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Section 3. SHIPBUILDING AND MARINE ENGINEERING

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Vibration of Marine Turbine Blading

The purpose of this paper is to give a general explanation of the vibration phenomena encountered in turbine design. Detailed methods have been omitted. The behaviour of a vibrating cantilever is expressed in terms of the exciting force and damping factor. The sources of damping and excitation are discussed. Impulse excitation is briefly considered and the reasons for its devastating effects are shown. The procedure of tuning low-pressure blading, to avoid resonance at the lower harmonics of the turbine speed, is explained in considerable detail. An apparatus for determining the natural frequency of blading, and another for applying both tensile and alternating bending stresses simultaneously, are described. The direct tensile stress in the test is plotted versus the alternating bending stress which shows the resistance of the blade to any combination of these stresses. The designed stresses for certain turbine blades and the stresses involved in several blade failures have been plotted on this diagram for purposes of comparison.—*Paper by R. W. Nolan, read at the 1949 Annual Meeting of the American Society of Mechanical Engineers. Paper No. 49-A-76.*

Single Gear Tooth Bending Fatigue Test

In this paper which is concerned with production and testing of induction hardening of gears, reference is made to a machine for a single tooth bending-fatigue test. In this test the torque in the gear is resisted by a loading pin acting against one tooth, and the stress is cycled from 0 to maximum. This test was developed in the hope that it would aid in the evaluation of shot-peening and heat treating technique and methods, and would provide a means for testing large gears which would ordinarily require large and expensive fatigue equipment. The test figure was purposely made rigid so that the test would discriminate between metallurgical variables present in the particular gear being tested and not be affected by variables in the surrounding gears and housing. Thus, variations in involute, tip relief, tooth interference and certain other mechanics of gearing should have negligible effect upon the test results. Preliminary data indicate that this new type of test can be used successfully for correlating bending-fatigue strength with shot-peening and heat-treating techniques and methods. The single-tooth test has certain advantages. Tests can be made at different torque loads on the same gear. Both shot-peened and unpeened teeth can be compared on the same gear. The fixturing is relatively simple and both large and small gears can be checked. The test is not delayed by breakdown of component parts, such as in the opposed transmission test. Also, residual stresses can be measured on the same gear which is tested in fatigue.—*H. B. Knowlton and H. F. Kincaid, S.A.E. Transactions, Vol. 4, January 1950, pp. 116-131.*

New Mirrlees Diesel Engine

The largest Diesel engine in the "J" range built by Mirrlees, Bickerton and Day, Ltd., which has now completed tests, is a 16-cylinder Vee engine with two banks of eight cylinders inclined at 45 degrees, and on test it developed 3,000 b.h.p. at 900 r.p.m. It is a high-pressure, turbo-charged engine with an air after-cooler specially designed for marine propulsion and auxiliary duties; industrial stationary duties; rail power traction and for oil field drilling rig applications. This size of engine was developed as the result of experience gained after some 1,600 hours of rigorous tests on an in-line 6-cylinder version of the "J" range. The bore was fixed at 9¾in. and the stroke at 10½in., allowing a speed range from 600 to 800 r.p.m. for in-line engines of six and eight cylinders. In special cases, where the intermittent load factor is predominant, it may be possible to increase the r.p.m. to 900. The speed, however, of the 3, 4, and 5-cylinder in-line engines and that of the 6 and 8-cylinder Vee engines will be limited to 750 r.p.m. The production model of the "JSS 16V" engine will be available in special cases to develop 2,270 b.h.p. at 900 r.p.m. Both bedplate and cylinder housing are of fabricated steel construction. The cylinder liners are of the wet type and chrome hardened. They have a continuous uninterrupted bore and are free to expand downwards through adequate water-tight joints. There are no cutaways for valve heads or connecting rod clearances. The pistons are of light alloy, with the combustion chamber formed in the crown. Each piston carries a large diameter floating gudgeon pin, three compression rings and two scraper rings. The engine is totally enclosed. There are no moving parts outside the covers, and, therefore, no external hand lubrication whatsoever is required.—*The Shipping World Vol. 122, 15th February 1950, p. 193.*

Residual Oil and Pulverized Coal

Investigations into the use of residual oil for gas turbine operation are carried out under the guidance of the Admiralty, in collaboration with the National Gas Turbine Establishment. Bunker fuel oil is undoubtedly the most convenient fuel for gas turbines, and if the problem of the vanadium content can be successfully solved it is likely to be widely used. Work on the use of solid fuel in gas turbines is the responsibility of the Ministry of Fuel and Power. The burning of coal as a fuel will probably be confined chiefly to applications of the gas turbine on land, though it may also be employed at sea. There are three main methods of consuming coal in the gas turbine. Firstly there is the direct internal combustion of coal in pulverized form. Arrangements have been completed with the English Electric Co., Ltd., for the construction of a 2,000 h.p. unit on this principle. This work is associated with research work already in hand at the Fuel Research Association, and it will also be integrated with other research into the internal combustion of coal on an existing 500 h.p. gas turbine built by C. A. Parsons and Co., Ltd., Newcastle. The second method is the external combustion of coal. This can be applied either to the closed or open cycle type of gas turbine. So far, however, the Ministry's plans include only the former, and arrangements have been made for the development of an external combustion system to be used in conjunction with an existing 500 h.p. gas turbine set which was built some time ago by John Brown and Co., Ltd., Clydebank. The third method is two-stage internal combustion. In this, producer gas is made from coal at or above the pressure required by the gas turbine and is then burned. A 2,000 h.p. set is being developed on this basis by the Metropolitan-Vickers Electrical Co., Ltd. From the fuel aspect, it would seem that the chief problem delaying the adoption of the gas turbine is that of the vanadium deposits from low-grade fuel oils. The difficulty is that the vanadium-bearing salts are present in the fuel in a soluble form, and cannot be removed by centrifuging. Up to the present, the only successful method of eliminating them has been by distillation, i.e., the production of gas oil, which increases the cost prohibitively. Active research programmes to investigate the incidence of so-called vanadium corrosion are in progress at the National Gas Turbine Establishment and elsewhere, and the basic factors of the case are becoming clear. It appears that vanadium pentoxide is the chief cause of trouble, and that this salt may be deposited on the surfaces of the ducting and turbine blades wherever the temperature is above its melting point, which is of the order of 650 deg. C. When in the molten form the vanadium pentoxide is believed to act as a "carrier" for oxygen and causes an inter-crystalline breakdown in the material, and this breakdown may take place quite rapidly.—*The Shipping World, Vol. 122 1st February 1950, p. 149.*

Gas Turbine Self-Starter

The first fully automatic "self-starter" for a gas turbine in central-station service has been completed by the Switchgear Division of the General Electric Company. The equipment, which is said to perform, automatically and in sequence, the starting and stopping operations, will be put in service by the Central Maine Power Company at the Farmingdale Station. A 3,500-kW. gas turbine, fuelled by Bunker "C" oil, will be put through its normal starting and stopping sequence by the equipment. The following functions will be automatically performed by the unit in properly timed sequence: (1) Energizes the turning gear to "break away" the unit. (2) Starts the cranking motor to bring the unit up to partial speed and actuate the air compressor. (3) Opens the fuel valve. (4) Ignites the fuel. (5) Allows the unit to accelerate under its own power. (6) Transfers for fuel supply from starting Diesel fuel to Bunker "C" for running. After reaching normal speed the unit is controlled by the operator in the same manner as a steam turbine to synchronize the generator with the bus. The stopping functions are automatically performed in the following sequence: (1) Fuel supply transferred back to Diesel fuel to purge the fuel lines of Bunker "C" oil. (2) Fuel supply is gradually reduced until the flame goes out. (3) Unit coasts to a standstill. While provision is made for annual testing of the individual steps, the starting and stopping sequence is always automatic. Protective features are included to provide emergency shutdown in case of abnormal temperatures, fuel, and air pressure.—*Mechanical Engineering, Vol. 72, February 1950, p. 155.*

Cleaning the Steam Side of a Condenser

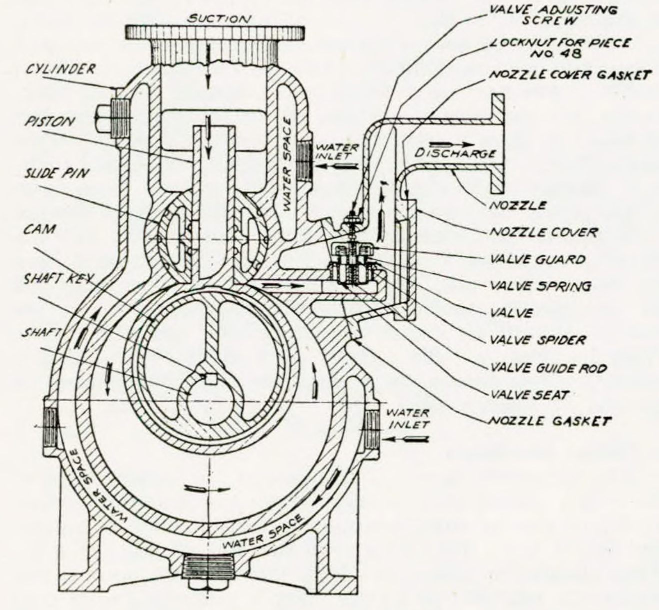
The steam side of surface condensers gradually acquires a coating of grease and dirt. This foreign matter has to be removed or the cooling water will not have a chance to condense the steam unless much more circulating water is used. That means poor heat transfer and poor efficiency. To remove the grease and foreign particles, the steam side of the condenser is boiled out with a strong solution of boiler compound or sal soda. Under normal conditions, boiling should not be necessary more frequently than every 2 or 3 years. Before boiling out, drain the salt-water side. Add 200 gallons of fresh water with 50 pounds of boiler compound or sal soda and 5 gallons of kerosene for each 1,000 gallons of water the condenser holds. That means, that if the condenser holds 2,000 gallons of water, you would add 500 pounds of boiler compound and 50 gallons of kerosene. All this is added to the steam side. Then fill the steam side with fresh water to the top row of tubes. Now cut in the live steam to the boiling-out connection at the bottom of the condenser. Boil the condenser for about 12 hours. Make sure the water boils, and not just makes a crackling sound. Drain the condenser to the bilges. Then flush out with fresh water until all the sludge or sediment is cleared from the bottom of the condenser. Before placing in service, make a thorough check for tube leaks. This boiling out need be done only once in 2 or 3 years, depending on the steam side condition.—*Marine Engineering and Shipping Review, Vol. 55, February 1950, p. 80.*

Dry and Wet Storage of Boilers

If a boiler is to be out of service for a long time and if there will be time for getting the boiler back into service when it is to be steamed again, it is recommended that it be stored dry. When storing dry, the boiler is emptied. Then it is cleaned thoroughly on both the fire and water sides. It is then dried and closed up tight so moisture cannot enter it. Trays of lime are placed in the drums and headers. The lime absorbs any moisture that might get into the boilers or that has been sealed into the boiler. Once the unit is closed, great care should be taken to keep all air, steam, and moisture out. Then the boiler should be opened up at regular intervals and the interior of pressure parts inspected for corrosion and for the condition of the lime. If the lime is deteriorating, it should be renewed. Not only the boiler, but the air heater, economizer, and superheater should be checked and prepared for long-time storage in this same way. If the boiler is to be stored for a short period and must be ready on a standby call, it should be wet stored. In order that boiler water conditions may be stabilized and oxygen bubbles cleared from the boiler water surface, the steaming rate of the boiler should be brought down while it is still cut in on the line. Then build up the alkalinity to a minimum of 20 grains per gallon. Open the air cock and see if the pressure is all out of boiler. When no steam issues from the vent, and before a vacuum starts to form in the boiler from cooling down, fill the boiler completely so the superheater is filled and until water flows from the air vent on the steam drum. Make sure that the make-up water is treated so that it also is 20 grains per gallon minimum alkalinity. It is good practice to de-aerate all make-up water for storing boilers. If there is a drain line to the steam stop check valve, open it so no water gets into the steam line. Fill the boiler until there is at least a 10-pound pressure showing on the steam gauge. Inspect the boiler connections for leaks. Then make frequent boiler water tests for proper alkalinity. If not up to standard, inject a dilute solution of caustic soda with a hand pump. To be on the safe side, make frequent inspections by emptying the boiler at set times to check for possible corrosion. If there is any danger that the engine room will be below the freezing point, the boiler should not be wet stored. During wet storage as well as dry storage, a cover should be placed on the stack to prevent moisture from harming the fire-side of the boiler.—*Marine Engineering and Shipping Review, Vol. 55, February 1950, p. 80.*

Mechanical Vacuum Pumps

In the last few years there has been an increasing interest on the part of power-house operators and designers in the use of mechanical rotary vacuum pumps on steam condensers in place of steam jet air ejectors. This interest has increased to the extent that there are now a number of units in operation, several more under construction and more under consideration. The rotary vacuum pump is capable of air removal over the entire range of pressures, from atmospheric to the lowest pressure desired. The pump normally used in this type of work, as shown in the cross-section, is liquid-sealed and of the duplex type, consisting of a water-jacketed cylinder in which there are two eccentrically mounted rotors or pistons, set 180 deg. apart on a horizontally rotating shaft. The shaft is



Section through pump

mounted on two bronze bearings included in the cylinder heads, and the shaft extends through one of the cylinder heads for the drive which is either through a V-belt from a high-speed motor or by direct connexion to a slow-speed motor. This type of pump requires some means of sealing and lubrication of the various working parts. For this purpose a vertical cylindrical tank is supplied and is attached to the pump discharge. This tank, which is generally called a separator tank, is about one-third full of the sealing and lubricating medium, which is supplied at the rate of a few gallons per minute. It enters the pump through the bearings and under about 15-lb. pressure. Inasmuch as the sealing and lubricating medium is under pressure, it enters the pump cylinder at a constant rate where it lubricates and seals the working parts and passes out through the discharge of the pump and back into the separator tank. In the separator tank there is a series of baffles whose function it is to separate the gases from the liquid. When large volumes of gases are passing through the separator tank, small amounts of the sealing medium are carried along with the discharged gases. In order to save this sealing medium an accessory separator is placed at the far end of the discharge line. This auxiliary separator is called a "whirl type of cyclonic" separator and functions much the same as a mechanical dust collector. At the discharge side of the pump there is a series of poppet valves through which the discharged gases pass. These act as check valves in preventing the back flow of the gases and sealing medium through the pumping cycle. Normally oil is used as sealing liquid.—*R. C. Webster, Combustion, Vol. 21, January 1950, pp. 49-51.*

Development of High-powered Diesel Machinery

In this review of modern trends in the development of high-powered Diesel machinery the author refers to the fact that the single-acting four-stroke engine is now used very little for the propulsion of larger ships, as it cannot compete with two-stroke engines regarding weight, space requirements, and price. Owing to its simple and robust design, this might, however, still be preferred in certain cases, viz., under particularly

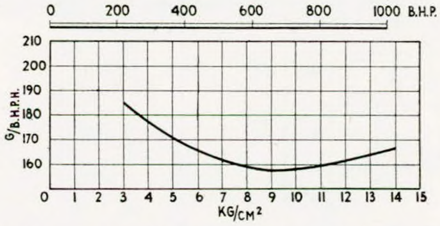


FIG. 1.—*Fuel consumption curve for 4-stroke engine with intercooled supercharge*

difficult service conditions, and with the advent of the high-pressure exhaust-turbo-charger with inter-cooling it is not too certain that the 4-stroke single-acting engine might not again prove competitive to the two-stroke engine. By high-pressure turbo-charging an output per cylinder of about 1,000 b.h.p. could be obtained, and thus without difficulty the output required for the plants in question. Tests with inter-cooled high-pressure supercharge conducted in the research department of the author's firm have shown that it is possible to increase the output of a standard 4-stroke engine, normally developing 420 b.h.p. without any undue heating or wear. Fig. 1 shows the fuel consumption at 500 r.p.m., and it will be noted that the consumption curve is remarkably flat over a large load range, and that the lowest fuel consumption has the favourable value of 159 grams per b.h.p. hour.—*H. Carstensen, Shipbuilding and Shipping Record, Vol. 75, 2nd February 1950, pp. 147-149.*

Experimental Closed-cycle Gas Turbine

The first part of this paper describes operating experiences obtained on the open cycle experimental 500 h.p. gas turbine plant built by John Brown and Co. Ltd. Following the completion of these tests the circuit was closed by removing the

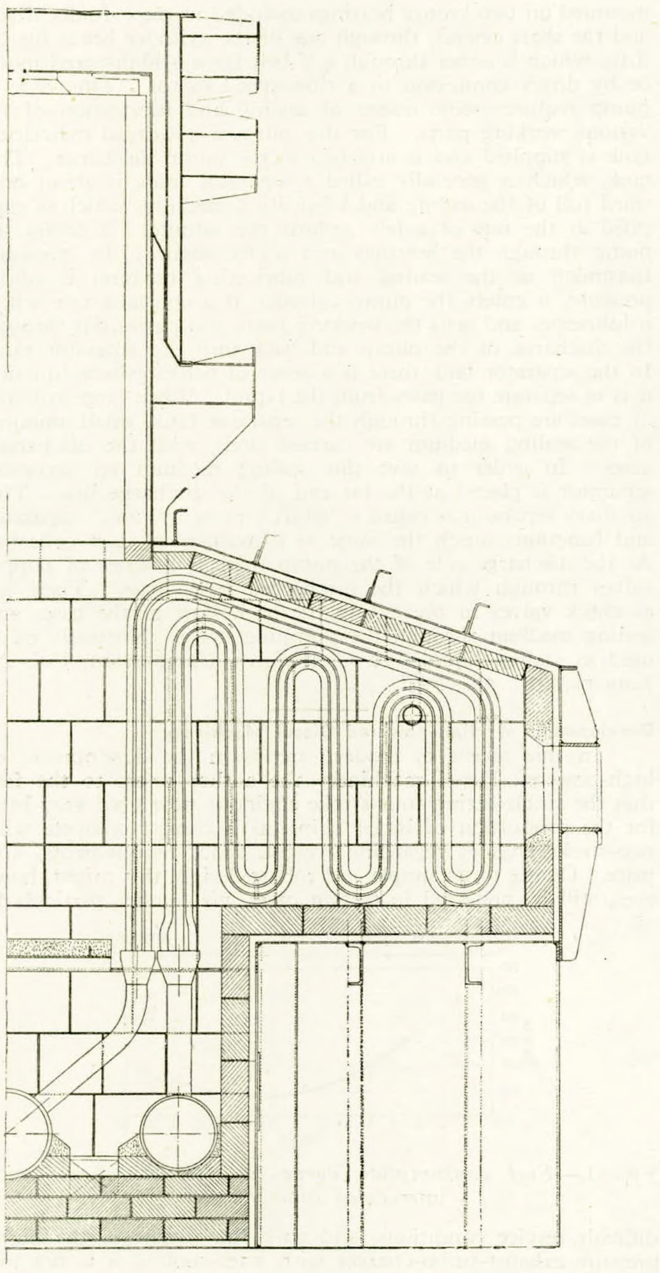


FIG. 20

open-cycle combustion chamber and replacing it with an air heater and by-passing the exhaust from the turbine into a cooler and from thence re-cycling it into the compressor inlet. In the set-up employed that proportion of the exhaust gases from the air heater which is used to warm the combustion air is re-cycled back into the usit, whereas normally the gases used to heat the combustion air would have been passed up the stack, whilst the uncooled remainder would have been re-cycled back into the combustion chamber. This circuit was used because it was not possible to obtain at short notice a re-circulating fan capable of handling at 600 deg. C. The arrangement used is therefore considerably less efficient than that which would have been employed on a plant for industrial use. In order to obtain an exact comparison with the open-cycle the turbine, the compressor and the re-generator were not modified in any way and consequently, as the turbine and compressor had initially been designed for atmospheric pressure on the outlet and inlet ends respectively, it was not possible to raise the pressure level of the closed circuit abov atmospheric on the inlet side. It was not, therefore, possible to incorporate all the advantages of the closed-cycle. Fig. 20 illustrates a half section through the air heater. The unit has been designed for atmospheric combustion and a forced-draught fan is used to take air from the atmosphere and pass it through the combustion air pre-heater into the top inlet flange on the combustion chamber. Inside the combustion chamber the air passes downwards between the outside and the intermediate walls and returns between the inner and intermediat walls to a space at the top of the air heater from which it is introduced into the combustion chamber proper through a burner of Todd design situated on the centre line of the unit. The gases coming from the re-circulating fan enter the combustion chamber via the middle or the lower flange and dampers are provided for controlling the amount of gases entering through these two. The gas which enters through the lower flange is introduced into the combustion space through holes in the brickwork surrounding the inner chamber walls; the gases which enter through the intermediate flange are mixed with the combustion air and pass downwards between the outlet and the intermediate walls and then upwards between the intermediate and the inner walls. On the closed circuit side, air from the re-generator is led into the central cylindrical header where it is divided up into a number of header tubes each of which in turn is divided into 12 heater tubes. The heater tubes are arranged in groups of three and after the initial radiation pass are led to the back of the unit before being wound forward in seven passes. . At their outlet ends the heater tubes are again assembled in groups of 12 into header tubes which lead into the cylindrical outlet headers. The combustion chamber behaved excellently and it was possible to run with over 13 per cent CO_2 and with an amount of re-circulating gas equal to 1·7 times the amount of fresh combustion air. It was very interesting to note that when this re-cycled gas was introduced through the brickwork the flame was quite normal but when it was all mixed with the combustion air prior to combustion the flame became completely transparent and indeed at the first test the observers were under the impression that the flame had been put out by the re-cycled gases and it was only when the temperatures remained steady that they realized that combustion was still carrying on. As gas analysis showed that the CO_2 was still unaltered and indeed the exhaust from the stack was invisible. Temperature readings taken on the brickwork showed practically no variation from the top to the bottom of the chamber, the average figure being just under 1,000 deg. C. when the plant was running with the designed combustion gas temperature of 1,180 deg. C. An examination after the initial run of 70 hours on gas oil showed the burners, flame cone and brickwork to be in excellent condition with absolutely no carbon deposits. After the closed circuit had completed some 80 hours running, the compressor was opened up for examination but was found to be in a very clean state with no deposits on the blading, except for a little fine dust which was found to be mostly ferrous oxide and can be assumed to have come from the new piping used on the circuit. No measurable difference in compressor performance was noted during this running period. A careful examination of the temperatures and pressures in other parts of the circuit showed that there had been no alteration to the turbine, the re-generator or to the cooler, in fact all the circuit conditions after test were exactly as they had been initially.—*Paper by J. B. Bucher, read at a Meeting of The Institution of Engineers and Shipbuilders in Scotland, 24 January 1950.*

Gas Turbine Installation

This invention relates to open-circuit gas turbine installations which involve two or more driving shafts, from which the outputs may be either constant or variable, both as regards speed and power. It is well known that the efficiency of a gas turbine plant is dependent to a large extent on the output; that is to say, the efficiency of a large plant is generally greater than that of a smaller one of the same design. Also, the manufacturing and installation costs of a large plant are normally less than those of two similar plants each of half the size of the large plant. In certain cases, such as in a ship having two screws, obviously two or more driving shafts will be required to drive these screws. It is a desideratum to make provision whereby two or more such driving shafts can be incorporated in a gas turbine installation with duplication or multiplication of the gas turbines, without duplicating the whole plant, and thereby lowering the efficiency of the plant. This invention is exemplified in Fig. 1 in which (41) denotes a low pressure compressor, (42) denotes an intercooler, (43) denotes an intermediate pressure compressor, (44) denotes a second intercooler, and (45) denotes a high pressure compressor. The heat exchanger is shown at (46), and (47) denotes a first combustion chamber, and (47A) a second combustion chamber. A high pressure turbine is shown at (48) and a low pressure turbine at (49), coupled to the propellers (50 and 51), respectively. (49′) denotes an additional turbine coupled to the propeller (51′), working fluid for driving the turbine (49′) being received from the exhaust of the turbine (49). The turbine (49′) exhausts to the heat exchanger

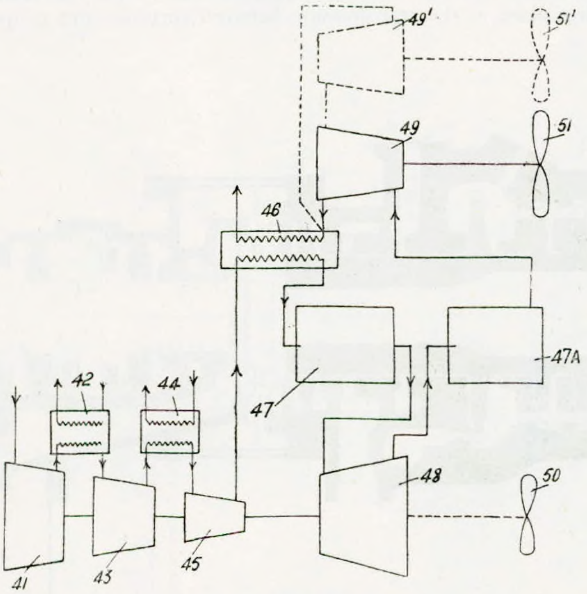


FIG. 1.

(46). In practice atmospheric air is compressed in the low pressure compressor (41) and is then cooled in the intercooler (42). On leaving the intermediate compressor (43), the air passes through the intercooler (44) to the high pressure compressor (45). From the latter the air passes through the heat exchanger (46) to the first combustion chamber (47), where fuel is burnt in the air at constant pressure. The heated air and the mixed products of combustion are partially expanded in the high pressure turbine (48) driving the compressors (41, 43, and 45), and the propeller (50), and are then reheated by the combustion of further fuel in the chamber (47A). The mixed gases are then expanded still further in the low pressure turbine (49) driving the propeller (51) and finally pass through the heat exchanger (46) before being exhausted into the atmosphere. The plant can be arranged to drive two screws, each having the power of the first screw by doubling the output of the air compressor, the coolers, heat exchangers, combustion chambers and high pressure turbine, and by adding a further low pressure power turbine with screw. The last mentioned turbine may be connected in parallel with the first low pressure turbine, in which case the turbine assembly is a series-parallel arrangement.—*British Patent No. 631,863, issued to John Brown and Co., Ltd., and T. A. Crowe. Complete specification accepted 10th November, 1949.—The Shipping World, Vol. 122, 8th February, 1950, p. 174.*

Bilge Pump

An electrically driven bilge pumping unit is illustrated in Fig. 2. The motor is of the totally enclosed type, driving a water pump as well as an air pump, the pump impellers being mounted on a common horizontal shaft (3). The air pump (2) is shown on the left and the water pump (1) on the right. The electric motor (4) drives these through a shaft (5) to which

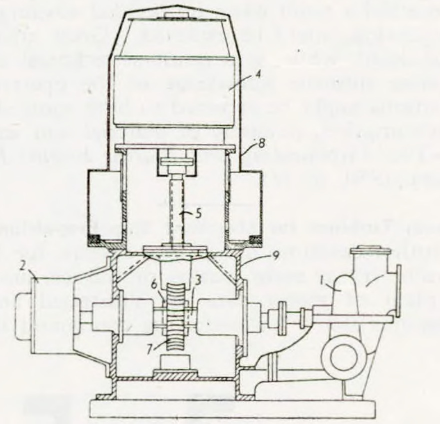


FIG. 2.

is secured a worm (6) meshing with a worm wheel (7). Watertight enclosures (8, 9) enable the pump to work submerged, if necessary. The overall height of the unit is less than normal for this class of pump, thereby providing the required clearance to permit a one-piece casing to be lifted clear of the driving motor.—*British Patent No. 626,272, issued to Drysdale and Co., Ltd., and J. Young.—The Motor Ship, Vol. 30, March 1950, p. 508.*

Hydraulic Drive for Trawl Winches

In collaboration with the Voith Turbine Works, the author has developed a trawl winch drive based upon the principle of the Foettinger transformer. This type of hydraulic coupling has an efficiency in excess of 80 per cent. In principle the Foettinger transformer consists of a centrifugal pump, driven by a Diesel engine in this particular case. The pump is built integral with a turbine driving the trawl winch. The working liquid discharged by the pump is passed to the turbine through stationary guide vanes. There is no mechanical coupling between Diesel engine and winch, the transmission of power being carried out by the liquid. A circular throttling valve is incorporated in the drive, affording speed control of the winch. The hydraulic drive is not reversible and a duplicate drive for reverse operation is provided. The author considers hydraulic couplings of the Vulcan type to be unsuitable for the trawl winch operation because of their torque characteristics.—*F. Süberkrüb, Hansa, Vol. 87, 25th February 1950, pp. 298-300.*

Aims and Achievements of Marine Engineering

In delivering the Watt Anniversary Lecture of the Greenock Philosophical Society, Dr. S. F. Dorey stated that the satisfactory application of the gas turbine to the propulsion of aircraft, did not necessarily point to its early adoption for marine purposes. Those familiar with attempts which had been made in this direction, foresaw only a specialized application in the next few years; and, even then, without any spectacular advantage—or even any advantage at all—over the fuel consumption of a modern oil engine. Nor need there be any anticipation of the early application of atomic energy for power production in the marine sphere. Even with rapid development, many years would elapse before nuclear fuel could be employed to supplement present sources of power. Though such plants might be attractive for naval vessels, so far as the mercantile marine was concerned, atomic energy would have little influence for a long time to come. Considering the near future, it would appear that marine engineering had reached a period of stability. "There are", he said, "a variety of choices open to a shipowner for his propelling machinery, and the merits of each will depend largely upon the services for which ships are required". Looking back, it was easy to choose a period in which progress resulted in a large percentage in saving of fuel—and even weight—without sacrifice of reliability. But the efficient operation of both steam and Diesel machinery had now reached a point when no decided advantage, nor even appreciable saving, could be expected. Great attention would be given to detail; while, as a result of technical and scientific research, more intimate knowledge of the operating and the stress conditions might be expected to have some slight bearing on fuel consumption, economy of material and smoothness of running.—*The Shipbuilding and Marine Engine Builder, Vol. 57, February 1950, p. 115.*

Geared Steam Turbines for Merchant Ship Propulsion

The author examines the general reasons for fitting steam turbine machinery as main marine propulsion units. Various types of plant of recent data are illustrated and described and the reasons stated for preferring compound turbines with below without detriment to the material, vanadium pentoxide becomes a serious problem at higher temperatures, as apart from fouling it destroys the material of which the tubes are formed. Fig. 33 shows a high-pressure turbine which is one of the turbines in a three-turbine arrangement for merchant-ship propulsion of 10,000 s.h.p. Some features are shown in this design which have been developed for gas-turbine work These can naturally be applied, as the working temperature is of the same order. At Pametrada a second-stage superheater to produce steam at temperatures up to 1,200 deg. F. and at a pressure of 1,100lb. per sq. in. gauge has already been installed. A turbine to operate under these conditions is in course of construction. In schemes utilizing steam at 1,200 deg. F. astern turbines would not be employed. By employing a separate astern casing in the l.p. for astern powers up to 55 per cent of the full ahead, and two asterns h.p. and l.p. for higher astern powers, it is felt that steam conditions up to 950 deg. F. can be handled successfully. Above some such temperature other devices such as reversible propellers, hydraulic couplings with or without idler gears would be employed. This will enable the turbine to run unidirectionally; all manœuvring taking place in the transmission between turbines and propeller.

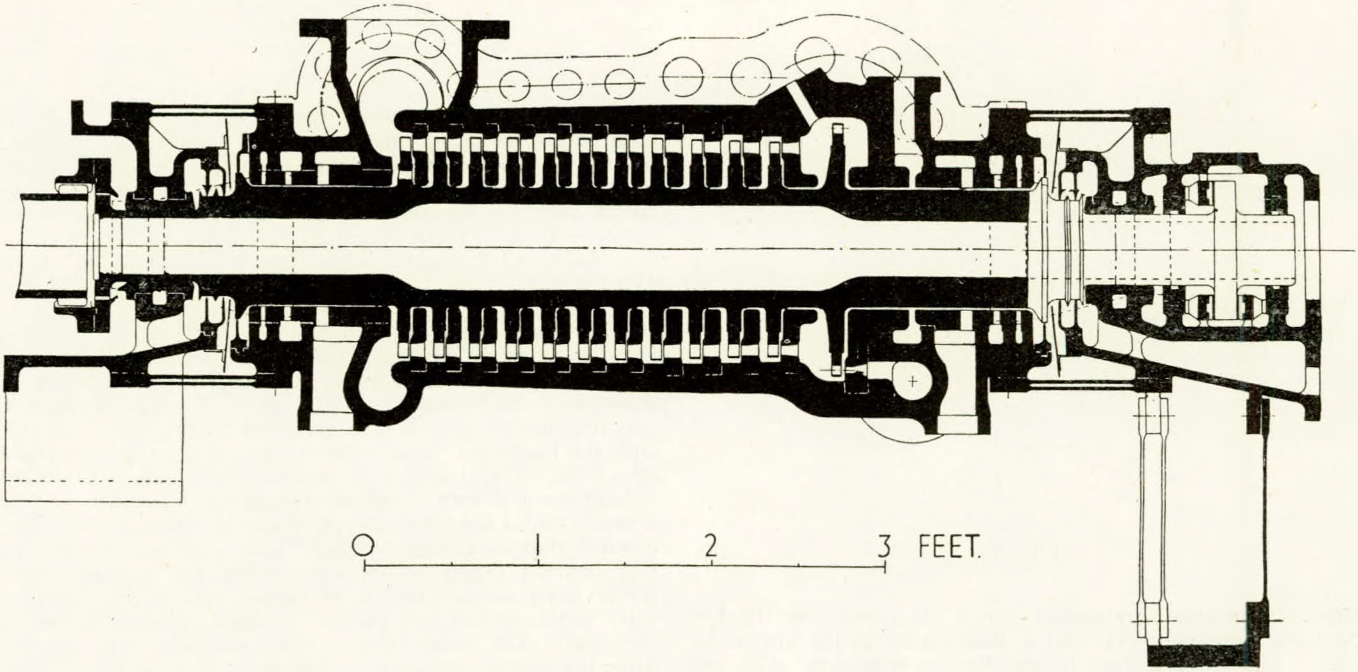


FIG. 33.

articulated double-reduction gearing. References are made to experiments concerning blade and rotor vibration, blade cascade tests, diphragm tests and turbine glands. The main developments to be expected in the future will centre on the use of still higher steam temperatures at the turbine inlet. In an Appendix to the paper, the author examines the evidence on the question of dissociation of steam and the effect of steam on alloy steels at a temperature of 1,200 deg. F. Observing that this temperature involves a tube temperature of about 1,350 deg. F. in the superheater, it represents nearly the limit for 100,000 hours life and reasonable stress values which can be given by materials developed for use in gas-turbine parts. Cooling of any sort at the high-temperature end of the steam cycle, would only reduce efficiency and it is felt that higher temperatures than 1,200 deg. F. cannot be expected at any rate for a long time. There is also a new difficulty to surmount on the furnace side of the superheater tubes as while the gases from crude residues carrying vanadium pentoxide can flow past superheater tubes at temperatures of about 1,000 deg. F. and The elimination of astern turbines, apart from removing manœuvring valves, pipes, etc., has the great design advantage of closing in the l.p. bearing centres and allows the critical speed of the rotor to be greatly raised; the l.p. turbine can thus run faster and handle greater heat drops while operating at a suitable margin below its critical speed.—*Paper by T. W. F. Brown, read at a Meeting of the North-East Coast Institution of Engineers and Shipbuilders, 10th March 1950.*

Gyro-propulsion

In the Gyro-drive developed by the Oerlikon Co. in Switzerland, energy is stored in a heavy flywheel revolving at high speed in a casing filled with hydrogen in order to reduce disk friction. The flywheel is brought up to speed by a three-phase motor of special design. Electrical power generation by withdrawal of kinetic energy from the flywheel is effected by using the afore-mentioned motor as alternator, the current generated being transmitted to the propulsion motor. A flywheel of 1,000 kg. weight revolving with a peripheral speed of

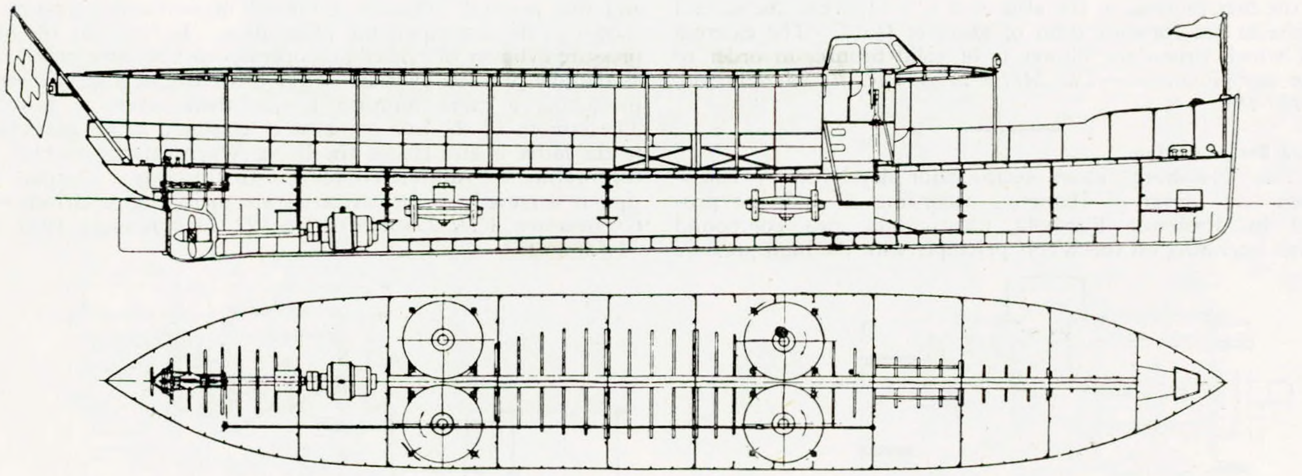


FIG. 4.

100 metres per second, possesses an energy of 500,000 mkg. By raising the speed to 200 metres per second, the energy is increased to 2 million mkg. A proposal to instal a gyro-drive in a ferry is illustrated in the accompanying drawing. The principal data of this vessel are as follows: length overall 26 metres, breadth 4·73 metres, displacement 44 tons, speed 15 to 18 km. per hour. There will be 4 flywheels each storing 6 kW.-hours and one propulsion motor of 100 h.p. at 480 r.p.m. At a speed of 15 km. per hour, the energy stored in the flywheels will be sufficient to propel the vessel over a distance of 12 km. Re-charging of the flywheels at the terminals will require 10 minutes.—*C. Züblin, Hansa, Vol. 87, 11th February 1950, pp. 242-244.*

Atlas Diesel Engine Governor

In Fig. 1 is illustrated a servo-motor converting a control movement initiated by the small force of a centrifugal governor into an operation carried out with the required degree of augmented energy and adapted to the fuel regulation of a marine engine. If it is desired to alter the speed of the engine, the shaft (3) is turned to vary the loading on the spring (9) and the piston valve (8) is displaced. The openings (12) of the piston valve (8) are uncovered and oil flows from the servo-motor cylinder (2) through the upper end of the sleeve (6). The spring (14) consequently moves the piston (13) down, in order to increase the fuel supply to the engine. Simultaneously, the springs (20, 21) are compressed and the flange (18), together with the sleeve (6), is moved downwards until a new position of equilibrium is reached. Since the sleeve (6) is connected to the springs (20), 21) at an intermediate position, a small movement of the piston valve (8), which is not actuated by forces other than those originating from the centrifugal weight (25) and the spring (9), results in a relatively large movement of the piston (13).—*British Patent No. 631,816, issued to A.B. Atlas Diesel, Stockholm.—The Motor Ship, Vol. 30, March 1950, p. 508.*

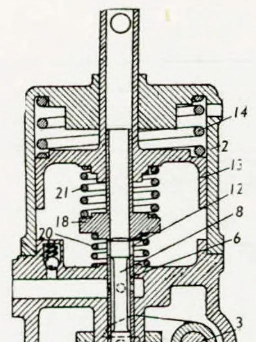


FIG. 1.

First Gray Polar Diesel-engined Ship

The Gray Polar Diesel engined M.S. *Spigerborg,* recently ran trials from the yard of William Gray and Co., Ltd , W. Hartlepool. The main particulars of the vessel are as follows:

Length o.a.	333ft. 2in.
Length b.p.	310 feet
Breadth	46ft. 10in.
Depth to shelter deck	27ft. 10½in.
Draught (summer)	19ft. 0½in.
Corresponding deadweight ...	3,530 tons
Gross register	2,325 tons
Immersion	28·65 tons per in.
Light displacement	2,087 tons
Light draught	8ft. 1in.
Corresponding immersion ...	24·9 tons per in.
Machinery	2,145 b.h.p.
Service speed	12·5 knots

The main engine has seven cylinders with a diameter of 500 mm., or 19·68 inch, the piston stroke being 700 mm., corresponding to 27·56 inch. The engine develops its rated power at 195 r.p.m. and the brake mean effective pressure is 74·4lb. per sq. in. (5·23 kg. per sq. cm.), the piston speed being approximately 900ft. per min. The main engine fuel consumption works out at 0·353lb. per b.h.p. hr. at full load, the daily expenditure for all purposes is expected to be under 11·4 tons. The lubricating oil consumption is 0·8 gr. per b.h.p. hr. There are two blower rotors, one for ahead and the other for astern, with an automatically operated change valve, the maximum air pressure being about 2·6lb. per sq. in. At the after end of the engine there is a train of gears driving the camshaft, as well as an external shaft for the blower drive, running the whole length of the engine and terminating in a second train of gears at the forward end, where the blower rotors are located. The blowers run at fourteen times the speed of the crankshaft, and the first increase at the after end is 4 : 1, while the second increase at the forward train of gears is 3½ : 1. The external shaft which drives the blower is of solid bronze, in order to damp any vibrations.—*The Motor Ship, Vol. 30, March 1950, pp. 480–485.*

Geared Steam Engine

The "Hamburg" steam reciprocator developed by Christiansen and Meyer of Hamburg according to a design proposed by Professor Rembold incorporates two compound engines operating on the Woolf principle with the high pressure and low pressure cylinders arranged diametrically opposite as shown in the accompanying illustration. In this way the high pressure cylinder of one engine operates on the same crankshaft as the low pressure cylinder of the other engine and vice versa, producing a more uniform torque than otherwise possible. The pinions of the two crankshafts engage a single gearwheel; to the latter is also geared the Bauer Wach exhaust turbine incorporating a hydraulic coupling. In this way a compact design is achieved. The auxiliaries are main engine driven.—*R. Christiansen, VDI Zeitschrift, Vol. 92, 11th January 1950, pp. 51-52.*

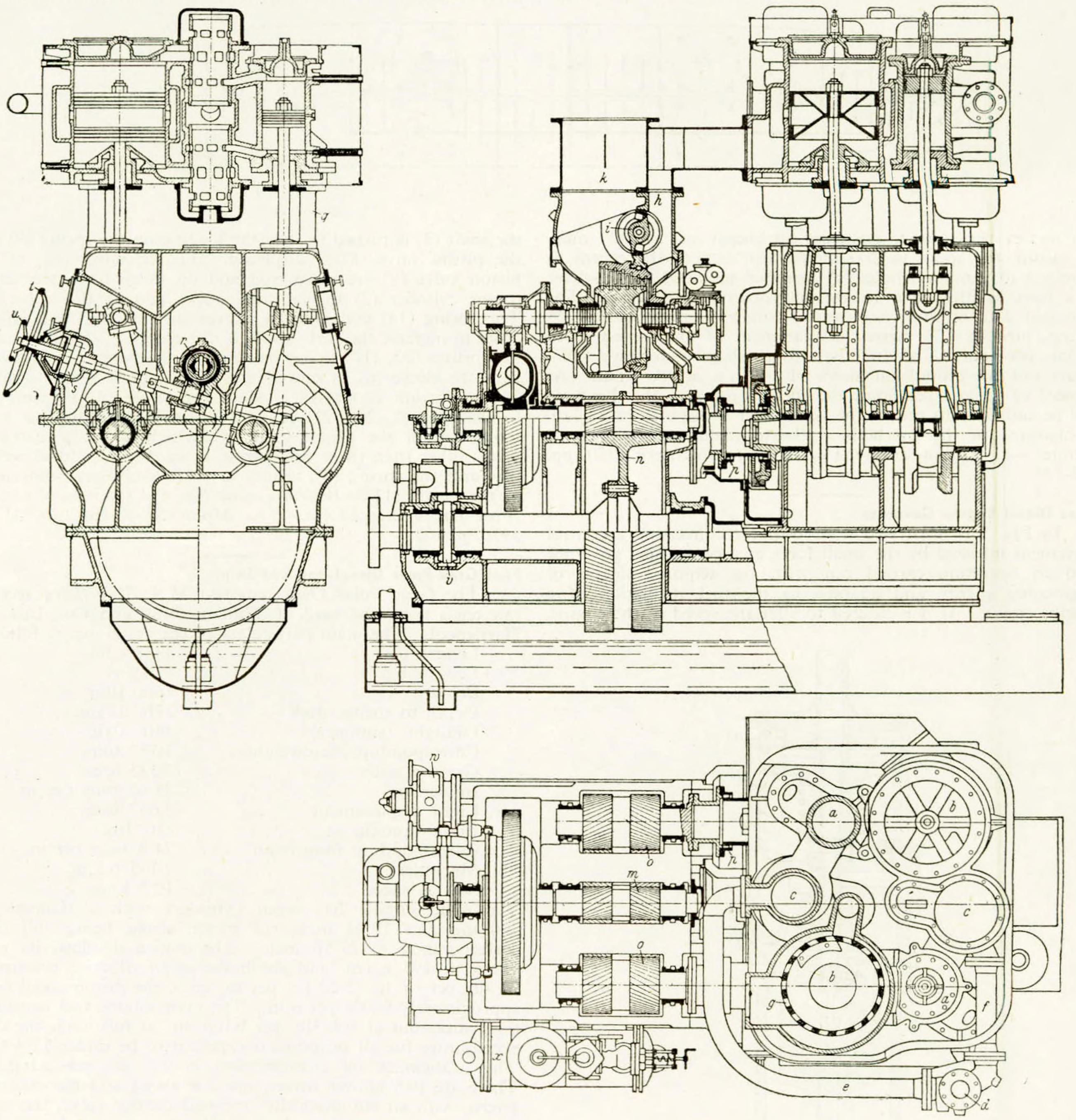
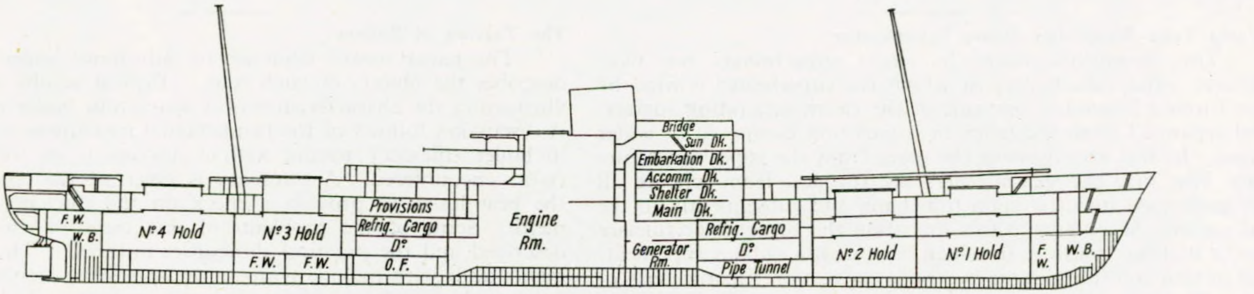


FIG. 1.

(*a*) H.p. cylinder.
(*b*) L.p. cylinder.
(*c*) Valve chest.
(*d*) Main throttle.
(*e*) Steam pipe.
(*f*) Steam pipe.
(*g*) Ports in l.p. cylinder.
(*h*) Exhaust turbine inlet.
(*i*) Change-valve.
(*k*) Exhaust turbine outlet.
(*l*) Hydraulic coupling.
(*m*) Turbine pinion.
(*n*) Gear wheel.
(*o*) Engine pinion.
(*p*) Claw coupling.
(*q*) Columns.
(*r*) Cam shaft.
(*s*) Planetary gearing.
(*t*) Reversing handwheel.
(*u*) Arresting wheel.
(*v*) Thrust bearing.
(*w*) Lubricating oil pump.
(*x*) Oil cooler.
(*y*) Worm gear.

Fast Argentine Motorship

The *Rio dela Plata* is the first of three fast 10,500 ton passenger and cargo motorships built by the Ansaldo S.A. Genoa-Sestri for the Argentine Flota Mercante del Estado and destined for the Genoa-River Plate service. The principal particulars of the vessel are:—

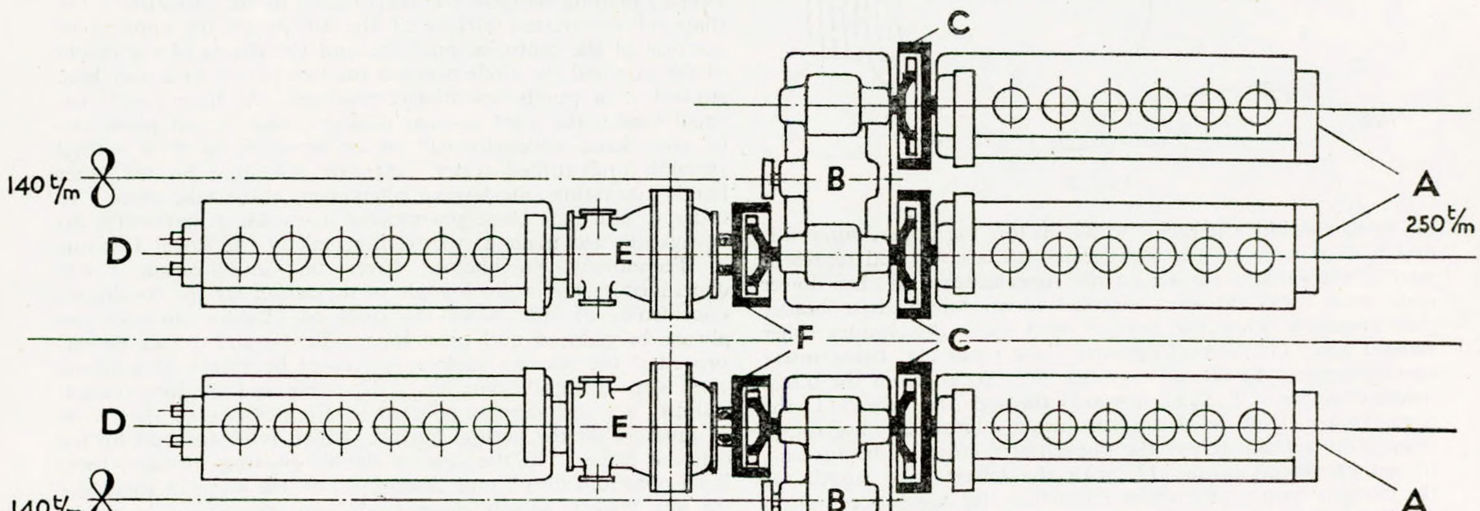


Length overall	547ft. 3in.
Length between perpendiculars ...	510ft. 0in.
Moulded breadth	65ft. 8in.
Moulded depth to shelterdeck ...	39ft. 9in.
Moulded depth to main deck ...	31ft. 1in.
Mean draught at full load ...	26in. 0in.
Gross tonnage	10,500 tons, approx.
Deadweight capacity	8,500 tons, approx.
Capacity, general cargo (bale) ...	269,600 cu. ft.
Insulated cargo capacity ...	80,000 cu. ft.
Speed on trials with 18,400 b.h.p.	20 knots
Cruising speed with 13,000 b.h.p.	18 knots
Fuel consumption at 18 knots for all purposes	68 tons per 24 hours

There are four main cargo holds, each served by one hatch, two of these being forward and two aft of the machinery spaces. The refrigerated holds and 'tween-decks adjacent to the engine room are notable in having cargo lifts instead of the more orthodox handling equipment. The article suggests that this is an apparently novel feature which might be more widely adopted as the actual "parcels" of cargo carried in refrigerated compartments are generally small and light, and the arrangement should lend itself to the adoption of a conveyor belt loading-discharge system from the top of the lift shaft to the cold store ashore. The main propelling machinery consists of two Fiat ten-cylinder two-stroke double-acting Diesel engines, with a bore and stroke of 650 mm. and 960 mm., and the power developed by each engine on works test was 12,500 b.h.p. at 180 r.p.m., a figure very nearly twice the rating of these engines at normal service speed.—*The Marine Engineer and Naval Architect, Vol. 73, February 1950, pp. 77-78.*

Multi-engine Propelled French Cargo Boats

Four 6,000-ton cargo boats each propelled by a multi-engine drive incorporating electro-magnetic couplings are nearing completion at the yards of the Chantiers de la Loire and Chantiers de Penhoet at Saint-Nazaire, each yard building two vessels. As outlined in the accompanying diagram, four reversible Diesel engines (A), each developing 1,550 h.p. at 250 r.p.m. are connected to reduction gears (B) through electro-magnetic couplings of the ASEA type, the propeller speed being 140 r.p.m. There are also two auxiliary generating sets (D) of 1,000 h.p. each at 250 r.p.m. which are rigidly coupled with the generators (E) of 600 kW. each. By means of the electro-magnetic couplings (E) these auxiliary units can be made to supply additional propulsive power whenever required, thus bringing the total propulsive output to 8,300 h.p. The reversible Diesel engines (A) and the non-reversing engines (D) are four-stroke cycle six-cylinder units of M.A.N.-S.G.C.M. design built by the Société Generale de Constructions Mécaniques, Courneuve, and are designated as the GV66 type having a cylinder diameter of 460 mm. and a stroke of 660 mm. The main engines and the auxiliary units are of identical design, the increased power of the main engines being due to the incorporation of Buchi superchargers. The latter were supplied by the Cie Electro-Mécanique, Le Bourget.—*A. Didierjean, Les*



Nouveautés Techniques Maritimes, 1949; Special Issue of Journal de La Marine Marchande; pp. 93-95. (See also Annales Techniques de la Marine Marchande, Vol. 3, No. 29, 1949, pp. 232-241).

Water Tube Boiler and Steam Superheater

This invention relates to steam superheaters for two-furnace water tube boilers in which the superheater is fired by one furnace located at one side of the steam generating surface, and separated from the latter by a partition composed of water tubes. In this arrangement the gases from the superheater furnace flow into the steam generating furnace, from whence all the gases pass into the main tube bank and subsequently to the gas outlet. According to this invention the superheater furnace has in its lower portion, in which it is fired, a radiant superheating section and in its upper portion a convection superheater section. The latter consists of element loops arranged horizontally. In Fig. 2 numerals (1) and (2) indicate the boiler and superheater furnace respectively, and (3) indicates the main evaporating tube-bank, there being a lower water drum (4) and an upper drum (5). The furnace partition consists of closely spaced water tubes (6) connecting the drums (4) and (5). The upper portions of the tubes (6) are more widely spaced in order to permit the flue gases from furnace (2) to enter furnace (1) and then to flow from that furnace over the bank (3) of the water tubes, the furnaces (1) and (2) being fired by the burners (7) and (8) respectively. Saturated steam from the upper drum

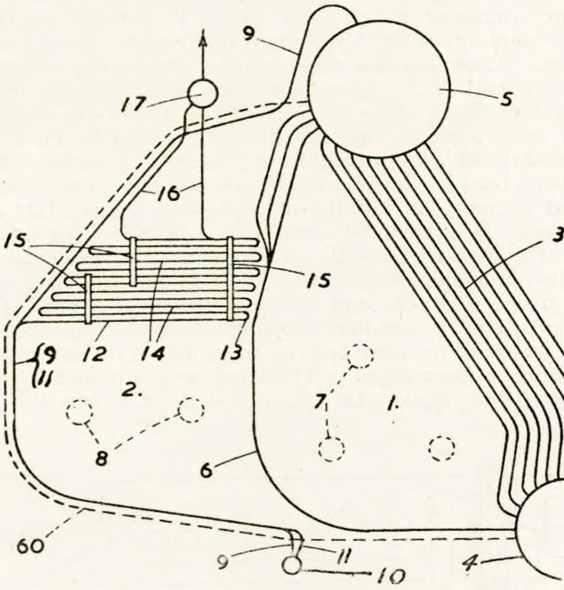


FIG. 2.

(5) flows through the tubes (9) which line the upper, outer side and bottom walls of the superheater furnace (2) and represent part of the radiant surface of the superheater. At their lower ends these tubes (9) are connected to an intermediate header (10) arranged below the surface, and from this header other radiant tubes (11) extend upward, these tubes (11) being interspersed among the downflow tubes (9). Steam from the intermediate header (10) flows upwards through the tubes (11) the upper parts (12) of which are bent into a substantially horizontal plane and extend across the superheater furnace, being connected by return bends (13) with the lowest tube lengths of the element loops (14), which constitute the convection section of the superheater. In this latter section each element comprises a number of horizontal tube loops (14), held together by the bands (15) or by slip spacers. As the convection section is suspended by its own element tubing, its supports are fluid-cooled. In a superheater of the design shown in Fig. 2, all the tubing will drain into the intermediate header (10) with the result that the whole superheater will be self-draining.—*British Patent No. 632,011, issued to The Superheater Co., Ltd., and L. C. Southcott. Complete Specification accepted 15th November 1949.—Engineering and Boiler House Review, Vol. 65, March 1950, pp. 101-102.*

The Testing of Boilers

The paper makes reference to Admiralty boiler tests and describes the objects of such tests. Typical results are shown illustrating the characteristics of a water-tube boiler under test. A discussion follows of the fundamental measurements involved in boiler efficiency testing with a discussion on the probable errors encountered. A summary is given of the estimation of the heat losses to provide a check on the efficiency measurement. Some of the problems of test measurement are then described and the practical difficulties outlined. The measurements dealt with include temperature, pressure, moisture, circulation, velocity, dust emission and gas analysis.—*R. L. Hayden, Transactions, The Institute of Marine Engineers, Vol. 62, February 1950, pp. 85-104; discussion, pp. 104-111.*

Shifting Boards for Ballasted Cargo Vessels

The author states that very little information appears to be available for the guidance of those faced with the problem of designing or fitting shifting boards for retaining solid ballast in ships. On the night of 14th-15th September 1948, the steamer *Leicester,* owned by the Federal Steam Navigation Co., Ltd., of London, on a ballast voyage from Tilbury to New York, and having shifting boards fitted in her 'tween decks where the ballast was carried, encounter a hurricane. She assumed a list of some 40 deg. to port, obviously owing to ballast having shifted, and was abandoned on the following night. After floating for some days, the ship was picked up by salvage tugs and towed to Bermuda, where it was found that the shifting boards had failed, and that a large shift of ballast had occurred. The present paper describes in detail the condition of the vessel on her arrival at Bermuda, and the nature of the failure of the shifting boards. The author attempts an analysis of the causes of failure, and ends by applying the conclusions reached to deriving proposed scantlings and suggesting precautions to be taken in designing and fitting shifting boards. *—Paper by R. A. Beattie, read at a Meeting of the Institution of Naval Architects, 30th March 1950.*

Model Tests on Single-step Planing Surfaces

The authors report that no information on tests made on stepped planing surfaces has been found in the literature. The shape of the wetted surface of the aft plane, the appropriate position of the centre of pressure, and the effects of the height of the step and the angle between the two planes have only been studied in a purely speculative manner. At high speeds (or small loads), the wave motion is slight, and the aft plane can be considered approximately to be separate, as if it moved through undisturbed water. At low velocities, on the other hand, interesting interference phenomena must take place. In order to elucidate these phenomena in detail, a systematic investigation was made at the testing tank of the Royal Institute of Technology, Stockholm. From this investigation it was concluded that at a given angle of incidence, for the conditions considered, in still water the drag of planing surfaces can always be reduced and their longitudinal stability can be improved if the planing surface is divided by means of a transverse step of an adequate size. The drag and the longitudinal stability are considerably affected by the position of the centre of pressure on the bottom surface, which is determined by the external forces. In the case of double planing surfaces, there is an optimum drag range determined by the angle of incidence for any velocity at any given load. An appropriate height of step and a suitable angle between the two planing surfaces are required in order that the planing surfaces should move within this range.—*C. Falkemo and J. Adlercreutz, Transactions, Royal Institute of Technology, Stockholm, No. 24, 1948.*

New Forms for Ships' Sterns

It is well known that a screw works by projecting aft from a ship's stern a column of water, but this column rotates in the same direction as the screw, and the energy needed to make it rotate is lost. In order to recover some of this energy many engineers have placed ahead of the screw a contra-propeller designed to make the water rotate in a direction opposite to that of the rotation of the screw. Up to the present such contra-propellers have been thought of as distributors in a helical turbine, and have been composed of a number of blades. The original idea of this author lies in finding a design which will allow the contra-propeller, and the screw which follows it, to be placed outside the boundary layer which envelops the hull, and outside the streams of turbulent fluid which the hull leaves behind it as it advances. A contra-propeller so placed in a flow having a velocity potential will work in quite a different manner from a turbine distributor, and the theory of its action can easily be derived from Prandtl's vortex theory of aeroplane wings. Thus the contra-propeller can consist of two blades only, so that its resistance to motion can be considerably reduced. In order to obtain this result the author designs the after portion of the hull like a "spoon-back" entirely convex, and without any sternpost, and places under it a "second hull" (which is of the form of a thick aeroplane wing jutting out under the "spoon-back") provided with its own stem, "the second stem", which divides the boundary layer enveloping the ship's hull, so as to throw it to one side and to the other of the second hull. In order to form a contra-propeller with two blades it is only necessary to twist the after sections of this "second hull" into the shape of an S. The screw-shaft runs through this "second hull" and emerges at the centre of its trailing edge at the meeting point of the two blades formed by the two half loops of the S-form mentioned above. The "second hull" also forms an excellent stabilization find at the ship's stern, and gives it good course-keeping qualities.—*Paper by P. Carlotti, read at a Meeting of the Institution of Naval Architects, 30th March 1950.*

Fisheries Research Vessel *Africana II*

Claimed to be the largest fishery research vessel in the world, the *Africana II* conforms in appearance to the latest trawler design with the exception that the port side of the vessel is occupied by various deep-sea sounding apparatus for hydrographic research work. The starboard side is fitted with the orthodox fishing gear. The principal dimensions of the vessel are:

Length overall	205ft. 6in.
Length b.p.	185 feet
Breadth (moulded)	33 feet
Depth (moulded)	17ft. 6in.
Tons gross	882
Tons net	249·37

The vessel is equipped with orthodox fishing gear along the starboard side, with portable fishponds occupying some 250 sq. ft. of deck space adjacent to the forward gallows. A standard heavy capacity "Bear Island" trawl winch, supplied by Robertson of Fleetwood, handles the trawl warps through three Tyne Metal Co. roller bollards, two of which incorporate fathom-recording sheaves and gauges. The trawl winch is capable of dealing with 1,000 fathoms of 2⅜in. F.S.W. rope. At the aft end of the poop deck is mounted a horizontal Clarke, Chapman steam scientific winch which is arranged to work in conjunction with a Mechan scientific davit. The horizontal winch has twin cylinders of 4-inch bore by 10-inch stroke driving a centre barrel and two warp ends through double purchase machine-cut gearing. It exerts a pull of two tons at 80 feet per min. from top layer of rope with a steam pressure of 180/220lb. per sq. in. The barrel, accommodating 1,000 fathoms of 1-inch circumference wire rope, is fitted with a disengaging jaw-clutch and screw band brake, the rope being evenly laid by a hand-operated spooling gear. The engine has piston-type valves with link motion reversing gear, all controls being grouped at the rear of the winch. A Kelvin electrically driven sounding machine is fitted just aft of the scientific winch and handles 360 fathoms of fine 1/16-inch 7-strand wire. On the port side working deck fore and aft are two plankton reels supplied by Ferguson Bros. (Port Glasgow), Ltd. Each reel is driven by an enclosed steam engine. The reels carry 4,790 fathoms of 4 mm. wire and are used to procure samples of water and life at various depths, using special nets, bottles and instruments. A hand-controlled "off and on" guiding arrangement for the wire is provided to ensure proper winding and unwinding on and off the reels. The wire runs out over special depth-recording fair-leads attached to specially designed davits. Incorporated in the forward set is a Lucas sounding machine driven from the opposite end of the engine to the reel. The main propelling machinery was supplied and installed by Aitchison, Blair, Ltd., Clydebank, and consists of a single set of triple expansion engines of 1,300 i.h.p.—*The Shipping World, Vol. 122, March 1950, pp. 289-290.*

Experiments with Models of High Speed Ships

The results are given of some experiments which have been carried out with a group of models of high-speed ships. The same models were used for the experiments described in a previous paper of the author, but they have now been tested at a beam-draught ratio of 2·75. The models have block coefficients varying from 0·535 to 0·605 and have been tested at speed-length ratios from 0·60 to 1·0. In the paper the following questions have been the main objects of the research. (1) How does the longitudinal centre of buoyancy affect the resistance? (2) Which total block coefficient can economically be used? (3) How should the displacement be divided between the fore-body and the after-body, i.e. which δ_F and δ_A should be selected?—*Paper by Professor A. F. Lindblad, read at a Meeting of the Institution of Naval Architects, 30th March 1950.*

American Ship Structure Committee

The Ship Structure Committee established in the United States represents a joint co-operative effort of five agencies—the U.S. Army, Navy, Coast Guard, Maritime Commission and American Bureau of Shipping—to promote stronger and safer welded steel ships through research in design, materials and welding methods. Assisted by co-operation with the Welding Research Council, the National Academy of Sciences, the National Bureau of Standards, the American Iron and Steel Institute and the British Admiralty Ship Welding Committee, it has been possible to reduce the number of fractures occurring annually in welded ships to one-tenth of the number originally experienced; to make progress in the introduction of a ship building steel that is more resistant to the type of fractures experienced than that formerly in use; and to accomplish refinements in detail ship design which lessen the susceptibility of welded ships to fracture. Although the incidence of welded ship fractures has been significantly reduced, the fracture problem has not yet been completely solved. Occasional serious casualties are still taking place. Thus, for instance, the Argentine T-1 Tanker, *Capitan,* broke in two off Cape Hatteras in December 1948. Serious shell and deck fractures continue to occur in various types of welded ships during the winter months. There is a great need for further study of the conditions which affect the resistance of welded ship to fracture. The welded ship problem will not be solved until completely safe, all-welded ships can be mass produced.—*Rear Admiral Ellis Reed-Hill, The Welding Journal, Vol. 29, February 1950, pp. 49-s, 73-s.*

Improvements in Mooring Anchors

The development of more efficient moorings for ships in exposed anchorages depends upon the provision of better mooring anchors. Investigation also shows that with better anchors considerable saving might be made of expensive chain cable required for less exposed anchorages for ships of the Reserve Fleet. Full-scale and model tests were made of the existing standard pick type mooring anchor and other standard designs to establish their performance. Several series of models were then tested and the best ones selected for larger scale test. Complete tests of one of these designs termed A.M.7 verified that it buries rapidly and has a holding pull some four times that of the old design mooring anchor. Anchors of the A.M.7 design are already in service and future requirements will be met by anchors of a similar but further improved type. A comparison is made at each stage between the ship's own anchoring equipment and the appropriate mooring equipment.—*Paper by K. P. Farrell, read at a Meeting of the Institution of Naval Architects, 30th March 1950.*

Thermal Expansion Effects in Composite Ships

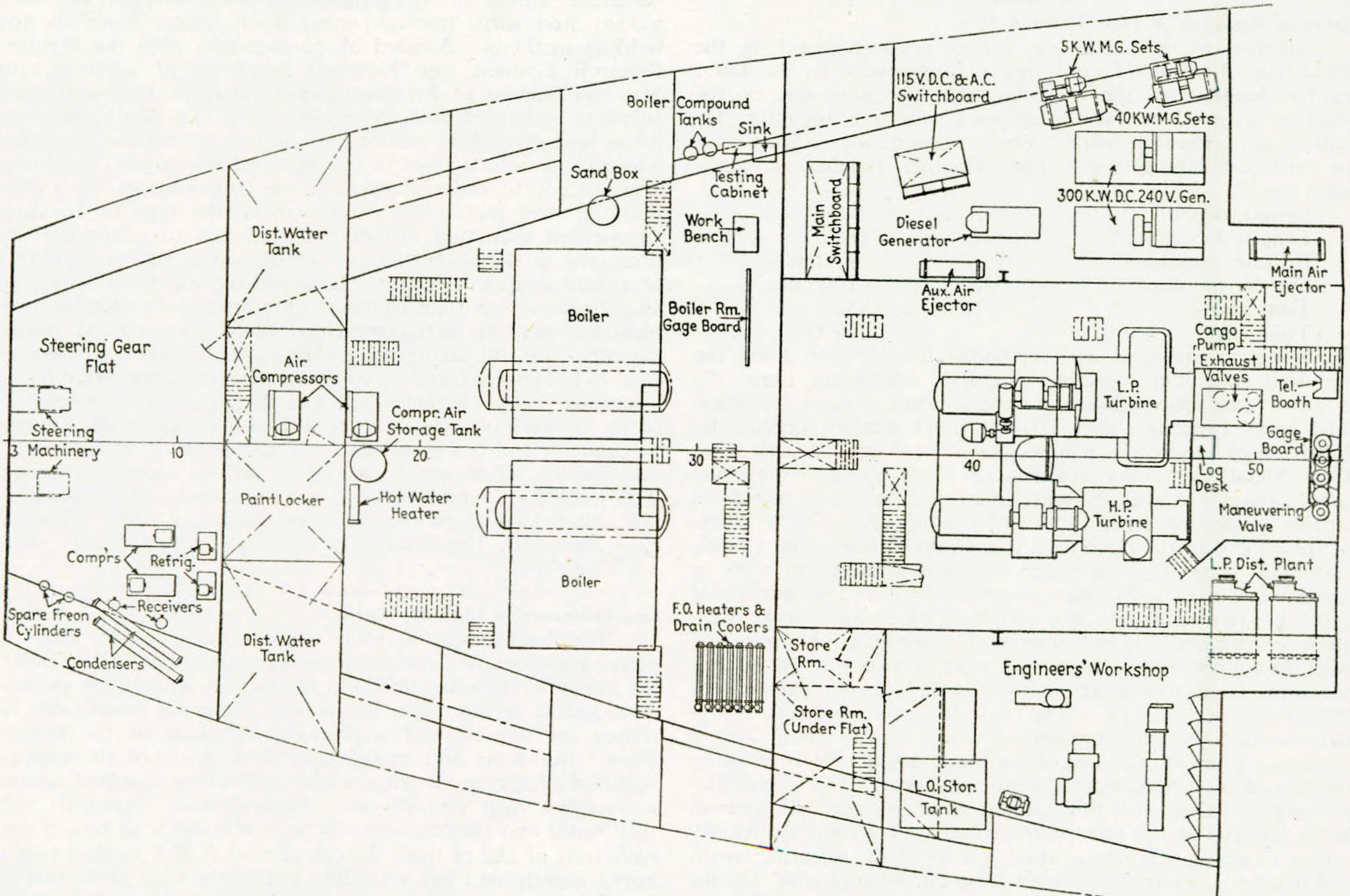
The problem of differential thermal expansion in ships of composite light alloy-steel construction is discussed in this paper, and it is shown that the longitudinal compressive stresses in the decks of such ships under certain climatic conditions are large enough to warrant consideration. A theoretical treatment suitable for drawing office use has been developed and is described. The results given by this method have been checked experimentally in a comprehensive experiment, and the agreement found was sufficiently close to justify confidence in the accuracy of the method. This experiment is described. Experiments were performed on ships to ascertain the nature of the temperature gradients likely to be found in practice and, based on these, suggestions are made as to standard temperature conditions to be used in association with the previously mentioned calculation. It is shown that considerable transverse stresses might be anticipated, but design, based on the quantitative results given by a method described, should nullify these effects of transverse expansion.—*Paper by E. C. B. Corlett, read at a Meeting of the Institution of Naval Architects, 31st March 1950.*

Investigations on Model Anchors

Investigations on model anchors have indicated possible methods of improving designs in order to obtain greater holding pull. The experiments cover anchors for permanent moorings and for ships' bower anchors. The holding pull of the Admiralty plan mooring anchor can be increased to at least four times its present value by minor modifications in shape and fluke angle. The separation between the fluke and shank is very important and there is an optimum value. The holding pull of the Admiralty standard stockless bower anchor and similar designs can be increased three-fold by small changes. The instability which occurs with nearly all designs can be overcome by a suitable arrangement of stabilizing fins and dihedral in fluke section. This superiority of holding pull is maintained at short scope. Breaking out force is shown to be roughly the same as that required for the Admiralty standard stockless anchor for the same holding pull. These conclusions are based on experiments in sand and apply equally well with minor modifications for any material in which the anchors can bury.—*Paper by H. L. Dove, read at a Meeting of the Institution of Naval Architects, 31st March 1950.*

27,000-ton Supertankers

Of seven 27,000-ton tankers ordered in January 1948 by Tankers Navigation Co. from Sun Shipbuilding and Dry Dock Co., Chester, Pa., four units had been delivered before the close of 1949. The leading particulars are: 628 feet length overall; 600 feet between perpendiculars; 82ft. 6in. breadth moulded; summer draft 32ft. 5in.; displacement 35,000 tons; contract speed 16 knots; speed on trial 16·85 knots. Each vessel is driven by a single screw with machinery aft. The design is featured by a cruiser stern, a single continuous steel deck, and a forecastle, bridge and poop erection. The hull is framed longitudinally, except that the peaks, forward deep tanks, double bottoms, after portion of the machinery space, and the forecastle and poop are tranversely framed. The hull is all-welded except for the following: On each side of the ship two deck seams, the deck stringer angle, the lower sheer strake seam, the upper and lower seams of the bilge strake, and the seam of the flat keel to garboard, are riveted from the vicinity of the forward cofferdam to the vicinity of the wing bunker tanks. Angle



connexions of two side keelsons and one deck girder each side are also riveted to the shell and deck, respectively, for about the same length as the seams. The main propelling machinery, built by De Laval Steam Turbine Co., consists of cross-compound steam turbines and double reduction gears, driving the single-screw shaft. It was designed for 12,500 normal shaft horsepower at 112 propeller revolutions per minute at contract draft, and 13,750 maximum continuous shaft horsepower at 125·7 r.p.m., with steam at 585lb. per sq. in. gauge and 790 deg. F. at the high-pressure steam chest and 28·5in. Hg vacuum, for both conditions, while bleeding 10,000lb. of steam per hour from the crossover pipe at about 50lb. per sq. in. abs. Flexible coupling of the enclosed, all-metal, gear type are provided between the turbines and their high-speed pinions. Each vessel has two Babcock and Wilcox boilers, two-drum type, each with an economizer, superheater, and internal desuperheater. (There is no external desuperheater.) They are set side by side on a boiler flat in the after part of the machinery space, with the firing aisles forward, and passage between boilers to the after boiler room. The furnace side wall, roof, and rear wall are water cooled. Each boiler is designed for a normal evaporative rate of 52,500lb. of steam per hour, with a superheater outlet pressure of 600lb. per sq. in. gauge and temperature of 800 deg. F., for a guaranteed efficiency of 87·5 per cent. The temperature of feedwater to economizer is 240 deg. F. The designed overload evaporative rate is 78,750lb. of steam per hour with a steam temperature of 825 deg. F. Inasmuch as the estimated steaming requirement for all purposes at 12,500 s.h.p. for normal sea conditions is 101,992lb. per hour, the boiler capacity has considerable margin.—*Marine Engineering and Shipping Review, Vol. 55, March 1950, pp. 40-50.*

Review of Ship Model Data

In a previous paper read two years ago the authors presented results obtained with single-screw ship models during the war. These results illustrated "optimum" curves of resistance and propulsive efficiency. Since that time there have been two major changes in the method of estimating ship powers at the National Physical Laboratory, Teddington; the first is using the propulsive coefficient at the ship-trial loading instead of at the model self-propulsion loading; the second is using a trip wire to obtain turbulent boundary-layer flow over the model surface. Neither change is completely satisfactory. There is some scale effect on the propeller thrust and torque coefficients and in the wake speed which makes the simple ship and model comparison inaccurate. With the turbulent boundary layer the model residuary resistance can be measured more precisely but the extrapolation to ship total resistance needs further research. These changes remove two of the main inconsistencies of the model experiments and represent a substantial step in ensuring that comparative model results are correct. With this firmer experimental basis, future changes in extrapolation to the ship will be less important. In the meantime this paper is intended to show that design changes and by comparisons of old and new methods to allow adjustment of marginal allowances.—*Paper by A. Emerson and N. A. Witney read at a Meeting of the North-East Coast Institution of Engineers and Shipbuilders, 24th February 1950.*

Testing Resistance to Cavitation Erosion

An article by Y. Bonnard and E. Josso in "Métaux et Corrosion" describes an apparatus for the rapid determination of the cavitation-erosion resistance of metals. The test specimen is tightly screwed into the end of a vertical nickel tube which vibrates in a H.F. alternating magnetic field. The test specimen is immersed (2 mm.) in the test liquid (sea-water, soft-water, etc.), and as a result of the rapidly alternating (7,000-10,000 c./s.) changes in pressure of the liquid film in contact with the lower face of the specimen, rapid cavitation erosion occurs, e.g. the loss in weight of a tin specimen after 30 min. in sea water at 20 deg. C. was 24 mg. Details of experiments with a high-nickel brass, two bronzes containing 12 and 16 per cent tin, respectively, and three steels including an 18 : 8 and a 13 per cent chromium steel, are presented, and it is shown that the most satisfactory criterion for assessing their cavitation-erosion resistance is the hourly loss in weight during the third and fifth hour of testing in soft water (20 deg. C.). The test appears suitable for giving indications with regard to choice of suitable materials and it is pointed out that neither the mechanical properties, microstructure, nor corrosion-resistance bear any direct relationship to the behaviour of the metals under conditions of cavitation erosion.—*Metallurgical Abstracts, Institute of Metals, Vol. 17, February 1950, Part 6, p. 444.*

Circular Cylinder Stresses

Although the theory of thermal stresses in thin tubes is well known, the equations for stress are not readily applicable to most practical problems. This paper is concerned with the elastically restrained cylinder with temperature expressed as a function of the axial length. Relatively simple stress equations may be obtained for the long cylinder with (1) linear thermal gradient, and (2) constant temperature difference between cylinder and end restraint. With the trend toward higher operating temperatures of industrial equipment, it becomes increasingly important to predict thermal stresses. Components of this high-temperature machinery are often cylindrical, as, for example, the hollow shafts and bearing casings of steam and gas turbines. Such parts may frequently be treated as elastically restrained thin cylinders, and thermal gradients approximated by (1) linear gradient or (2) constant temperature difference between cylinder and end restraint. (1) When heat transfer by convection and/or radiation is considerable, the temperature varies as a hyperbolic function of the axial length. A rigorous stress analysis becomes very cumbersome. Generally, however, conduction predominates, and departure from the linear gradient will be fairly small. (2) The case of a constant temperature difference between cylinder and attachment is rarely encountered; but its solution is mathematically identical with that for the shell and restraint at uniform temperature, but with different coefficients of expansion.—*M. Kornhauser, Journal of the American Society of Naval Engineers, Vol. 62, February 1950, pp. 55-62.*

Gear-tooth Stresses at High Speed

If the maximum stress in a gear tooth is less than the fatigue limit for the material, the tooth should not fail even after indefinitely prolonged running. Fatigue data collected in the conventional way suggest that the number of stress cycles required to cause failure of a given material under any particular stress is independent of the time-rate of repetition of stress. It should therefore be permissible to stress any gear, regardless of its speed, up to the fatigue limit for its material, although this suggestion may need modification because of the difference between the impulsive nature of the application of load to a gear, and the more gradual fluctuation of stress in a fatigue-test-specimen of the Wöhler type. Where high-speed gears have failed under stresses apparently lower than the fatigue limit, it becomes necessary to consider whether the actual stress was as low as had been supposed. Errors of pitch and profile in gear teeth may cause actual stresses to be higher than nominal stresses by an amount that increases with speed in any particular installation, up to a limit that would not be exceeded even at infinite speed. The nominal permissible stress (corresponding to the mean transmitted torque) should therefore take account of probable errors in the teeth. High tooth-loads may also be induced by running a geared system in a condition approaching that of resonance with some type of vibration. In general, this danger is more likely in high-speed installations than in others, and it is not always wise to follow the usual practice of ignoring its possibility.—*Paper by W. A. Tuplin, read at a Meeting of The Institution of Mechanical Engineers, 10th March 1950.*

Dry Film Lubrication

Considerable strides have been made in the past 10 years in the technique of lubricating rubbing surfaces by means of dry, slippery films. The original work of Huges and Whittingham carried out during the recent war showing the effect of chemical films on metallic surfaces has been more recently confirmed by work done in the United States. The British investigators showed that the formation of oxide, sulphide and other products on steel rubbing surfaces brings about a reduction of the coefficient of friction. Two types of iron oxide with different effects have been noted by the American workers, one of these oxides on steel facilitating lubrication, while the other contributes to friction. Though the beneficial effect of thin, chemical films on certain metals has been established by friction testing, the exact reasons why lubrication should be improved thereby are not clear. Various explanations have been put forward, which rely upon the fact that oxidation products on a metal surface discourage adhesion of rubbing faces. The crystal structure of these products undoubtedly has a bearing on the way they reduce friction, for such structure forms the hardness and ability, or otherwise, of the oxidation products to cleave when rubbed. Good examples of the two extremes are the sulphide film formed on iron, which facilitates lubrication, and the oxide film formed on aluminium, which is both hard, abrasive and non-lubricating. Recent studies by the National Advisory Committee for Aeronautics of the U.S.A. have confirmed some important facts about graphite films in lubrication. Their experiments showed that for lubricating dry films, the friction coefficient of rubbing faces falls sensibly with increasing speeds. The effect is so marked as to unmistakable and is in line with earlier work done in Britain and elsewhere. They have also shown that there is no appreciable increase in the coefficient of friction for increasing loads up to moderately high values. The fact that friction does not increase with load for drv rubbing surfaces is in accordance with theory. One singular advantage of a dry, slippery film is that there is little difference between static and kinetic friction. This is important where a mechanism must offer a sensitive response to input energy. A high static friction would lower appreciably the sensitivity of the equipment and where the rubbing faces are required to start and stop motion frequently, such a high friction value would prove a serious handicap. Why static friction should be high compared with kinetic friction for oil films is not quite clear but the fact remains that the two values are usually of a much different order of magnitude. This applies to journal bearings, plane rubbing faces and wherever the relative movement of two rubbing faces is involved. It has been known for many years that a thin film of pure graphite between rubbing faces gives a coefficient of friction, which compares favourably with fatty oils under boundary conditions. It is now customary to use a dispersion of "dag" colloidal graphite in alcohol, or other volatile liquid, to form a thin, dry film on rubbing surfaces. The parts to be treated can be brushed with this dispersion, dipped into a bath, or treated by any other convenient method. The film dries rapidly at room temperature and adheres well to most metals. In addition, it can be used on non-metallic faces such as plastics, rubber, glass and metallic faces such as plastics, rubber, glass and almost any solid surface being particularly useful for those materials, upon which oil cannot be used. One particular advantage of a thin, graphite film of this kind is that it resists temperatures up to 600 deg. C.—*Mechanical World, Vol. 127, 3rd February 1950, pp. 119-120.*

Arc Welding in Spanish Naval Shipyards

The development of electric arc welding as applied to ship construction can be considered in three successive phases, which are as follows: (1) The welding of miscellaneous small parts and structures of secondary importance. (2) The welded fabrication of bulkheads, oil fuel bunkers, engine and auxiliary seatings, deck-houses, etc. (3) The welding of the structural parts of primary importance such as shell plating, cellular double bottoms, tank tops, decks, etc., thus forming the all-welded ship. Spanish naval construction at the present time has arrived at the second phase, which is the combined welded and riveted type, and it is difficult to forecast whether the third phase will one day be reached in Spain, since it is not known whether the ship of the future will be all-welded or a combination of welding and riveting. In the Cartegena shipyard of the Empressa Nacional "Bazan" electric arc welding is being used for the construction of warships, both surface vessels and submarines, the first constituting a combination of units details of which cannot be disclosed on account of their secret nature. The following items, however, are welded: the shell and main deck, which form the main part of the ship girder, have their transverse joints welded while in the lower decks the transverse and longitudinal joints are also welded. The stringers and girders which so greatly contribute to the longitudinal strength of the ship have their butts welded and are riveted to the shell and deck longitudinally. The bulkheads, complete with stiffeners, are all-welded with the exception of the boundary angles, which are riveted to the bulkhead and shell. All the principal and auxiliary engine seats are welded, as well as their connections to the frames, for the purpose of greater resistance to vibration and also to improve the watertight qualities of the tanks formed by the walls of the seatings. The superstructures are also welded, together with miscellaneous items such as hatchways, pillars, ladders and all fittings, including watertight collars at bulkheads and decks. The Cartagena shipyard has also on hand the construction of a certain number of submarines with all-welded pressure hulls, in which riveting is used only on a few removable plates for the chamber of the principal motors, and also on the thinner plates of the outer shell.—*A. V. Nuñez, The Welder, Vol. 18, October/December 1949, pp. 81-84.*

Joining of Aluminium Alloys

At the present time riveting is the principal method of joining the aluminium alloys. A ⅝-inch diameter rivet in 5 cent aluminium magnesium alloy is the maximum size which can be closed cold, economically, to form a countersunk point. This can be done by hand hammer, pneumatic tool and hydraulic machine, although larger sizes can be closed by hydraulic means provided that the power is available. Heating of the rivet increases the speed of closing and makes it possible for larger diameters to be used; however, this requires to be done in a thermostatically controlled furnace as the temperature must not exceed 450 deg. C. and the alloys do not change colour with the rise of temperature, so there are no means of gauging the temperatures as is so readily done with steel. Although the fact that certain sizes of aluminium alloy rivets can be driven cold is of some economic benefit, there are arguments for ignoring this and for using hot driven rivets. In the case of riveted steel joints the faying surface rusts up and there is a strong clamping effect of the hot driven steel rivet when cold so that the joint will carry a considerable load before slip takes place. In contrast, the aluminium alloy joint does not rust up and the cold driven rivet has not the same clamping effect, so that there is almost immediate slip under load which will then be taken by the rivets loosening. From these considerations is appears that hot driven aluminium alloys rivets should be used for joints which take repeated stresses. Although aluminimum and its alloys have been welded for many years, the process has not been used in the structural field, owing to a number of causes. One of the main troubles has been the question of the use of flux. This is necessary to protect the weld, but it is equally necessary to ensure that all flux residue is removed from the weld owing to the danger of corrosive attack. This has meant that fillet welding could not be incorporated in the design owing to the possibility of the flux being entrapped, so that only butt welds were permissible. Another consideration is the strength of the weld itself, as the deposit, being of cast structure, is much weaker than the parts joined, although it is possible partly to compensate for this. Then there is the effect of heat upon the metal in the neighbourhood of the weld; in the case of the heat-treated alloys the strength gained through heat-treatment is lost and the strength of the aluminium magnesium alloys is as annealed, and these facts have placed an obvious restriction upon design. Gas welding has been widely accepted for many applications of the non-stressed type, such as ducting, and considerable technical progress has been made in this field. Electric arc welding has not made the same advancement and generally only down-hand welding is possible, although some vertical welds have been made. The speed of welding is high compared with that of steel so that the heat-effected zone is smaller. The advent of the Argonarc gas method of welding has opened the door to the much greater possibility of a wider acceptance of the alloys by shipbuilders, and it certainly holds out hope that the method will solve many of the existing difficulties. Argonarc welding has already been used for ship construction in Canada, and the nine vessels built for the Yangtze River service had aluminium alloy superstructures, and welding was extensively used in their construction, gas welding, spot welding, metal arc and Argon gas welding being employed.—*L. Watkins, The Shipping World, Vol. 122, 1st March 1950, pp. 227-229.*

Flame Cutting

Automatic flame-cutting processes can be divided into three major shape groups; straight lines, circles, and shapes other than the first two. The selection of the type of shape-cutting machine is largely determined by the kinds of shapes to be produced. Any one of the standard types of template tracing cutting machines can produce virtually any shape, but if sufficient volume of work is contemplated, special equipment will be found to have improved performance from the standpoint of speed of handling and accuracy in the finish piece. For example, a circular template will produce circular pieces, but it is significant to note that a circular bevel cannot be cut from a template because of the fixed angle of the torch. Therefore, if a part is to be bevelled, two complete set-ups are required with the accompanying loss of time. If, on the other hand, a machine is built whereon the piece is rotated, square edge cuts or bevel cuts at any desired angle can be produced with equal facility. Much has been said and written about the advantage of stack cutting—the process of cutting a number of pieces in a stack at one time. This method has been abandoned by the firm with which the author is connected in favour of multiple torch cutting whenever high production of identical pieces is required. In the first place, much time is consumed in the preparation of the stack of raw material. The plates or sheets must be in very close contact and hence must be more than usually flat. They must be pressed tightly together and secured by welding or clamping. Gaps between adjacent plates will usually interrupt the flame cut, with the resultant probable spoilage of some pieces and the necessity for reassembly of the stack for a new start. In the second place, it is very difficult to maintain constant dimensions in the finished pieces from the top to the bottom of the stack. If close dimensional tolerances and a high degree of accuracy are required, the author has found that stack cutting produces far too many rejects.—*J. T. Lewis, Jr., The Welding Journal, Vol. 29, February 1950, pp. 116-120.*

Brazing

A recent article published in *Tempil Topics* discusses brazing processes and their applications. Brazing is usually defined as a group of metal joining processes wherein the filler metal is a nonferrous or alloy whose melting point is higher than 1,000 deg. F., but is lower than that of the metals or alloys being joined. Brazing, unlike welding, does not require that the surfaces of base metals be melted; it differs from soldering, frequently termed soft soldering, in that the latter method makes use of filler metals which melt below 700 deg. F. Some confusion has resulted from the long use of the term "hard soldering" to denote brazing in which the silver alloys which melt from about 1,150 to 1,600 deg. F. are used. Brazing depends on the limited diffusion of the filler metal into the clean surface, hot, solid base metal. This is in contrast with fusion welding in which intermelting of the surfaces of the base metal and the filler metal takes place to form an integral joint. The brazing process is widely used because a great many metals can be joined effectively in that manner with a minimum of filler material and at relatively low temperatures compared with welding. Simple carbon steels as well as cast iron, copper, nickel, Monel, Inconel, brass, bronze and other metals are commonly brazed. Dissimilar metals, even with wide variations in melting points may be joined by brazing. Great economies often result from making repairs on broken massive machine parts of cast iron, malleable iron or steel by brazing "in place"; not only is the scrapping of expensive heavy parts thus avoided, but the "down time" of the machine is minimized. Brazing materials commonly used fall into two broad groups: silver base and copper base alloys. The relatively low melting points of the eutectics of the silver-copper and silver-copper-zinc alloys and their excellent flow characteristics allow their use for nearly all metals. The melting points of these alloys range from about 1,150 to 1,600 deg. F. The copper base brazing compositions generally contain a considerable percentage of zinc and may also include nickel, tin and phosphorus. These alloys melt from about 1,600 to 2,100 deg. F., and are generally available as wire, sheet, powder or lumps, to provide the form best suited for every application. Copper and copper alloys are commonly brazed with lower melting bronze filler metal. Nickel alloys are joined by using silver brazing alloys.—*The Welding Journal, Vol. 29, February 1950, pp. 155-156.*

Welding Metallurgy—Peening of Welds

Occasionally welds are peened. If the weld is peened at a white heat the steel will be so soft that the hammer will sink deep into the weld and the grain size will not be changed greatly. If peening is done at a dull red heat (slightly above the transition temperature) the grain size will be refined greatly. Below a dull red heat, peening cold works the metal and, while increasing its strength, impairs the ductility. Cold work distorts the grains and makes the weld subject to cracking. Peening will be most effective if the weld is at a dull red heat. However, after the welder has completed a section of the weld, the weld is hot at the end but relatively cold at the starting point. If the peening is to be done by the operator, he will probably cold work the start of the weld and hot work the hot end of the weld at too high a temperature to secure maximum grain refinement. If he can reheat the weld to the proper temperature or if a helper follows him at the proper distance with a hammer, peening can be done at the correct temperature, if at a considerable expense. If peening is to be effective, the coarse grains in the heat-affected zone also should be refined by hot work (peening). Any attempt by the operator to hot work the heat-affected zone so as to penetrate throughout will reduce the thickness at the heat-affected zone, which might be counteracted by upsetting the plate along the scarves before welding. Upsetting is expensive. Peening, therefore, must be confined to the weld, if it is employed at all. Fortunately, there is a simpler and more effective means of refining the grain size and raising the ductility of steel welds than peening. Instead of making the weld in a single pass, several smaller beads are used to fill the weld groove. The first bead and heat-affected zone is heated not far above the transition (critical) range by the second bead. In this way the grain size of the second bead is refined by the third bead, and so on. The final bead is a reinforcing bead, so that throughout its thickness the grain size of the weld and heat-affected zone is small. Single-layer submerged-arc welds are made in steels and under conditions such that their coarse grain size is offset by other factors. Peening may be used for other purposes than to refine the grain size. Peening spreads the weld and, if correctly applied, will counterbalance the normal contraction due to welding. Peening thus controls distortion, that is, the change in dimensions of the part due to welding. Peening also may be used to hammer down reinforcement. Blowholes, if present, may be sealed up by peening, but it is far wiser to correct the conditions leading to the blowholes than to accept them and attempt to remove them to an uncertain extent by peening.—*O. H. Henry, G. E. Claussen and G. E. Linnert. The Welding Journal, Vol. 29, February 1950, pp. 139-149.*

Buckling of Compressed Steel Member

This report contains a review of the fundamental problems met with in the design of compressed structural members. Buckling of centrally compressed bars is briefly dealt with, and methods are given for calculating the critical load both for buckling within the elastic limit and for buckling in the inelastic region. Bars subjected to the simultaneous action of an axial compressive force and a bending moment are treated at much greater length, and comparisons are made between the critical loads calculated on different assumptions regarding the elastic and plastic properties of materials. The theoretical results are verified by tests made on bars and frames on a model scale. The authors put forward general principles to be used as a basis of new Swedish tentative standard specifications for the design of compressed members. The main point of these specifications lies in the fact that compressed members are designed on the assumption of a definite initial deflection. A special chapters deals with built-up members, particularly batten-plate struts and columns.—*G. Wästlund and S. G. Bergström, Transactions of The Royal Institute of Technology, Stockholm, No. 30, 1949.*

Relative Ductility in Welded Ship-plate

An investigation was made to determine the dependence of zones of low ductility in weldments upon the steel and upon the welding conditions and heat treatment. The ductility was evaluated by means of eccentric notch-bar tension tests conducted at various low temperatures. A zone of low ductility was found in two low-carbon ship-plate steels at a distance of 0·3-0·4 inch from the weld centre line when the weldments were made on ¾-inch thick plate with 100 deg. F. preheat and interpass temperature. A 400 deg. F. preheat and interpass temperature improved the ductility in the critical zone, lowering the transition temperature from 20 to −45 deg. F. An 1,100 deg. F. postheat practically eliminated the zone of lowered ductility, the transition temperature being lowered to −70 deg. F. Temperature measurements made during welding showed that the critical ductility region had not been heated above the lower critical temperature. No change in micro-structure could be noted between the critical zone and the unaffected base plate. Micro-hardness tests showed only slight hardening in the embrittled region. The occurrence of the embrittled region is thought to be due to some subcritical temperature phenomena which may be the super-saturation and precipitation of carbon or carbides from the alpha phase.—*L. J. Klinger and L. J. Ebert, The Welding Journal, Vol. 29, February 1950, pp. 59-s-73-s.*

Effect of Fatigue on Transition Temperature

This report concerns an investigation to determine if prior fatigue would affect the transition temperature curves in impact of ship steels. Tests were made on two shipbuilding steels. A combination fatigue-impact specimen was designed and used. This specimen was round with a circumferential notch and could be tested in impact by sawing off the tapered ends necessary for holding in the fatigue machine. As testing proceeded, it was found that fatigue cracks were developing at the base of the notch both above and below the endurance limit. When specimens were cyclically stressed so as to avoid fatigue cracks, the resulting transition curve showed little deviation from the original curve presumably due to the low stress of the prior fatigue. A series of specimens of one of the two steels tested were prestrained in tension prior to testing in impact with a marked shift of the transition curve resulting.—*J. M. Lessels and H. E. Jacques, The Welding Journal, Vol. 24, February 1950, pp. 74-s-83-s.*

Stresses in Tanker Members

The Hardy-Cross method of moment distribution on frame rings is applied to the transverse frame rings of two modern tankers for several conditions of loading and draught. The variation of moment of inertia and section modulus over the length of the members is taken into account and curves of bending moment and stress distribution are drawn for each member. Positions of peak stress are clearly brought out. The methods used for obtaining the carry-over factor and the stiffness of each member and for calculating the fixed end moments are indicated in Appendices to the paper.—*Paper by H. J. Adams, read at a Meeting of the Institution of Naval Architects, 30th March 1950.*

Buoyancy Characteristics

This article compares the rate of water absorption within a 24-hour period for three low density cellular materials when immersed in water under two different water pressures or heads. The actual buoyancy remaining in these materials up to twenty-four hours under the same conditions is also shown. Curves depicting water absorption and retained buoyancy plotted against time graphically show the results of test. Phenolic foam, cellular hard rubber and balsa wood are the materials tested. Three different densities of phenolic foam (1·9, 2·6 and 8·8lb. per cu. ft.) were used. Cellular hard rubber and balsa wood having average densities of 6·9 and 8·9lb. per cu. ft. respectively were used in the tests. Significant conclusions drawn from the tests are (a) that cellular hard rubber is superior to the other two materials tested in rate of water pick-up and retention of buoyancy, (b) the rate of water pick-up and retention of buoyancy for the phenolic foams and balsa wood are fairly similar, and (c) the buoyancy retention of the lighter density (1·9lb. per cubic foot) is slightly higher than balsa wood and the other two densities of phenolic foam.—*H. J. Stark, J. A. Alpert and T. L. Shoemaker. Journal of the American Society of Naval Engineers, Vol. 62, pp. 139-144.*

Combustion in Vitiated Atmospheres

This series of papers deals with combustion in vitiated or oxygen deficient atmospheres, that is, in air which has lost a portion of its oxygen in a previous combustion process and which is contaminated with the products of combustion. There are three principal fields in the realm of gas turbine combustion which require an understanding of the effect of vitiation, namely, combustion research, aero-gas-turbine practice and industrial practice. It has been a common experimental technique in the study of combustion in hot gas streams to use a primary or "slave" combustion chamber to heat the air. The assumption has been made that when products of combustion are present in the air stream used, they have little effect upon the experimental results. To confirm or deny the validity of this assumption was one of the objects of the work described in this series of papers. The introduction of open cycles using reheat or combustion in stages raises the question whether the combustion process in the later stages is similar to that in pure air. Will flame length, stability and combustion efficiency remain unaltered if the same design of stabilizer is used? The use of "pressure-combustion furnaces" as air heaters in closed cycle plants raises further questions of interest. It has been suggested that exhaust gases should be recirculated to the furnace to dilute the incoming charge and maintain the gas temperature at a level compatible with the heat exchanger design. Will this adversely affect the flame length and stability of combustion? The authors conclude that the effects of vitiation arise chiefly from the decrease in flame temperatures involved and from the increase in the air/fuel ratio required for chemically correct combustion. In general, vitiation increases the time required for evaporation, mixing, heating and combustion of the fuel. In combustion by stages the later combustion chambers require special design. In particular a greater primary air flow and lower mean velocity through the primary zone are required. In an industrial gas turbine flame length may be shortened slightly if the recirculated gases are added before the primary combustion chamber. This involves the danger of non-inflammability if the recirculation is excessive.—*J. Burr and B. P. Mullins, Fuel, Vol. 28, No. 8, 1949, pp. 181-187; J. Barr, Fuel, Vol. 28, No. 9, 1949, pp. 200-205; B. P. Mullins, Fuel, Vol. 28, No. 9, 1949, pp. 205-207; J. Barr and B. P. Mullins, Fuel, No. 10, 1949, pp. 225-231.*