and from there to the delivery pipe (11). Returning from the injector the fuel enters the pump body at (18), flows through the circumferential groove (22) in the plunger, leaves the pump body at (19) and returns to the tank through pipe (6). When the plunger is lifted for the working stroke the holes (15) and the passage through groove (22) are closed and fuel is injected in the cylinder. The passages in the injector are shown in Fig. 29 (c). By this circulating system the fuel will be kept warm and fluid under all circumstances. The absence of a valve in the delivery pipe is no matter of importance because the distance from h.p. pump to injector is very small so that the influence of the pressure relief after the injection is not considerable.. The injector body and the nozzle are cooled by water entering at (7) and leaving at (10).—*Paper by F. G. van Asperen, read at a meeting of the North-East Coast Institution of Engineers and Shipbuilders, 25th March 1949.*

**Marine Water-tube Boiler with Divided Furnace**

This invention relates to watertube boilers and heating units for use in marine propulsion plants of the type in which reheating between stages of the prime mover is employed. In such a power plant the superheating requirements are complicated, since during ahead operation one degree of superheat together with a degree of reheat are required, while during astern operation no reheat and the same, or, generally, a reduced degree of superheat are required. Steam generated in the boiler illustrated is separated from water in the separators (45) in the drum (1) and then passed through a steam pipe (41) to the header (25) of the first superheater section (24), from whence it is passed through pipe (42) to the header (32) of the second superheater section (30), from whence it is then conducted to the prime mover. The lower drum (3) contains desuperheating pipes (43) into which steam from pipe (42) may be directed by opening the



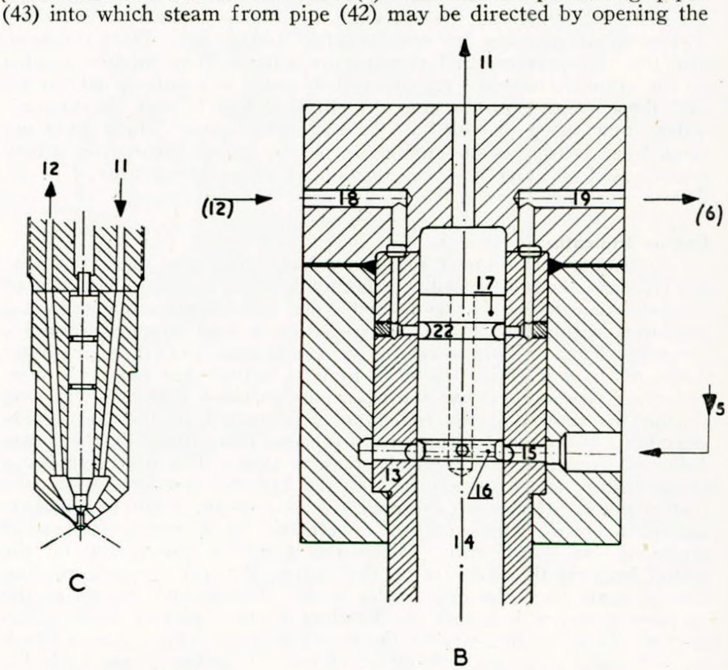
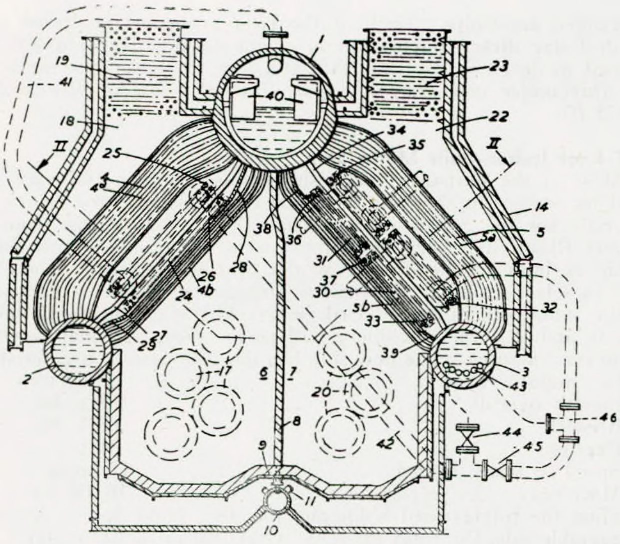




FIG. 29.—*Circulating system for heavy fuels*



hand valve (45). Upon cooling in the desuperheater coils the steam is returned to pipe (42) through the T-piece (46). By adjusting the valves (44 and 45), the steam entering the second superheater section can be attemperated to the desired degree, so that the steam leaves the second superheater at the required temperature. During ahead operation of the boiler both furnace chambers are fired and the rate of firing furnace chamber (7) below the reheater (31) is adjusted so that the reheater heats steam between stages of the prime mover to the required temperature. The total rate of firing of the two furnaces is so adjusted that the boiler produces the required steam flow at the required pressure, while the steam which is first superheated in the first superheater (24) and again heated in the second superheater (30) is brought to the required degree of superheater for use in the prime mover by adjusting valves (44 and 45) to pass a controlled proportion of the steam through the desuperheater (43). During astern operation no reheat is required, and the furnace (7) below the reheater (31) is not fired and the rate of firing furnace (6) alone is adjusted to produce the required steam flow at the required pressure, while the first superheater section (24) superheats the steam to the required temperature which, if required, may be attemperated by adjusting valves (44 and 45). It may be arranged, for example, that about 40 per cent of the total oil burnt shall, at maximum continuous rating for ahead operation, be burnt in the furnace (6), the steam reaching 620 deg. F. in the superheater section (24) and attaining without attemperation a final temperature of 850 deg. F. in the second superheater section. The reheater may reheat steam between stages of the prime mover from 590 deg . F. to 850 deg. F. During full speed astern operation the steam temperature attained in the first superheater section (24) may then be 800 deg. F..—Brit. Pat. No. 609,674 issued to Babcock and Wilcox, Ltd., and R. E. Zoller. Complete Specification accepted 5th October 1948.—*The Shipping World, Vol. 120, 9th March 1949, p. 302.*

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**Large Exhaust Gas Boilers**

The Clarkson exhaust gas boiler installed in the passenger liner *Angola* is one of the largest yet constructed, and a similar unit is being fitted in the sister ship *Mocambique* now being built by Swan, Hunter and Wigham Richardson. This vessel is launched and will shortly be completed. The boilers are 10ft. 3in. in diameter, with a height of 22ft. 6in. and have a heating surface of 2,400 sq. ft. The exhaust gases of both engines pass round the boiler, which is of the concentric composite type, but the gases from each engine are kept separate throughout their entire pass in the boiler, while simultaneous oil firing can be carried out in the centre of the boiler, if required. Normally, the whole of the services at sea can be taken care of by the exhaust gases only. The total estimated evaporation of the boiler is 12,000lb. of steam per hour at a working pressure of 100lb. per sq. in.—*The Motor Ship, Vol. 29, March 1949, p. 495.*

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**Marine Boiler Deterioration**

This paper comprises a survey of the several forms of boiler corrosion and deterioration encountered in marine boiler practice. Problems of design, operation and maintenance at sea are discussed in relation to metallurgical and electro-chemical phenomena involved. As regards water side-corrosion, pitting is more frequently encountered. "Soft Scab" pitting occurs mainly in superheater headers, superheater tubes, main steam piping, and on any steel components above the water line in the boiler liable to come in contact with water which has primed from the boiler. When such priming occurs, the water deposited will evaporate, leaving salts on the surfaces concerned. Under humid conditions, such as will exist when the boiler is shut down, droplets of condensate may form on these metal surfaces and will then dissolve any salts in their immediate vicinity, producing small droplets of quite concentrated electrolyte. This will give rise to intense local attack at favourably disposed areas. The initial and final stages in the development of a soft pit are illustrated diagrammatically in Fig. 11 from which it can be seen that electrolytic attack is initiated by a difference in potential between the inside and outside of the droplet, consequent on an oxygen content gradient. Subsequently, corrosion products forming at an intermediate position produce a semi-permeable membrane and establish the corrosion cell. Final products of electrolysis comprise black ferroso-ferric oxide ($Fe_3O_4$) on the inside and fully oxidized ferric oxide ($Fe_2O_3$) on the outside. Subsequent steaming will dry out moisture from the corrosion cap and further oxidize the iron compounds present to the "red rust" ($Fe_2O_3$) condition; this is a frequent characteristic observed in superheater tubes where operation has been intermittent. General wastage, corrosion fatigue, strain age embrittlement, and caustic embrittlement are in decreasing order of importance. Other forms of deterioration include high-temperature oxidation, bursts, high temperature creep, and corrosion on the fire-side. Several of each of

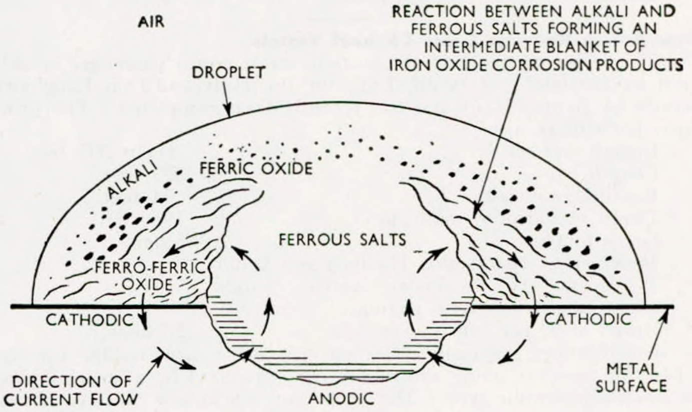


FIG. 11—*Formation of a soft scab pit*

the phenomena are illustrated in the paper and basic principles involved are discussed.—*Paper by I. G. Slater and N. L. Parr read at an Extra General Meeting of the Institution of Mechanical Engineers on 4th February 1949.*

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**Machine for Cleaning Metal Surfaces**

An American firm has developed a machine for cleaning metal surfaces by a process which can be regarded as a development of sand-blasting. Grit is directed through a "gun", on to the surface to be cleaned, and is then sucked back through the gun, together with the scale or paint removed from the surface, into a reclaiming tank. Here the dust is separated from the grit, and it is then passed into a dust collector. An alternative gun is provided for use at right-angled intersections. The machine consists of three units: the gun, the reclaiming tank, and the dust collector. Although the machine will operate at air pressures as low as 60lb. per sq. in., the rate of cleaning is considerably lower than at 85lb. per sq. in. The makers claim that up to 75 feet length of hose between the reclaimer and gun can be used. They also claim that the dust collector unit may be as much as 200 feet from the gun. For work on steel plate surfaces, a hard steel grit is recommended, of which 70 per cent will pass through a 0·028 inch-mesh screen. For efficient operation, the air supply must be dry, the surface must be reasonably dry, and it must not be raining. The tool would appear to be of most use on superstructures and other exposed steel surfaces of vessels which are to be laid up for a considerable period. Although for new shipbuilding and for ships in commission its general adoption would probably not be justified, it would probably lead to economy if it were used in areas of the upper works and the internal structure where access is difficult and sweating is likely.—*Admiralty Corrosion Committee, Hull Corrosion Sub-Committee. ACSIL/ADM/48/427. Journal of The British Shipbuilding Research Association, Vol. 4, March 1949, Abstract No. 2,527, p. 156.*

**Economy of Steel and Cast Iron by Welding**

The report of a special committee of the Institute of Welding includes a statement on the possible economies in steel that can be effected in marine engineering by changing over from riveting to welding. According to the statement made by Mr. J. A. Dorrat of the North Eastern Marine Engineering Co. (1938), Ltd., there is no great quantity of riveted work in marine engineering which can be readily changed over to welding. In steam installations on the light plate work associated with funnels, boiler uptakes, the boiler air heaters and ventilators, a certain amount has been done. The amount varies with the size and type of vessel, so that an average cannot be given. However, the following, which may be a guide, was taken from the machinery installation for an 8,000-10,000 ton cargo vessel:—

| Description. | Installation weight, tons. | Weight saving, tons. | Percentage saving. |
|---|---|---|---|
| Funnel ... ... | 4 | 0·6 | 5·5 |
| Ventilators ... ... | 11 | 0·2 | 5·0 |
| Boiler uptakes ... | 7 | 0·4 | 6·0 |
| Boiler air heaters ... | 6 | 0·5 | 8·0 |

From a review of figures taken on various jobs of this type Mr. Dorrat finds that the saving in material due to change-over from riveting to welding is between 5 and 8 per cent.—*Transactions of the Institute of Welding, Vol. 12, February 1949, pp. 3-5.*

**New British Railways Cross-Channel Vessels**

The *Hibernia*, the first of two twin-screw motor passenger vessels built by Harland and Wolff, Ltd., for the Holyhead-Dun Laoghaire service of British Railways, has recently been completed. The principal particulars are:—

Length overall ... ... ... ... ... About 397 feet
Length b.p. ... ... ... ... ... 375 feet
Breadth moulded ... ... ... ... 54 feet
Depth moulded to main deck ... ... 19ft. 6in.
Gross tonnage ... ... ... ... ... 5,200
Propulsion: twin-screw Harland and Wolff-B. and W. 8-cylinder, 2-stroke single-acting trunk-type engines ... ...
Speed at 90 per cent power ... ... ... 21 knots

A semi-balanced streamlined rudder aft and a bow rudder for use when the vessel is going astern are each operated by a steering gear of electric-hydraulic type. The propelling machinery consists of two trunk-type, two-stroke, single-acting, Harland-B. and W. Diesel engines of the builders' latest design with exhaust pistons of the same diameter as the working pistons. There are eight cylinders per engine, each having a diameter of 530 mm., the main stroke being 820 mm. and the exhaust piston stroke 360 mm. The engines are enclosed, forced-lubricated, direct reversing, and arranged for airless injection of fuel, and are coupled direct to the propeller shafting. They are conservatively rated, 90 per cent of the full power being sufficient to propel the vessel at 21 knots. The bedplates and frames are of fabricated steel throughout and the thrust blocks are incorporated in the after end of the bedplates. The cylinders are fresh-water cooled and the main pistons and the exhaust pistons are oil-cooled from the forced-lubrication system. The cooling of the lower part of the cylinder liners is done by the surrounding air in the scavenge belt. Scavenge air is supplied to each cylinder by four rotary blowers fitted to the back of the engine and driven from the crankshaft by roller chains.—*The Shipping World, Vol. 120, 23rd March 1949, pp. 341-343.*

**Twin-screw Cargo Motorship *Kaitangata***

The twin-cargo motorship *Kaitangata* is the first of a group of four sister ships ordered from Henry Robb, Ltd., of Leith, by the Union Steam Ship Company of New Zealand, Ltd. for the New Zealand coasting trade. The service is exacting, for not only does it involve the navigation of restricted waterways and the negotiation of difficult harbour entrances; there are, besides, offshore currents with which the ships have to contend. In view of these circumstances, the design of the hull and its appendages has been based on an extensive series of model experiments, carried out at the Ship Division of the National Physical Laboratory. In addition of the usual resistance and propulsion experiments, tests were carried out to determine the optimum dimensions and location of the propellers, the form and orientation of the shaft bossings, and the most suitable shape and area of the rudder. The non-balanced rudder is of relatively large area (about 80 sq. ft.), and is approximately square in outline. Apart from a moderate rise of the keel line at the extreme after end, the deadwood ahead of the rudder is retained. The propelling machinery is arranged amidships. Each of the twin propellers is driven by a British Polar direct-coupled Diesel engine having five cylinders and designed to develop 725 b.h.p. at 220 r.p.m. in continuous service.—*The Shipbuilder and Marine Engine-Builder, Vol. 56, March 1949, pp. 175-179.*

**A 17-knot Italian-built Motor Ship**

Most of the post-war motor ships constructed in Italy are propelled by engines of the Fiat type, and the M.S. *Sumatra*, which has run trials for the Swedish East Asiatic Co., Gothenburg, from the Cantieri Riunti dell'Adriatico, Monfalcone, is equipped with Sulzer machinery built at the Trieste works of the company. There are three vessels of similar dimensions and tonnage under construction for the same owners at the Eriksbergs yard at Gothenburg, but in these B. and W. type machinery will be installed, of equal power to that in the *Sumatra*. The new ship has the following characteristics:

Deadweight capacity ... ... ... ... 10,000 tons
Length overall ... ... ... ... ... 507ft. 4in.
Breadth ... ... ... ... ... ... 65ft. 4in.
Depth ... ... ... ... ... ... 31ft. 3in.
Speed on trials, loaded ... ... ... ... 17 knots
Machinery ... ... ... ... ... ... 10,000 b.h.p.

Including the refrigerated holds and the deep tanks for the carriage of vegetable oils, the total capacity is 630,000 cubic ft. (bale). The two engines are of the single-acting Sulzer standard design, each with seven cylinders, and at 130 r.p.m. they develop 5,000 b.h.p. each. The bore is 720 mm. and the stroke 1,250 mm.—*The Motor Ship, Vol. 29, March 1949, pp. 474-475.*

**Train Ferry *Mongibello***

A new train ferry was recently delivered by the Cantieri del Tirreno to the Italian State Railways for operation in the Messina Straits. This vessel, the M.S. *Mongibello* has a length of 90·5 m. b.p. and a breadth of 12·0 m., the displacement being 2,500 tons. Each of the two screw propellers is driven by a six cylinder two-stroke cycle single-acting Tosi Diesel engine of 1,250 s.h.p. at 230 r.p.m. There are three auxiliary Diesel generator sets of 100 kW. each. On trials fully loaded the vessel attained a sped of 14 knots.—*La Marina Italiana, Vol. 46, November-December 1948, pp. 182-183.*

**Mass Production of River Barges**

The reconstruction program of the French river fleet includes the building of 600 river barges at Strasbourg. 150 barges will be built by the Societe des Chantiers et Ateliers du Rhin, another series of 150 barges will be constructed by the Forges de Strasbourg, who will also assemble 300 barges prefabricated in three parts of Koenigshoffen and transported over a specially provided railroad line of 20 km. length. The first barge of 38·5 m. length, 5 m. breadth and 2·6 m. draft was recently launched.—*E. Renaudin, Journal de la Marine Marchande, Vol. 31, 17th March 1949, p. 448.*

**French 2,600 Ton Motor Cargo Vessels**

The reconstruction program of the French mercantile fleet includes the construction of eleven motor cargo vessels. Of these vessels three are built by the Chantiers et Ateliers de Provence at Port-de-bouc, two by the Arsenal de Lorient, and six by Marine Industries, Ltd., of Montreal. The six vessels built in Canada were delivered during the third quarter of 1948. One vessel, the *Tell*, built by the Arsenal de Lorient was delivered in December 1948, and one other, the *Atlas*, was launched in January 1949 at the Chantiers de Provence where another vessels of the series, the *Touggourt* will be launched this spring, to be followed by the third vessel, the *Auris* in autumn. The vessels of this series are of the shelterdeck type with an overall length of 95·345 m. and a moulded breadth of 14 m.; displacement when loaded is 4,385 tons and deadweight is 2,525 tons. The vessels built in France are equipped with Sulzer Diesels developing 3,000 s.h.p. at 150 r.p.m. The engines for the *Atlas*, *Touggourt*, and *Auris* were built by the Ateliers et Chantiers de la Loire at Saint Denis, while those for the *Tell* and *Tafna* were built at Winterthur. The French-built and the Swiss-built engines are substantially of the same design, though the Swiss-built units are equipped with oil cooled pistons instead of water cooled pistons. The Canadian-built vessels are furnished with Nordberg Diesels of Canadian manufacture. On trials the *Tell* achieved a speed of 14·64 knots with 2,630 s.h.p. at 148·4 r.p.m., displacement being 4,043 tons and deadweight 2,188 tons. At the speed of 14·13 knots with a deadweight of 2,188 tons, the fuel consumption was found to be 26·95 kg. per sea mile, based upon a a gas-oil having a lower calorific value of 10,030 calories per kg. The *Tafna* is to be fitted with a variable-pitch propeller of the Kamewa type.—*Journal de la Marine Marchande, Vol. 31, 10th March 1949, pp. 407-411.*

**New Tankers**

Three new tankers, soon to be built for Philadelphia Tankers, Inc., will not only be among the largest afloat, but will also employ special and unusual equipment. The ships themselves will be 625 feet long, and will carry almost ten million gallons of oil, enough to fill a line of tank trucks fifteen miles long. The propulsion unit will be a 16 500-s.h.p., 1,000 deg. F. geared turbine. This temperature rating is about 25 per cent larger than that of usual propulsion units, few of which operate at temperatures much in excess of 750 deg. F. Other equipment includes two 750 kW. a.c. turbine generators, a 700 kW. geared generator driven by the main turbine, boiler feed-pump turbines, transformers, switchboards, and motors.—*Westinghouse Engineer, Vol. 9, March 1949, p. 33.*

**Diesel Fireboat**

The 75-feet fireboat *Bernard Samuel* recently acquired by the City of Philadelphia is equipped with four pumps having a combined capacity of 6,158 U.S. gals. per min., at 150lb. per sq. in. pressure, when operated individually. With all pumps performing simultaneously when the vessel was on trials, 5,875 U.S. gals. per min. at 150lb. per sq. in. were delivered through six 2-inch nozzles. Average speed of the vessel was better than 17 m.p.h., the propulsion plant being an 800 h.p. General Motors Quad-6 Diesel. This consists of four six-cylinder engines mounted on a common base and driving the propeller shaft through a single 4 to 1 reduction gear. Due to the fact that the engines can be individually de-clutched and turned to pumping duty as the need demands, it was only necessary to install one engine plant to handle both pumping and propulsion power requirements. Each engine in the General Motors Quad-6 Diesel unit is equipped with a 166 b.h.p. heavy duty front power take-off with Rockford clutch and drives a separate De-Laval two-stage centrifugal pump. The De-Laval pumps have a capacity of 1,500 gals. per min. at 150lb. per sq. in. and operates at 1,800 r.p.m.—*Pacific Marine Review, Vol. 46, February 1949, pp. 37-39.*

**Shipbuilding in Sweden in 1948**

This article contains summary data on the output of Swedish shipbuilders, comprising some eighteen yards, during the year under review. Total construction in Sweden in 1948 amounted to forty-seven vessels totalling 431,700 tons d.w. and 294,000 tons gross, with 348,000 i.h.p. It is reported that only three vessels totalling 8,560 tons d.w. and 6,800 tons gross with 8,200 i.h.p. are equipped with steam propulsion plant.—*N. J. Ljungzell, Teknisk Tidskrift, Vol. 79, 19th March 1949, pp. 203-214.*

**Launch of the Wine Tanker *Bacchus***

The wine tanker *Bacchus*, which was recently launched from the yards of the Rotterdamsche Droogdak Mij., is intended to replace two vessels of this class lost during the war, the French owners being the Socièté Soflumar-Vins. The *Bacchus* has a capacity of 36,500 hectolitres of wine or alcohol. Entirely new principles have been followed in the design of this vessel, which was built under the special survey of Bureau Veritas. The principal data are: 101·8 m. length o.a.; 14·4 m. breadth; 5·88 m. draft; approximately 3,400 gross tons. The engine is a 3-cylinder Doxford Diesel of 2,100 b.h.p. at 110 r.p.m., giving the vessel a speed of 12½ knots. It was built by the Wilton works at Rotterdam, and is the first unit of this type built on the Continent.—*Journal de la Marine Marchande, Vol. 31, 3rd March 1949, p. 367.*

**New Icelandic Motor Trawler**

The motor trawler *Hallveig Frodadottir,* recently completed by the Goole Shipbuilding and Repairing Co., Ltd. for the Icelandic Government, is believed to be the largest motor trawler of its type. This, with three identical vessels now under construction, is part of the extensive new building programme undertaken by the Icelandic Government in the past two or three years. The principal features of the *Hallveig Frodadottir* and the *Jan Thorlaksson,* now fitting out at Goole, are as follows:—

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|---|---|
| Length b.p. | 170 feet |
| Breadth moulded | 29ft. 6in. |
| Moulded depth | 15ft. 6in. |
| Mean draught (light) | 11ft. 9in. |
| Speed on trials | 13·32 knots |

The double-barrelled electric trawl-winch, supplied by Clarke, Chapman and Co., Ltd., is designed for a duty of seven tons at 430ft. per min. from the top layer of rope from one barrel only, or 3½ tons from each barrel simultaneously. From the warping ends, it will pull 2½ tons at 180ft. per min. from one warping end at a time or 1¼ tons from each warping end simultaneously. However, when the ship is in port, the duty from the winch is: 2½ tons from one warp end at 180ft. per min.; 1¼ tons from both warp ends at 180ft. per min. When at sea, the power to the winch motor is supplied from a 220 kW. Ward-Leonard controlled generator driven by the engines but, when in port, the power is supplied from one of the 50 kW. generators which can be used either for this purpose or for feeding the ship's mains in parallel with the other 50 kW. set and the 15 kW. set. The main propulsion engine is a Ruston Mk. 5 VOXM four-stroke, single-acting, unidirectional, five-cylinder Diesel engine of the latest design. Pressure charging on the Buchi principle is employed, the pressure charger being supplied by Brown Boveri, Ltd. This engine develops 1,100 s.h.p. continuously at 420 r.p.m., and drives through a Vulcan-Sinclair fluid coupling of the scoop control type, manufactured by Hydraulic Coupling and Engineering Co., Ltd., of Isleworth, to an S.L.M. oil operated, reverse reduction gear of 4 : 1 manufactured by Modern Wheel Drive, Ltd. In addition, a drive is taken through an S.L.M. oil-operated clutch to a generator at the forward end of the engine for operating the trawl winch. Apart from eliminating the transmission of torsional oscillations from the engine to the gearing and vice versa, the fluid coupling enables the propeller speed to be reduced while the engine is kept at full speed to drive the generator serving the trawl winch. When manœuvring from ahead to astern or vice versa, the scoop tube is automatically withdrawn and the engine speed reduced in order to lessen the work on the oil-operated clutches of the reverse gear. On trials, a speed of 13·5 knots was attained, the engine developing 1,210 b.h.p. at 435 r.p.m., and a propeller speed of 107 r.p.m.—*The Shipping World, Vol. 120, 9th March 1949, pp. 299-302.*

**U.S. Navy Plastic Boat Programme**

The difficulty of procuring sufficient quantities of plywood and well seasoned, air dried lumber for an emergency boat building program led the Bureau of Ships to investigate the possibility of utilization of other materials. Metals have been used successfully for small boat construction, e.g. in lifeboats used by the Merchant Marine. Nevertheless, due to weight restrictions, the best metallic boats have, in spite of stiffening incorporated in the manufacturing process, had such thin skins as to make them unacceptable for Navy use. It appeared that a finished boat of plastic would weigh less than either a wood or a metal boat of the same dimensions, that there would be less framing and that the skin panel strength would be greater. A plastic boat would also be a monolithic structure easily repaired if damaged, not subject to those weaknesses of timber construction—failure of fasteners on account of fatigue around screws, nails and bolts; dimensional instability and distortion, dry rot and borer activity. Investigation also revealed the tremendous savings in equipment, time and money, if in another emergency boat building programme, plastic fabrication took precedence over the traditional material and construction method. The first boat of laminated plastic had a length overall of 28ft. 11in. and a beam (moulded) of 10ft. 3in. It was made in 24 hours, 10 men participating, but this time was reduced with subsequent hulls. The article describes the type of mould employed and the procedure followed in fabricating the hull. The plastic hull incorporates some twenty layers of fibrous glass mat, as tests had shown that fibrous glass mat and glass cloth have the highest tensile and flexural strength—specific gravity ratio. The hull weighed 2,666lb. In forecasting further developments, the author states that there is promise of much stronger glass laminates becoming available. The cost of glass cloth is, however, high; and it may be feasible to sacrifice strength to some degree and to sandwich other materials. It has been found that the use of cotton duck as an outer skin laminate provides better resistance against abrasion than glass laminates. Resin content is approximately 50 per cent by weight of a finished laminated structure. The moulds are expensive and difficult to make, but will last indefinitely. The author expresses the opinion that plastic construction could be applied to larger boats as small vessels. At the present time craft up to 110 feet in length could be produced efficiently, economically and expeditiously on a mass production basis.—*Commander A. C. Bushey, Jr., Journal of the American Society of Naval Engineers, Vol. 61, February 1949, pp. 16-22.*

**Sulphur in Diesel Fuels**

The authors give a brief historical account of the recognition of the undesirable effects of the sulphur content of Diesel fuels. Notwithstanding that it is, technically, quite feasible to desulphurize the fuels, other considerations compel the acceptance of a significant sulphur content for some time to come, and it may be expected that, as a result, engine wear and fouling will be accentuated unless appropriate precautions are taken. Engines of different types and design will be differently affected. An account of relevant physico-chemical reactions of sulphur in the engine focuses attention on the trioxide produced during combustion as the principal cause of ill effects. Amongst precautionary measures, certain beneficial rules of engine operation (particularly as regards coolant temperature) and design are proposed. Importance is attached also to metallurgical expedients such as chromium plating, and to the use of special additive-type lubricants. It is thus made evident that by suitable design, operation, and lubrication of the Diesel engine, it can be made to consume fuels of a higher sulphur content than has hitherto been usual, without adverse results. Instances of the satisfactory use of high sulphur fuels are given.—*Paper by J. J. Broeze and A. Wilson, read at a General Meeting of the Automobile Division of The Institution of Mechanical Engineers, 8th March 1949.*

**Filtration of Fuel and Lubricating Oils**

The internal-combustion engine presents the most difficult problem of keeping the lubricating oil free of abrasive material. Excessive wearing of pistons, rings, and cylinder liners is noticed in internal-combustion engines. This excessive wear is accompanied often with high fuel and lubricating-oil consumption. This wearing forms small particles of metal which mix with carbon from unburned fuel and partially burned lubricating oil and makes a gummy deposit which is the cause of ring sticking. It is essential, also, to filter the fuel oil as a means of maintaining correct injector operation. The contaminants or particles which form the deposits in the oil passages and inside an engine are the results of sedimentation and area of the size of particles which may be removed by filtration. Additives in the oil keep these small particles in suspension and prevent them from forming on the inside of the engine. Filters using absorptive elements are recommended for filtration of additive oils. High flow rate through a filter is obtained by connecting the filter in the engine lubrication system as full flow, that is, the entire output of the lube-oil pump will flow through the filter shell. Such a method is also called the shunt method. Filters connected as full flow should have a by-pass relief valve in the filter elements or installed in the filter shell which acts as an automatic pressure-relief valve. Lubricating oil filters should be connected as close to the engine as possible as hot oil tends for better filtration. Cotton waste has proved quite satisfactory for many years as a filter medium. The cotton waste should be of the long-fibre, unbleached type. Waste which contains fibres of rayon and scrap materials should not be used. As cotton waste has an affinity for moisture the cotton-waste type filter does much in maintaining a low neutralization number of the oil by preventing the moisture content of the oil from mixing with the sulphur content of the oil. In general, the application of oil filters to an engine is to maintain a clean operating engine. There is no great problem of filtering oil but to keep an engine clean effectively does present the problem of selecting an oil filter of such size which will remove foreign matter from the oil at the rate it is formed. It is also necessary to clean the filter when the dirt formation in the oil starts to accumulate beyond the dirt-holding capacity of the filter.—*Paper by F. L. Townsend, read at the 1948 A.S.M.E. Petroleum Division Conference, Paper No. 48-PET-10.*

**Two-phase Fuel Injection**

The Kammer pilot fuel-injection system injector is shown in Fig. 1. It is actuated by standard fuel injection pumps having normal camshafts. The fuel intake is at (A) and is led to a small cavity (C) via the fuel line (B), entering the annular space (D) formed by a control plunger (E) and its barrel (F). This annular space (D) communicates with a pilot spring-loaded checkvalve (H) which is opened by fuel under pressure. In Fig. 2 an enlarged section is shown. Fuel can flow past the checkvalve (H) and the small channel (W) into (J) located in the selector screw (K) and continuing in the selector pin (L); it is eventually discharged through the pilot orifices located in the nozzle teat (M). The control plunger (E) is loaded by a control spring (G); the strength of that spring is planned to yield under the pressure created by incoming fuel from the fuel-injection pump coupled with the limited discharge jet area provided for pilot injection. This spring is, therefore, compressed by the fuel acting on the control plunger (E) which recedes simultaneously. In this connexion it must be understood that the fuel pressure cannot exceed a value determined by the control plunger, because this involves an immediate further lift of the control plunger, nor can it fall below that value because that would cause the plunger to drop again. Eventually the plunger (E) uncovers a port (N) in the barrel (F) and permits fuel to enter the lead (O), open the main valve (P) and reach, via lead (Q), flutes (R) and the pocket (S) in the bulge of the nozzle above the teat; therein are situated teh main orifices, which are connected to the pilot orifices in the discharge of fuel. Up to the end of the desired injection period both sets of orifices continue to inject fuel, but whilst the injection pressure during the pilot period was governed by the control spring, the pressure of main injection is only governed by the fuel-injection pump cam contour, as in conventional injectors, because the plunger (E) is stopped against the lower face of the nozzle holder (T). When spill occurs in the fuel pump and pressure drops under the control plunger, the valves (P and H) snap shut and the plunger returns to the position shown in

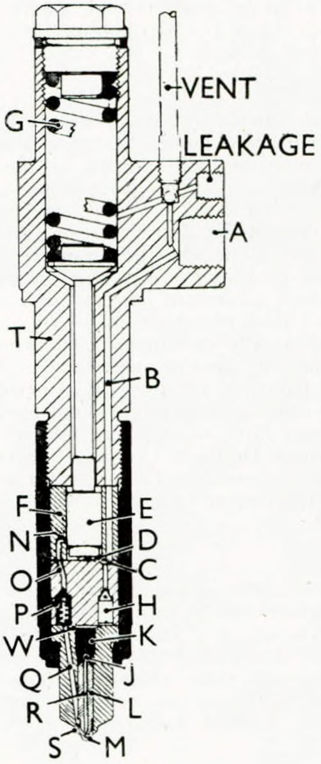



Fig. 1.

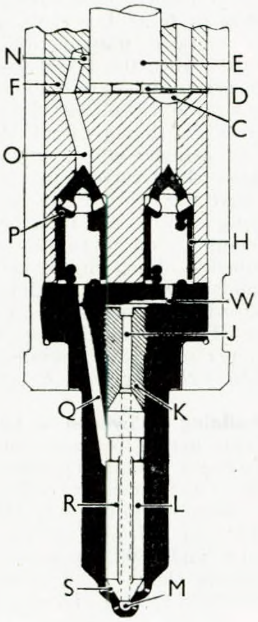



Fig. 2.

the Figs. 1 and 2. A special delivery valve at the pump end provides line-pressure control and permits fuel back-flow from under the plunger (E). The valves (H and P) are forced closed by small springs located within them, but they are also forced on to their seats by the cylinder gas pressure which acts on their lower surfaces through pressure exerted upon the fuel by way of the nozzle orifices. This is claimed to be most beneficial in providing a sharp, clean, fuel cut-off and in preventing secondary pressure waves (which occur during a period of highest cylinder pressure) from re-opening the valves.—*The Oil Engine and Gas Turbine, Vol. 16, March 1949, pp. 388-389.*

**Oxidation of Turbine Oils in the Laboratory**

In connexion with the introduction of inhibited turbine oils into Naval Service the behaviour of typical oils under oxidation test I.P. 114/47 has been observed. This test method was devised by a panel of the Lubricating Oil Pool Technical Committee. The duration of this test is 90 hours; and although it is very much shorter than some other test methods, it would be of considerable advantage if an even shorter test period could be devised. With this end in view, six typical turbine oils have been examined by the standard method as well as by a modified test. The possibility of substituting oxygen for air in this test is considered. The variations of the standard conditions were made in order to get a better understanding of the course of the reaction and hence to develop a more rapid test procedure, and also to collect information about the correlation of the oxidation test with the deterioration of oils in practice. In this connexion the significance of peroxide value and interfacial tension is discussed. It was found that the oxidation-corrosion test of Pope and Hull is not a suitable corrosion test for turbine oils, and the use of interfacial tension as a criterion of the extent of oxidation is not advised in view of the lack of correlation between interfacial tension and demulsification number. Promising results were obtained when the behaviour of four pre-selective gearbox oils in oxidation test I.P. 114/47 was compared with service experience.—*D. Wyllie and G. C. N. Cheesman, Journal of the Institute of Petroleum, Vol. 35, January 1949, pp. 61-72.*

**Fuel Oil Service System of Victory Ships**

The article discusses the type of fuel-oil service system installed in Victory ships designated VC2-S-AP2. These ships are equipped with main turbines developing 6,000 s.h.p. However, the fuel-oil service system installed in Victory ships of the VC2-AP3 design (those equipped with 8,500-horsepower geared-turbine propulsion machinery) is quite similar to the system described in this article, the major differences being ones of size and capacity. In addition to this specific information the article contains general information which provides material for a study of the operation of a simple fuel-oil service system.—*Marine Engineering and Shipping Review, Vol. 54, February 1949, pp. 52-53.*

**Cavitation Analysis by Electrical Analogy Method**

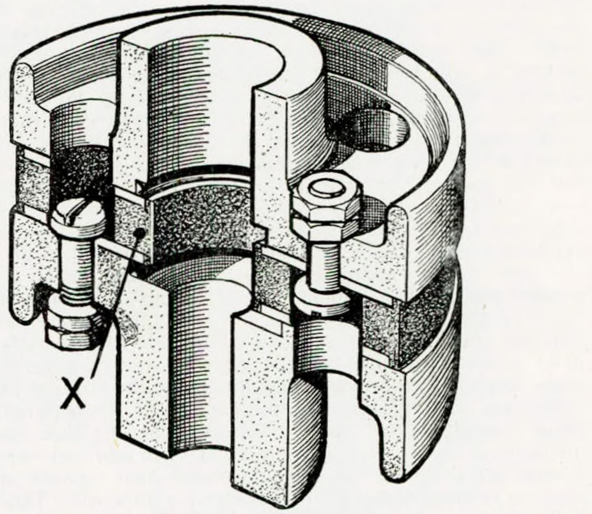
There exists no simple analytical method of designing for cavitation-free performance or even of predicting the performance of a given design. The electrical analogy method has been used quite frequently in the past for the convenient solution of problems in two-dimensional fluid motion. In principle it is based upon the fact that the flow of an electrical current through a homogeneous conductor follows the same pattern as the irrotational (i.e., non-viscous) flow of a fluid through a passage having a similar boundary form. Use of the electrical analogy method as described in the article makes it possible to study the pressure characteristics at any given boundary profile. The technique is also readily applicable to the study of many free-surface phenomena.—*H. Rouse and M. M. Hassan, Mechanical Engineering, Vol. 71, March 1949, pp. 213-216.*

**Analysis of the Contra-flow Regenerative Heat Exchanger**

The regenerative heat exchanger was first proposed and constructed by Dr. Robert Stirling in 1816, in connexion with a regenerative hot-air engine. Since then its use has been mainly confined to furnace air preheaters. With the advent of the gas turbine, however, the need has arisen for a compact heat exchanger of high thermal ratio and low pressure loss. This, of necessity, tends to the use of gas passages of small hydraulic diameter. In the recuperator, a limit to the size of the gas passages is imposed by the difficulties of producing and supporting in position long thin passages which must perform the dual function of transferring heat from one gas to another and of keeping separate the two gases at different pressures. Small gas passages in a recuperator also tend to become blocked with the deposits from the hot gases of a gas-turbine unit. The renewed interest recently taken in the regenerative heat exchanger is due to the promise which it shows of meeting these objections. The requirement that the gas passages must keep the gases separate is removed, so that these passages may, for instance, be merely the voids in any porous material or in any aggregate of elements. The manufacture and task of supporting in position such a "matrix" is comparatively simple. Information to date suggests that the tendency to blockage of the gas passages in a gas-turbine regenerator should be very much reduced because of the blasts of clean air which alternate with every gas discharge. So far as the author is aware, however, no regenerative heat exchanger has yet been set to work in conjunction with a gas-turbine installation. The paper makes no attempt to deal with pressure losses or the application of the regenerator to the gas turbine. The treatment of these aspects of the problem follows fairly conventional lines, once the thermal behaviour of the regenerator has been investigated.—*Paper by G. E. Iliffe, submitted to the Institution of Mechanical Engineers for written discussion.*

**Rubber-metal Flexible Couplings**

A British firm has developed flexible couplings of a new design which are available in two types, one known as the disk type and the other as the barrel type. Both have been specially designed to accommodate irregularities in power transmission due to load fluctuations or misalignment. These fluctuations may be large, due to shock loads, or they may be small, due to vibration. Metal-sprung couplings may sometimes be incapable of sufficient deflexion to accommodate large shock loads, and may not sufficiently reduce the transmission of high-frequency oscillations. Rubber, on the other hand, has inherent damping properties and provides ample flexibility to cope with all transmission conditions. The D.S.P. disk coupling is suitable for drives from 0·8 h.p. to 64 h.p. per 100 r.p.m. for oil engines, blowers, generators, compressors, pumps and many other applications. It is designed to withstand occasional shock loads up to one-and-a-half times the rated load and is made in sizes for shaft diameters up to 5 inch. The rated angular deflexion for this range of couplings varies from 3 to 5 deg., but under occasional shock loads the deflexion may be as high as 8 deg. A disk unit consists of a pair of robust steel circular plates between which is bonded an element of specially compounded rubber. The rubber is stressed primarily in shear to give large deflexions under torque loading, whilst the flexibility of the coupling reduces axial loads, resulting in improved bearing life. It also allows for misalignment conditions and can accommodate large irregularities in power transmission caused by load fluctuations. Bolt holes in the plates permit the coupling disk being readily assembled between metal flanges. For special applications, the elastic torsional stiffness of the coupling can be varied over a wide range by the use of specific rubber compound controlled within fine limits.—*The Oil Engine and Gas Turbine, Vol. 16, March 1949, p. 375.*




*Disk-type coupling. X indicates the rubber-bonded coupling disk sandwiched between two flanges*

**Testing of High Temperature Alloy Blades in Turbo-supercharger**

The authors describe briefly a series of jet tests, utilizing gas produced from the combustion of Diesel fuel oil, as a means for comparing the resistance of a number of high temperature alloys to hot gas impingement. The deficiencies of this method for simulating conditions in a gas turbine are discussed. The main part of the paper is concerned with comparative tests of a number of high temperature alloys when tested in the form of blades in a turbo-supercharger. The test rotor contained 142 blades representing twelve different alloys. Both wrought and precision cast blades were included. Tests were made at eight temperatures ranging from approximately 1,200 deg. to 1,500 deg. F., test runs at each temperature being of 50 to 150 hours duration except in the case of the 1,500 deg. F. test run which was continued for 1,000 hours. After several of the test runs, measurements were made to determine the amount of permanent extension in the blades and disk. The extension of the blades and disk accompanying progressively higher testing temperatures is shown graphically. A procedure for correlating the metal temperature of the blades with that of the combustion gas in the nozzle chamber of the supercharger is described.—*Paper by W. C. Stewart and H. C. Ellinghausen, read at the 1948 Annual Meeting of the American Society of Mechanical Engineers, December 1948.*

**Behaviour of Deck Plating after Buckling**

The author presents a simple theory of the behaviour of a panel of plating after buckling. An attempt is made to calculate the total stress in such a panel of plating and to ascertain when permanent set is likely to occur. The investigation is extended to the case where plating buckles over a large area. The results are compared with compressive tests on steel plates. A method of calculating the section modulus of the section of a ship whose deck has buckled is described. The question of degree of fixity of plating at beams is considered. The limitations of the theory are discussed and some general conclusions given.—*Paper by W. Muckle, read at a meeting of the Institution of Naval Architects, 7th April 1949.*

**Ventilation of Living Space in Ships**

According to the author there exists a simple and effective method, which can be applied in design, and whereby passengers and personnel can be assured of a greater measure of comfort than is possible by the observance of the prevailing principle which employs a "flat" rate of air change per hour. It can be claimed for this system—known as the balanced system—that the benefit derived from its adoption will be more than proportionate to the additional work entailed in design, and any extra cost incurred. In the balanced system recognition is afforded to the fact that the structure of a vessel is such that it offers less resistance to hea transmission (both inwards and outwards) than does the usual shore structure. Moreover, the extent of the exposed surfaces on board ship often carries considerably from one space to another, so that the amount of air supplied for heating or ventilating must be related to the amount and character of the surface exposed; the greater the area exposed, the greater must be the amount of heat (or air-containing heat) supplied to offset the losses in cold weather, even though the spaces may be adjacent and of similar capacity. Similarly, the quantity of air supplied in hot weather should be apportioned to the various spaces in accordance with the extent and nature of the surface exposed. The conclusion is, therefore, that the amount of air required is governed more closely by superficial measurement than by cubic capacity. The quantity of air to be supplied in cu. ft. per min. per person, is easily determined from the amount of sensible heat emitted by the human body, in relation to the permissible temperature rise. The sensible heat emission per person varies from 200 to 250 B.Th.U. per hr., and the value 220 B.Th.U. per hr. is commonly used. The number of persons in habitation and the amount of lighting installed contribute largely to the heat gain in confined spaces, and these factors should be taken into account in considering the ventilating or air-conditioning installations. Many vessels operate between arctic and tropical latitudes; and, while the provision of heating for the former does not present insuperable difficulty, conditions at and near the equator have occasioned increasing concern, and have resulted in a demand for air-conditioning. Without doubt, air conditioning is the ideal; but it is not the sole and inevitable alternative to the mechanical system of ventilation generally installed in ships. In the balanced system—admittedly not so effective as air-conditioning—the quantity of air supplied to each individual space is balanced against the aggregate rate of heat gain through transmission and emission from the various sources which have been enumerated, viz., the structure, lighting, occupants, etc., in order to control the rise of temperature within the space to a reasonable limit in the most unfavourable tropical conditions. For public rooms, such as dining saloons, air-conditioning is almost a necessity, because of the high percentage of casual heat gain from occupancy; but, in living spaces in which the heat gained by the structure is the main cause of radiation, the balanced system offers a solution which discounts the need for air-conditioning, and particularly if thermal insulation is applied to the interior surfaces of those parts of the structure which are more directly exposed to the heating effect of the sun.—*R. McDonald, The Shipbuilder and Marine Engine Builder, Vol. 56, March 1949, pp. 166-167.*

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**Aspirating Air Diffuser**

This article points out the importance of proper air distribution in air-conditioned spaces in order to secure maximum comfort. When cold air enters an area through conventional outlets—grilles, registers, or perforated panels—it usually sweeps to the floor and this forces the warmer room air to the ceiling. Until the velocity of the cold incoming air subsides, it cannot mix with the warmer room air. This results in drafty, turbulent conditions; temperature differentials are great; humidity is unequalized, and stagnant air pockets are prevalent. The author describes a case in which such unsatisfactory conditions were rectified by installing "aspirating" air diffusers. These devices are composed of a series of scientifically designed metal cones assembled in definite relation to each other. Air entering a space such as a cabin passes through these cones and, because of their unique design, is reduced instantly in velocity within the device. Simultaneously, air from the cabin—equal to about 35 per cent of the incoming air—is siphoned into the diffuser where it is mixed with the incoming air. The pre-mixed air then leaves the diffuser at a low velocity and spreads over a predetermined area well above the occupancy zone, and finally reaches this level of the cabin as a slow-moving low-pressure blanket. Because the primary air-mixing action takes place within the diffuser, and because all major air turbulence is limited to its immediate vicinity, no drafts are perceptible to occupants of the cabin. Because the air-mixture spreads slowly through the cabin instead of sweeping in, as it does when conventional fixtures are used at duct openings, obstacles such as columns, equipment, and furnishings do not deflect air flow.—*F. Honerkamp, Marine Engineering and Shipping Review, Vol. 54, February 1949, pp. 65-68.*

**Design of Refrigerated Ships**

The paper gives the result of experience gained over some years with this type of vessel. In referring to the seriousness of oil leakage, the author points out that with the introduction of Diesel machinery this leakage caused considerable trouble in way of double bottom oil-fuel tanks, and more especially at the bilges in way of refrigerated holds. With the earlier form of flanged margin plate and overlapping gusset plates it was almost impossible to obviate this oil leakage through the rivet connecting the gusset plate to the margin plate. A later arrangement was to carry the tank top plating flat out over the bilge, thus incorporating the gusset plate with the tank-top plating, the margin plate being fitted to this extension with a hooked angle bar which permitted better riveting and the caulking of the heel and toe of the vertical flange of the tank top margin bar in way of the bilge; this certainly gave much improved results. Nowadays the whole of this detail can be a welded construction, which appears to be the only satisfactory answer to the question of oil leakage in way of refrigerated cargo hold bilges. It is also good practice to arrange all seams and butts of the tank top of welded construction in way of double bottom oil fuel tanks. There is no doubt that cofferdams were necessary with riveted construction as it was impossible to prevent oil leaks through the allegedly oil-tight bulkhead. It would appear, however, that with the development of welding technique it may be possible to omit these cofferdams with advantage to the carrying capacity of the ship. It should be pointed out that all access manholes from refrigerated spaces to double bottom and deep oil-fuel tanks should be of double cover type to minimize oil leakage in way of these manholes.—*Paper by J. Baird, read at a meeting of The Institution of Engineers and Shipbuilders in Scotland, 22nd March 1949.*

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**Air Driven Ship Cleaning Brush**

An American firm of paint makers has developed a new wire brush driven by compressed air for cleaning ships' bottoms and boottop belts and other surfaces preparatory to painting. The brush itself is roller shaped, 12 inch in width, 6-inch diameter, and houses a 4/10 h.p. air motor. Almost the entire unit is made of aluminium and is very light. It is operated from the drydock bottom with tubular aluminium handles of varying lengths through which the compressed air is passed. The construction of the brush itself involves an entirely new principle. In earlier tests with a similar ship cleaning brush, flat spring steel wires were vulcanized in a rubber base. However, the severe flexing of the wires when passed over rivet heads, laps of plates, pitted plating, etc., caused crystallization and breakage of the wires. In the new design the brush itself consists of short flat spring steel wires attached to the ends of rubber straps. Therefore, as the brush revolves at about 2,000 r.p.m., the flat steel ends do the cleaning while the rubber straps provide for the flexing. The wires, of course, are held against the surface to be cleaned by centrifugal force, while the rubber straps allow complete flexibility. In operation, while it does not remove good adhering paint, it cleans all loosely attached paint and rust down to bare metal. The flexibility of the rubber straps, together with their stretching properties removes rust from river heads, welded seams and deeply pitted pockets, exposing the bare metal ready for priming and painting. The brush also removes every vestige of fouling, both grass and shells, and the flat wires remove any dampness from the surface of the paint, thus eliminating the need for hosing down when time does not permit.—*Pacific Marine Review, Vol. 46, February 1949, pp. 86-87.*

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**Fireproofing Wood and Deck Coverings**

The fire protection of wood can be carried out by surface coating or by impregnation. The former has the advantage of easy application, especially to timber ***in situ***, and offers high resistance to flame penetration and the spreading of fires, but impregnation with fire-proofing chemicals is much more effective. The best results are obtained with pressure impregnation, the most important process in this country being the "Oxylene" process which utilizes an aqueous solution of mono-ammonium phosphate containing a small proportion of boric acid. The proofed timber is, however, supplied in bulk, and when cut up or shattered the portions may be less fire resistant. Whenever possible, therefore, wood will be replaced by less inflammable substitutes. Any fireproofed material for deck covering must retain the typical physical properties of flexibility and resilience of the linoleum normally used. The most satisfactory fireproofed linoleum so far consists of a material in which part of the linoxyn is replaced by chlorinated resin and part of the filler by antimony oxide. This material has given good results in extended fire trials at the Admiralty Fire Testing Ground, Haslar, and it is proposed to submit it to sea trials to assess its durability and general suitability. The investigation of entirely new types of deck covering based on materials other than cork and linoxyn, or in which these ingredients are partly or wholly replaced by less inflammable substitutes, is being carried out. —*Admiralty, Dept. of Chief of Naval Information, Bulletin No. 12. Journal, The British Shipbuilding Research Association, Vol. 4, February 1949; Abstract No. 2,442, pp. 94-95.*

**Causes of Flue Gas Deposits and Corrosion in Modern Boiler Plants**

This paper is intended to be amplification and continuation of the author's previous paper entitled "Causes of High Dew-point Temperatures in Boiler Flue Gases" read in 1943, in which attention was drawn to the phenomenon of the catalytic production of sulphuric acid in flue gases by their passage over heated iron surfaces. At that time the full importance of these findings was not apparent, but evidence is now submitted to show that, in addition to causing deposits and corrosion by the deposition of acid in air heaters and economizers, this phenomenon is primarily responsible for the flue-gas deposits and corrosion in all parts of boiler plants. Experiments are described which show that when flue gases are passed over sand-blasted steel-surfaces which are maintained at the gradation in metal temperatures which can occur through a modern boiler plant, sulphur trioxide is produced at the high temperature surfaces, causing sulphuric acid to condense on the cooler surfaces. When the maximum surface temperature is moderated no appreciable formation of acid occurs. It is believed that the interaction of the sulphur trioxide in the generation zone, and the condensed acid in the cooler zone, with the various constituents of the fuel ash and the metal of the heating surface, can explain all the numerous deposit and corrosion effects which have been the subject of research work both here and abroad during recent years. The high surface temperatures which are necessary for the catalytic action to occur are largely due to the general advances in operating temperatures and pressures, and in some measure to "surface combustion" taking place on the tube surfaces, elevating their temperature above that which they would acquire if swept only by inert gases. Although the advance in steam and water temperatures is considered to be a primary factor in these difficulties, many other contributory factors, particularly the nature of the fuel, play an important part.—*Paper by W. F. Harlow, read at a joint meeting of The Institution of Mechanical Engineers and The Institution of Electrical Engineers, 4th March 1949.*

**Corrosion-erosion Tests on Ni-Resist in Sea Water**

According to a report published in *Nickel Topics,* Ni-Resist has for many years past been successfully used to provide resistance to the corrosive and erosive effects of salt water at high velocity, e.g., in such applications as casings and impellers for salt-water pumps. In order to supplement service records, and to provide engineers with quantitative data for comparing high-alloy cast iron with bronze, trials have been carried out in a corrosion-erosion test apparatus at The International Nickel Co.'s Marine Testing Station at Kure Beach, N.C. Specimens used in the tests were in the form of bars measuring about $4\times\frac{3}{4}\times\frac{1}{4}$ inch. They were attached, in duplicate pairs, to the periphery of a Bakelite disk which was mounted on a Monel shaft so that it could be rotated at high speed while submerged in sea water in the tank of the machine. The speed of rotation was 380 r.p.m., providing a calculated speed of 27ft. per sec. at the tips of the test bars. The action of the testing machine is such as to whip large quantities of air bubbles into the water. The violent agitation of the water by the specimens also served to heat it, so that it was necessary to keep the temperature down by the continuous addition of cooled water pumped directly from the ocean into the testing machine from which it overflowed to waste. By this means, the temperature of the water was held at about 86 deg. F. during the test and at the same time the continuous replacement of the water being used for the test prevented exhaustion of corrosion products which might have affected the corrosion reactions. The test ran for a total of 1,440 hours without interruption for more than a few hours at a time. The extent of corrosion was measured in terms of weight-loss, and the effects produced were also recorded by photographs of the leading edges, which were subject to the most highly concentrated attack. Comparison was made of the behaviour of the various grades of Ni-Resist with that of a bronze of the type generally used in applications requiring resistance to the corrosive and erosive effects of sea water. Results confirm the suitability of Ni-Resist cast irons for such applications as sea-water pumps, valve bodies, etc.—*The Nickel Bulletin, Vol. 22, January-February 1949, pp. 30-31.*

**Crankshaft Damping**

The author attempts to give a correct physical explanation of natural damping by torsional vibrations, and also to obtain approximate formulæ for pre-calculation of the damping in any given case. The paper describes experimental work with a single-cylinder engine driven by external power, and excited to torsional vibrations by a spring-loaded dam disk. In this way the damping from the moving parts could be investigated separately, and it was found that the damping was almost entirely due to hysteresis in the crankshaft, and oil damping, due to lateral shaft movements in the main and crankpin bearings, which was directly proportional to the bearing clearance. The paper also gives a simple and practical method for the calculation of damped vibrations in arbitrary elastic systems, and the calculation of hysteresis and bearing damping in a single-cylinder engine. Formulæ are given for the total damping in multi-cylinder engines, with or without heavy flywheels, and the results are compared with the measured damping in a number of oil engines in service.—*Paper by P. Draminsky, read at an Extra General Meeting of the Institute of Mechanical Engineers, 25th February 1949.*

**International Conference on Safety of Life at Sea, 1948**

This paper is primarily a record of the principal amendments and additions to the technical provisions of the International Convention on Safety of Life at Sea, 1929, which were decided upon at the International Conference held in London in 1948. The technical matter covered by the conference is contained in the Regulations of the Convention and these are commented upon wherever important and interesting decisions in relation thereto were arrived at. Particular reference is made in the paragraphs under construction to the water tight sub-division of short international voyage ships which carry large numbers of passengers in excess of their lifeboat capacity, and to discussion which arose at the conference on the criterion of service numeral which is used to determine the factor of sub-division appropriate to any particular ship. The stability of passenger ships in a damaged condition and fire protection in accommodation spaces in passenger ships are also mentioned in these paragraphs. Under the heading of Life-saving Appliances reference is made to changes affecting the types of lifeboats carried in ships and their equipment, stowage and launching arrangements, and to particular regulations relating to whale factory ships and oil tankers.—*Paper by G. Daniel, read at a meeting of The Institution of Naval Architects, 6th April. 1949.*

**The Effect of Heat on Electro-deposited Chromium**

The object of this investigation, which was carried out at the Thornton Research Centre of the Shell Refining and Marketing Co., was to examine the effect of heat and friction on the physical properties of electro-deposited chromium in the temperature range from 100 deg. C. to 450 deg. C. and to find, if possible, the factors, other than poor adhesion to the basis metal, responsible for those cases in which satisfactory wear resistance is not obtained. The wear resistance was investigated with a reciprocating-wear machine. Wear was examined at room temperature, and at 180 deg. C. and 250 deg. C. The most obvious feature of the test was the scoring found on the chromium surface when the test temperature was above room temperature. Scoring was found to occur even at 100 deg. C. after a period of time and must be due to some physical characteristics of the chromium itself. The mechanism responsible for the scoring is postulated as follows. It was found in preparing cross-sections of chromium deposits for microscope examinatinons, that extreme care was essential in rubbing the sections on fine emery papers; otherwise the sections were found to contain angular irregular voids due to chromium breaking out of the free surface. It has also been found that chromium contracts when heated and that the grain size of the electro-deposited metal is exceedingly fine. The cracking of chromium deposits on heating will lead to small pieces of chromium becoming detached, owing to the small grain size and brittleness of the deposited metal. Hence in subjecting continuous chromium surfaces to friction above room temperature, the physical structure is altered with a distinct possibility of the breaking-out of small pieces of metal. It is true that 300 deg. C. is quoted as the temperature at which significant contraction of the deposit takes place, but it is not necessary for the bulk of the metal to reach this temperature. The heat generated by friction can raise the temperature locally, and so initiate a small amount of contraction and cracking. Since the deposit is under a contractile stress, whenever these minute cracks exceed the critical depth, they will propagate till the decrease in stress falls below the critical value, but, in reaching this depth, small pieces and sharp edges will be left to be broken off and cause wear and scoring. It is possible also, but not so probable, that cracking can arise in an engine from the sharp edge of a piston ring pressing on a plated cylinder wall, the action being similar to that shown in Figs. 5 and 6 of the original article where a diamond impression initiated large cracks at temperatures above 80 deg. C. It is suggested that hydrogen is the operative factor in the wear of chromium at elevated temperatures. It follows from this that trouble may be experienced down to 80 deg. C., where hydrogen is first evolved, but would be longer in developing. This is borne out by crack formation being found at 250 deg. C., but pronounced at 300 deg. C., and also by scoring being found in wear tests at 100 deg. C. after six hours but not after one hour. If the suggestions made above concerning the wear of chromium are correct, then it should be possible to overcome the difficulties by adopting a definite procedure of heating and grinding combined with reverse-current etching.—*R. Graham, K. R. Williams and R. W. Wilson, Engineering, Vol. 167, 18th March 1949, pp. 241-243; 25th March, 1949, pp. 265-267.*

**Auxiliary Power Plants for Large Tankers**

A number of the new American 12,500 h.p. super tankers will each be equipped with two General Electric auxiliary a.c. generating sets. The steam conditions are to be the same as those of the propulsion turbines, i.e. 835lb. per sq. in. g., 840 deg. F. at the throttle, and 28·5in of vacuum at the exhaust The specific steam consumption will be 11lb. per kW. hr. Each auxiliary generator set consists of a 6-stage condensing turbine, a single-reduction gear with speed reduction ratio of 10,059 to 1,2020 r.p.m., a 400 kW. a.c. 0·8 power factor, 450-volt, 60-cycle, 3-phase generator, and a shunt-wound exciter generator rated 125 volts, d.c., 1,200 r.p.m., all coupled together and mounted on a common bedplate. Each set exhausts into its own condenser.—*F. V. Smith, Pacific Marine Review, Vol. 46, February 1949, pp. 30-33, 92.*

**Cavitation of Screw Propellers**

The author presents cavitation characteristics as deduced from ship trials and model experiments. The trial results of a destroyer show 15 per cent loss of thrust and 7 per cent loss of propulsive efficiency at high speed attributed to cavitation. Generally cavitation in warships proceeds beyond the incipient stage if the ship speed exceeds 30 knots and propeller tip speed exceeds 105 knots. Unital thrust of propellers is not a good criterion, the range being 0·5 to 1·0 tons per sq. ft. of developed blade area. The thrust loss in motor torpedo boats, if driven by fast running engines, is large and may be 60 per cent, or even 80 per cent in extreme cases but is reduced to a small amount if suitable reduction gear is fitted. Loss of efficiency may be 20 per cent in severe cases, but small and even negligible with suitable machinery. Tests of a model of the propeller fitted to the destroyer show breakdown in thrust coefficient and, to a less extent, torque coefficient and efficiency when cavitation ensues. This is marked at high speed and high slip. Cavitation loss is small if the propeller operates at moderate slip at the peak of the efficiency curve. The growth of cavitation with increase of speed is illustrated by photographs. The critical loading coefficient for thrust breakdown increases from 0·67 at low cavitation number to 0·81 at large number, i.e., 0·77 to 0·93 tons per sq. ft. of developed blade area. Thrust breakdown curves are plotted for two series of model propellers covering a range of blade width. Propeller cavitation criteria are deduced from theory of cavitation breakdown for profile sections applied to model propeller tests. Comparison is made between tests of the model propeller in a tunnel and in a ship tank. At working slip the thrust coefficient is 6 per cent greater in the tunnel than in the tank and the propeller efficiency ½ per cent less. A theoretical correction for tunnel wall interference helps to correlate tank and tunnel results when there is no cavitation. Cooperative research is in progress in eight cavitation tunnels in different countries, to determine the effect of tunnel wall interference and other factors on cavitation. This research was arranged at the recent meeting of the International Conference of Ship Tank Superintendents. Cavitation tests are compared with ship trial-results. An empirical correction is made to cavitation number to allow for more severe conditions in the ship. There is a limit to the present extent to which cavitation tests simulate ship conditions but important advances are made. This is confirmed by examples for two-bladed propeller, five-bladed propeller and erosion.—*Paper by R. W. L. Gawn, read at a meeting of the North-East Coast Inst. of Engineers and Shipbuilders, 11th March, 1949.*

**Theoretical Study of a Methodical Series of Propeller Tests**

The author gives details of some theoretical calculations carried out for the Dutch Tank B.4.40 screw. It was considered that a method of calculation should be adopted which would allow of the examination of each individual screw section on its own merits, the combination of such individual results producing the required performance characteristics of the propeller. For this reason, the combined momentum-blade element theory, in the form presented by Professor Burrill, was chosen. Certain small modifications to the method were devised in order that the calculations could be carried out in a clear and concise form. Six typical propeller sections were investigated, namely, at 0·3, 0·5, 0·7, 0·8, 0·9 and 0·95 radius. This meant that, together with spots of zero value at root and tip, curves of thrust and torque grading over the propeller blade could each be defined by eight spots—enough, it was considered, to represent fairly the shape of each grading curve. The results are discussed, and suggestions put forward as to how the basic theory might be modified in order to eliminate certain discrepancies between theory and experiment. These modifications are then tried out on the unity pitch ratio screw.—*Paper by J. F. Leathard, read at a meeting of the Institution of Naval Architects, 8th April 1949; Paper No. 7.*

**New Fuel Transfer Pump**

The transfer of the heavier fuels presents a problem which can be solved in various ways. At least one large oil company is equipping all new installations with heating arrangements to bring the temperature of the fuel oils up to 150 deg. F. or more, to reduce the viscosities so as to enable the oil to be pumped into the ship. Pumping problems are being experienced also in the transfer of the oil from the double-bottom tanks often through long suction lines incapable of allowing the specified quantity to flow to the pump. In exceptional cases, due to the high viscosity, long line and inadequate pipe size, the maximum quantity which will flow to the pump may be very small, and this quantity cannot be increased by any size or type of pump, since the supply of oil is solely determined by the pressure of the atmosphere to push the oil along the suction line. Under these conditions the fuel oil transfer pump will be required to adjust itself to this decreased capacity and, at the same time, pump against a high vacuum of approximately 25in. Hg.. To meet such demands, Stothert and Pitt, Ltd., are manufacturing a variable-stroke, rotary displacement, piston pump direct coupled to a variable speed electric motor. Such a variation to displacement and speed is designed to make this pump suitable for the handling of fuel oils of all viscosities likely to be encountered. In addition, it is capable of creating, and pumping against, the high vacuum which will be necessary to draw the oil up the suction line. Designed for a maximum capacity of 75 tons per hour against a discharge pressure of 45lb. per sq. in., the pump, with correct adjustment of the control wheel, is capable of continuously working against a vacuum rising to 25in. Hg. The pump is directly coupled to a 16 h.p. variable speed motor with speed variation down to half speed. A description of the mechanical features of the pump is given.—*The Shipping World, Vol. 120, 13th April 1949, p. 417.*

**New Approach to Paddle Wheel Design**

The author reports on the results of his theoretical and experimental investigations into the operation of paddle wheels. He makes the point that the existing theories do not furnish a complete explanation of the phenomena involved in the production of the propulsive effect by the movement of a paddle through the water. A new type of feathering wheel is proposed and detailed data regarding the functioning of the new wheel type, which is called the "Süberkrüb" wheel after its inventor, are given. The advantages of the new wheel type are claimed to be increased efficiency in combination with higher speed and consequently reduced wheel dimensions.—*F. Süberkrüb, Hansa, Vol. 2, 12th February 1949, pp. 161-165.*

**Machinery of the *Oslofjord***

The Norwegian American Line's transatlantic passenger liner *Oslofjord*, built by the Netherland Dock and S.B. Co. at Amsterdam, is equipped with the two highest powered engines of the Stork type yet constructed. They are each designed to develop 8,175 b.h.p. at 130 r.p.m., and on the test bed are reported to have shown a fuel consumption approximating to 150 gr. (0·33lb.) per b.h.p. hour. The scavenging air is supplied from separate blowers, and the power required for these machines has to be taken into consideration. The seven cylinders have a diameter of 720 mm., with a piston stroke of 1,100 mm. The engine is of the double-acting two-stroke type, but the cylinder covers, instead of being in two sections, as previously, are now in one part and of molybdenum steel. There is a single centrally placed fuel valve in the upper cylinder cover, and four valves symmetrically disposed around the centre in the bottom cover. Each lower cover has a starting air valve, but there is none in the upper cover, and, at starting, fuel is supplied to the top fuel valve at the time when starting air is being delivered to the cylinders. Rapid manœuvring is thereby ensured. The pistons are oil-cooled, and the cylinder liners and covers are cooled by circulation of fresh water. Fuel oil is utilized for cooling the fuel valve nozzles. Scavenging air is supplied from rotary scavenging blowers. There are three auxiliary

Diesel engines for this purpose, and in normal operation one will supply the air to each propelling engine, the third unit being a spare. The capacity of the blowers is 1,100 cubic m. per min., against a static head of 1,250 mm. of water, with the blower running at 2,040 r.p.m. In all, there are four Diesel generating sets with engines of the four-stroke Stork type, with six cylinders 420 mm. in diameter and a stroke of 600 mm. They develop 1,100 b.h.p. at 330 r.p.m. Two of the engines are coupled direct to 450 kW. 220-volt Diesel dynamos, and the other two to 600 kW. dynamos. Two of the 450 kW. units and one of the 600 kW. sets are each coupled to a scavenging air blower through a Vulcan-Sinclair hydraulic coupling. In the ship the auxiliary engines are installed in a separate auxiliary engine-room, and the shaft of each is taken through a bulkhead to drive the blowers, which are installed in a narrow and well-insulated compartment drawing air either from the engine-room or from the deck. The vessel has a gross register of about 16,000 tons and a displacement of 16,500 tons.—*The Motor Ship, Vol. 30, April 1949, pp. 6-7.*

**Apparatus for Indicating Piston Condition**

A special apparatus for indicating the condition of a piston and cylinder of an oil engine is illustrated in Fig. 1. A certain amount of rotation is given to the piston as it reciprocates, and slip occurs when there is any abnormal resistance to this rotation, an electrical indicating device being provided. To improve the running qualities of the piston (2), a ratchet imparts rotation by means of an operating

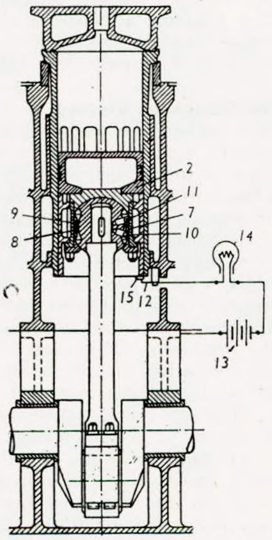



Fig. 1.—*(British Patent 610,209.)*

drum (7) fixed to the piston, slotted expanding rings (8, 9) with operating pins (10, 11) being fitted. The pins reciprocate, due to the oscillation of the connecting rod, which has a ball-shaped upper end. When the connecting rod rocks in one direction, one of the pins (10) expands the ring (8) by moving away from the slot, while the second ring (9) is compressed. The expanded ring (8) engages the operating drum (7) frictionally and imparts rotation to the piston while the ring (9) slides over the inner surface of the drum. When the connecting rod oscillates in the reverse direction, the operating rings exchange their actions. It is stated that in the course of service the piston acquires a highly polished surface with an increasing hardness. The indicator comprises a switch (12), a battery (13) and a lamp (14). The contact is made by a projection (15) on the piston at the lower dead centre.—*Brit. Pat. No. 610,209, Sulzer Freres, Winterthur, Switzerland. The Oil Engine and Gas Turbine, Vol. 16, April 1949, p. 433.*

**Fiat-engined Liberty Ships**

Following the re-engining of two of the Liberty class ships among the large number acquired by Italy from the American Government, a plan of converting many of these vessels to Diesel propulsion has been prepared, and the first of the series will probably be taken out of service for the work of installing a six-cylinder Fiat engine to be carried out in the near future. The standard engine for this method of conversion is a six-cylinder Fiat, double-acting two-stroke unit with cylinders 680 mm. in diameter. According to the detailed calculations which have been made by the builders, taking the average service speed with the Liberty machinery as 10·6 knots, with a corresponding output of 1,900 b.h.p. at 72 r.p.m., when a 3,600 b.h.p. Fiat engine operating at 125 r.p.m. is installed the increase in speed will be 1·9 knots; this takes into account the loss of efficiency of the propeller due to the higher revolution speed. The vessel should, therefore, maintain 12½ knots when loaded. It is believed, however, that as the loss in propeller efficiency has been calculated on the well-known Taylor formulæ, the final results will be better than 12·5 knots, as it has been ascertained in several tank tests that the actual speed in service is more favourable than that anticipated from tank trial results. It is claimed by the builders that this engine will use the same boiler oil as employed in the oil-fired Liberty steamers, and that the daily consumption of 27 tons for a speed of 10·6 knots will be reduced to 15·5 tons for 12½ knots, a reduction of 2,810 tons per annum. Beyond this, there will be an increase in deadweight capacity of 530 tons.—*The Motor Ship, Vol. 30, April 1949, p. 22.*

**Exhaust-gas Turbo-charging of Diesel Engines**

This article discusses the application of pressure charging to four-stroke Diesel engines and deals with standardized exhaust gas turbo-chargers supplied by the Brown Boveri Co. It is claimed that observations made by customers over a period of years have shown that pressure-charged engines are quite as reliable in operation as uncharged engines and require less maintenance than the latter. As an example, the case of the turbo-chargers of the two 800 b.h.p. main engines of the M.S. *Lochfyne,* owned by the British shipping firm David McBrayne, Ltd., Glasgow, is quoted. These turbo-chargers have operated without a single breakdown or trouble of any kind since being put into service fifteen years ago, during which time the ship has travelled over 380,000 miles. The original gas turbine blades are still in very good condition. Apart from the ball bearings, which have been changed about every two years, no parts have had to be replaced. Both engines still operate with the original valves, due to the effective cooling action of the scavenging air.—*Th. Egg, The Brown Boveri Review, Vol. 35, Nos. 9/10, 1948, pp. 259-262.*

**Diesel Engines for Fishing Boats**

Fishing boat engines have to meet widely varying conditions of service. They must be capable, for instance, of running just as long as is necessary at very low speeds without any harmful consequences. During the voyage from the home port to the fishing grounds they are set to the normal full speed ahead. But for the actual fishing this speed must be greatly reduced, as a boat which is trawling a net only makes about 3 or 4 knots. The drifters which fish for herring and mackerel in the North Sea move even slower. Although under these circumstances the engine must run far below normal speed, lubrication must never be insufficient or excessive. Cooling must also be adapted to service at all times, so that the most favourable cylinder temperatures are maintained. The choice of propeller speed is affected by factors such as propeller diameter, vessel speed and the water resistance to hull and nets. It is all the more important, however, because it is instrumental in deciding whether the requisite propeller thrust is obtained with the lowest possible fuel consumption; in other words, it seriously influences the economy of operation. The first question to be faced is whether the propeller speed is to be chosen for the outward run or for the fishing proper. In open water, the speed of vessels below 100ft. in length should be about 10 knots; larger vessels should make 11 or 12 knots. These requirements would call for fairly high propeller speeds. Once in fishing waters, however, the boat becomes a kind of tug and must tow dragnets at depths of as much as 700 or sometimes ecen 1,000ft. The resistance of the cables, the gear used for keeping the net open and above all the net itself demands a powerful thrust from the propeller, and if this is not forthcoming the net will not remain fully open. Now the propeller efficiency is higher when its speed, for a given thrust, is lower, so that in this case obviously a fairly low speed would be preferable. The only completely satisfactory answer to these conflicting requirements would be the provision of two propellers with different characteristics for the two duties. In practice the propellers used are mostly fitted with fixed blades and therefore with constant pitch, so that there is no alternative but to find the best possible compromise. For fishing near the coast priority is given to the actual fishing performance, and the propeller is dimensioned for this work. Under these circumstances the upper speed of 300 r.p.m. arrived at from considerations of piston speed proves to be rather high. In practice speeds between 150 and 200 r.p.m., according to the size of the vessel, have been found to give the best results. The engine of a fishing boat with direct propeller drive is consequently so chosen that its normal speed lies within these limits. As already stated, this factor is of special significance for fishing vessels. The Sulzer two-stroke Diesel engines of the types TS and TD, which comply with the aforementioned requirements are single-acting, direct-reversible models running at the low speed of 250 r.p.m., which permits them to be coupled direct to the propeller shaft in most cases. The mean piston speed is only 16ft. 6in. per sec., and the unit weight does not exceed 90lb. per b.h.p. The new

TD type is equipped with lateral scavenge pumps and is shorter than the corresponding type with a single scavenge pump at the end of the engine. Apart from the double-acting scavenge pumps, the engine drives a cooling-water pump, a piston-type bilge pump, a gear-type oil pump and a two-stage air compressor for a pressure of 425lb. per sq. in., all of them being direct-coupled. Reversing is done with compressed air, special attention having been paid to the simplicity and reliability of the system. This is an advantage specially appreciated in drifters, as these vessels often have to carry out a large number of manœuvres within a short time.—*P. Kindt, Sulzer Technical Review, No. 2, 1948, pp. 6-11.*

**Natural Torsional Vibrations of Engine Shafts**

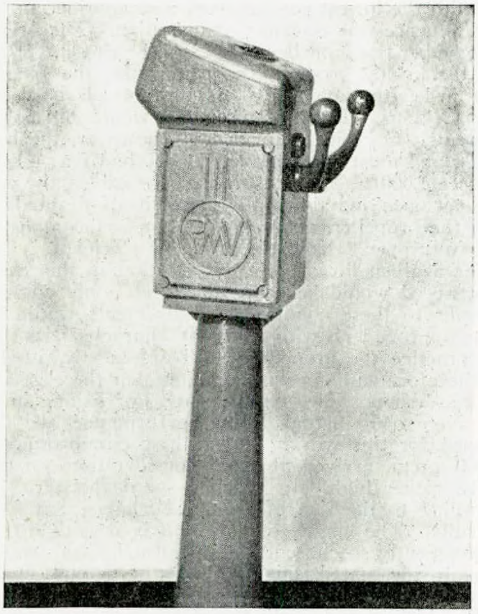
An article by K. Klotter in Ingenieur Archiv describes in detail the various existing methods of determination of the natural torsional frequencies of engine shafts. There is a list of the different methods, with references in chronological order, and a discussion of their usefulness. The methods are grouped into (a) methods that replace the original dynamic system by an equivalent one (a one-dimensional shaft with rotating disks of corresponding moments of inertia), and (b) methods that split up the engine shaft into sub-systems, each with one or two disks only. The problem is restricted to undamped free vibrations only. All methods are explained with fully worked-out numerical examples to demonstrate the analytical or graphical details and the system of tabulation.—(K. Klotter, Ingen.-Arch., 1949, p. 1). *Abst. No. 2,606, Journal, The British Shipbuilding Research Association, Vol. 4, No. 4, April 1949.*

**Pulverized Coal Fired Marine Installation**

An article on early developments in pulverized coal firing in the November 1948 issue of "Combustion" had stated that none of the units fitted with pulverized coal firing in the period extending from 1927 to 1937 is still in service with this type of firing. It is now stated that the *Berwindglen,* listed as having been changed to oil firing, was later converted back to coal and is now being operated with pulverized coal by the Berwind Coal Co., her boilers being fired with two B. and W. type B mills which replaced the Fuller-Lehigh type C mills in 1937.—*Combustion, Vol. 20, March 1949, p. 46.*

**Cargo Vessels with Single Hold**

For some time past the Porsgrund Mek. Vaerk. of Porsgrund, Norway, has built a standardized type of steamer, an "easy trimmer" single decker, with the engine aft and one large hold, the total cargo capacity being in the neighbourhood of 2,500 tons. It is stated that this type has proved eminently successful a series of five or six being now in service. Some time ago four such ships equipped with Diesel machinery have been laid down. The first of these, the *Dido,* recently ran trials, reaching 12·6 knots in ballast. Length overall is 259 feet and length b.p. is 243 feet. Beam is 37ft. 9in. and gross register is 1,612 tons. The shipbuilders employ an electro-hydraulic steering gear of their own design, which is claimed to be simple and to occupy small space. On the bridge the steering wheel is replaced with two handles which control the steering gear motor. The engine is of the Nohab two-stroke type with six cylinders; it has an output of 1,080 b.h.p. and runs at 300 r.p.m., driving the propeller through a reduction gear at 120 r.p.m.—*The Motor Ship, Vol. 30, May 1949, pp 54-55.*



*The steering wheel is replaced with two handles which control the steering gear motor.*

**Steam Propulsion Economy of Small and Medium Size Vessels**

The author discusses the various factors influencing the economy of reciprocating machinery for powers up to 3,000-4,000 i.h.p. and stresses the advantages offered by superheated operation and the employment of exhaust turbines of the Bauer-Wach type. A new design of compound engine combined with Bauer-Wach turbine is described. This plant, which was developed by a concern in Bremen, is rated at 1,100 i.h.p. The reciprocating part consists of one high pressure cylinder of 450 mm. diameter and one low pressure cylinder of 900 mm. diameter, the stroke being 900 mm. Live steam condition is 14 atmos. and 300 deg. C. and the vacuum is 92-95 per cent. The reciprocator speed is 100 r.p.m. and that of the exhaust turbine is 6,000 r.p.m. The article also illustrates a triple-expansion engine equipped with an integral oil fired resuperheater for reheating the steam between the high pressure and medium pressure cylinders. The reheater is contained in a casing attached to the engine frame and the fuel oil pump is driven off the engine, so that the supply of fuel to reheater is contained in a casing attached to the engine frame and the down. The reheat temperature is manually controlled.—*Prof. Dr. G. Bauer, Hansa, Vol. 2, 19th February 1949, pp. 184-188.*

**Some Problems in the Design of Rigging**

In introducing his subject the author points out that the design of ships' rigging involves a great variety of problems; but with the exception of the design of small parts such as shackles and eyeplates, the whole subject appears to have been ignored for many years. The absence of failures in rigging indicates that all shipbuilders have some method of solving their problems, but as with many aspects of shipbuilding, their solutions may err on the side of safety to an unlimited extent. No shipbuilder will experiment with a well-tried standard method of design on such a vital subject without the backing of the strongest theoretical justifications that have been thoroughly criticized and amended by the opinions of others. The paper covers the analytical approach to a number of specific problems, as, for instance, the design of derrick posts without and with stays and a mast with stays at two levels. The effects of preventers when fleeting cargo are analysed and the choice of the numerical value for the factor of safety is discussed. The author points out that the tension modulus of steel wire rope is a factor of vital importance in the design of stayed posts and masts; yet it is not mentioned in any of the accepted handbooks. Accordingly an experiment was carried out by the author. A section of wire rope was put under tension in a testing machine and the elastic extension for a given change of load measured. From this it was deduced that the tension modulus is between 4,000 and 6,000 tons per sq. in. for rope of 6/19 construction. The test was done on wire of this construction as it had been customary to fit such wires as standing rigging on previous vessels. The value 4,000 tons per sq. in. was used in subsequent mast design problems. An experiment on a mast was carried out on a vessel fitting out at Barrow when the routine tests of the derricks were being held. It was thought that the tensions in the standing rigging could be obtained if the wires were made to vibrate at their natural frequency. After a preliminary trial to determine the range of frequencies likely to occur, two observers were entrusted with the task of measuring the changes in frequency and noting the nature of loading. It was found that the majority of wires could be made to vibrate in the binodal form by impressing oscillations with the hands. This binodal form was desirable as it kept the frequency down to a figure that could be counted with ease over a period of half a minute. Thus the only equipment required to measure the changes in tension was a stopwatch. Owing to the difficulty of determining the position of the load, it was not possible to check the results by finding the deflexion of the mast at the points of attachment of the various stays from the bending moment diagram, and relating these deflexions to the defiexions to the theoretical extensions of the wires. It is hoped to carry out similar experiments on each ship as the derrick tests are performed, which will provide both a check on the design and useful data on the initial tensions in standing rigging imposed by setting up with rigging screws.—*Paper by R. V. Turner, read at a meeting of The Institution of Engineers and Shipbuilders in Scotland, 12th April 1949.*