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Research in Marine Auxiliary Machinery

The paper describes several investigations that have been or are being performed in a large engineering group whose main products are marine and power plant auxiliary machinery and it is appropriate that some introductory remarks be made as to the organization of the research and modes of procedure. A typical problem investigated concerns pump packings. Limited information was available on the surface wear between metal sleeves and textile packing materials. Occasional service failures accentuated the need for classified information and experiments were commenced several years ago. In the hope of acquiring data in a comparatively short time it was first proposed to incorporate a minute quantity of cobalt 60 (half-life 5·3 years) in the sleeve material and to measure the radioactivity of the emergent lubricating water as an indication of the amount of wear, showing up long before mechanical, optical or electronic methods. In this way, relative wear in a given time might give a criterion of the quality of any packing/sleeve combination. It was soon found to be no use speeding up the detection of wear unless the wear bore a known relationship to time, and that wear was generally catastrophic as soon as a certain type or amount had taken place. A sudden increase and discontinuity could take place at any instant—after 1 hour or after 10,000 hours—and no method of time reduction (or load increase) could be suggested to give results capable of interpretation. This method was abandoned, therefore, in favour of endurance testing. No two tests were comparable when assembled by different operators; all packings were inserted, therefore, by the same man, and in this way the amounts of compression of each ring of packing, and, where the rings were to be made, the gap between the ends of each ring, etc., were controlled: a packing could not be considered as a seal, as when a complete seal was achieved, the material was quickly burned and the sleeve damaged. All packing tests now allow leakage (anything up to 2 gal. per hr.—although 1/10th of this is more usual) and, therefore, lubrication (and the provision of a reasonable temperature at the working surface). Heat transfer problems in marine auxiliary machinery range over practically the whole field of the subject but include mostly aspects in which gases and liquids transfer heat to each other either directly or indirectly and with and without change of state. The problems are often dynamic and the heat transfer concerns moving fluids in conditions where the transfer coefficients increase with speed. The design of most equipment is a compromise between the attainment of highest heat transfer and the requirements of least power to maintain motion against the mechanical loss, with which heat transfer is associated. Many factors enter this generalization before economic design is reached; these are other parameters upon which the total transfer depends, namely, time, surface area and potential difference, besides practical considerations of maintenance and cost, and physico-chemical relationships, for example, deposition, erosion, corrosion, etc., which affect the heat transfer surfaces. A condenser tube from a large ocean liner was returned for examination and set up inside a large tube, giving the same hydraulic means depth as existed in its original bundle. With temperature and pressure measurements along the whole length—some 8 feet—and with steam conditions and specific condensation rate as close to practice as possible, heat transfer coefficients for the tube (*i*) as returned from the ship; (*ii*) cleaned on its outer surface only; and (*iii*) cleaned on both outer and inner surfaces, were measured for different rates of tube flow. It was found that the resistance to heat transfer had increased by about 50 per cent from the original clean condition and emphasized sharply the need for periodic cleaning. Considerations such as these enter the design and hence allowance for running with a reasonable amount of fouling is made (thereby increasing the size of plant) to reduce maintenance costs.—*Paper by S. Weinberg, read at a meeting of the Institution of Engineers and Shipbuilders in Scotland, 26th January 1954.*

A Century of Coaster Design and Operation

The paper deals with the evolution of the coasting tramp. Some figures of the tonnage and type of cargo carried coastwise are given, and the influence coal cargoes have on the design of coasters is mentioned. Reference is made to proportions, power and speed and the tendency for these vessels to increase in size. A technical comparison is made between two vessels, one built about the beginning of this century and one recently constructed. Plans are shown indicating the gradual improvement in accommodation and general layout. Reference is also made to stranding, navigational aids and future tendencies in design. In referring to future tendencies the fact is mentioned that, according to the authors of the past generation, the raised quarter-deck originated in certain small vessels where the engine and boiler rooms were situated amidships. It was found that the tunnel through the after hold cut away so much capacity that, when the vessels were loaded homogeneously, they tended to trim by the head; the raised quarter-deck was fitted to increase the capacity of the after hold and correct this tendency. In the small coaster with the engine and boiler room aft, the raised quarter-deck originated to get height aft to accommodate the machinery. As this type of vessel got larger and the bridge house was shifted forward towards amidships, the raised quarter-deck was continued forward to this bridge house to increase the hold capacity, and perhaps also to suit trim. With the very modern coasters, the raised quarter-deck in many instances still prevails in spite of the fact that the Diesel machinery does not call for the height, and the officers' accommodation and bridge may be situated aft. This leaves a very small well-deck forward and appears to result merely from custom or tradition. Obviously with Diesel machinery, the single flush-deck coaster is the correct answer and perhaps in due course the term "raised quarter-deck" will disappear from the vocabulary of shipbuilders and shipowners. Although a large number of typical coasting tramps are employed in the coal trade, many vessels operating in the coal trade are especially designed for that purpose, such as the "flat-iron" type designed to pass under the Thames bridges. Other specially designed colliers are fitted with hopper-sided tanks in place of the normal hold bilges, and a number of such colliers are not fitted with winches or cargo working gear since these are not required in their particular trading and discharging berths. It is clear from the above that the modern coasting vessel constructed specially for the carriage of coal is not ideally suited for general tramping, and since about 80 per cent of the cargo carried by the coasting tramp consists of coal, it is clear that there will be progressive decline in the number of true coasting tramps as we have known them for a century, and a greater tendency towards specialization.—*Paper by J. C. Robertson, and H. H. Hagan, read at a meeting of the Institution of Engineers and Shipbuilders in Scotland, 1st December 1953.*

Turbine-Generator Sets for Shipboard Service

The authors briefly examine the factors influencing the design of marine turbo-generator sets, and then comment on the individual elements of the complete unit. It is common practice to supply steam to the turbine from the same boilers as are used for the main engine, and at the same temperature and pressure. The turbine rotor is often manufactured from a solid forging; this avoids the high disc-base stresses due to shrunk-on discs and, by eliminating disc hubs, shortens the span between the bearings. As regards the gearing, it is usual in modern designs to have a reduction ratio of the order of 10:1; single-reduction gears are preferable because of their simplicity. Modern marine practice strongly favours the use of A.C. in the generating plant and distribution system, with conversion to D.C. when speed control of the motors is necessary, as, for instance, in cargo winches. The generators are usually three-phase, 60-cycles, 450-volt machines rotating at 1,200 r.p.m. The present trend is toward totally enclosed generators with surface air coolers, particularly for sizes of 500 kW. and over. The generator speed of 1,200 r.p.m. which is necessary with a six-pole machine has often been questioned, and a higher-speed two-pole generator proposed instead. This would involve a change from the salient-pole rotor to the cylindrical rotor with a smaller diameter; but repeated studies have shown that for a comparable level of efficiency there will be no reduction in the physical size of the complete unit and, in fact, the unit will be smallest when a six-pole generator is used. The arrangement of the individual units forming the set is briefly considered. Since the baseplate cannot be completely stiff, some tolerance on alignment is needed, which can be provided by a special quill shaft in the coupling; this is illustrated by a sketch.—*Paper by A. G. Gale, and H. J. Chase, read at the 1953 Annual A.S.M.E. Meeting. Paper No. 53-4-89. Journal, The British Shipbuilding Research Association, December 1953; Vol. 8. Abstract No. 8,371.*

Fishing Vessel Developments

Since the autumn, international publicity has sharply outlined trends in the development of fishing vessels of all kinds. The reason for such attention is the success of the International Fishing Boat Congress, 1953, held under the auspices of the Food and Agricultural Organization of the United Nations in Paris from 12th to 16th October, and in Miami from 16th to 20th November. The object of duplicating the session was so that people from the eastern hemisphere could gather in the French capital, and those from the western hemisphere in one of the most popular resorts on the Florida coast. A gratifying number of delegates attended both conferences. The idea behind the Congress was to table knowledge at international level which would advance the design and construction of fishing vessels of all kinds. It was felt that fishing boat construction had in the past been too severely localized in certain districts, and that one-half of the world did not know anything about the technique employed in the other. No attempt was made—as obviously this would have been impossible—to evolve a standard fishing vessel. There was, however, a great deal of valuable interchange of information concerning hull shape, sea-kindliness, methods of propulsion, details of equipment, and last, but not least, methods of refrigeration. The subjects of debates ranged all the day from small open boats of beaching type to the largest and most complete of fish factories; it covered methods of construction from catamarans to highly advanced craft made of laminated wood. It dealt with steel prefabrication and with more common methods of construction. It compassed the employment of light metals in funnels and superstructures; and, perhaps naturally, one of the most keenly debated sessions was that which dealt with propulsion. The main arguments centred around the choice of two-stroke or four-stroke cycles, of direct or indirect drive. There was considerable difference of opinion as to the speed of rotation ideal for a trawler engine. Vigorous support for the old-fashioned hot bulb crankcase compression unit came from Scandinavia, the reason being its simplicity and reliability of operation. On the other hand American opinion was almost universal in supporting the high speed engine. Questions of deck gear drive, and whether a special generator unit, electrical or hydraulic, should be supplied; or whether in view of the increase in auxiliary load a complete Diesel-electric installation should be fitted, were also discussed. The question of the free-piston gas generator and gas turbine was also considered; though the degree of interest was not shown which one would have expected, in view of the fact that a licence to build the free-piston engine had been taken out in Great Britain by a well-known firm of trawler machinery builders.—*The Shipping World, 13th January 1954; Vol. 130, pp. 54-56.*

General Review of Welding in Shipbuilding

After a review of the history and economics of electric welding as applied to marine engineering, the causes of failures experienced in service are discussed. Practically all failures have been caused, fundamentally, by notch brittleness of mild steel. Revised specifications have been drawn up, therefore, as a result of research and experience; and electric welding has become a major feature of ship construction. The level of efficiency reached enables welded ships to be built which are as reliable as their riveted predecessors, and which possess the many additional advantages of welded construction. Continued vigilance is necessary, however, to ensure that the lessons already learnt by hard experience are kept in mind. There is room for improvement in all stages of development of the all-welded ship, and this can only be attained by a close study of the behaviour of the ships already in service. The fundamental requirements for good design are explained and illustrated by diagrams of typical sections. A rational shipyard layout is also illustrated. There is a comprehensive bibliography.—*Paper by F. C. Cocks, read at the Conference of Welded Structures, Institution of Civil Engineers and Institution of Structural Engineers, 24th November 1953. Journal, the British Shipbuilding Research Association, December 1953; Vol. 8. Abstract No. 8,296.*

Direct Contact Jet Condenser

Steam air ejectors are commonly used for the maintenance of moderate and high degrees of vacuum. For a moderate degree of vacuum, a single ejection stage (comprising a single ejector or a number of ejectors, operating in parallel) is provided; for higher degrees of vacuum, a number of stages acting in series are provided. It is usual for condensers to be provided both between stages (inter-condensers) and after the last stage (after-condensers). Inter-condensers serve to decrease the load on the next ejector stage by reducing to a minimum the amount of steam which passes over with the non-condensable gases. In order to achieve the maximum efficiency both inter- and after-condensers must be designed so as to lead to maximum steam condensation and air cooling consistent with a low pressure drop. Many different forms of condensers have been used more or less successfully, some of the surface type in which the heat exchange is indirect and some of the type in which the heat exchange is direct, the steam being caused to rise through falling water. The invention is concerned with condensers of the latter type. In most forms of direct contact jet condensers, the cooling water is caused to cascade over a series of trays or baffles while the steam and air rise in contact with it. The condensing performance of such condensers leaves much to be desired, the mediocrity of the result obtained being, at least in part, due to the difficulty experienced in maintaining the water in contact with the steam and air for a sufficiently long period. Also, the air cooling performance of such condensers is not all that it might be. This is frequently the result of the relative disposition of the steam inlet to the condenser and the air outlet which results in the cooled air being brought into contact with the heated surfaces before it leaves the condenser. The body of the condenser depicted in Fig. 8, is a cylindrical casing (10) with a grating (12) on which are supported a jumbled mass of brass ferrules (14). In the top of the casing is a spray head (16) with six nozzles (18) (only three are visible) through which the cooling water entering at inlet (20) is discharged on to the ferrules. The water finally leaves the casing through an outlet (22) below the grating (12). The mixture of steam and air from the ejector enters the casing (10) laterally through an inlet (24) below the grating and, of course, flows upward into the mass of ferrules where it comes into contact with the water from the nozzles (18). Any gases which are uncondensed, continue to flow upwards and finally leave the condenser casing laterally and through an outlet (26) at a level above that of the water nozzles (18). The ferrules (14) must be arranged in a jumbled mass such, for example, as that which would be formed if they were shovelled into the casing. Their effectiveness is thought to be due to the fact that they cause the water to drip irregularly from one ferrule to another so that very intimate and prolonged contact between the water and the steam is ensured. Also, the ferrules provide very extensive cool surfaces on which the steam can condense. The condenser thus combines the desirable features of surface condensers and of direct heat exchange condensers.—*British Patent No. 700,747, issued to Foster Wheeler Limited. Complete specification published 9th December 1953. Engineering and Boiler House Review, February 1954; Vol. 69, pp. 60-61.*

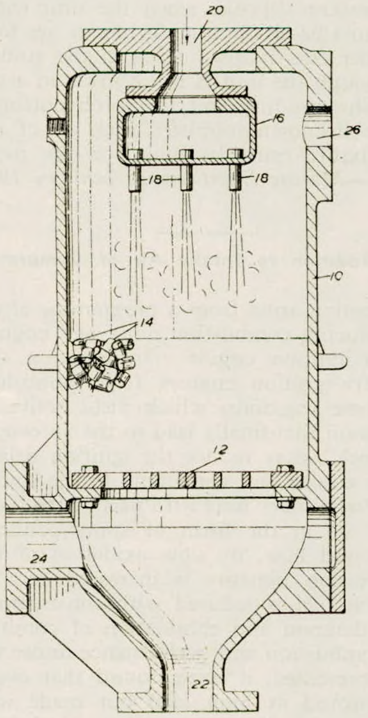


FIG. 8

Functions of Materials in Bearing Operation

In this paper, the numerous factors affecting the operation of plain bearings, principally crankshaft, main, and pin bearings, are considered, and the problems of bearing design, from the aspect of selection of the various materials which are combined in the construction of a bearing assembly, are discussed. An attempt is made to bring together conclusions reached by studying the factors referred to singly or in groups, and the performance of bearings in practice. The latter is based on a study in summary of the three kinds of bearing failure, namely, seizure, excessive wear, and mechanical breakdown. Comment on recent work by Lunn and by Roach is made, and a working hypothesis on the effective fatigue strength of a bearing lining is proposed.—*Paper by P. P. Love, P. G. Forrester, and A. E. Burke, read at a general meeting of the Automobile Division of the Institution of Mechanical Engineers, 8th December 1953.*

Motion of Ships in Confused Seas

In this paper the authors have attempted to show that theoretical studies of ship motions need not be confined to those experienced by a vessel in a seaway whose pattern is rhythmic and regular and, therefore, unreal. The motions that a vessel undergoes in a confused seaway—such as occurs within, or close to, a storm generating area—can be derived provided one seeks only knowledge as to the amplitudes of motion and forgoes (for the time being) any pursuit of phase relationships between vessel and sea. It is possible to treat phase relationships by more advanced techniques, but then the mathematics become extremely difficult. In confused seas phase relations are of lesser importance in most practical applications. Confining the interest to amplitudes alone makes for a powerful extension of the available theory of ships' motions. It then becomes possible to make direct use of a statistical definition of the seaway based on its energy spectrum. Through application of the proper response amplitude operators and frequency mappings, the statistical definition of the seaway is converted into a quasi-homogeneous Gaussian process with a known spectrum. An immediate consequence is a statistical definition of the motions of the vessel for any course and speed. Indeed, the root-mean-square values of the motions and a measure of the occurrence of extreme amplitudes are obtained in this manner. However, these are quite sufficient to define the real motions that a ship experiences in a real sea. The essential idea expounded in this paper is that the motions of a ship are a Gaussian process and that they are completely characterized by a response spectrum defined by an equation. This representation leads to the conception that the response of a ship in a confused sea is a steady-state process rather than a continuous succession of transients. The advantages to be gained from such a conception are powerful for they lead immediately to practical results. The findings of this paper may be summarized as follows: The representation of the oscillatory motions of a ship in confused seas by a response spectrum based on a statistical definition of the seaway is sufficient to give a characterization of these motions of such completeness that the naval architect may obtain practical solutions to many problems in this field which he now deems important.—*Paper by M. St. Denis, and W. J. Pierson, read at the Annual Meeting of The Society of Naval Architects and Marine Engineers, New York, 12th November 1953.*

Fluid Flow in Turbo-machinery

The value to the turbo-machinery designer of air flow tests on models of the various components is discussed, and the facilities for carrying out such tests at Pametrada are described. Model testing is illustrated with descriptions of flow investigations in the ducting of the Pametrada gas turbine. The uses and limitations of cascade testing of turbine and compressor blades are discussed and the importance of the new high-speed variable-density wind tunnel is explained. The main features of the air turbine and its instrumentation are described.—*Paper by H. S. Fowler, read at a meeting of the Institution of Engineers and Shipbuilders in Scotland, 3rd November 1953.*

Metallurgical Aspects of High Temperature Steam and Gas Turbine Plants

In the first place a brief account is given of the properties that are important in metals used at temperatures above 850 deg. F., i.e. resistance to creep, rupture, high-temperature fatigue, relaxation, thermal shock and high-temperature corrosion. This is followed by a discussion of the composition, treatment and properties of the steels and other alloys available for high-temperature applications, i.e. non-stainless ferritic steels, stainless ferritic steels, austenitic steel and alloys based on nickel and cobalt. The paper concludes with a section on welding in which those aspects of welding that are particularly relevant to high-temperature applications are discussed.—*Paper by J. M. Robertson, read at a meeting of the North-East Coast Institution of Engineers and Shipbuilders, at Newcastle on Tyne, 15th January 1954.*

Novel Gas Turbine Plant

In most of the known gas turbine processes the high temperature of the driving gas imposes a severe punishment upon the turbine blade material. It is necessary, therefore, to reduce artificially the temperature of the driving gases, for example, by the admission of cooler air, which obviously causes considerable losses in cycle efficiency. It has already been proposed in cases where solid fuels and fuels with high ash content are to be burned, to connect the turbine in front of the combustion chamber, in order to protect the turbine blades against wear by fly-ash, but this arrangement has certain inherent disadvantages in other respects. A recent patent describes a method by which these disadvantages can be overcome. In a turbine cycle of this type, the compressed air delivery by the compressor is passed through a heat exchanger in which it is heated by heat absorption from the hot gases of a combustion chamber of the oscillatory gas-column type discharging directly into the heat exchanger. The heated air is then passed into the turbine, from which it is discharged into the combustion chamber, where it is used as combustion air and from which the products of combustion are then passed into the heat exchanger, as already explained. In the combustion chamber, gas oscillation is affected by the production of positive pressure waves travelling towards the open end of the combustion chamber, while a flow in the opposite direction towards the inlet of the combustion chamber is prevented by means of a non-return throttle member. At the open end of the combustion chamber, the advancing positive pressure is converted into a returning negative pressure wave, which draws fresh air into the combustion chamber for the subsequent cycle, and also causes ignition of the newly formed mixture.—*The Engineers' Digest, January 1954; Vol. 15, p. 3.*

Three-dimensional Flame-cutting Equipment

The Milwaukee Shipbuilding Corporation has placed on the general market the first "three-dimensional" flame-cutting equipment for scarfing the edges of curved metal pieces in preparation for welded assembly of larger units. The method opens the door for designing increased strength into curved metal structures made up of welded parts which themselves are curved. The semi-automatic "3-D" equipment speeds up and makes more uniform the preparation of welding joints, in comparison with previous hand-held flame-torch methods of scarfing curved pieces. Application has been made for patents. Initial production of more than 80 "3-D" units has been absorbed for manufacturing of curved welded metal parts for armed-forces equipment. That demand now has been met and Milwaukee Shipbuilding finds itself with a substantial addition to its original line of bore grinders. The foundry, shipbuilding and welding fields are highly interested, according to company officials. Milwaukee Shipbuilding calls the flame-cutting equipment three-dimensional because it operates vertically, horizontally and at angles from the first two as it scarfs the edges of curved metal pieces. The third dimension is the operation of the cutting equipment around curves. In this way it is a departure from previously developed semi-automatic flame cutters which operate vertically and horizontally in two dimensions only as they scarf the edges of flat metal. Milwaukee Shipbuilding's new equipment centres on a flame cutter which moves under power as it follows the curves of the metal under process. The cutting flame slants at a constant angle from the desired apex of a scarfed edge. A system of twin rails was devised whose rise, fall, tilts and turns match those of the metal under process. Four spring-loaded drive rollers and a spring-loaded idler roller keep the moving unit true to the path of the rails. The drive rollers move at speeds up to 15in. per min., powered by a 1/25 h.p. electric motor. The speed range can be varied to suit customers' needs. One of the rollers in the drive group is serrated to prevent slippage when the unit enters a curve, where control must be absolute if deviations are to be avoided. The serrated roller cuts its own track in the guide rail which it follows. Although the unit is being made in a standard size by Milwaukee Shipbuilding and the torch cutting equipment obtained from well-known suppliers, each set of rails and the accompanying fixture must be built for the piece of metal being processed.—*Marine Engineering, January 1954; Vol. 59, p. 68.*

Effect of Fuel Addition to Intake Air of Compression Ignition Engine

The investigation arose from a programme of research into the means of reducing combustion noise and engine roughness in a compressor-ignition engine. Information on pre-flame reactions in spark-ignition engines from published literature indicates that these reactions, which yield active partial products of combustion that finally lead to the auto-ignition of the end gas, or "knock", may reduce the ignition delay in a compression-ignition engine and contribute to the smooth running of the engine. Preliminary tests with part of the fuel introduced with the intake air in the form of spray confirmed such a view. They showed how, by slow oxidation of the aspirated fuel, the compression pressure is increased and the ignition delay of the main fuel is reduced with consequent smoothing of the pressure diagram and elimination of combustion noise. The effects on combustion and performance under various loads and speeds are presented, it being found that overall thermal efficiency is improved at high load but made worse at low load. Further investigations were then carried out on the various factors that may have effects on the result of introducing fuel with the intake air. Among these were the types of fuel used for the aspirated as well as the main charge, combustion chamber design, compression ratio, and timing of both parts of the charge. An analysis is made of the processes of combustion under such conditions and a theory advanced to account for the phenomena observed. It is concluded that the lean aspirated mixture does not proceed to complete combustion. The use of intake spray was found particularly effective when the main fuel was of low ignition quality. The technique of using the slow oxidation of the aspirated fuel was then applied to study the ignition of various hydrocarbons in a compression ignition engine. It is shown that the differences in ignition delay of various fuels lie chiefly in the chemical part of the delay while the physical part remains essentially constant in spite of widely different physical properties of the fuel. It is shown also that with the use of the intake spray, the compression ignition engine can run on any fuel irrespective of its cetane number and with only small differences in thermal efficiency. Finally, the effect of some additives on the reactions of the aspirated fuel and the ignition of the main fuel were studied. The results showed that the reactions were suppressed by the addition of formaldehyde and tetraethyl lead and slightly accelerated by nitrogen peroxides. The indication was that the reaction was of the "low temperature" mode. This is discussed in relation to the recently published works on oxidation of hydrocarbons in general and the reactions leading to knock in the spark-ignition engine.—*Paper by W. T. Lyn, read at a general meeting of the Institution of Mechanical Engineers, 8th January 1954.*

Effect of Auxiliary Fuels on Smoke-limited Outputs of Diesel Engines

It is well known that one of the factors limiting the power output from Diesel engines is the tendency to produce exhaust smoke. In consequence only some 65-70 per cent of the available air can be burnt. By reducing the full-load quantity of Diesel fuel injected and aspirating a volatile fuel, such as gasoline, into the intake air, power outputs of 20 per cent or more above the maximum rated load can be obtained without producing more smoke than when the engine is operated at full load under normal conditions. About 80 per cent of the available air can then be utilized. The effect of the octane number of the auxiliary fuel is important, maximum smoke-limited powers being obtained with the higher octane number fuels, while with fuels of low octane number, power increase may be limited by the occurrence of knocking when the air-fuel ratio is too low. By limiting the power on Diesel fuel alone to about 80 per cent of the normal maximum and aspirating gasoline into the intake, it is possible to obtain a power increase of about 12 per cent at all speeds between that of maximum torque and maximum power without increased smoking or the occurrence of knock. Owing to improved economy, the fuel cost per brake horsepower per hour using gasoline aspiration, is not increased. Modifications required to the engine are slight and the fuels required to operate are readily obtainable; even if the auxiliary fuel should run out no damage would be done to the engine.—*Paper by L. D. Derry, E. M. Dodds, E. B. Evans, and D. Royle, read at a general meeting of the Institution of Mechanical Engineers, 8th January 1954.*

Testing Refrigeration Compressors

A new method for measuring the capacity of refrigeration compressors has been developed. This method has been used in Norway since 1942. A piping diagram for the system is shown in Fig. 1, while Fig. 2 shows the process in the temperature-entropy plane. The evaporator and the greater part of the condenser are eliminated. The hot refrigerant gas from the compressor (a) is only cooled sufficiently, so that direct throttling to the suction pressure will give the desired gas state in the suction line. The amount of gas circulated can, of course, be measured in the usual way by means of a calibrated orifice or similar equipment. It is difficult to get accurate results this way, however. The nozzle coefficient will often be considerably influenced by even a slight oil coating, and elimination of pressure vibrations is not always too easy. A special calorimeter (c) is therefore used for this purpose. In the calorimeter the gas is cooled by water to a temperature (point 3 in Fig. 2) well above the saturation temperature. In order to eliminate every possibility of condensation in the calorimeter, the water inlet temperature should also always be maintained above this limit. The gas and water inlet and outlet temperatures are measured, and the quantity of cooling water determined by means of a flow meter (g). Is the heat transferred in the calorimeter Q_{cal}, and the refrigerant enthalpy difference $i_2 - i_3$, the weight of refrigerant circulated will be $G_k = \frac{Q_{cal}}{i_2 - i_3}$, and the refrigerant capacity $Q_o = G_k (i_u - i_s)$. When the specific heat of the refrigerant in question is known with sufficient accuracy for the superheated region, the capacity

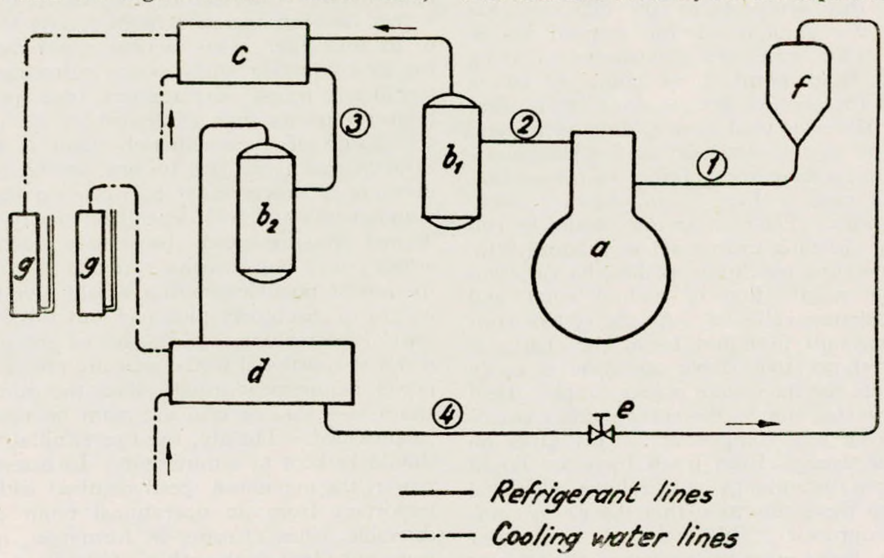


FIG. 1—*Diagram of the circulation test method*

(a) Compressor; (b_1 and b_2) Oil interceptors; (c) Gas flow calorimeter; (d) Gas cooler; (e) Expansion valve; (f) Surge tank to reduce pressure vibrations; (g) Flow meters for cooling water. The number of designations refers to the states in Fig. 2.

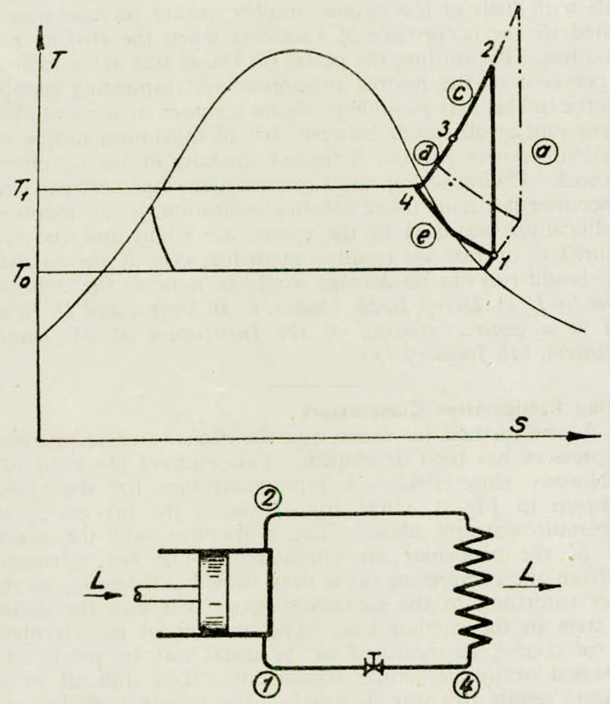


FIG. 2—*Temperature-entropy diagram of the circulation process; thin lines indicate the complete refrigeration cycle*

can thus be found with good precision, without any requirements to the measuring equipment which are difficult to fulfil The pressure drop through the calorimeter should be checked, so that correction can be made for the influence on the temperature in point 3 if necessary. It is not difficult, however, to maintain the pressure drop below the level where such a correction is required. Lubricating oil following the gas stream may affect the results considerably, and result in overestimation of the capacity. It is therefore always necessary to install an efficient oil interceptor (b_1) in the gas line before the calorimeter. To check its efficiency it is advisable to install a second interceptor (b_2). With this arrangement no difficulty has been experienced, and the amount of oil trapped in b_2 has been negligible. The necessary additional cooling to bring the gas state from point 3 to point 4, giving the desired condition in the suction line, is done in the heat exchanger (d) (Fig. 1). Water is used as a cooling agent, and the heat removed is found in the same way as for the calorimeter. It is necessary to increase the volume on the suction side by installation of a tank f (Fig. 1), in order to avoid unusual pressure fluctuations. The suction line should be run from the bottom, so that the tank cannot act as a liquid trap. Adjustment to desired operating conditions is done by variation of the refrigerant charge and the flow of cooling water, and by manipulating the regulating valve (e). As the system contains practically no refrigerant in liquid form, the charge is very small. Experience shows that stable operation is easily obtained. The heat balance for the system is very simple. Heat given off to the cooling water and to the surroundings equals the power consumption of the compressor. This gives an excellent check on the accuracy. Even when there are liquid particles in the suction gas, the enthalpy can easily be calculated from the heat balance, on the assumption that the gas is completely dry after the compressor. This is an advantage over the usual testing methods, based on measurement of the amount of refrigerant circulated. On the other hand, the method is, of course, limited to such cases, where there is a considerable superheat after the compressor.—*G. Lorentzen, Annexe 1953-1 au Bulletin de l'Institut International du Froid; pp. 47-52.*

Combination Propulsion Plants for Naval Vessels

The full speed required of any vessel fixes its total installed shaft horsepower. However, total shaft horsepower of combatant Naval vessels is utilized only a very small percentage of the total underway time. Thus weight penalties are incurred during the entire life of the ship to meet full power requirements that exist only occasionally. This problem has always been recognized. Many schemes have been devised to maintain high efficiency at cruising powers and at the same time to meet the full speed power requirements of Naval combatant vessels. None has resulted in a drastic reduction in the total weight of machinery plus fuel; in fact some have resulted in an increase in total weight. The prospects of accomplishing major weight reductions in the foreseeable future with conventional steam or Diesel propulsion plants is not promising for large powers. It is vital, therefore, that serious consideration be given to other arrangements, such as the application of simple, lightweight booster type gas turbines in combination with base load machinery, which has good thermal efficiency under cruising conditions. Substituting booster gas turbine power for the upper portion of the presently installed shaft horsepower would result in a plant of reduced cost, less maintenance, lower weight and less space. A booster gas turbine is defined as a simple open cycle gas turbine with separate power turbine, but without refinements, such as intercooling and regeneration. Application of such units to propulsion of Naval vessels offers a sound and practical method of decreasing the machinery weights of some types of combatant vessels by as much as 20 per cent. At the same time fuel consumption at cruising power is appreciably reduced, thereby increasing the endurance by as much as 25 per cent. Part of this increase in endurance is due to the better efficiency of the cruising plant which would operate at a higher percentage of its maximum power. In addition, the engine room length can be decreased and an approximate reduction of about 25 per cent of the moulded volume will result. This is important because space is also a major military characteristic. The risk involved in construction of an experimental vessel is negligible, as will be demonstrated. Consideration already has been given to this type of plant, and much more will be given in the future. The gas turbine suited to such an arrangement will have low specific weight (about 2lb. per s.h.p.), a rating of 5,000 to 10,000 s.h.p., a specific fuel consumption of about 0·7lb. per s.h.p.-hr., the lowest feasible air mass flow and a full power life of approximately 500 hours. The fundamental prerequisite for advantageous application of a combination plant is that the ship operate at high powers for only a small portion of its total life. This tactical requirement actually does exist for all combatant ships except submarines. It is particularly significant in light displacement, high speed vessels with limited bunker capacity such as destroyers. A major consideration in the design of a combination plant is the division of power between steam and gas turbine booster power. The optimum division of power must be made on the basis of operational requirements and will depend on the type of vessel, maximum desired cruising speed, the relative machinery weight, and the astern power requirement. At first it would appear that use of the lowest possible cruising would give the maximum gain in weight of machinery plus fuel, but this is not quite true for at least three reasons. First, the weight of auxiliary machinery is not proportional to the cruising power. Secondly, the astern power requirement usually fixes the minimum capacity of the steam base load or cruising plant because boiler capacity must be provided. Thirdly, the operating time of the gas turbines should be kept to a minimum. In determining the division of power, the maximum speed required with the cruising plant is important from an operational point of view because it is desirable, when cruising in formation, not to have to change from one plant to the other. Calculations indicate that a steam to gas turbine ratio of about 50 per cent is probably desirable for most light displacement high speed combatant vessels. This higher percentage of steam power reduces the maintenance of the gas turbine portion of the plant and adds appreciably to its life. The advantages of combination machinery plants are as follows: 1. A decrease in overall weight and space, resulting in more speed for the same displacement, increased endurance or greater installed power. 2. A decrease in maintenance and repair; also easier handling during maintenance of the smaller, lighter machinery. 3. Quick starting: the ship can get under way on the gas turbines alone. This is a limited advantage because the ship cannot be backed down unless controllable pitch propellers or reverse gears are installed. 4. Fast acceleration from maximum cruising power to full power. The gas turbines require a very short warm-up period and will provide quick power. It is important to note that in attaining the above advantages, the steam plant is entirely conventional, and is capable of driving the ship at about 80 per cent of full speed. Moreover, the overall plant reliability is greatly enhanced because of the presence of two separate and independent sources of power, tied together only at the reduction gear. The main disadvantages of combined plants are: 1. The necessity for selecting a fuel acceptable to the gas turbine booster units. When the base load plant does not burn the same fuel as the gas turbines, the percentage of gas turbine fuel to base load plant fuel must be decided by operational requirements. 2. The arrangement requires a rather special gear configuration to accommodate the larger number of pinions. 3. Some extra ingenuity is required in disposing the various turbines around the gear.—*Paper by Commander J. J. McMullen, U.S.N., presented at the Semi-annual Meeting of the Society of Mechanical Engineers, Los Angeles, 28th June 1953. Journal, The American Society of Naval Engineers, Inc., November 1953; pp. 681-687.*

Cathodic Protection

This paper reviews the methods of cathodic protection used in the Royal Navy. Cathodic protection of the outer bottoms of H.M. ships is at present applied only to the Reserve Fleet, because on active ships the disadvantages of the extra weight and resistance caused by cathodic protection equipment is considered to be excessive. Single ships are protected by magnesium anodes, groups of ships by impressed currents. The special problems of ships laid-up in fresh water are mentioned. Cathodic protection is often provided for floating docks, jetties, and other structures in the sea, as well as to underground pipes. A special form of silver-silver-chloride electrode and a voltmeter for use in measurements of pipeline potentials are described. In enclosed spaces on board ship, which may be filled with sea water, cathodic protection involves a risk that hydrogen may collect in pockets; in such spaces zinc anodes seem to be the only safe and simple agents. Laboratory experiments with zinc anodes are described which show improved performance with high purity zinc, or with zinc alloyed with one per cent magnesium. Amalgamation of either material gives a major improvement in active life. The effects of cathodic protection on anti-corrosive and anti-fouling paints are briefly discussed. Finally, experiments on the cathodic protection of aluminium and its alloys, and of stainless steel in sea water, are described.—*Soc.Chem.Ind. paper by J. T. Crennell, read at Symposium on Cathodic Protection, 13th November 1953. Journal, The British Shipbuilding Research Association, December 1953; Vol. 8. Abstract No. 8,390.*

Notes on Cargo Handling

The boom and winch is still the universal standard on ships today, and the burton or married fall is the standard method of using this gear. Like so many things about the ship this gear looks very crude when compared to modern handling equipment used in other industries, and yet nothing revolutionary has come along to displace it. Some European ships have been equipped with shipboard rotating cranes, but this idea does not seem to be spreading very rapidly. The author has made some paper studies of crane applications to new cargo ship designs in an attempt to evaluate their possibilities. The first problem is the number and location of the cranes. Keeping within the available deck space between adjacent hatches it appeared that the choice would be limited to one of the three schemes shown in Fig. 2. Scheme A with cranes mounted port and starboard is the best arrangement, but is also the most expensive. It also has the disadvantage that in the event of all loading being done from the pier side, one of the two cranes would be idle. Scheme B with a single crane mounted port and starboard is the best arrangement, for a small ship, but to make such an application to a ship the size of the *Mariner* would result in an unreasonably long boom. Scheme C represents a compromise between the first two, and visualizes a single crane mounted on tracks so that it

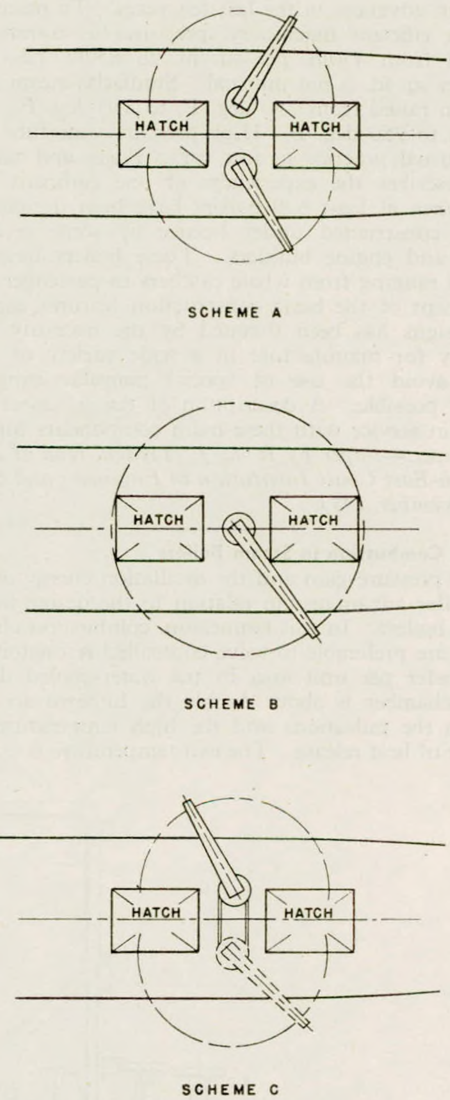


Fig. 2—Shipboard Rotating Cranes.

could be moved to serve either side of the ship. With this arrangement there has been added the complication of providing tracks, and the mechanism for the translatory motion of the crane. Taking the best arrangement, Scheme A, it is still necessary to load from pier and lighter simultaneously in order to get two machines working one hatch. Loading from one side only, the one machine would have to operate at double the speed of a burton rig to give the same loading rate. Comparing the available data on the burtoning cycle with the cycle of the proposed crane, it did not appear that this could be achieved. On the other side of the ledger, the crane has the decided advantage of greater flexibility. It can pick up a load from any point within its radius of operation, and deposit it exactly at the desired spot. Also, with the usual hoisting arrangement of a crane, a greater range of loads can be handled without the delays incident to the doubling up of tackle on a boom-winch arrangement. The only conclusion that could be drawn from this study was that it would be necessary to make an actual installation in order to obtain a reliable answer to the possibilities of cranes. It is certainly hoped that such an installation will be made.—*F. G. Ebel, Bulletin, The Society of Naval Architects and Marine Engineers (New York), February 1954; Vol. 9, pp. 19-27.*

Ten Years in Design of Marine Boilers

The design of marine oil-fired watertube boilers has made significant advances in the last ten years. To meet the demand for more efficient machinery, pressures in common use have increased from 450lb. per sq. in. to 650lb. per sq. in., and 850lb. per sq. in. is not unusual. Similarly, steam temperatures have been raised from 750 deg. F. to 850 deg. F., and in some instances to 950 deg. F. High-pressure watertube boilers have become usual practice in fast cargo liners and tankers. This paper describes the experiences of one company during this period when at least 600 boilers have been designed and have all been constructed under licence by some seventeen shipbuilders and engine builders. These boilers have been fitted in vessels ranging from whale catchers to passenger liners. The development of the basic construction features common to all these designs has been dictated by the necessity to maintain suitability for manufacture in a wide variety of boiler shops and to avoid the use of special manufacturing machinery wherever possible. A description of the successes and defects revealed in service with these main components forms the bulk of the paper.—*Paper by R. L. J. Hayden, read at a meeting of the North-East Coast Institution of Engineers and Shipbuilders, 27th November 1953.*

Pulsating Combustion in Steam Boilers

The pressure-gain and the oscillation energy of a pulsating burner offer advantages in relation to the design and operation of steam boilers. In this connexion, combustion chambers with diffusers are preferable to valve-controlled resonator tubes. The heat transfer per unit area in the water-cooled diffuser combustion chamber is about double the hitherto accepted figure, owing to the pulsations and the high temperature due to the high rate of heat release. The exit temperature is extremely high for such a small combustion chamber, and it would be dangerous, therefore, to allow the ash-laden hot gases to come into contact immediately with the convection surfaces of the boiler. A radiation chamber is, therefore, interposed between the burner and the boiler heating surfaces, the design being such that pulsations of the gas occur also in the radiation chamber, with consequent improvement of the heat transfer there as well as at the secondary heating surfaces. In conventional boilers the heat transfer in the radiation part takes place almost entirely by radiation, the proportion by convection being relatively small. As the radiation from the gas depends only on its temperature, the radiation heat transfer is not affected by the pulsations. The latter do, however, greatly improve convection heat transfer by breaking up the stagnant layer of gas at the heat-exchange surface, and thus a much smaller radiation part will be needed for the same total heat transfer than with conventional boilers. If the degree of cooling in the radiation part is sufficient, and there is enough space available there, the convection surfaces can also be installed in that section of the boiler. The convection surface required will be diminished, with pulsating combustion, even more than the radiation surfaces. The propagation of the vibrations is little affected by the tube nests. This section of the boiler may be increased in size sufficiently to permit the installation of adequate heat-transfer surface for the flue-gas exit temperature.—*Engineering and Boiler House Review, Vol. 69, January 1954, pp. 4-7; February 1954, pp. 34-38.*

Free Piston Type Gas Turbine Plant

This paper is intended to set forth the latent possibilities in heavy-duty free-piston machinery. The cycle promises Diesel efficiency from a simple low-temperature gas turbine having low first costs. Many features are described in the paper. Cooper-Bessemer has built a test plant and is now investigating its operating characteristics. No conclusions have been announced. Economic evaluations are being made requiring additional field work which includes an examination of the general acceptance by engineering and operating people who ultimately will decide the actual utility. Detailed cost studies of complete plants, including careful estimates of the machinery, have been made for electric power generation, pipeline pumping, and marine installations. This propulsion plant

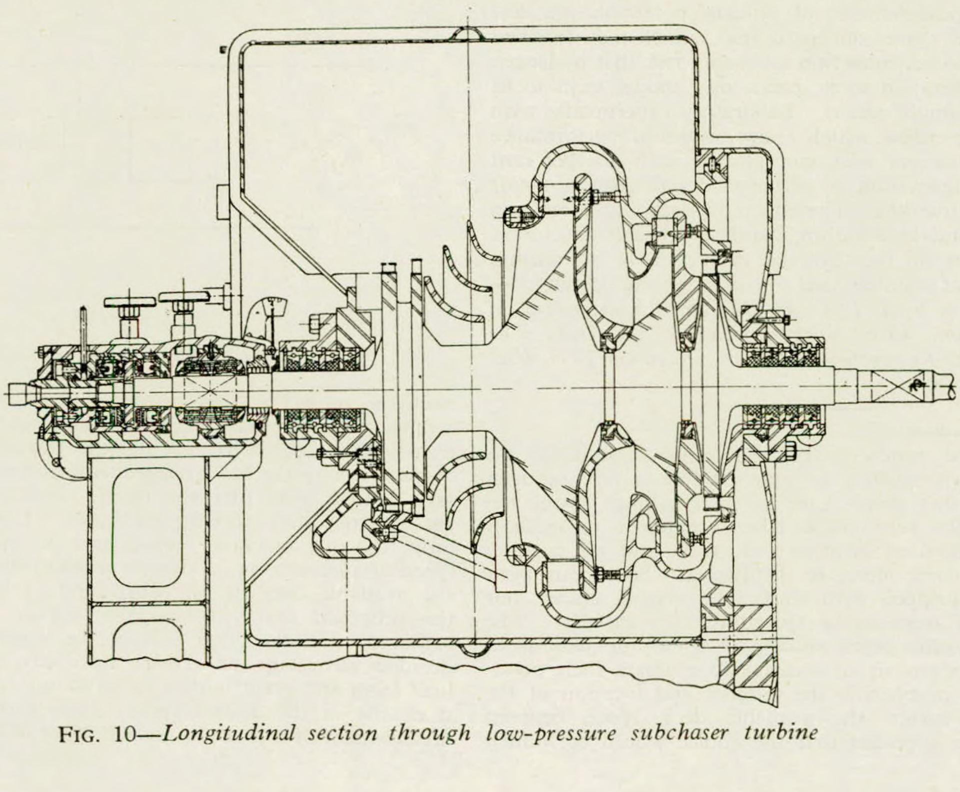


FIG. 10—*Longitudinal section through low-pressure subchaser turbine*

combines all of the desirable characteristics of the steam turbine with the economy of the Diesel engine and avoids most of the undesirable elements. The outstanding advantages of the free-piston turbomotorship are as follows: 1. Combination of reliability and high thermal efficiency with reduced weight and space requirements. 2. Low installed cost with low maintenance and other operating costs. 3. Great flexibility of arrangement permissible. The gas generators may be placed in a wide variety of positions since the coupling to the gas turbine is pneumatic through the exhaust gas lines. Moreover, under-way maintenance to individual gas generators can be accomplished. 4. Instant availability of power. No particular warm-up period is required to get the plant under-way. 5. Easy accessibility for repair and maintenance. Head-room of only 7 feet above floor level is required to remove major components. Low number of moving parts, all easy to reach. 6. Complete absence of mechanical vibration. Such foundations as are required are light and simple. 7. Low centre of gravity for installed machinery. 8. Astern operation is accomplished with a controllable-pitch propeller or an astern turbine either in the same or a separate casing with a 1 per cent loss in overall efficiency. The reversal can be accomplished without shutting down the gas generator. Flexibility here is about the same as in a steam plant. Furthermore, the gas generator-gas turbine combination is well suited to electric drive. The high speed of the gas turbine is conducive to the use of lightweight, compact, electrical equipment. 9. Rapid response characteristics well suited to the manœuvring demands for a vessel. 10. Good part-load performance. In addition to the high part-load efficiency of the individual gas generators, it is necessary to operate only the number required to meet load requirements. The shutdown units can be started immediately as required. 11. Combination of all the outstanding features of motorships and, at the same time, elimination of many of the complications of Diesel engines. The free-piston unit has good economy with simple equipment designed for a low-temperature turbine and low maintenance expense. 12. Relative freedom from smoke under all normal conditions. 13. Good adaptability to various types of ships. Because of the anticipated lower costs and the high thermal efficiency at all loads, there are excellent opportunities for application in such ships as auxiliaries, transports, tankers, cargo vessels, and tugs. 14. Reserve torque overload capacity comparable to a steam-turbine plant. The gas generator, for instance, can be made to deliver great overloads in mass flow almost instantly and without the thermal inertia of a steam boiler. This can develop high torques at reduced turbine speeds. 15. Low-cost construction and the elimination of all critical materials. An established standard size of gas generator may be used for a large range of powers by multi-unit application to selected turbine sizes of relatively simple construction and design. The mass-production possibilities of such a programme, in the event of a national emergency, are obvious. A preliminary layout of the free-piston gas-turbine installation in a C-3 cargo vessel is shown in Fig. 10. It should be remembered that the ideal method of applying any prime mover to a vessel would be to design the ship for the specific power plant. Fig. 10 is a layout of a conversion. It includes many compromises which would not be necessary in an original design. For example, in a vessel designed for free-piston turbomotors, the engine-room height could be much less than that for steam turbines with their boilers and Diesel-engine installations. Also, the ship's service generators might be coupled up with a spare gas generator, and the donkey boiler could be "fired" by the exhaust gases of the gas turbines. All of the gas generators are located at one level. Ample space is provided for disassembly and servicing. The exhausts are run through individual lines to a common turbine and the free-piston motors rest on rails so situated that operation and adjustment are easy at the operating level. The scavenging headers, gas generator to turbine interconnecting piping, and most of the cooling piping run below this level, where they are easily accessible from below. The gas turbine and reduction gear are located at the lower level, so that suitable connexion can be made to the propulsion shafting. The turbine-nozzle arc is divided into six sections, one connecting to each gas-generator exhaust. The volume in all exhaust lines is approximately the same; surge chambers are added to the shorter ones to balance. The gas-generator intakes are grouped for minimum interference without excess size, and a 4-ft. diameter stack connexion suffices for this arrangement. The turbine exhaust requires a 3·25-ft. diameter or equivalent area.—*J. J. McMullen and R. P. Ramsey, Transactions, A.S.M.E., January 1954; Vol. 76, pp. 15-29.*

The Length of Oil Flames

The combustion of a fuel involves two processes; the fuel has to be mixed with the combustion air before the chemical reactions which comprise combustion can take place. Combustion will occur as soon as air, fuel, and active reaction carriers from a region where combustion is already taking place are mixed in suitable proportions. Since the chemical reactions are very rapid at flame temperatures, it is reasonable to assume that these occur within a very short period after the combustible mixture is formed, so that the rate of combustion, and therefore the flame length, is mainly determined by the rate of the mixing process. In a previous paper the theory of mixing in turbulent jets was used to derive an equation relating the length of a free turbulent jet flame to the flow conditions at the nozzle. This equation was found to hold for vertical oil and gas flames in free air, provided that the initial momentum of the jet was large compared with that resulting from the buoyancy force acting on the flame. In this paper, a modified form of the flame length equation has been obtained, applicable to fuels of widely varying calorific value, and also relationships to correct the free flame for the different conditions found in a combustion chamber.—*Journal of the Iron and Steel Institute, March 1954; Vol. 176, pp. 270-273.*

Apparatus for Checking Contours of Propeller Models

Models of marine propellers are usually checked with an apparatus which determines the pitch and the blade thickness at certain points. It may be desirable, however, to obtain a more extensive check of the correctness of the blade contours. This can be achieved by means of the apparatus described below, which automatically plots out the blade sections on paper. The main features of the equipment are shown in Fig. 1. The propeller *P* and the recording drum *T* are mounted on a shaft *B*, which is slowly rotated by the motor

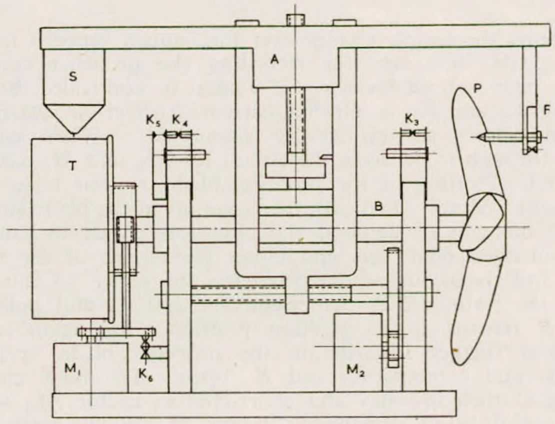


FIG. 1

M_1 through spur and worm gearing. A feeler *F* and a stylus *S* are mounted at opposite ends of a horizontal arm *A*. This arm can be raised or lowered by rotating a micrometer screw, so as to obtain accurate adjustment for each cylinder-section diameter. The arm *A* is supported by a frame on the shaft *B*, and can be displaced axially along the shaft by motor M_2 driving a lead screw. As the feeler and stylus are both rigidly attached to the arm A, the distance between them is always the same, so that the stylus will give an accurate reproduction on the recording drum of the relative motion of the feeler around the propeller. The feeler and stylus controls are shown schematically in Fig. 2. The point of the feeler F is mounted on a leaf spring f. When the feeler touches the propeller blade, it closes an electric circuit through the propeller blade, the feeler, and coil L_1 in the quick-acting polarized relay R, which requires very little current, thus avoiding high-intensity sparks which might otherwise damage the point of the feeler and burn holes in the propeller blade surface. When current flows through coil L_1, the relay is triggered and closes contact K_2. As a result of the current passing through the stylus relay, the stylus is pressed downwards on to the recording drum and perforates a small hole in the paper. As the feeler moves further along the propeller blade, spring f is deflected and interrupts the current at one of the contacts K_0 and K_1. In tests, it has been found that the motion of the feeler is less than 0·01 mm. during the period of effective contact at K_0 and K_1 and between the feeler point and the propeller blade. As the polarized relay R does not return to its initial position when coil L_1 is not energized, this relay is provided with another coil L_2. The circuit of L_2 is closed by contact K_2 when the stylus is depressed. This causes relay R to return to its starting position, thus breaking the circuit in the stylus relay. A spring returns the stylus to its initial position. Fig. 2

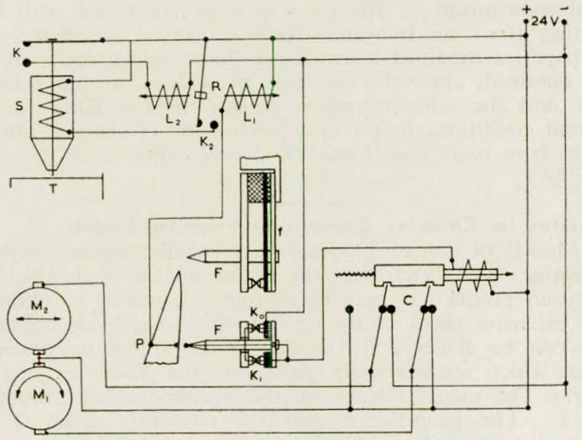


FIG. 2

also shows the simple change-over mechanism between motors M_1 and M_2 necessary for recording the propeller contour. When relay circuit-breaker C, which is controlled by the contacts K_0 and K_1, is without current, it short-circuits motor M_1 and closes the circuit for motor M_2. When current passes through C, it closes the circuit for M_1, and M_2 is short-circuited. Plotting of the propeller blade contour takes place as follows: Motor M_2 moves the feeler up to the blade surface. At the moment of contact, the circuit is closed by coil L_1. The polarized relay acts and closes the circuit of the stylus relay, and the stylus needle perforates the paper. Contact K above the stylus closes the circuit for coil L_2 and polarized relay R returns to its starting position. M_2 again moves the feeler further inwards on the propeller blade, spring f deflects, and contacts K_1 and K_2 open. The relay circuit-breaker C then operates and short-circuits motor M_2, which stops rapidly, and switches in motor M_1, which rotates the propeller and the recording drum in such a way that the feeler will move a slight distance away from the blade, depending on the pitch of the propeller. As soon as the feeler point leaves the blade, a new current impulse is obtained through the polarized relay, actuating the stylus relay and thus producing a hole in the paper. Motor M_1 is short-circuited and stops rapidly, then motor M_2 starts again, and the same operation is repeated. Thus, the motors are alternately switched in and out of circuit, with M_2 moving the feeler towards the blade and M_1 rotating the blade away from the feeler. Motor speeds can be individually adjusted by means of rheostats, so as to obtain variations in operating speed, and to space perforations as desired on the paper. A set of recording drums of various sizes makes it possible to record true blade contours at various radii. Two feelers, one for the pressure side and one for the suction side of the propeller, can be swung alternately into position.—*P. Langsethmo, Teknisk Ukeblad, 24th September 1953, Vol. 100, pp. 763-764; Abstract in The Engineers' Digest, February 1954, Vol. 15, p. 55.*

New 10,000-ton Tramp Motor Ship

During the last few years the Flensburger Schiffbau Ges. has paid especial attention to the development of 10,000-ton tramp motorships, and the following vessels were built: M/S *Luise Leonhardt* with five hatches, of the open shelterdeck type, about 10,300 tons d.w., propelled by a M.A.N. two-stroke cycle six-cylinder Diesel engine developing 3,500 b.h.p. at 110 r.p.m.; M/S *Nordland* with five hatches, of the closed shelterdeck type, about 11,500 tons d.w., with the same engine as above; M/S *Schwanheim,* with five hatches, about 11,500 tons d.w., same engine as above, but arranged aft. A fourth vessel of the open shelterdeck type of 10,230 tons d.w. with the same type of engine arranged aft, is building. The leading particulars of the *Schwanheim* are: 144·46m. overall, 132m. between perpendiculars, breadth moulded 18·4m., draught 8·58m., service speed 13 knots. Ten 3-ton and two 5-ton winches are installed, all electrically driven.—*Hansa, 2nd January 1954; Vol. 91, pp. 130-134.*

Under-water Welding

Although under-water welding does not, at present, have the widespread application of under-water cutting, it is a procedure of great potential value. Recent advances in the art, including simplification of the welding technique and the use of larger electrodes in all positions, have increased the potentialities of under-water welding. Under-water welds in mild steel, made under test conditions at several United States Navy Yards, have consistently developed over 80 per cent of the tensile strength and 50 per cent of the ductility of companion welds made under normal conditions. For most under-water purposes, the 20 per cent loss in strength is not objectionable; but, in some applications, the 50 per cent loss in ductility may be a serious disadvantage. The loss in ductility is explained by the hardening effect, caused by the very drastic quenching action of the surrounding water. In all under-water welding operations, care must be taken because of the dangers attendant on the use of the electric arc under water, and strict adherence to the safety rules must be observed. There are a number of limitations to under-water welding and these fall into four main categories:—(*a*) Size of the patch to be applied, (*b*) purpose of patch, (*c*) contours of the hull, and (*d*) repairs to protuberances. Except for repairs to leaky rivets and split seams, welding under water should only be an additional method for the securing of a patch on any hull. The size of the patch that can be safely welded differs with all types of work to be performed. An unusual instance was reported, wherein a steel plate of about 10 feet by 20 feet was successfully welded on a ship's hull during operations in Italy. Numerous other instances have been brought to light, however, in which welds of more than 5 feet by 9 feet have failed. The safest rule to follow is to use a cofferdam where practical. When that is impossible, a patch should be used, secured with bolts, and then welded to ensure greater watertightness. Even while a ship is in dry-dock, it is a major operation to replace a steel plate on a ship's hull; and to carry out a job of this magnitude under water would, of course, mean considerable work, because of the curvatures of the hull. Measurements of the damaged area, and the weight and bulkiness of the plate, must be dealt with at the same time. Direct current is preferred for under-water welding, mainly because it is safer to use than alternating current. A welding generator of at least 300-amp. capacity is usual, although a 200-amp. machine set for peak, or near peak, load may serve in an emergency. A 300-amp. A.C. transformer may be used where a D.C. generator is not available, but, again, under such conditions, special attention must be paid to safety measures. The electrode-holder must be fully insulated, durable, and allow for easy electrode changing. Electrode-holders with metallic jaws, such as are used under normal conditions, are difficult, if not impossible, to insulate fully, and their use in water should, therefore, be restricted to emergencies when the standard holder is not available. Too much importance cannot be attached to proper preparation of the surface to be welded. No satisfactory weld can be obtained over thick paint, rust or marine growth. In a multi-pass weld, each bead must be thoroughly cleaned before the following bead is applied. Plates to be added can be prepared with a sander or grinding wheel, or by flame-cleaning top-side; but cleaning the necessary base-metal surfaces under water will involve strenuous application of the scraper, chipping hammer and/or wire brush. The development of pneumatic tools for the under-water cleaning of surfaces to be welded is under way, but is not yet complete. The possibility of under-water flame-cleaning has also been given consideration. The diver must be so attired that his body is fully insulated from the work, the torch and the water itself. Experience shows that, when employing the less hazardous D.C.-power source, the standard deep-sea diving dress is suitable and may be considered reasonably safe. Rubber gloves, attached or unattached, are an essential part of the dress. When using A.C., it is recommended that precautions be taken to insulate the diver's head from the helmet. This may be done by the diver wearing a skull cap, and by covering the button on the exhaust valve with rubber tape.—*D. Brooks, The Shipbuilder and Marine Engine-Builder, February 1954; Vol. 61, pp. 105-107.*

German Geared Diesel-engine Cargo Liner

The cargo-passenger liner *Schwabenstein* is the first of six similar ships designed and building for the Roland Linie. The *Schwabenstein* has a dead-weight-carrying capacity of 9,100 tons, a gross tonnage of 9,200 and has berths for a maximum of 86 passengers, all arranged in outside cabins. She was built by Bremer Vulkan, Bremen-Vegesack, and the main characteristics are as follows:

Length overall	537ft. 9⅜in.
Length b.p.	492ft. 1⅝in.
Extreme breadth	63ft. 7¾in.
Depth to first deck	39ft. 4½in.
Depth to second deck	31ft. 4in.
Draught	26ft. 2in.
Service speed	17¼ knots

The hull of the ship, which has been welded to an appreciable extent, is sub-divided to form five cargo holds, the total bale capacity being 608,565 cu. ft., and the grain capacity 670,967 cu. ft. Special strengthening for navigation in ice has been incorporated in the structure. All of the winches serving the derricks are electrically driven and are of Atlas Werke manufacture. Schat-type davits are fitted on the boat deck, and carry aluminium alloy lifeboats, the lifeboat winches being electrically driven. The propelling machinery comprises two seven-cylinder M.A.N. double-acting two-stroke engines, developing a total of 10,500 s.h.p. in service at 220 r.p.m. and each driving through Vulcan hydraulic couplings and reduction gearing to a single propeller shaft. The reduction gear was manufactured by A.G. Weser and has a ratio of 1·77:1, which gives a propeller speed, at the above-mentioned power, of about 124 r.p.m. The b.m.e.p. at this output is about 4·9 kg. per sq. cm. in the bottom cylinder and 5·8 kg. per sq. cm. in the top. Each engine has a cylinder bore of 530 mm. in diameter, the stroke being 800 mm. A useful feature of the engine-room, and one which should receive greater attention by other shipbuilders, is the thorough manner in which every auxiliary in the engine-room is name-plated. Plastic plates are fitted to each pump, giving the names of the manufacturers of the pump and motor, the rated outputs, and the services for which the pump can be used. Similar plates are fitted to all heat exchangers, giving the heating surface and other useful information, while all cross connexions are clearly marked.—*The Motor Ship, March 1954; Vol. 34; pp. 525-529.*

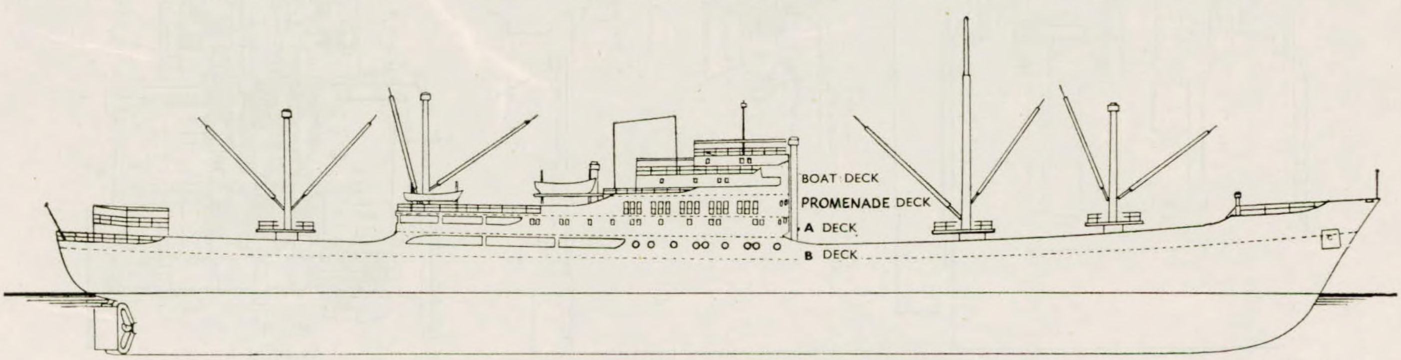
New Liner for Australian Service

The twin-screw turbine steamship *Arcadia,* which at 29,734 tons gross is the largest liner in the fleet of the Peninsular and Oriental Steam Navigation Company, has been delivered to her owners by John Brown and Co. (Clydebank), Ltd. In general, the *Arcadia* is similar to the *Himalaya,* built in 1949, but there are certain individual features which enable her to be readily distinguished. Perhaps the most prominent is the new style of funnel, designed to eliminate the danger of smuts falling on the wide expanse of sports decks, by co-operation between the builders, the owners, and Thermotank, Ltd., in whose wind tunnel smoke tests were carried out.

Principal Particulars of the "Arcadia"

Length overall	720ft.
Length b.p.	668ft.
Breadth moulded	90ft. 6in.
Depth to C deck	49ft. 9in.
Tons gross	29,734 tons
Complement:	
First class	679
Tourist	735
Officers and crew ...	711
Total	2,125
Cargo capacity:	
General (bale)	211,000 cu. ft.
Insulated (bale)... ...	158,500 cu. ft.
Normal power	34,000 s.h.p. at 130 r.p.m.
Service speed...	22½ knots

The under-water form is of normal design based on model experiments carried out at the builders' experimental tank. The bower anchors are housed in recesses port and starboard. The stem is curved in contour and raked forward. The stern, which is of the cruiser type, has an overhang of 34 feet 6 inches



Profile of the passenger liner Schwabenstein, *of 9,200 gross tons* abaft the after perpendicular. The rudder, which is of the semi-balanced type, and of streamlined form, is supported on a pedestal bearing at the steering gear flat. The bilge keels are of plate construction. Shaft bossings of the overhang type have been designed to give an uninterrupted flow of water to the propellers. Frames are spaced 33 inches apart, reduced forward and aft to Lloyd's requirements. Main frames are generally 12-in. channels extending from the bilge to E deck with 10-in. channels in the tweendecks above extending to C deck and 7in. channels from C deck to A deck. Deep web frames are introduced at intervals throughout the length in the tweendecks from C to A decks. Deck beams are 7-in. channels up to and including A deck, cut at the toe of the frame and bracketed with beam knees. The side and bottom frames are out-joggled to suit a normal arrangement of "in and out" shell strakes. Side shell plating is 0·77 inches thick and bottom shell 0·84 inches thick for half length amidships, reduced at ends to suit Lloyd's requirements. The flat keel is 1·14 inches thick, having a 1-in. doubler all fore and aft. Electrical power has been used for every conceivable service. For the management of the hull there is electric steering gear, windlass, capstans, cargo and boat winches, baggage and service lifts, etc. In the engine and boiler rooms all auxiliaries, with the exception of some feed pumps, are electrically driven; and electric lighting, heating and ventilation calls for an electrical power station of considerable size. This station is situated over the forward end of the engineroom and has a total capacity of 3,600 kW., made up of three British Thomson-Houston 1,200-kW. generators each with a 220-volts D.C. dynamo driven by a steam turbine through single reduction gearing. The main propelling machinery was constructed by the shipbuilders and consists of a twin-screw installation of impulse reaction geared turbines designed to give a normal power of 34,000 s.h.p. at 130 r.p.m. and a maximum power of 42,500 s.h.p. at 140 r.p.m. The astern turbines are capable of developing about 65 per cent of the normal ahead power. Each screw is driven by an independent set of turbines, each set comprising h.p., i.p. and l.p. turbines working in series. The turbines are designed for an inlet pressure of 500lb. per sq. in. gauge and a temperature of 800 deg. F. and a condenser vacuum of 28·0 inches at 86 deg. F. sea. The main gearing is of the double helical type with double reduction for the h.p. turbine and single reduction for the i.p. and l.p. turbines, each turbine being connected to the gearing by a claw-type flexible coupling. Electrically operated turning gear is fitted on the l.p. turbine pinions and it is able to turn the main shafting one revolution in 10 minutes. The boiler installation, built by the shipbuilders, consists of three two-furnace Foster Wheeler controlled superheater type watertube boilers designed for a steam outlet pressure from the superheater of 530lb. per sq. in. gauge and a superheat temperature of 850 deg. F. which can be varied down to 600 deg. F. Each boiler has a generating surface of 20,355 sq. ft. and a superheater surface of 2,525 sq. ft. The normal evaporating rate is 105,000lb. per hour and the maximum continuous rate is 165,000lb. per hour. *—The Shipping World, 17th February 1954; Vol. 130, pp. 203-208.*

Valve Gear for Uniflow Engine

In engines of the uniflow type it is practicable and most desirable that control of power output should be effected by varying the ratio of expansion rather than by varying the admission pressure by throttling, while moreover if the engine is governed by a centrifugal or other type of speed responsive governor it is important that frictional losses between the governor and the apparatus actuated by it shall be as low as possible. In the case of control by a throttle valve, the necessity for a steam-tight gland through which its spindle passes with the friction inseparable from such a gland, imposes an additional disadvantage of such a method of control. An object of the present invention is to provide admission valve apparatus for a uniflow fluid pressure engine which will enable a sharp cut-off of working fluid to be obtained, will enable the cut-off point to be controlled to the required degree, while imposing little load due to friction on the governor, and will tend to avoid leakage through the admission valve. According to the present invention a uniflow fluid pressure engine has admission valve apparatus comprising two valves arranged in series. One of these constitutes a cut-off valve and is in the form of a piston valve having a circumferentially extending cut-off edge which co-operates with the cut-off edge of an associated port in its cylinder in such a manner that the point in the reciprocating movement of the valve at which it closes its associated port can be varied by rotational movement of the valve relatively to its cylinder. The other valve constitutes an admission when the piston valve is already open. In some cases the rotational movement of the piston valve relatively to its cylinder to vary the cut-off point may be effected by rotation of the cylinder while the valve remains rotationally stationary, but in most cases would be effected by rotation of the piston valve, the cylinder remaining stationary. A side elevation, partly in section, of the complete engine is given in Fig. 1, while Fig. 2 is an end elevation; and Fig. 3 is a cross-section through the valve apparatus of the

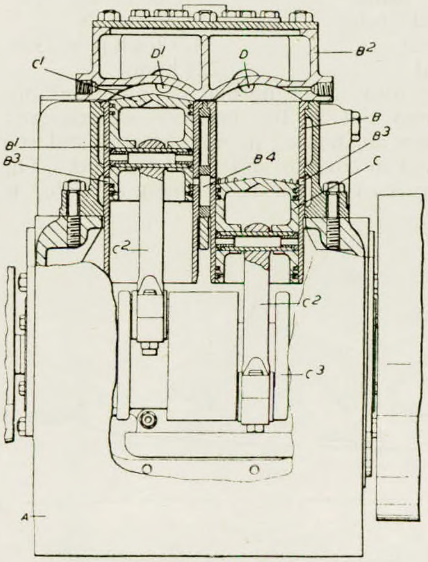


FIG. 1.

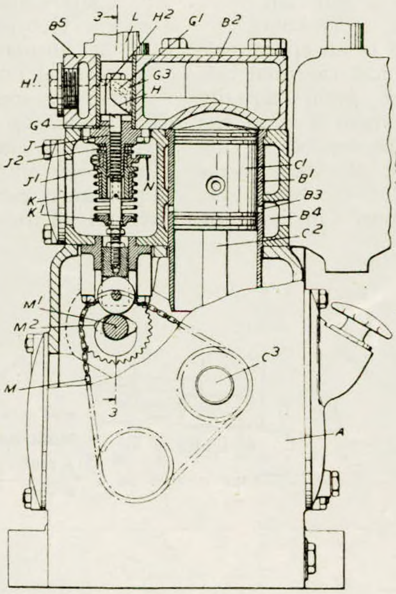


FIG. 2.

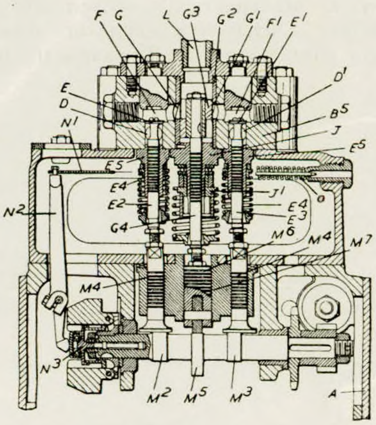


FIG. 3.

engine taken on the line 3-3 of Fig. 2. The operation of the valve apparatus is as follows:—At approximately the moment when each of the working pistons C, C^1 reaches its top dead centre position, the appropriate cam M^2 or M^3 begins to lift its associated poppet valve E or E^1, at which time the associated port G or G^1 is uncovered by the co-operating cut-off edge H or H^1 so that working fluid is admitted from the supply passage L through the port G or G^1, the chamber F or F^1, the valve E or E^1 and the passage D or D^1 to the appropriate working cylinder. As the piston moves downwards on its working stroke the cam M^5 appropriately raises or lowers the piston valve G^3 until at a point in the working stroke of the piston valve determined by its rotational position the appropriate cut-off edge H or H^1 on the piston valve G^3 closes the appropriate port G or G^1 and thus effects cut-off of working fluid from the working cylinder. When the working piston reaches the end of its working stroke exhaust takes place through the exhaust ports B^3 in the normal manner for a uniflow steam engine, and the other piston will then be at its top dead centre position. The other of the cams M^2, M^3 now begins to lift its associated valve E or E^1 so that working fluid is now admitted to the other working cylinder and is subsequently cut off therefrom by the piston valve G^3 at an appropriate point in the stroke of the working piston, again determined by the rotational position of the piston valve G^3. The rotational position of the piston valve G^3 is controlled by the governor so as to retard the cut-off point with reductions in speed of the engine and vice versa, while the speed setting can be controlled by adjustment of the spring O which acts upon the governor mechanism. If the load on the engine exceeds some predetermined value the governor will rotate the piston valve G^3 to a point at which the grooves H^2 come into register with the ports G, G^1 so that these ports remain permanently open, whereupon both the admission and the cut-off of working fluid is effected solely by the valves E, E^1, the conditions under which this can occur being the extreme overload conditions. The cams M^2 and M^3 are therefore so designed as to cause the valves E and E^1 to close at the points in the working strokes of their respective working pistons representing the cut-off points for maximum overload.—*Brit. Patent No. 700,821 issued to H. R. Ricardo. Complete specification published 9th December 1953. Engineering and Boiler House Review, March 1954; Vol. 69, pp. 88-89.*

Italian Tanker

The tankers now building in Italian shipyards under the provisions of the Cappa Law include a series of large steam-driven vessels of some 32,250 tons deadweight. The first of these vessels to complete is the *Mirella d'Amico*, one of a group of three sister ships building for Soc. di Navigazione d'Amico by Cantieri Riuniti dell'Adriatico at their Monfalcone yard. These ships have a service speed of 16 knots with 13,000 s.h.p., but the *Mirella* d'*Amico* achieved a speed of 17·32 knots trials at full load and over 18 knots at half load. The principal particulars of the ship are as follows:

Length o.a.	659ft.
Length b.p.	621ft.
Breadth moulded	86·2ft.
Depth moulded	45·7ft.
Draught at full load	34·2ft.
Deadweight	32,250 tons
Full load displacement	41,500 tons
Cargo tank capacity...	1,520,000 cu. ft.

The ship complies with the requirements of the Registro Italiano Navale, Lloyd's Register of Shipping and the American Bureau of Shipping for the highest class of bulk oil carriers on long routes, and also with the requirements of the Safety of Life at Sea Convention. There are two longitudinal bulkheads in the cargo tanks, dividing the 10 tanks into 30 compartments. From forward to aft, transverse bulkheads divide the ship into forepeak, water ballast tanks, forward fuel oil deep tanks with forward pump room at centre, forward cofferdam, the cargo tanks, after cofferdam, after deep fuel oil tanks and pump room, boiler room, engineroom and after peak. The main propelling machinery consists of a three-stage arrangement of CRDA-Parsons steam turbines, developing 13,000 s.h.p. in service at 100 r.p.m. to give the service speed of 16 knots. Steam is supplied at 595lb. per sq. in. by two Foster Wheeler boilers. Of the deck auxiliary machinery, mention has already been made of the steam winches. The windlass and two warping capstans, the latter being located aft, are also steam driven, while the steering gear is of Hastie electro-hydraulic type.—*The Shipping World, 3rd March 1954; Vol. 130, pp. 260-261.*

Ship's Speed Indicator

This invention concerns a method for measuring the speed of vessels through water by comparing the dynamic and the static pressure beneath the ship. Conventional types of speed indicators working on this principle are provided with air extracting device to separate any air. With this method, deaeration of the water column is effected automatically in the vicinity of the tapping point, i.e. outboard of the vessel. The dynamic pressure meter consists of a streamlined tubular body extending downwards from the bottom of the ship. When the ship is moving this meter measures not only the dynamic pressure x but also the static or draught pressure y, i.e. the value $x+y$. The static pressure y is produced by the

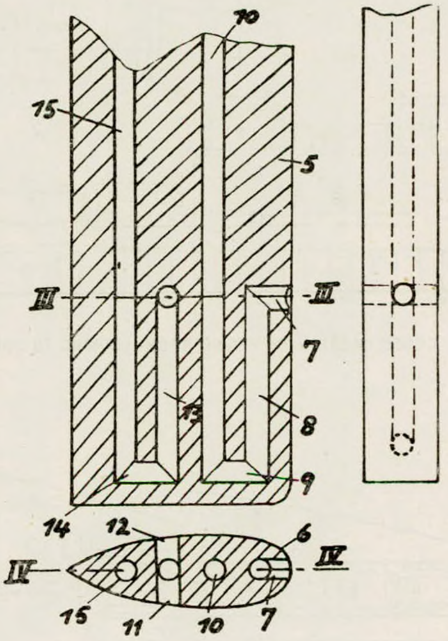
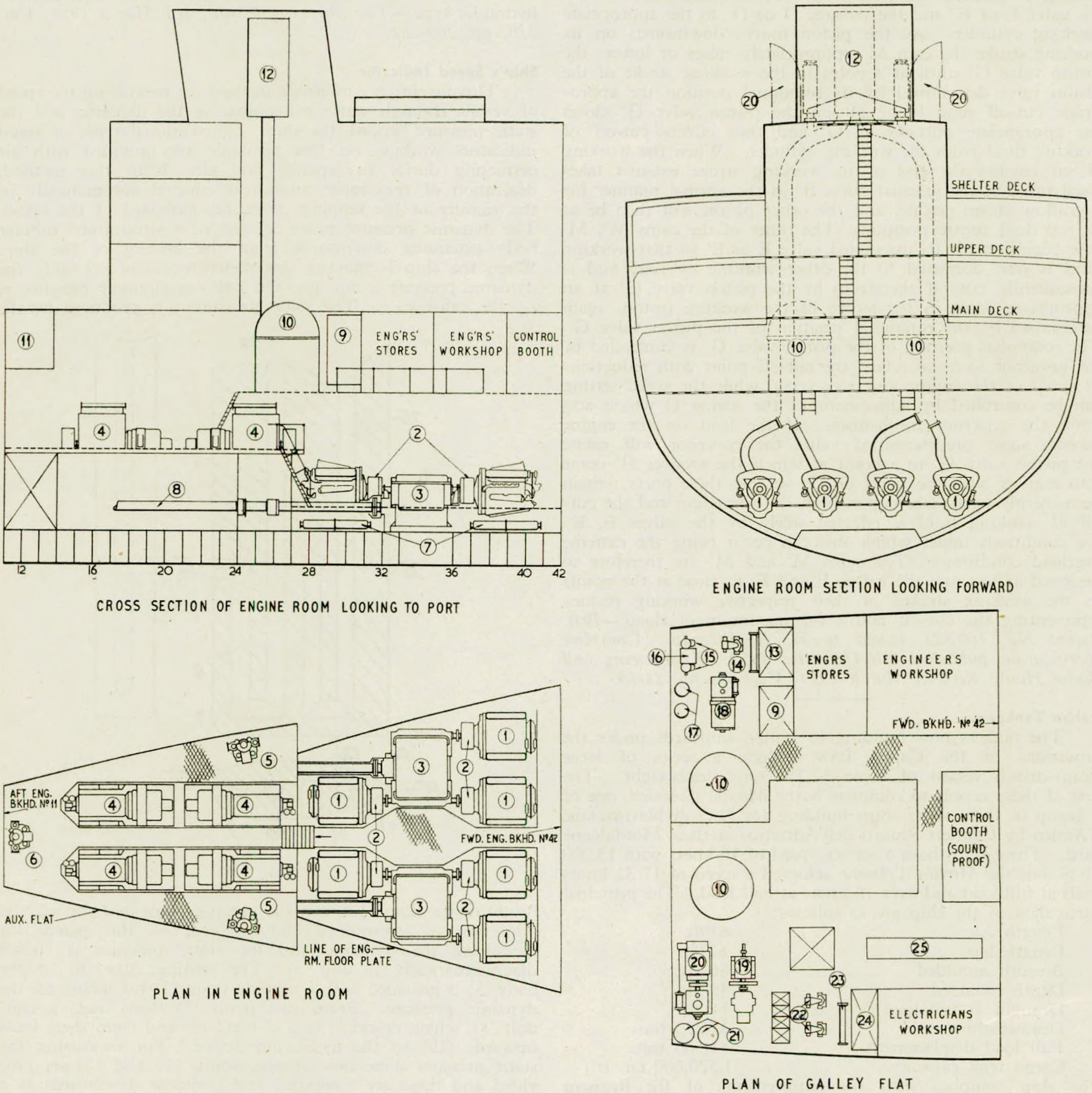


FIG. 1.

height of the liquid between the tapping point and the surface. The type of streamlined body covered by this patent for measuring the dynamic and the static pressure is shown diagrammatically in Fig. 1. The leading edge (6) of the body (5) is provided with a tapping point (7) for taking off the dynamic pressure. From this point (7) there leads a conduit (8), which descends to a U bend (9) and from there leads upwards (10) to the measuring device. For measuring the static pressure alone two tapping points (11 and 12) are provided and these are combined and continue downwards as a conduit (13), which again passes round a U bend (14) and leads upwards to the actual measuring device. If suitably dimensioned, this disposition of conduits ensures the object of the invention.—*Patentee: H. Hoppe (British Patent No. 699,939). World Shipbuilding, February 1954; Vol. 4, p. 29.*

Multi-engined Propulsion

A proposal for the propulsion of a 17-knots twin-screw cargo and passenger liner has recently been prepared at the request of a shipowner by D. Napier and Son, Ltd., It is proposed to install six Deltic high-speed Diesel engines, of the type which has been designed and developed for the Royal Navy, in a machinery compartment arranged aft. The vessel for which this proposal has been put forward is of 6,000 tons gross and has a displacement of 8,500 tons at 21ft. draught. She will carry 430 tons of fuel oil in the midships and after double-bottom tanks. The arrangement of three engines geared to one shaft takes up very little room and the number of auxiliaries required is reduced. It will be seen from the drawings that the vessel for which the proposed scheme has been put forward has a small baggage or cargo hold arranged forward, but the bulk of the cargo space is arranged in holds amidships. The concentration of weight in this position, balanced by the extremely light weight of the engines, and the fact that the



Machinery arrangement in proposed Deltic-engined liner

1 Deltic engine.
2 Fluid coupling.
3 S.L.M. gearbox.
4 175 kW. generator.
5 Gen. service pump.
6 Fire and bilge pump.
7 Combined L.O. and F.W. coolers.
8 Propeller shaft.
9 Lub. oil tank.
10 Composite boiler.
11 Fuel service tank (auxiliaries).
12 Silencer.
13 Lub. oil tank.
14 Lub. oil purifier.
15 Lub. oil filters.
16 Purifier hot F.W. tank.
17 Aux. air receivers.
18 Emergency Diesel-driven air compressor.
19 Emergency Diesel-driven generator.
20 Main air compressor.
21 Main air receivers.
22 Fuel oil purifiers.
23 Fuel oil heater.
24 Fuel oil service tank.
25 Switchboard.

varying weight of the main fuel storage tanks is also amidships, enables the ship to be trimmed in any condition of loading by cargo distribution and without excessive use of trimming tanks. A cargo capacity of 191,000 cu. ft. is provided in midships holds Nos. 2 and 3. This is approximately two-thirds of the total space available in a conventional ship of this size designed to carry cargo only, and furthermore, 278 passengers can be accommodated. The propelling machinery consists of six Napier Deltic 18-cylinder marine engines with a total output of 10,350 b.h.p. or 9,650 s.h.p. Three engines are geared to each shaft through Vulcan-Sinclair fluid couplings and reverse/reduction gearboxes of the S.L.M. type. The operating speed of the engine is 1,500 r.p.m. at maximum output, and a 10:1 reduction is obtained in the gearbox, giving a maximum propeller shaft speed of 150 r.p.m. Three speed ranges, very economical in terms of fuel consumption per b.h.p. are provided by this system, and correspond roughly to ship's speed of 6 to 8 knots, 10 to 13·5 knots and 15 to 17 knots. Conditions of loading, etc., will, of course, determine the speeds at which maximum overall economy is achieved. When manœuvring, rapid ahead and astern engagement can be carried out on the clutches without draining the fluid couplings or stopping the engines. Starting and manœuvring air is consequently not required, and a considerable reduction in the capacity of the ship's compressed air system can be effected. The weight of the proposed machinery, including engines, gearboxes and couplings, is given as 95 tons, compared with a corresponding figure of 750 tons for an equivalent direct-coupled installation. Of considerable importance in a passenger vessel is the almost complete lack of vibration of the 18-cylinder two-stroke power unit, which fires at every 20 degrees of the crankshaft and is consequently extremely smooth in operation. The triangular arrangement of the cylinders results in almost perfect balance and, in addition, each engine is flexibly mounted so that virtually no vibration is transmitted to the hull.—*The Shipping World, 10th February 1954; Vol. 130, pp. 187-189.*

French Marine Boilers

French marine boilers can be divided into three distinct classes, i.e. specifically French, developed from the Du Temple-Guyot type of boiler; Velox and Lamont types, built under licence; and, finally, certain boilers specially requested from abroad, such as Foster-Wheeler boilers, built at the Chantiers de Penhoet. Penhoet Boilers, Type P41:—These boilers have the following characteristics:—They belong to the asymmetric three-drum type, with natural circulation; the cross-section of the combustion chamber is a parallelogram, with tubes arranged around the walls; most of the tubes terminate above the level of the upper drum; natural circulation is ensured by suitably dimensioned water return tubes. The superheater usually consists of primary and secondary parts. A thermostatically controlled three-way valve in the steam circuit between the two superheater elements makes it possible to divert part of the steam flow to the tubular desuperheater immersed in the side drum of the generating circuit. This arrangement permits regulation of superheat temperature. In other designs, this regulation is effected by a desuperheater of the water-injection type. After leaving the superheater, the hot gases strike the following heat-exchange surfaces:—A bank of generating tubes, an economizer consisting of horizontal spiral elements; and, in certain models, a tubular air preheater of circular contour. The boiler is surmounted by a layer of air, which ensures recovery of heat diffused through the inner casing and which permits cooling of the latter. These boilers are designed to operate at relatively low combustion rates of the order of 3 kg./m.²/hr. (0·6lb. per sq. ft. per hr.) of total heating surface, i.e. evaporator, superheater and economizer. As a result, their efficiency is very high. The Chantiers de Penhoet have constructed P41 boilers for 8,000-tons cargo vessels in great numbers. These boilers have a simplified layout, compatible with economy in initial cost, without attempting to obtain very high efficiency. In view of this, steam conditions were limited to 37 kg./cm.² (525lb. per sq. in.) at 400 deg. C. (752 deg. F.). They have given complete satisfaction in service. The boilers of the passenger vessel *Bretagne* are designed to obtain better characteristics than those mentioned above and for steam conditions of 40 kg./cm.² (570lb. per sq. in.) at 450 deg. C. (842 deg. F.). They, too, have given complete satisfaction since they were put into service. The following table gives the main particulars of the boilers described:—

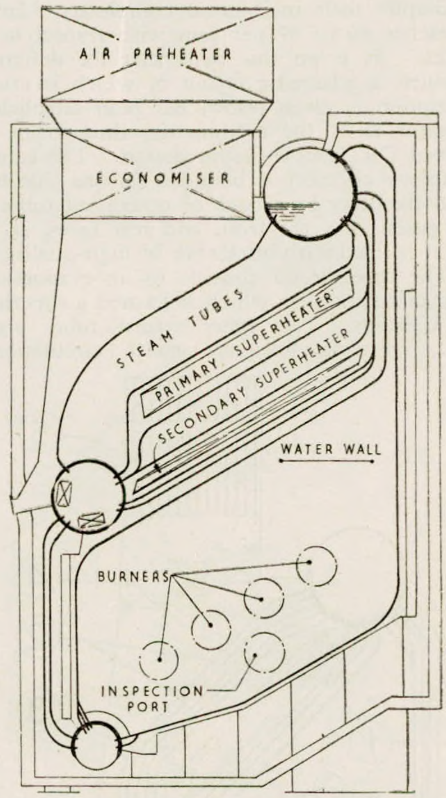
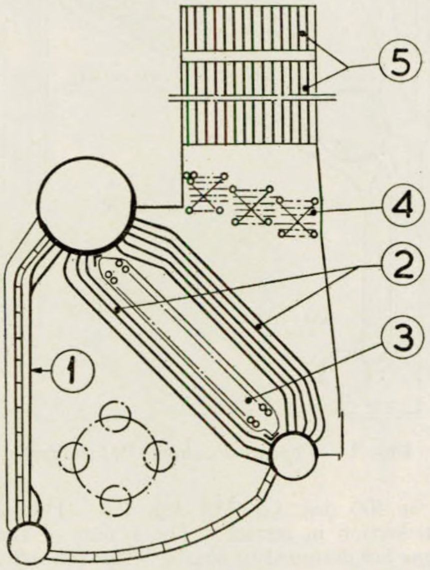


FIG. 1—*Typical Penhoet P41 boiler*

Particulars.	Cargo vessels *Mont Agel, Mont Viso.*	Passenger liner *Bretagne.*
Pressure, kg./cm.² (lb. per sq. in.)	37 (525)	40 (570)
Superheat temperature, deg. C. (deg. F.)	400 (752)	450 (852)
Feedwater temperature, deg. C. (deg. F.)	100 (212)	150 (302)
Evaporation, metric tons per hr. (tons per hr.)	15 (14·8)	25/35 (24·6/34·4)
Efficiency related to a gross calorific value of 10,300 kcal. (40,900 B.Th.U.), per cent	82·7	87·1
Evaporative heating surface, m.² (sq. ft.)...	226 (2430)	375 (4030)
*Total heating surface, m.² (sq. ft.)...	410 (4410)	672 (7230)
Heating surface of air preheater, m.² (sq. ft.)	—	475 (5110)
Heat release, kcal. per m.³ (B.Th.U. per cu. ft.) ...	$3{\cdot}15\times10^4$ (3530)	$4{\cdot}31\times10^4$ (4820)
Heat release, kcal. per kg. (B.Th.U. per lb.)	87 (157)	109 (196)

*Sum of evaporator, superheater, and economizer surfaces.

In general, boilers used by the Merchant Navy are of large dimensions and are not of the high-duty type, from the point of view of combustion rates, thus giving a high degree of efficiency. Where the space available is limited, some shipowners have adopted Velox boilers with pressurized combustion chambers, despite their mechanical complexity. In these the efficiency reaches 86 to 87 per cent with respect to the gross calorific value. It is on this basis that the design of high-efficiency boiler, a schematic layout of which is attached and whose description is given below, has been established. This steam generator is of the asymmetric, three-drum type, and is of improved Du Temple-Guyot design. The comparatively large combustion chamber is bounded on one side by a water wall, and on the other by a bank of generating tubes, traversed by the hot gases. On the front and rear faces, as well as at the bottom, it is lined with brickwork of high-quality refractory material. The tube circuit consists of an evaporator section, with natural circulation, in which is located a superheater with horizontal U-bends. The water return tubes are liberally dimensioned, ensuring adequate water circulation in the



High efficiency boiler project

1. Closely spaced water wall.
2. Steam tubes.
3. Superheater.
4. Economizer.
5. Air heater.

generating tubes, with their consequent protection. Below these tubes is located an economizer with horizontal tubular elements, and a tubular air preheater which, according to the space available can be mounted either vertically or obliquely. This bank of tubes is in two sections, the upper section being constructed of ordinary steel, and the lower of stainless steel and removable, so as to permit replacement in case of tube corrosion. Boilers of a similar type, at present in service, show that notable reliability and ease of operation can be expected from them. The disposition of the tubes, as well as the use of conveniently located flues, permit efficient cleaning. Their weight and space requirements, related to heat release, are very moderate compared with other models, and their efficiency is comparable with that of the Velox boiler.—*G. M. Pommelet, World Shipbuilding, February 1954; Vol. 4, pp. 9-12.*

Protective Propeller Fins

This patent covers a device to prevent damage to ships' propellers by ice or submerged objects which might otherwise foul the propeller blades. In the invention two or more fins are provided on either side of the hull, one above the other, each so dimensioned that at its after end it extends out approximately to the circle described by the tips of the propeller blades. The propeller is thus protected against objects which extend well below the water, for example, ice which may be tilted by the ship. Owing to their shape, the fins push any object which is encountered outwards, so that it is not necessary for their outer edges to extend out as far as the propeller blades. In shape they should preferably follow the stream lines of the hull. In Fig. 2, the stern (1) of a ship carries the propeller (2), and the rudder (3), the latter being omitted from the view from aft. The protective device

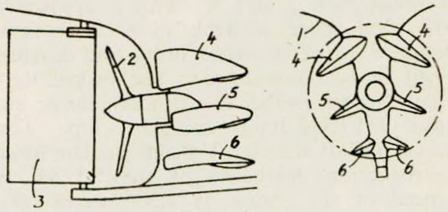


FIG. 2.

comprises the fin-shaped members (4, 5 and 6), which are arranged in pairs on either side of the ship.—*Patentee: N. J. Liaaen (British Patent No. 700,276). World Shipbuilding, February 1954; Vol. 4, p. 29.*

Torsional Vibration of a Geared Turbine Propulsion System

The Victory Ship programme, initiated and carried on during World War II, began on the basis of a great number of ships and involved several shipyards and several manufacturers of propulsion engines. In the interest of production, the steam turbines and reduction gears were required to have common seating arrangements and connexions to ship's piping. Turbines from any of four manufacturers (one had two designs), and gears from any of four manufacturers, were required to be interchangeable in these respects. It was not possible, therefore, to tune the turbine branches to a common frequency as was the general practice when a two-node resonant condition was predicted in the operating speed range at fairly high power. Calculated torsional characteristics, based on generally accepted methods, indicated the possibility of objectionable torsional vibration at the two-node resonant speed. Four of the first ships in the programme were prepared, during manufacture of the reduction gears, for extensive testing on sea trials to determine by measurement the location and severity of the two-node resonance. Results of the tests show that both the one-node and two-node resonant speeds were found quite close to the calculated values and that the vibratory torques at the two-node resonance were only about 16 per cent of the predicted values in spite of a stimulus indicated as greater than used in the original calculation. Evidence from three inter-related considerations indicates that the friction damping coefficient, which is most effective in limiting the two-node type of oscillation, should be about ten times as large as previously used in calculation. These considerations are (1) the amplitude of measured torque variations, (2) the shape of the calculated vibratory torque peaks which were fitted to the test data, and (3) the phase relationships of the vibratory torques recorded for the two turbine branches and the propeller branch of the mass-elastic system. In addition to the analysis of the two-node type of torsional vibration, which was the prime objective of the tests, some interesting observations were made in the region of the one-node resonant speed. These were not considered to be as conclusive as the two-node data for several reasons; there was indisputable evidence of loss of gear tooth contact and the analysis is based on the assumption that this does not occur; at no time did the three branches of the system oscillate in unison at one-node frequencies, and the propeller damping coefficient was indicated to be smaller than usually assumed, possibly because the propeller was barely submerged or breaking the surface during the tests. Higher frequency torque variations recorded during the tests are attributed to the three-node type of system torsional vibration and the vibration of propeller blades as wedge-shaped cantilever beams during conditions of high propeller slip which is effective as a random stimulus. The Victory Ship engines are considered operable at any possible speed without reservation insofar as torsional vibration is concerned.—*Paper by A. D. Sutton. abstracted in Bulletin, The Society of Naval Architects and Marine Engineers (New York), February 1954; Vol. 9, p. 34.*