ENGINEERING ABSTRACTS

Section 3. SHIPBUILDING AND MARINE ENGINEERING

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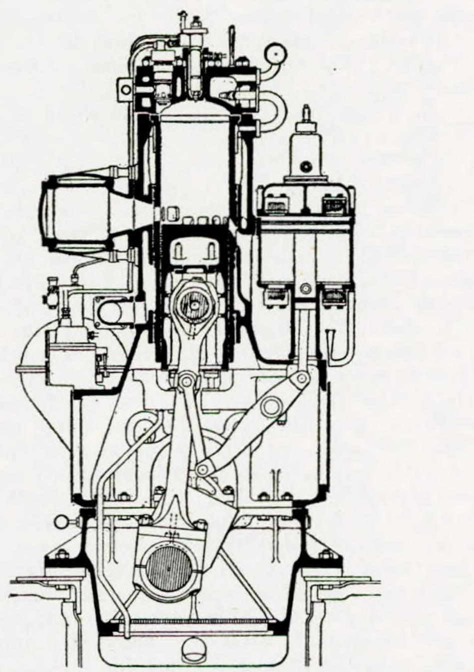
First Clark-Sulzer Engine

This article reports test bed results obtained on the first Clark-Sulzer Diesel engine recently completed at the Southwick, Sunderland, engine works of George Clark (1938) Ltd. This engine is the first of four such engines to be built for installation in four self-trimming colliers of 2,500 tons d.w., being built for Stephenson Clarke, Ltd. The engine is an 8-cylinder reversible, trunk-piston, single-acting unit working on the two-stroke cycle with direct fuel injection. It develops 1,125 b.h.p. at 220 r.p.m., under full power conditions. Each working cylinder has its own scavenge pump driven by a rocking-lever from the main connecting rod. This arrangement with separate scavenge pumps driven from each cylinder gives a uniform delivery of scavenge air, quiet suction and favourable balance of the reciprocating masses, and reduces the length of the engine to a minimum. Each cylinder and pump form an identical unit. In the scavenging system employed, the working piston uncovers the ports shortly before bottom centre and covers them again at the commencement of the compression stroke. Two small air compressors supplying compressed air for starting and reversing, are fitted to two of the scavenge pumps. The engine-driven pumps are built on the forward end of the engine and comprise two reciprocating piston-pumps for cooling water and bilge, and a gear-type oil pump for bearing lubrication and piston-cooling. The controls consist of one combined starting and reversing lever and one lever with fine adjustment for speed regulation. The camshaft is driven through spur wheels which are fitted at the after end of the engine. The starting valves on the cylinder covers are operated pneumatically and the control valves required for this purpose are fitted in a common casing.



Cross-section through the Clark-Sulzer engine

The fuel pumps, one for each cylinder, are incorporated in one block, and situated as close as possible to the crankshaft, thus shortening the drive, which is by gearing from the after end of the crankshaft. The pumps are designed to give variable timing of injection, depending on the engine load.—*The Shipping World, Vol. 121, 23rd November 1949, pp. 556-557.*

Fast Cargo Vessel

The fast single-screw motorship *Wanstead* is the first of three similar ships built by the Caledon Shipbuilding and Engineering Co., Ltd., for the Antwerp-Canada service of Watts, Watts and Co., Ltd. The *Wanstead* is something of a new class of ship altogether, for although designed for a specific trade, her dimensions, speed and cargo equipment could ensure her full employment on alternative services, should economic circumstances so dictate. She is in effect halfway between the 13-knot general cargo vessel and the 17-knot fast cargo liner. The ship has been built to the highest class of Lloyd's Register, "with freeboard", and has the following leading particulars:—

Length overall	468 feet
Length between perpendiculars ...	440 feet
Breadth moulded	64 feet
Depth moulded to shelter deck ...	43ft. 3in.
Depth moulded to second deck ...	27ft. 6in.
Gross tonnage	5,664 tons
Net tonnage	2,745 tons
Deadweight capacity—as open shelter deck ship on 25ft. draught	8,300 tons
Block coefficient	0·66 approx.
Stowage rate	72·5 cu. ft. per ton
Speed in service	15 knots

The *Wanstead* is propelled by a five-cylinder Doxford opposed-piston engine with cylinder bore and combined stroke of 670 mm. and 2,320 mm. respectively; in service it develops 5,500 s.h.p. at 110 r.p.m. This engine is of the in-line scavenge pump type and has been built by Scotts' Shipbuilding and Engineering Co. Ltd., Greenock. A feature of the engine room is that, except in the gangways and at the control position, gratings are used instead of the customary chequer plating. All the pipework beneath, which is painted in distinctive colours, according to duty, can be seen.—*The Marine Engineer and Naval Architect, Vol. 72, December 1949, pp. 555-565.*

Passenger and Cargo Line *Helenus*

The single-screw turbine driven passenger and cargo liner *Helenus*, the first of a new design to be built for Alfred Holt and Co., Liverpool, was recently completed by Harland and Wolff, Ltd., Belfast. The principal dimensions of the vessel are as follows:—

Length overall	about 522ft. 6in.
Length b.p.	„ 485 feet
Breadth moulded	„ 69 feet
Depth moulded	„ 38ft. 6in.
Gross tonnage	„ 10,125 tons

The vessel is arranged with one complete steel deck, a main deck forward and aft of the machinery space and poop, centrecastle, forecastle, promenade and boat decks. There are six main cargo holds, four forward and two aft of machinery space; two of the forward holds (Nos. 3 and 4) are arranged for the carriage of insulated cargo. In addition, No. 5 hold is arranged as a deep tank, and the space between this and the machinery space is divided to form four oil fuel bunkers. Oil fuel side bunkers and settling tanks are also arranged at the fore end of the machinery space, while the fore peak and after peak tanks are arranged for oil fuel or water ballast. The cargo handling arrangements include one heavy derrick for dealing with 50-ton loads, four derricks for 10-ton loads, four for 7-ton loads and sixteen for 5-ton loads. These are served by twenty-four electric cargo winches. The propelling machinery consists of a single-shaft arrangement of triple-expansion, double-reduction geared turbines, made by Harland and Wolff, Ltd. This installation is capable of developing a service power of 14,000 s.h.p. ahead at a propeller speed of 106 r.p.m. Impulse blading is fitted for the h.p. ahead and the h.p. and l.p. astern turbines, while the m.p. and l.p. ahead turbines are of the reaction type. The h.p. and l.p. astern turbines are housed in the m.p. and l.p. ahead casings. The gears are of the interleaved type and the lower portion of the gear-case is incorporated in the ship's structure. The main condenser, of the regenerative type, is slung underneath the l.p. turbine casing and is supported from the tank top by springs. This maintains a vacuum of 28½ inch Hg. under service conditions. Steam, at a pressure of 525lb. per sq. in. gauge and at a temperature of 850 deg. F., is supplied from two oil-fired Foster-Wheeler controlled superheater boilers made by Harland and Wolff, Ltd., complete with air heaters and economizers and the necessary fans for forced and induced draught. The boilers are in the engine room, which has no separating bulkhead; the boiler floor is on a slightly higher level than, and directly accessible from, the turbine manœuvring platform.—*The Shipping World, Vol. 122, 4th January 1950, pp. 14-16.*

New Rudder Design

A new rudder design evolved by Mr. H. A. Wilson, of the Esso Transportation Co., Ltd., has recently been tried out on the Anglo-American Oil Co.'s 18,000-tons twin-screw tanker *D. L. Harper*. This development, entailing an addition to the main rudder of two specially located and specially designed wing rudders, has been produced to improve the steering of several of the company's twin-screw tankers in confined and shallow waterways, where slow speeds are necessary. The steering of twin-screw ships under these conditions has been a long-standing problem. In some cases ships' rudders have to be turned over as much as 20 to 25 degrees, before they enter the propeller race and before the rudders feel the reaction from the propellers. A critical examination of all the circumstances suggested that if something could be devised to use more quickly the benefit of the propeller races, steering at the smaller angles of helm would be improved. It was evident that no great benefit in this direction could be obtained by increasing the length of the rudder, and the experiment was accordingly tried of attaching two small auxiliary rudder blades in the main rudder in a winged-out fashion, so that, with the rudder amidships, the auxiliary blades were just within the propeller races. Subsequently trials disclosed that in the tanker *D. L. Harper*, thus experimentally fitted, the degree of helm necessary to maintain the ship's course at reduced speeds of eight and six knots, even with a strong beam wind blowing, was less than half that previously required. The recent growth of traffic through the Suez Canal has focused attention in general on the problem of steering large twin-screw ships accurately at slow speeds, and the tests which took place during the passages of the *D. L. Harper* through that waterway were of great interest. The master reported that both when the ship was in ballast and when she was laden it was possible to hold the vessel on her course for most of the time with not more than 5 degrees of helm. The steering qualities were described as being not only good but exceptional. Since the successful trials carried out on the *D. L. Harper*, the new design of rudder has been fitted to several Esso tankers.—*The Shipping World, Vol. 121, December 1949, p. 625.*

Survey Tender *Aestus*

The new survey tender *Aestus* was recently completed by W J. Yarwood and Sons, Ltd., for the Mersey Docks and Harbour Board. This ship embodies many requirements such as high speed for her length, extended beam to accommodate the machinery, etc., and limited draught. The hull form and propeller are the outcome of tests in the experimental tank of the National Physical Laboratory at Teddington. The funnel is of Birmabright aluminium alloy. All the main strucfure has been riveted, but welding has been introduced in many of the internal structural parts. The main particulars of this vessel are as follows:—

Length overall	81ft. 3in.
Length, b.p.	75 feet
Breadth mounded	19ft. 6in.
Depth moulded	10 feet
Draught loaded	7 feet

The main propelling machinery comprises two sets of Diesel engines driving a single screw through magnetic couplings and reverse gear. These engines are Paxman Mark 6RPLM six-cylinder, four-stroke units, each having a cylinder bore of 9½ inch and a stroke of 12 inch. Each of these non-reversible engines develops 220 b.h.p. at 590 r.p.m., and as each engine has clockwise rotation, all working parts are interchangeable. The throttle controls are arranged to synchronize engine speeds and torque, and to reduce the engines automatically to idling speed when they are declutched from the propeller. The couplings, by the British Thomson-Houston Co., Ltd., are of the squirrel-cage electric-magnetic type, with one fitted at the after end of each engine. These couplings, either or both of which can be operated by push button, provide for a small percentage of slip when engaging a second engine so that engagement without shock is assured. Should the propeller be fouled, they are flexible enough to protect the machinery from damage. This flexibility, in association with the reverse-reduction gear, practically eliminates engine torsional vibration effects from the gear teeth and propeller shafting. Though not necessarily a feature of these couplings, it has been found possible to start a second engine from the first through the gear box without overloading the equipment. The gear box, by Modern Wheel Drive, Ltd., is of the S.L.M. oil-operated reverse-reduction type, having two input shafts running at engine speed and capable of transmitting continuously the maximum power of both main engines either ahead or astern. The gear ratio is 1·5 : 1.—*The Shipping World, Vol. 121, 30th November 1949, p. 577.*

Oil Tanker Progress

A vast programme of tanker tonnage is proceeding in British shipyards. During the war, this country lost two million deadweight tons of tanker shipping, but by the middle of 1947 these losses had been made good, and by the beginning of 1948 the British tanker fleet had reached a figure of 600,000 deadweight tons more than in the middle of 1939. In January 1949 it was estimated that the world's tanker fleets comprised a total of 1,872 vessels representing a deadweight carrying capacity of 23,815,000 tons with an average speed of 13 knots. At that date there were 440 tankers of 7,453,080 tons deadweight under construction or on order. Of this total, 38·3 per cent were under construction in the United Kingdom, 28·1 per cent in the United States, and 17·8 per cent in Sweden; 20 per cent of these new ships will have a speed of 16 knots or more; 48·2 per cent are of 17,000 tons deadweight and over; 11·2 per cent are of 28,000 tons and over, and 6·7 per cent are of 30,000 tons and over. From these statistics can be seen the trend towards increased size and speed which is taking place, as well as the importance of the British contribution to world tanker construction.—*R. Hammond, Petroleum, Vol. 12, November 1949, pp. 277-280.*

Design of Hotel Services for Passenger Ships

This paper considers in detail some of the problems involved in the design of hotel services for passenger ships. The engineering required may be considered as two separate, but related, problems. The first is the design of the plant and equipment which supply the hotel services, such as electricity, air conditioning, heat, and fresh water. The second is the problem of distributing these hotel services throughout the accommodations. Distribution must be accomplished without compromising ceiling heights; and the piping, ducts, and wiring must be located so as to be readily accessible for maintenance without adversely affecting appearance. The installation costs are important and good design can do much to reduce these to a minimum. The addition of air conditioning, the improvements in interior treatment, and government regulations concerning watertight subdivision and fire zoning increase the difficulties involved in distributing the hotel services. The point is made that the fundamental characteristics of the various systems of distribution should be established in the early stages of the design of the ship so that they may be integrated effectively with the subdivision, the accommodation, and the design of the machinery. The importance of early settlement of these features becomes greater with increase in size of ship, total complement, and number of decks. In the larger vessels this must be extended to include adequate studies of the principal leads for distribution of air, water, steam, and electricity to obtain the greatest simplicity, economy, sightliness, and ease of maintenance. In the larger vessel, having several passenger decks, the use of horizontal transverse zones and vertical trunks, properly integrated with the arrangements of quarters and service spaces, will afford great advantages in simplicity of ducts, piping, and cables, with improved facility for construction and maintenance and better appearance.—*Paper by E. P. Worthen and W. H. Muller, read at the Annual Meeting of the Society of Naval Architects and Marine Engineers, 10th-11th November 1949.*

Cargo Distribution in Tank Vessels

Regarding longitudinal strength, the bulk oil carrier is designed on certain broad assumptions with respect to distribution of light ship weight, cargo distribution, cargo density, and sea conditions. It has appeared perhaps generally impractical to investigate the stress conditions associated with the innumerable loading and weather conditions which may occur in service. The decisions as to loading distribution have been considered largely operational problems, and it has been assumed that the operator would control the distribution of cargo so as to operate under favourable stress conditions. Little guidance has been available to assist in favourable selections of cargo distribution. Without such guidance, it is extremely difficult to judge the effect of a chosen cargo distribution on hogging and sagging stresses in a seaway, bearing in mind that the weight of cargo carried is normally from three to four times the light ship weight. The problem is much more complicated when heavy density cargoes are carried, as some tanks are necessarily empty, and sound judgment is essential in selecting the locations of slack or empty tanks. With the recent marked increase in dimensions of oil tankers, particularly in length, cargo distribution becomes an even more important consideration from a longitudinal stress point of view. The authors demonstrate the following major principles: (1) The basic design stress increases with length of ship. (2) The basic design stress may be considerably exceeded in operation, in some cases by as much as 50 per cent. This excess can and should be avoided by suitable cargo distribution. (3) The carriage of heavy cargoes increases the risk of high operating stresses. This risk is accentuated in the larger vessels. (4) With favourable cargo distribution, heavy cargoes permit operation with equal hog and sag stresses. These stresses may be as much as 30 per cent below the basic design stress. (5) Operating stresses in the ballast conditions may exceed those in some fully loaded conditions. This can be obviated by advantageous distribution of the ballast. (6) Stresses at intermediate drafts in still water may exceed the most favourable wave stresses when fully loaded. This can be avoided by control of distribution and sequence of loading and unloading tanks. The paper suggests the most favourable cargo distribution from the aspect of longitudinal stress for several representative modern tankers, so that cargo distribution patterns may be established for similar types of tankers. The effect of varying cargo density over a wide range is illustrated.—*Paper by J. H. McDonald and D. F. MacNaught, read at the Annual Meeting of the Society of Naval Architects and Marine Engineers, 10th November 1949.*

Shipboard Lubrication Problems

The authors relate from their own experiences and from records of the U.S. Navy Department instances of machinery troubles on naval vessels which were, or appeared to be, lubrication failures. Some of the difficulties arose from the use of the wrong lubricant or its improper application; some were from improper bearing design, problems of metallurgy, or a combination of these and other factors. A few cases show that some limiting quality of the lubricant was not fully evaluated in the design of the unit. However, the majority of the difficulties were due to disregard of some simple principle of lubrication. Since lubrication depends on designs as well as on materials, a change of lubricant alone was seldom sufficient to ensure trouble-free operation.—*Paper by W. D. Leggett, Jr., G. L. Neely and J. B. Ritch read at the Annual Meeting of The Society of Naval Architects and Marine Engineers on 10th-11th November 1949.*

Gate Valve Motor

A new fluid-motor operator for gate valves which has been developed in the United States is said to make the wider use of motor-operated valves possible, since the actuating unit can be adapted easily to standard valves in stock or to valves already installed. The new device can be operated by water, oil, compressed air or gases, and consists of a motor, gear box, and yoke, and is bolted to the valve bonnet. The motor operates the stem through the gear box, deriving its power from five flexible diaphragms mounted radially about an eccentric on the drive shaft. Operating fluid is admitted in rotational sequence to the five diaphragms which actuate pistons that transmit the thrust to the eccentric through a roller bearing mounted on the shaft. The motor is built for valve sizes from 4 to 30 inch.—*Petroleum, Vol. 12, November 1949, p. 293.*

Off-design Performance of Turbo-compressor Stage

Turbine or compressor stages are often designed by comparison with a mathematical model with a perfect fluid and a particular flow quantity. Empirical correction factors are required for the various sources of discrepancy between the mathematical model and an actual stage, such as wall and blade friction losses and tip clearance losses, but such models are of considerable use as a basis of comparisons. In multi-stage compressors or turbines that are required to operate over a wide range of speeds and flow quantities, the individual stages are required to operate over a wide range of velocity ratios. So far no satisfactory model has been proposed for the effect of variation of flow quantity from the design conditions. This paper proposes a numerical method and considers by way of example the simple problem of an isolated stage working in a long, parallel annulus with a perfect incompressible fluid.—*Paper by W. Merchant, submitted to The Institution of Mechanical Engineers for written discussion, 1949.*

Gaskets and Bolted Joints

This paper consists of a study of the loading requirements of gaskets in bolted joints, with the object of developing a rational basis for design of such joints. Starting with an analysis of gasket conditions for tightness, the gasket factor *m* is defined, and its variation with initial gasket stress and gasket width is predicted. These trends are confirmed by a survey of the available-literature data. In a bolted joint, gasket stress becomes a function of the elastic constants of the system. Equations are derived to predict gasket and bolt stresses resulting from the application of internal fluid pressure, and typical elastic recovery curves for an asbestos gasket are presented. Consideration is given to the effect of gasket creep in a bolted joint, and to the problem of distribution of bolt load, for which an approximate theory is derived. On this basis a tentative new design procedure is proposed. Finally, a summary of data which should be obtained for use with the new design procedure is given.—*Paper by I. Roberts, read at the 1949 A.S.M.E. Annual Meeting, Paper 49-A-2.*

Repair of Iron Castings

The author describes a recently developed method for repairing cast-iron sections without any preheating other than simply warming to remove the chill before commencing the actual welding. The type of joint concerned was designed to meet a demand for quick and effective repairs to castings that usually cannot be welded without dismantling, complete preheating, and consequent slow cooling. Even then it often happens that although every precaution has been taken, the finished casting either becomes badly distorted, or it may fracture on cooling due to the unequal stresses in the main body of the casting. The improved method consists in using mild steel tubing to fit inside the joint or to be placed alongside the fracture. The method has been adopted for the repair of heavy flywheels, pulley wheels, castings with offset spokes and two machined faces as well as for cracked Diesel engine cylinder blocks. In each instance there was an abrupt change in section. Bronze arc welding electrodes (using d.c.) are to be recommended although good quality cast-iron electrodes may also be employed.—*G. G. Musted, Welding, Vol. 17, December 1949, pp. 539-543.*

Flame Straightening and Forming Structural Steel

Straightening complicated pieces and assemblies with the employment of local heating by the oxy-acetylene flame and the corollary process of forming or cambering steel parts by local heating, save time and cost and accomplish results that could not be attained otherwise without tremendously large and costly machinery. An early and still important use of the oxy-acetylene flame in structural work is for heating steel to visible red heat so it can be straightened with hammer blows. A more recent use of the oxy-actetylene flame has been for locally heating steel to hot bending temperatures preparatory to bending in a press or by other mechanical means. This application, which is possible where there is a plentiful supply of low cost acetylene, pits the rapid heating of the oxy-acetylene flame against the slower, more soaking heat of less expensive fuels burned in torches or in open top, slot-type furnaces. Heating time is reduced and sharper bending and more accurate location of bends can be attained. Heating time is reduced directly by the rapid heating rate of the intense oxy-acetylene flame, but there is further saving of time and fuel because a short heating time minimizes the loss of heat into contiguous metal that it is not necessary to heat. Sharper bending and more accurate location of bends result from the ability to heat only the metal which must be deformed in bending. It is very important, however, in this use of the oxy-acetylene flame to heat sufficient metal to permit bending without excessive reduction in the thickness and width of the piece being bent. For unsymmetrical sections this demands heating of triangular- or trapezoidal-shaped areas. Another very important precaution for this method of heating for bending or for any other scheme of local heating is to avoid heating to temperatures of 400 to 700 deg. F. any steel that has been severely cold-worked, as by cold-forming, punching or shearing. Inattention to this precaution can allow the development of strain-ageing and consequent brittle fracture of the steel. An example of this can be cited from the records of a large fabricating shop in the fracture of several 8×8×1⅛-inch flange angles for girders. The angles were punched and then curved by cold-forming before they were torch-heated for making a sharp angular bend. They broke in brittle manner through punched holes about a foot from the centre of the bend while they were being fitted and assembled with the web and other parts of the girders.—*F. H. Dill, The Welding Journal, New York, Vol. 28, November 1949, pp. 1,067-1,069.*

Cooling Rates in Arc Welds

As a result of quantitative temperature measurements in the neighbourhood of arc welds made in ½-inch steel plate, a series of differential equations was developed. These equations express the temperature distribution adjacent to an arc weld as a function of time, distance from weld centre line and welding variables. Graphical solutions have been included which permit calculation of the heating and cooling cycles experienced by any point in the vicinity of an arc weld in ½-inch steel plate. These solutions provide the basis of subsequent reports involving the design and application of a time-temperature control device for exact duplication of weld heat-affected structures.—*E. F. Nippes, L. L. Merrill and W. F. Savage, The Welding Journal, New York, Vol. 28, November 1949, pp. 556-s-564-s.*

Percussion Welding—Iron and Steel

In summarizing the various welding processes in use or in process of development, the authors make brief mention of percussion or percussive welding, which has been used only to a limited extent. The equipment is difficult to build and operate, and the process is confined to butt-type joints of limited size. Nevertheless, the process is unique and of considerable interest because butt welds can be made with incredible speed, in almost any combination of dissimilar materials, and without the expulsion of a fin or flash around the joint. The pieces to be joined are held a short distance apart in clamping dies which carry current and apply pressure. The ends to be welded are carefully prepared for accurate mating. An extremely heavy electric current is delivered (electrical energy delivered by discharging condensers) to the pieces as a very short impulse, perhaps for only one-thousandth of a second, and flows across the gap between the pieces as an arc. The heat of this high-energy arc produces superficial melting over the entire end surfaces of the bars. An instant after the arc has struck, the pieces are brought together with an impact blow to complete the weld. The weld fusion zone and base metal heat-affected zone together comprise only a narrow band a few thousandths of an inch wide.—*O. H. Henry, G. E. Claussen, and G. E. Linnert, The Welding Journal, New York, Vol. 28, November 1949, pp. 1,056-1,064.*

Bonding Aluminium to Cast Iron

A process for the bonding of aluminium to cast iron has been developed in America. In providing a cylinder with aluminium cooling fins a suitably cleaned ferrous liner is immersed in a bath of molten aluminium. When the liner has reached the temperature of the aluminium, it is chemically attacked by the light alloy and an aluminium rich alloy forms on the face. The cylinder is removed from the molten aluminium bath, placed in a mould, and the aluminium casting is poured about it. It is possible to save machining operations when fins of sufficient thinness are cast on cylinder barrels. An improvement of 50 per cent in heat dissipation and decrease of 25 per cent in weight have been achieved over all-steel construction. The bond is chemical or molecular, transfers heat from one metal to the other without loss at the interface, and also transmits stress. It is strong and runs up to Vickers 875 diamond Brinell hardness. Bond strengths have reached 17,000lb. per sq. in. and shear 8,000lb. per sq. in. on tests.—*S.A.E. Journal, Vol. 57, November 1949, p. 82.*

Percussive Welding—Aluminium

Recent improvements have been made in percussive welding; but because of the patent situation full disclosure of the nature of these improvements cannot yet be made. However, it can be said that stored energy welding current is used, carefully metering the exact amount of energy discharged during the weld and thus permitting close control of amount of metal melted at the weld interface. Also, in order to obtain the impact desired immediately after application of the welding current, the work is hammered together by using the energy stored in an air cylinder whose piston is connected to the moving electrode. The piston is held back until the moment of impact by means of an electric solenoid. Welding control circuits open the solenoid circuit at the moment of impact, so air pressure then is free to ram home the piston and movable electrode connected to it. Such a system provides precise control of every factor involved in the welding operation. Amount of metal melted is controlled by amount of energy stored up and dissipated in making the weld. Movement of the electrodes and pressure exerted prior to hammer blow can be controlled by usual methods. Hammer blow timing can be precisely controlled by electronic circuits connected with the actual application of welding current, and can be applied at any desired point succeeding application of welding current. Strength of hammer blow can be adjusted easily by selection of air cylinder and pressure maintained therein. The result is a precision flash-welding machine. The article includes micrographs of a percussion-welded tube joint between copper and aluminium alloy tubing of $\frac{5}{16}$-inch outside diameter and $\frac{1}{16}$-inch wall. The weld was made with 1,200 microfarad condensers charged to 2,200 volts, discharging to the weld through a 1,000 : 1 ratio step down transformer giving approximately 80,000 amperes at 2·2 volts during the second flashing period during which about $\frac{1}{4}$-inch was burnt off each piece. The weld zone measured only 0·0004-inch and the original structure adjacent to the weld was found unchanged. Some slight cold working of the aluminium due to the hammer blow had occurred. There was no internal flash on the welds.—*C. Bruno and C. W. Birdsall, The Welding Journal, New York, Vol. 28, November 1949, pp. 1,070-1,075.*

Refrigerant Leak Detector

In any refrigeration system the loss of refrigerant through a comparatively small leak can soon render the entire equipment ineffective. Where Freon or methyl chloride is used as refrigerant, one of the servicing difficulties has been to locate small leaks quickly and accurately. With the Type H leak detector, manufactured by the British Thomson-Houston Co., Ltd., minute leaks can be rapidly detected. The instrument is highly sensitive to vapours of halogen compounds and gives both a visual and audible indication of a leak as small as one at the rate of even less than one-fiftieth of an ounce per year. The instrument consists of a detector unit and a control unit. The detector is a hand-held probe with a handgrip and a plastic-tipped metal nozzle. The unit contains an element sensitive to vapours of halogen compounds (those containing chlorine, bromine, iodine, and fluorine) and a motor-driven impeller which draws air through the element. A small loudspeaker built into the detector unit emits an audible clicking sound, and when the sensitive element detects the presence of vapour of a halogen compound, the frequency of the "clicking" increases, thus giving an audible indication. A visual indication of a leak is given by an increase in the meter reading on the control unit. The apparatus requires 190 to 260 volts a.c. supply, the consumption being 150 watts. In addition to the testing of refrigeration systems, the detector can be used for the pressure testing of tanks, pipes, joints, and welds. The method is to pressurise the system with air carrying a small quantity of "tracer gas", such as chloroform vapour, carbon tetrachloride, Freon, etc. One advantage is that as soon as the required pressure is attained, testing for leaks can begin immediately; there is no waiting for long periods for water to seep through any minute holes, as in the conventional hydraulic method of testing.—*The Shipping World, Vol. 122, 4th January 1950, p. 13.*

Portable Leak Detector

A portable device has been developed in the United States for quickly locating leaks in pipes and tanks. It is said to detect even the smallest pin-point leaks by a simple vacuum principle. Seams are covered with soapsuds and the inspection box with a clear glass top and soft rubber base is laid over the seam. A strong vacuum created within the inspection box draws air or gas through any leaks, causing soap bubbles to form. These instruments are available in a number of sizes and shapes for inside corners, outside corners, as well as circumferential and straight seams.—*Petroleum, Vol. 12, November 1949, p. 292.*

Surface Preparation of Mild Steel Prior to Fabrication

In dealing with various methods of surface preparation the author refers to hand-operated compressed-air-propelled shot blasting equipment which is the only type of shot blasting machine which can be used on site. A recent development in the United States is the Vacu-Blasting equipment, which is shown diagrammatically in Fig. 5. In this case the shot blast nozzle is surrounded by a suction hood and a flexible brush is provided around the circumference to provide a seal. The claim is that in this way blasting can be carried out in shop without creating a dust nuisance,

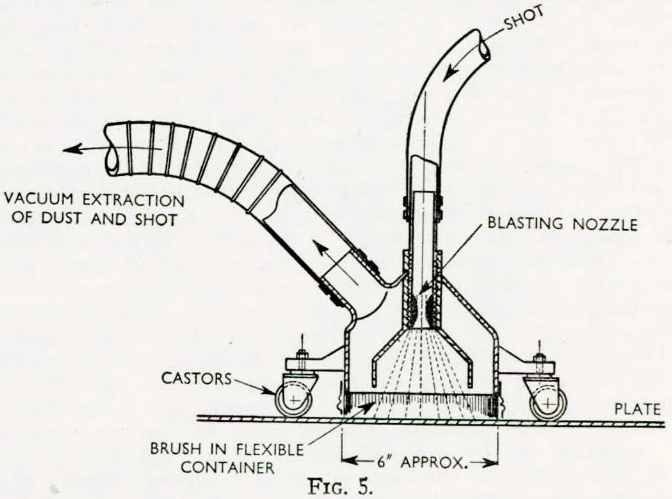


FIG. 5.

and it seems likely that, provided the cost of maintenance and spares, which is admittedly high, is not excessive, some development on these lines might prove to be very useful for touching-up plates and sections where the clamps had screened them from the shot, and furthermore it might find use on site for cleaning welds prior to the application of the protective coating.—*Paper by W. A. Johnson, read at a meeting of The Institution of Mechanical Engineers on 11th November 1949.*

Hot Air Turbine Ship's Propulsion Plant

This recent invention relates to power plants incorporating elastic fluid (e.g. hot air) turbines for use in the propulsion of ships and other purposes. As will be seen from the accompanying illustrations, the high pressure turbine (a) is arranged to drive two compressors (b and c) with an intercooler (d) between them and a low pressure turbine (i) driving an alternator (k) supplying power to a propeller motor (l). Alternatively the turbine may be arranged to drive a system of reversible gears or a reversible propeller, etc. Atmospheric air taken in at (b^1) is compressed in several stages to a final pressure which may be, for example, about 400lb. per sq. in. at full power, the air being cooled between the compression stages so that it leaves the compressor at about 200 deg. F. The air leaving the last compressor (g) passes to an air reservoir (m) from which it passes through a control valve (m^1) to an independently fired heater. In the design shown, the heater comprises a first section (n) for heating the air for the h.p. turbine, a second section (n^1) for heating the air exhausted from the h.p. turbine, and a third section (n^2) for heating the air exhausted from the intermediate turbine before passing to the low-pressure turbine. The heater sections are provided with two independently fired oil furnaces (o^1 and o^2)m, the products of combustion from the furnace (o^1) passing first to the heater section (n^1) and the products of combustion from the furnace (o^2) passing first to the heater section (n^2), while the flue gases from both furnaces pass to the heater section (n) for the h.p. turbine. Should it be found that the temperature of the air supplied to the h.p. turbine is excessive, some of the flue gases may be by-passed round part of the heater section (n) by means of damper (o^3). The air may, for example be heated to 1,300 deg. F. at the three turbine inlets, and the air exhausted from the l.p. turbine is then passed through the pipes (p^1) to the furnaces (o^1 and o^2), to be used as combustion air. The heater is preferably constructed with an external jacket (p^2) traversed by the exhaust from the l.p. turbine as it passes to the burner. So that the component parts of the three turbines and their pipe connexions and accessories may be subjected to the design temperature, they must be gradually heated before putting the plant into active operation. In the case of marine propulsion it is particularly important to avoid temperature fluctuations when manœuvring or in changing load, since such fluctuations would cause rapid expansion or contraction with possible detrimental consequences. To prepare the plant for service, an auxiliary Diesel generator (r^3) is used to drive the air compressor (s) for the purpose of pumping air into the air reservoir (m). The oil pumps (t^1 and t^2) are started up, the burners ignited, and the refractory linings and heater elements raised to a suitable temperature before the valve (m^1) is opened to allow air to flow through the first section of the heater to the h.p. turbine control valve. When the air has reached a temperature suitable for admission to the turbines, the h.p. control valve is opened, valve (u^1) (see Fig. 3) is partly opened and valve (u^2) is partly closed, thereby enabling the exhaust from the i.p. turbine to be suitably apportioned between the l.p. turbine and the furnace (o^1). The rate of combustion in the two furnaces is slowly increased until all three turbines and their connexions have reached the desired temperature and the plant is then ready for operation. When manœuvring, the air temperature will be reduced appreciably below that used for maximum power under stable conditions, for instance, from 1,300 to 1,100 deg. F., to provide a margin of safety when rapid changes in power are required and fluctuation in temperature may be difficult to control. The power delivered to the propeller will be varied, first, by the opening or closing of the h.p. turbine control valve, thereby increasing or diminishing the amount and pressure of air passing through all three turbines, and, second, by increasing or reducing the speed and output of the fuel pump serving furnace (o^2) and consequently the temperature of the air entering the l.p. turbine. The temperature of the air entering the h.p. and i.p. turbines will be controlled within close limits of co-ordinating the speed and output of the fuel pump of furnace (o^1) with that temperature. At small loads or no load, valve (u^1) may be used to bypass part of the exhaust from the i.p. turbine or furnace (o^1) instead of going to the l.p. turbine, and this arrangement also facilitates the use of part of the plant for restoring full pressure in the air reservoir (m), which on occasions may be advantageous.—*Brit. Pat. No. 624,948, issued to J. Johnson. Complete specification accepted 20th June 1949. The Shipping World, Vol. 121, 14th December 1949, p. 624.*

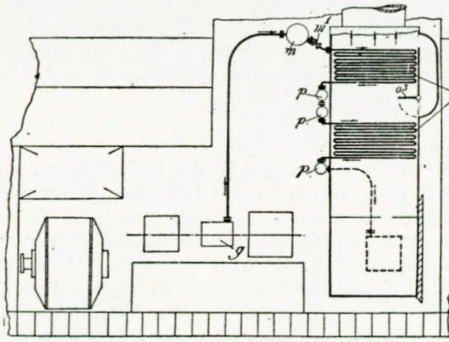


FIG. 1.

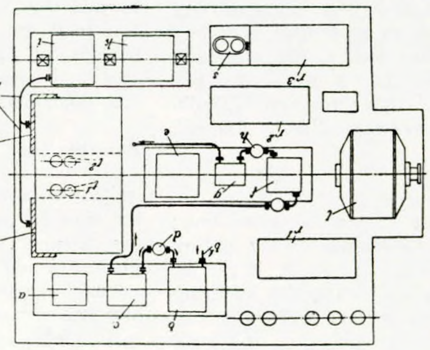


FIG. 2.

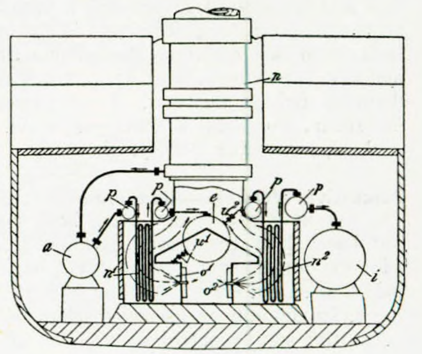


FIG. 3.

Reversible Transmission for Gas Turbines

A reversible transmission system for gas turbines is shown in Fig. 6. The mechanism is intended to ensure synchronism of the engaging clutch elements of a reverse gear automatically with the braking of the turbine for reversing. The gas turbine (1) drives the propeller shaft (4) through gearing incorporating clutches (2, 3). By disengaging the clutch (2) of the ahead shaft and engaging the clutch (3) a reverse drive is obtained. When it is desired to drive astern, the gas supplied to the turbine through the pipe (11, 12) is cut off at the valve (13). The clutch (2) of the ahead shaft (8) is disengaged, and the astern clutch (3) is engaged only when the appropriate clutch elements are in synchronism. A compressor (14) provides the necessary braking action, gas being admitted to the high-pressure end (20) through a pipe (15). When the momentum of the turbine and compressor rotors has been absorbed, the gas is diverted by the valve (16) to the low-pressure end (21) of the compressor by way of the pipe (17). The result is that the turbine (1) is rotated in the opposite direction. The clutch element (19) is driven in the same direction as the element (18), which is compelled to rotate in the ahead direction,

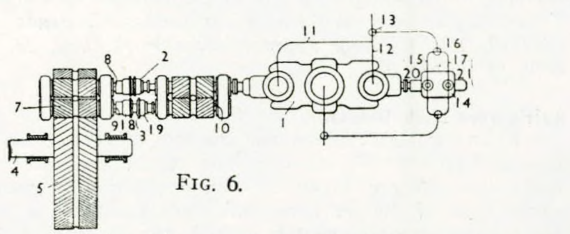


FIG. 6.

due to the way of the ship. As the turbine builds up speed, the revolutions of the elements (9, 10) of the astern shafting synchronize, and the clutch elements (18, 19) act accordingly, whereupon the clutch (3) is engaged. Gas to the compressor is then cut off by the valve (13) and admitted to the turbine, finally driving it in the ahead direction and thus through the pinion (7) to the astern shaft, driving the gearwheel (5) of the propeller shaft (4) astern.—*Brit. Pat. No. 624,180, issued to Vickers-Armstrongs, Ltd. and G. Wood. The Motor Ship, Vol. 30, December 1949, p. 366.*

Gas Turbine Operation at 1,500 deg. F.

Successful operation with inlet gas temperatures of 1,500 deg. F. has been accomplished in tests on the Allis-Chalmers 3,500 h.p. gas-turbine plant at Annapolis, Md. This turbine unit is the first to be designed for long-time service with operating temperatures as high as 1,500 deg. F. Installed at the U.S. Naval Engineering Station in 1944, the elements of this gas-turbine unit have since been operated in more than 3,000 turbine-hours of rigorous testing, and have been started and stopped more than 400 times. Successful operation of the unit at 1,350 deg. F. was revealed in 1946, and subsequent tests have included operation for some length of time at temperatures ranging up to 1,524 deg. F. The performance of the test unit, which includes two gas turbines, an axial-flow compressor, a regenerator, and auxiliary equipment, has been very satisfactory thus far, the operators report. Despite the high temperatures at which the unit has been operated, there have been no failures of blading in the high-temperature turbine zones. Frequent inspections have revealed no indications that the unit will not have a satisfactorily long operating life. Heretofore, temperatures as high as 1,500 deg. F. have been achieved only in short-time tests of turbines operating above their normal design temperatures, or in short-life-cycle machines such as aircraft jet engines, in which the advantages of extremely high power concentration in a small volumetric space and very low specific weight offset the reduced operating life resulting from the high initial gas temperatures applied to lightweight machines. Thus a great deal of previously unavailable information is being obtained from the tests on the Annapolis equipment. Among the many design innovations being tested in this installation are various methods of applying cooling air whereby high-temperature parts may be cooled to protect them against undue weakening effects of extremely high temperatures. Other design features of the turbines have demonstrated their soundness in enabling the unit to withstand successful abnormal operating conditions which have occurred during some of the tests. Probably the greatest significance in the successful operation of this unit at a temperature as high as 1,500 deg. F. lies in the importance of high-temperature operation in increasing gas-turbine thermal efficiency.—*Mechanical Engineering, Vol. 71, December 1949, pp. 1,042-1,043.*

Transient Flow of Gases

Although problems relating to non-uniform flow of a compressible fluid have been tackled by a number of workers, investigations have generally been limited to the study of one-dimensional non-viscous adiabatic flow. This type of flow, however, is associated with so many problems encountered by mechanical engineers that an easy and reliable method of solution is extremely important. The method of characteristics described in this paper provides a comparatively simple method of solving such problems by a graphical process. It consists essentially in the simultaneous construction of two corresponding diagrams: (a) the state diagram showing the changes in the state of the fluid produced by spreading disturbances, and (b) the position diagram depicting the spreading of the disturbances. The graphical construction of these diagrams is based upon certain special mathematical properties of the equations which describe the motion and enable boundary conditions corresponding to any particular problem to be readily taken into account. Applications include the phenomena occurring in internal combustion engine exhaust and intake pipes, in particular, those relating to the Kadenacy system of scavenging. It is also applicable to the type of flow encountered during the emptying of cylinders, in pulsating ram jet engines and, in general, in any engine making use of the intermittent flow of gases. The phenomena occurring in long indicator passages during tests on high-speed engines also come within its scope.—*Paper by J. Kestin, submitted to The Institution of Mechanical Engineers for written discussion, 1949.*

Heat Transfer and Fluid Friction in Viscous Flow Across Tube Banks

In the course of a research programme on tubular heat exchangers, pressure drop and heat-transfer, data are reported for heating and cooling a medium-viscosity oil flowing across banks of vertical tubes in seven test exchangers. The apparatus variables include equilateral triangle, in-line square, and staggered-square arrangements; tube sizes of ⅜-inch and ¾-inch outside diameter; and pitch ratios of 1·25 and 1·50. The results are shown both in simple plots of pressure drop and coefficient of heat transfer versus the rate of flow and in generalized correlations. Tentative correlations are provided for friction and heat transfer which bring the data for these seven tube banks somewhat closer together than previous correlations. In a comparison of the heat-transfer coefficient versus pumping power loss per unit surface area, the smaller-diameter tubes at the smaller pitch ratio gave the best performance. The data include the following. When the pitch is increased for a given arrangement and tube size: at constant velocity the pressure drop is lower, at constant velocity the coefficient of heat transfer is lower, and at constant pumping power loss the coefficient of heat transfer is slightly lower. When the tube diameter is increased for a given arrangement and a constant-pitch ratio: at constant velocity the pressure drop is lower, at constant velocity the coefficient of heat transfer is lower, and at a constant pumping power loss the coefficient of heat transfer is considerably lower. The highest coefficients of heat transfer were obtained with the smaller tube sizes and the small tube pitches in the staggered arrangements.—*Paper by O. P. Bergelin et al, read at the 1949 A.S.M.E. Annual Meeting; Paper No. 49-A-87.*

Sea Water Laundering

During World War II emergency measures were taken by the United States Navy and the United States Army Surgeon General to do laundering aboard ships with sea water. Toward the end of 1944 the problem became so urgent that the Army Quartermaster General undertook an intensive research programme to provide the best available products and to devise processes for this task. This paper describes the work done by and for the Quartermaster to this end. The lack of accepted laboratory methods for measuring detergency was considered a bigger handicap than the critical shortage of detergent products. An extensive programme was undertaken to advance the test methods, to evaluate all detergents that could be made available in quantities necessary for military requirements, and to devise optimum washing procedures for utilizing the most promising ones. The work was done in three phases: (1) laboratory detergency tests, (2) full scale laundry operations with synthetic sea water, and (3) confirmation of pertinent findings under actual use conditions with sea water. As a result, four products were approved as suitable for military needs of sea water laundering, and improved laundry formulas were developed for all the major classifications of military clothing.—*T. H. Vaughn, et al., Industrial and Engineering Chemistry, Vol. 41, January 1949, pp. 112-119.*

Boiler Feed Pumps for Marine Service

The construction of boiler feed pumps of the centrifugal type for marine service has experienced a rapid evolution in recent years. Boiler pressures, feedwater temperatures and speeds have gone up and will continue to go up. The space available and the weight required have been reduced. The pump designer has used the experience gained in this and other fields to produce pumps and turbines to meet the requirements of continuous duty, with high efficiency asd reliability. The pressure per stage is limited to the strength of the rotating parts and resistance to erosion of the materials used, providing available suction conditions do not impose restrictions on the speed and capacity. Normal engineering conservatism of not using stage pressures in excess of some arbitrary values frequently limits the pressure per stage, although the development of the chrome steels, together with improved techniques in casting, fabricating, and balancing permit appreciably greater stage pressures than those used in present marine practice, without danger of corrosion, erosion, or vibration. The necessity of having cavitation-free operation over a wide range of capacity

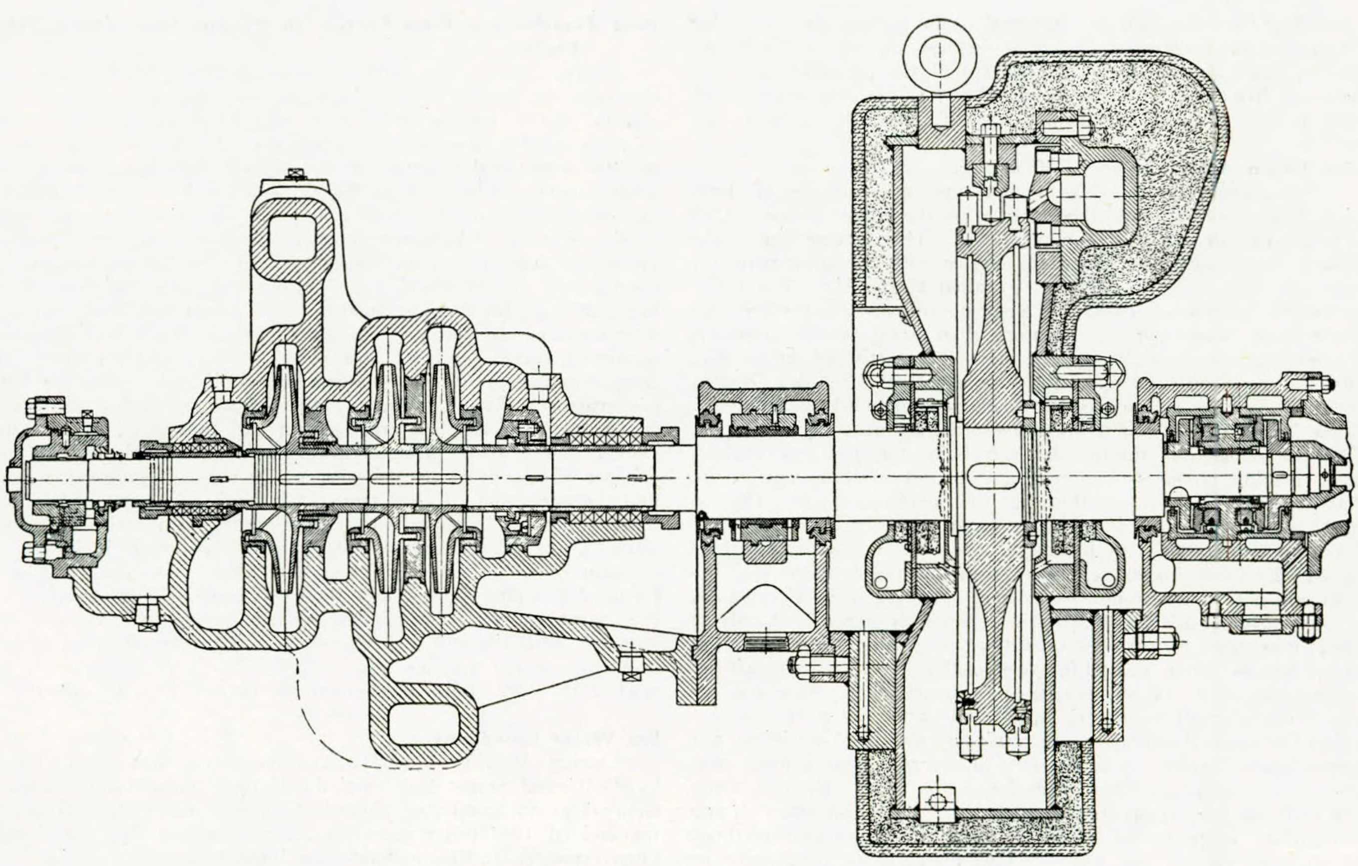


FIG. 4.—*Three-stage pumping unit with three bearings and integral shaft for pump and turbine.*

requires a definite minimum suction pressure in excess of the vapour pressure. This excess pressure is affected by fluctuations in water temperature, suction pressure, piping losses and other factors, and is called the "net positive suction head", its required value depending upon speed and capacity of the pump. The pumps must be designed to meet these conditions without causing vibration or pressure surge. Aside from strength considerations of the rotating parts and material erosion, the deciding factors are "net positive suction head" and the specific speed of the pump impellers. Specific speed is a guide to the obtainable pump efficiency and to the head per stage that is practical, and to the number of stages required. There is a close interrelation, for most satisfactory operation, between head per stage, capacity and the net position suction head. For shipboard installation the smallest unit, without sacrifices of reliability and serviceability is usually the most desirable one. This in effect, calls for a high rotative speed and compact design. The author shows a four-stage pump with flexible coupling. All four impellers are of the single-suction type, the first- and third-stage impellers face toward the outboard end, the second- and fourth-stage impellers face toward the inboard side, thus giving axial balance of the rotor. Water-cooled stuffing boxes are used. On this four-stage pump, the interstage connections are made by welding on crossover connections. This type of unit is particularly adapted for high-pressure and high-temperature applications, similar to those used in recent tankers and passenger vessels. Fig. 4 shows a 3-stage pumping unit with three bearings and integral shaft for pump and turbine. All impellers are of the single-suction type. Axial balance is obtained by arranging the first- and second-stage impellers so that the inlets face in opposite directions and the third-stage impeller with a wearing ring on the back. This design is suitable for higher discharge pressures than can be handled by a two-stage pump or for a lower net positive suction head at the same discharge pressure than a two-stage pump.—*F. Fritscher, Marine Engineering and Shipping Review, Vol. 55, January 1950, pp. 62-66.*

New Swedish Oil Engine

The new Bolinder type WA marine model is a low compression two-stroke crankcase scavenging engine, built in powers of 145 b.h.p. at 325 r.p.m., and 200 b.h.p. at 280 r.p.m. No reduction gears are being used with these new engines, reversing being by means of a reverse gear or by reversible-blade propeller. Starting is by electric starter or cartridge and air, and Bolinder rapid-heating lamps are fitted as a standby to assist starting in cold weather. The fuel injectors employ a new Bolinder principle —which provides two different types of spray according to load requirements. A hand-operated lever fitted to each injector closes the central spray orifice at light loads and the fuel is then injected through two side orifices whose jets impinge on the hottest part of the combustion chamber surface and thus maintain a sufficiently high temperature to enable the engine to accept full load immediately. At full load the lever is moved to a position which allows the centre spindle to lift and uncover the full-load central spray orifice.—*Gas and Oil Power, Vol. 44, December 1949, pp. 374-375.*

Storage Additive Type Lubricants

In this article, which deals with the storage and handling of lubricants in general, reference is included to precautions to be taken in storing additive type lubricants in the shipping drum itself, using a hand pump for removal of the contents when needed. This means that oil drums must be set up on end. The procedure is perfectly satisfactory provided there is no chance of water getting into the lubricant, especially if it contains an additive. Straight mineral oils are not adversely affected by a little moisture —they may become cloudy, but after standing for a few days they clear up as the moisture settles to the bottom. Moisture, however, is a serious contaminant when an oil contains additives, most of which are water sensitive. Contact with water may remove some types of inhibitors, and also affect their stability. If, therefore, a turbine or hydraulic oil, or an additive motor oil, or preservative oil, is to be stored in the shipping drum, with the drum standing on end to enable removal of the contents by pump or ladle, extra precautions must be taken to see that no moisture accumulates on the head of the drum, or can be splashed or sprayed into it. A good tight bung and seal on a drum is excellent assurance that this won't occur; a tight oil pump fitting in the bung hole of an oil drum helps protect this type of lubricant. Above all, do not store additive type lubricants where they can be exposed to moisture either from rain, hosing or drip. Keep drums on their side until the contents are to be used. Keep them under cover, preferably indoors, because even the best of closures do not guarantee freedom from water contamination if a drum stands "heads-up" with water on top.—*Lubrication, Vol. 4, No. 7, 1949, pp. 1-12.*

Progress in High Duty and Alloy Cast Iron

It has long been known that if the carbon in cast iron is retained in the combined form, that is the iron is white as cast, then with suitable compositions, by a special type of annealing heat treatment known as malleability, the graphite can be made to separate gradually from the metal, and under these conditions it assumes a nodular form, or as compact clusters of fine graphite. The beneficial effect of rendering the graphite nodular is seen at once in the enhanced properties of malleable cast iron. According to the processes used, the tensile strength of malleable varies from about 20-30 tons per sq. in., and the iron has a measurable elongation of 5 per cent or upwards; of greatest importance is, however, its increased toughness or shock resistance. For many years past foundrymen have realizezd that there would be a great advantage in any process which would lead to the production of grey cast iron, as cast, which would have the properties of malleable, as a result of the graphite being in a more compact form. It is this goal which has now in recent times been realized. Studies made by the investigators of the British Cast Iron Research Association led to the development of a process in which nodular structures could be produced by the addition of cerium to the molten alloys. Another process for the production of spheroidal graphite cast iron has been developed in the United States by the International Nickel Company. The basis of this work was the discovery of effective methods of introducing magnesium. The magnesium process can be applied with great advantage to all engineering types of grey cast iron produced in the foundry. For optimum results, however, careful control over composition is essential. The process is simple and safe, and can be applied in ordinary foundry practice without interfering with normal production. The foundry properties of the metal are good, although special precautions are necessary to meet the particular characteristics of the new iron, such, for example, as its rather higher shrinkage as compared to ordinary grey iron. Experience to date, however, has shown that it can be applied without substantial modification, to the normal types of castings made in the foundry. The fluidity of the metal is good, and it has now successfully been applied to the production of single castings up to 20 tons weight on the one hand, and to light repetition castings on the other, where production runs of many thousands have been successfully made. The new spheroidal graphite cast iron, in the cast condition, has a tensile strength of over 40 tons per sq. in., whilst under particularly favourable conditions much higher strengths may be achieved. As cast, this tensile strength is associated with a high yield point, and elongation generally from 2-5 per cent for irons with a pearlitic matrix. Of greater importance, however, is the marked shock resistance shown by the iron as measured by an impact test; it is usually five to ten times that of ordinary iron.—*A. B. Everest, Metallurgia, Vol. 41, December 1949, pp. 84-88.*

Steam Atomizing Burners

Steam atomization was the earliest successful method of burning fuel oil. However, the inefficiency of the early steam atomizing burners and the successful development of a pressure atomizing burner made the old steam atomizing burners less popular. It was used mainly for burning very heavy oils and residues from cracking processes. Recently, however, the steam atomizing burner has become more popular. The type of oil which has been sold in recent years known as Bunker C has caused formations of slag on superheater tubes requiring frequent outages and cleanings. The steam atomizing burner has reduced considerably the tendencies for building up this slag. The large amount of water consumed by the old burners was a disadvantage, some types using up to 10 per cent of the steam generated. The present-day steam atomizing burners use less than one per cent of the steam generated for atomization. This means that for every 1,000lb. of steam generated about 7½lb. is used for atomization. With the regular straight mechanical atomizing burner, the range of capacity is about 2 to 1. With the so-called wide range plunger type or return flow type burners, the range of capacity is in the neighbourhood of 4 to 1. However, with the steam atomizing burners, a capacity range of 10 to 1 or 15 to 1 is easily obtained. Steam atomizing burners are entirely suitable for either hand or automatic operation. It is unnecessary for the firemen or the combustion control to adjust the atomizing steam pressure. The firemen or control merely regulate the oil pressure as required and the atomizing oil pressure automatically follows at 20lb. per sq. in. higher than the oil pressure. This relationship is maintained up to a maximum atomizing steam pressure of 130lb. per sq. in. g. At this pressure, the oil pressure is 110lb. per sq. in. g. The oil pressure can build up from there to its maximum, normally 300lb. per sq. in. g., and the atomizing steam pressure will remain constant at 130lb. per sq. in. g.—*W. B. Hill, Pacific Marine Review, Vol. 46, December 1949, pp. 108-109, 128.*

Stabilization Reduces Ship's Roll

Ship stabilization tests are now being carried out aboard the mine sweeper U.S.S. *Peregrine* (see Engineering Abstracts, September 1949, p. 95). A model of the device has been tested at Stanford University. As a result of the testing, instrumentation and analysis of the model, the engineers have made some discoveries about locating the stabilizer aboard ship. They have found that the cross duct of the stabilizer—the line through which water is made to flow from one tank to the other—should be placed above a ship's centre of gravity. In this position, the force of the water travelling in the duct aids the stabilization process. This means that the duct should be relatively small in order to increase the acceleration force of the water being transferred from one tank to the other. If the duct must be placed below a ship's centre of gravity, it should be made as large as possible to minimize the force of the water's acceleration in the duct—a force which in this case is in the opposite direction of the force needed to stabilize the ship.—*Pacific Marine Review, Vol. 46, December 1949, pp. 81, 130.*

Considerations in Provision and Operation of Overseas Mail Service

The author discusses a number of operating problems and their solutions. Referring to the abnormal incidence of propeller shaft failures during the past few years, the point is made that stress investigations have not indicated inevitable failure. Whatever the measure of the analyses, the effect of corrosion, if present even in slight degree, is seriously to aggravate proneness to fatigue failure. Rubber waterstops at the forward face of the propeller are of two main types. In one case compression of the rubber is fixed by the propeller nut, in the other, by a gland ring independently adjusted. Both types have several modifications. Rubber is the only suitable material. The rubber ring should be endless, and consequently must be placed on the shaft before the propeller. It cannot be renewed without removing the propeller, an operation which generally extends the normal time in dry-dock. Survey requirements are met by examination at intervals of three years. Physical features of rubber have wide variations in hardness, resilience, and in plasticity. All rubber is subject in time to plastic flow under pressure. Change of form consequent upon plastic flow diminishes resistance water seepage. The nature of rubber is such that the ring must be finished to dimensions in a suitable machine before it reaches the dry-dock. If delay is to be avoided the ring should be available when the propeller is withdrawn. During the past five or six years, coincident with the loss of propellers in vessels built in wartime, shafts of other vessels of varying ages were condemned on account of fatigue cracks, either adjacent to the after end of the liner, or to the forward end of the taper bore of the propeller boss. At each of these positions there is stress concentration in conjunction with galvanic action between copper and steel if sea-water is present to act as electrolyte. Quantities of water were regularly found when the propeller nut was removed. The propeller cap is usually of light structure with widely spaced fastening to the boss, making a poor joint. Investigation followed as to the possibility of water passing from the inside of the cap between the face of the nut and that of the propeller. Slight damage was identified from the box wedges set between nut face and propeller face to drive the propeller home on the taper before the nut was finally set up. None of these investigations discovered fault sufficient to explain the abnormal change. War-time workmanship associated with propeller-shaft examination in foreign ports was at first sought as explanation, but the condition was found later, in cases where apparently proper care had been given when the propeller was previously removed and replaced. The cause of the change was obscure. Further search for the cause of the change recalled that in the years immediately preceding the 1939-45 war there had been a general change from propellers with bronze blades and separate steel bosses to integral solid-bronze propellers. The change was favoured by higher propulsive efficiency of the solid propeller, on account of the smaller boss, a difference of the order of 3 per cent or an annual saving in the cost of fuel of £3,000-£4,000 for a ship of 25,000 gross tons with boiler fuel at £4 per ton. Further investigation established not only that the corrosion was confined to the solid propellers, but that it applied to each solid propeller in a greater or less degree without exception. The cored space in the solid propeller contains air at atmospheric pressure when the propeller is assembled in dry-dock. Immediately the vessel floats, there is an external static head of 10-15 feet of water, and added pressure impulse as the ship pitches in service. Balance of pressure inside and outside the propeller boss requires that one-third to one-half of the cored space inside the boss shall be filled with water. Access may be past minute error in the fit of the rubber ring, fault in the mating faces of propeller nut or boss, or past the backlash clearance of the screw thread on the shaft. In time, pressure will be balanced by water entering the boss. There is no corresponding condition in the boss with separate blades where the boss fits over the whole of its length. The obvious means of prevention is to omit the cored recess or to ensure that after assembly the space is completely filled with non-corrosive grease or other suitable material. It is suggested that, had the propeller bosses been fitted over their whole length in "Liberty" and other vesels, few propellers would have been lost.—*Twenty-second Thomas Lowe Gray Lecture, delivered by J. Gray, C.B.E. at the Institution of Mechanical Engineers, 20th January 1950.*

Design and Performance of Air Injector

The air injector, in its various forms, is a device which has many applications in engineering practice, and several attempts have been made to analyse its mode of action, some of these having been supported by experimental work. Most of the experimental results available are related to ejectors in which relatively high-pressure steam is utilized as the driving fluid, but even in these cases the information provided is restricted to a narrow field. The investigation described relates to an air ejector employing as the driving fluid air at a relatively low pressure, not exceeding 40lb. per sq. in. (abs.), and covering a wide range of operating conditions by means of interchangeable nozzles. Two distinct experimental arrangements were built—one for the set of conditions in which the ejector draws in a relatively small quantity of suction fluid and pumps it through a relatively high pressure-ratio, and the other covering conditions in which the quantity of suction fluid is much larger, but the pressure-ratio is quite small. For a given initial pressure and quantity of driving fluid, the rate of mass flow of suction fluid depends chiefly on the diameter of the combining tube, in which the driving and suction fluids mix; in the experiments, the ratio of combining tube area to driving nozzle area was varied in twelve steps, covering a range of area ratios from 1·44 to 1,110·0, and compression ratios ranging from about 2 to about 1·001. Efforts were made to find the best proportions of those parts of the ejector which exert a major influence on performance, and certain conclusions are drawn from the results of the experiments. Theoretical aspects of the problem are briefly discussed.—*Paper by L J. Kastner and J. R. Spooner, submitted to The Institution of Mechanical Engineers for written discussion, 1950.*

Fluid Flow Measurement

Several advantages over the primary elements in current use are claimed for a new primary element for fluid flow measurement manufactured by the Bethlehem Foundry and Machine Company. Among these is greater simplicity in construction and operation arising from the short length of the tube (normally less than two pipe diameters) and its freedom from straight run limitations in regard to pipe bends and elbows. Greater economy is secured through the negligible resistance to flow of the tube, which reduces loss of head. Greater range, higher accuracy and simplified metering of gases and vapours are other advantages of the new element. The flow tube consists essentially of a short tube equipped with two groups of pressure nozzles around the inner periphery—one group pointing upstream, the other downstream. These nozzle groups are connected by common pressure rings, from which connexions are made to the high and low side of a standard flow meter. When the tube is in operation, a total head equal to the static head minus some component of velocity head is impressed. The differential head impressed on the meter is the difference between these two heads.—*Abstract from Energy, U.S.A., Vol. 11, No. 5, 1949, p. 5; Engineers' Digest, Vol. 11, No. 1, January 1950, p. 31.*

Diesel Electric Propulsion of Ships

The author surveys the field of application of Diesel-electric propulsion and describes the various electric transmission systems available. Of recent installations those in the dredgers *Mersey No. 26* and *Mersey No. 27* are discussed at some length. The author concludes that for ships up to about 4,000 h.p. Diesel-electric propulsion is justified when special features are required. This form of propulsion offers advantages not available with any other form when there is a large amount of manœuvring compared with full power running, when there is a large deck machinery load which may be required simultaneously with propulsion or when a special propeller characteristic is wanted. If none of these features is present and if Diesel propulsion is required, either the direct Diesel or some form of mechanical transmission is generally the more economical solution.—*Paper by E. L. N. Towle, read at a Meeting of The Institution of Engineers and Shipbuilders in Scotland, 1st November 1949.*

Arc Underwater Welding

The application of arc underwater welding, though it is not as widely practised as it might be, has already many practical applications in the erection, repair, and salvage fields of marine and civil engineering industry. There exists a vast field of possible applications as regards ship repairing below the waterline. Ship repairing includes the welding of leak-proof patches to ships' hulls, the strengthening of buckled plates with stiffeners, the welding of strained plate joints and the sealing of leaking rivets. It is reported that Lloyd's Register of Shipping has approved the welding of underwater patches only until such times as the next routine drydock examination of the vessel concerned. In the marine salvage sphere the fixing of lifting lugs, towing hooks and other attachments which may assist in the raising of sunken ships must be mentioned. The majority of underwater applications are in comparatively shallow waters. By that is meant water with depths of approximately 30 feet, so that the welder diver can work for lengthy periods at a time, whereas in deep water a greater portion of the divers' time is necessary in reaching the required depth and resurfacing; thus the operating time is considerably reduced. It is essential to bear in mind that the speed of underwater welding is dependent on the working conditions of the welder diver. Some of the important factors that contribute to satisfactory tests are: the accessibility and location of the work, the visibility above and the turbulence below the waterline, the visibility of water, the speed of the current if there is one, and the confidence of the diver in his attendant, equipment and surface plant. As could be reasonably expected, the repair of a ship lying on a muddy reef, or ledge, in very low temperature and fast running water, with the diver having to work in a cramped or horizontal position, could not be performed as quickly nor as satisfactoriiy as a straightforward welding operation in still water with a readily accessible joint. Underwater welding is equally effective in salt or fresh water, though the former has a far higher electrical conductivity. This higher conductivity has its drawbacks insofar as it has increased electrical corrosion problems to be considered. Underwater welding is limited in depth of operation more by the human element rather than electrical or mechanical factors. It is considered that where a diver is capable of working under water welding is capable of being done, although allowances have to be made for the voltage drop which is about 5 per cent per 100 feet of cable length. As to the welding plant and equipment, any portable d.c. welding generator with a minimum of 80 volts open circuit, with a 400 amp. output and capable of rapid voltage recovery will be found satisfactory. On an average, 500 amps. should be considered as a maximum, as above this figure greater currents do not produce better welds and there is difficulty in ensuring insulation and protecting the diver. Though satisfactory welds have been produced with an a.c. welding transformer, the use of a.c. current for underwater welding is not recommended owing to the more elaborate precautions that are necessary. In addition to the stringent regulations that are required by the Home Office, it has been established that when using a.c. current, as with d.c. using reversed polarity, there is a greater danger from electrolytic corrosion.—*C. C. Bates, The Shipping World, Vol. 122, 11th January 1950, pp. 85-87.*

Application of Aluminium Alloys to Shipbuilding

This report summarizes present day knowledge with regard to light alloys insofar as that knowledge is applicable to shipbuilding. Only aluminium-base alloys are considered, athough some information on available magnesium-base alloys is given in an appendix. It does not appear, however, that the magnesium-base alloys will be of much use in marine work. The available aluminium alloys are discussed generally and reference is made to their physical and mechanical properties. The alloys are mainly of two kinds: those whose strength is increased by cold working, and those whose strength is increased by heat treatment. In the former come the alloys containing small percentages of magnesium and the latter include the duralumin type alloys. The most suitable for shipbuilding purposes appear to be non-heat-treatable alloys containing 3½ per cent and 5 per cent magnesium known as A.W.5 and A.W.6 and a heat-treatable alloy containing silicon known as A.W.10. These are medium strength alloys with minimum ultimate strengths ranging from 14 to 20 tons per inch and have very high resistance to marine corrosion. Stronger alloys of the duralumin type are available, but are not considered suitable for marine work in view of their reduced corrosion resistance. Corrosion and fouling are discussed. With regard to the latter the usual anti-fouling compositions used on steel ships are unsuitable as the mercury and copper compounds attack the alloys. The development of suitable compounds is being pursued and preliminary work suggests that a satisfactory solution will be found. Painting and surface treatment are dealt with; this differs from the treatment given to steel plates and details of various processes are given. With regard to the joining of light alloys riveting appears to be the only practical proposition at the moment, although the practical development of arc welding is being pursued. Possible immediate applications to shipbuilding are suggested such as: superstructures, lifeboats and davits, masts and derricks, minor bulkheads, hatch covers, etc. Problems requiring further study are discussed and suggestions put forward for future investigations. Principal among these are: strength and watertightness of riveted joints in thick plates, arc welding, temperature effect in composite steel-aluminium structures due to differential expansion, etc. Work on most of these is already under way.—*W. Muckle, The British Shipbuilding Research Association, Report No. 22, November 1948.*

Threading of Stainless Steel

An article by J. J. Robert in *Machinist* gives the following recommendations for the threading of stainless steels: Correct selection of tools is of major importance; they should be of high toughness, and be kept well ground and sharp. Tungsten high-speed steels are suitable in many cases; for higher speeds of work cast-alloy tool materials are preferable. Cemented carbides are recommended for many continuous-cutting applications. Cutting speeds for stainless steels are usually considerably lower than those suitable for carbon steels, and the power required is approximately double. Good results are obtained by using a sulphur-base oil, containing a small percentage of vegetable or animal fat, to serve as lubricant. Machine threading has been found to be most successful, with a self-opening tangential-chaser die-head. It is recommended that the lip rake angles range from 22 deg to 25 deg. Chasers which have leads of about three to four threads are found to give long life. If necessary, short-lead chasers may be used for threading to shoulders, or in other applications where space will not permit long leads. Additional space, to permit ejection of the chips and small particles, should be left between the land of the die and the work, and also in front of the die-hard. A cutting speed of 15 to 40 f.p.m. is used for most production work involving die-hard threading, precise selection of speed being dependent on the pitch of the thread, the type of thread (taper or straight), and the condition of the material being cut. The same general practice is suitable for tapping, and, where the hole-size permits, a collapsible tap is preferred. Lathe threading of stainless steel involves the use of tools having a 6 deg. to 9 deg. top rake and a 9 deg. to 12 deg. side rake. Starting with a depth of cut of 0·025 to 0·035 inch, and gradual reduction to 0·005 inch in the finishing cut, gives a smooth, even thread. To demonstrate the application of the above general principles and recommendations, details are given of methods suitable for threading of three typical pieces of work in stainless steel: (1) bar stock in the form of a union nut; (2) a valve casting; and (3) a cast valve body.—*The Mond Nickel Bulletin, Vol. 22, December 1949, pp. 204-205.*

Performance of Crankcase Lubricating Oils

The paper deals in turn with the basic requirements of a crankcase oil; chemical and physical tests on new and used oils; viscosity, S.A.E. numbers, and viscosity index; common problems arising in lubrication; ring sticking; engine wear; oil consumption; bearing failures; engine deposits; the grouping of oils; the influence of Diesel fuel and petrol on engine deposits and wear; engine laboratory tests; the rating of oil performance on an engine test; and finally, the correlation between field and laboratory tests. It attempts to show that unsatisfactory engine life is not necessarily connected with the quality of the lubricating oil, and that many of the faults often attributed to the lubricant are, in fact, controlled by incorrect engine design, excessive operating conditions, or poor maintenance. Better results can usually be obtained by the use of additive treated oils. Reference is made to the "Supplement 1" series of oils which effectively deal with high-sulphur Diesel fuels, and examples are given of bench and field tests carried out on straight and treated oils.—*Paper by A. Towle, read at a meeting of The Institution of Mechanical Engineers, 14th February 1950.*

Lost Wax Process of Precision Casting

The paper describes a casting process which differs from standard foundry practice in that it uses a wax pattern in a high refractory one-piece mould to produce metal castings with a good surface finish to an accuracy of ±0·002 inch. The process involves making a master pattern in either hard wood or metal, relating it to a soft metal die by precision casting technique, and then the production of wax patterns from the die on an injection machine. Finally, the wax patterns are invested in refractory moulds, the wax is melted out, the mould baked, and the metal component is cast. The "lost wax" process is advantageous in cases where (a) the metal is unmachinable, or (b) where the component is of an unmachinable shape, or (c) where production by other methods takes too long. One of the most common applications is in the manufacture of gas-turbine blades. The tool costs are relatively low compared to the costs involved in alternative methods of manufacture, the die cost being a function of the number of castings required. The production of cheap castings is necessarily dependent on the scrap percentage being kept to a minimum; at present the scrap from the manufacture of gas-turbine blades is less than 30 per cent, and the author surmises that it would not be unreasonable to expect it to be less than 10 per cent in two years' time.—*Paper by J. S. Turnbull, read at a Meeting of The Institution of Mechanical Engineers, 6th January 1950.*

Test Apparatus for Studying Loop Scavenging

An apparatus is described for studying the process of scavenging the cylinders of loop-scavenged two-stroke engines. This consists essentially of a full-scale single-cylinder model of the engine with means for driving it for a single porting cycle. Tests upon the apparatus are made at atmospheric temperature and different gases are used at appropriate pressures to simulate the correct densities of the engine air charge and exhaust gas. The choice of suitable gases permits an easy analysis of the trapped charge for the determination of the efficiency of the scavenging process. Some experimental results and conclusions are presented, including the effects of operating with high charge densities and exhaust back pressures. A method is developed of expressing, from the experimental results, the scavenging performance of a given design of ports. This takes the form of curves relating the volumetric total charge supplied and the charging efficiency with the pressure ratio across the ports. These relationships are otherwise independent of the mass charge, the intake density, the exhaust back pressure, and the pressure loss across the cylinder. The influence of crankhaft speed is examined, as also are some causes of lost efficiency.—*Paper by H. Sammons, read at a Meeting of The Institution of Mechanical Engineers, 25th November 1949.*

Overlaying with Aluminium-bronze Electrodes

Where metal-to-metal contact is involved, hard grades of aluminium-bronze electrodes (160 to 300 Brinell) are rapidly assuming an important place in overlaying new and worn steel parts because they provide low-friction, non-seizing surfaces and wear remarkably well. Further, aluminium-bronze electrodes can be applied to high-carbon, alloy or tool steel without danger of spalling off or cracking the parent metal. Most grades of cast iron and bronze can also be overlayed. Overlays are successful on difficult-to-weld metals, such as large tool steel, medium carbon or cast-iron gears, because bronze inherently possesses a characteristic of penetrating into the pores of the metal and bonding firmly. Also, since aluminium bronze melts at approximately 1,950 deg. F., 700 degrees less than steel, it can be applied rapidly without materially affecting the heat treatment in the parent metal and the deposit feathers out as welded and leaves no craters when the arc is broken. Much of the long wearing characteristics of aluminium bronze can be attributed to the high polish attained by the bronze as it wears. Excellent life is obtained even when the bronze overlay is working against hardened steel parts due to the non-seizing effect of dissimilar metals working against each other. Bronze overlaid shafts operating against pump packing last three to six times longer than the original shafting which is usually harder than the bronze overlay. It is interesting to note also that packing is tremendously increased when shafts are overlayed with hard aluminium bronze because bronze wears smoothly and does not pit or corrode. When shafts are to be operated against bronze bearings, bronze overlays are not recommended because the dissimilar metal effect would not be present.—*J. A. Cunningham, The Welding Journal, New York, Vol. 28, December 1949, pp. 1162-1165.*

Cathodic Protection of Marine Tractor

This article summarizes the results obtained with the experimental installation of galvanic magnesium anodes to counteract hull corrosion. The equipment on which experimentation was conducted was a Chrysler Marine Tractor (Sea Mule) operating in the sea water of the Freeport harbour. This equipment could be dry-docked at convenient intervals for inspection, and its construction was such that magnesium anode attachments could be made readily to the hull. Underneath surface protection consisted of three coats of a vinyl-type paint followed by a vinyl antifouling top coat. Six 50-lb. high purity magnesium alloy anodes then were installed in the tunnel section under the deck. Attachment of the anodes was accomplished by welding extension rods to the ends of the ⅜-inch pipe which runs through a standard 50lb. sea water anode, and then bolting these rods to angle iron cross braces. Four days after the launching of the "Mule", a potential survey was made with a special Cu-$CuSO_4$ reference electrode and it was found that no point on the hull was below a potential of 1·0 volt, which indicated that the underwater surfaces were under effective protection. The Sea Mule operated in the Freeport harbour for five months and was then dry-docked for inspection. Observations indicated that highly satisfactory results were being obtained, because there was no significant blistering of the antifouling paint and the coating breaks caused from abrasion showed no signs of corrosion. A protective coating had formed on the hull where the original paint film was injured in welding on the channel sections. It is calculated that the average overall current density applied was 12·8 ma. per sq. ft. As indicated by a minimum Cu-$CuSO_4$ potential reading of 1·2 volts taken after three months of operation, the above current density is considerably in excess of the amount actually required for effective protection.—*O. Osborn, Corrosion, Vol. 5, December 1949, p. 416.*

Chromium-plated Piston Rings

Chromium has long been known to be extremely wear- and corrosion-resistant, and the plating of wearing parts in internal-combustion engines is by no means new, and has proved successful. However, its success in this field has always been hampered by the problem of maintaining an oil film on the surface, difficulties of processing and accurate sizing, and of relatively high cost. The alternative proposition of chromium plating the piston rings has been studied and one of the results has been that such rings not only reduce ring wear but show an equally good improvement in cylinder bore wear. Studying wear results with different materials, it is usual that, other things being equal, cylinder bore wear is generally less the harder the ring material. In carrying this to the extreme, when using chromium plating, it is found that bore wear can be reduced to one half the normal rate and frequently has been reduced to more than a quarter of the usual value. The hard ring seems to reduce bore wear for two main reasons. In the first place, it does not allow particles of loose abrasive to embed into the surface of the ring and thus lap the bores. Secondly, the hard ring does tend to resist wear, and in so doing there is less abrasive material being removed, which in turn would wear the cylinder bore. A third feature of the chromium-plated ring is its resistance to scuffing. It is felt by many that the greater part of normal cylinder wear takes place by attrition. This process occurs in the main at the top position of the piston ring when relative motion is nil, and therefore the oil film is replaced by boundary lubrication when operating temperatures are highest and when a sudden rise in gas pressure forces the ring out against the cylinder wall. This must inevitably cause some metal-to-metal contact and local microscopic seizures. Chromium seems to be resistant to the local seizure and hence improves the life of the affected parts. In developing chromium-plated rings there have been many difficulties to overcome. In the first place, the problems of adhesion between cast iron and chromium presented—as the liner-plating specialists found before the war—a multitude of difficulties which have now been solved. The next difficulty was to avoid sharp ragged edges at the corners of the rings, and the standard procedure now is to use a small chamfer before plating; this has entirely solved the problem. The next troubles were found in operation where, because the chromium is so wear resistant, it takes a very long period to bed into the cylinder. To overcome this, small diameter rings are manufactured with a tapered periphery of ½ deg. to 1 deg., and are lightly lapped in a dummy cylinder to establish a narrow land of contact around the full circumference. These rings rapidly bed into the cylinder and at the same time establish immediate oil control. For the larger diameter rings, such as are used in Diesel engine, taper turning is not practicable, and alternative means are being used quite effectively. Chromium-plated rings are now in regular production for many types of petrol and Diesel engines and are standard fitment by a number of makers with engines up to 8½ inch bore size. The employment of such rings in engines of appreciably larger cylinder bore is new, but progress is being made.—*Gas and Oil Power, Vol. 44, December 1949, pp. 370-371.*

Roller Chain Drives

The power capacity or rating of a roller-chain drive is an important feature to the user, designer, and manufacturer of the drive. Many successful drives have been and are in operation, and much operating experience has been gained. There is a real need, however, for a clearer picture of the mechanics of the roller-chain drive. If a rational study were available, it would help in the organization of limited data for general use, and it would help in solving the many special cases that arise. An analytical study of a simple type of roller-chain drive considering wear and vibration is presented by the author. The operating range of the drive is divided into two regions. Power-speed relations are given for these two regions. In one region the impressed tooth contact frequency is equal to or less than the highest natural frequency of the strand vibration. In this region vibration characteristics may be correlated with roller impact and such possible consequences as roller breakage, noise, heating, and wear of sprocket teeth. In the other region the impressed frequency is above the highest natural frequency of strand vibration, and the major factor is chain wear and elongation. The complete mechanics of the roller-chain drive is quite complicated, and thus a completely rational solution of the power-capacity problem appears difficult. It seems that the best that can be done with the present available information is to block out a general analytical method or framework. Numerous factors have to be considered for the general case of power rating. For example, one service requirement on the drive may be a certain limit on the chain wear elongation over a certain period of time. For an exceptionally severe application a limit may be placed on roller breakage and abuse. Another limit may be placed on the noise of the drive, and a limit may also be placed on the overall dimensions of the drive. Thus, in the general case, it may be necessary to consider various separate factors and to select one drive which will satisfy all the separate limits imposed. A review of the numerous factors involved indicates that two main ones are wear and vibration. They have somewhat the character of major independent factors. It appears that other factors can be correlated with these two, and that the analytical framework should include these two factors, at least in major roles.—*Paper by R. C. Binder, read at the 1949 A.S.M.E. Annual Meeting; Paper No. 49-A-10.*

Steam Cycle at 1,600 deg. F.

As the cost of fuel increases, it becomes more important to evaluate the savings which might be effected by increasing the steam temperatures in the power plant of the future. Because of the wide general interest in this problem, the A.S.M.E. Special Research Committee on High Temperature Steam Generation requested the authors to prepare a paper covering two points: (1) the probable thermal gains by the use of high-temperature steam in the regenerative cycle, and (2) the possible fuel savings that might result. The regenerative-steam-cycle heat-run gains that may be realized at temperatures up to 1,600 deg. F. are presented. These gains are calculated for a theoretical cycle and also for a practical cycle wherein such losses as extraction piping pressure drop, heater terminal temperature differences, etc., are considered. An economic evaluation includes such factors as fuel costs, load characteristics, auxiliary power requirements, boiler efficiencies and annual fixed charges. A method of comparing the heat-rate gains due to higher steam temperatures with those made possible by resuperheating is provided. It is found that the practical cycle shows less improvement with increasing pressure, and more improvement with increasing temperature, than the theoretical cycle. This relationship results from the fact that the practical turbine loses in efficiency with increasing pressure, and gains in efficiency with increasing temperature, while the theoretical turbine is assumed to be always 100 per cent efficient. These trends are not at all new; the only novel feature is the extrapolation of the trends to show what economies may be expected at temperatures up to 1,600 deg. F. and pressures up to 3,000lb. per sq. in. abs. Use of steam temperatures in power plants above the present maximum of 1,050 deg. F. will depend on the development of suitable materials for the higher temperatures, and on the design of acceptable equipment, particularly boilers and turbines, at a cost commensurate with the thermal gains to be realized. Since the costs of such materials and equipment are not presently available, this paper has presented only the total investment which could be justified on the basis of the thermal gains expected at the higher steam conditions.—*Paper by P. H. Knowlton and R. W. Hartwell, Jr., read at the 1949 A.S.M.E. Annual Meeting; Paper No. 49-A-33.*

Modernization of the Michigan Tank

The Experimental Naval Tank at the University of Michigan was built in 1904 to serve as a laboratory for the Department of Naval Architecture and Marine Engineering. It was the second tank to be constructed in the United States and was 300ft. long by 22ft. wide at the water surface, by 10ft. deep. With the recent addition of 60ft. to its length it still remains the largest tank on the American Continent owned by an agency other than the U.S. Government. There is a movable false bottom which can be lowered on threaded rods during deep water tests to 7 feet or raised to the water surface for shoal water tests. This bottom extends from the starting end of the test section for a distance of 133 feet. The remaining portion of the test section is of full depth so that it is possible to obtain both shoal and deep water resistances during the same test run. The false bottom does not extend to the tank sides, thus permitting water circulation around the edges as the model passes over. It is believed that this feature contributes greatly to the ability of the tank to predict full-scale performance accurately in shoal water One of the difficulties most prevalent in model basin work is that of producing turbulent flow over all or nearly all of the model surface in order that the full scale resistance can be predicted accurately. The tank has used a needle spray for several years to condition the water so that turbulent flow will be easier to produce. There is a spray pipe across the front of the car in which are tapped 15 nozzles 0·106 inch diameter. A 20 gal. per min. centrifugal pump supplies water at 20lb. per sq. in. in the spray pipe just ahead of the nozzles. The nozzles are directed vertically into the water and the jets may be felt to a depth of about 15 inch. This equipment is operated on the backtrip only, being idle during the actual resistance tests. This is effective in removing stagnant dust layers which may have some influence on test results. The scum is forced over the beach to the gutter at the north end of the tank and thus cannot work its way back over the surface.—*C. W. Spooner, Jr., Marine Engineering and Shipping Review, Vol. 55, January 1950, pp. 42-46.*

New Propeller Dynamometer

A new dynamometer for tank-testing of model propellers has recently been installed at the Experimental Towing Tank of Stevens Institute of Technology, Hoboken, N.J. The new dynamometer incorporates electrical resistance strain gauges to measure torque and thrust with minimum movement of the propeller relative to the hull model. All readings are taken on dials at a central control point, an arrangement that is found to be a great improvement over the existing gravity-type dynamometer. Suitable for propellers from 3 to 6 inch in diameter, the dynamometer is driven synchronously by a three-phase motor controlled by a special General Electric "Thymotrol" variable-frequency generator set. The revolutions are recorded automatically at the control point. For multi-screw models, duplicate three-phase motors installed to drive the other shafts, run at exactly the same speed as the shaft on which measurements are being taken.—*Marine Engineering and Shipping Review, Vol. 55, January 1950, p. 78.*

Self-bailing Self-righting Lifeboat

A new motor lifeboat with the unusual features of being able to right itself and bail itself out if capsized in a heavy sea, recently underwent tests given by the U.S. Coast Guard at Curtis Bay, Md. The tests effectively demonstrated that, even if the new boat was turned completely up-side-down, it would roll to an upright position, drain itself dry through baling scuppers and be under way again within a matter of seconds. The craft's power plant continued to function normally during all stages of the operations. The boat was designed by Coast Guard Headquarters and built in 1948. It is 36ft. 8in. in length, has a beam measurement of 10ft. 1in. and draws 3ft. 3in. of water, light. Power for propulsion is furnished by a 4-cylinder General Motors Diesel engine with hydraulically actuated reverse gear. Drive to the 26-inch by 21-inch propeller is through a 2 to 1 reduction gear. At 8 knots cruising speed the engine turns at 1,500 r.p.m. and consumes 5·5 gals. of fuel per hour.—*Marine Engineering and Shipping Review, Vol. 55, January 1950, p. 40.*

Aluminium Dipping Process

The aluminium process is primarily applied for the purpose of the scaling of steel at moderately high temperatures. In this process the iron or steel surfaces are immersed in a bath of molten aluminium. The penetration of the aluminium into the iron as required for the achievement of an adherent coating will take place only if the iron surface has been completely freed from oxides. Moreover, the surface of the aluminium bath must also be free from an oxide layer, as such a layer would keep the liquid aluminium from coming into contact with the surface of the metal to be treated; and it is this requirement which makes the practical application of the liquid bath process most difficult. Immersion of the work surface in a molten salt bath has been found to provide an effective method for cleaning the surface of the work prior to the dipping process and various salt bath compositions have been proposed. In the Dellgren process the access of atmospheric oxygen at any stage of the aluminium dipping process is prevented by special means. Thus a protective atmosphere is provided during the preliminary salt bath treatment, during the transfer of the work piece into the bath and also during the cooling process. The protective atmosphere may be either hydrogen or towns gas. These precautions greatly complicate the execution of the process and may even make its application impossible where bulky and complicated metal surfaces are concerned; also, treating plants of this type are costly to install and maintain. The Dellgren process is said to be employed in Sweden with good results. In the treatment of steel tubing, an aluminium layer of some 0·0025 inch thickness is, it is reported, produced, and the brittleness of the aluminium layer can be minimized in this process by employing an aluminium bath containing small additions of zinc. A relatively recent process which is claimed to be meeting with considerable success in the United States is the Moeller process, which in this country is covered by a British Patent and in which the bath of molten aluminium is carried on top of a layer of molten salts, the liquid aluminium floating on top of the heavier salt bath. The process is carried out by first immersing the article to be coated in the bottom layer of molten salt for the purpose of deoxidizing the surface and heating the article to proper process temperature. After that, the article is raised into the layer of molten aluminium where the surface receives its coat of aluminium. The underlying principle of the process is to deoxidize the iron surface and then to expose it to the molten aluminium without intermediate exposure to any kind of atmosphere.—*Engineering and Boiler House Review, Vol. 65, January 1950, pp. 13-15, 25*

Stress Corrosion Tests

The paper describes a series of tests carried out using an apparatus specially designed for stress-corrosion investigations at high temperatures. The specimens, which were of hollow form, were inserted into the base of an autoclave and loaded by means of a lever system, through a push rod inserted in the specimen. This arrangement permitted the production of accurate notch-forms on the external surface in contact with the sodium hydroxide solutions. It was found possible to produce failure of notched specimens rapidly and consistently, but homogeneously stressed specimens were immune. The fractures were inter-crystalline and typical of those occurring in practice. Dilute solutions, though not entirely impotent, were found to be very much less effective than those of high concentration. The presence (or absence) of silica in the solutions exerted no appreciable influence. Neither the addition of tannin nor the use of fine-grain steels was completely effective in preventing intercrystalline failure, though in the case of the latter, increased resistance was noted in some cases. Cathodic polarization was found to be protective while anodic polarization did not prevent, and possibly hastened, failure.—*Paper by C. D. Weir, submitted to The Institution of Mechanical Engineers for written discussion, 1950.*

Resin Coating for Metal

It is announced in the United States that a method of making tetra-fluoroethylene resin adhere to metals and a process for preparing the finish in a sprayable form have been discovered. The developments are expected to be of considerable interest to the electrical and chemical industries. The material may be used for corrosion-resistant linings for chemical tanks, as an insulating material for wire in electric motors, transformers, and for coating fabrics made of glass fibre. The product is claimed to resist attack by almost every chemical (but not molten alkali-metals) up to 500 deg. F. Its heat resistance is said to permit the production of higher capacity electric motors without any substantial increase in size.—*Petroleum, Vol. 13, January 1950, p. 19.*

Examination of Surfaces

Permanent and accurate three dimensional records of surface finish and damage can now be made. The process is simple and can be used by inexperienced personnel without weighing or heating equipment. The record is made by casting on to the surface under study a thermosetting resin which hardens in approximately twenty minutes after the addition of a catalyst. The resin and catalyst are preweighed and packaged in small containers ready for immediate use. External heat and pressure are not required. After the resin hardens the casting is removed for study. The negative replica may be tested with a profilometer or similar surface measuring instrument without damage to the casting. Gear teeth, shafts, bearings, journals, headers, valves, cylinders, rolls, screw threads, machine ways and other parts may be inspected and facsimile records made by use of resin casting. This method of recording surface conditions is considered an invaluable tool for research, test, development, manufacturing and inspection facilities.—*J. W. Sawyer, Journal of the American Society of Naval Engineers, Vol. 61, November 1949, pp. 819-827.*

Double-acting Two-stroke Engine Installation

The m.s. *Sirius* built in Holland for the Finska Angfartygs A.B. is fitted with a double-acting two-stroke Sulzer engine, of which few have been built in recent years for marine service, especially of the size in question. The engine of the *Sirius* has seven cylinders with a diameter of 530 mm. the piston stroke being 800 mm. The output is 3,500 b.h.p. at 180 r.p.m. giving the vessel a speed of 16 knots. The deadweight capacity of the vessel is 4,000 tons.—*The Motor Ship, Vol. 30, January 1950, p. 422.*

Small Geared Diesel Ship

The *Knut Jarl* which has been built for the North Sea and coastal trade of Det Nordenfjeldske D/S by James Lamont and Co., Ltd., Greenock, is a vessel of 235 feet overall length with a moulded breadth of 35 feet, and a gross register of 1,026 tons. She is equipped with a Swedish-built Nohab two-stroke eight-cylinder engine, developing 1,440 b.h.p. at 300 r.p.m., and driving the propeller at 140 r.p.m. through a reduction gearbox. Three 60 kW. Diesel engined dynamos are installed, one of them driving a two-stage air compressor through a friction clutch. The engine room auxiliaries are electrically driven; as are the winches on deck and the anchor windlass. The designed speed of the vessel when fully loaded is 12½ knots and on trials a speed of 13¾ knots was attained.—*The Motor Ship, Vol. 30, January 1950, p. 411.*

Vibration of Piping Caused by Gas Pressure Pulsations

Practically all compressors and their associated piping will show evidence of certain dynamic forces which are the result of the reciprocating action of the compressor pistons. One of the most important of these is the presence of pulsative flow in the gas phase of the system. Accompanying these more or less rapid periodic vibrations in flow rate are corresponding periodic changes in pressure of the compressed fluid known as pressure pulses. Some of the energy contained in such pulses, which are propagated through pipes with approximately the speed of sound, may be converted to physical vibration of piping, associated equipment, and supporting structures, depending upon the amplitudes and frequencies of the sinusoidal components making up the pulse wave shape. This conversion is due to the presence of mechanically resonant components in the piping and associated structures sympathetic to the frequencies present in the pressure pulses. Acoustical resonance effects of chambers and pipe lengths involved in the flow path can be of considerable importance and actually often prevent the accurate measurement of the pulsation wave shape produced by the compressor action because of its masking influence. When using reciprocating compressors, it is practically impossible to separate the effects of compressor valve action and piping resonance. The resonance effects of piping and manifolding chambers can often produce economic loss in that they may alter the effective static-pressure conditions at the juncture of compressor valve and piping so that a real and undesirable increase in engine fuel consumption or horsepower requirement becomes necessary This is a standing wave condition whereby the reflected pressure wave or pulsation produces an increase in the average pressure at the outlet of the compressor cylinders at approximately the time when the valve opens to discharge another compressed "slug" of gas into the line. As might be expected, the resulting erratic valve action may also be recognized by an increase in the rate of wear with consequent rise in valve-replacement costs.—*R. C. Baird and I. C. Bechtold, Transactions A.S.M.E., Vol. 71, November 1949, pp. 989-995.*

Large French River Tug

The large river tug *Fréderic Mistral* of the Lloyd Rhenan has been lent to the Cie Nationale due Rhone for the purpose of carrying out tests with screw propelled tugs on the River Rhone. The *Fréderic Mistrel* was built in 1948 at the Normandy yard of the Soc. des Chantiers et Ateliers de St. Nazaire (Penhoet) and proceeded from Le Havre via Brest-Gibraltar-Port St. Louis-du-Rhone under her own power. Length between perpendiculars is 70 metres, breadth is 8·80 metres, draft is 1·25 metres, block coefficient is 0·694, and displacement is 535 tons. The vessel is propelled by four 4-stroke cycle engines of the S.G.C.M.-M.A.N. type of 8 cylinders, built by the Société Génerale de Constructions Mécaniques at La Courneuve. Cylinder bore is 300 mm. and stroke is 380 mm. Each engine develops 550 b.h.p. at 470 r.p.m., but is capable of developing 700 b.h.p. at 550 r.p.m. M.e.p. is 4·9 kg. per sq. cm. and piston speed is 5·95 metres per second. There is a reduction gear giving the screw propeller a speed of 290 r.p.m. with an engine speed of 470 r.p.m. Propeller diameter is 1·70 metres. Auxiliary equipment includes two 90 e.h.p. Diesels, each driving a three-phase alternator rated 75 kW. at 1,000 r.p.m. Three-phase alternating current was chosen because delivery of this type of machinery could be obtained more rapidly than of d.c. machinery. Also the weight of a.c. machinery is less than that of corresponding d.c. machinery. With a tractive effort of 16 tons the vessel maintains a speed of 18 km. per hour.—*G. Garric, Les Nouveautés Techniques Maritimes en 1949 (Special Issue of Journal de la Marine Marchande), pp. 167-174.*

Ventilated Thermal Insulation for Marine Gas Turbine Plant

The orthodox design of thermal insulation for power plant service is based primarily upon the objective of conserving the energy of the thermodynamic working substance. However, additional functions must be satisfied for many specialized installations. If a high temperature plant must be installed in a limited space, where operating personnel are in continuous attendance, as in a marine installation, a most important function is to reduce the heat leak into the machinery space. In conventional insulation structures the heat leak from the working substance is of the same magnitude as the heat leak to the machinery space. Consequently, minimizing one, will minimize the other. A comparison of steam turbine with gas turbine plants shows that the hot duct flow areas of the gas turbine plant tend to be 40 to 50 times greater than the corresponding areas in steam plants. The result is that hot-duct surface areas are of the order of seven times greater for a given length of duct. Further, the temperature difference to be insulated against is perhaps two to three times greater than that for a comparable steam plant. It is evident that the application of a conventional type of insulation to a gas turbine plant, of a thickness suitable for a steam plant, will result in a machinery space leak of 10 to 20 times that of the steam plant. Even with a greatly increased machinery space ventilation, a suitable thickness of a conventional insulation structure will be of the order of 15 to 25 inch. It is believed that the bulk and weight of insulation of this thickness will be a serious handicap in the adoption of a gas turbine as prime mover. The system proposed in this paper consists of a layer of primary insulation several inches in thickness adjacent to the hot surface. This layer controls the magnitude of the heat leak from the working substance. Next to this layer is a ventilated annulus, and outside of the ventilated annulus is layer of blanket insulation, ¼ to ½ inch thick, which, with the cooling air flowing through the annulus, serves to minimize the heat leak into the machinery space to the desired magnitude. Such a structure, with an overall thickness of 4 to 5 inch for a hot-wall temperature of 1,500 deg. F. will reduce the heat leak into the machinery space to substantially less than that of the conventional installation in a steam plant. Some preliminary test results of such an insulation structure are included in the paper.—*A. L. London and S. R. Garbett, Transactions A.S.M.E., Vol. 71, No. 7, 1949, pp. 817-824.*

Metals at High Temperature

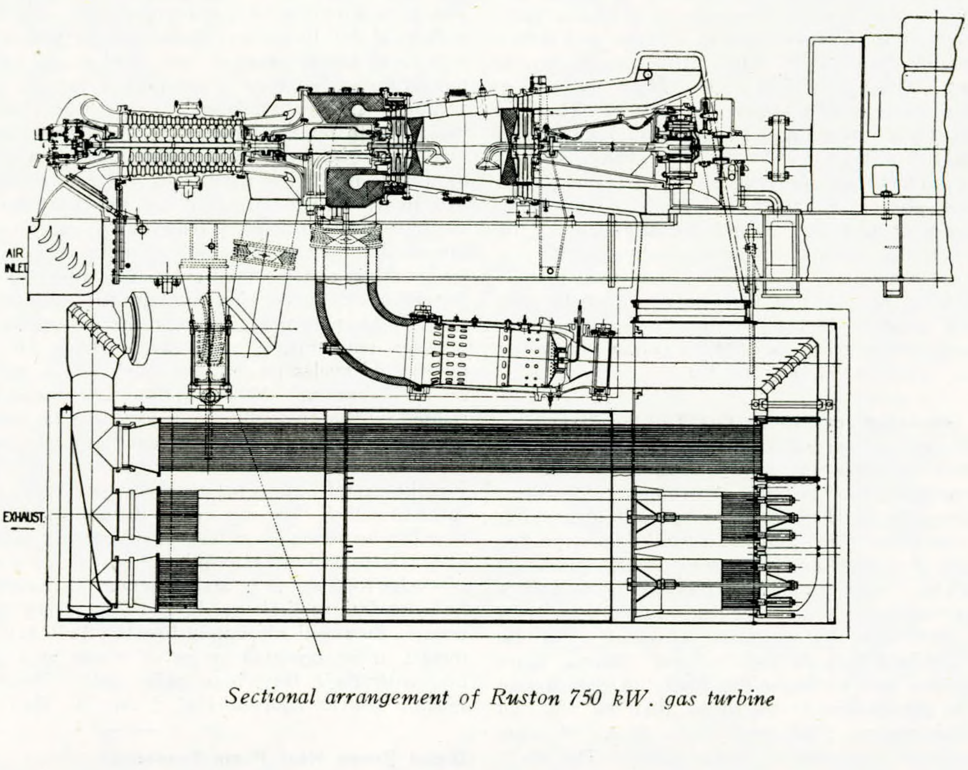
When metals ae subjected to repeated cycles of stress at high temperatures, their performance is affected by changes in section, surface finish and surface condition (i.e. the mechanical properties of the outer layer of metal and the presence or absence of defects in it) just as it is at ordinary temperature. Most knowledge about the fatigue of metals relates to steel a atmospheric temperature. Steel does not creep at this temperature, and the complications that are introduced when the behaviour of steel is considered under repeated cycles of stress at high temperature arise from the fact that the application of repeated cycles of stress to a metal that can creep have to be dealt with. If changes in section, surface finish and surface condition are ignored, the study of the fatigue of steel at atmospheric temperature is simplified by the fact that steel at atmospheric temperature has a definite fatigue limit in that a combination of mean-stress and range of stress that can be withstood for 10 million cycles, can be withstood for an indefinite time, and longer tests do not need to be carried out. At high temperatures, however, several other factors have to be taken into account. In the first instance, as the mechanical properties vary rapidly with temperature in the high temperature range, temperature has to be included as an important variable, and in addition, as time is in a sense equivalent to stress in the range of temperature in which a metal can creep, this has also to be considered as an important variable. Furthermore, under high temperature conditions components can fail by deforming too much as well as by breaking, and repeated cycles of stress can produce deformation as well as fracture. Consequently the study of the fatigue of steel at high temperatures involves the investigation of the relations between temperature, mean-stress range of stress, time and the number of cycles on the one hand, and deformation and fracture on the other, and although time and the number of cycles are related by the frequency of the cycles, the position with regard to the study of the fatigue of steel at high temperatures is much more complicated than that which arises at ordinary temperatures. Finally, metals subjected to repeated cycles of stress at temperatures at which they can creep, do not have definite fatigue limits that can be revealed in tests involving ten million cycles. If they have a fatigue limit it requires tests of some hundreds of millions of cycles to discover it, and in general it is best to assume that they have no fatigue limit at all, and that if any test is continued long enough they will ultimately break. This brings the behaviour of metals under repeated cycles of stress at high temperatures into line with their behaviour under static stress.—*J. M. Robertson, Heaton Works Journal, Vol. 5, No. 29, 1949, pp. 303-315.*

Diesel Driven Heat Pump Evaporator

In view of the increasing importance of the economic operation of all types of marine vessels, attention is drawn by the author to a new type of Diesel-driven heat pump evaporator which offers a method of distilling fresh or sea water more economically than by any other method. The type of evaporator described requires no chemical knowledge, can be operated by crews of cargo ship standard and, if necessary, can be completely automatic, possesses highly economic characteristics combined with ease of installation and maintenance, with minimum repair cost. The effluent from the evaporator may be used in complete confidence for high-pressure boilers, making the advantages of high-pressure steam available to the small ship. The fundamentals of heat pump evaporation are explained by the author. The numerical examples given include data on the expected performance of a Diesel driven evaporator plant distilling 10,000lb. per hour from raw sea water. Two types of vapour compressor are described, one being of the rotary design and the other a centrifugal machine with single stage impeller. The author gives analyses of raw water supplied to a heat pump evaporator and its distillate. Although the heat pump evaporator was operating under overload conditions 99·4 per cent of the original total solids was found to be removed. The removal of 98·2 per cent of silicon (SiO_2) suggests the elimination of encrustation of superheaters and turbine blading due to carry-over through these components with consequent improvement of boiler and turbine efficiency. The paper includes a diagram circuit of a plant of the type referred to.—*S. B. Jackson. Transaction of the Institution of Marine Engineers, Vol. 61, December 1949, pp. 225-230.*

Ruston Gas Turbine

The Ruston and Hornsby Gas Turbine (see Engineering Abstracts. December 1949, p. 124) has shown a measured fuel consumption at half full load and full load of 0·77lb. per b.h.p. per hr. and 0.59lb. per b.h.p. per hour respectively, these figures agreeing closely with the predicted thermal efficiencies at these loads. The consumption of lubricating oil is negligible. With a maximum gas temperature of 1,340 deg. F., a compression ratio of 4 : 1 and a heat exchanger of 75 per cent effectiveness, the full-load plant thermal efficiency is 24 per cent, and at 40 per cent load the efficiency is 18½ per cent. If desired, the same basis components may be arranged to run a non-recuperative gas turbine (i.e., without the heat exchanger) in which case the plant efficiency at full load would be 17½ per cent falli,ng to 11 per cent at 40 per cent full load. With the heat exchanger the maximum rated output is 1,070 b.h.p., and without the heat exchange the maximum rated output is increased to 1,250 b.h.p. at the gearbox output shaft. As shown in the accompanying drawing, the 13-stage axial compressor is driven by a turbine of built-up construction similar to that used for the compressor rotor. Relatively thin section material has been used for the turbine inlet ducting to allow the metal temperature to adjust itself rapidly to the prevailing gas temperature without the occurrence of large thermal stresses. This ducting is surrounded by lagging which prevents the outer casing supporting the turbine stator from being subjected to large temperature changes. This ensures accurate control of the position of the stator blades relative to those of the rotor. A circumferential belt has been cast round the compressor casing which is used for collecting air from the ninth stage of the stator blading. This air, being at suitable temperature and pressure, is used for cooling between the disks of the compressor-driving turbine, while the high pressure face of this turbine is cooled by air taken from the compressor discharge after passing through a small external cooler in which lubricating oil is used as the cooling medium. Measurements of the turbine disk temperatures immediately after running have shown that this cooling system is most effective in service. The heat exchanger —designed to house nine identical tube bundles which operate in parallel—are arranged in three rows, each of three bundles; each bundle is spaced so as to provide passages around their ends for the admission of the exhaust gas which passes outside the tubes. The design of a marine gas turbine with an output of 1,000 b.h.p. has been proposed. This unit has an installed weight, including the reverse reduction gear, of 25 tons. The thermal efficiency at the propeller shaft is given as 23 per cent, compression ratio 4 : 1 and propeller speed 120 r.p.m. The overall length, including reverse-reduction gear, is given as 22 feet, width overall 11ft. 6in., and height overall to the top of the heat exchanger 26ft. 6in., although an alternative arrangement of the heat exchangers which would appreciably reduce this height could be arranged.—*The Shipping World, Vol. 122, 18th January 1950, pp. 111-112.*



Sectional arrangement of Ruston 750 kW. gas turbine