ENGINEERING ABSTRACTS

Section 3. Shipbuilding and Marine Engineering

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Gas Turbine Using Two Types of Machine

In the gas turbine plant shown in Fig. 2, the gas first passes through Francis-type turbines, which operate the admission to the rotor wheels from the periphery in the radial direction and with axial exhaust. The succeeding turbine is an axial flow machine. The two types of turbine have independent shafts, so that one works at the highest temperatures and speed for which it is most suited, whereas the other operates satisfactorily at lower speeds. The working medium is drawn into the low-pressure compressor (2), passing thence to the high-pressure machine (6). The discharge is taken through a heat exchanger

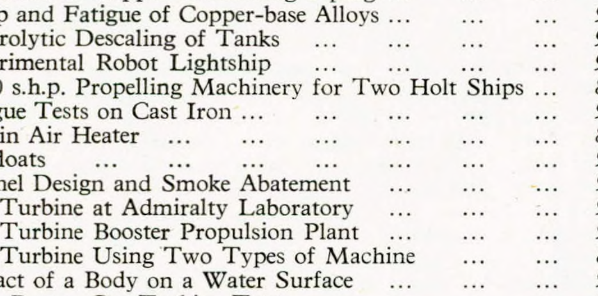


Fig. 2.

(8) where it is supplied with heat given off by the medium flowing from the turbine (17). From the heat exchanger, the gas flows to the combustion chamber (10), where it is further heated by fuel supplied through the burner (11). Thus, the working medium attains a high pressure and temperature, and is led to a Francis-type turbine (13) driving the compressor (6). The turbine (17) is an axial flow machine with a coupling (18) for driving a generator. The turbine (13) has a connection to the turbine (15) which drives the compressor (2). In the construction of the Francis-type turbine, the guide vanes (27) are adjustable and are hollow. The cavities in the vanes communicate with channels (32, 34) in the shafting (28), so that part of the cooling medium can be led to the space (33) between the walls of the turbine casing. Another part is led from the space (30) through channels (35) to the cavity (36) for cooling the rotor (24) and further through channels (37) to a space (38) for cooling the bearing (23).—*Brit. Patent No. 628,697, K. E. Hansen, Lyngby, Denmark. The Motor Ship, Vol. 31, April 1950, p. 39.*

Balanced-pressure Engine Indicator

Balanced-pressure indicators are used for obtaining indicator cards on reciprocating engines, compressors, pumps, and the like. The basis for this method of indication consists of a pick-up transducer containing a diaphragm exposed to the engine cylinder pressure (or manifold pressure, and so forth) on one side and an applied calibrated pressure on the other. (Fig. 1.) When the cylinder pressure exceeds the applied pressure, the diaphragm deforms to close an electrical circuit. Likewise, when the cylinder pressure diminishes the circuit is broken. An electrical circuit is required to convert the resulting pulses into a visible trace; this is accomplished by inducing a spark through spark-sensitive paper each time the diaphragm makes or breaks contact. A stylus, which comprises one side of a spark gap, is mounted on, or actuated by, a pressure-sensitive servo (such as a piston-cylinder or a Bourdon-tube arrangement), which is motivated by the same calibrated pressure as applied to the pick-up transducer. The stylus moves adjacent and parallel to a

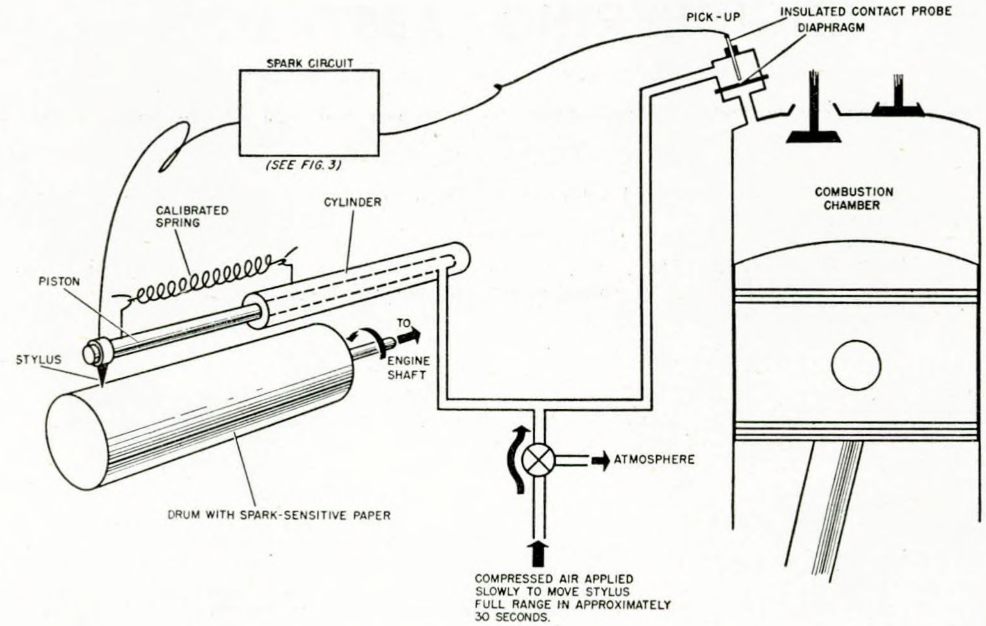


FIG. 1.

drum on which the sensitive paper is wrapped, the drum being coupled to the engine shaft to run at the same speed and in accurate index to the engine. Provisions are also made for obtaining top-dead-centre and constant-pressure lines on the diagram. Instruments of the type under discussion have been available commercially for a number of years. The spark circuits on these instruments, however, are such that it is necessary to make two passes of the stylus; once with the spark circuit set to function when the pick-up diaphragm makes contact, and once with the circuit set for break. Since approximately 30 sec. are required per pass, a total time of about one minute is required to obtain a card with a sufficient number of spark marks to make a clear line. In some instances it is important to obtain a diagram in a shorter period of time. In connexion with the author's work, the necessity arose for obtaining cards faster than commercially obtainable balanced-pressure instruments would permit. A new spark circuit, which is the subject of this paper, was therefore developed. The new circuit permits obtaining a diagram with one pass of the stylus, thereby reducing, by approximately one half, the time required to obtain a complete card. As in a previous model, the spark is produced by the discharge of a condenser through a thyratron and through the spark-transformer primary. However, in this unit two thyratrons are used, one being triggered by the closing of the pick-up contacts and the other by the opening of the contacts. The problem that dictated the use of two rather than one thyratron was not the circuit complexities, but the limitations subjected by the recovery time of a single firing condenser. At the peak of an indicator-card diagram, the pick-up contact closes and opens again instantly, allowing no time for firing, recovery, and refiring of a single thyratron and condenser. Even with the new indicator circuit there will remain occasions, such as studies of cyclic variations, where greater recording speed is required. For these applications, the oscillographic methods of indicating will be required.—*D. R. Church and D. K. Hart, Mechanical Engineering, Vol. 72, May 1950, pp. 389-391.*

Overloading of Propulsion Reduction Gear

In the modern turbine gear propulsion unit it is quite possible to overload the gear directly to a detrimental amount. There is an excess of nozzle area in the turbine inlet to provide for operation at reduced inlet steam pressures. If extra hand control valves are opened and the inlet pressure is maintained, outputs considerably in excess of the normal rated or even maximum rated are obtainable. In fact, for most installations the maximum output that can be obtained will be limited probably by the boilers' capacity to supply steam to the turbine rather than by the available turbine nozzle areas' inability to pass steam. Any operation at excess output will overload the gears and, if continued, may cause wear and damage to the gear teeth. Overloading of gears by operation of the turbine above rated power is, moreover, only part of the story, for the gear teeth transmit the power output of the turbine by means of a sustained torque, and it is the torque that determines the actual tooth loading and not the power alone. In other words, applying the power equation

$$\text{Power} = K \times \text{torque} \times \text{r.p.m.}$$

$$\text{or Torque} = K \times \frac{\text{power}}{\text{r.p.m.}}$$

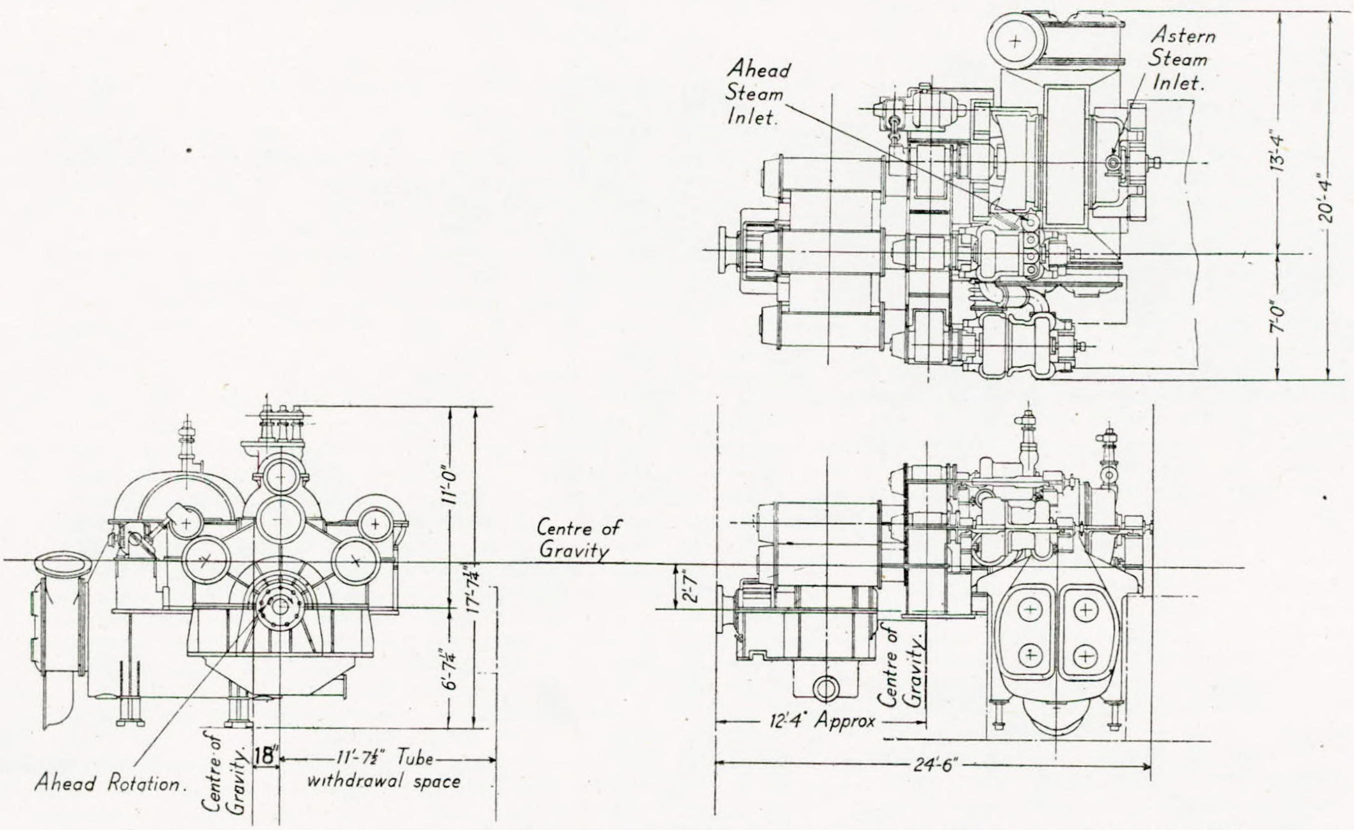
and the tooth loading is proportional directly to power and inversely to r.p.m. It is a well-known fact that for fixed input power to the propeller shaft of a vessel the designed revolutions per minute of the propeller will be decreased, if the propeller pitch is too great or if the vessel is slowed down by increased resistance to motion due to any of a number of causes such as deep draught, fouled hull, head winds and seas, or any combination of these. On one type of vessel on which draught tests were made, the revolutions per minute drop with constant power was about 10 per cent in going from the light draught condition to the heavy draught condition. Taking into account the additional drag due to bottom fouling and head winds and seas, it will be apparent that very appreciable drops in propeller revolutions per minute may occur. Any drop in revolutions per minute not accompanied by a proportionate drop in power results in a proportionate increase in torque. Considering that the steam turbine output is comparatively little affected by changes in speed, it will be appreciated that even with the normal power valve opening it is quite possible to overload a propulsion gear if the revolutions are down appreciably. If now the revolutions per minute increased arbitrarily the power required would also increase, not in proportion but appreciably more so with even further overloading of the gear then taking place. Thus the fallacy of operating a geared vessel on a revolutions basis. Revolutions per minute are not synonymous with power, although many operators have apparently disregarded this fact and as a result have overloaded propulsion gears, with wear then taking place. Good operation of propulsion gears requires that consideration be made of not only power output but also the revolutions per minute which occur. The criterion of power operation should be the gear torque factor of power/r.p.m., and continued operation above the maximum rating torque factor should not be done, bearing in mind that if operation at powers above rated is done, an additional increment of loading on the low-pressure side due to uneven load variation between the high-pressure and low-pressure turbine may also occur.—*R. J. Brown, Marine Engineering and Shipping Review, Vol. 55, May 1950, pp. 65-69.*

8,000 s.h.p. Propelling Machinery for Two Holt Ships

The main propelling machinery for two vessels for Alfred Holt and Co. now being built by the Caledon Shipbuilding and Engineering Co., Ltd., is being supplied by Metropolitan-Vickers Electrical Co., Ltd. Each of the two single-screw vessels will be propelled by geared marine turbines of the three-cylinder type, designed to operate with steam at a pressure of 600lb. per sq. in. and a temperature of 950 deg. F. and exhausting at 28½-inch of vacuum with cooling water at 75 deg. F. The maximum power of each set is 8,000 s.h.p. at 112 r.p.m. with an economical power of 7,250 s.h.p. The l.p. turbine houses the astern stages, which will develop 3,500 s.h.p. The top half high-pressure cylinder casing is of cast molybdenum vanadium steel, and contains a velocity of compounded stage followed by eight impulse stages. The rotor consists of a molybdenum steel forging with integral wheels and at the maximum output runs at 6,000 r.p.m. The intermediate-pressure cylinder is of cast carbon steel and houses 13 impulse stages; the rotor, which at maximum output runs at 4,500 r.p.m., also has the wheels forged integral with the shaft. The low-pressure cylinder is of fabricated construction and at the after end houses eight low-pressure ahead stages and at its forward end the astern stages, consisting of a velocity compounded stage and two impulse stages. The speed of the l.p. rotor will be 3,000 r.p.m. at maximum load. Unlike the h.p. and i.p. rotors, this rotor consists of separate forged wheels shrunk on and keyed to a forged steel shaft. The admission to the high-pressure cylinder will be by three nozzle groups, each controlled by an admission valve. One group alone carries the load to 5,000 s.h.p.· with the second in use the load reaches 7,250 s.h.p.; while with the three groups in service, the load of 8,000 s.h.p. is obtained. The astern turbine has only one nozzle group. The ahead and astern valves are controlled by two hand wheels mounted concentrically on the manœuvring platform. The blading is of stainless steel throughout, all blading and nozzles being fully machined except the diaphragm blades of the final low-pressure stages. The condenser has a fabricated shell and has a cooling surface of 6,170 sq. ft. The gears, which are also being made by Metropolitan-Vickers, are of the double-reduction type, the first reductions being carried in cases separate from the main final reduction case. In all instances the cases are of fabricated construction.—*The Shipping World, Vol. 122, 24th May 1950, p. 469.*

Lubricants for Steam Engines

Lubricating systems on reciprocating steam engines may be divided into two classes: (1) Those designed for the internal contact surfaces of pistons, cylinders, valves, and stuffing boxes. (2) Those designed for the external contact surfaces not enclosed within the parts of the engine exposed to steam. Any oil for internal lubrication must: (1) Blend thoroughly with the steam. (2) Adhere to metal surfaces under conditions of friction, high temperatures, pressure, and steam velocity. (3) Lubricate metal surfaces which must necessarily fit tightly to prevent steam leakage. With low-pressure units where steam condensation occurs in the cylinders, cylinder oils containing a small percentage of fatty oil had been found most effective. The fatty oil enhances the ability of the lubricant to adhere to the metal surfaces and resists the washing action of condensed steam. With higher pressure units where steam condensation does not occur in the cylinders straight mineral oils have been found to be satisfactory lubricants. External lubrication to reduce friction and wear to a minimum and to keep bearing temperatures within reasonable limits may be carried out by hand, drip-feed, wick-feed, gravity or forced-feed delivery of the oil,



General arrangement of the 8,000 s.h.p. steam turbine by the Metropolitan-Vickers Electrical Co., Ltd.

or a combination of these systems. For many years external moving parts of marine reciprocating steam engines were lubricated with mineral oils compounded with blown rapeseed oil in various percentages up to 20 per cent.—*Paper by D. D. Lewis, Bulletin, The Society of Naval Architects and Marine Engineers, Vol. 5, May 1950, pp. 12-13.*

Torsional Vibrations in Geared Turbine Plant

This paper is concerned specifically with the analysis of geared turbine marine propulsion drives. In such systems torque reversals at the gears accompanied by objectionable noise and wear can occur even though the corresponding vibratory shaft stresses do not result in failure. Thus, for satisfactory operation both vibratory stress and torque must be kept below certain limits over the entire operating range. A procedure is outlined in detail for obtaining a numerical solution of the problem. This makes use of a form which permits rapid solution of a problem by relatively inexperienced technical personnel. A typical geared-turbine system has seven masses and six natural frequencies, but in general only the lowest two or three frequencies have corresponding critical speeds within the operating range. The first three frequencies, however, can be estimated from simplified systems which are dynamically equivalent to the original. For the second and third modes, the seven-masses can be reduced to four by replacing each turbine, pinion, and first reduction gear by a single mass, and by combining all second reduction elements with the bull gear. For the fundamental frequency the estimate can be made by means of a two-mass system. Of the three lowest frequencies mentioned, it is possible to arrange matters so that at least one of these will be eliminated. This suppression of certain criticals is achieved by "tuning" and is of particular interest for marine drives of the type discussed. The average propulsion system has one to two criticals which occur below maximum speed. A complete analysis thus involves an estimate of vibratory shaft torques in the resonant zones in addition to a determination of the critical speeds. For this purpose excitation and damping torque values must be known. Both of these, insofar as the propeller is concerned, have been established experimentally for a variety of blade shapes and stern structures. For calculation purposes, values on the conservative side are generally adopted and refined estimates are resorted to only when inconvenient or prohibitive changes are indicated as necessary. Torsional vibration may be accompanied by a number of undesirable effects, but these generally can be avoided by proper analysis in the early stages of design.—*Paper by A. D. Andriola, Bulletin, The Society of Naval Architects and Marine Engineers, Vol. 5, May 1950, p. 21.*

Lengthening Valve Life

Mechanical means of augmenting valve life—such as valve rotation—merit serious consideration, today as never before. It is getting harder and harder to create new alloys to withstand the mechanical and thermal stresses of present-day engines. Metallurgy is no longer able to meet the problem as in the past. Rotating valves offer at least three important benefits. First, the stem will be free of deposits. This prevents sticking, a common cause of valve burning. Second, rotation imparts a light wiping action between the valve face and seat, making for cooler operating valve heads. Preventing such face deposits also precludes the flaking off of some portions, which would allow blowby and subsequent burning. Third advantage with valve rotation is that it prevents prolonged exposure of any sector of the valve face to a local hot spot on the seat. This produces lower and more uniform valve-face operating temperatures. Reducing peak valve temperatures substantially improves valve life. Among the secondary benefits of valve rotation are the prevention of stem scuffing. It distributes whatever lubricant there is around the valve stem diameter and prevents metal-to-metal contact, usually responsible for scuffing. Rotating the valve also cuts down undue tip wear and eliminates the grooving often found on non-rotating valves.—*A. L. Pomeroy, Journal, Society of Automotive Engineers, Vol. 58, April 1950, p. 70.*

Seven-cylinder Opposed Piston Diesel Engine

At the Greenock works of John G. Kincaid and Co., Ltd., a 7,600 b.h.p. marine Diesel engine is under construction for the Alfred Holt motorship *Bellerophon,* which will thus be the first ship to be fitted with a 7-cylinder engine of this type. The new engine is an opposed-piston, single-acting unit, the seven cylinders having a diameter of 750 mm., main piston stroke of 1,500 mm. and an exhaust-piston stroke of 500 mm. The maximum service rating is given as 7,600 b.h.p. at 110 r.p.m. A 6-cylinder engine of this type has been in service for more than a year now in the M.V. *Braeside,* built by Barclay, Curle and Co., Ltd., for Australian owners. The principal advantages claimed for this type of engine are greater simplicity and, since there are fewer moving parts, lower initial cost. No reduction in weight is claimed over its predecessors; in fact, the weight is believed to be slightly higher, but it is believed that this will be offset by fewer renewals and the ability to burn lower grades of fuel oil.—*The Shipping World, Vol. 122, 17th May 1950, p. 455.*

Boiler Fuels for Marine Diesel Engines

This paper gives the results of further experience at sea with the M.S. *Auricula,* including the burning of boiler fuels with a viscosity up to 3,000 secs. Red. I at 100 deg. F.; a comparison in cylinder liner and piston ring wear when burning Diesel and boiler fuels; materials of construction, with particular reference to the wear-resisting properties of various metals used for piston rings are discussed, as are the causes of cylinder liner wear, carbonization of fuel valve nozzles and the various matters which make for proper combustion of fuel. The following subjects are also dealt with: the essential properties for cylinder lubricating oils; methods to prevent contamination of crankcase oil; the cost to equip new and existing motorships to burn boiler fuels; heating facilities necessary to ensure the transfer of boiler fuels; and experience obtained in the burning of boiler fuels in opposed piston type engines.—*Paper by J. Lamb, Transactions, The Institute of Marine Engineers, Vol. 62, No. 6, 1950, p. 217.*

Fire in Air Heater

The Proceedings of the Merchant Marine Council, U.S. Coast Guard, report a recent case of a soot fire in the air preheater of a watertube boiler. The investigation showed that shortly before the occurrence of the fire, difficulty had been experienced in maintaining the water level. This difficulty was caused by trouble with the feed pump and necessitated putting the fires out and eventually resulted in loss of steam. There was an ample supply of water in the boiler not in use, and, therefore, this latter boiler was cut in while adjustments were being made to the feed pump. In the course of an hour or two these adjustments were made and it was decided to switch the auxiliaries back to the original boiler. Shortly thereafter a soot fire was discovered in this boiler's air heater. This fire resulted in extensive damage to the tubes in the air heater. On this ship it was part of the routine to blow tubes once a day while in port and they were last blown on the damaged boiler at 2 p.m. on the preceding day. It was also found that at the time of the casualty the air heater in the boiler concerned was being by-passed. It was concluded that the fire was caused by the ignition of an accumulation of soot deposits on the fire side of the air-heater tubes, and that these soot deposits were probably saturated with unburned particles of fuel oil which caught fire and in turn ignited the soot. As the air heater was being by-passed at the time of the fire it is evident that it was less able to resist action of the fire. This is another one of the expensive and damaging accidents caused by a failure to keep the fire side of the air-heater tubes clean. Although the chief engineer of the ship in question recommended that the tubes in the air heater be larger and spaced farther apart, in order to reduce the likelihood of soot accumulation, it is felt that the fact that many of these boilers are in successful operation with no trouble indicates that poor cleaning is the basic cause for this accident. Had the tubes been clean no fire would have taken place, and it therefore behoves all personnel in charge of boilers equipped with air heaters to make sure that the gas passages in such heaters are kept free of soot deposits. The fact that a soot blower is fitted does not necessarily mean that it is fitted correctly. Minor misalignment on the part of steam jets may reduce their cleaning effectiveness greatly. Therefore, engineering personnel should be sure that cleaning operations are doing the job for which they were intended. The cure for this type of accident is very simple—no soot—no air heater fires.—*Marine Engineering and Shipping Review, Vol. 55, May 1950, pp. 74-75.*

British Built Swedish Steam Tanker

The steam tanker *Soya-Christina,* built by Short Brothers, Ltd., Sunderland, for Rederi A/B Soya, of Stockholm, recently completed extensive and successful loaded trials. A speed of 13¼ knots was attained over the measured mile. The new vessel is the only large steam-driven tanker in Swedish registry, and the biggest steamship of any type under the Swedish flag. She has been chartered by the Anglo-Saxon Petroleum Co., Ltd., for five years. The *Soya-Christina,* with a length b.p. of 440 feet, breadth moulded 58ft. 9in., and depth moulded 33 feet, is designed to carry 10,300 tons d.w. on a draught of 26ft. 3in. The cargo tanks are divided into nine main centre tanks and ten side tanks for the carriage of bulk oil cargo. The main cargo tanks are of a special design arranged with two fore and aft bulkheads placed 41 feet apart, and represent a new departure in tanker construction which will enable the vessel to load a full deadweight cargo of lubricating and other heavy grades of oil in these main tanks. The vessel is also able to load a full deadweight cargo of light oil in the centre tanks with the addition of the wing tanks on port and starboard sides. All cargo tanks are fitted with heating coils. There are two pump rooms, one at each end of the bridge space dividing the cargo into three sections. The main machinery space is placed aft, and ample bunker capacity is arranged in the double bottom under the boilers, and in a cross bunker in the boiler room; a deep tank is also arranged under the dry cargo hold forward for the carriage of oil fuel or water ballast. The main cargo lines are on generous lines and arranged on the ring main system. Each tank is controlled by double shut-off valves. A Butterworth tank cleaning system is installed by which the structure inside the tanks is washed down with hot water under high pressure; a heater and pump are installed in the engine room for this purpose. The cargo tanks are fitted with a system for gas freeing together with steam fire-smothering lines. The main propelling machinery was installed by George Clark (1938),. Ltd., and consists of their latest reheat design with cylinders 26 inch by 41 inch by 73 inch with a stroke of 48 inch. Steam to the main engines is provided at 220lb. per sq. in. by three single-ended forced draught boilers 16ft. 6in. diameter by 11ft. 6in. long and arranged for oil burning only.—*The Shipping World, Vol. 122, 10th May 1950, p. 433.*

Shipbuilding in the U.S.A.

In his Annual Report to the American Bureau of Shipping, J. Lewis Luckenbach, retiring president who becomes chairman of the Board, gave a detailed analysis of the shipbuilding situation in the United States. Welding, in place of riveting, was developed in the United States to the point where the largest vessels are now practically all-welded, speeding up construction time by permitting the assembly of large sections, and reducing the total weight. The Bureau's requirements for Hull Steel which were adopted in 1947, as a result of the comprehensive investigations which had been carried on up to that time in connexion with failures in the larger types of welded vessels, have now become universally accepted in the steel making industry in the United States. Since the adoption of these new requirements there have been completed and are now in operation eleven large ocean-going tankers using steel in accordance with these new Rules. In addition, there are nine large ocean-going tankers in operation which were under contract at the time of the adoption of the new specifications and for which it was possible to change the steel orders to the extent that the improved steel was used for the principal structural parts. All of the large ocean-going merchant vessels at present under construction in United States shipyards are being built with steel in accordance with the new specifications and many repair yard orders are being filled with this steel. Samples of the steel obtained from the material as actually being supplied to the shipyards have been tested in the Laboratory of the Bureau, and these tests have indicated a very marked improvement in energy absorption values and transition temperatures over and above the values which were being obtained with steel produced in accordance with previous specifications. Corresponding tests are being carried out by the steel makers themselves and such preliminary information as has been released substantiates the results of the tests carried out by the Bureau. There has been a continuing trend in engineering design in new construction to the use of higher steam temperatures, 1,020 deg. F. for one group of ships, which necessitates more extensive use of alloy steels. A study disclosed that none of the recognized standards, such as the A.S.M.E. Boiler Code, A.S.M.E. Power Piping Code, the A.S.A. Standards for Valves and Fittings and the Welding Society Codes, covered installations for temperatures in this range and that while the codes were in the process of revision it could not be expected that the revisions would be completed in time for application to the installations now in hand. Therefore, a number of the leading metallurgists, welding experts and engineers who have been associated with high temperature installations in both central power stations and the oil refineries were invited by the Bureau to serve as an Advisory Panel. As a result of this activity, the Bureau has issued, in mimeograph form, an Interim Guide for the Installation of High Temperature Steam Piping on Ships, which is presently being followed for the new construction.—*The Welding Journal, New York, Vol. 29, April 1950, p. 342.*

Australia's Towing Tanks

Australia is now a shipbuilding as well as a seafaring nation, and needs a few ship model towing tanks including a major one at the Commonwealth Scientific and Industrial Research Organisation in Melbourne for naval and mercantile ship model tests, as well as small tanks at Sydney and Brisbane for the instruction and training of students and cadets of naval architecture, marine engineering and possibly aeronautical engineering. Ten years ago, when the new hydromechanical laboratories of the University of Sydney were being planned, provision was made for two tanks suitable for the towing of ship models. The smaller, or "junior" tank, is of welded steel, 60 feet long, 4 feet wide and 3 feet deep. It is provided with a small cable-driven truck running on rails suspended from the ceiling. Small towing dynamometers of various designs can be mounted on the truck for measuring the drags of the models. Two other methods of towing small models are being designed. The first is the falling weight type described by Wellenkamp in 1908 and used a century earlier by Beaufroy. In this form of dynamometer an endless loop of taut line or wire passes over a pulley at each end of the tank and the model is attached to a string wound round the axle of one of the pulleys. The second is formed by a looped wire stretched between two pulleys as before, but driven by a small winch. The drag measuring balance is, in this case, mounted either in the model or on a frame supported by the wire. The smaller tank will normally take models about 3 feet long, at speeds of about 3 feet per sec. Larger and smaller models, however, will be tried out in this tank at lower and higher speeds for the sake of obtaining data on the so-called "tank wall effect". The special balance will be called upon to measure very small drags in the neighbourhood of a pound. The flow of the water round the models will be "laminar" or " transitional", of very great experimental interest, especially for students. Design of other auxiliary apparatus for this tank, such as artificial beaches and a wave making machine is being considered. The larger or "senior" tank of reinforced concrete, is 175 feet long, 9 feet wide and 5 feet deep. It is expected that two types of towing apparatus will be designed for this tank. The first will be a larger edition of the wire loop type of the smaller tank, and will enable light mowing forces at high speeds or heavy forces at low speeds to be dealt with for a driving effort of a few horsepower. The second will be a light cable-driven truck with pneumatic rubber-tyred wheels which will support the main part of the weight of the trunk and its dynamometer and run on the concrete verge of the tank. The remaining small part of the weight will be supported on lubricated guiding skids bearing on accurately levelled rails which will prevent vertical or sideways movements of the truck. The rails will also be used if necessary to support a very light sledge which can be coupled to the cable system, in place of the wheeled truck, for high speed tests. The cable drive for this form of towing gear may absorb about 20 h.p., during the speed-up periods. Other possible items include wave traps and a wave-making machine.—*Professor G. G. McDonald, Commonwealth Engineer, Vol. 37, March 1950, pp. 308-309.*

Pametrada Progress

The survey published by The Parsons and Marine Engineering Turbine Research and Development Association, known as Pametrada, on the completion of their first five years' service, includes reference to the development work carried out in steam turbine design. Until 1945 the majority of designs were generally of the orthodox Parsons type, usually comprising three-casing reaction turbines driving through single-reduction gearing, or compound turbines in which the high-pressure casing comprises a two-row wheel followed by reaction blading with a low-pressure turbine of all-reaction type. Early in 1945 the design section began to develop its own designs, beginning with a high-pressure turbine of impulse type employing a solid rotor with disks formed by gashing out from the forging, together with all-machined diaphragms of built-up construction. The first contract for a vessel employing this type of machinery was placed in 1945. This vessel, the *Cortona,* built by R. and W. Hawthorn, Leslie and Co., Ltd., began her maiden voyage in December 1947 and has since been in continuous service. Ten vessels with machinery of a similar type are now in service and thirty-six more under construction. It is stated that the first design inquiry received by the Association, on 2nd January 1945, became a building contract. The machinery to this design was built by Vickers-Armstrongs, Ltd., Barrow, and installed in the P. and O. liners *Surat* and *Shillong.—The Shipping World, Vol. 122, 24th May 1950, p. 472.*

Experimental Robot Lightship

An experimental lightship which will have no crew and will be entirely controlled from shore is shortly to be ready for service in the United States. It will be stationed near Scotland Light Vessel, one of the three lightships in the approaches to New York Harbour, and will be controlled from the U.S. Coastguard Station at Sandy Hook. All equipment on board is duplicated, and in the case of the main generators triplicated, and change over is made automatically in the event of failure, and can also be controlled from on shore. The vessel has an electronic foghorn forward of the mast. It has no propelling machinery. Its cost of $375,000 compares with $750,000 for a normal lightship, and a total saving in annual expenses of $59,000 should be made.—*The Shipping World, Vol. 122, 10th May 1950, p. 435.*

Funnel Design and Smoke Abatement

In Part I of this paper, Mr. Ower explains the causes of the descent of smoke on the deck. It is shown that the trouble can be avoided either by increasing the height of the funnel, or by imparting enough upward momentum to the funnel gases to carry them into the smooth flow of air above the disturbed, eddying zone immediately over the deck and superstructure. Momentum can be given either by speeding up the funnel gases themselves or by blowing additional air out of the uptake; this air need not be passed through the boilers. The best shape of funnel is cylindrical, because the conventional streamlined casing is no longer streamlined in winds inclined at more than 10 deg. or so. Other possible methods of improving conditions are discussed, including a recently patented French funnel, which seems to be promising. In Part II Mr. Burge describes the experimental technique by which the problem is studied with models in the wind tunnel, and the results are discussed. One of the methods of prevention hitherto investigated on the model scale only consists of surrounding the gases from the uptake with an annular sheath of high-velocity air.—*Paper by E. Ower and C. H. Burge, read at a joint meeting of The Institution of Naval Architects and The Institute of Marine Engineers, 25th April 1950.*

Large Molasses Tankers

The Athel Line have recently taken delivery of two tankers of special type built to their orders by R. and W. Hawthorn, Leslie and Co., Ltd., of Hebburn-on-Tyne. While representing the most modern trends in tanker design, both in size and arrangement, the *Athelking* and *Athelmonarch* design contains many points of special interest. They are dual-purpose ships, designed for the carriage of molasses but suitable for chartering to oil companies for long periods. As molasses forms a very heavy cargo (26 cu. ft. per ton, compared with about 40 cu. ft. per ton with oil), considerable care has been taken in the structural arrangement. This has dictated the choice of the two-deck summer tank type, as it has been found, during the long experience of the owners, much more successful in resisting the various stresses set up by molasses than the more usual two-bulkhead type. A considerable weight of owners' extras has been embodied in the design at crucial points. All bulkhead connexions within the tanks are welded. The frames are riveted to the hull. The longitudinal frames stop at the transverse bulkheads, with brackets slotted through and welded. The girders are continuous. Points of interest in the general arrangement are the roomy pump room amidships, preventing an undue concentration of cargo when loaded, and the inclusion of the after pump room entrance in the poop which, together with a breakwater at the poop front, gives maximum shelter from the weather. A rotary pump is used for molasses only and, it is reported, has been found most successful in service. These two vessels, in common with other tankers built for the Athel Line since the war, have no telemotor, the steering gear control being all-electric, with three alternative systems from the bridge and one electric and one manual system aft. Principal particulars of the vessels are:—

Length overall	544 feet
Length b.p.	510 feet
Breadth moulded	67 feet
Depth moulded to upper deck ...	36 feet
Load draught	28ft. 9in.
Corresponding deadweight	15,630 tons
Cargo capacity (100 per cent full) ...	760,000 cu. ft.
Oil fuel capacity	1,383 tons
Gross tonnage	11,182 tons
Net tonnage	6,515 tons

The main propelling machinery, constructed in the St. Peter's Works of R. and W. Hawthorn, Leslie and Co., Ltd., consists of a Hawthorn-Doxford four-cylinder oil engine developing 4,450 b.h.p. at 112 r.p.m. in service. The cylinders are 670 mm. bore and the combined stroke is 2,320 mm. The scavenge pump between Nos. 2 and 3 cylinders is driven from the main crankshaft by means of a connecting rod and the forced lubrication pump, jacket and piston cooling pump, and sea water circulating pump are driven by means of levers from the scavenge pump crosshead. The standby pumps for these services are all steam driven. Steam for the molasses cargo pumps and steam auxiliaries is supplied by two cylindrical boilers working at a pressure of 180lb. per sq. in. and arranged for burning oil fuel with cold air forced draught. Each boiler is 15 feet in diameter and 12 feet long with three furnaces, the wing furnaces being arranged to take the exhaust from the main engines. At sea all steam for auxiliaries in normal use is obtained from this exhaust.—*The Shipping World, Vol. 122, 10th May 1950, pp. 427-430.*

Welded Aluminium Tripod Masts

This article describes the construction and installation of lightweight aluminium all-welded tripod masts aboard U.S. Navy radar picket ships at the San Francisco Naval Shipyard. The new type masts compensate for the increased amount of weight being added to the radar equipment aboard destroyer-type picket ships. To offset this increased weight on the mast, the new aluminium mast is constructed in the form of a tripod. With the addition of two horizontal braces, cable and other antenna wires, the new mast is capable of supporting the tremendous equipment load and at the same time withstand the terrific side stress brought on by a ship's roll and pitch. To make the main tripod leg, 10-foot lengths of ⅜-inch aluminium (standard aluminium alloy 61ST6) plate are rolled into 20-inch tubing and longitudinally welded. Next, the tubes are butt-welded together to make up the mast's 38-feet height. Twenty-foot lengths of ¼-inch drawn aluminium tubing (diameter 10¾-inch), butt-welded, comprise the two smaller (but longer) legs of the tripod mast. In all welding on the masts, welders utilize the shielded, inert gas electric arc. Two radar antenna platforms are attached as a unit to the tripod. Finally wire and other accessories are added with a coat of Navy grey over a primer undercoat of zinc chromate, and the mast is ready for installation by yard workers.—*Pacific Marine Review, Vol. 47, April 1950, pp. 29-30.*

Transition Temperature of Ship Plate in Notch-tensile Tests

During the past few years the Ship Structure Committee in America has sponsored a series of research projects in which notched, full-thickness specimens of ship plate were tested in tension. One of the principal goals of these efforts was to determine the minimum specimen width which would simulate the brittle-to-ductile transition characteristics of a "wide" specimen or, by inference, a welded ship. In these investigations, specimens whose widths ranged from 3 to 108 inch were tested, and it was concluded that a 12-inch wide specimen would provide a reliable index to the ductility-transition behaviour of large plates. Consequently, the 12-inch wide, notch-tensile specimen has become a standard of reference in the research programme of the Ship Structure Committee and currently is being used at Swarthmore College and at the David Taylor Model Basin for tests to evaluate the notch ductility of ship steel. However, since the 12-inch wide specimen is an expensive one and is too large to be broken in the testing machines available in most laboratories, it cannot be used for routine testing. It is desirable, therefore to establish the correlation between results from tests of this 12-inch wide reference specimen and results from tests of the various types of small specimens; to date, however, insufficient data exist to establish such correlation with certainty. The work reported in the present paper was planned with cognizance of this history of the 12-inch wide specimen and is part of the programme of wide-plate tests sponsored by the Ship Structure Committee. The objectives of this investigation were: (1) To increase the volume of data from tests of the 12-inch wide specimen, which data are useful for correlation with results of tests of various small specimens. To this end, six steels conforming to the old A.B.S. (American Bureau of Shipping) specification for hull steel, three steels conforming to the new A.B.S. specification for hull steel (Class B), and two modified A.B.S. Class B steels were tested. (2) To determine the transition temperature of steels removed from two fractured ships; in this phase of the work 12-inch wide specimens from six plates were tested. (3) To determine the effect on transition temperature, as determined by the 12-inch wide specimen, of adding aluminium to two otherwise typical A.B.S. Class B heats of steel. (4) To develop small, economical tensile specimens for determining transition temperature of steel; eleven of the steels which had been tested with the 12-inch wide specimen were used for this purpose.—*E. M. MacCutcheon, C. L. Pittiglio and R. H. Raring, The Welding Journal, New York, Vol. 29, April 1950, pp. 184-s-194-s.*

New Canadian Lighthouse and Buoy Tender

The new Canadian lighthouse and buoy tender *Alexander Mackenzie* was built at the yards of the Burrard Dry Dock Co. at North Vancouver. Principal dimensions and description are:—

Length overall	150 feet
Length b.p.	138 feet
Beam moulded	30 feet
Depth moulded to main deck	13ft. 6in.
Height of poop bridge and forecastle ...	7ft. 6in.
Displacement moulded at 10ft. 3in. mean draught salt water, approximately ...	720 tons
Draught, 11 feet aft; 9ft. 6in. forward.	
Engines, full Diesel; total horse-power, both engines	1,1000
Maximum propeller revolutions (approximately) per minute	250
Speed, maximum	13½ knots

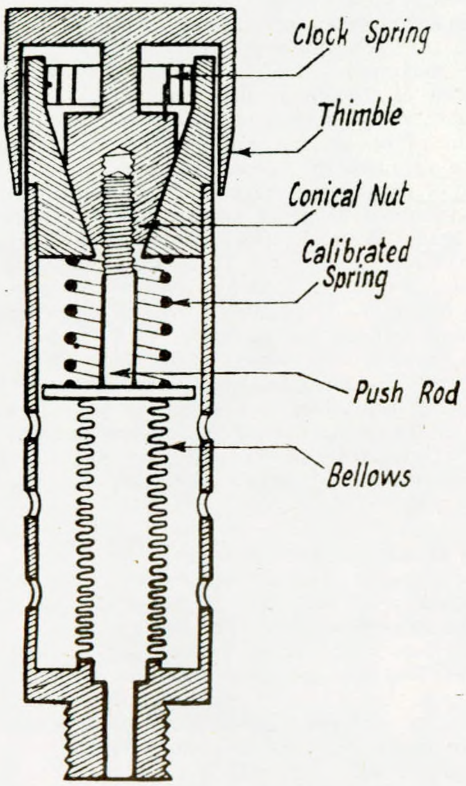
The foremast of the sturdy little "aids to navigation" vessel is rigged with a steel derrick of 10 tons safe working load; the arrangement of blocks and running and standing rigging is designed to facilitate the hoisting of both buoys and general cargo. All of the derrick's running rigging is led to the hoisting winch, controls of which are handled at a station on the forecastle head. A small steel mast fitted aft carries a steel boom of half-ton capacity for lifting stores, etc. The windlass is electrically driven by a 25-h.p. motor and is designed to handle the two 1,200-lb. anchors as well as warping operations. The main propelling engine consists of twin 10-cylinder, 9 inch × 12 inch Vivian engines of the pressure-charged non-reversing type. Each engine develops 500 b.h.p. at 500 r.p.m. continuously at rated speed. The engines are directly coupled by stub shafts to two-to-one S.L.M. oil-operated reverse reduction gears which have a thrust built integrally and are directly connected to the line shafting.—*Canadian Shipping, Vol. 21, April 1950, pp. 17-19.*

Lubrication of Ball and Roller Bearings

In the correctly designed and manufactured ball bearing, sliding friction is so reduced as to be of no real consequence either to life or performance. The balls or rollers in a bearing are in themselves the most perfect self-aligning members known; they require no guiding other than the groove or raceway in which they roll, and the function of the retainer or separator is, as the name implies, merely to space the balls or retain them in the bearing. If sliding friction is so reduced, then it might be asked why was lubrication necessary. Unfortunately sliding friction does occur between ball and separator, although it is only in the vicinity of the poles of the balls, where the surface speed is low. It is therefore essential to maintain a film of lubrication at these points if friction is to be efficiently reduced. The function of a lubricant in relation to ball and roller bearings may be classified as follows: (a) To reduce friction against adjacent parts; (b) to act as a heat absorber and carry away any heat developed in the bearing during running; (c) to act as a form of auxiliary seal, thus preventing moisture and dirt from reaching the bearing; (d) to act as an agent against corrosion; this has been found necessary, since under ordinary atmospheric conditions the highly polished surfaces soon corroded unless protected by a film of oil. Since anti-friction bearings do not employ a lubricant in the same manner as do plain bearings—i.e., as a medium for direct support of the load—it is evident that for a given period of operation they will not only require very much less lubricant, but that its application will also be less critical to bearing performance. Too few people realize that it is just as harmful to a bearing to over-lubricate as to under-lubricate it. Fluid friction is caused by movement of the lubricant. If a bearing housing is filled entirely with grease or oil, the lubricant is violently agitated by the revolving parts and excessive heating may result if the speed is high enough. For the correct operation of ball and roller bearing, the lubricant should be reduced so as to fill not more than one-half of the space around the bearing; this will give normal temperature running.—*T. B. Sansom, Petroleum, Vol. 13, May 1950, pp. 112-114.*

Combustion and Compression Pressure Indicator

A well-known British firm of engine indicator makers have recently put into production a new instrument for measuring the combustion and compression pressures of internal-combustion engines. It is known as the Dobbie McInnes peak-pressure indicator and is being made under licence from a Swiss concern. The indicator consists of an outer casing which is provided at its lower end with couplings or connecting nuts designed to fit any well-known engine indicator cock or valve. A bellows unit contained within the casing is secured at its upper end to a



Diagrammatic arrangement of the indicator

screwed push rod and at its lower end to the outer shell; cylinder pressure acts on the interior of the bellows. The screwed push rod is fitted at its upper end with a conical nut which is held in place on a tapered seat by means of a calibrated compression spring interposed between the lower face of the seating and upper face of the push rod flange. A graduated thimble is machined integrally with the conical nut and a pre-loaded clock spring housed within the upper part of the tapered seating tends to turn the nut when this is not firmly seated on its base. Referring to the diagrammatic sketch, engine pressure acts on the bellows and pushes upwards the spring loaded push rod with its conical nut. The pre-loaded clock spring winds the nut downwards again on to its seat. This is repeated over a few engine cycles until no further movement of the nut takes place and the push rod is locked by the nut in its uppermost position. The total angle through which the nut is turned is a measure of the vertical travel of the push rod, the graduations on the thimble having been arranged in a similar manner to those on an ordinary micrometer, and hence the maximum engine pressure applied to the bellows can be observed. The graduated scale on the thimble may be in any desired units, such as lb. per sq. in. or atmospheres.—*Gas and Oil Power, Vol. 45, April 1950, pp. 88, 97.*

Mechanical Engineering in Atomic Energy

"If the atomic energy industry, particularly in the power field, is to be successful, the mechanical engineering profession has a tremendous job ahead of it.. We must learn how to handle liquid metals at high temperatures. We must solve the problems of heat transfer rates which at the present time seem out of this world. We not only must learn how to build these atomic power plants at an economical figure, but we must also learn how to operate them economically and how to maintain them over long periods of time. The stress analysis work, the heat transfer work, and plain mechanical ingenuity required to pump liquid metal, to say nothing of the problem of how materials behave under intense radio-activity, are all major mechanical engineering problems. Those of us in the atomic energy business feel sure that these problems will be solved; but we know that it is no easy job, and that atomic power is certainly not just around the corner."—*R. S. Niblett, in a talk before the Machine Design Division of A.S.M.E., Washington, D.C., January 1950. Nucleonics, Vol. 6, April 1950, p. 47.*

Measuring Elongation of Pre-stressed Parts

With the pre-stressed parts shown in Fig. 4, the comparing element used for measuring elongation and calculating the stress is a tube for the conveyance of liquid, thus forming a permanent part of the mechanism. Referring to the lower diagram on the left, the section indicates a bolt (3) in which is a comparing element (1) in the form of a tube secured by a weld (2) at one end. A dial gauge (6) is fixed to a block (7) at the free end of the bolt and its feeler pin rests on the free end of the tube (1). Alterations in the length of the bolt (3) will thus be indicated on the gauge which, together with the block (7), is removed after the pre-stressing operation. The right-hand diagram shows the piston (8) of a double-acting oil engine fixed to a hollow piston rod (9). The cross-sectional area at the upper end of the rod is reduced, so that pre-stressing caused by tightening the nut (20) will produce a greater elongation. This is determined by the comparing element (21) which projects into the unreduced part of the rod. The cross-sectional area of the lower end of the rod is also reduced where it passes through a hole in the crosshead (23). This reduced part of the rod is pre-stressed by a nut (14) and a comparing element (15) is provided. When the piston rod ends are pre-stressed by the required amount, the end surfaces of the comparing elements lie flush. The pipe bracket (16) can be attached only

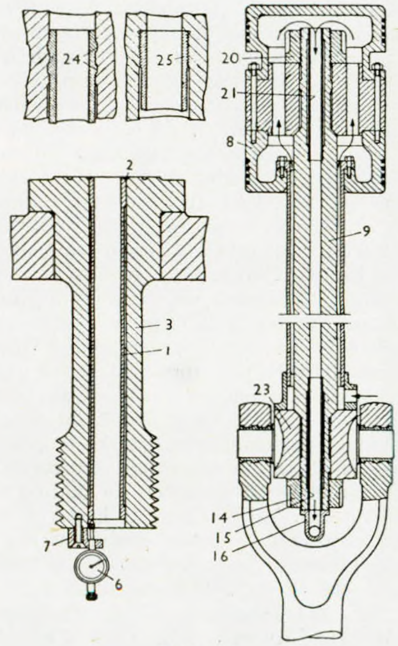


FIG. 4.

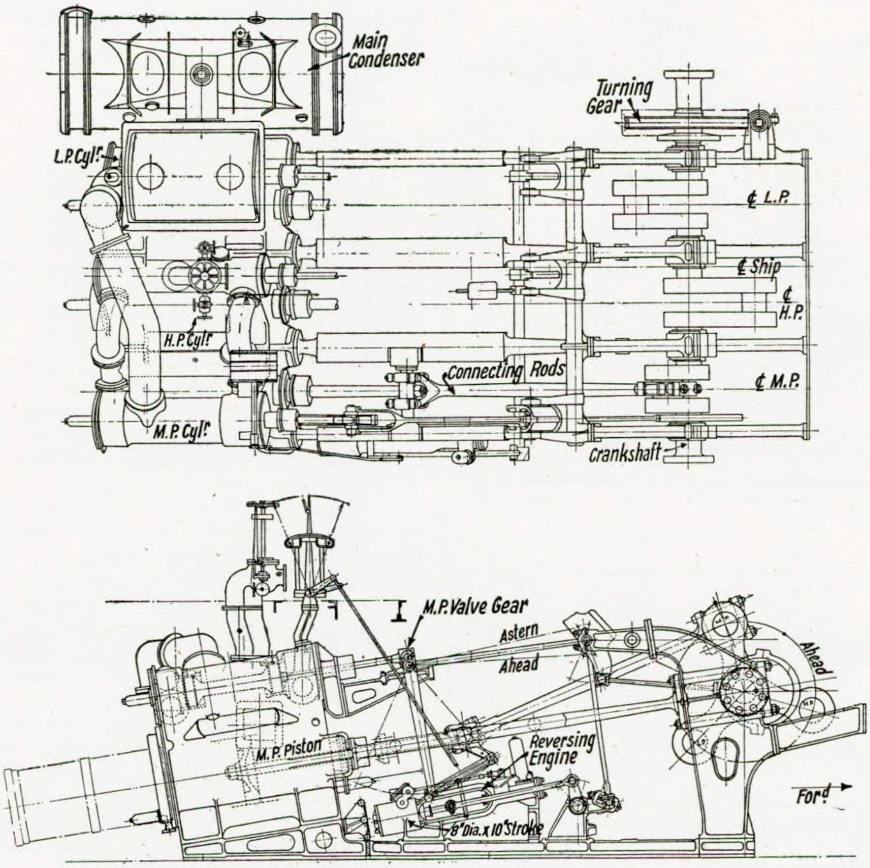
when the lower end of the rod has been pre-stressed to the correct amount. In the small diagrams two methods are shown of securing the tubes. In one case the tube is rolled into grooves (24), and with the other method a screwed attachment (25) is employed.—*Brit. Pat. No. 632,601, issued to Sulzer Frères, Winterthur, Switzerland. The Motor Ship, Vol. 31, May 1950, p. 79.*

Paddle Steamer for River Ganges

The paddle steamship *Yamuna,* built to the order of the High Commissioner for India on behalf of the Oudh-Tirhut Railway, has recently been completed by Yarrow and Co., Ltd., Scotstoun, Glasgow. The vessel has been built—after tank tests at Teddington—to the highest class of Lloyd's Register, complies with the latest Government of India regulations and has the following technical particulars:—

Length overall	225ft. 6in.
Length between perpendiculars	215ft. 0in.
Depth over paddle boxes	53ft. 6in.
Depth moulded	32ft. 0in.
Breadth moulded	10ft. 6in.
Draught loaded	4ft. 3in.
Displacement loaded	644 tons
Deadweight capacity	150 tons
Speed on trial	12 knots
Passenger accommodation	1,500 (total)

The propelling machinery, manufactured by the builders, consists of a three-crank triple-expansion diagonal paddle engine, having cylinders of 16 inch, 35½ inch and 41 inch diameter, with a stroke of 4ft. 6in. The valve gear is of the Brock single-eccentric type, in which part of the valve motion is derived from a link attached to the crosshead. The reversing gear weigh shaft is operated by a Brown's steam and hydraulic servo-cylinder controlled from the starting platform, at main deck level. The paddle wheels, 14ft. 3in. diameter over the floats, are of modern design. The wheels are fitted with eight elm floats, 9 feet long by 3ft. 3in. deep. Wooden floats are usual for river work, where they are likely to strike floating obstructions which might bend a steel float and perhaps cause damage to the shell plating. Steam is supplied by two marine type locomotive-type boilers designed for a working pressure of 185lb. per sq. in.—*The Marine Engineer and Naval Architect, Vol. 73, May 1950, pp. 193-196.*



Details of Yamuna's *main engine*

Torsional Vibrations

The forced and resonant-vibration characteristics of a torsional system are easily enough determined by the well-known Holzer methods, in the case where the only major damping influences available in the system are those of the prime-mover itself, or of the propeller, in the case of a propeller-propulsion system. Where a damper exists within the system, the classic solution confines itself to the determination of the resonant-vibration characteristics under optimum damping conditions, and utilizes the principle of energy balance in conjunction with an equivalent system. The problem of forced vibration in a system having a damper is not susceptible, however, to such simple treatment regardless of whether or not the damping is optimum. This paper outlines an exact method, whereby both the forced- and the resonant-vibration characteristics of a viscously damped system may be calculated for any existing damping condition, optimum or otherwise, and which utilizes the ever-popular Holzer form.—*T. W. Spaetgens, Journal of Applied Mechanics, Vol. 17, March 1950, pp. 59-63.*

Control of Distortion in Arc Welding

The author opens his paper with the statement that (1) when a piece of metal is heated it expands; (2) when a piece of metal cools it contracts; (3) when a piece of metal is heated it loses strength but gains ductility; (4) when a piece of metal is cooled it gains strength but loses ductility. These four statements are the fundamental rules which govern the control of distortion in welding. The problem is complicated, however, by the fact that in welding the material is not heated at a constant temperature throughout, but is subjected to very great heat concentrated at certain areas, the volume of heat bearing no definite relationship to the parts involved. It is thus impossible to produce mathematical formulæ which will cover the problem and it is necessary to fall back on general experience to devise methods enabling one to predetermine the nature and extent of the distortion produced. When a piece of metal is heated it expands. What happens if that piece of metal is prevented from expanding? If a metal bar is held in a strong vice and heat is applied, the tendency of the bar to expand is resisted by the vice. As the temperature of the bar rises its strength decreases and the force required to hold it bcomes less; in other words, the specimen yields by plastic

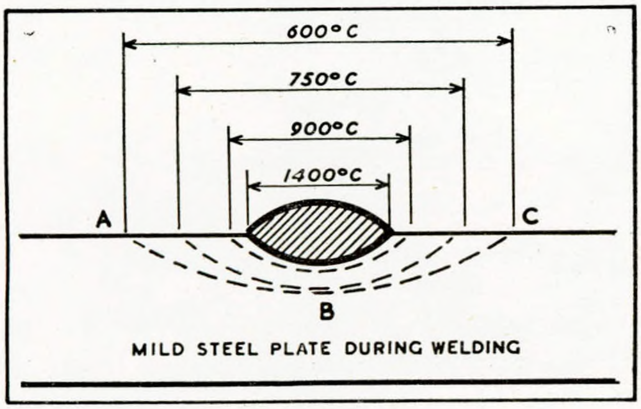


FIG. 1.

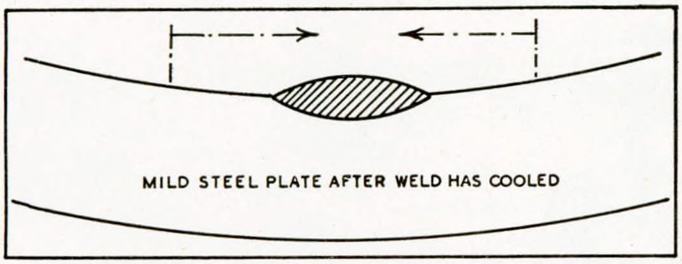


FIG. 2.

flow. When, however, it begins to cool, there being nothing to prevent its contraction, the specimen shrinks and falls out of the vice, when it will be appreciably shorter than before. Assuming that the vice is absolutely rigid and will not yield under the stresses imposed and that the bar is of sufficient proportions not to buckle, it is possible to predetermine the amount of contraction exactly, and to calculate the stresses imposed on the vice during heating. But the results have no practical value in the solution of the welding problem. It is, however, possible to obtain some insight into the probable behaviour of a weldment by considering the weld and the adjacent heat affected zone as the bar, and the surrounding parent metal as the vice. In this case the vice may not be sufficiently rigid to withstand the stresses, the bar may not be proportioned to resist buckling, and it certainly will not be heated to the same temperature throughout its length. If a bead of weld metal is deposited on a piece of steel plate, the plate will be heated locally somewhat on the lines indicated in Fig. 1. That portion of the plate within the line A.B.C. will be heated to the plastic range, but will be restrained from expansion by the surrounding cold area. When it cools and starts to contract, it will produce a state of strain in the parent metal, sufficient to pull the plate together and cause it to buckle. (Fig. 2.) The amount of buckle will depend upon the size of the bead, the thickness of the plate, the rate of deposition. For a given size of bead a weld laid down with an 8-gauge electrode at 140 amps. will produce far more distortion than a similar weld laid down with a ¼-inch diameter electrode at 320-330 amps. The reason is that with a slower rate of travel the heat-affected zone is wider than would be caused by a larger electrode with its greater speed of travel. "For a given size of weld, the larger the electrode that can be used, the less the distortion." This statement is made in spite of many assertions that it is not so in practice. The reason why it does not always come out right is that the welder tends to lay a heavier bead when using a bigger electrode. The operative requirement is that the weld bead size must be the same. So in setting out to correct distortion by using a bigger gauge electrode, it is necessary to see that the operative is trained to lay the correct size bead. *R. G. Braithwaite, Transactions, Institute of Welding, Vol. 13, April 1950, pp. 64-70.*

Fatigue Tests on Cast Iron

This report contains the results of fatigue and static tests on twenty cast irons, including nine pearlitic and acicular flake graphite irons, seven cerium-treated nodular irons, and two as-cast and annealed magnesium-treated nodular irons. Seven of the irons were used to study the notch sensitivity of flake and nodular graphite irons. The endurance ratio of both flake and nodular graphite irons tended to decrease with increasing strength, the ratio being greater for the nodular irons. The nodular irons were much more notch sensitive in fatigue than the flake graphite irons and were sometimes weaker than the flake graphite iron when a notch was present, in spite of their greater static strength. The results are compared with other published data on the fatigue properties of cast iron. The tests reported below were all carried out on the same rotating cantilever machine. Owing to difficulties encountered in the early tests, particularly in machining the test pieces, there were sometimes only six specimens for test, but experience has shown that to obtain a fatigue curve at least ten specimens should be tested. The actual stresses in a cast iron test piece in bending are known to deviate from the theoretical stresses, whether it be tested as a cantilever or a beam. For instance, a round transverse bar has an ultimate bending stress, as calculated from the elastic bending formula, of approximately twice the tensile stress determined in a direct tensile test. Just as the transverse rupture stress of a beam is much higher than the actual stress, so is the derived fatigue higher than the true fatigue stress. The considerations of theoretical and actual stresses in fatigue tests and the effect on the endurance ratio are discussed at the end of the report.—*J. W. Grant, Research Report No. 251, British Cast Iron Research Association, Journal of Research and Development, Vol. 3, April 1950, pp. 333-354.*

The New Metals—Molybdenum, Titanium and Zirconium

The commercially available materials are not always adequate for the severe high temperature or corrosive conditions which are encountered more and more in present-day technology. Progress in the development of the metals molybdenum, titanium and zirconium has been stimulated because of this need. Molybdenum and titanium are now obtainable in limited quantities and shapes, but zirconium research has not as yet advanced to the point of making that metal readily available. Techniques for producing and fabricating these metals are progressing rapidly, so that within the next few years they should take an important place in industry alongside the other so-called "special metals", such as the stainless steels, super-alloys, tantalum and others. Each of the new metals has properties that make it valuable for certain applications but, of course, each also has its limitations. In the table given in the article the common characteristics of molybdenum, titanium and zirconium are compared with those of nine other well-known metals. Molybdenum, which is available commercially, is particularly noteworthy because of superior strength at temperatures above 1,900 deg. F. as compared with the best present-day metals, and for its high modulus of elasticity. It is probable that molybdenum alloys of even more desirable properties will be announced before long. The increased availability of sheet, forgings and fairly massive parts of high molybdenum alloys will enable increased efficiency of many processes at high temperatures to be achieved. A decided limitation of molybdenum, however, is the severe oxidation of the metal at red heat and above. Surface coatings that will overcome this drawback are being intensively studied and will probably provide a solution to the problem. Welding of molybdenum presents problems due to its high conductivity and melting point, and development work in this field is required. Titanium appears to have a future paralleling that of aluminium. It is more abundantly available in the earth's crust than many other commonly used metals. By alloying or cold working, its tensile strength can be increased to 150,000lb. per sq. in. or more. It is relatively light (about 67 per cent heavier than aluminium, but over 40 per cent lighter than the stainless steels. The metal is weldable and does not present any unusual fabrication problems. Corrosion resistance is excellent, approximately equal to that of the stainless steels, which makes titanium a highly promising material for industrial atmospheres and possibly for salt-air applications. Titanium development is proceeding rapidly and one large metal-producing company in the United States has announced the availability of limited quantities of titanium in sheet and other forms. Zirconium development appears to be in its early stages and while the metal is available for special small-scale uses, it is not being produced in commercial quantities at the present. The metal is 20 per cent lighter than plain carbon and stainless steels, but is about 40 per cent heavier than titanium. It is superior to titanium and the stainless steels in corrosion resistance to many media. Future applications appear excellent in view of its high strength, relatively light weight and excellent corrosion resistance, and these attractive properties should spur research and development looking to the production of zirconium on a commercial scale.—*Extracted from Tempil Topics, The Welding Journal, Vol. 29, April 1950, pp. 321-322.*

Powder Cutting of Stainless Steel

With the advent of powder cutting, the torch cutting of stainless steel is on an almost equal basis with the torch cutting of mild steel. There are two types of powder used, an iron of about 98 per cent. purity and sodium bicarbonate containing 1-1½ per cent calcium phosphate to prevent caking, both of which are injected into the flame. In the iron-powder method, two things occur which assist the cut, first, the oxidation of iron to iron oxide is an exothermic reaction and increases the flame temperature, and secondly, the iron oxide fluxes the chrome oxide allowing a fresh steel surface to be presented to the flame at all times. The sodium bicarbonate method is purely a fluxing operation, no additional temperature being generated. A modified method makes use of a mixture of nitrogen and hydrogen as the carrying medium for the iron powder. When using iron powder, for 4-inch slabs the speed will be 6 to 8 inch per minute, although 11 inch per minute has been reached. Powder usage is at the rate of 6 to 7oz. per minute of torch time, cutting oxygen up to 13·0 cu. ft. per minute, preheat oxygen, 0·9 cu. ft. per minute and acetylene 0·8 cu. ft. per minute. The width of kerf is approximately ¼ inch. These figures of course will vary with the operator, type of steel, size of cutting tip and gas pressures. The figures given would probably be average on a production basis. This process produces a good quality cut and it does not require any stationary preheat to start the cut. The process gives off brown fumes which are non-toxic and are not objectionable in well-ventilated rooms. One objectionable feature of this process is that there is a bead of fused powder and slag deposited on the top edge of the cut, sometimes extending over an inch back from the cut, which is very tenacious and difficult to remove. The equipment used for the sodium bicarbonate method, as in the case of the iron powder method, consists of a standard 3-hose tubular cutting torch using standard-sized tips. In this case the powder is introduced directly into the cutting-oxygen stream, at cutting pressures, in the powder dispenser. The powder dispenser is of the vibratory type. Some typical operating data for this method cutting regular 18-8 analysis show the powder usage to be 1·1oz. per minute of torch time (this powder costing 5/6 that of iron powder), cutting oxygen up to 15·0 cu. ft. per minute, preheat oxygen 2·5 cu. ft. per minute and acetylene 2·2 cu. ft. per minute. Average cutting speeds for a 2¾-inch section would be 6 to 11 inch per min. with corresponding lower speeds for heavier (4½-inch) sections. Both methods of cutting produce changes within the steel that must be considered. The cutting temperature is such that, except in carbide stabilized analyses (Cb and Ti), there is an intergranular carbide precipitation extending in approximately 0·020-0·025 inch from the cut surface. With subsequent welding and/or annealing this presents no problem. The author also reports that extending from the cut surface from 0·1 to 0·125 inch the typical austenitic irregular many-sided crystal has been changed to a columnar structure.—*C. W. Powell, The Welding Journal, New York, Vol. 29, April 1950, pp. 308-310.*

Clad Metal for Heat Resisting Applications

"Rosslyn Metal" is a copper core Inconel or stainless steel clad metal. The copper core imparts high thermal conductivity to the material, thus eliminating dangers from hot spots or heat zoning. The material may be formed with ease and all the common types of welding have been applied successfully although special techniques are required in some cases. The lateral conductivity of the material is demonstrated and information given on the high temperature physical properties of the material, forming and welding information, and oxidation and embrittlement characteristics.—*Index Aeronauticus, Vol. 6, May 1950, p. 49.*

Electrolytic Descaling of Tanks

Results of further experimental electrolytic descaling of cargo petrol and oil tanks, as conducted aboard the tanker U.S.S *Cimarron* at San Francisco Naval Shipyard, have been reported to the Navy's Bureau of Ships. The method consisted of supplying power from welding machines to an arrangement of anodes suspended in the tanks to be cleaned. In general, enough current was supplied to give a current density of 0·235 amps. per surface foot of tank area at a potential of 20-30 volts. The results indicated the method to be practical as well as economical. No dangerous hydrogen concentrations were observed. Use of mild steel anodes and sea water as the electrolytic fluid assured economy of materials for the tests. The vessel's cargo tank No. 2 (used for petrol) required two operations, using two types of anodes and two power setups. Results were judged "fairly good" after two treatments. Operation 1 used twelve straight, flat bar anodes, arranged with four anodes getting power from each of three welding machines. It showed clean surfaces parallel to the anode axes and in the vicinity of the anode only. Increasing the length and adding extra fins at the bottom of the anodes led to much better and more complete overall cleaning on Operation 2, which had three anodes powered by each of four machines. For operation 3 (the ship's No. 8 wing tanks, port and starboard) six anodes in each wing tank were divided with three anodes being powered on each of four machines. An important feature of the test was the proof that a soluble anode of mild steel, rather than an insoluble such as carbon, produces practically no hydrogen and no detectable chlorine. Some molecular hydrogen (an almost negligible amount) could be detected escaping through ½ inch standpipe vents threaded through holes in the deck. These were installed for three purposes: (1) point of entrance and support for cables suspending anodes, (2) combination vents and standpipes to assure a full tank of electrolyte, and (3) vent points to check for hydrogen every hour. With simple precautions, it is believed welding and burning could be carried on on the deck immediately above tanks being cleaned, if tanks are vented to open deck areas. The chemical deposits left on the tanks (flakes of calcium magnesium and sodium salts) would of necessity require special cleaning before use of tanks for carrying aviation or other light white petrol products. Wire brush cleaning or use of a stronger washdown than sea water after cleaning were recommended for this problem. The experimenters also recommended further study to learn if a period of short reversal of current near the end of the process would assist in removing the deposits without affecting the magnetic state of the ship's hull. Further recommendations for study of establishment of Navy ship descaling stations, and even for shipboard installations aboard moving vessels were made following the tests.—*Pacific Marine Review, Vol. 47, April 1950, pp. 36-37.*

Bend Test for Aluminium Sheets

Specifications for the thinner gauge light alloy sheets usually call for a bend test in place of an elongation determination as evidence of adequate ductility. The requirements are usually well within the capacity of the material, and in this article the author describes a free cone bend test which is capable of determining more accurately the limiting radius of bend which the material is able to withstand.—*W. Thompson, Metallurgia, Vol. 41, April 1950, pp. 341-345.*

Creep and Fatigue of Copper-base Alloys

At the request of the Brass and Copper Tube Association an investigation of the creep, fatigue and scaling properties of copper-base alloys suitable for use in gas turbine tubular heat exchangers has been initiated. The creep and fatigue properties are studied in the Association's laboratories; oxidation and scaling resistance will be determined in Professor Preece's laboratory at Durham University. In the first instance the creep properties of two commercial alloys at present in use for heat exchanger tubes have been determined at 450 deg. C. and 500 deg. C. The tests have been made on the materials in the "as drawn" and annealed states. In addition stress-rupture tests have been made at the same temperatures.—*British Non-ferrous Metals Research Association, Annual Report, 1950, p. 14..*

Gas Turbine Booster Propulsion Plant

In this paper, which deals with the prospects of gas turbines in naval applications, the authors make reference to the proposed scheme of a combined steam turbine-gas turbine propulsion plant of 30,000 h.p. in which the cruising power requirements would be covered by a 9,000 h.p. steam turbine plant. Fig. 2 shows the arrangement discussed, which, compared with a 100 per cent steam turbine plant, represents a net saving of about 260 tons, equal to 28 per cent reduction, in the weight of the propulsion plant. The length of the overall machinery space would be reduced, relative to the former steam plant, by more than 10 feet. The steam component included in this arrangement would consist of a cross-compound turbine with a single-flow low-pressure element having the astern wheels in its exhaust casing. This turbine could be designed to give the maximum possible economy at the 20-knot point without undue sacrifice either at 9,000 s.h.p. per shaft (which is the maximum power at which it would operate without gas turbines), or at 30,000 s.h.p. per shaft when operating in combination with the gas turbines. Only one boiler per shaft would need to be provided, since, in the event of a boiler derangement, the two steam plants could be cross-connected, or the gas turbines operated alone, if at least one of the five units per shaft were provided with a starting motor. Overheating of the steam turbine during operation with the gas turbines alone could be avoided by maintaining a vacuum in the condenser. The boilers, turbines, condensers, and auxiliaries are visualized as being conventional, non-experimental in nature, and having normal design factors. The reduction gears likewise would

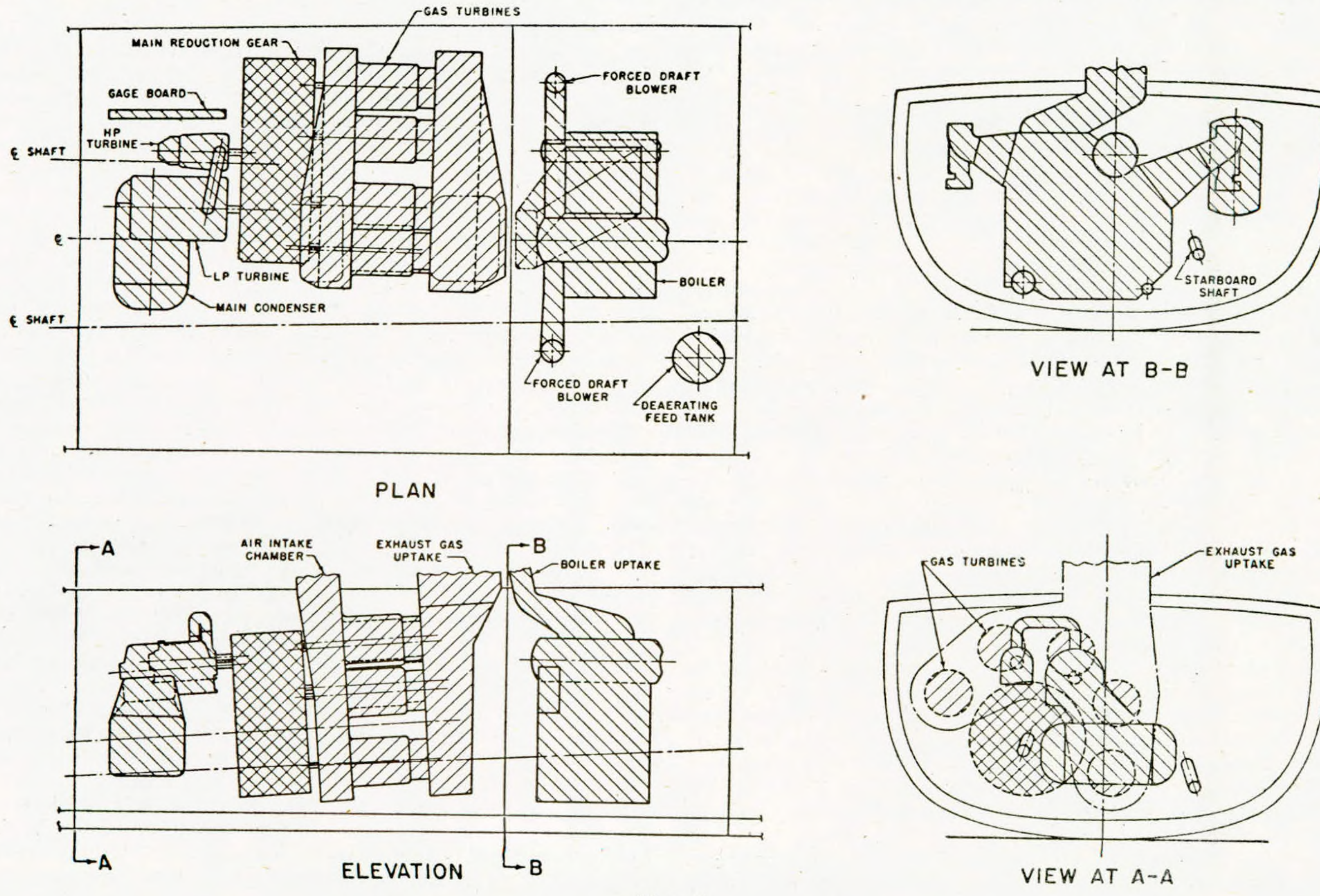


FIG. 2.

follow conventional design practice, except as to arrangement. Each gear would have five single-helical first-reduction pinions and gears and five double-helical second-reduction pinions spaced at 45 degrees around the 130-inch bull gear. Two of the high-speed pinions would transmit power from a gas turbine as well as from one element of the steam turbine. The other three pinions would transmit power only from gas turbines. A friction clutch of the Fawick "Airflex" type could be incorporated in each first-reduction gear in the gas-turbine power train. The five gas-turbine elements on each shaft would be interchangeable. They would be adaptations of the now familiar 500-hour open-cycle 2-shaft aircraft turbo-propeller engine. The compressors probably would be of the axial-flow type and have a 70-1 pressure ratio. Combustion is accomplished in the familiar can-type chambers clustered about the central axis. The turbine-inlet temperature would be 1,900 deg. F., giving a design fuel rate of 0·6lb. per h.p. hr. at full power. The gas turbines should burn the same fuel oil as that used in the steam boiler. The fuel economy of this combination steam-and-gas arrangement would be at least equal to that of comparable steam machinery at all speeds, and, at the most important points, that is, between 20 and 25 knots, it would be from 10 to 13 per cent better.—*R. T. Simpson and W. T. Sawyer, Marine Engineering and Shipping Review, Vol. 55, May 1950, pp. 52-56.*

John Brown Gas Turbine Tests

To supplement the information contained in his recent paper, Mr. J. B. Bucher of John Brown and Co., Ltd., has supplied details of further test work on the builder's small closed-cycle plant at Clydebank. A 120-hour continuous run has been completed, using a special 200-seconds oil having an ash content of about 0·04 per cent, of which more than 80 per cent was vanadium pentoxide. Laboratory tests performed some time ago at Clydebank and Sheffield suggested that vanadium might cause very heavy corrosion on austenitic steels at temperatures above 650 deg. C. (1,202 deg. F. and, accordingly, a fuel oil with an exceptionally high vanadium content in the ash was selected for further trials on the plant at Clydebank. During the whole of the 120-hour test, a small proportion of carbon tetrachloride was continuously injected into the combustion air stream; the underlying idea was that chlorides of vanadium might be formed, which would be gaseous at temperatures of 200 deg. C. (392 deg. F.) and above, and would consequently pass through the air heater without damaging the tubing. When the air heater was opened up, the tubing was found to be considerably corroded, particularly on the fire row tubes. An examination of the scale suggested that the corrosion was caused by (a) vanadium attack, (b) chlorine attack and (c) excessive temperatures. With regard to the vanadium and chlorine troubles, it would appear that, at the combustion-chamber temperatures of about 800 deg. C (1,472 deg. F.), any chlorides of vanadium which may have been formed broke down again to vanadium pentoxide and chlorine. The injected carbon tetrachloride, far from being advantageous, apparently increased the corrosion by adding chlorine to the combustion products.—*The Oil Engine and Gas Turbine, Vol. 18, May 1950, p. 13.*

Auxiliary Diesel Engines for Increasing Gas Turbine Compressor Output

With the arrangement illustrated in Fig. 1 it is proposed to employ the auxiliary Diesel engines in a ship's installation to augment the compressor with a gas turbine. The compressor, normally driven by the gas turbine, has an output limited to a load smaller than the maximum required. Thus when the capacity of the turbine-driven compressor (2) is insufficient, an additional supply is obtained through the pipe (9) from one or more of the compressors (10, 11, 12, 13) driven by the auxiliary Diesel engines (15, 16, 17) through clutches (21, 22, 23). At sea, the Diesel generators (18, 19, 20) will all be available to run at a reduced load, as current will not be

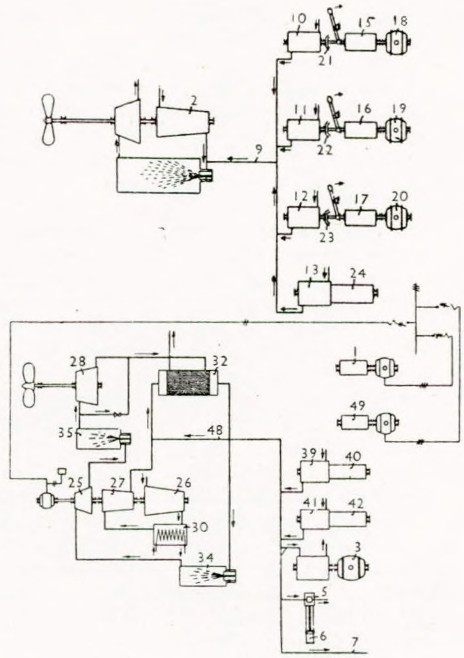


FIG. 1.

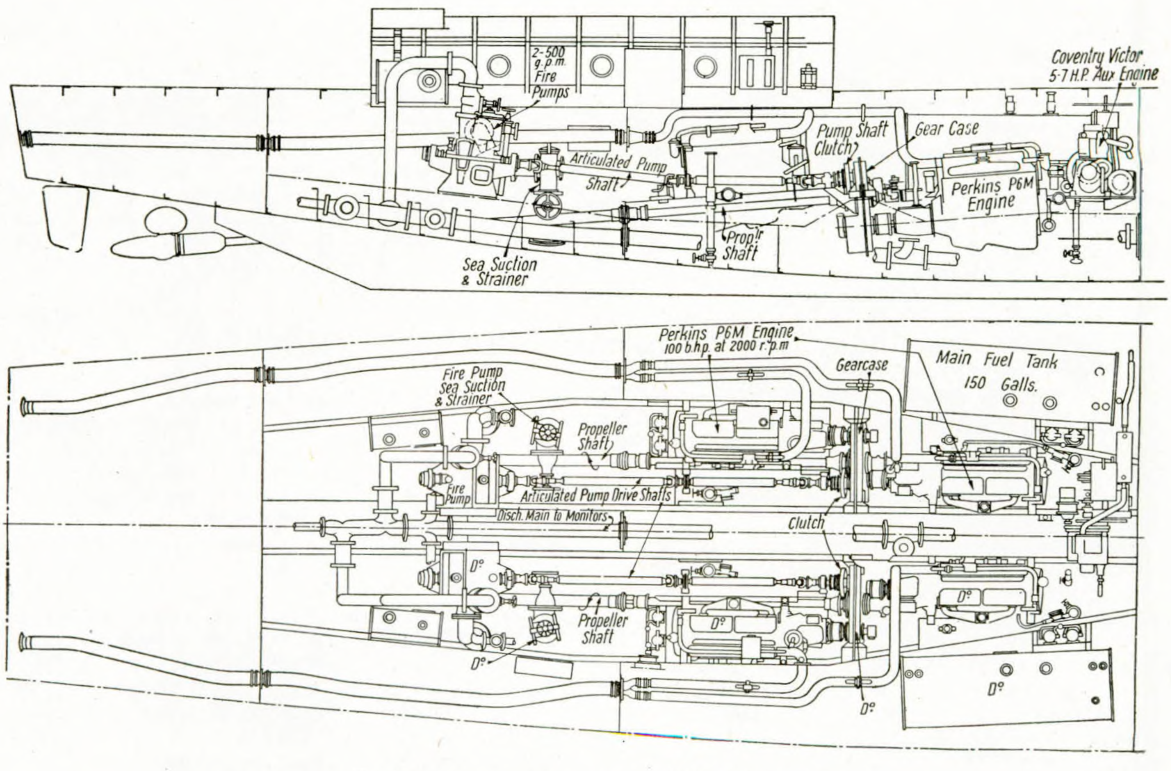
required for running the winches. It is proposed to install a fourth compressor (13) driven by a Diesel engine (24). With the arrangement shown in the lower diagram, compression and expansion occur in two stages and a heat exchanger is employed. The l.p. compressor (26) supplies the h.p. compressor (27) through an intercooler (30). The discharge from the compressor (27) is taken through the heat exchanger (32) and thence to the h.p. combustion chamber (34). The gas is supplied to the compressor turbine (25) and is reheated in the l.p. combustion chamber (35), finally being completely expanded in the propeller-driving turbine (28). The exhaust is led to the heat exchanger (32) and passes to the funnel, in the case of an open cycle arrangement. When required additional air is supplied by the compressors (39, 41) driven by the engines (40, 42). If these machines are pressure-gas producers, the air is mixed with the exhaust gases, so that the medium flowing to the gas turbine through the pipe (48) is pressure gas. Current for the ship's mains is supplied by Diesel generating sets (49, 1). The installation includes a gas turbine-driven dynamo (3) and a service pump (6) coupled to a gas-driven engine (5). The winches and steering gear in this installation are supplied with compressed gas through a pipe (7).—*British Patent No. 630,835, issued to Phillips' Gloeilampenfabrieken Eindhoven, Holland. The Motor Ship, Vol. 31, May 1950, p. 79.*

Gas Turbine at Admiralty Laboratory

The equipment at present in use at the Admiralty Engineering Laboratory consists of a Metropolitan Vickers *Gatric* engine, similar to that installed in M.G.B. 2009, and coupled to a dynamatic brake, a pressurized combustion-chamber testing rig, and apparatus for the gravimetric analysis of the combustion gases. Air for the combustion rig is supplied by two Rolls-Royce *Merlin* engines, driving *Griffon* superchargers, the maximum output of each of which is 5lb. per second at a pressure of $3\frac{1}{2}$ atmospheres, and by a Perkins Diesel engine driving a Roots blower, with an output of about $1\frac{1}{4}$-lb. per second at a pressure of 5lb. per sq. in. (gauge). Patternator equipment for testing fuel sprayers is in the course of installation. Additional equipment on order includes a number of turbo-blower units, which will be set up as self-supporting gas turbines for use in testing blade materials, and two machines for carrying out fatigue tests on single blades at high temperatures. The supply of air for the combustion rig and other purposes will be increased by the installation of a second Roots blower and a reciprocating compressor, both electrically driven. The latter will have a capacity of 0·64lb. per second at a pressure of 100lb. per sq. in. Facilities are also being provided for brake-testing small turbines, of up to 200 s.h.p. at speeds between 15,000 and 40,000 r.p.m. A transmission dynamometer, incorporating a 10 to 1 ratio epicyclic gear, is on order, and an immediate requirement is being met in part by the conversion of a spare *Merlin* supercharger to an air brake. Accurate measurement of rotational speed will be secured by the use of a five-decade crystal-gated electronic counter. Up to the end of 1949 the *Gatric* engine had been run for about 120 hours on Pool gas-oil. This period included runs to establish the characteristics of the engine, and to improve th accuracy of temperature and speed measurement; a series of acceleration tests, with extensive electronic recording of temperature, pressure, speed and fuel changes for guidance in the design of governor gear for a gas-turbine-driven alternator; and, finally, experiments in washing the compressor blades with various liquids, in attempts to recover the efficiency lost through deposition of atmospheric dirt. Running has recently recommenced after stripping and cleaning the engine and fitting a combustion chamber of new design. The future programme includes work with high-temperature strain gauges (to measure turbine-blade vibration while running) and operation with boiler fuel, the latter in extension of the work already carried out by the Metropolitan-Vickers Electrical Co., Ltd., on a similar engine. Up to the present, the combustion rig has been operated principally on kerosene, initially to develop satisfactory operating technique and instrumentation, and, latterly, to provide data regarding the rate of oxidation of various blade materials. These data are required for combustion with the behaviour of similar blade samples in the combustion products of boiler fuel.—*The Shipbuilder and Marine Engine-Builder, Vol. 57, May 1950, p. 372.*

Firefloats

Two firefloats of new design recently demonstrated their fire-fighting and manœuvring capabilities in Portsmouth harbour. The performance of the two craft is similar, but the machinery arrangement is different, the only point of "similarity" in this respect being that neither is fitted with orthodox propellers. In each vessel the firefighting equipment includes two specially designed monitors manufactured by the Pyrene Co., Ltd., and mounted on the centreline, one amidships and one aft; these are of the dual-purpose type and can be used for either water or foam. The capacity is 350 gals. per min. of water, or 1,800 gals. per min. of foam at a pressure of 130lb. per sq. in. There are also two smaller monitors of standard R.A.F. type mounted port and starboard side just forward of the break of the fo'c'sle. These fittings are also dual-purpose ones and can deliver 150 gals. per min. of water, or 900 gals. per min. of foam under the same working conditions. There are also two foam pourers, or spreaders, arranged in the side of the ship just forward of amidships and approximately one foot above the waterline. These are used to lay a blanket of foam on the surface of the water, which will act as a most effective barrier against the spread of a floating oil fire. Prototype No. 1, which has been given the number R.A.F. No. 80, has been built under contract in accordance with Air Staff requirements, and has the following principal dimensions: length overall, 55 feet; length on waterline, 52ft. 2in., beam moulded, 12ft. $6\frac{1}{2}$in.; depth moulded, 5ft. $4\frac{1}{8}$in.; draught aft loaded, 3ft. 9in. Four Perkins S6M Diesel engines provide power for pumping and propulsion and are so arranged that each pair drives a propeller shaft and also a fire-fighting pump. Each engine has six cylinders, $4\frac{3}{8}$-inch bore by 5 inch stroke, and is rated 100 b.h.p at 2,000 r.p.m. with a short-time overload capacity of 130 b.h.p. at 2,250 r.p.m The engine room arrangement of this craft is shown in the accompanying illustration. This vessel is fitted with Rotol variable pitch propellers. Firefloat No. 81 is slightly larger, with a hull constructed of aluminium alloy to British Standard Specification AW5 and has dimensions as follows: length overall, 60 feet; length on waterline, 56ft. 5in.; beam moulded, 14 feet; depth moulded, 6ft. 5in.; draught aft loaded, 3ft. 7in. In this vessel the propelling and pumping machinery are entirely separate, there



Engine room arrangement of "Firefloat 80" being two General Motors type 64 HN-5 Diesel engines, each fitted with a clutch giving neutral and ahead position and a 2·04 to 1 reduction gearbox. The output shafts from the gearboxes pass aft to the stern of the ship where they drive Voith Schneider propellers.—*The Marine Engineer and Naval Architect, Vol. 73, May 1950, pp. 179-185.*

Corrosion of Copper Circulating Piping

This paper describes some of the troubles which operators of C type ships are experiencing due to corrosion and wastage of copper circulating piping. Examples are presented to indicate the extent and frequency of the troubles encountered, the remedies tried, theories projected, etc. The cases related indicate that the maximum life of bare copper piping is from two to four years, and that holes appear frequently, generally during every voyage. The various types of repairs and replacements made are described. These include repairs using copper, tinned and solder or lead wiped, or sheet lead lined, or lead sprayed and coated with a bakelite type of varnish; also replacements using cupro-nickel, monel, and rubber-lined pipe. All operators from whom information was requested, have reported unsatisfactory experience with copper piping. It is concluded that materials more satisfactory for salt water service than copper must be used to avoid excessive maintenance.—*Paper by D. C. Macmillan, Bulletin, The Society of Naval Architects and Marine Engineers, Vol. 5, May 1950, pp. 15-16.*

Impact of a Body on a Water Surface

An approximate method presented, determines the pressure distribution during impact on the surface of a body which strikes a horizontal water surface at an arbitrary angle. The effect of the splash is neglected and the pressure on the free boundary is assumed proportional to the potential, as if the process were an impulse. The shape of the submerged portion of the striking body is approximated by a semi-ellipse (two dimensions), by a hemisphere, half an ellipsoid of revolution and half a general ellipsoid. Under those conditions, explicit results from the pressure distribution are found. As a special example, the impact pressures on a sphere striking at an angle of 45 deg. F. are computed in detail.—*L. Trilling, Journal of Applied Physics, Vol. 21, February 1950, pp. 161,-170.*

Speed of Rise of Air Bubbles in Liquids

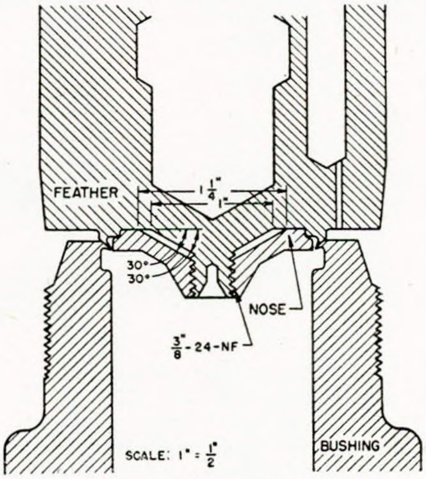
Technique and results of an experimental investigation of the speed of rise of air bubbles in liquids are presented. Classification of bubbles as "large" or "small" was made on the basis of their behaviour, the large bubbles becoming mushroom-shaped and the small bubbles remaining spherical. It was found possible to develop an empirical formula for the velocity of large bubbles dependent only upon the volume of the bubble and upon Laplaces's constant for the liquid, independent of viscosity. Contrary to expectations, the resistance of small bubbles in certain liquids was considerably less than that of solid spheres, a phenomenon which may be explained by the assumption of partial and continuous replacement of the boundary layer while the bubble is rising.—*T. Bryn, Applied Mechanics Review, Vol. 3, February 1950, Abstract No. 301.*

Optimum Screw Propulsion for Self-Propelled Barges

Increasing numbers of river and canal barges are motorised nowadays, and their cargo capacity and dimensions are growing steadily. Vessels capable of carrying 1,000 tons are under discussion, and it is also intended to use such large units for towing a number of dumb barges. A high power of about 1,000 h.p. may be required for the purpose, and this may confront the designer with difficult propulsion problems, since the screw diameter should hardly exceed 5 to 6 feet for river-going ships, even if the propeller is tunnelled. Otherwise, an unduly large proportion of the blade are may emerge in the light condition. The author illustrates the condition for four-bladed propellers of one of the Schaffran series in a curve of optimum efficiencies plotted against the pitch-diameter ratio and a modified, non-dimensional form of Taylor's power coefficient. He then discusses three different solutions of the problem of designing a screw of limited diameter that is capable of absorbing a high power: (1) A single propeller; (2) twin screws of normal design; and (3) twin screws displaced as far as possible outwards and behind the stern; this is a new proposal in order to obtain a low wake fraction and a low thrust-deduction coefficient. The author has plotted curves for the useful thrust and of the thrust per s.h.p. against the engine power for a maximum screw diameter of 5·6 feet, and also for the maximum revolutions of standard engines at that output.—*Völker, Schiff und Hafen, Vol. 2, February 1950, p. 33. Abstract No. 3628, Journal, The British Shipbuilding Research Association, Vol. 5, May 1950.*

Sealing of High-Pressure Steam Safety Valves

The essential elements of a safety valve, in so far as sealing is concerned, are the feather and the seat bushing. The seat bushing is simply a hollow tube connected to the boiler, with a carefully finished seat on the top surface, while the feather is essentially a cover with a similar seat held against the seat bushing by pressure from a spring. When the steam pressure on the bottom of the feather extends the force of the spring, the design of the valve is such that the feather pops up and steam is released until the pressure drops to a predetermined value. At that point the feather is forced down by the spring, and the valve is again closed. A fundamental investigation of sealing with high pressure steam safety valves has shown that poor sealing is a result of self-induced growth of tiny initial leaks. Expansion of the leaking steam cools local areas of the valve seat, causing contraction of the seating surfaces in

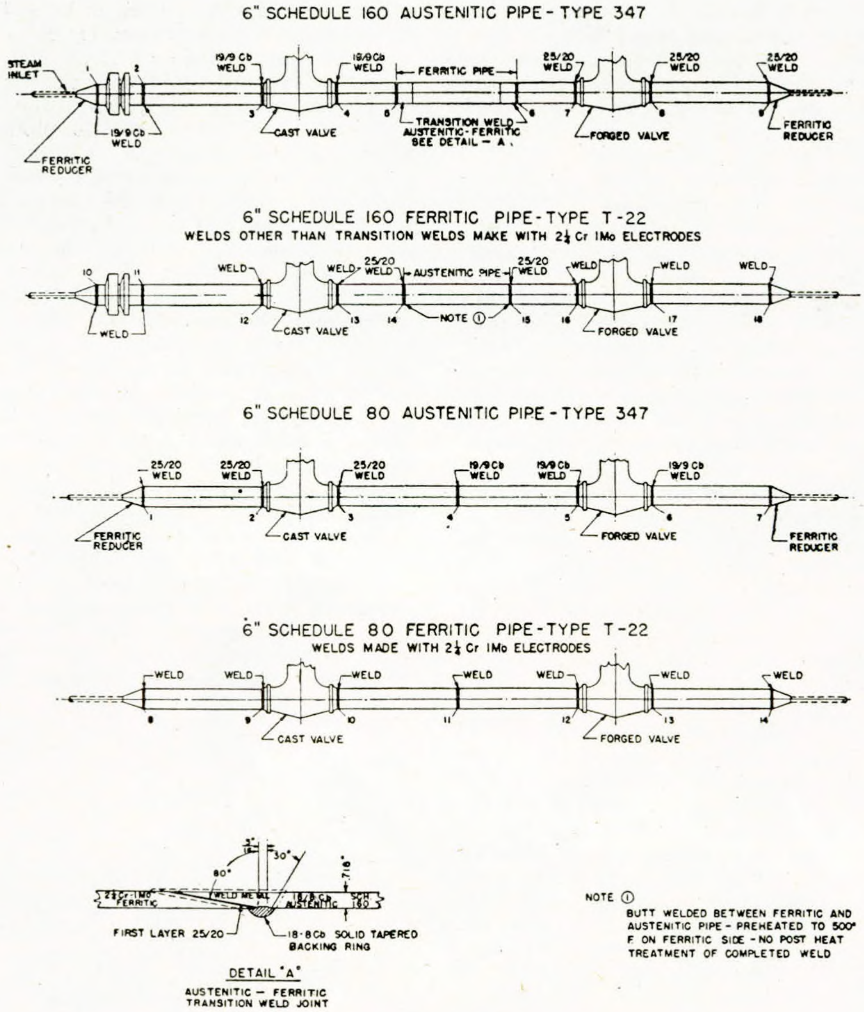


New design valve seat

a manner which increases the size of the leak. A new design of valve seat was developed incorporating thin flexible seating surfaces. The cooling effects of the leaking steam were minimized by providing better heat transfer from the high-temperature steam. The new design resulted in considerable improvement in sealing, and service tests have shown excellent performance, it is claimed.—*R. E. Adams and J. L. Corcoran. A.S.M.E. Spring Meeting Paper No. 50-S-24.*

Thermal Shock Tests of Austenitic and Ferritic Steam Piping

Thermal shock tests have been conducted on 6 inch pipe and valve assemblies representing combinations of ferritic and austenitic pipe lengths joined by welding. Each assembly contained two valves, one cast and the other forged of similar composition to the pipe material. The thermal shock tests were designed to simulate the effect of carry-over if boiler feed-water into main steam lines operating at 900 or 2,000lb. per sq. in. pressure and 1,050 deg. F. steam temperature. The assemblies were shocked by introducing either 60 or 88 lb. of boiler water at the saturation temperature along with the flow of superheated steam. Each assembly was subjected to 100 or more shocks. Temperature differences in the pipe walls show that the



Details of Thermal-shock specimens

maximum temperature difference between the inside and the outside of the wall was obtained for the ferritic steel assemblies. Moreover, the maximum temperature difference occurred in a much shorter time in the case of the ferritic steel. All thermal-shock specimens withstood 100 or more shocks without failure by rupture. Each assembly was subjected to 4,000 or more deflexions corresponding to expansion cycles that would obtain on heating a piping system from room temperature to 1,050 deg. F. once every two days for approximately 20 years.—*W. C. Stewart and W. G. Schreits, U.S. Naval Engineering Experiment Station, Annapolis, Md. 1950 A.S.M.E. Spring Meeting Paper No. 50-S-23.*

"Molykote", a New Lubricant

A new lubricant, "Molykote", is described, which consists of a smooth-textured molybdenum disulphide powder and has many valuable properties as a dry lubricant and an anti-seizing compound. It is stated to prevent galling and seizing at bearing pressures of over 100,000lb. per sq. in. and at either high or low sliding velocities. It is recommended for all types of running-in surfaces, either rubbed into the bearing surface during assembly and used with conventional lubrication, or mixed with a light oil into a paste and brushed into the surfaces; and for treatment of threaded connexions that have to be dismantled. It is also thought to counteract the effects of fretting corrosion. At temperatures of over 750 deg. F. the rate of oxidation assumes practical significance, but not below this temperature. Oxidation produces metal oxides which have no lubricating properties, although they may have anti-seizing or anti-welding properties. In the absence of air, however, lubricating properties are maintained at temperatures in excess of 750 deg. F. Molykote is non-magnetic. It is estimated that 10 ounces will liberally cover 500 sq. ft. of smooth metal with a tenacious coating.—*Scientific Lubrication, Vol. 2, April 1950, p. 16.*